Interim Research Report

on Safety Relief Valve Stability and Piping Vibration Risk

2003-2012

AIChE DIERS

The Design Institute for Emergency Relief Systems
an AIChE Industry Technology Alliance

Dear Colleague:

As you may already know, the Design Institute for Emergency Relief Systems (DIERS) is an Industry Technology Alliance (ITA) formed under the auspices of the American Institute of Chemical Engineers (AIChE). DIERS through the Users Group is active in all aspects of applied research pertaining to relief and flare systems across the wide variety of industries represented by our membership. We hold two technical meetings per year to promote the exchange of process safety and relief systems best practices.

DIERS is currently considering funding of additional research relating to safety relief valve stability and piping vibration risk. This interim research report presents a useful collection of selected papers from our meetings that discuss these important topics. These papers illustrate some of the current thinking related to safety relief valve stability and piping vibration risk.

As we try to refine the objectives, desired success factor(s), and outcomes for additional research, we would very much appreciate hearing your thoughts and ideas regarding these important topics. Please send your comments and/or suggestions to my attention.

Please note that the opinions presented in these papers do not reflect the opinions or the endorsement of AIChE or of the member companies. They are provided as is without any warranties of any kind.

If you have received a copy of this document and you are not a member of DIERS, please consider joining. Membership is free. Please visit our web site to find more about DIERS and company membership.

Sincerely,

H. G. Fisher, Chair
DIERS Users Group

Table of Contents

Safety Relief Valve

Feedback on the 3% Rule for the Inlet Pressure Drop at the Entry to a Relief Valve *- S. Egan | Rhodia*

Safety Relief Valve Stability an Engineering Analysis *- H.G. Fisher | FisherInc*

Body Bowl Choking in Pressure Safety Valves *- R. D'Alessandro, P.E. | Evonik Industries*

Stability Tests *- S. Kostos and J. Kim | Bayer*

2010 Fall DIERS Users Group – Reno, Nevada

Odds and Ends Relief Valve Stability *- M.A. Grolmes | Centaurus Technology, Inc.*

2009 Fall DIERS Users Group – Houston, Texas

Some Further Considerations of Upstream Parameter Effects on Relief Valve Stability *- M.A. Grolmes | Centaurus Technology, Inc.*

2009 Spring DIERS Users Group – Orlando, Florida

Deal with Controversial Topics in Pressure Relief Systems *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

Controversial Topics Unknowns in Emergency Relief Sizing *- H.G. Fisher | FisherInc*

Best Practices for Flare Systems Evaluation *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

Control Valve Flow Coefficients *- M.A. Grolmes | Centaurus Technology, Inc.*

General Notes on the Use of Control Valve Flow Coefficients *- M.A. Grolmes | Centaurus Technology, Inc.*

2008 Spring DIERS Users Group – Las Vegas, Nevada

Some SRV Performance Findings from the BP Incident Investigation *- H.G. Fisher | FisherInc*

Pressure Relief Sizing - Odds and Ends *- M.A. Grolmes | Centaurus Technology, Inc.*

Comments from ISO 4126 Committee Meeting *- A. West | ISO 4126-10 Committee*

2007 Fall DIERS Users Group – Manchester, New Hampshire

Development of Spring Loaded Safety Valves *- R.D. Danzy | Dresser, Inc.*

Backpressure Effects on Spring Loaded Safety Valves *- R.D. Danzy | Dresser, Inc.*

2007 Spring DIERS Users Group – Philadelphia, Pennsylvania

BP Incident Scenario *- J.Lay | Chemical Safety Board and H. Fisher | FisherInc*

2006 Fall DIERS Users Group

Understand Flare Noise *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

2005 Spring DIERS Users Group – Charleston, South Carolina

Back Pressure *- Biel | LESSER GmbH & Co.*

2005 3rd International Symposium

Pressure Safety Valve Thrust Forces for Compressible Gas or Vapor Flow *- R. D'Alessandro, P.E. | Evonik Industries*

2004 Fall DIERS Users Group – Denver, Colorado

Simpson Charts for PSV Thrust Force Revisited *- R. D'Alessandro, P.E. | Evonik Industries*

2003 Fall DIERS Users Group – Las Vegas, Nevada

Practical Guidelines for Dealing with Excessive Pressure Drop in Relief Systems *- G.A. Melhem, Ph.D. | ioMosaic Corporation and H.G. Fisher | FisherInc*

Piping Vibration Risk

2012 Spring DIERS Users Group – Kansas City, Missouri

Estimate Vibration Risk for Relief and Process Piping *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

Assessment of Piping Vibration Risk *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

2011 Spring DIERS Users Group – Hamburg, Germany

Estimate Acoustically Induced Vibration Risk in Relief and Flare Piping *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

2010 Spring DIERS Users Group – Houston, Texas

Assessment of Piping Vibration Risk *- G.A. Melhem, Ph.D. | ioMosaic Corporation*

2007 Fall DIERS Users Group – Manchester, New Hampshire

Screening for Vibration Induced Fatigue in Process & Relief Piping *- D. Nguyen, P.E. | ioMosaic Corporation*

Analysis of PERF-99-5 Data

DIERS Fall 2012 Meeting

Concord, MA

October 10, 2012

Jim Lay, PE
Senior Process Safety Engineer
USDOL-OSHA, Enforcement Programs

OSHA adds value to business, work and life

Disclaimer

This presentation represents the work and views of the author and does not necessarily represent the views, policy, or endorsement of the Occupational Safety and Health Administration

Summary:

- PERF 99-05 data indicates that relief valve blowdowns vary widely from valve-to-valve, are not tightly grouped around 7% of set
- Ranking of PERF test results by [Valve Blowdown – Inlet Pressure Loss] suggests that a margin of at least 4% may be needed to ensure stable valve operation
- Future work on RV stability should consider & address these findings

OSHA
adds value to business, work and life

Topics:

- Brief overview of statistical sampling
- Statistical distribution of PERF-99-5 blowdown data
- “Margin” [blowdown – inlet loss] for stable operation
- Implications for future experimental / modeling work
- Q&A

OSHA adds value to business, work and life

Sampling:

- A means of estimating population characteristics from a limited number of data points (a sample)
- Two major concerns:
  - The sample must be *representative of the population being studied*
  - *The sample must be large enough to provide useful precision (i.e., have adequate statistical "power")*

OSHA adds value to business, work and life

Relief Valve Blowdown:

- Defined as the pressure below the set pressure at which a relief valve closes, expressed as a % of set pressure
- A “classic” example: A relief valve with a set point of 100 psig closing at 93 psig has a blowdown of 7%

Relief Valve Blowdown:

- A measurable characteristic of relief valves
  - May require high flow test apparatus, especially for larger valves
- Expected to be a “random variable” due to variation in relief valve component tolerances, e.g., springs, disc, ring, others…
  - And in fact blowdowns varied from valve-to-valve when tested in PERF-99-05

OSHA adds value to business, work and life

Relief Valve Blowdown:

- Identified as an important input to valve stability modeling (Darby, Melhem)
- Relevant to determining “safe operating margin”
  - The difference between blowdown and inlet pressure loss needed to assure stable operation
  - The “3% rule” implies a 4% safe operating margin based on the widespread assumption that blowdown is 7% for API 526 valves

OSHA adds value to business, work and life

Relief Valve Blowdown:

- The distribution of blowdown in conventional process service relief valves is thus highly relevant to current API, DIERS, & other attempts to develop engineering tools and guidance to assure stable operation in new and existing relief valve installations
- Statistical blowdown distribution can be used as input to computer models or in “margin” calculations

Relief Valve Blowdown:

- There is very little published blowdown data
- National Board blowdown data from certification testing not published
  - Per Code, blowdown is first determined…
  - …and then adjusted to **no more** than 5% (capacity certification UG-131(3)(a)) or 7% (facility certification UG-136(c)(3)(b))
- PERF 99-05 is the largest published blowdown dataset identified

PERF-99-05 Blowdown Data:

Testing Program Results

Valve Number	**Valve Size & Set Pressure**	Inlet Piping Length and Pressure Drop						Measured Blowdown
		2 Feet Valve Initial Lift (Test)	Inlet Pressure Drop % of Set Pressure	**4 Feet Valve Initial Lift (Test)**	Inlet Pressure Drop % of Set Pressure	**6 Feet Valve Initial Lift (Test)**	Inlet Pressure Drop % of Set Pressure	
1	1E2 - 50 psig	Stable	1.2	Stable	2.0	Stable*	2.7	5.3
2	1E2 - 250 psig	Stable	1.0	Unstable	1.7	Unstable	**2.2**	**5.6**
3	2J3 - 50 psig	Stable	2.2	Stable	3.2	Unstable	**4.3**	**8.0**
4	2J3 - 250 psig	Stable	2.0	Test Failed	3.0	Not Tested	4.0	7.6
5	3L4 - 50 psig	Stable	2.3	Stable	3.3	Stable	**4.5**	**4.3**
6	3L4 - 250 psig	Stable	1.8	Stable	2.7	Not Tested	3.5	11.9
7	1E2 - 50 psig	Stable	1.1	Stable	1.9	Stable	2.6	7.1
8	1E2 - 250 psig	Stable	**1.0**	Unstable	1.7	Unstable	2.3	**2.5**
9	2J3 - 50 psig	Stable	2.3	Stable	3.5	Stable	4.6	9.1
10	2J3 - 250 psig	Stable	1.9	Not Tested	2.8	Not Tested	3.8	6.4
11	3L4 - 50 psig	Stable	2.3	Stable	3.3	Stable	**4.6**	**5.6**
12	3L4 - 250 psig	Stable	1.8	Not Tested	2.7	Not Tested	3.6	5.7
13	1E2 - 50 psig	Stable	1.1	Stable	1.9	Stable	2.7	9.9
14	1E2 - 250 psig	Stable	1.0	Stable	1.7	Stable	2.2	6.4
15	2J3 - 50 psig	Stable	2.3	Stable	3.4	Stable	4.5	11.8
16	2J3 - 250 psig	Stable	1.9	Not Tested	2.8	Not Tested	3.7	5.6
17	3L4 - 50 psig	Stable	2.5	Stable	3.5	Stable	4.6	10.6
18	3L4 - 250 psig	Stable	1.8	Unstable	2.7	Not Tested	3.5	4.4

PERF-99-05 Blowdown Dataset:

- Valves were "designed, built, and tested to the requirements of both ASME Boiler and Pressure Vessel Code Section VIII, Div. 1 and API Standards 526 and 527"
- "PRVs within this scope constitute the majority of the pressure relief devices in service in the hydrocarbon processing industry throughout the world."

From PERF 99-05 Phase III report, October, 2002, reissued 2012

OSHA adds value to business, work and life

PERF-99-05 Blowdown Dataset:

- **A screening study** intended to be broadly representative, although not exhaustive
  - Included multiple manufacturers, sizes, set pressures
- **A reasonable and representative dataset for estimating the blowdown distribution of relief valves currently installed in U.S. process plants**

OSHA
adds value to business, work and life

PERF-99-05 Blowdown Dataset:

- **Statistical "power"** is primarily based on data quality and the number of data points
- When fitting a distribution to data, it's not meaningful to talk about "statistical significance"
  - No hypothesis testing involved
- Goodness-of-fit & confidence interval width are the key measures of stat "power"

OSHA
adds value to business, work and life

Statistical Distributions:

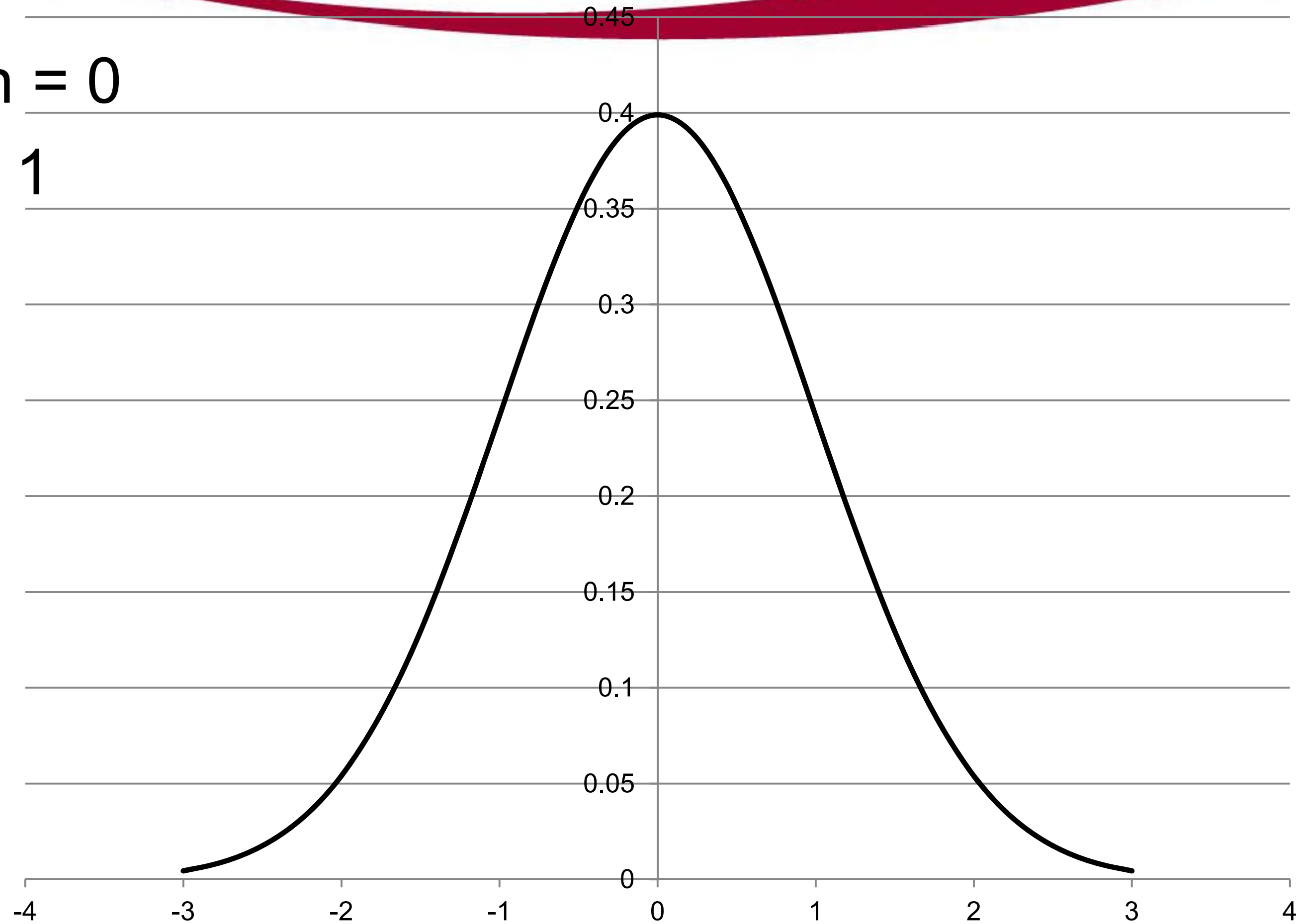
- Describe the expected distribution of attribute values over a population
- Very useful for characterizing manufacturing process capability, often as a first step in process optimization
- Best known distribution is the “normal” distribution – the classic “bell shaped” curve

OSHA
adds value to business, work and life

Standard Normal Probability Density Function

Mean = 0

SD = 1



$F(x, \mu,\sigma)=(2\pi\sigma^2)^{-0.5}\exp(-(x-\mu)^2/(2\sigma^2))$

OSHA adds value to business, work and life

Statistical Distributions:

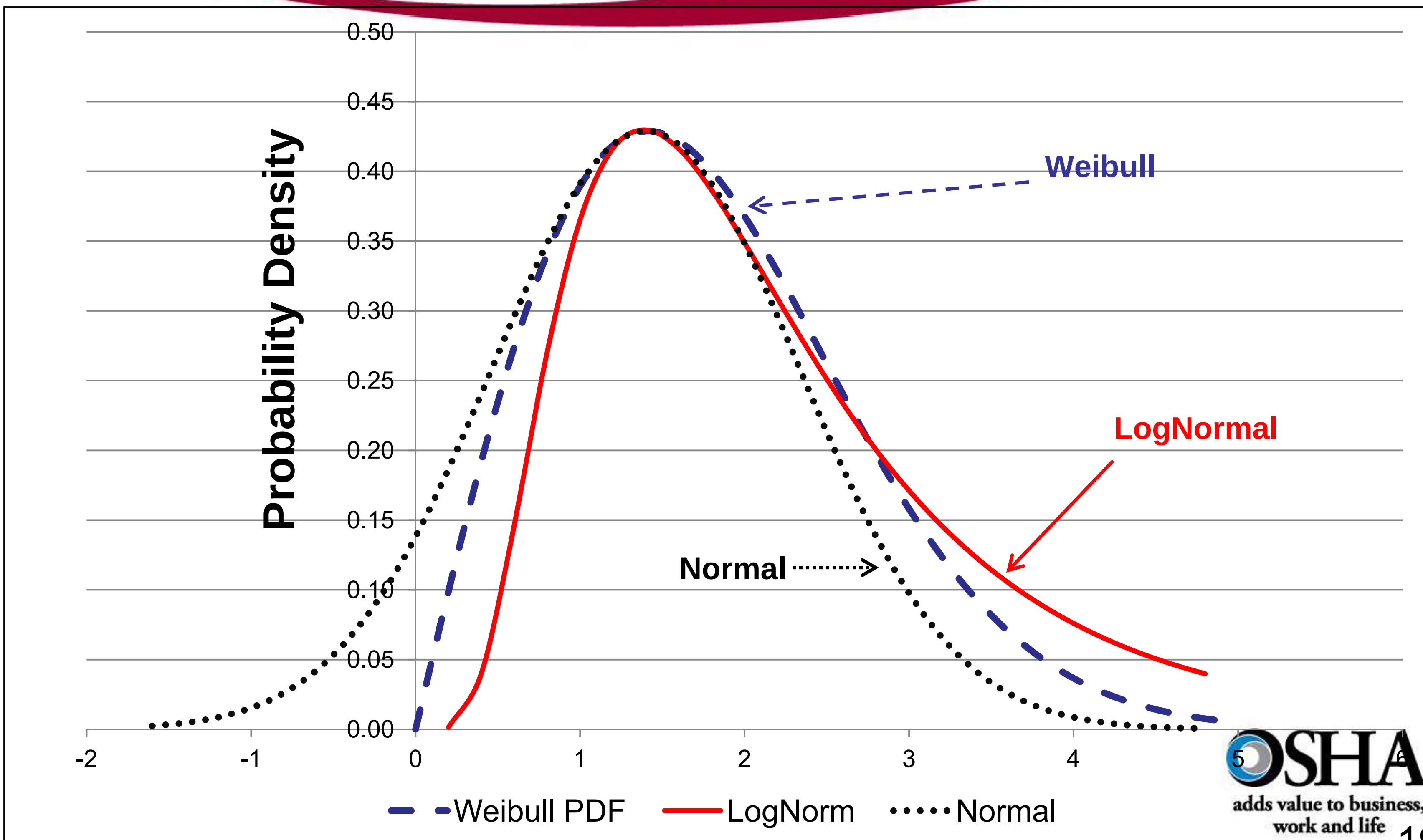
- Not all data follow a normal distribution
  - May have only positive values
  - Can be skewed or exhibit “spread” wider or narrower than the “normal” distribution
- Manufacturing & reliability data or attribute distributions that are dominated by one major factor often follow a “Weibull” distribution

Statistical Distributions:

- Assemblies with multiple components affecting the attribute of interest frequently follow a “LogNormal” distribution
- Both Weibull and LogNormal are constrained to positive values and have varying degrees of skew and heavier or lighter “tails” depending on the value of their parameters
- All three are 2 parameter distributions

OSHA
adds value to business, work and life

Statistical Distributions: Weibull, LogNormal, & Normal PDFs (typical)



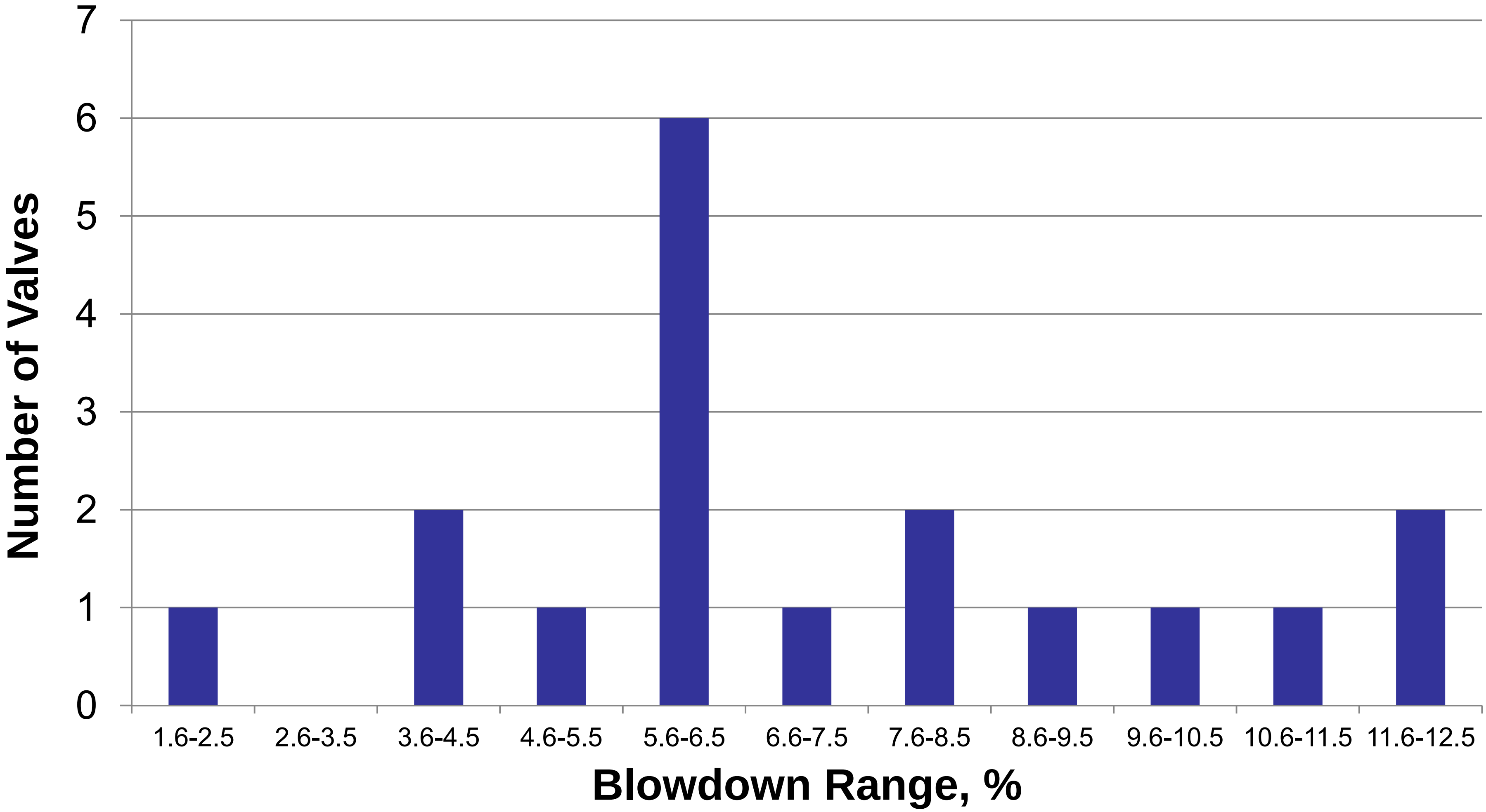
OSHA adds value to business, work and life

Statistical Distributions:

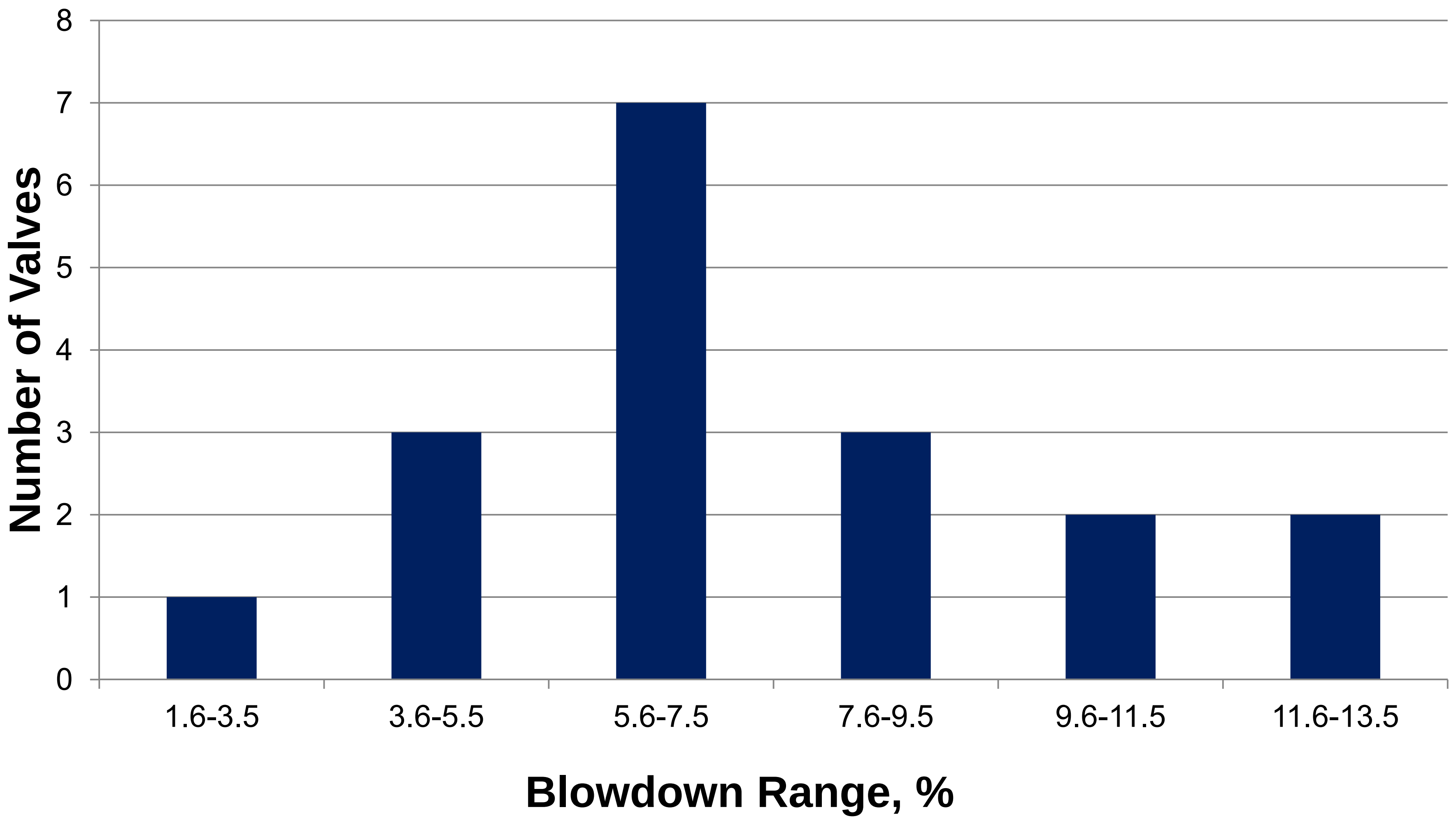
- Probability density functions are how most people visualize distributions
- In statistics texts these are often shown built up from histograms
- This is inefficient use of data, and very sensitive to the number and location of the histogram "buckets" used
- But they can be useful for preliminary "eyeball" evaluations

OSHA adds value to business, work and life

PERF 99-05 Blowdown Histogram 11 “buckets”:



PERF 99-05 Blowdown Histogram
6 “buckets”:

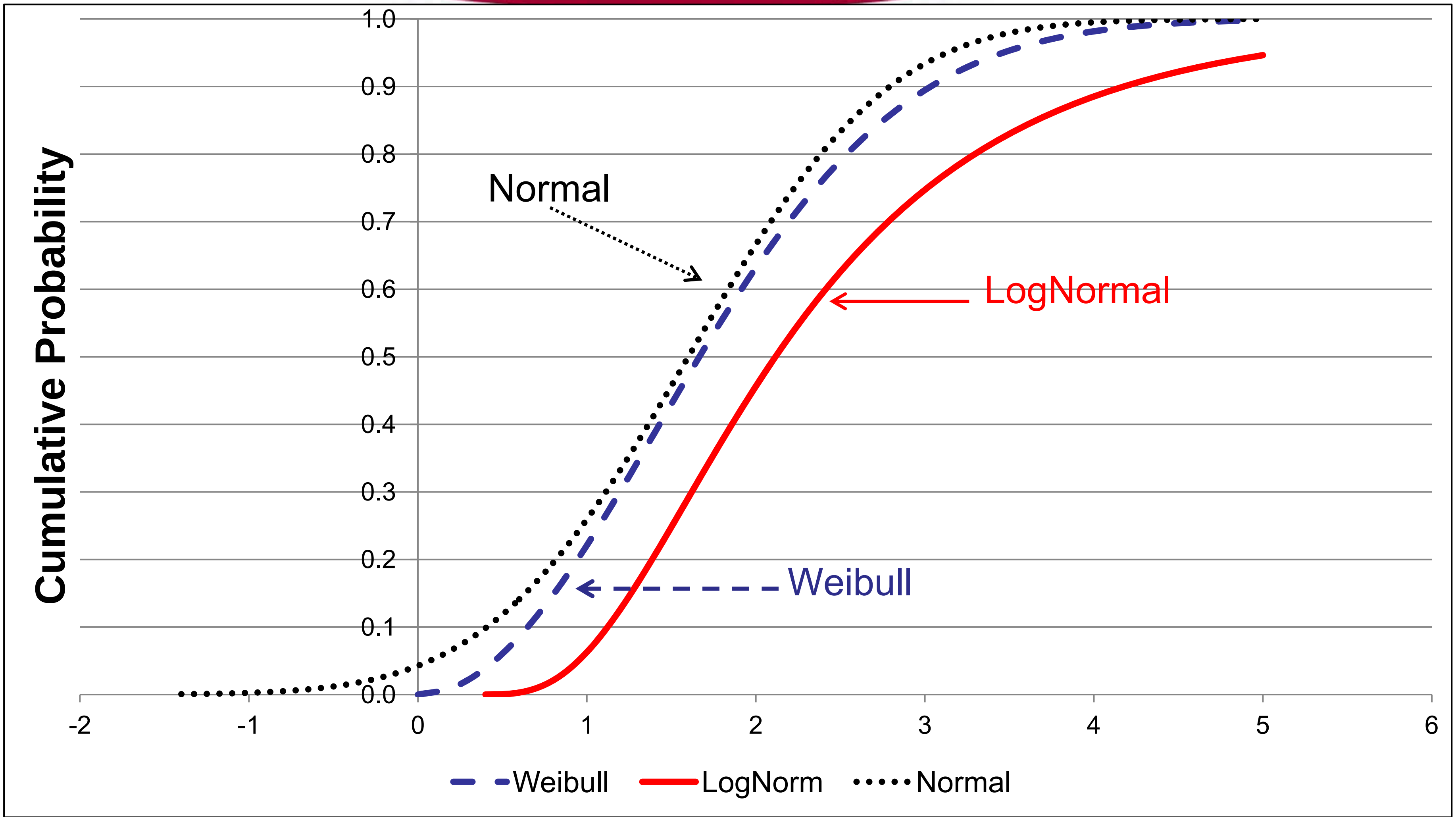


Statistical Distributions:

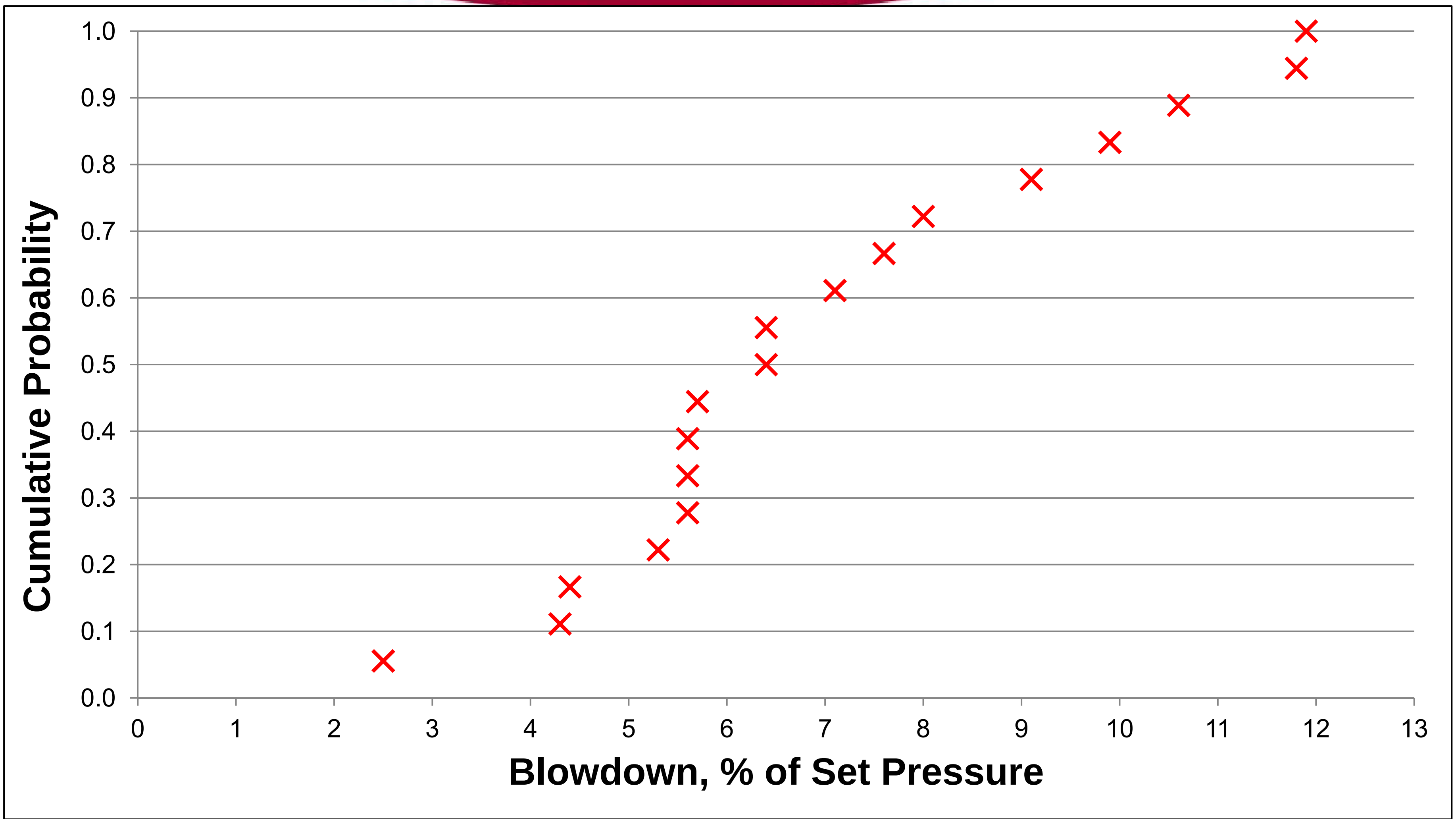
- A better approach is to plot the data as an ordered series with the cumulative probability plotted against blowdown value
- This makes full use of the data – not "diluting" it into buckets
- The cumulative probability function for a distribution is just the integral of the Probability Density Function

OSHA adds value to business, work and life

Statistical Distributions – Cumulative Probability Distributions:



Statistical Distributions – PERF Cumulative Probability Data Plot:

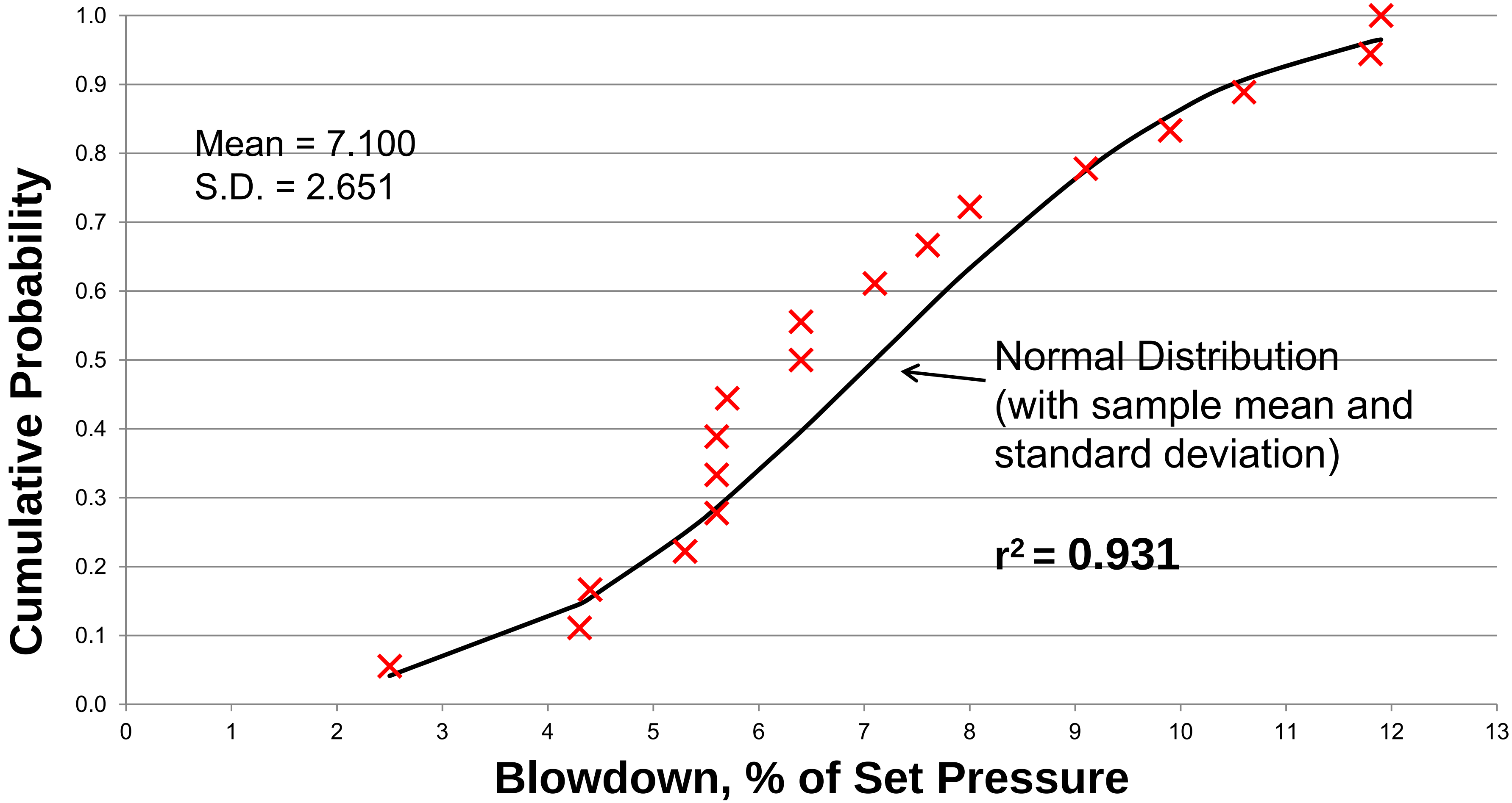


PERF 99-5 Blowdown Data – Raw Data Descriptive Statistics:

For all 18 data points:

- Mean of 7.1%
- Median of 6.4%
- Mode of 5.6%
- S.D. of 2.65%

Statistical Distributions – Normal Distribution CPD Fit to Data:

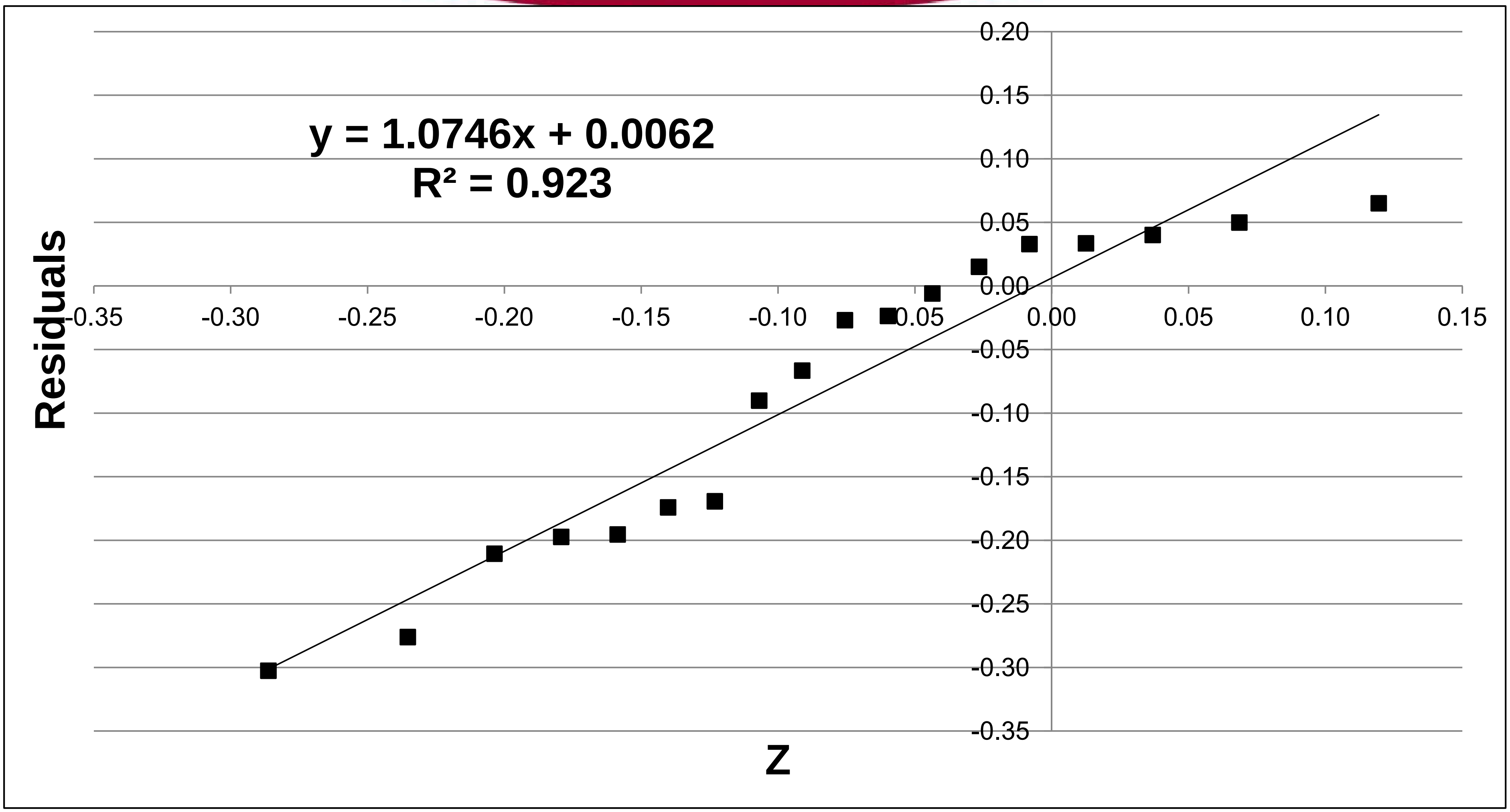


Residuals: The differences between the data points and the fitted distribution

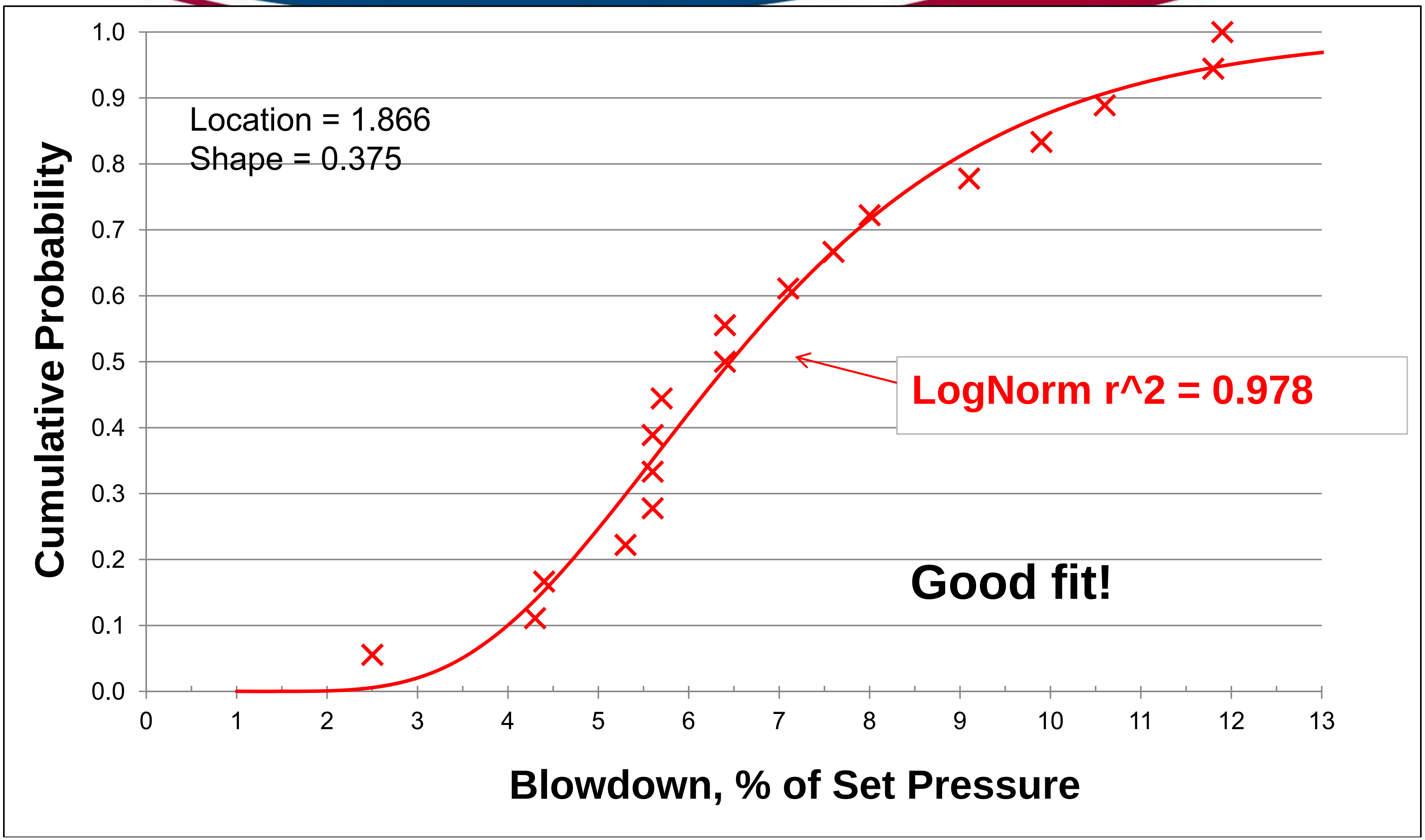
- If deviations from the fitted distribution are random (noise), then the residuals should be normally distributed
- Plotting randomly distributed residuals on a "normal distribution" plot should yield a straight line
- Goodness of fit and visual observation are both useful in distinguishing random from systemic effects

OSHA adds value to business, work and life

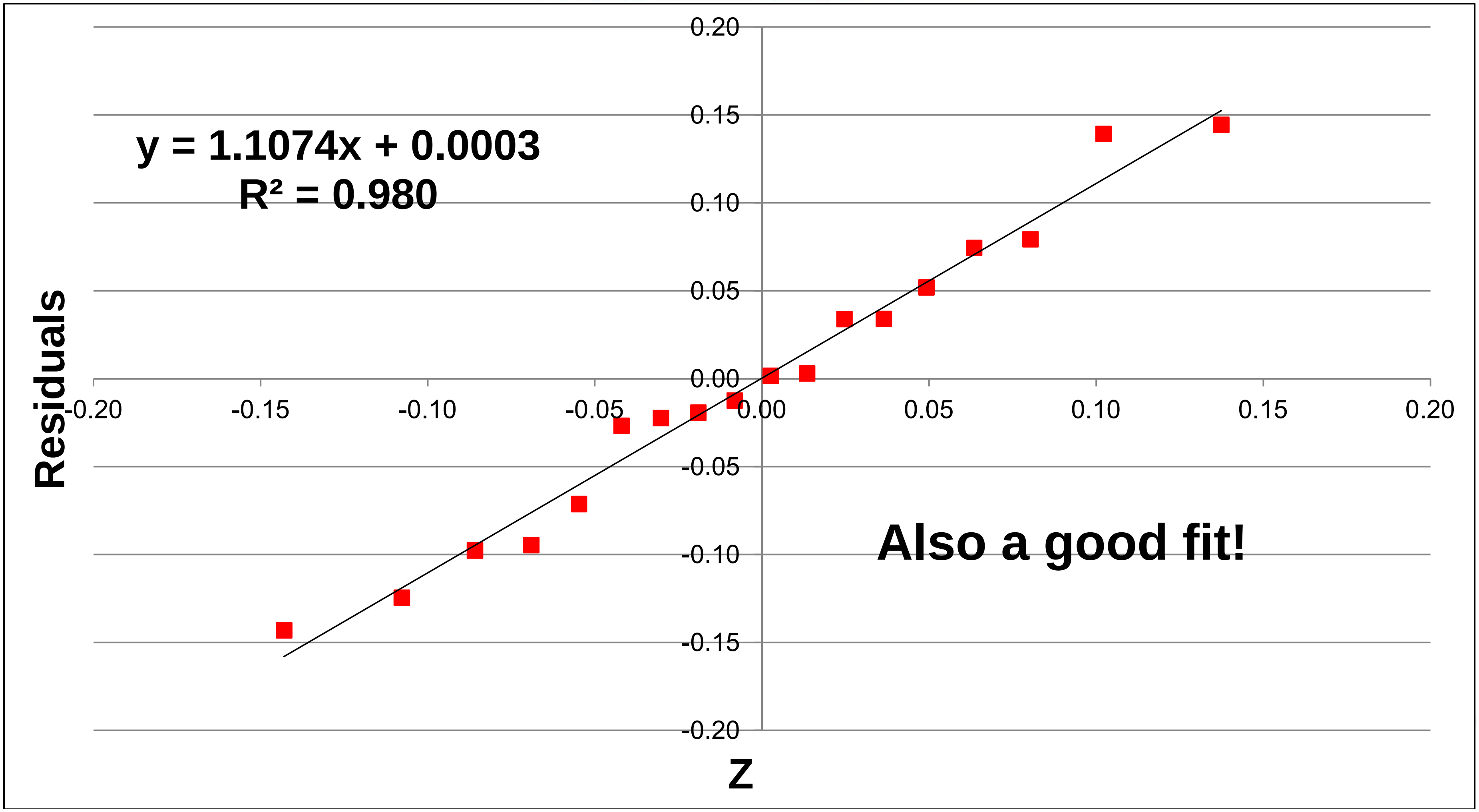
Sorted Residuals vs Z for the Normal Distribution Fit to the PERF Data:



Statistical Distributions – LogNormal Distribution Fit to Data:



Sorted Residuals vs Z for LogNormal Distribution Fit:



Selection of Distribution for Follow-up:

- Multiple distributions were tried, including least-squares fitted Normal and Weibull distributions.
- LogNormal had the best fit to the data by r^2 and the best behaved residuals
- LogNormal was also conservative in predicting a smaller fraction of valves with low blowdowns than other distributions

OSHA
adds value to business, work and life

So, How is this Useful?

- Can make predictions about the fraction of valves expected to have blowdowns at or below (or above) any specified value
- These can be used as inputs to computer simulations to more accurately reflect expected valve characteristics
- By calculating confidence intervals we can quantify the uncertainty in these predictions
- Can also estimate the sample size needed to reduce this uncertainty to a specified level

OSHA adds value to business, work and life

Expected Valve Fraction vs Blowdown:

Blowdown % Set Pres	LogNorm Cum Prob
2	0.001
3	0.020
4	0.100
5	0.247
6	0.422
7	0.585
8	0.716
9	0.812
10	0.878
11	0.922
12	0.951

OSHA adds value to business, work and life

Discussion:

- Having 10% of valves with BDs < 4%, and 25% of valves with BDs < 5% should give relief system experts pause
- Weibull predicts 15% of valves with BD < 4%, 7% of valves with BD < 3%
- Fraction with BD > 10% also significant
- The long standing assumption that relief valve blowdowns are tightly grouped around 7% appears to be in error, possibly badly in error

OSHA adds value to business, work and life

Discussion:

- This BD distribution can potentially explain many historical stability observations – such as chatter at < 3% inlet pressure loss and stability at higher inlet losses
- Low BDs are expected to reduce valve stability
- High blowdowns could cause sustained discharge for processes with small operating pressure margins

OSHA adds value to business, work and life

Statistical Power of PERF Analysis:

- Statistical power was addressed by calculating 95% confidence intervals for the fraction of valves with blowdowns at or below various values
- Based on the standard deviation of the LogNormal distribution being Chi-Squared distributed

MS Excel Formulas for Upper & Lower Confidence Intervals:

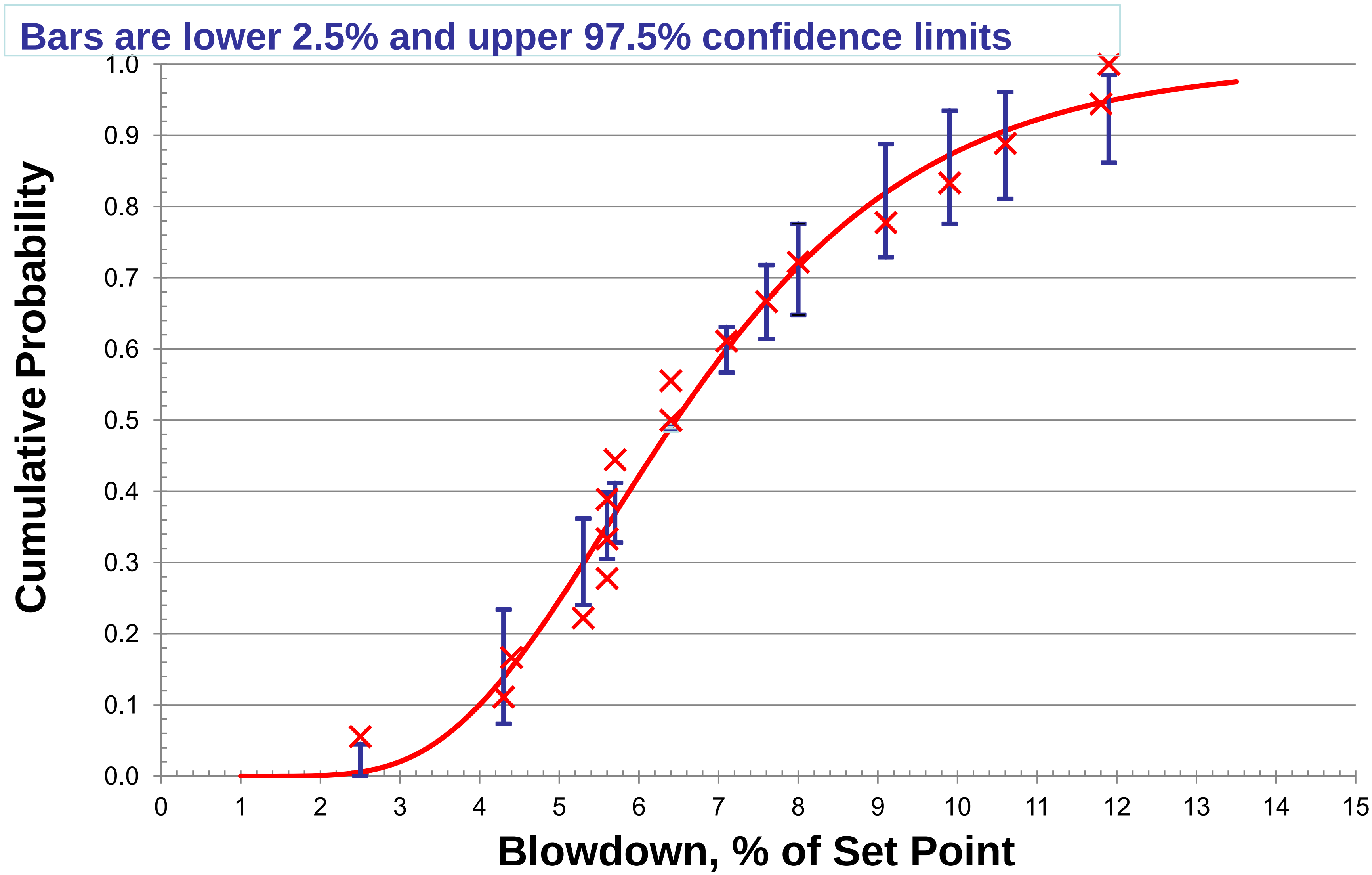
Lower limit: =SD***SQRT**((N-1)/**CHIINV**((alpha/2), N-1))

Upper limit: =SD***SQRT**((N-1)/**CHIINV**(1-(alpha/2), N-1))

95% Confidence Intervals for Fitted LogNormal:

Blowdown	Expected Fraction	Lower 2.5% CI	Upper 97.5% CI
≤6%:	42%	39.6%	44.8%
≤5%:	25%	18.1%	32.4%
≤4%:	10%	4.4%	19.7%
≤3%:	2.0%	0.3%	8.6%
≥9%:	18.8%	11.9%	27.8%
≥10%:	12.2%	6.0%	21.8%
≥12%:	4.9%	1.4%	13.5%

LogNormal Confidence Interval Graph
(95% within bars)



Confidence Interval Sensitivity to Number of Data Points:

- *At moderate blowdowns the "spread" between upper and lower confidence limits is approximately proportional to $1/\sqrt{N}$*
- *Reducing the confidence ranges shown by a factor of 4 would thus require blowdowns from a representative sample of ~288 relief valves (4^2 x 18)*
- *Use of archival data may be a cost effective way to improve precision*

OSHA
adds value to business, work and life

[Blowdown – Inlet Pressure Loss] "Margin":

- The 44 completed PERF tests were sorted by [Measured Blowdown – Inlet Loss]
- 6 of 18 (33%) of tests with a "margin" < 4% chattered
- 26 of 26 (100%) of tests with a "margin" > 4% were stable
- 4% was a break point below which some valves chattered, and above which none did

OSHA adds value to business, work and life

[Blowdown – Inlet Pressure Loss] "Margin":

- 10 planned tests were reported as "not tested"
- Hopefully the reasons for this can be clarified by the PERF experimental team.
- At least some of the valves not tested were damaged in earlier tests

OSHA adds value to business, work and life

[Blowdown – Inlet Pressure Loss] "Margin":

	Valve Type	Piping Length, ft	Test Pressure, psig	Measured Blowdown	Inlet dP	**Margin**	Stable? (+ or -)
1	3L4	6	50	4.3	4.5	-0.2	+
2	**1E2**	**6**	**250**	**2.5**	**2.3**	**0.2**	**-**
3	**1E2**	**4**	**250**	**2.5**	**1.7**	**0.8**	**-**
4	3L4	4	50	4.3	3.3	1	+
5	3L4	6	50	5.6	4.6	1	+
6	1E2	2	250	2.5	1	1.5	+
7	**3L4**	**4**	**250**	**4.4**	**2.7**	**1.7**	**-**
8	3L4	2	50	4.3	2.3	2	+
9	3L4	4	50	5.6	3.3	2.3	+
10	1E2	6	50	5.3	2.7	2.6	+
11	3L4	2	250	4.4	1.8	2.6	+
12	3L4	2	50	5.6	2.3	3.3	+
13	1E2	4	50	5.3	2	3.3	+
14	**1E2**	**6**	**250**	**5.6**	**2.2**	**3.4**	**-**
15	2J3	2	250	5.6	1.9	3.7	+
16	**2J3**	**6**	**50**	**8**	**4.3**	**3.7**	**-**
17	**1E2**	**4**	**250**	**5.6**	**1.7**	**3.9**	**-**
18	3L4	2	250	5.7	1.8	3.9	+

[Blowdown – Inlet Pressure Loss] "Margin":

- This analysis does not break out acoustic coupling from other chattering mechanisms
- In the absence of acoustic modeling a ***minimum*** 4% margin appears necessary to ensure stable operation
- This margin is based on ACTUAL VALVE BLOWDOWN, not "nominal" or "average" blowdowns for a class of valves!

OSHA
adds value to business, work and life

Implications for Future Work - Blowdown:

- The PERF-99-5 data are from a screening study that selected valves that are widely representative of those installed in the process industries
- The data are very well fitted by the LogNormal statistical distribution with high correlation and no evidence of lack-of-fit
- The quality and quantity of the data give reasonably tight confidence intervals around the predicted distribution
- The data strongly support blowdown variability as a significant, potentially very significant, factor in relief valve instability

OSHA adds value to business, work and life

Implications for Future Work - Blowdown:

- It is critical to firm up our understanding of the blowdown distribution, including factors such as:
  - Size, set pressure, manufacturer
  - New versus in-service
  - Other variables?
- Multiple data sources should be explored, e.g., experimental and archival
- Experiments should be based on representative normal-production run valves

OSHA adds value to business, work and life

Implications for Future Work - Blowdown:

- A sufficiently large dataset should be developed to achieve predetermined confidence intervals
  - This should be looked at “up front” in any proposed study
- A representative and adequately powered study will strengthen the usefulness and credibility of the tools developed to predict and ensure valve stability

OSHA adds value to business, work and life

Implications for Future Work - Blowdown:

- Keep in mind the potential need for establishing performance tolerances on relief valve blowdown
  - What is desirable?
  - What is achievable?
  - Coordinate with the ASME pressure relief sub-committee
- Precision of the PERF blowdown measurements should be good (PTC 25 test rig, average of three readings) – this needs to be confirmed

OSHA adds value to business, work and life

Implications for Future Work – Safe [BD-IPD] Margin:

- A minimum 4% margin based on actual blowdown appears necessary in the absence of more detailed analysis
- Blowdowns for individual valves are not currently available. Basing margins on conservative BD values may give impractical results
- "Safe margin" may be highly interactive with other causes of valve instability, such as acoustic coupling and excessive back pressure
- Recommend addressing explicitly in future work

OSHA adds value to business, work and life

Any Questions??

Jim Lay, PE
OSHA – Directorate of Enforcement Programs
General Industry
202.693.1827
Lay.Jim@dol.gov

OSHA
adds value to business, work and life

ioMosaic

Business Confidential Document

PRV Stability Requirements

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

50th DIERS Users Group Meeting

October 2012

Concord, Massachusetts

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

The design and evaluation of relief systems is highly constrained

- The relief system must prevent the failure of the vessel due to overpressure or underpressure
- Damage to vessel, piping, and valve must be prevented by design
  - Fluid reaction forces, steady and dynamic loading
  - Vibration risk, especially to discharge piping
  - Fatigue failure caused by PRV chatter to valve, piping, and piping components
  - Large pressure fluctuations caused by acoustic resonance
  - Vortex shedding (singing relief valve problem) for some specific installations
- If we can address all the above requirements, we also need to properly handle and treat the effluent
- Finally we must properly document the design and keep the documentation up to date and easily accessible

Typical causes of chatter include

- Excessive inlet pressure loss
- Excessive backpressure
- Oversized valve
- Bad installation

ioMosaic

To accurately assess whether a "relief system" will operate in a stable manner (chatter free) you must consider the following important system time constants and how they interact

- Valve time constant
  - How fast does a pressure relief device close and open?
- Vessel or pressure source time constant
  - How fast does a vessel de-pressure and re-pressure after a pressure relief device opens and re-seats?
- Inlet line time constant
  - How long does it take for a pressure wave to propagate upstream from a pressure relief device to the pressure source and back?
- Outlet line time constant
  - Acoustic barriers may be established due to "body bowl choking"
  - Note that acoustic barriers, such as the presence of control valves, change in diameter, etc. can cause standing waves that can lead to acoustic coupling or resonance with relief systems components

ioMosaic

How can we establish if PRV stability is an issue for a specific installation?

- It has been shown that very long inlet lines will result in stable PRV operation
- It has been shown that a PRV will go through a short period of instability during closing and by inference during opening as the valve gets to full lift
- During valve closing, if the inlet line is short enough, the returning compression wave can keep the valve open
- It is expected that the coefficient of restitution can change the calculated valve opening time
- A slower valve is better
- Low flow rates cause less damping for the piping
- Resonance amplification factors can be very large, 50 to 100 times

The key is to decouple PRV frequency from the piping and other important system frequencies or to slow down the valve response (closure) time

ioMosaic

In order to understand PRV stability we first need to understand some fundamentals about pressure pulsations/oscillations, the energy source for standing waves

- A pulsation is a pressure fluctuation that travels through the fluid and/or piping system
- Pulsations are propagated as traveling acoustic waves, i.e. at the speed of sound in the fluid/piping system
- When an acoustic wave encounters a change in area, new reflected and transmitted waves are generated
- The superposition of two or more traveling waves that have the same frequency generates a standing wave at the same frequency
- Amplitudes of standing waves can grow to extremely large values when resonance occurs

The acoustic velocity estimates can be subject to large uncertainties

- This is most significant for liquids
- Pipe flexibility can lower the value of the acoustic velocity
- The presence of minute amounts of entrained gas in liquids can drastically reduce the value of the acoustic velocity
- Adding a small amount of air, say 0.1 % by volume can reduce the value of the acoustic velocity for a liquid-air system by a factor of 1/2

ioMosaic

The acoustic velocity of a traveling wave can be calculated based on the fluid properties and the flexibility of the piping supports

$$u_{ac} = \eta u_{sonic} = \frac{1}{\sqrt{1 + \frac{K}{E}\psi}} u_{sonic}$$

$$u_{sonic} = \sqrt{\frac{C_p}{C_v}\frac{1}{\kappa\rho}} = \sqrt{\left[\frac{\partial P}{\partial \rho}\right]_S} = \sqrt{\frac{1}{\kappa_S \rho}}$$

Pipe condition	ψ
Rigid	0
Anchored against longitudinal movement through its length	$\frac{d}{\delta}(1-\nu^2)$
Anchored against longitudinal movement at the upper end	$\frac{d}{\delta}(1.25-\nu)$
Frequent expansion joints present	$\frac{d}{\delta}$

ioMosaic

The acoustic velocity of a traveling wave can be calculated based on the fluid properties and the flexibility of the piping supports

Material	E (GPa)	Poisson's Ratio ν	$K = \frac{1}{\kappa}$ (GPa)	ρ in kg/m3
Aluminum	69	0.33		
Brass	78-110	0.36		
Carbon steel	202	0.303		
Cast iron	90-160	0.25		
Concrete	20-30	0.15		
Copper	117	0.36		
Ductile iron	172	0.30		
Fibre cement	24	0.17		
High carbon steel	210	0.295		
Inconel	214	0.29		
Mild steel	200-212	0.27		
Nickel steel	213	0.31		
Plastic / Perspex	6.0	0.33		
Plastic / Polyethylene	0.8	0.46		
Plastic / PVC rigid	2.4-2.75			
Stainless steel 18-8	201	0.30		
Water - fresh			2.19	999 at 20 C
Water - sea			2.27	1025 at 15 C

Let's consider the impact of piping flexibility on acoustic velocity reduction

Material	Piping Schedule US	$K = 1/\kappa$ GPa	d/δ	η
Liquid Water	5	2.19	52.2	0.799
Liquid Water	10	2.19	35.5	0.850
Liquid Water	40	2.19	13.4	0.934
Liquid Water	80	2.19	11.3	0.944
Liquid Water	160	2.19	6.47	0.967
Liquid Propane	5	0.11	52.2	0.986
Liquid Propane	10	0.11	35.5	0.991
Liquid Propane	40	0.11	13.4	0.996
Liquid Propane	80	0.11	11.3	0.997
Liquid Propane	160	0.11	6.47	0.998
Vapor Propane	5	6.80E-04	52.2	1.000
Vapor Propane	10	6.80E-04	35.5	1.000
Vapor Propane	40	6.80E-04	13.4	1.000
Vapor Propane	80	6.80E-04	11.3	1.000
Vapor Propane	160	6.80E-04	6.47	1.000

Propane data at 293 K and 8.35 bara

Piping flexibility is most important for liquids that are highly incompressible where thin wall piping is used

ioMosaic

It is well established that small amounts of vapor can drastically reduce the mixture speed of sound

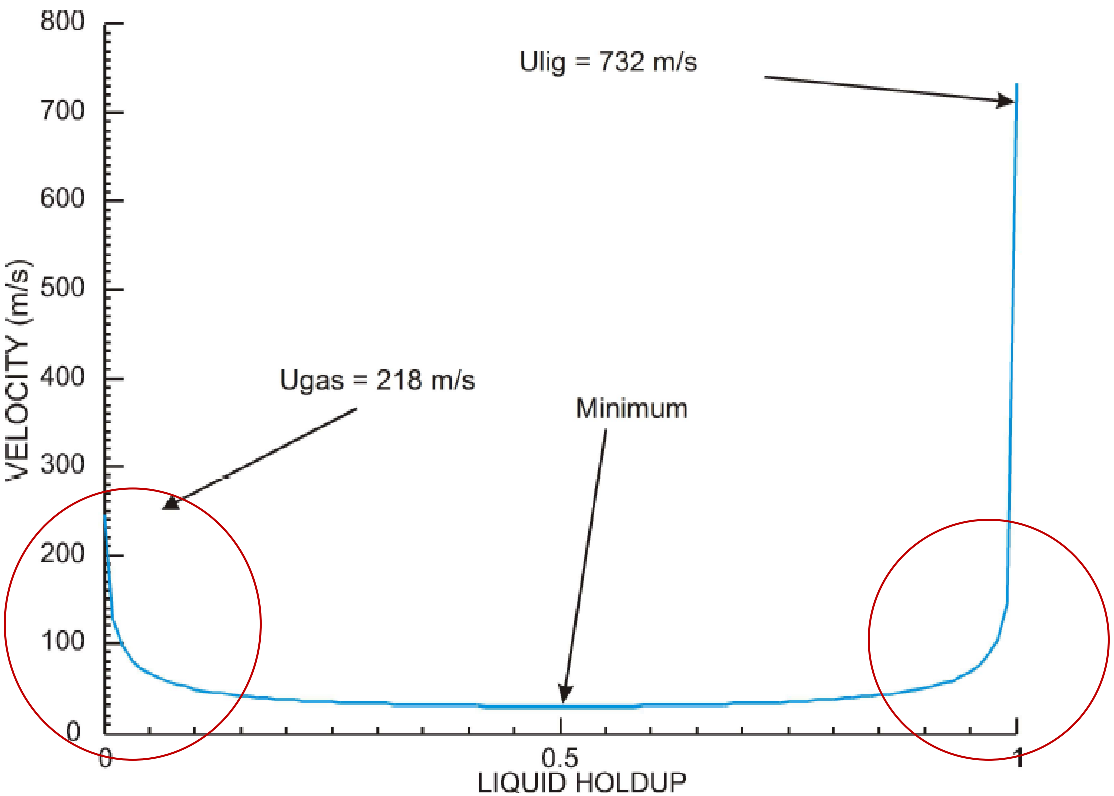


Table 2.1: Propane properties at 293 K and 8.35×10^5 Pa.

Property	Liquid	Vapor
Density, kg/m^3	523	18.1
κ, Pa^{-1}	9.18×10^{-9}	1.47×10^{-6}
β, K^{-1}	0.00408	0.00558
u_{max}, m/s	733	218

Table 2.2: Calculated propane mixture maximum velocities

ϵ_l	$u_{m,max}$ in (m/s)	ϵ_l	$u_{m,max}$ in (m/s)
0.00	218	0.60	81.7
0.05	145	0.70	87.6
0.10	118	0.80	100
0.20	95	0.90	133
0.30	85.1	0.95	181
0.40	80.7	0.99	357
0.50	79.7	1.00	732

$$u_m^2 = \frac{1}{\frac{\alpha}{u_{g,max}^2} + \frac{1-\alpha}{u_{l,max}^2} + (\rho_g - \rho_l)\left[\frac{\partial \alpha}{\partial P}\right]_s}$$

ioMosaic

Resonant piping frequencies can be estimated for a constant diameter pipe if the acoustic velocity can be calculated reliably

Pipe End Conditions	f_n Hz	Wavelength
Open - Open Closed - Closed	$u_{ac} \dfrac{i}{2L}$	1/2
Open-Closed Closed-Open	$u_{ac} \dfrac{[2i-1]}{4L}$	1/4

Note that L is the effective "acoustic length" of the pipe and depends on the presence of acoustic barriers such as valves, pumps, change of flow area, etc.

Rapid opening and closing of a PRV can excite these pipe frequencies

Resonance is achieved when the PRV opening/closing frequency matches the piping frequency

This is a dangerous mechanism and should be considered separately from PRV chatter that can potential cause fatigue failure damage to PRV and piping components due to force of impact of the disk on the seat

Resonance is more likely at low flow because of the acoustic damping caused by pressure losses at high flow rates

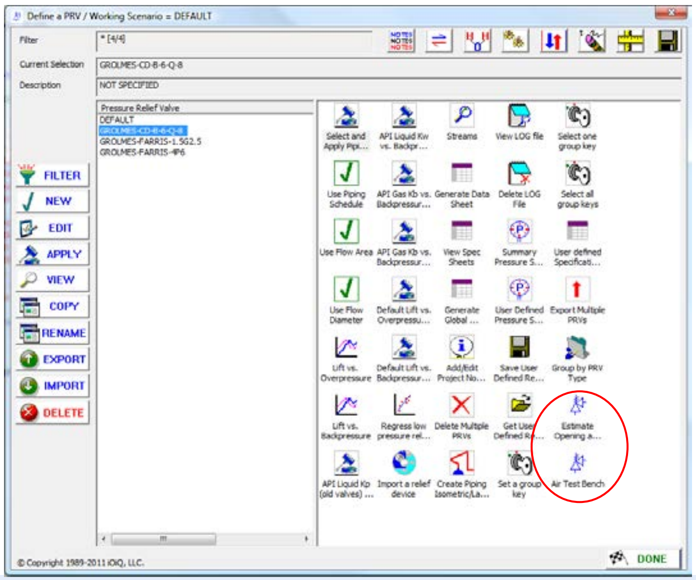
- Low flow rates cause less damping for the piping

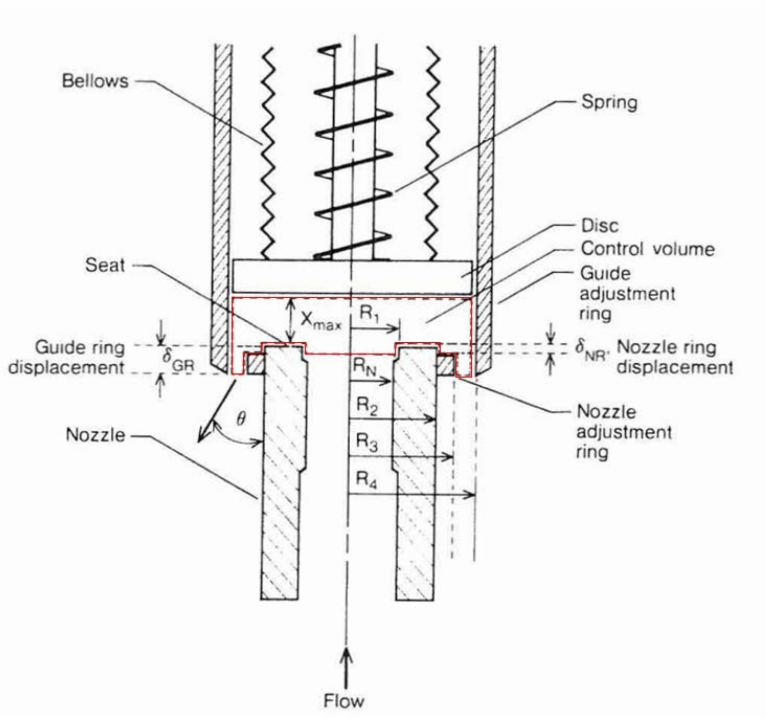
- Resonance amplification factors can be very large, 50 to 100 times

Using a single degree of freedom analysis, we can describe the motion of the valve disk

$$
\begin{aligned}
F_{NET} &= F_{Up} - F_{Dn} \\
&= \left(P_I - P_{set} + P_{atm}\right) A_I + \eta P_* \left(A_D - A_I\right) + \dot{m}^2 \left[\frac{\cos\theta}{2\pi R_1 x C_d \rho_e} + \frac{1}{\rho_I A_I}\right] \\
&\quad - P_B A_D - K_s x - \left(P_{atm} - P_B\right) A_{bel}
\end{aligned}
$$

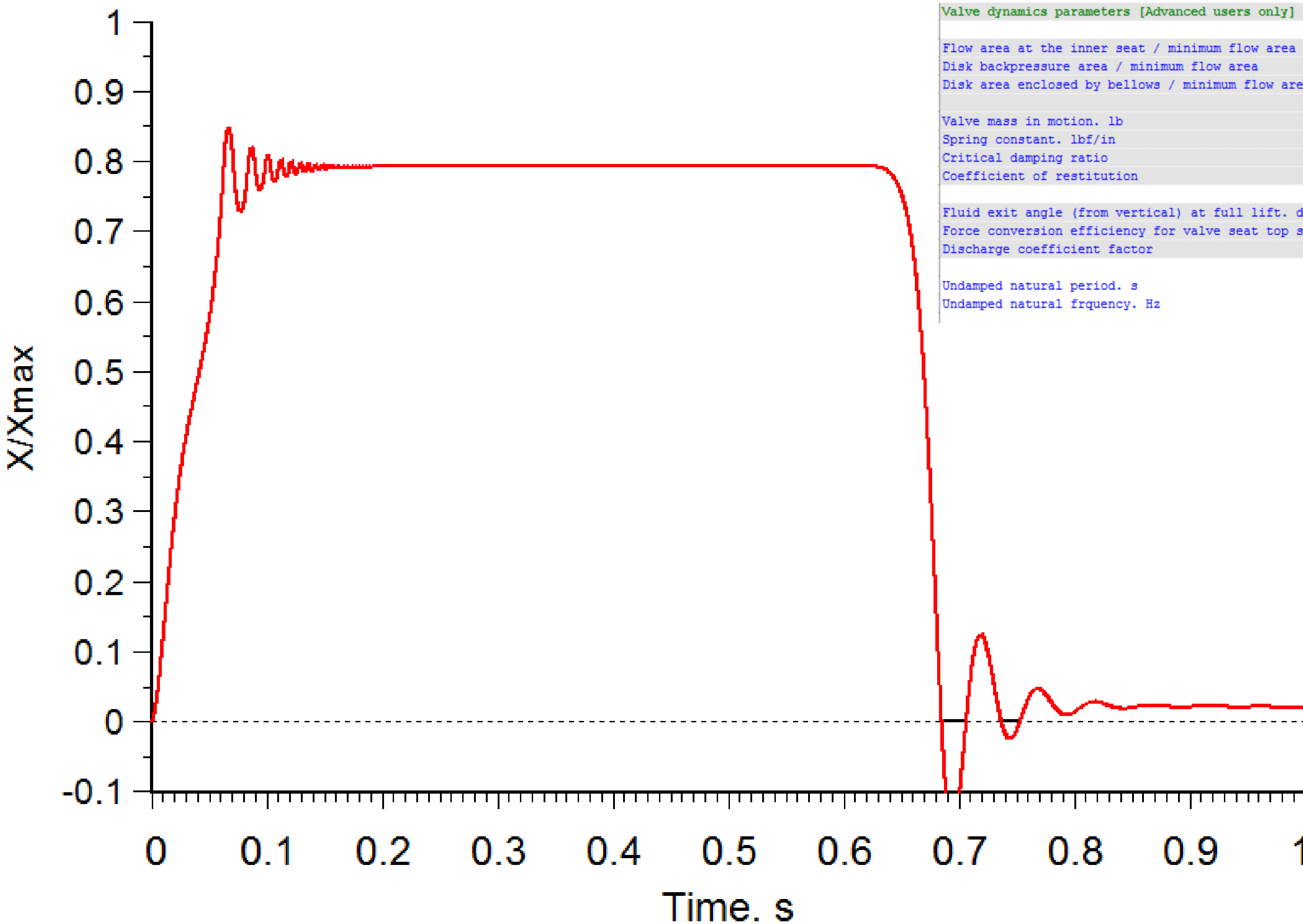
$$
\begin{aligned}
\frac{dx}{dt} &= u_d \\
m_D \frac{du_d}{dt} + C u_d &= F_{NET}
\end{aligned}
$$





ioMosaic

Let's examine the dynamics of a 5 m³ vessel full of air at 25 C and 55 psig using the Farris 4P6 PRV considered by Grolmes earlier

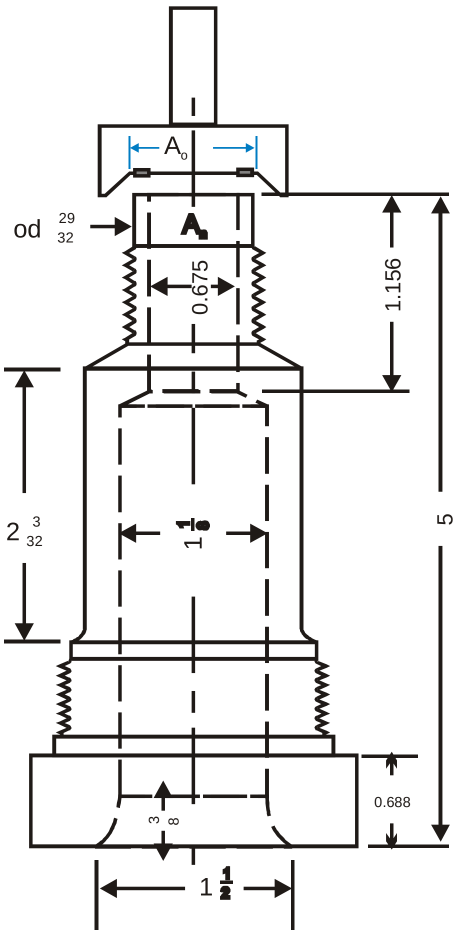


Valve dynamics parameters [Advanced users only]	
Flow area at the inner seat / minimum flow area	1.000
Disk backpressure area / minimum flow area	1.300
Disk area enclosed by bellows / minimum flow area	0.000
Valve mass in motion. lb	11.54
Spring constant. lbf/in	571.96
Critical damping ratio	0.200
Coefficient of restitution	0.010
Fluid exit angle (from vertical) at full lift. degrees	10.00
Force conversion efficiency for valve seat top surface pressure	0.60
Discharge coefficient factor	0.25
Undamped natural period. s	0.0454
Undamped natural frquency. Hz	22.0173

Source: SuperChems Expert

ioMosaic

Grolmes provided a simple correlation to calculate k and m from available valve data based on actual measurements he has conducted on numerous real valves up to 6Q8



$$k = 1.08 \frac{P_{set} \; A_{redbook}}{y_{lift}}$$

$$m = 0.025 \, M_{valve}$$

$$t_c \approx \frac{1}{2 f_n} = \pi \sqrt{\frac{m}{k}} = 0.478 \sqrt{\frac{M_{valve} y_{lift}}{P_{set} A_{redbook}}}$$

M_{valve} $\uparrow$ $\rightarrow$ $t_c \uparrow$
y_{lift} $\uparrow$ $\rightarrow$ $t_c \uparrow$
P_{set} $\downarrow$ $\rightarrow$ $t_c \uparrow$
$A_{redbook}$ $\downarrow$ $\rightarrow$ $t_c \uparrow$

Use consistent units – Pa gauge, s, kg, m, and m²

ioMosaic

If we can calculate the PRV opening and closing times, and if we can estimate the acoustic piping length, we can impose a length criteria on the inlet and discharge piping that is independent of the 3 % or the 5 % rules

- It is not easy to estimate the acoustic length of most relief piping because of the presence of expanders, reducers, etc.

- Its has been suggested that acoustic piping length can be established by following the 2:1 diameter rule

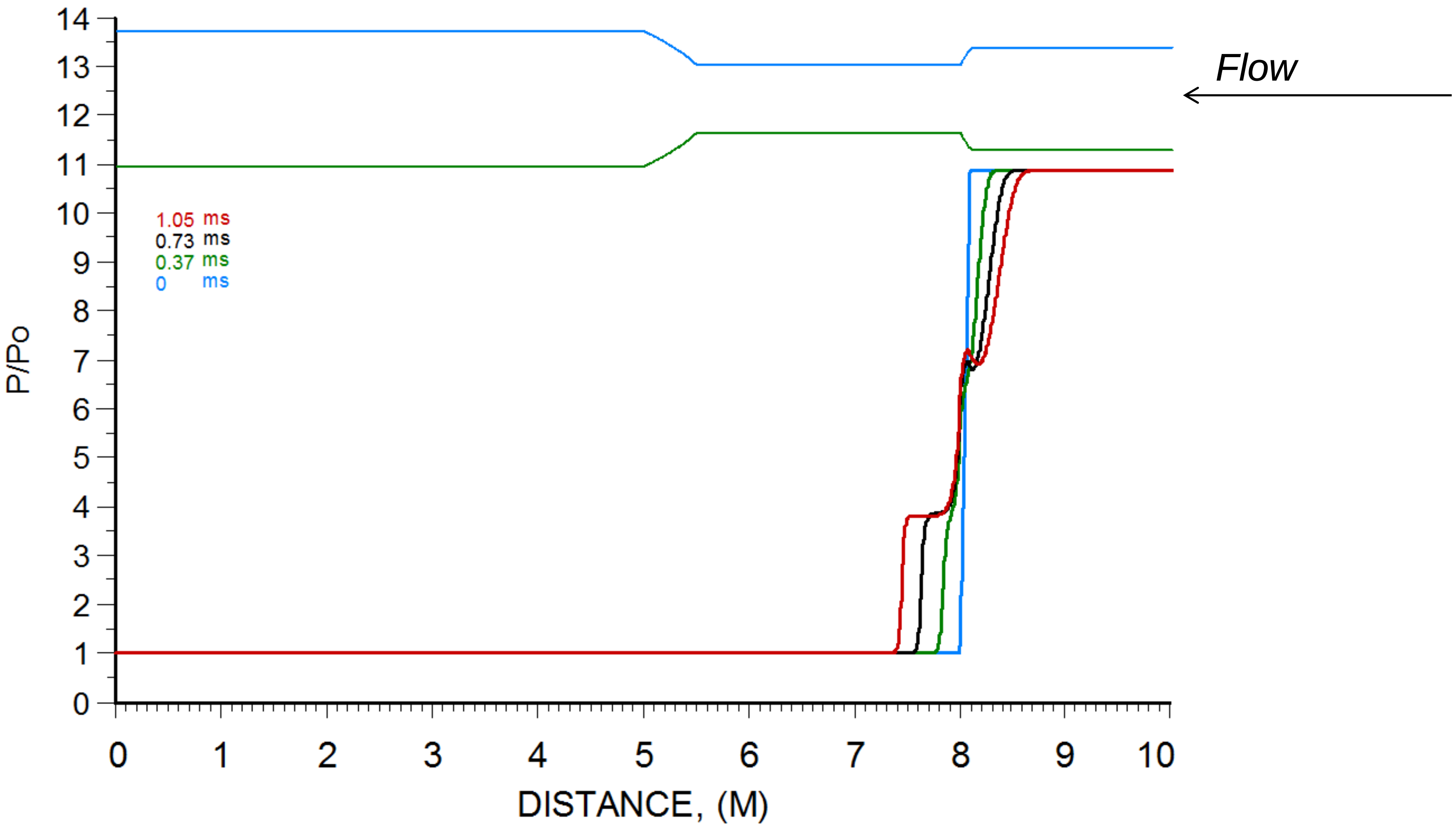
- Real piping frequencies can be much different than the theoretical frequencies that can be calculated using the simple methods outlined here. This can lead to a false sense of security and non-conservative designs or evaluations

- The best way forward is to resolve the flow in the inlet and discharge lines by solving the fluid dynamics equations coupled with the PRV SDOF equations

- This solution is easily implemented for single phase flow and is more difficult for multi-phase flow

ioMosaic

Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

ioMosaic

Consider a simplified inlet line with a single constant diameter pipe and allow for a 20 % margin between the piping frequencies and the valve frequency

- Most PRVs have an opening / closing time of approximately 25 ms (0.025 s)
- Typical acoustic velocities for liquid, gas, and two-phase systems for rigid piping are 1500, 400, and 25 m/s
- To decouple the frequencies with a 20 % margin we must have:

$$L \leq 0.8 \frac{u_{ac} t_v}{4} \quad \text{or} \quad L \geq 1.2 \frac{u_{ac} t_v}{4}$$

- Typical Liquid: **L < 7.5 m or L > 11.25 m**
- Typical Gas: **L < 2.0 m or L > 3.0 m**
- Typical Two-phase : **L < 0.13 m or L > 0.2 m**

ioMosaic

The pressure spike caused by rapid valve closure due to chatter is much more dangerous for liquid flow and least dangerous for flashing liquid flow

$$P_{\max} = P_o + \rho_o u_{\max}\left[u_o - u_c\right]$$

Example – Liquid, water like flow at 20 m/s

P_{max} = P_o + *1000*2000*[20-0]* = ***P_o + 400 bars!***

Example – Gas, air like flow

P_{max} = P_o + *5*350*[350-0]* = ***P_o + 6 bars***

Example – Two phase, water like, high void fraction, flashing flow

P_{max} = P_o + *5*30*[30-0]* = ***P_o + 0.045 bars***

ioMosaic

As the fluid moves through piping components, vortices are formed and swept into the main stream

- Vortex shedding can create standing waves
- Vortex shedding frequency (Mach No < 1.4) $f_p = \frac{N_{Str} u}{D} = 0.2 \frac{u}{D}$
- Frequency of standing waves
  - Open / Open $f = \frac{i * u_{ac}}{2L}$
  - Closed / Open $f = \frac{i * u_{ac}}{4L}$

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1 + \frac{K}{E}\psi}}$$

- If the vortex shedding frequency couples with piping components frequencies, the potential for damage can become substantial
- Note that L is the effective “acoustic length” of the pipe and depends on the presence of acoustic barriers such as valves, pumps, change of flow area, etc.

It is common to install relief devices on column overhead and process lines

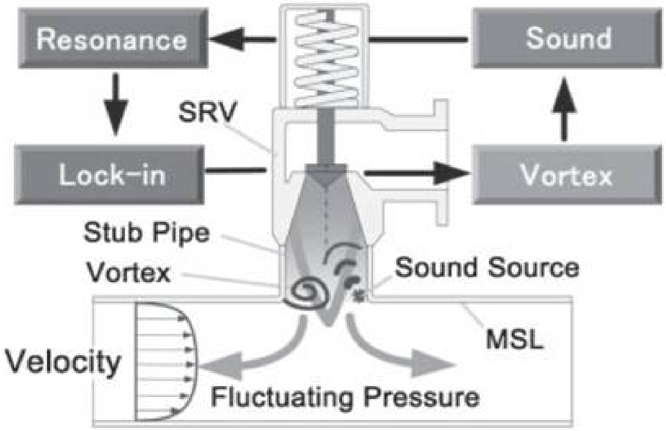
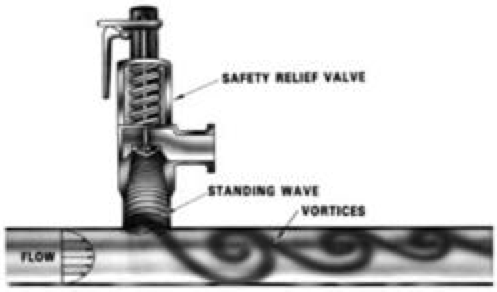
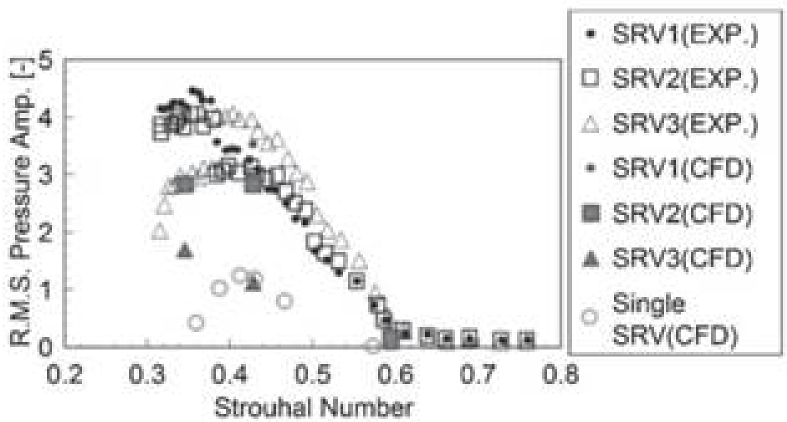


Fig. 4 Flow-induced acoustic resonance of SRVs in MSL

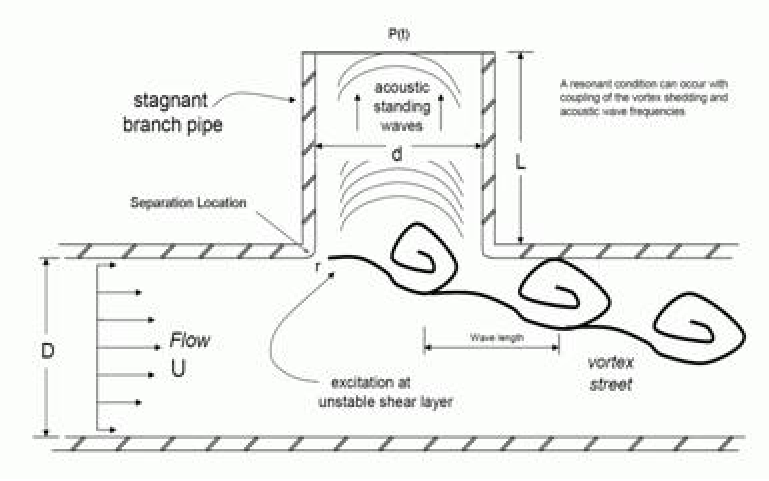
"Singing" Safety Relief Valve



Reference: Hambric, S.A., Mulcahy, T.M., Shah, V.N., et. al., "Flow-Induced Vibration Effects on Nuclear Power Plant Components Due to Main Steam Line Valve Singing," Proceedings of the Ninth NRC/ASME Symposium on Valves, Pumps, and Inservice Testing, NUREG/CP-0152, Vol. 6, pp. 3B 49-3B 69, July 2006



Increasing Velocity



ioMosaic

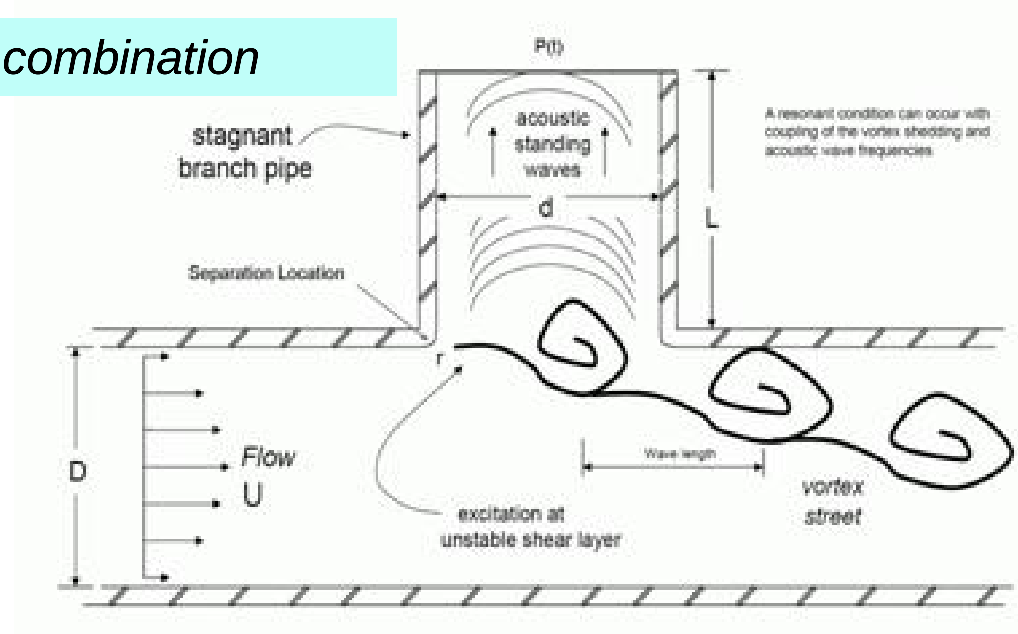
Resonance occurs when the vortex shedding frequency coincides with the acoustic frequency of the standpipe

Natural frequency of standpipe / valve combination

$$f_a = \left(\frac{2n-1}{4}\right)\frac{c_0}{L+L_e}$$

n = 1 for 1st mode, 2 for 3rd mode, etc., c_0 = fluid speed of sound, and r = radius of inlet chamfer

L_e = End correction corresponding to Rayleigh's upper limit = 0.425 d



Frequency of pressure oscillations (sound) created by vortex shedding

Vortex shedding creates pressure oscillations – the energy source for standing waves

$$N_{St} = \frac{f_s}{U}(d+r)$$

N_{St} = Strouhal Number where 0.63 >= N_{St} >= 0.3

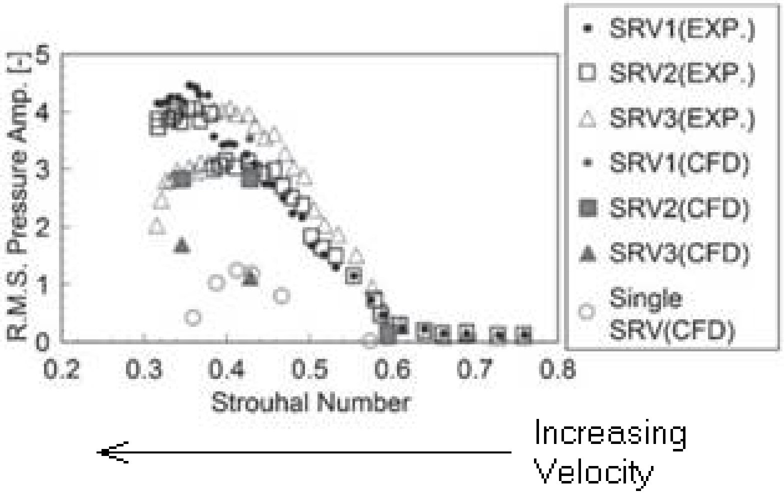
ioMosaic

Resonance can cause fatigue failure from cyclic loads and can cause leaking and chatter of the valve

$$f_s = N_{St}\left(U/D\right) \approx 0.33\left(n-0.25\right)\left(U/D\right)$$

For n = 1

$$U = \frac{f_s}{N_{St}}\left(d+r\right) = \frac{1}{4}\left(\frac{c_0}{L+L_e}\right)\left(\frac{d+r}{N_{St}}\right)$$



N_{St} = Strouhal Number where $0.63 \geq N_{St} \geq 0.3$

Peak oscillations occur around N_{St} = 0.4

RMS is the ratio of pressure oscillations divided by dynamic pressure = ½ ρu^2

RMS begins increasing at a specific onset Strouhal Number and flow velocity depending on acoustic speed, pipe diameter, and pipe length, reaches a peak value and then decreases

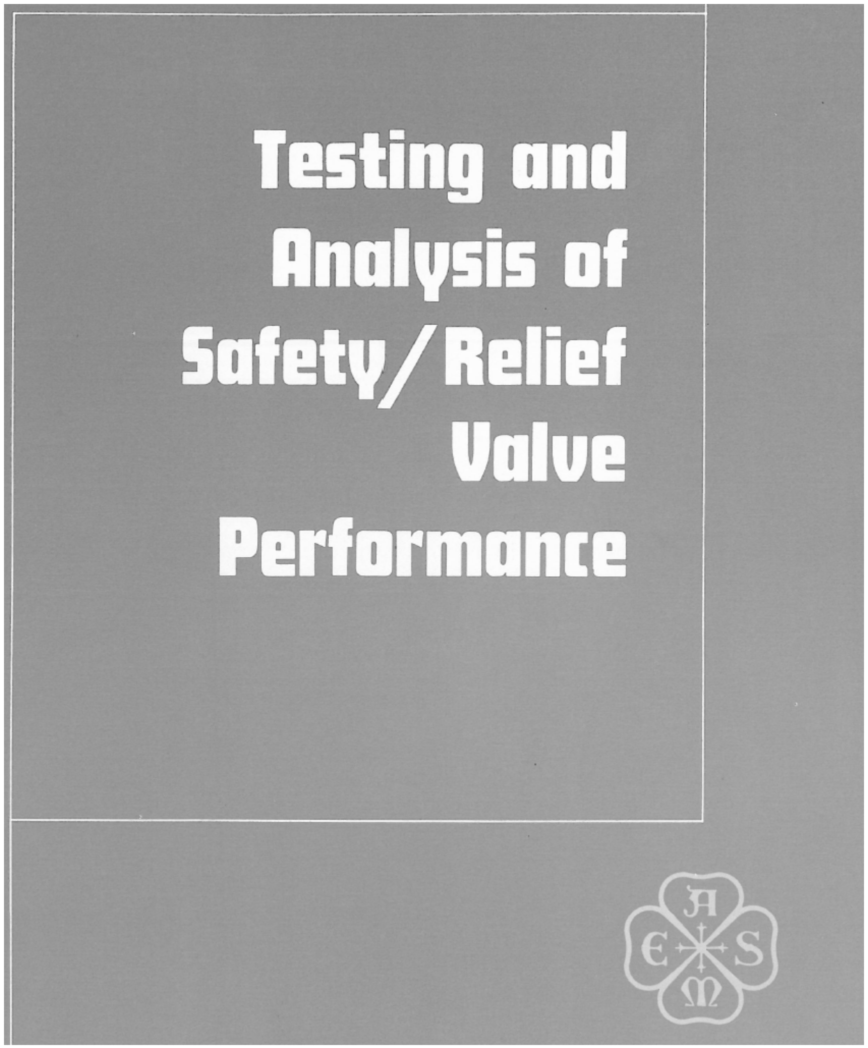
ioMosaic

The most stable relief device is one that is "slightly undersized"

- Even with gas or two-phase chatter, a system might actually survive, especially if the inlet line length is $<> 4L/u_{ac}$, regardless of how much pressure drop or what caused the valve to close
- Inlet pressure loss ONLY is not a sufficient criterion to guarantee valve stability
- Relief dynamics are required to determine relief system stability, especially for oversized valves
- Inlet line length can be restricted to ensure stable valve operation even when the inlet pressure drop is excessive
- Inlet line length limits required for stable valve operation (Shortest to Longest)
  - Flashing Two Phase → Gas → Liquid
- Relief systems damage risk from chatter (High to Low)
  - Liquid → Gas → Flashing Two Phase

ioMosaic

Key References



CONTENTS

Related Presentations

- G. A. Melhem and Harold G. Fisher, "Guidelines for dealing with excessive pressure drop in relief systems", DIERS Users Group, 2003
- G. A. Melhem, "Deal with controversial topics in pressure relief systems", DIERS Users Group, Spring 2009, Orlando, Florida.
- G. A. Melhem, "Estimate acoustically induced vibration risk in relief and flare piping", Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany
- G. A. Melhem, "Pressure Relief Valve Stability", Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

Relief Device Stability Screening

For Spring Loaded API STD 526 Valves

Dustin Smith, P.E.
Craig Powers, PhD

Smith & Burgess
Process Safety Consulting

5700 NORTHWEST CENTRAL DR. SUITE 301
HOUSTON, TX 77092
SMITHBURGESS.COM

AGENDA

- Introduction
- Background
- Goals of the screening method
- Methodology
- Sample problems
- Review compared to known data
- Recommendations for any future work
- Discussion

INTRODUCTION

Dustin Smith P.E.

PRINCIPAL ENGINEER

SMITH & BURGESS

PROCESS SAFETY CONSULTING

Craig Powers PhD

Senior Engineer

SMITH & BURGESS

PROCESS SAFETY CONSULTING

BACKGROUND - Problem

- There are known chatter incidents that resulted in a "loss of containment"
- Relatively rare occurrence
- Industry // regulatory difference of opinion on "Relaxing" the 3% rule

Historically, un-managed (or studied) change leads to increased problems

GOAL OF THE SCREENING METHODOLOGY

1. Focused on vapor / gas systems
2. Categorize installations into two buckets
   - Free from chatter
   - May chatter
3. Equations that can be done by hand
4. Relies on minimal valve specific information
5. ***All criteria must be passed***

The methodology does not predict chatter intensity, or frequency

Methodology Basis

- Based on known work
  - 80’s ASME / EPRI Research
  - 99-02 Research (From Germany)

- Validated (to date)
  - Published API Perf Data
  - Zahorsky’s ASME/ EPRI Data

Mechanisms of Chatter – Literature Review

1. Inlet line length
2. Excessive inlet pressure losses
3. Standing waves
4. Oversized relief devices
5. Improper relief device installation

All criteria must be met to be considered acceptable

Inlet Line Length – Literature Review

1. Theory
   1. Valve opens
   2. Reduced pressure area forms
   3. Pressure wave travels to some point
   4. Gets reflected back and "Supports" the disk
2. Published equation basis (Source 9)

$$t_{open} > \frac{2L}{c}; \quad \Delta P \leq \frac{2t_{wave}}{t_{open}}; \quad L_{Allowable} = f\left(\Delta P_{Chosen}, t_{open}\right)$$

Inlet Line Length – Various Equations

1. Direct solution of the basis equation

$$L < 111.5 t_{open} \sqrt{\frac{kT}{MW}}; \quad t_{open} > \frac{2L}{c}; \quad c = 223\sqrt{\frac{kT}{MW}}$$

2. Frommann & Friedel (1998, Source 6)

$$L_i < 9{,}078 \frac{d_i^2}{w_{\%O}} \left(P_s - P_B\right) t_o$$

Assumes a 20% sudden pressure loss is acceptable

Inlet Line Length – Various Equations

3. Frommann & Friedel (1998, Source 6)

$$L_i < 45{,}390 \frac{d_i^2}{w_{\%O}} \left(\frac{P_s - P_{rc}}{P_s} \right) (P_s - P_B)\, t_o$$

Assumes sudden pressure loss is limited by blowdown

4. Cremers, Friedel, Pallaks (2001, Source 9)

My implementation was not substantiated by the 99-05 PERF PRV Stability Project

Inlet Pressure Losses– Literature Review

1. Theory (EPRI / ASME)
   1. Valve opens
   2. Pressure develops (both acoustic and frictional)
   3. Valve closes (repeat)
2. Published equations (Source 32)

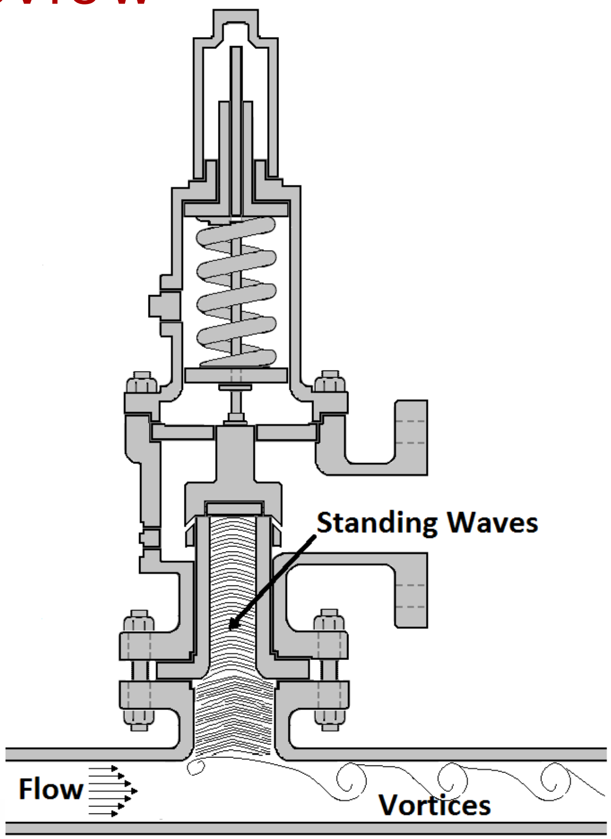
$$P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Acoustic}$$

$$\Delta P_{Acoustic} = \frac{L w_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2$$

Standing Waves – Literature Review

1. Theory
   1. High process flow velocity
   2. Vortex Shedding occurs at the tie-in point
   3. Standing waves form
2. Published equations (Source 10)

$$L_i < \frac{d_i c}{2.4U}$$



It has been speculated that Helmholtz resonance may occur (34) but generally is not considered to cause destructive chatter (35, 36).

Oversized relief devices

1. Conventional wisdom – concern when the capacity is less than 25% (Sources 22, 29)
2. Valve operation
   1. Pressure in vessel increases
   2. Valve opens, capacity depends on inlet/outlet conditions
   3. If flow to vessel is more than capacity pressure increases if not it decreases.
   4. Cycle time related to flow and volume (not only rate)

Oversized relief devices

1. If destructive chatter was caused by oversize devices:
   1. Problem would be extensive
   2. No solution
2. High frequency chatter > 1 hz (per manufacturers)

$$w_{PSV} > 4w_{required}$$

And,

$$1 > t_{cycle} = t_{P_{Blowdown} \to P_{Set}} + t_{P_{set} \to P_{Blowdown}}$$

Installation Guidelines

1. No inlet restrictions [UG-135(b)(1), Source 15]
2. No outlet restrictions / backpressure issues (Sources, 3, 9, 12, 23 ,25)
3. Balanced Bellows vents open (Source 24)
4. Pocketed outlet piping (Source 1)

Supporting Equations // Assumptions

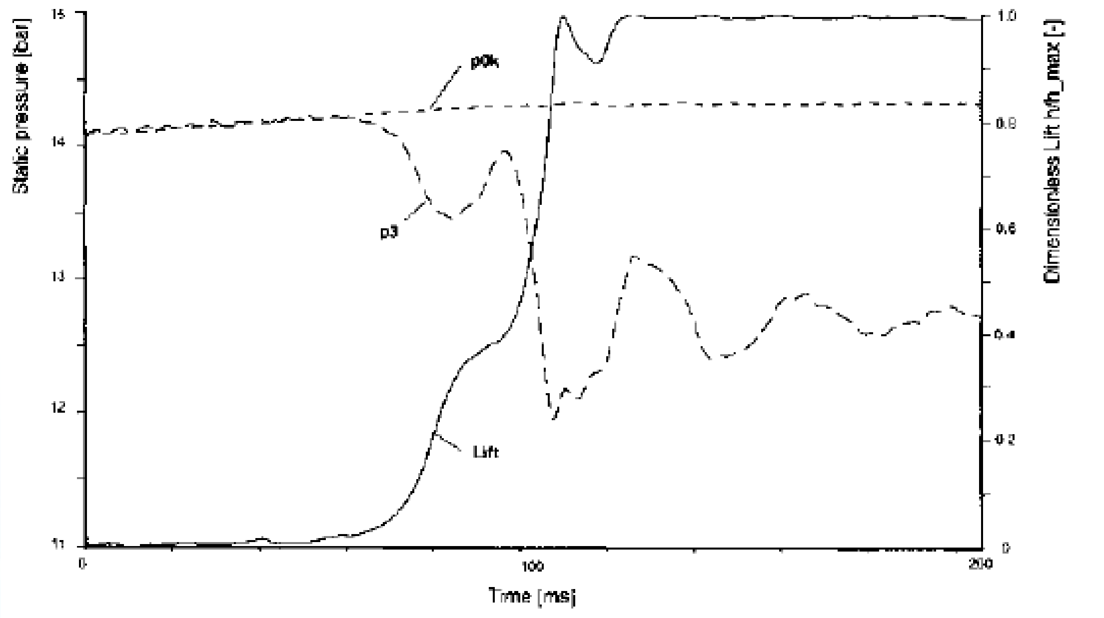
1. Relief valve opening time [Source 9]

$$t_O \approx \left(0.015 + 0.02\frac{\sqrt{2d_{PSVi}}}{\left(P_s/P_{ATM}\right)^{2/3}\left(1-P_{ATM}/P_s\right)^2}\right)\left(\frac{h}{h_{\max}}\right)^{0.7}$$

2. Speed of sound in a perfect gas (Source 29)

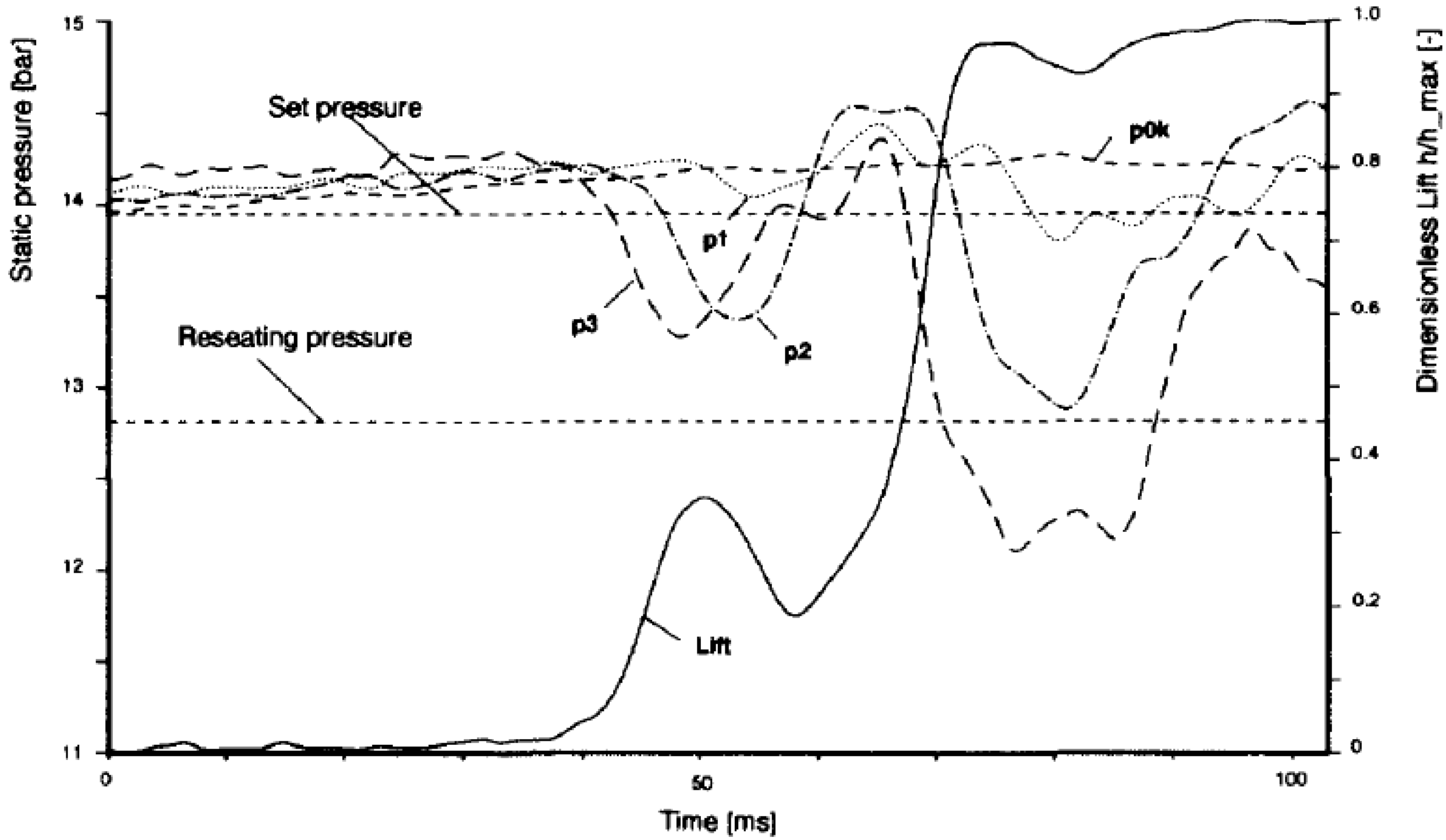
$$C = 223\sqrt{\frac{kT}{MW}}$$

3. Valve "pops" to about 60% open (Source 6)



RELIEF DEVICE STABILITY

Semi-Validated Method from Literature



Sample Problems (Criteria 1.1 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L < 111.5 t_0 \sqrt{\frac{kT}{MW}} < 111.5\left[\left(0.015 + 0.02\frac{\sqrt{2d_{PSVi}}}{(P_s/P_{ATM})^{2/3}(1-P_{ATM}/P_s)^2}\right)\left(\frac{h}{h_{\max}}\right)^{0.7}\right]\sqrt{\frac{kT}{MW}}$$

$$L < 111.5\left[\left(0.015 + 0.02\frac{\sqrt{2\times 2.1}}{(64.7/14.7)^{2/3}(1-14.7/64.7)^2}\right)(0.6)^{0.7}\right]\sqrt{\frac{1.4(85+460)}{28.8}}$$

$$L < 111.5[0.028]\sqrt{\frac{1.4(545)}{28}} < 16.1\,ft$$

Sample Problems (Criteria 1.1 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L < 111.5\left(0.015 + 0.02\frac{\sqrt{2d_{PSVi}}}{(P_s/P_{ATM})^{2/3}(1-P_{ATM}/P_s)^2}\right)\left(\frac{h}{h_{\max}}\right)^{0.7}\sqrt{\frac{kT}{MW}}$$

$$L < 111.5\left(0.015 + 0.02\frac{\sqrt{2\times 0.957}}{(264.7/14.7)^{2/3}(1-14.7/264.7)^2}\right)(0.6)^{0.7}\sqrt{\frac{1.4(85+460)}{28.8}}$$

$$L < 111.5(0.014)\sqrt{\frac{1.4(545)}{28.8}} \quad < 8.0\ ft$$

Sample Problems (Criteria 1.2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L_i < 9{,}078 \frac{d_i^2}{w_{\%O}} \left(P_s - P_B\right) t_o < 9{,}078 \frac{2.1^2}{7{,}060 \times 0.6} \left(50 - 4\right) 0.028$$

$$L < 12.2\ ft$$

Sample Problems (Criteria 1.2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L_i < 9{,}078 \frac{d_i^2}{w_{\%O}} \left(P_s - P_B\right) t_o < 9{,}078 \frac{0.957^2}{4{,}470 \times 0.6} \left(250 - 20\right) 0.014$$

$$L < 10\ ft$$

Sample Problems (Criteria 1.3 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L_i < 45{,}390 \frac{d_i^2}{w_{\%O}} \left(\frac{P_s - P_{rc}}{P_s} \right) (P_s - P_B) t_o \quad < 45{,}390 \frac{2.1^2}{7{,}060 \times 0.6} (0.08)(50-4)\, 0.028$$

$$Where, \quad blowdown = \left(\frac{P_s - P_{rc}}{P_s} \right) = 8\%$$

$$L < 4.9\ ft$$

Sample Problems (Criteria 1.3 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$L_i < 45{,}390 \frac{d_i^2}{w_{\%O}} \left(\frac{P_s - P_{rc}}{P_s} \right) (P_s - P_B) t_o < 45{,}390 \frac{0.957^2}{4{,}470 \times 0.6} (0.025)(250 - 20) 0.014$$

$$L < 1.25\, ft$$

Sample Problems (Criteria 2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Accustic}, \quad \Delta P_{Accustic} = \frac{L w_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2$$

$$C = 223\sqrt{\frac{kT}{MW}} = 223\sqrt{\frac{1.4(85+460)}{28.8}} = 1150\frac{ft}{s}, \quad \rho = \frac{P_{Set} MW}{RT} = \frac{64.7\times 28.8}{10.73\times 545} = 0.32\frac{lb}{ft^3}$$

$$\Delta P_{Accustic} = \frac{2\times 1.17}{12.6\times 2.1^2\times 0.028} + \frac{1}{10.5\times 0.32}\left(\frac{1.17\times 2}{1{,}150\times 2.1\times 0.028}\right)^2, w_{PSV} = \left(\frac{7{,}060\times 0.6}{3{,}600}\right) = 1.17\frac{lb}{s}$$

Sample Problems (Criteria 2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$\Delta P_{Accustic} = \frac{L w_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2 = 2.512 + 0.00036 = 2.5\,psi, \quad \Delta P_{Friction} = 5.1\,psi\ (measured)$$

$$P_S \times BD = P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Accustic}, \quad 50 \times 0.08 > 5.1 + 2.5$$

$$P_S \times BD > \Delta P_{Frictional} + \Delta P_{Accustic}, \quad 9\ psi > 7.6\ psi$$

Sample Problems (Criteria 2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Accustic}, \quad \Delta P_{Accustic} = \frac{L w_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2$$

$$C = 223\sqrt{\frac{kT}{MW}} = 223\sqrt{\frac{1.4(85+460)}{28.8}} = 1150\frac{ft}{s}, \quad \rho = \frac{P_{Set} MW}{RT} = \frac{264.7\times 28.8}{10.73\times 545} = 1.3\frac{lb}{ft^3}$$

$$\Delta P_{Accustic} = \frac{2\times 0.745}{12.6\times 0.92^2\times 0.014} + \frac{1}{10.5\times 1.3}\left(\frac{0.745\times 2}{1{,}150\times 0.96\times 0.014}\right)^2, w_{PSV} = \left(\frac{4{,}470\times 0.6}{3{,}600}\right) = 0.745\frac{lb}{s}$$

Sample Problems (Criteria 2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$\Delta P_{Accustic} = \frac{Lw_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2 = 15.37 + 0.001 = 15.4\, psi, \quad \Delta P_{Friction} = 22.5\, psi\ (measured)$$

$$P_S \times BD = P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Accustic}, \quad 250 \times 0.025 > 15.4 + 22.5$$

$$P_S \times BD > \Delta P_{Frictional} + \Delta P_{Accustic}, \quad 31.3\ psi > 37.9\ psi$$

Sample Problems (Criteria 2 – Line Length)

Tag Number	PSV Capacity (lb/hr)	Initial Lift (%)	Inlet Pipng (ft)	P_{Set} (PSIG)	P_{Back} (PSIG)	T_{inlet} (°F)	MW
PSV-3 (2J3)	7,060	60%	2	50.0	4	85	28.8
PSV-8 (1E2)	4,470	60%	2	250.0	20	85	28.8

$$\Delta P_{Accustic} = \frac{Lw_{PSV}}{12.6 d_i^2 t_O} + \frac{1}{10.5\rho}\left(\frac{w_{PSV} L}{c d_i t_O}\right)^2 = 15.37 + 0.001 = 15.4\, psi, \quad \Delta P_{Friction} = 22.5\, psi\, (measured)$$

$$P_S \times BD = P_S - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Accustic}, \quad 250 \times 0.025 > 15.4 + 22.5$$

$$P_S \times BD > \Delta P_{Frictional} + \Delta P_{Accustic}, \quad \cancel{31.3\ psi > 37.9\ psi}$$

Sample Problems – Summary Results

Tag Number	Eq. 1.1	Eq. 1.2	Eq. 1.3	Eq. 2.0	PERF Exp. Between
PSV-3 (2J3)	16.1	12.2	4.8	2 – 4	4 - 6
PSV-8 (1E2)	8.0	10.0	1.2	< 2	2 - 4

1. Equations 1.1 and 1.2 are "optimistic"

$$L < 111.5 t_{open}\sqrt{\frac{kT}{MW}}, Eq\,1.1;\; L_i < 9{,}078\frac{d_i^2}{w_{\%O}}(P_s - P_B)t_o, Eq\,1.2$$

2. Eq. 1.3 is most "accurate" $L_i < 45{,}390\frac{d_i^2}{w_{\%O}}\left(\frac{P_s - P_{rc}}{P_s}\right)(P_s - P_B)t_o$

3. Eq. 2.0 (Acoustic & Friction ΔP) conservative

Experimental Validation

Comparison to API PERF Study (Source 15)

Model Correlation	PERF Results	Model Prediction	Eq. 1.3	Eq. 2.0	No. Of Cases
Agreement	Chatter	Chatter	9	9	9
Agreement	Stable	Stable	26	14	14
False Negative	**Chatter**	**Stable**	**0**	**0**	**0**
False Positive	Stable	Chatter	12	24	24
Agreement[1]	Not Tested	Chatter	7	7	7
Percent Correlation %			74 (78)	49 (56)	

Note 1: *There are a number of cases that were not tested, but were assumed to chatter as the reason for not being tested was not included but assumed to be damage from previous runs.*

Experimental Validation

Comparison to the Zahorsky Data (Source 31)

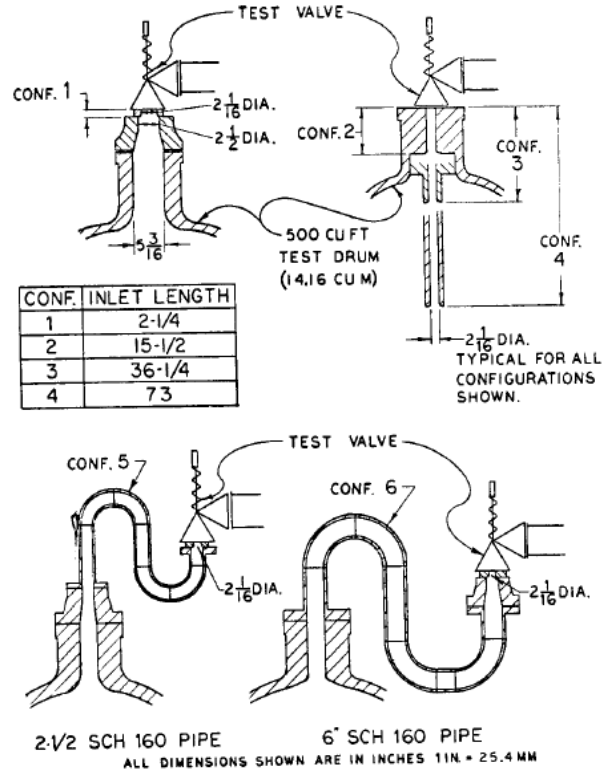


Figure 2. Schematic of Inlet Piping Configurations

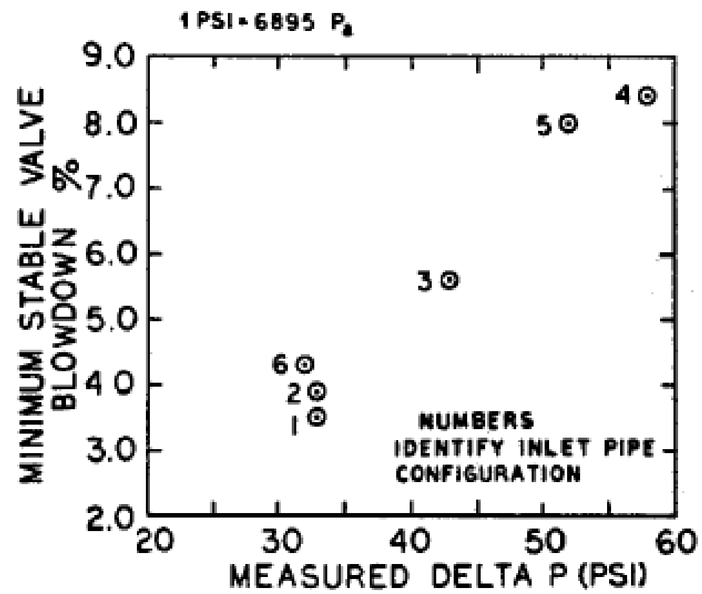


Figure 8. Relationship of Inlet Pressure Drop to Minimum Obtainable Valve Blowdown on Various Inlet Pipe Configurations

Comparison to the Zahorsky Data (Source 31)

Run	**Exp. Determined Blowdown**	**Predicted Blowdown[1]**	**Δ Blowdown (Pred. – Exp)**
1	3.9%	4.0% (0.3 / 4.0)	0.1%
2	3.9%	5.6% (2.0 / 5.6)	1.7%
3	5.6%	9.7% (4.7 / 9.7)	4.1%
4	8.4%[2]	16.7% (9.4 / 16.7)	***8.3%***
5	8.3%	12.6% (6.3 / 12.6)	4.3%
6	4.3%	5.3% (0.3 / 5.3)	1.0%

Note 1: *The values are (Eq. 1.3 / Eq. 2.0) in percent.*

2: *The only case with agreement for Eq. 1.3*

Recommendations For PERF-II

1. Is increasing PSV blowdown enough?
2. How do pipe diameter changes affect stability…

$$R_\pi = \frac{(A_1-A_2)^2}{(A_1+A_2)^2} > {}^2/_3, \text{ Reflection for acoustic boundary}$$

$$T_\pi = \frac{4A_1A_2}{(A_1+A_2)^2}, \text{ Transmission of acoustic losses}$$

3. Do acoustic losses degrade with distance?
4. What is the opening time relevant to chatter?
5. Do the valves pop to XX% open?
6. Does backpressure affect chatter? For bellows?
7. This is generic, is it conservative enough?

PRV Stability Round II

PRV Stability
Research Program

Project outline

Agenda and Objectives

- Agenda
  - Background into PERF 99-05 (Round I)
  - Next round goals and objectives
  - Progress of Round II setup
- Objectives
  - Raise awareness
  - Communicate progress
  - Garner interest

Background

- PERF 99-05 to investigate stability
  - Open project soliciting any participants
    - 12 industrial participants eventually joined
    - API was also associated by MOU
  - Petroleum Environmental Research Forum project vehicle for anti-trust reasons
  - Concluded 2012
    - Majority of calendar time was consumed by corporate legal departments

PERF 99-05 Project (Round I)

- Goals
  - Determine if API guidance provides assurance against unstable operation
    - Answer is no because of highly non-linear interactions among various parameters beyond irreversible pressure loss in the inlet line
  - Create validated engineering tools (screening criteria and software) for evaluating PRV installations for stability
    - Model created and validated, but limited

PERF 99-05 Project (Round I)

- Experimental program to confirm valve stability model
  - 18 conventional pressure relief valves
    - 3 manufacturers
    - 3 valve sizes (1E2, 2J3, 3L4)
    - 2 set pressures (50 and 250 psig)
  - Valve characteristic testing
  - Testing with inlet and outlet piping

PERF 99-05 Project (Round I)

- Recommendations from Round I
  - Estimation of key parameters
  - Testing at higher set pressures
  - Testing of other manufacturer / models
  - Investigate dynamic response at closing
- Update API RP 520 Part 2
  - Attempted qualitative caution/guidance
  - Some have expressed concerns

Round II Goals

- Some companies have expressed interest in continued research
- Develop industry guidance
  - Whether PRV installations may be subject to instability
  - Predicting consequences (e.g. damage to the valve) in the event of instability
  - Identify mitigating or corrective actions
- Address some outstanding questions

Outstanding Questions

- Frequencies and/or amplitudes of oscillating behavior without damage?
- Screening heuristic/map possible?
- Is blowdown vs. irreversible inlet losses an appropriate boundary condition in the screening? If so, what margin?
- Sufficient similarity among valve sizes for a particular mfg/model?

Outstanding Questions

- Reasonable range of values for key parameters in stability model?
- Tighter range placed on ‘acceptable’ blowdown values on production valves?
- Effect of body bowl choking?
- Effect of other relieving phases?
- Effect of restitution important?

Path Forward – Setup Phase

- Solicit potential Participants, Consociates, Informed Parties
- Parallel tracks
  - Project charter development
  - Draft participation agreements
  - Request for proposal process
- Sponsor chair – Aubry Shackelford
- Sponsorship coordinators – Clark Shepard and Dustin Smith

Current Status - Charter

- Completed tasks
  - Several interested participants identified
  - Project charter drafted
  - Sharepoint site established
- Ongoing tasks
  - Incorporating feedback on charter
  - Determine project ‘umbrella’ (PERF or CCPS)
  - Establish Organizational Task Group

Current Status - Agreements

- Completed tasks
  - Outline how to approach PERF
  - Strawman for Partnership Agreements
- Ongoing tasks
  - Outline how Consociates may be involved
  - Draft Participation Agreements
- Future tasks
  - Submit PERF request, anti-trust notifications
  - Execute Participation Agreements (critical path?)

Current Status - RFP

- Completed tasks
  - Identify how to release PERF 99-05 data
  - Identify potentially interested researchers
- Ongoing tasks
  - Secure PERF 99-05 data release
  - Strawman for RFP
- Future tasks
  - Draft and review of RFP; RFP process
  - Deliberation and selection of proposal

Budgeting

- Preliminary budget estimated
  - $40,000 - $50,000 / participant / year
  - 2-3 years anticipated
- Budget to be updated upon receipt of proposals

DIERS Users Group

- Interest in participation
  - Companies as Participants
  - DUG as Consociate or Informed Party
- Consociate is limited vs. Participant
  - Designed for groups or associations
  - Attempt to deal with funding incentives and other constraints on either party
  - Intended to be customizable

DIERS - PRV Stability Research

Clark Shepard – Chair of API SCPRS

DIERS Spring Meeting 2012 – Kansas City

Objective

Solicit active support/participation in PRV Stability Research as a project sponsor and/or technical lead that develops a research proposal for the Joint Industry Project.

Background

- Many countries including the US require operators to meet “Good Engineering Practices”. In the US, CFR 1910.119 outlines that the employer shall document that equipment complies with “**recognized and generally accepted good engineering practices” (RAGAGEP).**

- API 520 Part II provides guidance on **Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries** and is viewed by many as “Good Engineering Practice”

- API 520 Part II should outline the causes and associated design criteria to achieve PRV Stability

API SCPRS - PRV Stability

- Number of API members supported research through the Petroleum Environmental Research Forum (PERF) initiative
- API Fall '11 meeting, presentations were provided to outline the current state of research with regard to PRV Stability
  - PRV stability appears to be affected by several parameters including valve specifications, system design, and acoustical effects
  - Based on testing, mathematical models of valve lift response was possible
  - Models have been developed and testing complete but uncertainty exists for a few variables
  - Additional research is required to focus on the relationship of PRV system arrangement and stability
- SCPRS agreed to update guidance in API 520 based on best available information/science

SCPRS – Sub-Committee on Pressure Relief Systems

API 520 Part II - Update

- API 520 Part II working draft sent to committee and other stakeholders in March 2012 to develop direction for ballot
- Review of comments began during API Spring meeting (April 2012)
- Comments for draft API 520 Part II include:
  - Detail requirements for an Engineering Analysis
  - Provide additional guidance with regard to identifying PRVs with modulating characteristics
  - Understand the difference between the API guidance found in the appendix and that published by Chiyoda for analyzing PRVs in liquid service
- Comments support need for additional PRV Stability research

PRV Stability Research

Industry needs to support continued research to improve the guidance related to PRV Stability

PROPOSAL:

•Support a final phase of research to better define the causes of PRV instability and provide design criteria

•Request proposals for a Joint Industry Project (JIP)

•Form a task group to support and review JIPs

•Recognizing company/organization budget cycles beginning for 2013, fast track JIP proposals by July 2012 or delay program to 2014

Project Organization - Example

Project Technical Steering Committee (TSC)
REPRESENTATIVES FROM:
ACADEMIA VALVE MANUFACTURERS INDUSTRY

Project Sponsors and Stakeholders

Project Manger

System Mechanical Research Technical Lead

Objective:
Determine influence of PRV system mechanical design elements (e.g., inlet and outlet piping, blowdown setting, set pressure, valve size, natural frequency, valve's internal dimensions, etc.) on valve stability.

Main Activities:
1. Conduct a comprehensive literature search on the PRV instability related to PRV system mechanical characteristics.
2. Perform statistical analysis on the available test data runs investigate the possibility of establishing multi-dimensional correlations of the system mechanical parameters vs. valve stability. Techniques of model regression such as Quantitative Structure-Property Relationship (QSPR) and Super Vector Machine (SVM) may be considered.
3. Identify gaps in the available test data runs and recommend potential expanded test program, including designing the experimental program. Consider the following new experimental program elements:
- Vapor and liquid services
- Low pressure and high pressure system
- Larger than "L" orifice valves
- Variable inlet piping lengths that generates pressure drops up to 10% of the valve's set pressure
- Variable blowdown settings
- Variable system depressuring rates to evaluate stability at closing
- capture disc characteristic to understand the PRV response based on demand (i.e., modulating characteristics)
4. Conduct parametric study to simplify the mathematical model of the disc spring system to allow imperial data to solve unknown inputs.

Milestones & Deliverable:
1. Literature search report and findings (Month 2)
2. Statistical analysis & Modeling (Month 8)
3. Potential experimental program design (Month 10)
4. Experimental program – testing (Month 18)
5. Testing data analysis and statistical modeling (Month 24)
6. Mathematical modeling simplification (Month 30)
7. Stable valve system design criteria and evaluation approach (Month 36)

System Acoustic Coupling Research Technical Lead

Objectives:
Validate the adequacy of the acoustical concepts and guidelines to predict valve's instability provided in the API 520 Part II Standard and provide new guidance as needed.

Main Activities:
1. Conduct a comprehensive literature search on the PRV instability related to PRV system acoustical coupling.
2. Evaluate the PRV system during operation at required capacity and closure to understand the contribution of the following system mechanical elements on valve chattering
- Inlet line length
- Outlet line length
- Disc spring system (e.g., natural frequency)
3. Determine the inlet pipe length that may cause acoustic coupling and validate via experimental testing.

Milestones & Deliverable:
1. Literature search report and findings (Month 2)
2. Theoretical modeling and potential experimental program design (Month 8)
3. Experimental program – testing (Month 18)
4. Testing data analysis and statistical modeling (Month 24)
5. Stable valve system design criteria and evaluation approach (Month 36)

Valve Design Research (optional) Technical Lead

Objectives:
Identify valve design characteristic (e.g., bowl chocking, inlet to outlet flanges ration, etc.) that will lead to inherent instable performance and develop guidance that can be provided via the API 520 Part II Standards to identify these valves.

Main Activities:
1. Conduct a comprehensive literature search on PRV inherently instable designs.
2. Identify sample of potential instable valves by design to be included in the experimental testing program.

Milestones & Deliverable:
1. Literature search report and findings (Month 2)
2. Theoretical modeling and potential experimental program design (Month 8)
3. Experimental program – testing (Month 18)
4. Testing data analysis and statistical modeling (Month 24)
5. Stable valve system design criteria and evaluation approach (Month 36)

Roles and Responsibilities

Steering Committee
•Three members with equal voting to set direction
•Develops charter with scope and objectives
•Approves Experimental Plans
•Liaisons with sponsors
•Responsible for budget
•Semi-annual reports to sponsors
•Quarterly stewardship meetings
•Reviews monthly progress reports

Project Manager
•Responsible for the development of the experimental plans
•Responsible for testing schedule and costs
•Provides monthly progress reports to Steering Committee
•Attends quarterly stewardship meetings
•Meets monthly (F2F) with Technical Lead(s) for report out and witnessing tests as needed (9 – one day meetings)
•Reviews weekly progress reports by Technical Leads

Technical Leads
•Develops the Experimental Plan(s)
•Assembles team required to develop Experimental Plan
  – (Project Mgr, Experimental Statistician, Testing Cord., Instrument Rep.)
•Responsible for executing Experimental Plan
•Oversees testing
•Responsible for data analysis and reporting conclusions
•Develops weekly progress reports
•Meets monthly with Project Manager
•Available for quarterly stewardship meetings

SAFETY RELIEF VALVE STABILITY

PRESENTED TO THE DIERS USERS GROUP MEETING

by

HAROLD G. FISHER

INDEPENDENCE, MO

May 8, 2012

RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICE (RAGAGEP)

ELEMENTS:

1. RECOGNIZED
2. GENERALLY ACCEPTED
3. GOOD ENGINEERING PRACTICE

RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICE

“Recognized and generally accepted good engineering practices are analogous to the legal term “standard of care”. In the case of professionals, the standard of care refers to the level of care that a reasonable professional in the same circumstances would take to prevent harm or injury to another person.” (US CSB STAFF MEMBER – PSP)

“RAGAGEPs encompass the whole body of guidelines, standards, generally accepted principals (both taught in school and learned from others) that establish the ways in which responsible engineers accomplish their tasks.” (US CSB STAFF MEMBER – PSP)

RAGAGEPs provide guidance on engineering, operating, and maintenance practices developed by industry and technical experts. They are usually based on and / or revised to reflect, lessons learned from industry. The most common RAGAGEP references by OSHA in RNEP inspections … are codes and standards from the American Society of Mechanical Engineers (ASME), the American Petroleum Institute (API), and the publications from the Center of Chemical Process Safety (CCPS).” (ABSG CONSULTING, INC. – PSP)

RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICE (US CSB STAFF MEMBER - PSP)

"Licensed professional engineers are bound by regulations established by the states in which the engineers are registered. A sampling of states reveals that most states include codes of ethics and principles of engineering practice into the states laws for professional engineers. For example, the Texas regulation states

Engineers shall not perform any engineering function which, when measured by generally accepted engineering standards or procedures, is reasonable likely to result in the endangerment of lives, health, safety, property, of welfare of the public."

"The Engineering Code of Ethics contains a requirement for engineers to practice within their area of expertise, and to ensure their designs meet standards and established practices. This code of ethics is a legal requirement for licensed engineers."

"Engineers shall approve only those engineering documents that are in conformity with applicable standards." "Each engineers has a responsibility to practice only within his / her area of expertise and to approve only those designs that comply with applicable standards." Knowledge of codes and standards of applicability to ones work is a hallmark of engineering competency."

PRESSURE RELIEF DEVICE RAGAGEP

ASME BOILER AND PRESSURE VESSEL CODE (BPVC)

API RP 520, PART II – INSTALLATION

ISO4126-1 – SAFETY RELIEF VALVES
ISO4126-3 – SAFETY RELIEF DEVICES
ISO4126-9 – SAFETY DEVICES
ISO4126-10 – SAFETY RELIEF VALVES

ASME BOILER AND PRESSURE VESSEL CODE (BPVC)

PRESSURE RELIEF VALVES (UG-134)

CHATTERING – NOT ALLOWED

FLUTTERING – NOT ALLOWED

"Pressure relief valves SHALL be designed and constructed such that when installed per UG-135, the valves will operating without Chattering and SHALL not Flutter at the flow-rated pressure in a way that would either interfere with the measurement capacity or would result in damage."

ASME BOILER AND PRESSURE VESSEL CODE (BPVC)

CERTIFICATION OF CAPACITY (UG-131)

"Pressure relief valves for compressible fluids having an adjustable blowdown construction SHALL be adjusted prior to testing so that the blowdown does not exceed 5% of the set pressure of 3 psi (20 kPa) whichever is greater."

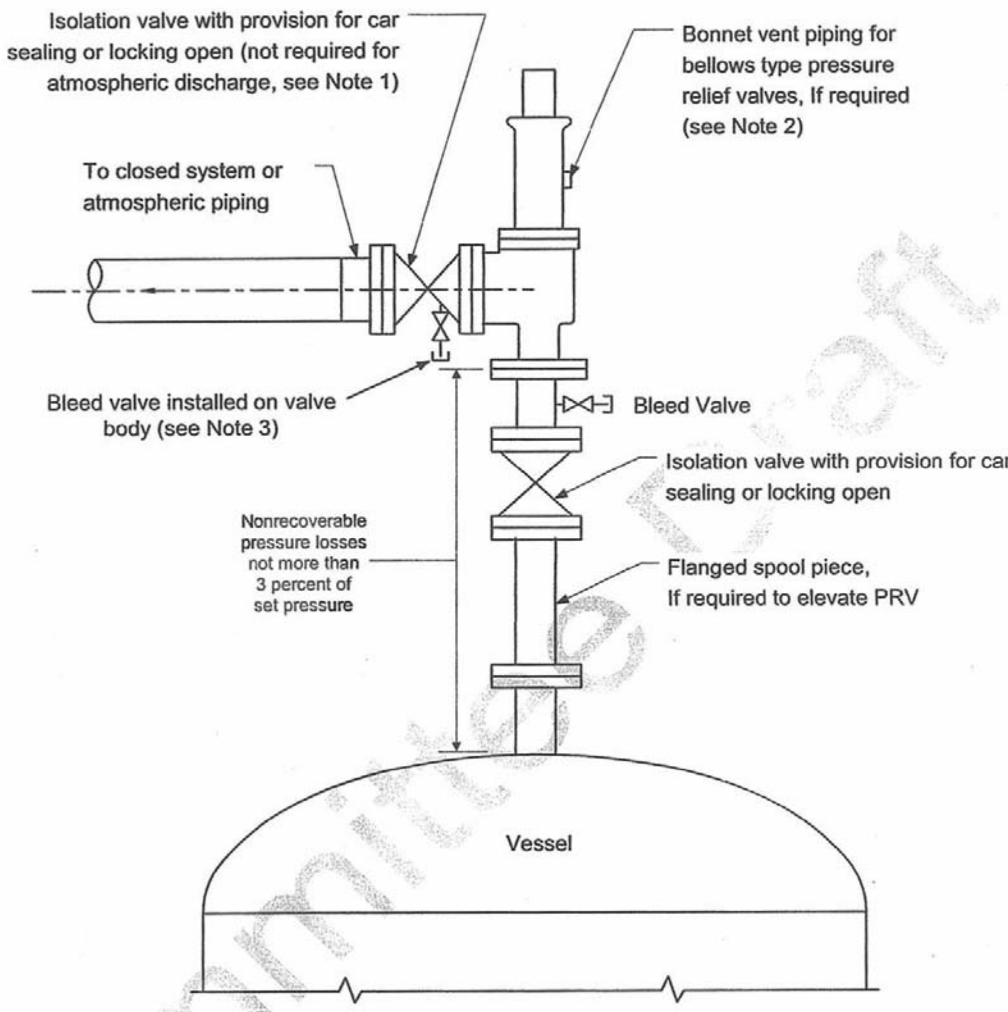
OPERATIONAL AND CAPACITY TESTING (UG-136)

"If a pressure relief valve with adjustable construction selected from a Manufacturer exhibits a blowdown that exceeds 7% of the set pressure or 3 psi (20 kPa), then an adjustment SHALL be made to meet this requirement."

SAFETY RELIEF VALVE INLET PIPE IRREVERSIBLE PRESSURE LOSS RULE (3% RULE)

ASME BPVC – APPENDIX M-6

"The nominal size of all piping, valves and fittings, and vessel components between a pressure vessel and its safety, safety relief, or pilot operated pressure relief valves SHALL be at least as large as the nominal size of the device inlet, and the flow characteristics of the upstream system SHALL be such that the cumulative total of all nonrecoverable inlet losses SHALL not exceed 3% of the valve set pressure. The inlet losses will be based on the valve nameplate capacity corrected for the characteristics of the flowing fluid."



Notes:

1. See Section 6 for the use of isolation valves in pressure relief system piping
2. See Section 7.
3. Alternatively, a pipe spool with bleed may be provided

Figure 10 – Typical Pressure Relief Device Installation with an Isolation Valve

SRV INLET PIPE IRREVERSIBLE PRESSURE LOSS (ENLARGED PIPE DIAMETER REQUIRED)

COX AND WEIRICK. CEP (NOVEMBER 1980)

4P6 / 6R8 / 6R10 / 8T10	ALWAYS
SRV WITH RD IN SERIES	ALWAYS
1.5H3 / 2J3 / 3L4 / 6Q8	WITH SHUTOFF VALVE AND L/D GT 5 AND VESSEL SQUARE-EDGED NOZZLE

SRV DISCHARGE PIPE BACK PRESSURE (ENLARGED PIPE DIAMETER REQUIRED)

6R8	ALWAYS
3L4 / 4P6 / 8T10	CONVENTIONAL SRV, PSET GT 100 PSIG, AND LENGTH GT 10 FEET

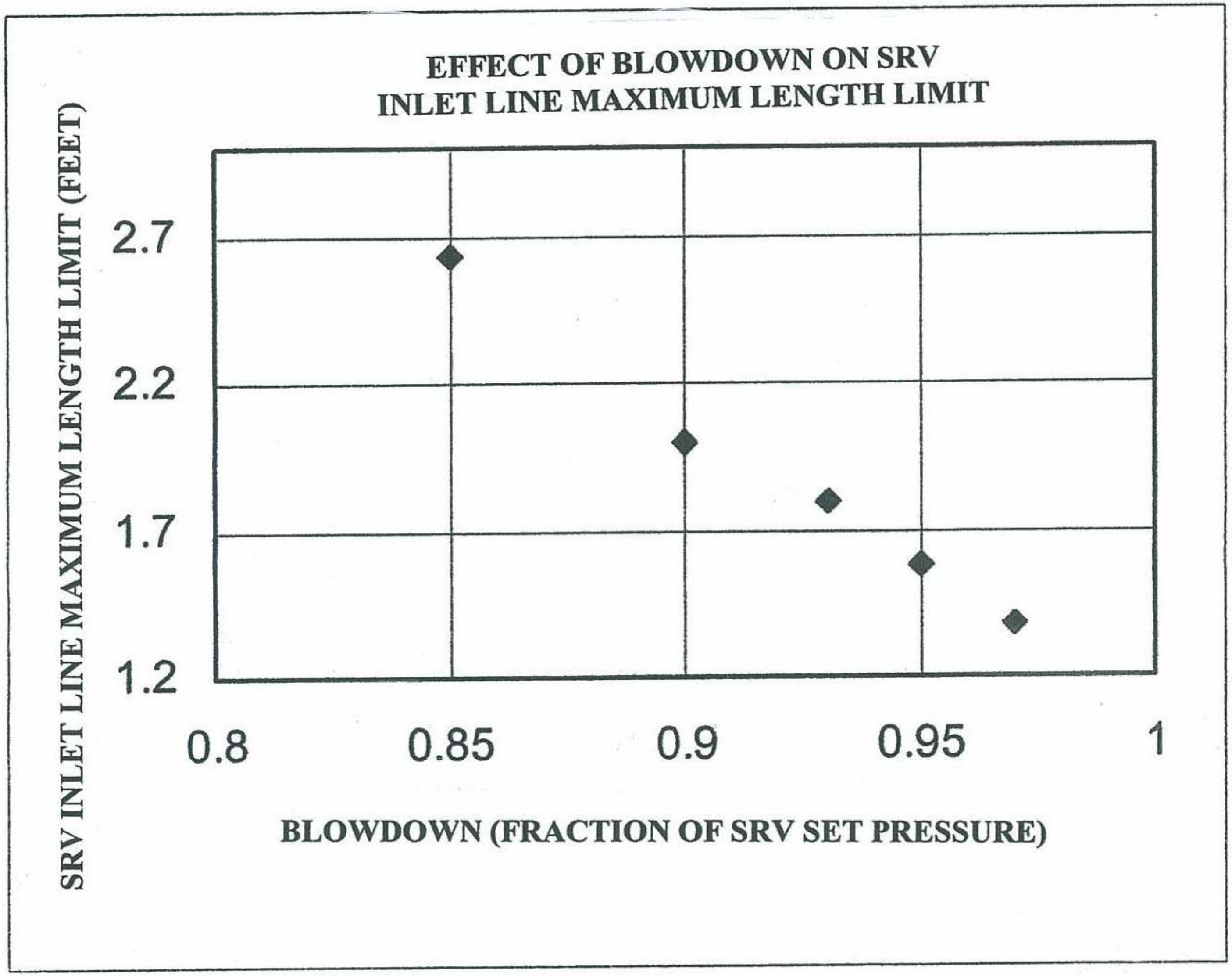
SAFETY RELIEF VALVE DESIGN PROBLEM

CROSBY JOS CONVENTIONAL 1.5H3 (0.887 SQUARE INCHES)

INLET PIPE:	DIAMETER = 1.338 INCHES (S160)
	LENGTH = 1.8 FEET
	FITTINGS – NONE
DISCHARGE PIPE:	DIAMETER = 2.624 INCHES (S160)
	LENGTH = 10 FEET
	FITTING – NONE
SET PRESSURE:	138 PSIG
FLOW:	1.1 PSET (166.5 PSIA)
FLUID:	AIR @ 70°F (21.1°C)
BLOWDOWN:	7%

SAFETY RELIEF VALVE DESIGN PROBLEM RESULTS

	FLOW (PPH)	IPD (PSIG)	IDP (%)	DBP (PSIG)	DBP (%)
SQUARE ENTRANCE	9,190	11.6	8.4	9.7	7.0
IDEAL ENTRANCE	9,555	5.0	3.7	10.5	7.6
PIPE ROUGHNESS (ε = 0.0001)	9,579	4.6	3.3	10.0	7.2
PIPE ROUGHNESS (ε = 0.00005)	9,613	4.0	2.9	9.3	6.8


EFFECT OF BLOWDOWN ON SRV
INLET LINE MAXIMUM LENGTH LIMIT
SRV INLET LINE MAXIMUM LENGTH LIMIT (FEET)
2.7
2.2
1.7
1.2
0.8
0.85
0.9
0.95
1
BLOWDOWN (FRACTION OF SRV SET PRESSURE)

SAFETY RELIEF VALVE INLET PIPE IRREVERSIBLE PRESSURE LOSS RULE (3% RULE)

ASME BPVC – COMMITTEE CORRESPONDANCE

1. Now is not the time to open up the long-standing 3% Rule to the use of unsubstantiated, untested methodologies that have a high probability of ruining the safety record of the 3% Rule that provides the required pressure integrity when the valve is called upon to activate.

2. Blowdown is neither tested nor reported by the National Board or the valve manufacturers or the end user. Approval for the use of an engineering analysis must be based on a recommendation of the ASME Committee of Safety Valve Requirements (BPV-SRV) developed via the ASME consensus process.

SAFETY RELEF VALVE BLOWDOWN DATA – ASME SECTION VIII

CONVENTIONAL SAFETY RELIEF VALVES

DOSSENA (VALVE WORLDS, 2002)

5.3, 6.7, AND 7.3 % OF SET PRESSURE

API PERF STUDY PRESENTATION (NOVEMBER 2011)

CONSOLIDATED 1900, FARRIS 2600, CROSBY JOS

1.5, 4.3, 4.4, 5.3, 5.6, 5.6, 5.6, 5.7, 6.4, 7.1, 7.6, 8.0, 9.1, 9.9, 10.6, 11.8, 11.9 % OF SET PRESSURE

SAFETY RELIEF VALVE BLOWDOWN DATA – ASME SECTION VIII

PATENT 5,515,884 (May 14, 1996) – DRESSER INDUSTRIES, INC.

Provide Chatter free operation with a blowdown of less than 10 percent for all fluids (gas, vapor, and liquid) flowing through the valve.

PATENT 7,744,071 (June 29, 2010) – MERCER VALVE COMPANY, INC.

Low blowdown valves will have a blowdown valve of about 15% or less, preferably 10% or less.

A particularly preferred valve will have a blowdown of 5 to 10% of set pressure.

Low blowdown valves are desirable because they can minimize the amount of gas that is lost from the pressurized system into the atmosphere during venting, thereby addressing environmental concerns.

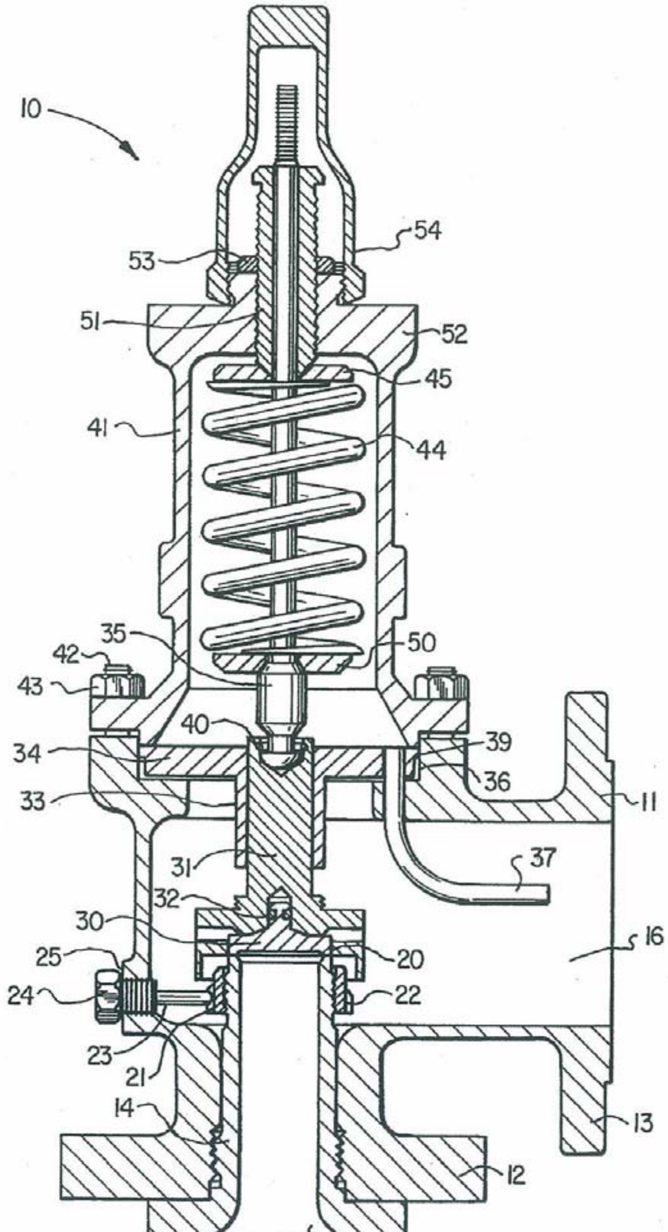

10
54
53
51
52
45
41
44
35
50
42
43
40
39
34
36
33
11
31
37
32
30
16
25
20
24
22
23
21
14
13
12
15

FIG. 1

SAFETY RELIEF VALVE BLOWDOWN DATA – ASME SECTION VIII

PATENT APPLICATION US 2011/0284092 (May 14, 1996) – DRESSER, INC.

CONSOLIDATED MODEL 1900LA CONSOLIDATED DUAL PRV 100

TABLE 3

Performance Comparison of PRV 100 to Conventional Pressure Relief Valve for Liquid Application

Conventional Pressure Relief Valve designed for Liquid Application				PRV 100			
Media	set pressure (psi)	closing pressure (psi)	% Blowdown	Media	set pressure (psi)	closing pressure (psi)	% Blowdown
Water	81.8	72.9		Water	101.8	96.9	
	81.8	76.2			102.1	98.8	
	81.8	76.1			102.4	97.2	
Average	81.8	75.1	8.2%	Average	102.1	97.6	4.4%
Air	80.2	67.9		Air	103.1	96.9	
	80.1	67.4			102.7	96.8	
	80.3	68.4			102.5	96.8	
Average	80.2	67.9	15.3%	Average	102.8	96.8	5.8%
Water	156.5	144.0		Water	189.2	179.7	
	155.7	143.3			189.4	178.7	
	155.5	143.1			189.3	179.8	
	155.5	143.2					
Average	155.8	143.4	8.0%	Average	189.3	179.4	5.2%
Air	152.2	125.0		Air	189.2	177.9	
	152.3	125.4			189.1	176.3	
	152.2	125.1			189.1	175.2	
Average	152.2	125.2	17.8%	Average	189.1	176.5	6.7%

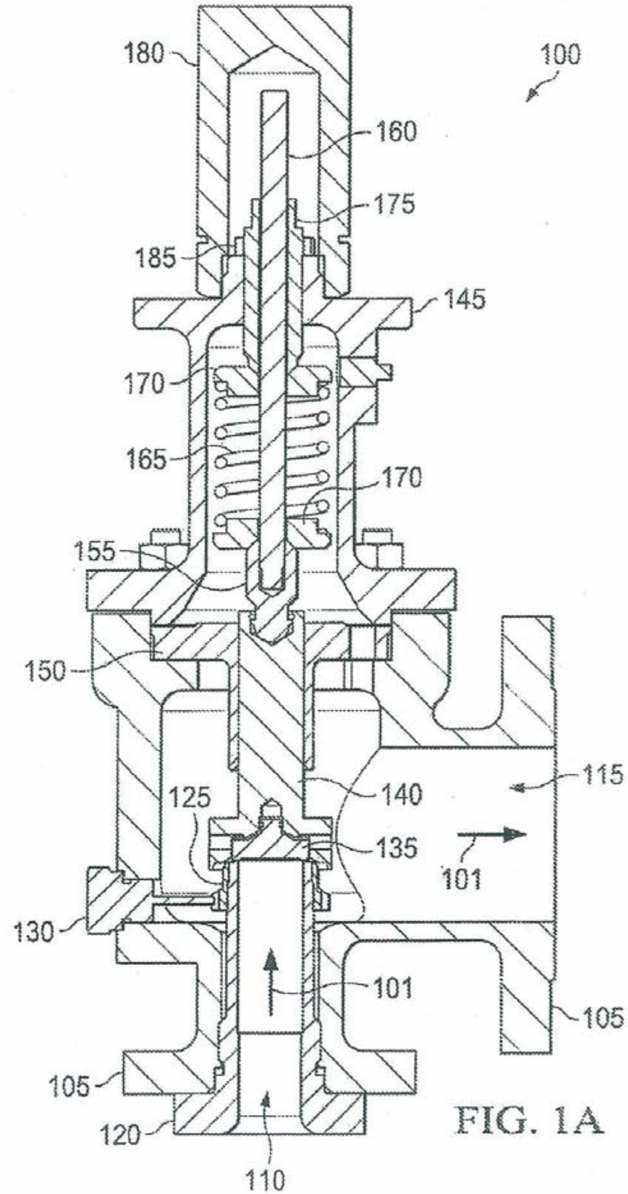

180
100
160
175
185
145
170
165
170
155
150
115
140
125
135
101
130
101
105
105
120
110

FIG. 1A

Development of Spring Loaded Safety Valves

Fall 2007 DIERS

Roger D. Danzy

Dresser Inc.

MANCHESTER, NH (OCTOBER 8 – 10, 2007)

Testing Program Results

Valve Number	Valve Size & Set Pressure	Inlet Piping Length and Pressure Drop						Measured Blowdown
		2 Feet Valve Initial Lift (Test)	Inlet Pressure Drop % of Set Pressure	**4 Feet Valve Initial Lift (Test)**	Inlet Pressure Drop % of Set Pressure	**6 Feet Valve Initial Lift (Test)**	Inlet Pressure Drop % of Set Pressure	
1	1E2 - 50 psig	Stable	1.2	Stable	2.0	Stable*	2.7	5.3
2	1E2 - 250 psig	Stable	1.0	Unstable	1.7	Unstable	**2.2**	**5.6**
3	2J3 - 50 psig	Stable	2.2	Stable	3.2	Unstable	**4.3**	**8.0**
4	2J3 - 250 psig	Stable	2.0	Test Failed	3.0	Not Tested	4.0	7.6
5	3L4 - 50 psig	Stable	2.3	Stable	3.3	Stable	**4.5**	**4.3**
6	3L4 - 250 psig	Stable	1.8	Stable	2.7	Not Tested	3.5	11.9
7	1E2 - 50 psig	Stable	1.1	Stable	1.9	Stable	2.6	7.1
8	1E2 - 250 psig	Stable	**1.0**	Unstable	1.7	Unstable	2.3	**2.5**
9	2J3 - 50 psig	Stable	2.3	Stable	3.5	Stable	4.6	9.1
10	2J3 - 250 psig	Stable	1.9	Not Tested	2.8	Not Tested	3.8	6.4
11	3L4 - 50 psig	Stable	2.3	Stable	3.3	Stable	**4.6**	**5.6**
12	3L4 - 250 psig	Stable	1.8	Not Tested	2.7	Not Tested	3.6	5.7
13	1E2 - 50 psig	Stable	1.1	Stable	1.9	Stable	2.7	9.9
14	1E2 - 250 psig	Stable	1.0	Stable	1.7	Stable	2.2	6.4
15	2J3 - 50 psig	Stable	2.3	Stable	3.4	Stable	4.5	11.8
16	2J3 - 250 psig	Stable	1.9	Not Tested	2.8	Not Tested	3.7	5.6
17	3L4 - 50 psig	Stable	2.5	Stable	3.5	Stable	4.6	10.6
18	3L4 - 250 psig	Stable	1.8	Unstable	2.7	Not Tested	3.5	4.4

Figure 1 B

API PERF 99-05 PRV Stability Experimental Project Overview [31] (2011)

Test No. Pipe Length (Feet)	Valve Size Pset (psig)	Result	Blowdown (Psi)	Inlet Pipe Loss (Psi)	BD - IDL (Psi)
1 - 2	1E2 - 50	S	5.3	1.2	4.1
1 - 4		S		2.0	3.3
1 - 6		S		2.7	2.6
7 - 2	1E2 - 50	S	7.1	1.1	6.0
7 - 4		S		1.9	5.2
7 - 6		S		2.6	4.5
13 - 2	1E2 - 50	S	9.9	1.1	8.8
13 - 4		S		1.9	8.0
13 - 6		S		2.7	7.2
2 - 2	1E2 - 250	S	5.6	1.0	4.6
2 - 4		U		1.7	3.9
2 - 6		U		2.2	3.4
8 - 2	1E2 - 250	S	2.5	1.0	1.5
8 - 4		U		1.7	0.8
8 - 6		U		2.3	0.2
14 - 2	1E2 - 250	S	6.4	1.0	5.4
14 - 4		S		1.7	4.7
14 - 6		S		2.2	4.2

S - Satisfactory Performance

U - Unsatisfactory Performance

Figure 1 C

API PERF 99-05 PRV Stability Experimental Project Overview [31] (2011)

Test No. Pipe Length (Feet)	Valve Size Pset (psig)	Result	Blowdown (Psi)	Inlet Pipe Loss (Psi)	BD - IDL (Psi)
3 - 2	2J3 - 50	S	8.0	2.2	5.8
3 - 4		S		3.2	4.8
3 - 6		U		4.3	3.7
9 - 2	2J3 - 50	S	9.1	2.3	6.8
9 - 4		S		3.5	5.6
9 - 6		S		4.6	4.5
15 - 2	2J3 - 50	S	11.8	2.3	9.5
15 - 4		S		3.4	8.4
15 - 6		S		4.5	7.3
4 - 2	2J3 - 250	S	7.6	2.0	5.6
4 - 4		Test Failed		3.0	4.6
4 - 6		Not Tested		4.0	3.6
10 - 2	2J3 - 250	S	6.4	1.9	4.5
10 - 4		Not Tested		2.8	3.6
10 - 6		Not Tested		3.8	2.6
16 - 2	2J3 - 250	S	5.6	1.9	3.7
16 - 4		Not Tested		2.8	2.8
16 - 6		Not tested		3.7	1.9

S - Satisfactory Performance

U - Unsatisfactory Performance

Figure 1 D

API PERF 99-05 PRV Stability Experimental Project Overview [31] (2011)

Test No. Pipe Length (Feet)	Valve Size Pset (psig)	Result	Blowdown (Psi)	Inlet Pipe Loss (Psi)	BD - IDL (Psi)
5 - 2	3L4 - 50	S	4.3	2.3	2
5 - 4		S		3.3	1
5 - 6		S		4.5	- 0.2 **
11 - 2	3L4 - 50	S	5.6	2.3	3.3
11 - 4		S		3.3	2.3
11 - 6		S		4.6	1.0
17 - 2	3L4 - 50	S	10.6	2.5	8.1
17 - 4		S		3.5	7.1
17 - 6		S		4.6	6.0
6 - 2	3L4 - 250	S	11.9	1.8	10.1
6 - 4		S		2.7	9.2
6 - 6		Not Tested		3.5	8.4
12 - 2	3L4 - 250	S	5.7	1.8	3.9
12 - 4		Not Tested		2.7	3.0
12 - 6		Not Tested		3.6	2.1
18 - 2	3L4 - 250	S	4.4	1.8	2.6
18 - 4		U		2.7	1.7
18 - 6		Not Tested		3.5	0.9

S - Satisfactory Performance

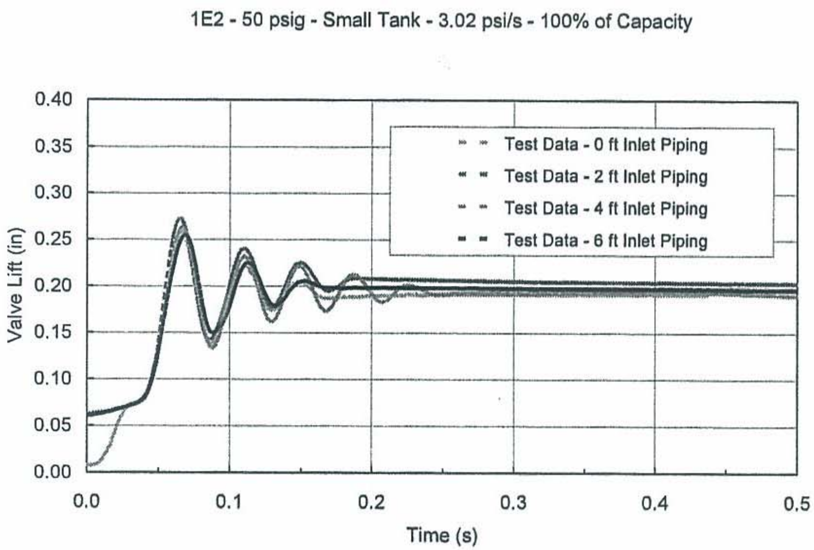
U - Unsatisfactory Performance

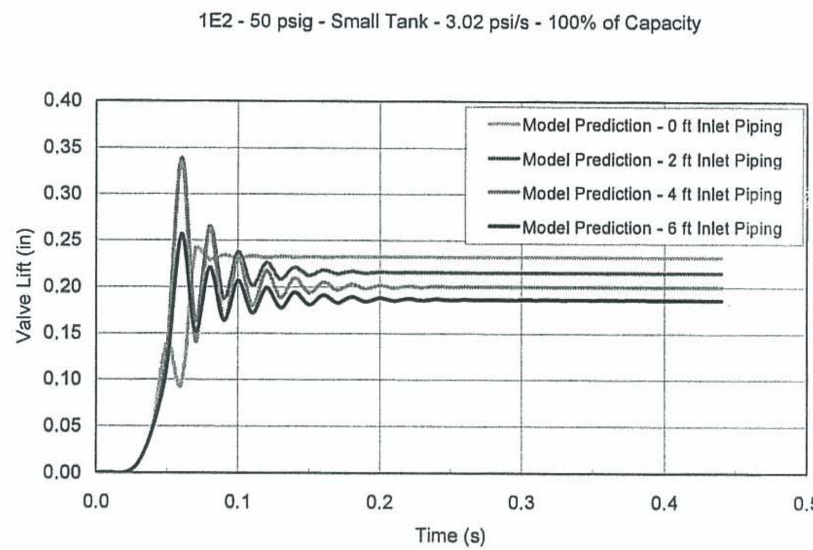
** Impossible

1E2 Conventional Safety Relief Valve

Flowing Pressure (50 Psig)			Flowing Pressure (250 Psig)		
BD - IPL (Psi)	Pipe Length (Feet)	Operation	BD - IPL (Psi)	Pipe Length (Feet)	Operation
2.6	6	S	0.2	6	U
3.3	4	S	0.8	4	U
4.1	2	S	1.5	2	S
4.5	6	S	3.4	6	U
5.2	4	S	3.9	4	U
6.0	2	S	4.2	6	S
7.2	6	S	4.6	2	S
8.0	4	S	4.7	4	S
8.8	2	S	5.4	2	S

Model Prediction - Examples

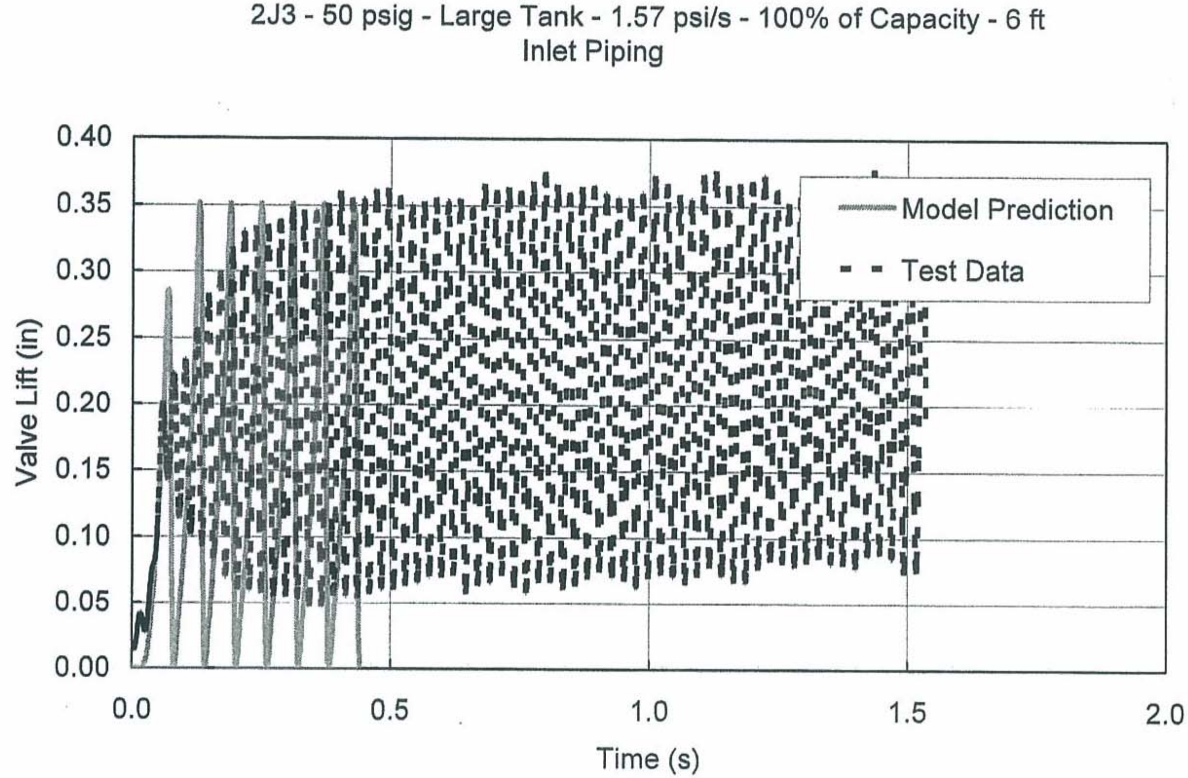




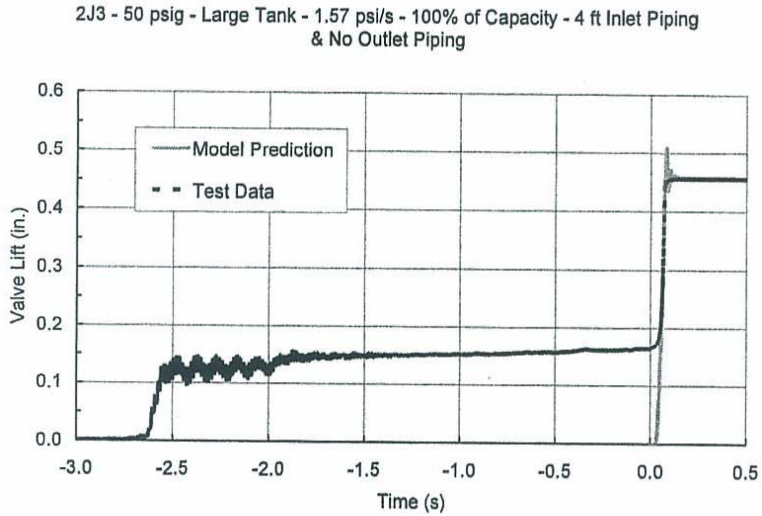
2J3 Conventional Safety Relief Valve

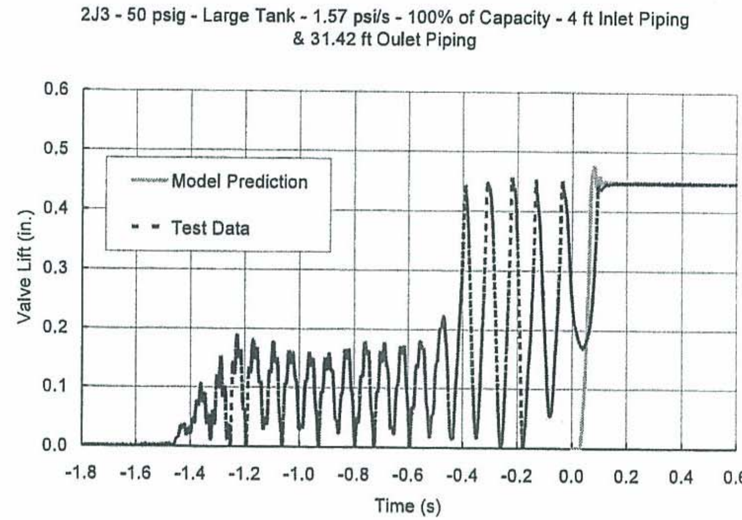
Flowing Pressure (50 Psig)			Flowing Pressure (250 Psig)		
BD - IPL (Psi)	Pipe Length (Feet)	Operation	BD - IPL (Psi)	Pipe Length (Feet)	Operation
3.7	6	U	1.9	6	NT - U
4.5	6	S	2.6	6	NT - U
4.8	4	S	2.8	4	NT - U
5.6	4	S	3.6	6	NT - U
5.8	2	S	3.6	4	NT - U
6.8	2	S	3.7	2	S
7.3	6	S	4.5	2	S
8.4	4	S	4.6	4	FAILED
9.5	2	S	5.6	2	S

Model Prediction - Examples

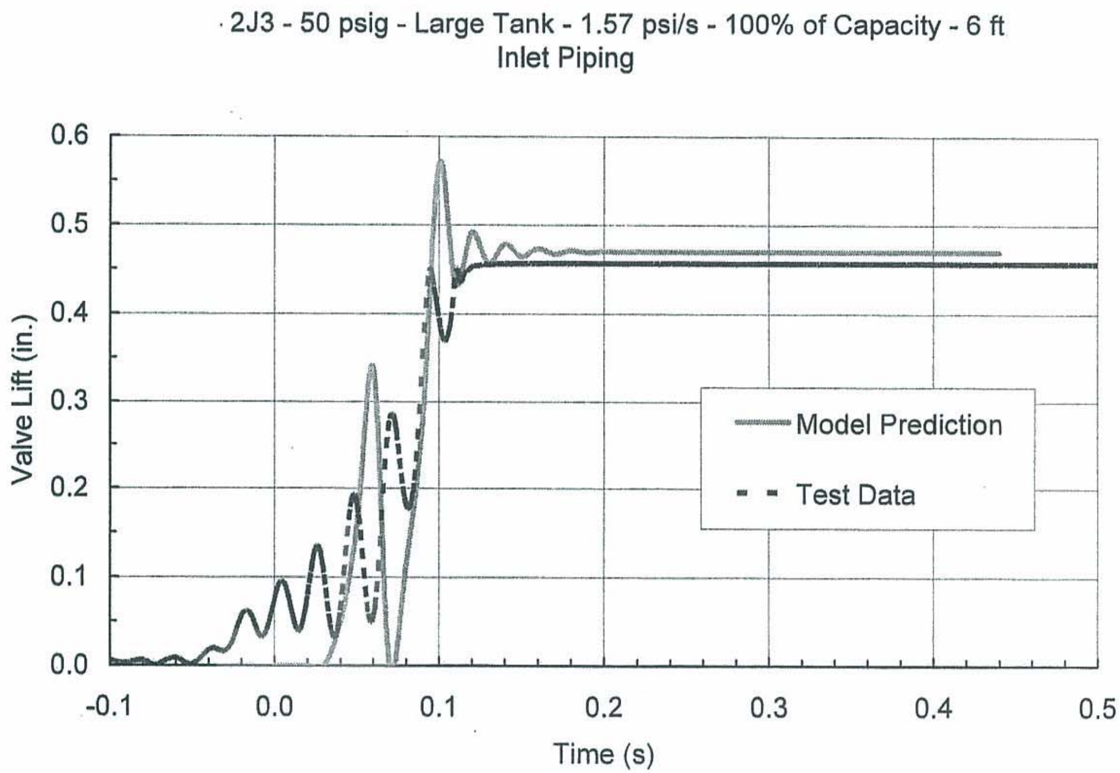


Model Prediction - Examples

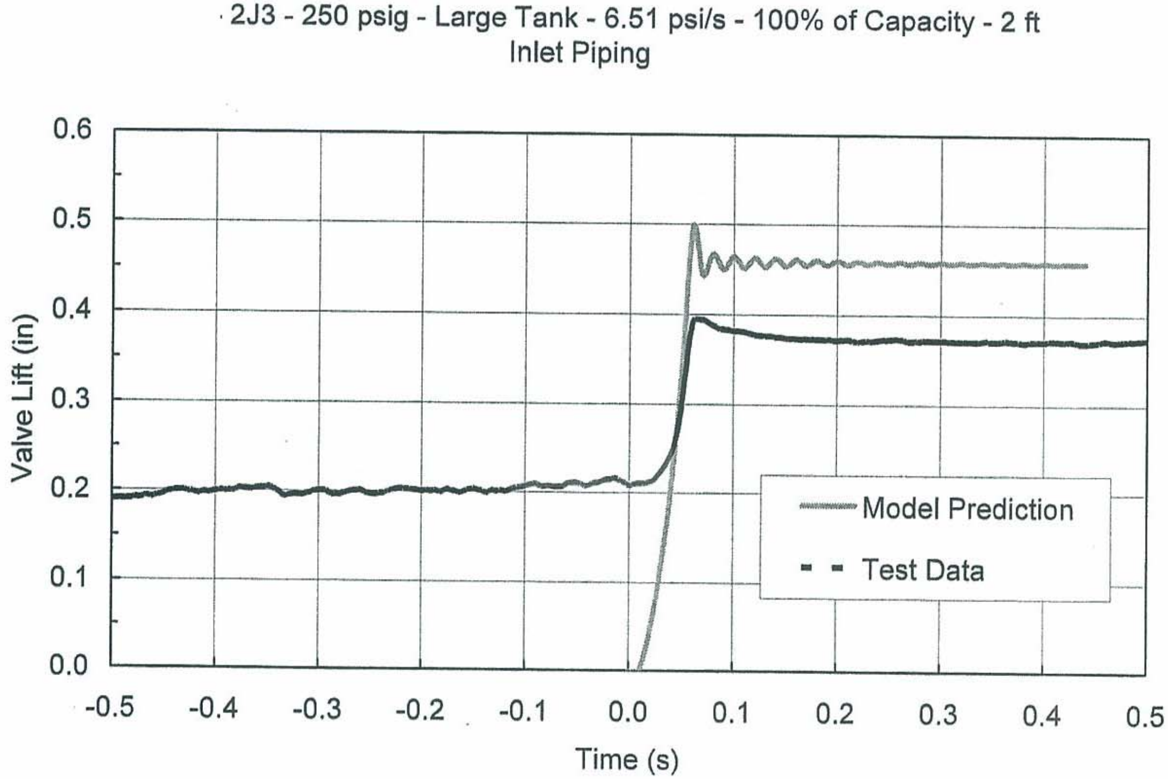




Model Prediction - Examples



Model Prediction - Examples



3L4 Conventional Safety Relief Valve

Flowing Pressure (50 Psig)			Flowing Pressure (250 Psig)		
BD - IPL (Psi)	Pipe Length (Feet)	Operation	BD - IPL (Psi)	Pipe Length (Feet)	Operation
-0.2	6	S - ?	0.9	6	NT - U
1.0	4	S	1.7	4	U
1.0	6	S	2.1	6	NT - U
2.0	2	S	2.6	4	S
2.3	4	S	3.0	4	NT - U
3.3	2	S	3.9	2	S
6.0	6	S	8.4	6	NT - U
7.1	4	S	9.2	4	S
8.1	2	S	10.1	2	S

PRESSURE RELIEF DEVICE INSTALLATION

API RP 520, PART II – INSTALLATION STABILITY

- SAFETY RELIEF VALVE AND PIPING
  - INLET PIPING
  - SAFETY RELIEF VALVE
  - DISCHARGE PIPING
- RUPTURE DISK PIPING

SAFETY RELIEF VALVE INLET PIPE
IRREVERSIBLE PRESSURE LOSS ENGINEERING ANALYSIS

ENGINEERING	SCIENCE and MATHEMATICS
ANALYSIS	SPECIFIC TO THE INSTALLATION
	SPECIFIC TO THE SAFETY RELIEF VALVE

- MANUFACTURER
- VALVE MODEL
- FLOWING FLUID

"NO GENERIC ONE SIZE FITS ALL ANALYSIS"

"NOT A MANAGEMENT OR COMMITTEE DECISION"

API PERF STUDY – DARBY SPREADSHEET

DIERS SUPERCHEMS PROGRAM

CHIYODA PAPERS AND PROGRAM

SAFETY RELIEF VALVE CAPACITY REDUCTION / DAMAGE

- CAPACITY REDUCTION:
  - CHATTER / FLUTTER
- DAMAGE:
  - SEATS (NOT LEAK TIGHT – CLOSE VALVE / SHUTDOWN)
  - STICK OPEN / CLOSED
  - CATALTROPHIC FAILURE
- CONSEQUENCE:
  - PSM INCIDENT
  - LOSS OF CONTIANMENT OF A HIGHLY HAZARDOUS CHEMICAL
  - FIRE, EXLOSION, TOXIC EXPOSURE
  - SINGLE FATALITY / MULTIPLE FATALITY

1976 API PRV Chatter Data

1976 API Survey Results

Size of PRV	Incidents	Leaks/Failures
¾ x 1	1	0
1 ½ x 2 ½	1	1
1 ½ x 3	1	1
2 x 3	3	1
2 ½ x 4	2	0
3 x 4	9	4
4 x 6	16	6
6 x 8	7	5
Unknown	2	1
Total	**42**	**19**

- 4 of the leaks/failures led to fires

Other PRV Chatter

Year	PRV Design Service	Phase During Incident	Liquid Certified Valve	High Inlet Loss	High Back Pressure	Valve Oversized	Other	Incident Severity	Consequences
1964	Vapor	Liquid	No					4	4" line failure and fire
1974	Liquid	Liquid	No					3	Flange leak and fire
1976	Liquid	Liquid	No	Yes		Yes		3	Flange leak and fire
1978	Vapor	Liquid	No	Yes				5	2" pipe failure and VCE
1980	Liquid	Liquid	No					3	Flange leak and fire
1981	Liquid	Liquid	No				Relocated PRV	1	Flange leak
1981	Vapor	Liquid	No					1	Flange leak
1982	Vapor	???	No				Revised Piping	1	Flange leak
1983	Liquid	Liquid	No				Improved supports	1	Small bore pipe failure
1983	Vapor	Vapor	No				Improved supports	2	Pipe failure
1983	Liquid	Liquid	No					1	Flange leak
1983	Liquid	Liquid	No				Changed pipe thk	1	Fitting failure
1983	Liquid	Liquid	No				Piping revisions	1	Flange leak
1983	Vapor	???	No	Yes	Yes	No		1	Flange leak
1986	Liquid	Liquid	No	No	Yes			1	Flange leak
1986	Vapor	???	No					1	Flange leak
1998	Liquid	Liquid	No	No	No	Yes		2	Flange leak and fire
2002	Vapor	Liquid	No	No	No	Yes		1	Small bore pipe failure
2005	Vapor	Vapor	No	No	No	No	Acoustics	3	Large piping failure
2009	Liquid	Liquid	unknown	No	No			2	Flange leak
2009	Liquid	Liquid	unknown				Revised pipe fitting	1	Fitting failure
2010	Liquid	Liquid	No	No	No	Yes		1	Small bore pipe failure
2010	Liquid	Liquid	No	No	No	No		1	Pipe leak

PROSAF Inc.

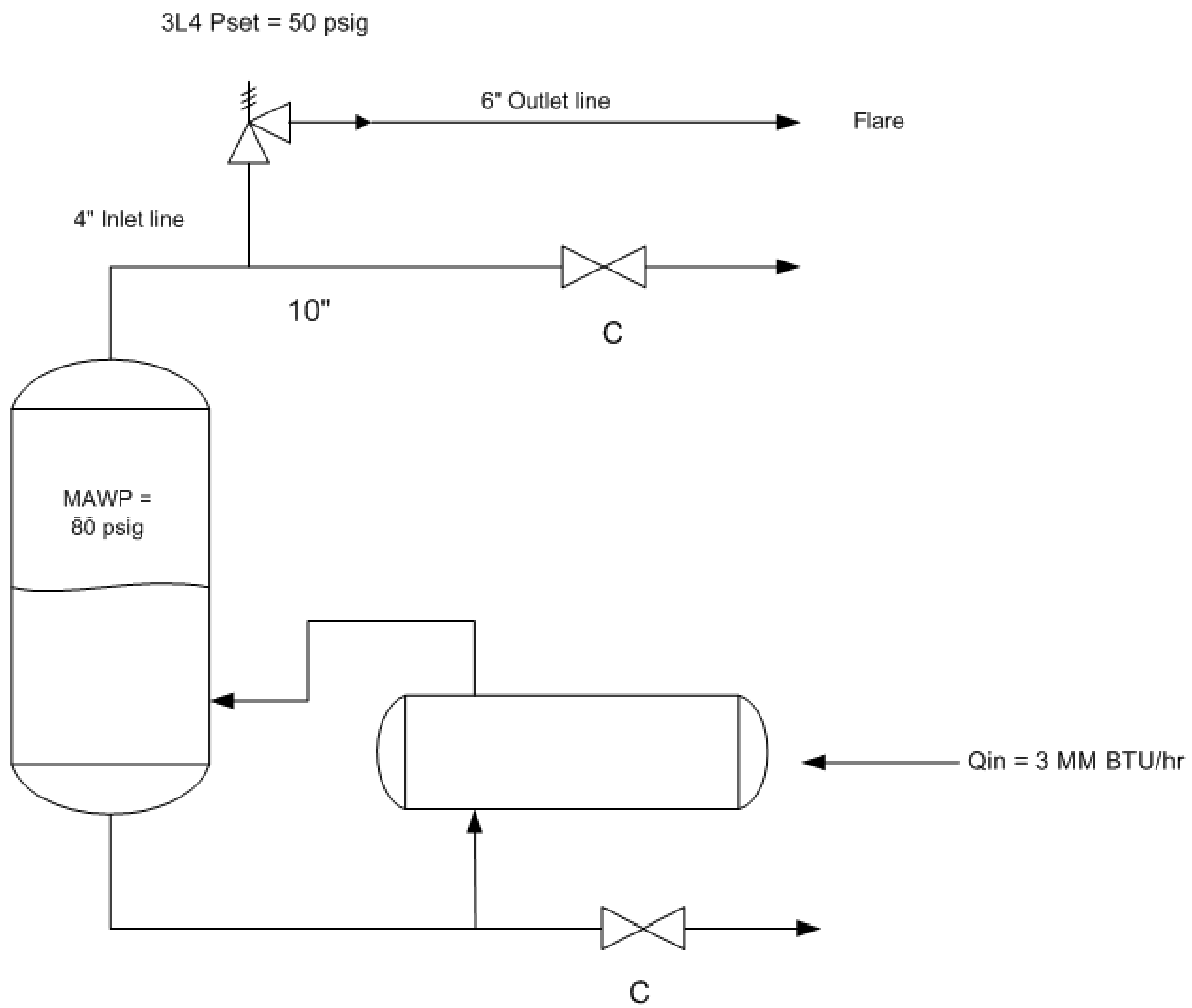
DIERS Users Group

Chicago, IL
October, 2011

3% Rule Basics

Current RAGAGEP

3% Rule Basics:
Current RAGAGEP - example



3% Rule Basics: Current RAGAGEP

- **Example details**
  - **Fluid is hexane, MW = 86.18**
  - **Pset = 50 psig**
  - **P110 = 55 psig**
  - **MAWP = 80 psig**
  - **Pmaap = 88 psig**
  - **Patm = 14 psia (not at sea level)**
- **Scenario is blocked in with heat continuing**

3% Rule Basics: Current RAGAGEP

- **API Std 520, Eqn 1**

$$\frac{P_{cf}}{P_1} = \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}}$$

  - **k_{ideal} = 1.06 (Table 7) (note this is ideal gas at std conditions)**
  - **k_{real} = 1.09 (real gas, saturated at 69 psia)**
  - **k_{real} = 1.108 (real gas, saturated at 102 psia)**

3% Rule Basics: Current RAGAGEP

- **API Std 520 advises use of heat capacity ratio of ideal gas at relief temperature**
- **Stability conditions:**
  - $P_{stability} = P_s = (55 + 14) = 69$ psia
  - $T_s = 128$ C $= 722$ R
  - $k_s = 1.048$
  - $Z_s = 0.86$
- **Capacity conditions**
  - $P_{capacity} = P_{cap} = (88 + 14) = 102$ psia
  - $T_{cap} = 147$ C $= 756$ R
  - $k_{cap} = 1.046$
  - $Z_{cap} = 0.816$

3% Rule Basics: Current RAGAGEP

- **Solve API Eqn 1 for stability and capacity pressures:**
- **Stability: P_{cf} = 41.1 psia**

$$P_{cf} = 69\left[\frac{2}{(1.048+1)}\right]^{\frac{1.048}{(1.048-1)}} = 41.1\ psia$$

- **Capacity: P_{cf} = 60.8 psia**

$$P_{cf} = 102\left[\frac{2}{(1.046+1)}\right]^{\frac{1.046}{(1.046-1)}} = 60.8\ psia$$

3% Rule Basics: Current RAGAGEP

- **Next, solve API Eqn 8 for term C for both stability and capacity purposes:**

- **Stability: C_s = 321.0**

$$C_s = 520\sqrt{1.048\left(\frac{2}{(1.048+1)}\right)^{\frac{(1.048+1)}{(1.048-1)}}} = 321.0$$

- **Capacity: C_{cap} = 320.7**

$$C_{cap} = 520\sqrt{1.046\left(\frac{2}{(1.046+1)}\right)^{\frac{(1.046+1)}{(1.046-1)}}} = 320.7$$

3% Rule Basics: Current RAGAGEP

- **Next, determine other parameters for the sizing equation, API Eqn 2 for both stability and capacity purposes:**
- **K_d = 0.953 * 0.9 = 0.858**
- **K_c = 1 (no rupture disk present)**
- **K_b = 1 (assumption; to be checked later)**

3% Rule Basics:
Current RAGAGEP

- **Next, determine other parameters for the sizing equation, API Eqn 2:**
- **λ_{cap} = 112 BTU/lb**
- **v_{gcap} = 1.32 ft3/lb**
- **v_{lcap} = 0.0302 ft3/lb**

$$W_{cap} = \frac{3{,}000{,}000}{112} \frac{(1.32 - .0302)}{1.32} = 26{,}173\ lb/hr$$

3% Rule Basics: Current RAGAGEP

- **Next, solve the sizing equation, API Eqn 2 and pick a PSV size**

$$A_{cap} = \frac{26,173}{(320.7 * 0.858 * 102 * 1 * 1)} \sqrt{\frac{(756 * 0.816)}{86.18}} = 2.495\, in^2$$

- **Select a 3L4 PSV with orifice area of 3.17 in2 which provides the required capacity**

3% Rule Basics:
Current RAGAGEP

- **Next, use the sizing equation, API Eqn 2, to determine PSV capacity at 10% OP:**

$$W_s = 3.17 * 321 * 0.858 * 69 * 1 * 1 \sqrt{\frac{86.18}{(722 * 0.86)}} = 22{,}443\ lb/hr$$

- **This is the capacity to use for IPL/OPL calcs**
- **Note: PSV capacity inadequate for required relief at 10% OP; pressure will have to rise above 69 psia**

3% Rule Basics: Current RAGAGEP

- **Note also: this is not BEF, use derated discharge coefficient in sizing equation, API Eqn 2, to determine PSV capacity at 10% OP:**

$$W_s = 3.17 * 321 * 0.858 * 69 * 1 * 1 \sqrt{\frac{86.18}{(722 * 0.86)}} = 22{,}443\ lb/hr$$

- **Why?**

- **Yes, PSV will flow more than this at 10% OP, but the bar (3%) , is based on “valve nameplate capacity” which is not BEF.**

3% Rule Basics: Current RAGAGEP

- **IPL: What is the inlet line? In this example, from the vessel to the PSV**
- **Because the vessel provides the source of flow or inventory that is needed to keep the PSV open**
- **To use Comflow to calculate IPL, convert 10” piping resistance coefficient total (entrance, elbow and 20’) to 4” basis. Add 4” fittings (tee branch, elbow and reducer) plus 5’ length to provide Comflow input.**

3% Rule Basics: Current RAGAGEP

- **From Comflow:**
  - **IPL = (68.39 – 67.67)/50 = 0.0144 or 1.44% OK**
- **From SC Lite:**
  - **IPL = 2.1% (difference in flow rate and in treatment of tee branch resistance coefficient)**
  - **OPL = 3.8% (shows OK to use conventional PSV and validates assumption of K_b of 1)**
  - **Note that OPL also done at 10% OP, not at higher allowed OP**

```
           *****  COMPRESSIBLE GAS FLOW IN PIPING  *****

  1>  Choose: (1) U.S. Cust.-F (2) U.S. Cust.-C (3) Metric Units  [2]

  2>  Est.: (1) W  (2) P1  (3) D  (4) P2 from P0  (5) P2 from P1  [4]

  3>  D, W, P0, T0  = 4.026,22443,69,128

  4>  MW, K, MU, Z  = 86.18,1.048,.009,0.86

  5>  L, KF, ES     = 5,.844,0.006


 ====  RESULTS  =================  PIPE EXIT  ========================

  P1 =     68.39           P2 =     67.67

  RE =  3.912D+06          P2S =     68.28

  FF =   0.00544           T2 =    127.83

                      V2 =     80.56


 ===========  (1-5) CHANGE DATA LINES 1-5  (R) RERUN  (X) EXIT  [ ]  =

  NOMENCLATURE & UNITS                     U.S. Cust.

  --------------------                     ----------   -

  D,ES,L      pipe Diam., roughness, Length          in,in,ft

  K,MW,Z      Cp/Cv, Mol. Wt., compress.              -

  P,T         Pressure, Temperature                psia,F/C

  MU,W        viscosity, mass flowrate             cp,lb/hr

  F,V         Force, Velocity                      lbf,ft/s

  FF,RE,KF,N  Fan. f, Rey. no., fitt. loss, total loss    -
```

WZD-DIERS EXAMPLE PSV AND PIPING FLOW

SECTION: 1.1 **nerated scenario for Detailed Piping Isometric - Vapor Discharge**

VAPOR/GAS FLOW FROM PIPES

Piping Isometric / Layout	DIERS
Liquid Density Method	Databank
VLE/PVT Method	Equation of state

**** INLET CONDITIONS**

Temperature. C	128.0
Pressure. psig	55.50

**** EXIT CONDITIONS**

Temperature. C	114.871	
Pressure. psig	2.016	
Surroundings/back pressure. psig	2.0	
Barometric pressure at site elevation. psig	-3.974E-06	DEFAULT site
Speed of sound. m/s	192.061	
Exit Mach Number	0.237	
Maximum Mach Number	0.979	** WARNING: Ma
Exit rho*u*u in kg/m/s2 or in Pascals	6574.579	
Maximum rho*u*u in kg/m/s2 or in Pascals	33902.811	
Exit Reynold's Number	2.6036E+06	
Exit steady state impulse. lbf	510.364	
Maximum steady state impulse difference. lbf	5293.217	** WARNING: >
Mass flow rate. lb/s	5.946	
Volumetric flow rate. SCFH	87848.144	

21,406 lb/hr

2.27% of Pset of 50 psig

COMPONENT	MW	FLOW IN lb/
n-HEXANE	86.177	5.94

Relief device found at segment 3	**EXAMPLE 3L4: Enter Descr**	**l. Service= Gas**
Last iteration Kb correction	1.0	
Pressure at device inlet. psig	53.789	
% Inlet pressure drop relative to effective set point	3.180	
% Irrev. inlet pressure drop relative to effective set point	2.106	
% back pressure relative to effective set point	3.806	

Segment #, Name, Type	Start Elevation. ft	End Elevation. ft	Length. ft	Exit Diameter. ft	Exit/Flow Area. ft2
001, 10 INCH INLET LINE	10.0000	10.0000	20.0000	0.8350	0.5476
002, 4 INCH INLET LINE	10.0000	15.0000	5.0000	0.3355	0.0884
003, EXAMPLE 3L4, PRV (orifice)	15.0000	15.5000	0.5000	0.1674	0.0220
004, 4 INCH OUTLET LINE	15.5000	15.5000	0.2000	0.3355	0.0884
005, 4 X 6 EXPANDER, EXP, Tapered:45.00 deg,	15.5000	15.9102	0.4102	0.5054	0.2006
006, 6 INCH OUTLET LINE	15.9102	15.9102	15.0000	0.5054	0.2006

Segment #, Name, Type	Inlet Temp. C	Inlet Pres. psig	Irrev. dP. psia	Total dP. psia	Exit Friction Factor
001, 10 INCH INLET LINE	128.0000	55.5000	0.4037	0.9366	0.0044
002, 4 INCH INLET LINE	127.7100	54.5634	0.7290	0.7739	0.0054
003, EXAMPLE 3L4, PRV (orifice)	127.6101	53.7895	25.5549	25.5549	0.0000
004, 4 INCH OUTLET LINE	115.0188	2.0473	0.0318	0.0454	0.0054
005, 4 X 6 EXPANDER, EXP, Tapered:45.00 deg,	114.9998	2.8046	0.0056	0.0066	0.0038
006, 6 INCH OUTLET LINE	114.9982	2.7980	0.7381	0.7819	0.0049

Segment #, Name, Type	Exit NRe
001, 10 INCH INLET LINE	1479361.35
002, 4 INCH INLET LINE	3689351.76
003, EXAMPLE 3L4, PRV (orifice)	7395647.28
004, 4 INCH OUTLET LINE	3921024.18
005, 4 X 6 EXPANDER, EXP, Tapered:45.00 deg,	2601759.40
006, 6 INCH OUTLET LINE	2603644.12

** NOTE : Backpressure calculation is enabled

Last Specified: 04:21:18 PM, Mon Sep 05 2011
Last Executed: 04:21:26 PM, Mon Sep 05 2011

Segment Index and Name	Type	AXIAL DISTANCE. ft	Elevation with respect to vessel bottom. ft	STATIC PRESSURE. psig	FLUID TEMPERATURE. C
001, 10 INCH INLET LINE	Pipe	0.000	10.000	55.500	128.000
		10.001	10.000	55.492	127.999

3% Rule Basics: Current RAGAGEP

- **Some interpret API 520, para. 5.3.3.1.3 allows BUBP above 10% in this case because max allowed OP is 88/50 or 76% OP**
- **Yes, at 76% OP, the PSV will be stable with BUBP above 10%**
- **However, you MUST FIRST meet the first sentence of para. 5.3.3.1.3 which says BUBP ≤ 10% at 10% OP**

3% Rule Basics:
Current RAGAGEP

- **Example:**
  - **Fire case; P_{MAAP} = 121% of MAWP**
  - **Allowed OP is then 21%**
  - **IPL is calculated at 3% at 21% OP**
  - **OPL is calculated at 21% at 21% OP**
  - **is this OK per API 520, 5.3.3.1.3?**
- **What happens as this system rises through the 10% OP point?**

3% Rule Basics: Current RAGAGEP

- **Simple Interpolation shows:**
  - **IPL drops just below 3% - good**
  - **OPL drops to about 17% - not so good**
- **This PSV is likely unstable at 10% OP even though it probably will stabilize at 21% OP – if it can get there**

3% Rule Basics:
Current RAGAGEP

- **Conclusion:**
  - **Show IPL ≤ 3% at 10% OP**
  - **Show OPL ≤ 10% at 10% OP**
  - **Check bellows PSVs for stability (≤ 50% BUBP) at 10% OP**
  - **Check PSV capacity against required capacity at allowed OP**
- **If software is not set up to check stability at 10% OP and capacity at allowed OP, this is NOT an indication that the required method is wrong…..**

DIERS Users Group Mtg.
October, 2011   Burr Ridge, IL

“Odds and Ends”
Relief Valve Stability-Part 4
Welcome to the Morgue

Michael A. Grolmes

Centaurus Technology, Inc.

4590 Webb Road

Simpsonville, KY 40067

502-243-9678

ctimag@att.net

Centaurus Technology, Inc.

Diplomacy

Is the art of saying

"Nice Doggie"

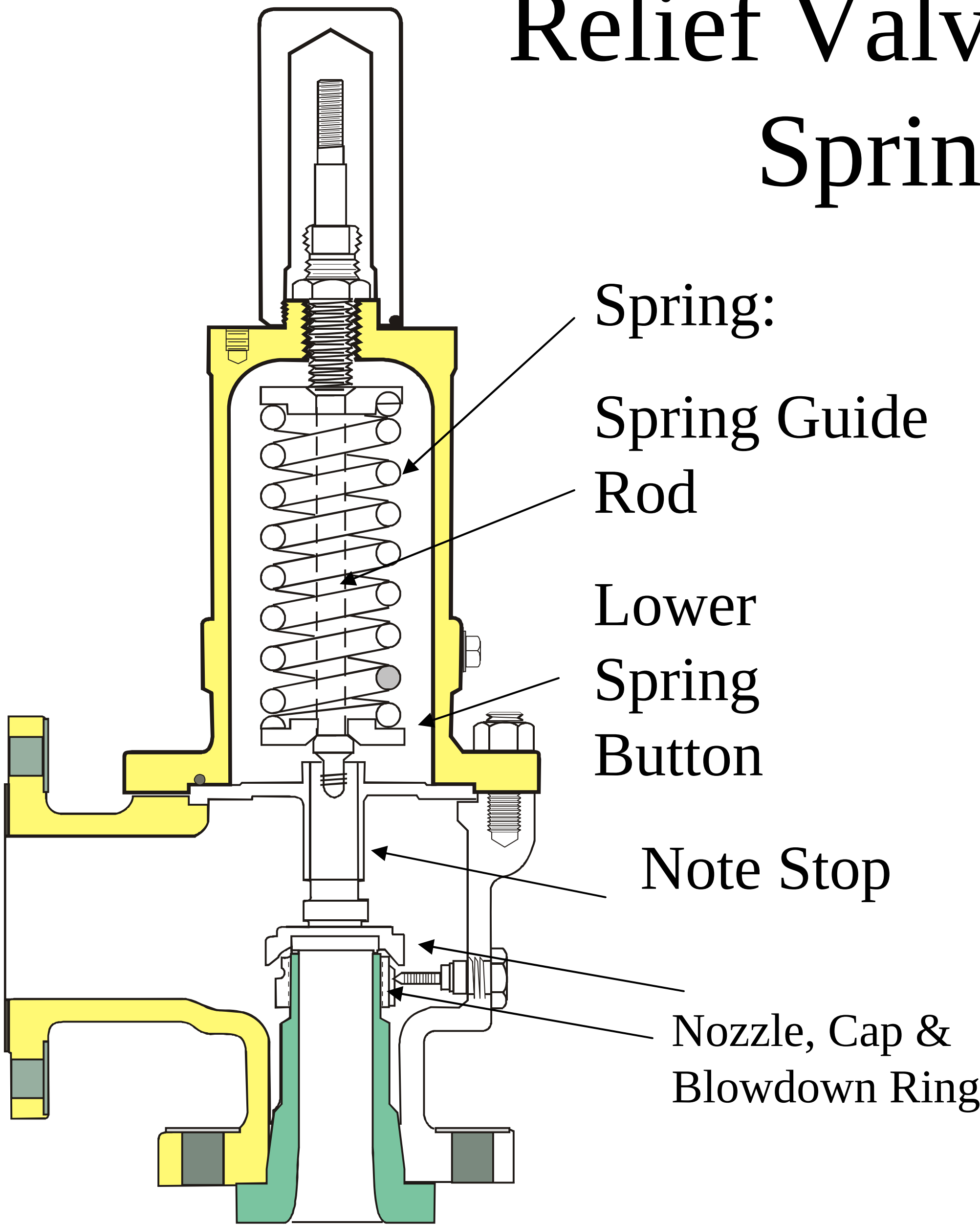
Until you can find a rock.

Will Rodgers

Relief Valve Nozzle Illustrations

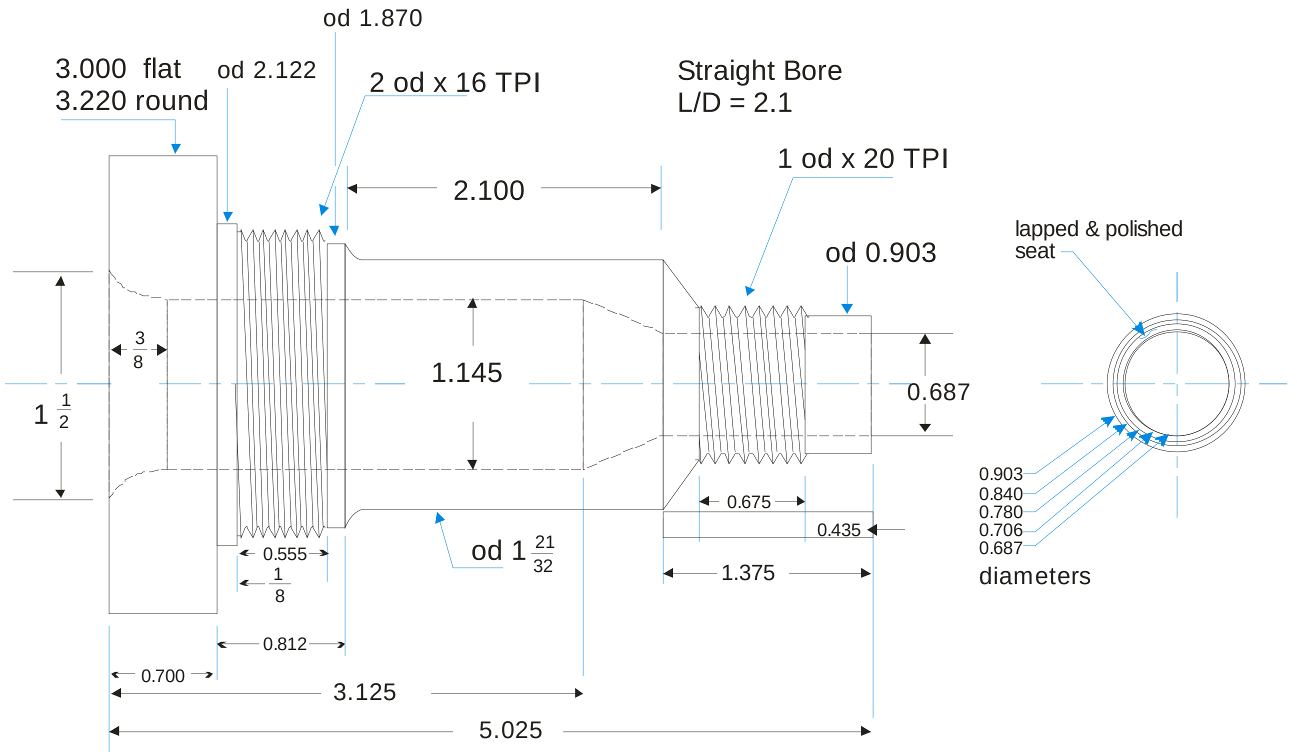
With many thanks to those who provided relief valves for examination.

Relief Valve Operation as a Spring –Mass System



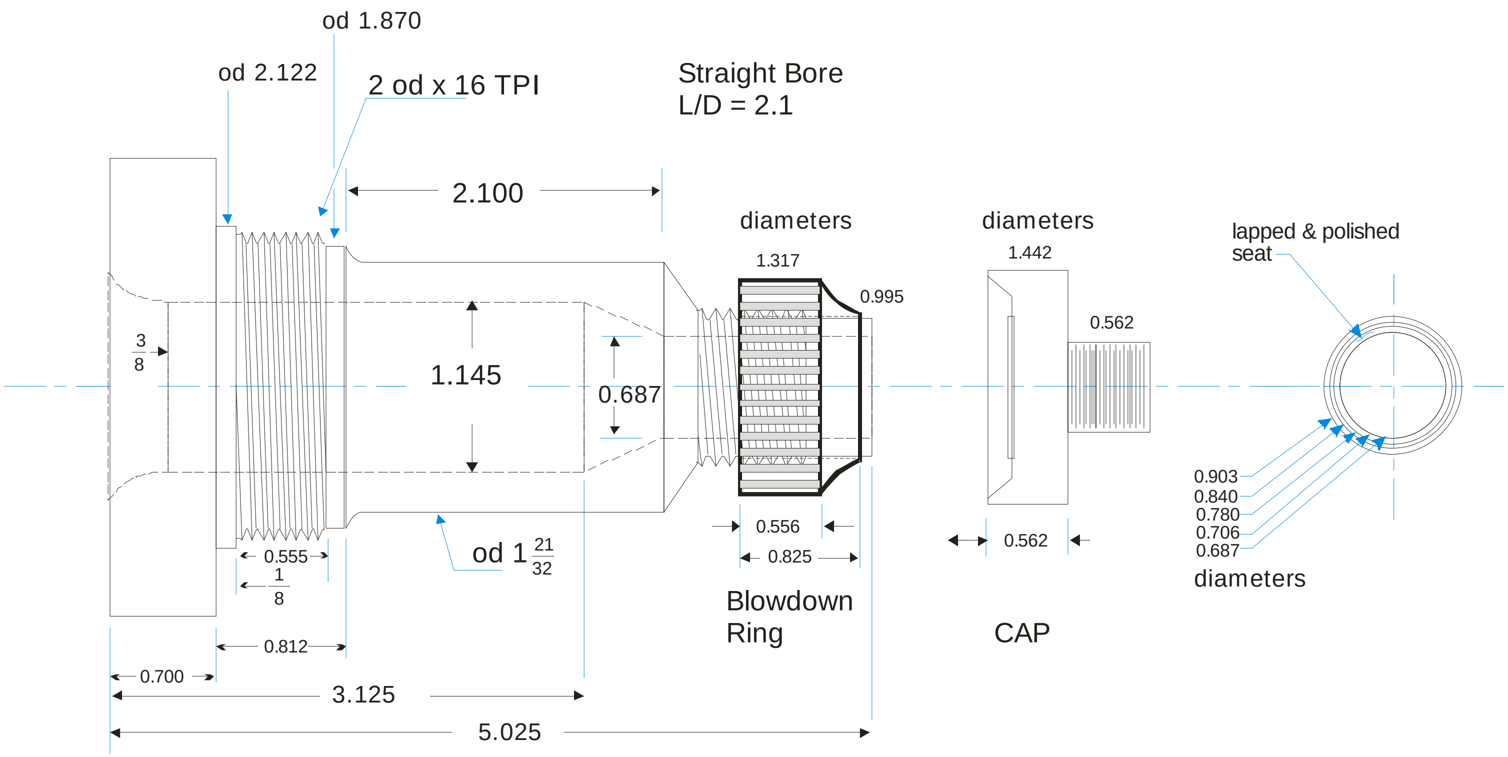
The mass is there, but not in one lump. There is the valve cap, a slider piece, the spring rod, a restraining button and some part of the spring itself. (1/3 total Spring Wgt.)

Vendor		Consolidated Dresser	Consolidated Dresser	Farris	Farris	Farris	Consolidated Dresser
Model		**1906Fc**	**1905Fc**	**26FA10**	**26GA10L**	**26PB10-120**	1910-30-Qc
S/N		**TJ46345**	**TH85105**	**67041-A10**	**545342-1-A14/G**	**7511-A9**	TC78912
Bellows:		No	No	No	No	Yes	Yes
Inlet	[inch]	1½	1½	1½	1½	4	6
Flange		300# RF	150# RF	150# RF	150# RF	150# RF	300# RF
Outlet	[inch]	2	2	2	2½	6	8
Flange		150# RF	150# RF	150# RF	150# RF	150# RF	150# RF
Nozzle	[Letter]	F	F	F	G	P	Q
API Area	[sq. in.]	0.307	0.307	0.307	0.503	6.38	11.05
Set Pressure	[psi]	100	120	180	100	50	185
Name Plate Rating		695 scfm air	818 scfm air	1239 scfm air	145 usgpm water	7932 scfm air	
Catalog Rating		698 scfm air	821 scfm air	1239 scfm air	145 usgpm water	7877 scfm air	43000
Date of Mfg.		Oct-91	Jan-90	unk	Oct-08	unk	unk
Dead Weight	[lbs]	43	39.5	38.5	39	185	
Catalog Weight	[lbs]	45	45	44	50	190	530
Redbook: Nozzle Area	[sq. in.]	0.375	0.375	0.371	0.559	7.087	12.85
Measured Nozzle Area	[sq. in.]	0.363	0.363	0.371	0.567	6.4108	12.347
Redbook: Lift	[inch]	0.182	0.182	0.206	0.326	0.901	1.09
Measured Component Weights							
Spring Wgt.	[g]	103	112	360	241	2555.2	14968
Cap and bellows	[g]	322	322	96	231	3137.7	19731
Spring rod	[g]	269.5	230	205	126	513	2268
Spring Button	[g]	51	51	107	103	731.7	5216
Total wgt. In motion	[g]	676.8	640.3	528.0	540.3	5234.1	32204.3
Spring:							
Free Length	[inch]	2.737	2.000	4.1875	4.275	7.6875	12.125
OD	[inch]	1.343	1.380	1.7620	1.605	3.5590	7.091
ID	[inch]	0.960	0.975	1.1670	1.125	2.5120	4.875
Wire Dia.	[inch]	0.192	0.202	0.2975	0.240	0.5250	1.108
Check	[inch]	-0.001	0.000	0.0000	0.000	-0.0015	0.0000
Pitch	[inch]	0.407	0.413	0.4670	0.486	0.9750	2.25
Coil Angle	degrees	8.0	8.0	9.0	7.0	9.0	9.0
G	[psi]	1.16E+07	1.16E+07	1.16E+07	1.16E+07	1.16E+07	1.18E+07
Mean Dia	[inch]	1.1515	1.1775	1.4645	1.365	3.0355	5.983
C or D/d	[-]	6.00	5.83	4.92	5.69	5.78	5.40
Active Coils	[#]	5.7	3.8	8.0	7.8	6.9	4.4
Spring Constant	[lbf/in]	225	385	454	243	572	2365
	[N/m]	39474	67383	79478	42483	100166	414119
Wgt in Motion	[kg]	0.677	0.640	0.528	0.540	5.234	32.204
Nat Freq by Formula	[Hz]	**38.4**	**51.6**	**61.7**	**44.6**	**22.0**	**18.0**
Estimated Nat Freq							
Ratio of Cap area to nozzle area		1.3	1.3	1.3	1.3	1.3	1.2
Spring Constant; k est	[lbf/in]	289	347	455	241	552	2827
	[N/m]	50654	60785	79695	42155	96685	494925
Wgt in Motion; m_est	[kg]	0.677	0.612	0.499	0.680	5.171	31.253
Nat Freq by estimate	[Hz]	**43.54**	**50.14**	**63.61**	**39.62**	**21.76**	**20.03**
Glide Surface Area	[sq inch]	5.77	5.77	2.38	2.00	16.27	49.14
Cap/nozzle impact area	[sq inch]	0.169	0.169	0.075	0.201	1.12	2.686
Moving Wgt / Valve wgt	[%]	3.31586	3.13704	2.64550	2.38242	6.07321	13.39570



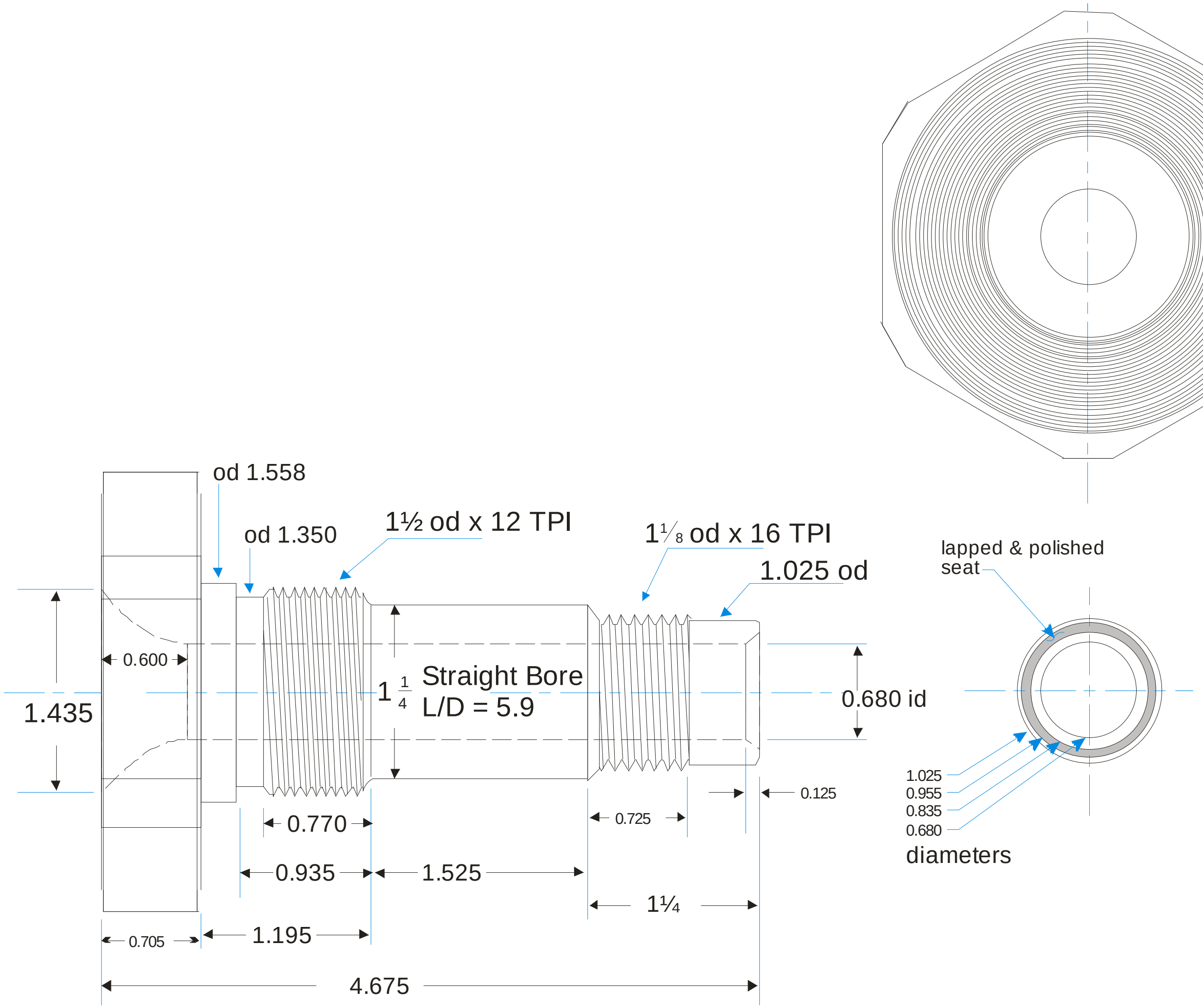
(ALL DIMENSIONS in INCHES)

("F" Nozzle from Farris 2600 1 ½ F 2)



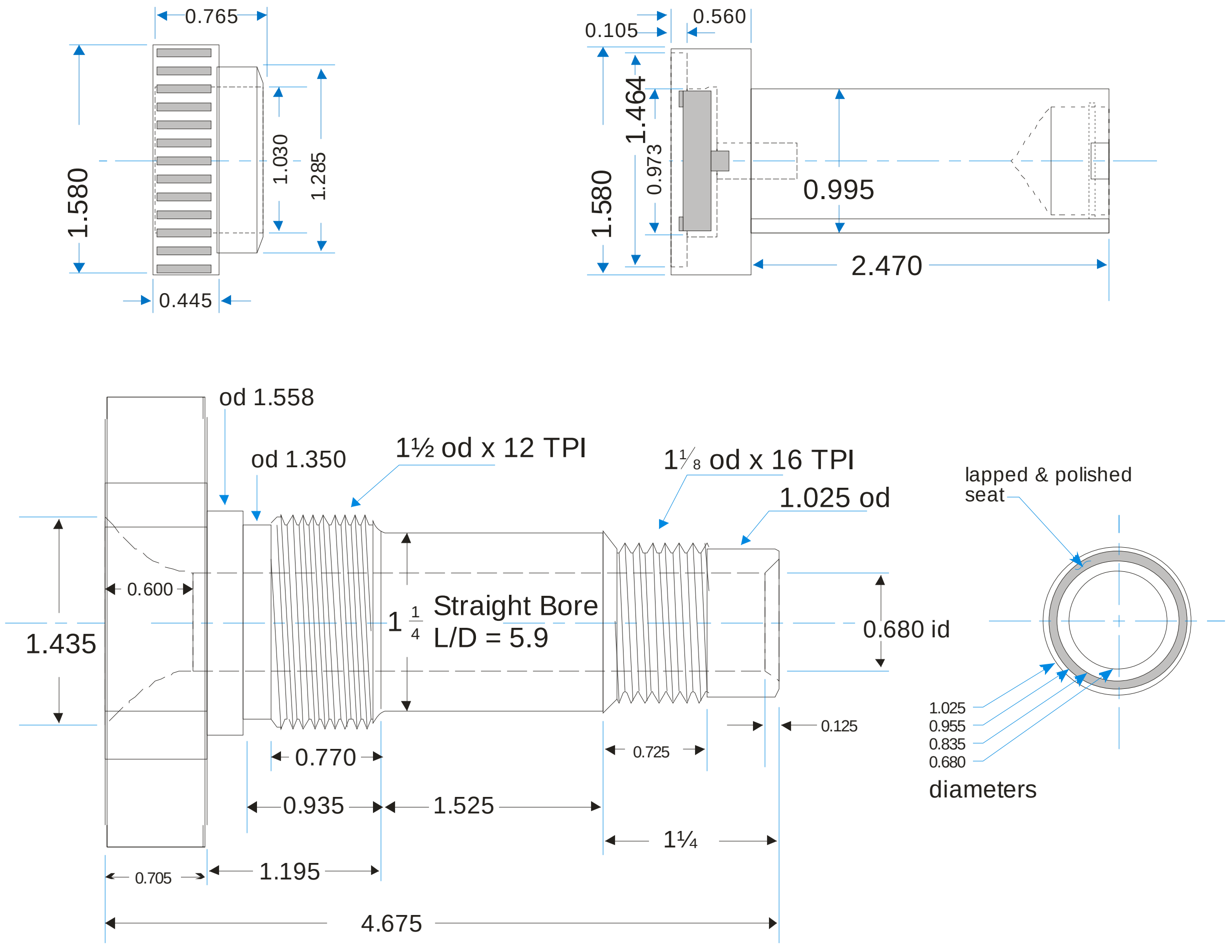
(ALL DIMENSIONS in INCHES)

("F" Nozzle from Farris 2600 1 ½ F 2)



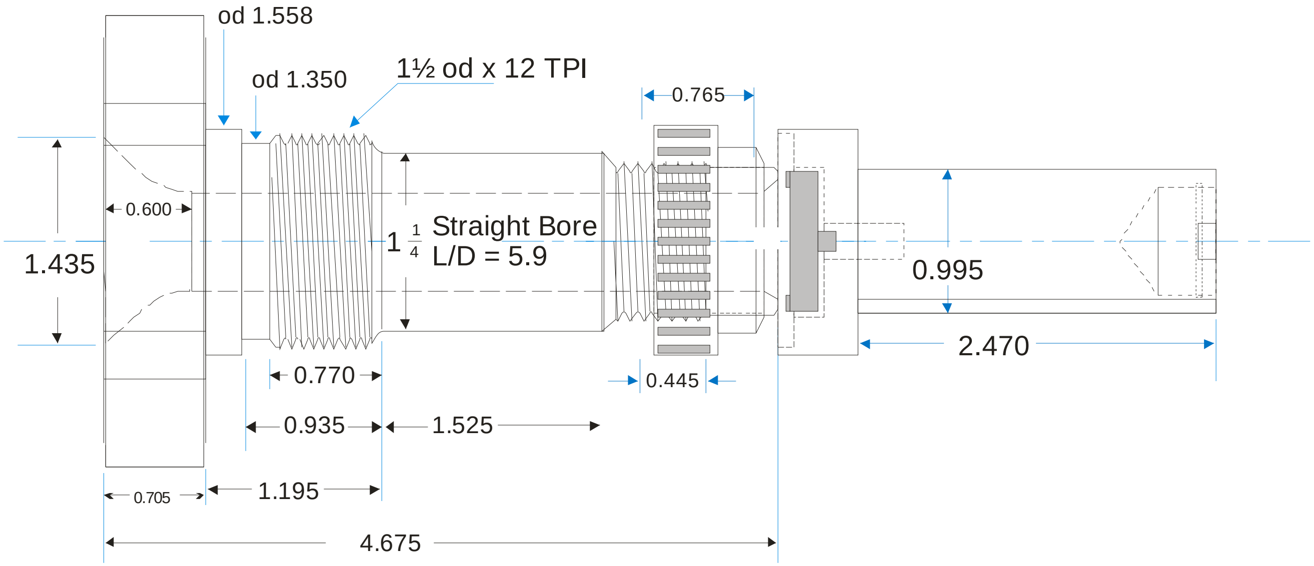
(ALL DIMENSIONS in INCHES)

(Nozzle from Consolidated 1905 or 1906 Fc; *1 ½ F 2*)



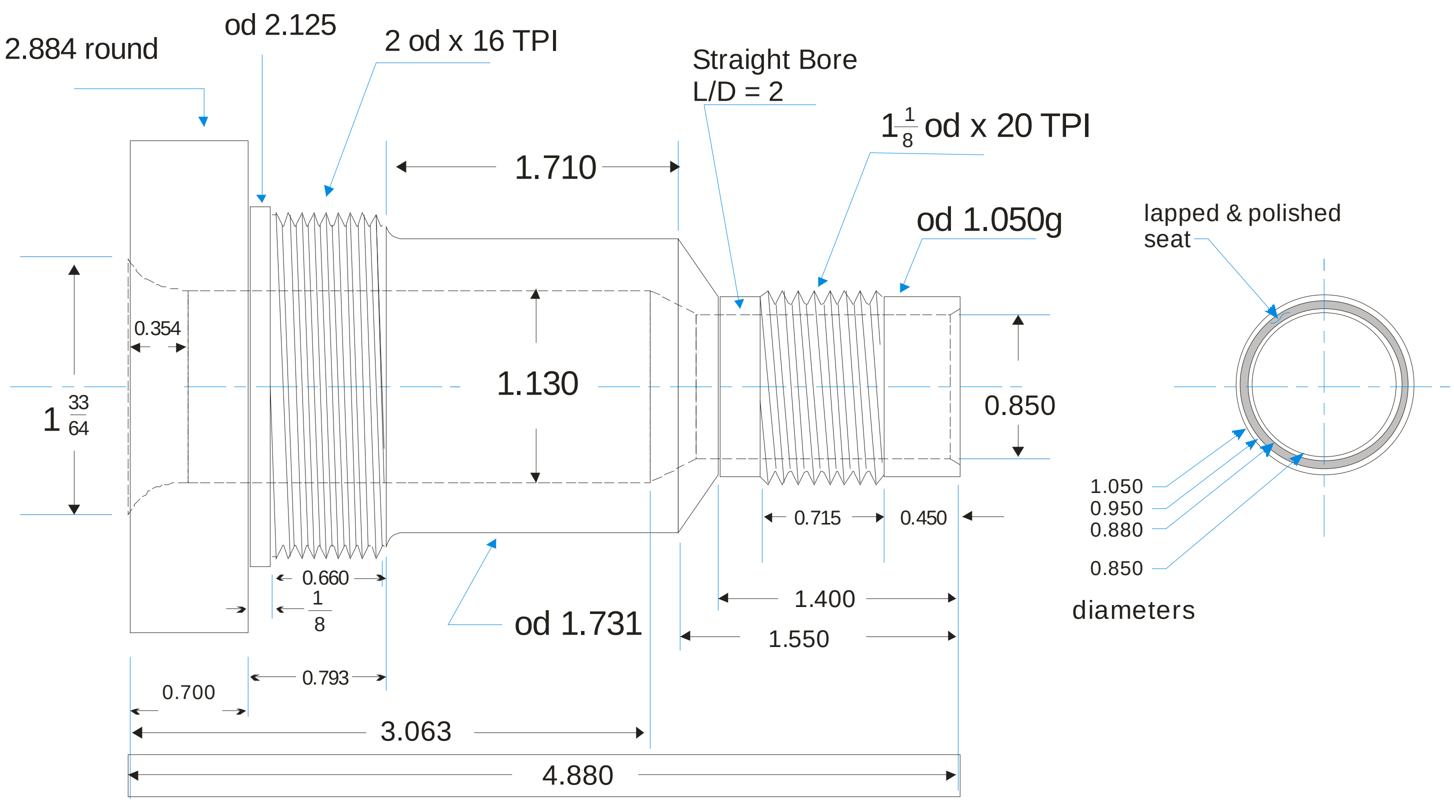
(ALL DIMENSIONS in INCHES)

(Nozzle from Consolidated 1905 or 1906 FC *1 ½ F 2*)



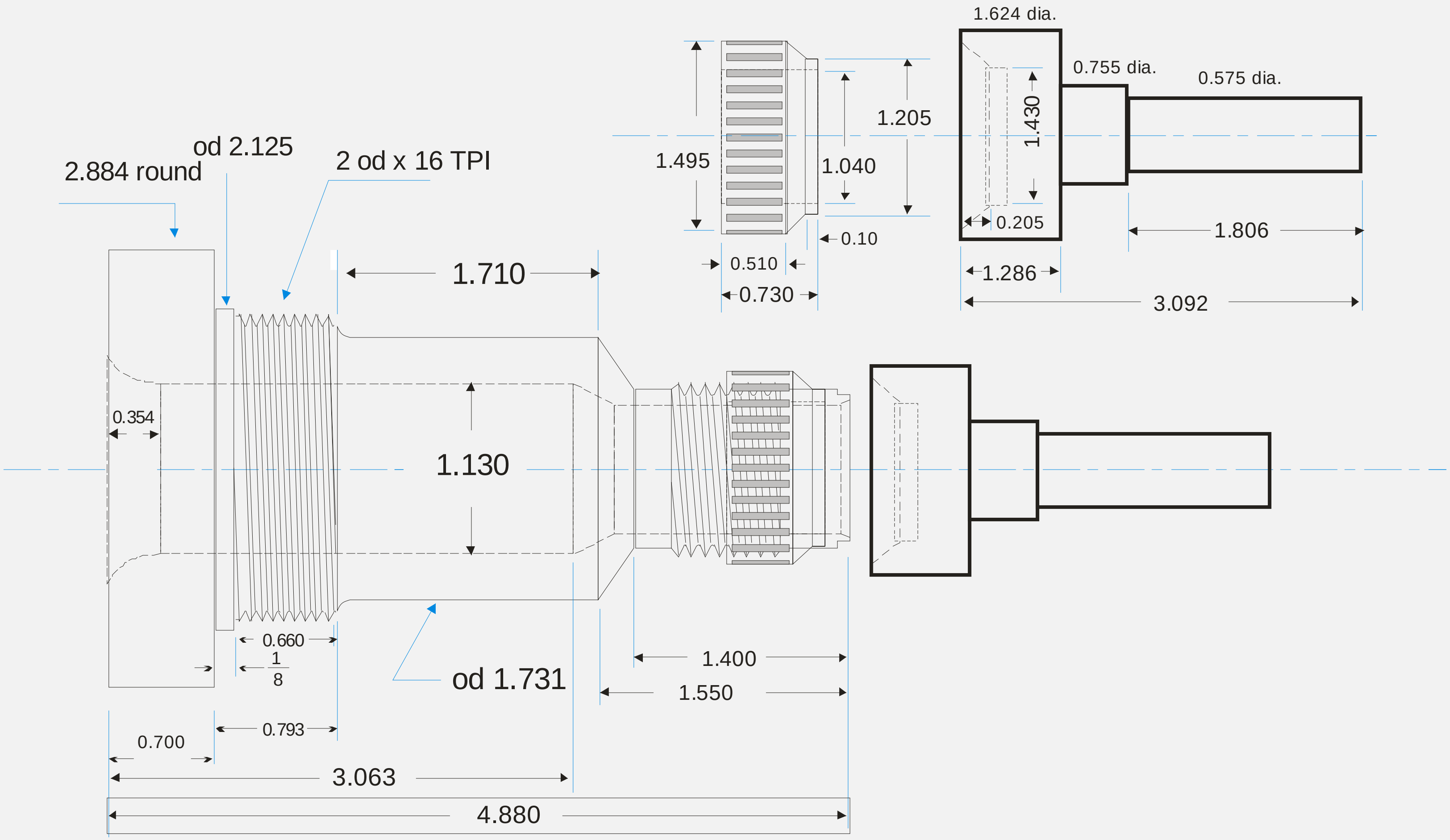
(ALL DIMENSIONS in INCHES)

(Nozzle from Consolidated 1905 or 1906 FC *1 ½ F 2*)



(ALL DIMENSIONS in INCHES)

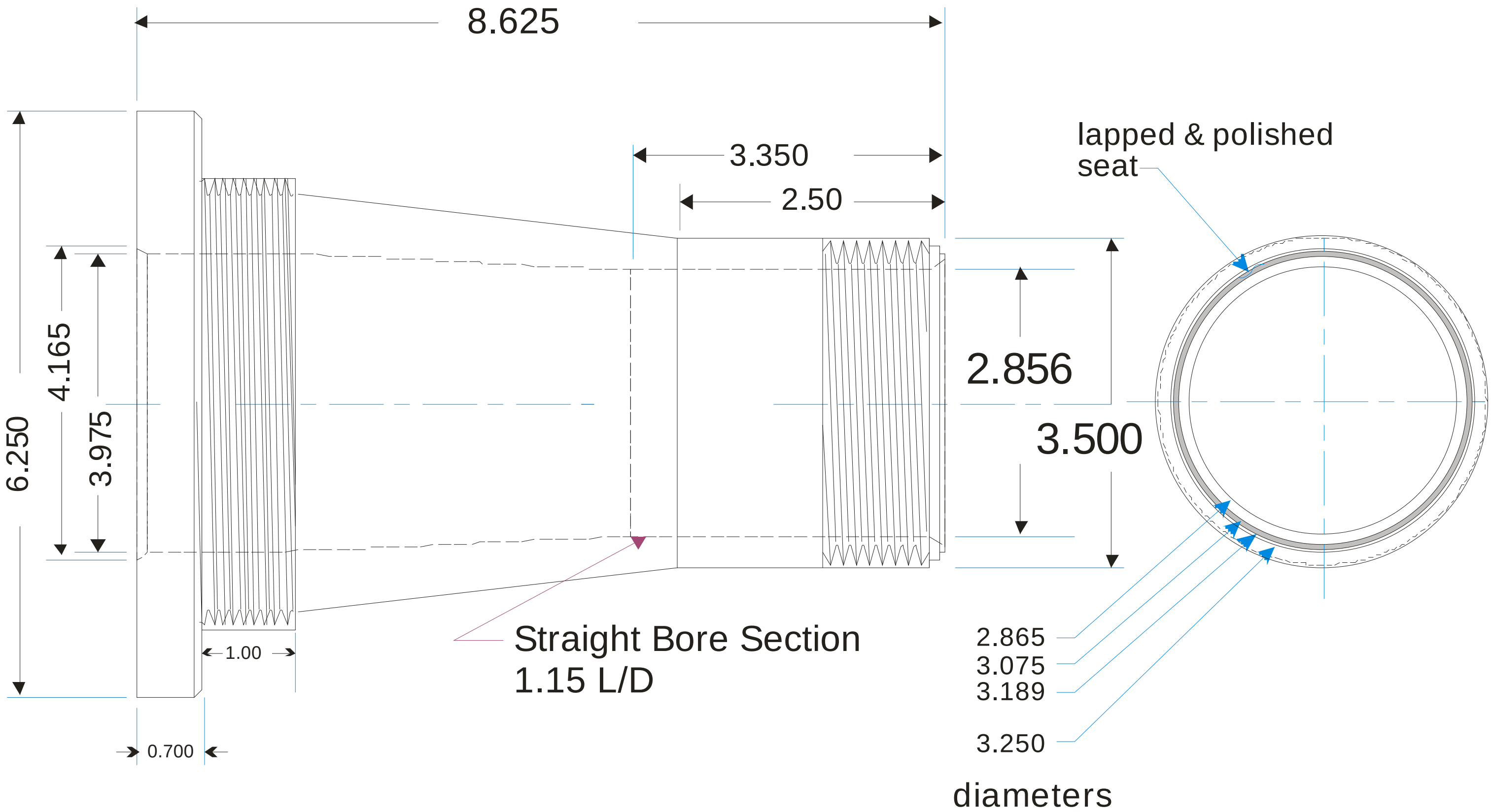
("G" Nozzle from Farris 2600 1 ½ G 2½)



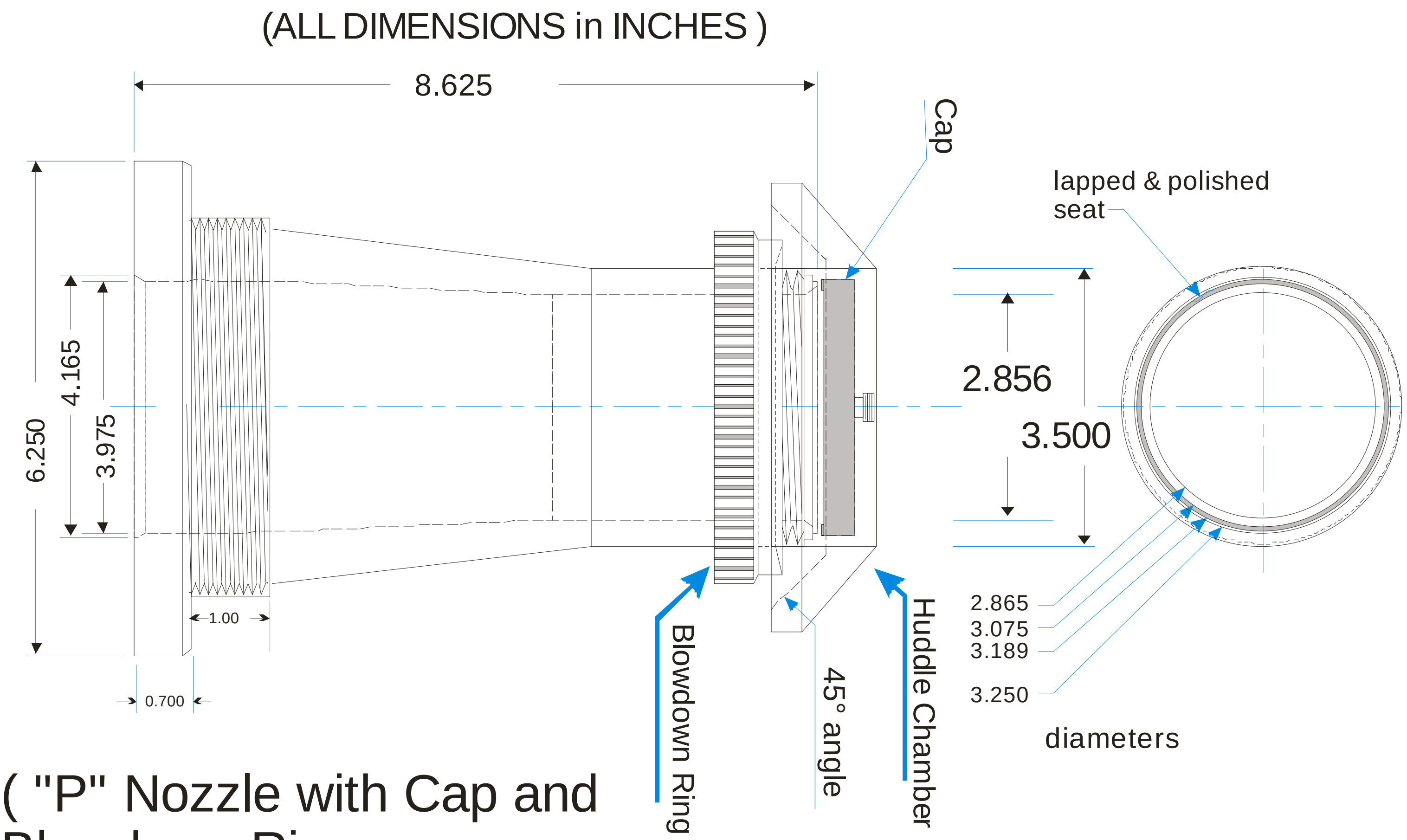
(ALL DIMENSIONS in INCHES)

("G" Nozzle from Farris 2600 1 ½ G 2½)

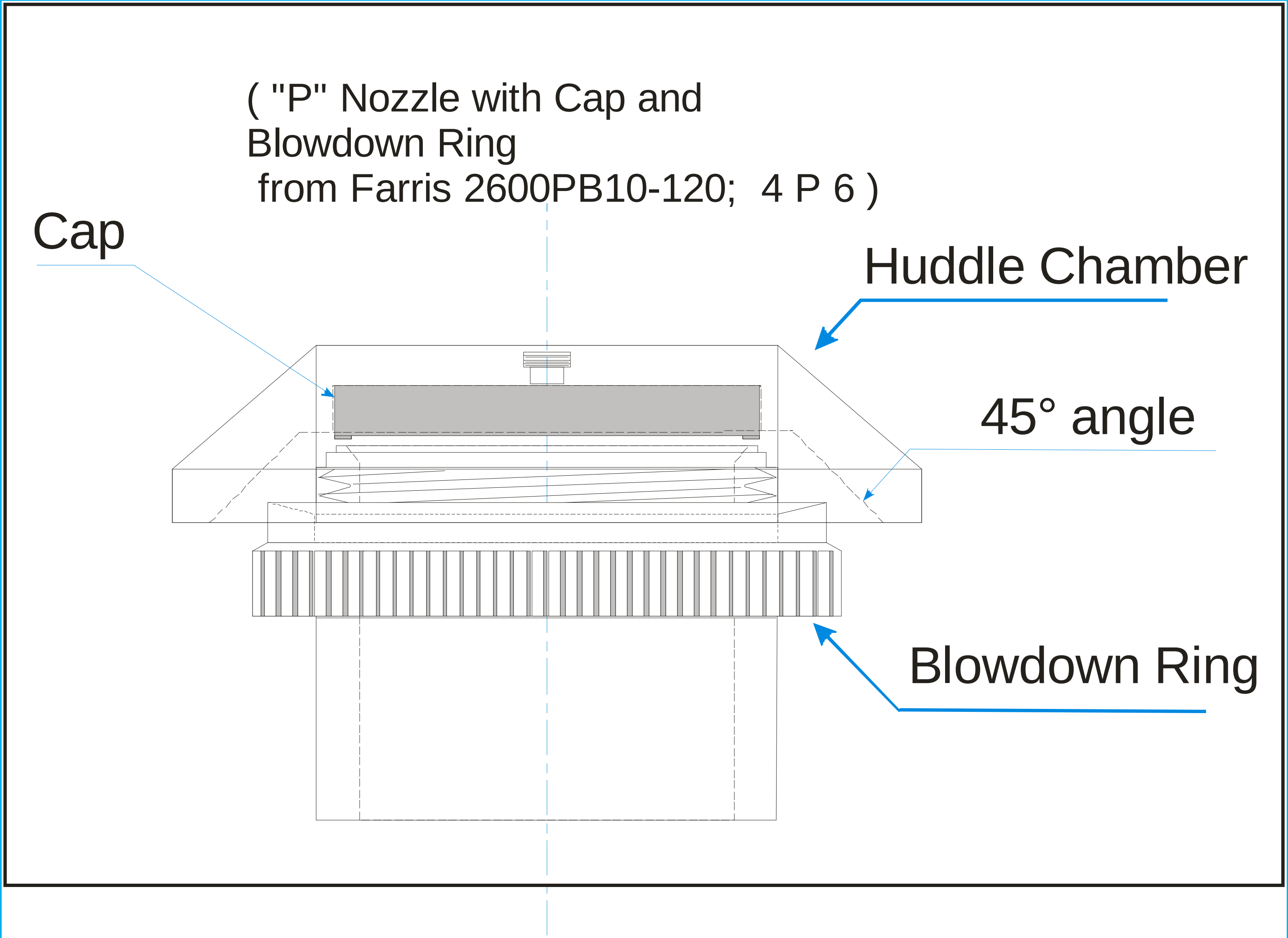
(ALL DIMENSIONS in INCHES)

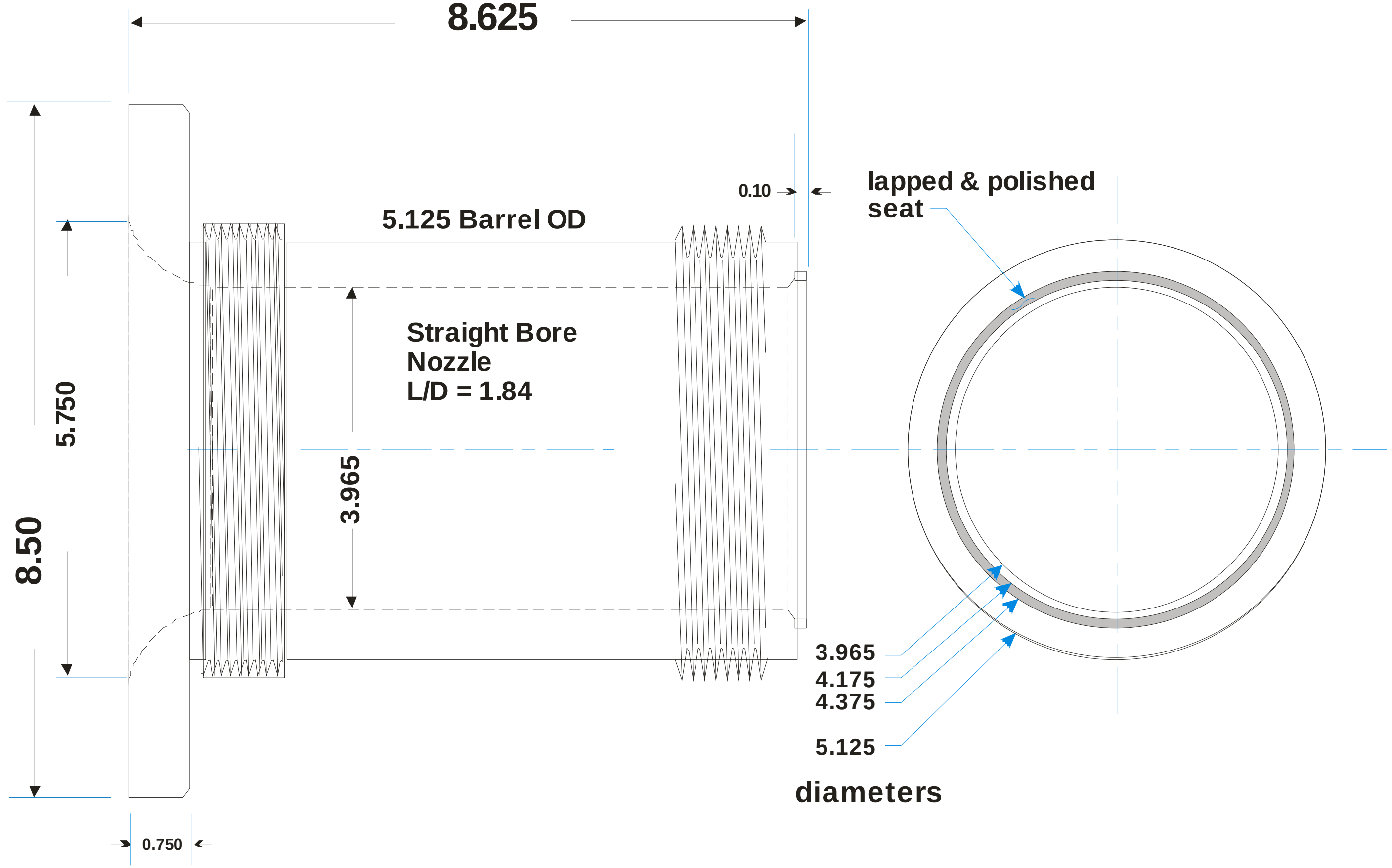


("P" Nozzle from Farris 2600PB10-120; 4 P 6)



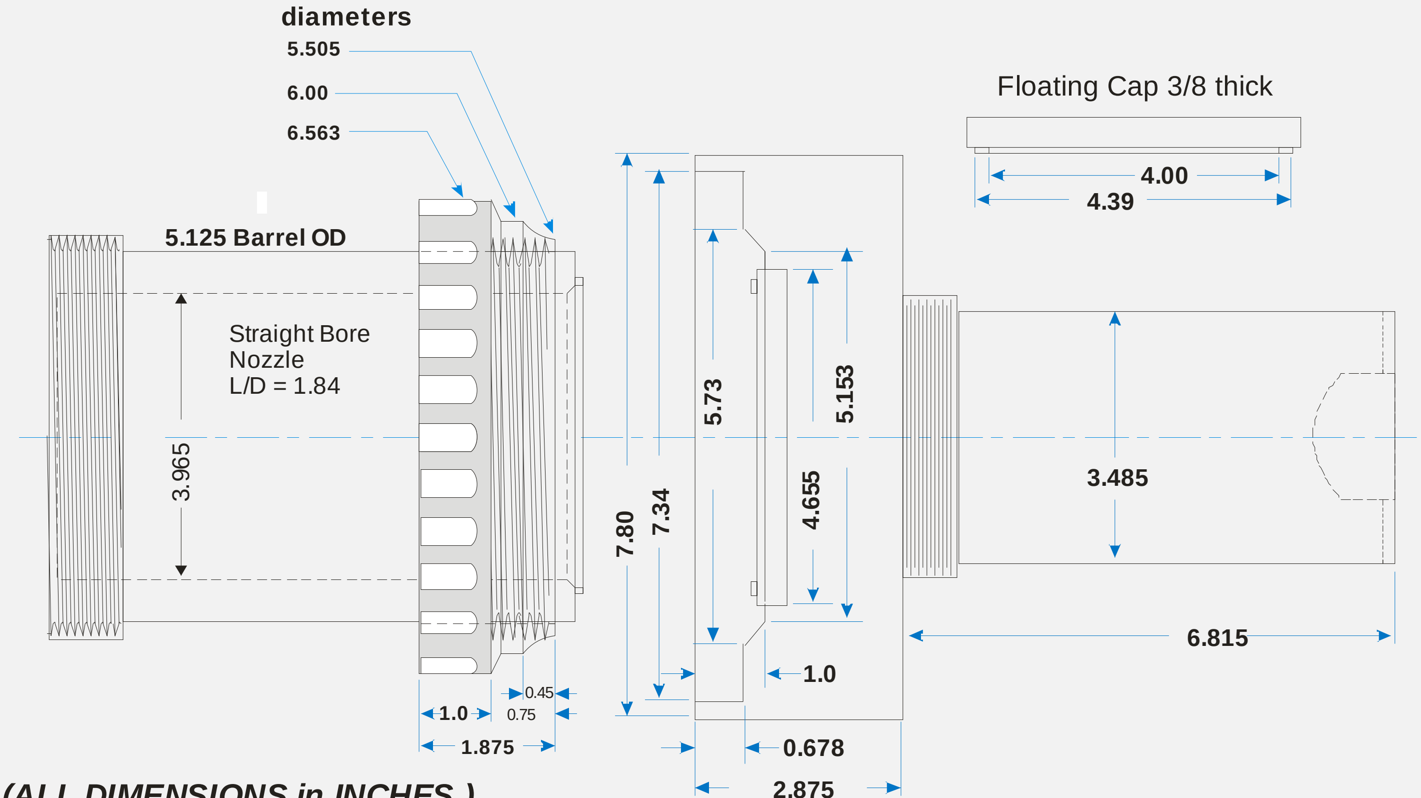
("P" Nozzle with Cap and Blowdown Ring from Farris 2600PB10-120; 4 P 6)


("P" Nozzle with Cap and
Blowdown Ring
from Farris 2600PB10-120; 4 P 6)
Cap
Huddle Chamber
45° angle
Blowdown Ring



("Q" Nozzle from Consolidated 1910-30-Qc; 6 Q 8)

(ALL DIMENSIONS in INCHES)



(ALL DIMENSIONS in INCHES)

**("Q" Nozzle Cap and Blowdown Ring
from Consolidated 1910-30-Qc;   6 Q 8)**

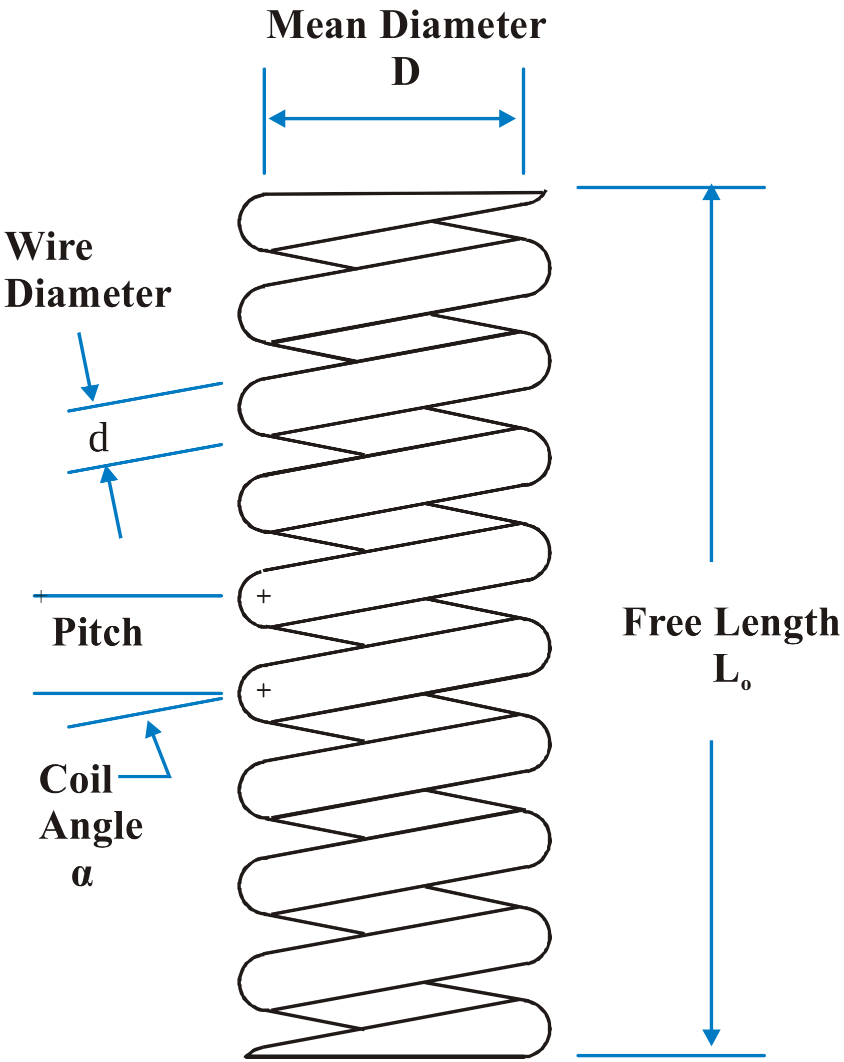
What can one do with this data ?

Perhaps more than might be expected

Spring Constant
Natural Frequency
Valve Opening Time
Cycles to Failure Projections

SPRING CONSTANT

Helical Springs

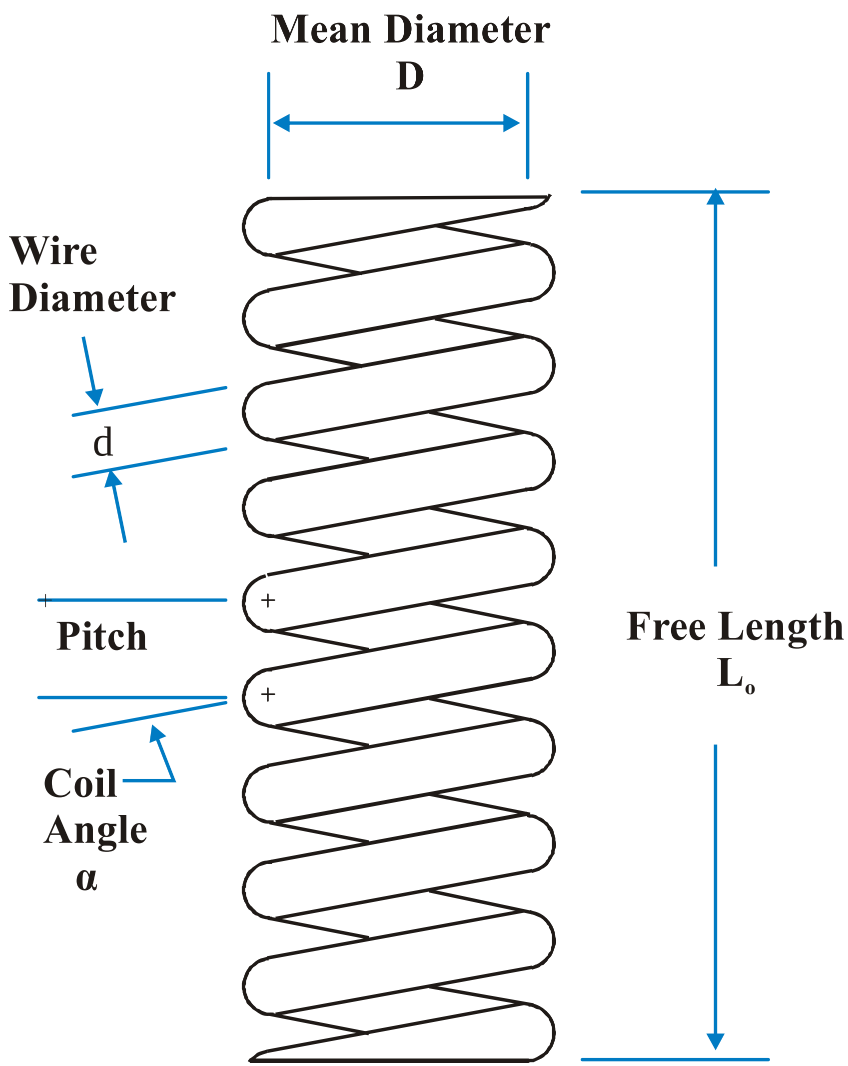


Relief Valves typically

have springs with closed ends ground such that the number of active coils is $Na = Nt - 2$

Where Nt is the total number of coils

Helical Springs



Nomenclature:

N_a is the number of active coils

G is the modulus of torsion or rigidity

C is the diameter modulus; C= D/d

D is the mean diameter

d is the coil diameter

Spring Constant k

$$k = \frac{G\,d}{8\,C^3\,N_a}$$

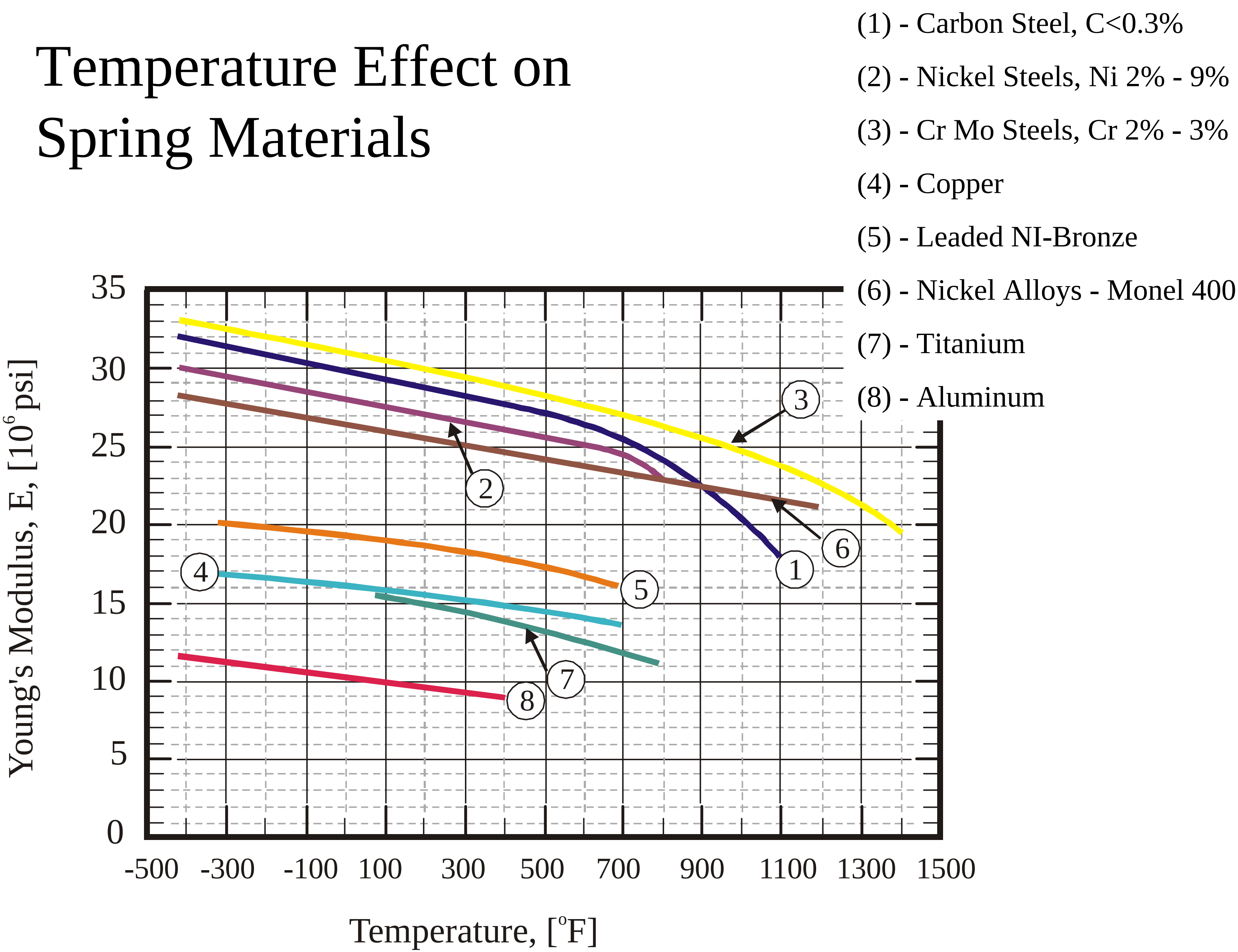
Young's Modulus (E) and Modulus of Torsion (G)

Metal	**Poisson's Ratio**	**E [10^9 Pa]**	**G [10^9 Pa]**
Aluminum	**0.330**	**69**	**27**
Copper	**0.360**	**117**	**43**
Ni-Steel	**0.310**	**213**	**76**
Stainless Steel 18-8	**0.300**	**201**	**73**
Carbon Steel	**0.303**	**202**	**79**
High Carbon Steel	**0.295**	**210**	**81**
Inconel	**0.290**	**214**	**79**

$$G = \frac{E}{2(1+v)}$$

Poisson's Ratio

Temperature Effect on Spring Materials



Approximation to SPRING CONSTANT

$$k = C_1 \left[\frac{P_{set} \; x \; A_{noz}}{Lift} \right]$$

$$C_1 = C_2 \; x \; C_3$$

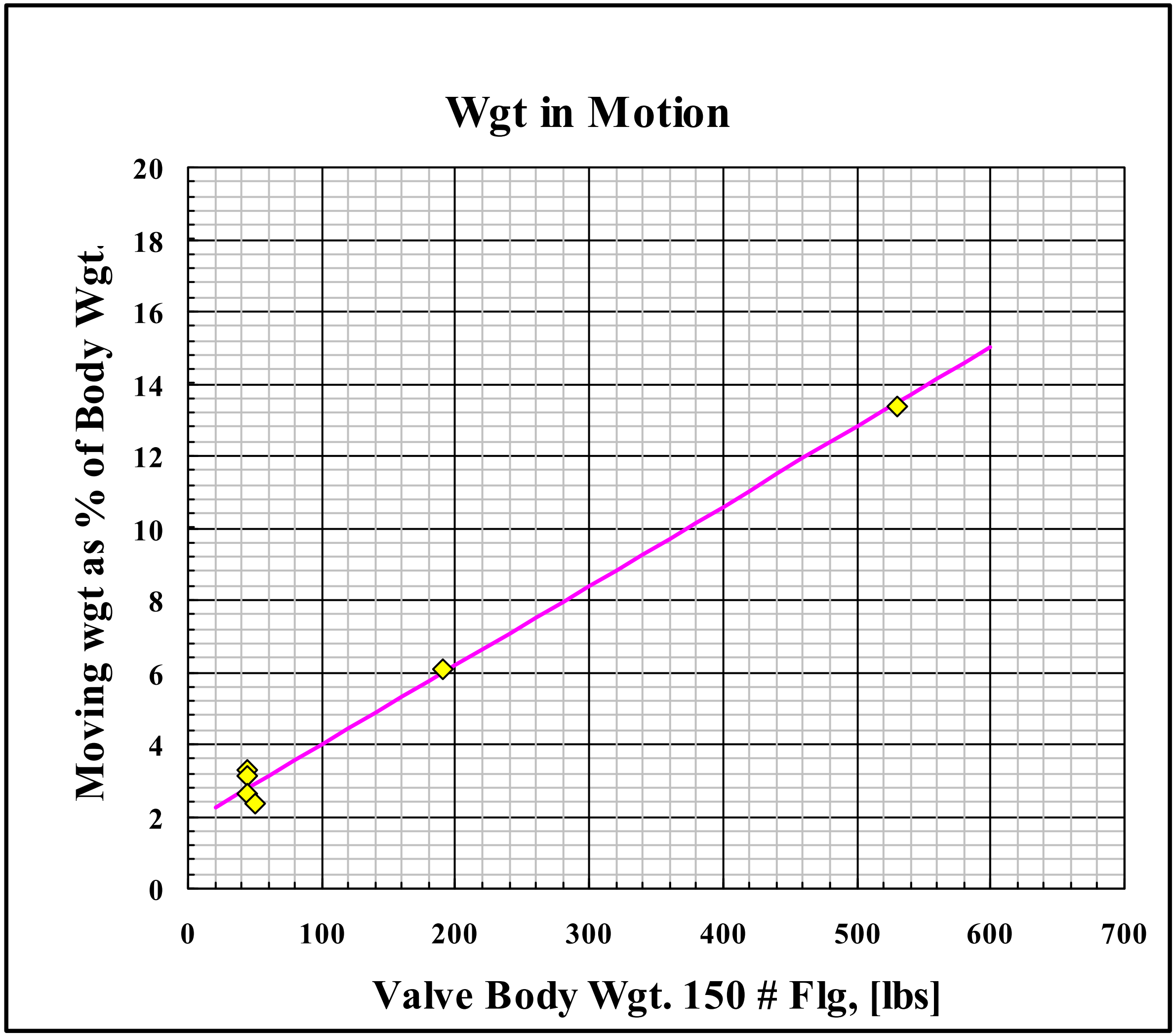
$$C_2 = \frac{P_{fullflow}}{P_{set}}$$

$$C_3 = \frac{A_{pop}}{A_{noz}}$$

C1 , C2, C3 are dimensionless constants that should be close to 1 in order of magnitude

Vendor		Consolidated Dresser	Consolidated Dresser	Farris	Farris	Farris	Consolidated Dresser
Model		1906Fc	1905Fc	26FA10	26GA10L	26PB10-120	1910-30-Qc
Type		11/2 F 2	11/2 F 2	11/2 F 2	11/2 G 21/2	4P6	6Q8
P_set	[psig]	100	120	180	100	50	185
Measured Spring k	[lbf/inch]	229.3	384.8	453.9	242.6	572.1	2365.0
Redbook: Nozzle Area	[sq. in.]	0.375	0.375	0.371	0.559	7.087	12.85
Redbook: Lift	[inch]	0.182	0.182	0.206	0.326	0.901	1.09
C_1	[-]	1.113	1.556	1.400	1.415	1.455	1.084
C_2 (*assumed*)	[-]	1.05	1.05	1.05	1.05	1.05	1.05
C_3 (A_pop / A_noz)	[-]	1.060	1.482	1.334	1.348	1.385	1.033
C_1 average	[-]	1.337					
C_3 Average	[-]	1.274					
Approximation to							
Spring Constant	[lbf/inch]	275.5	330.6	433.5	229.3	525.9	2916.5
Error	[%]	-20.15	14.08	4.50	5.49	8.06	-23.32

Mass in Motion



Mass in Motion

$$\%\,wim \;=\; 1.8 + 0.22\ x\ BW$$

$$wim \;=\; (\%\,wim\ x\ BW)\,/\,100$$

wim   wgt in motion
BW   body wgt for unit with 150 # flg

Mass in Motion

Vendor		C-D	C-D	Farris	Farris	Farris	C-D
Model		1906Fc	1905Fc	26FA10	26GA10L	26PB10-120	1910-30-Qc
Bellows:		No	No	No	No	Yes	Yes
Type		11/2 F 2	11/2 F 2	11/2 F 2	11/2 G 21/2	4P6	6Q8
P_set	[psig]	100	120	180	100	50	185
Catalog Weight 150# FLG	[lbs]	45	45	44	50	190	530
Total wgt. In motion (measured)	[g]	676.8	640.3	528.0	540.3	5234.1	32204.3
Total wgt. In motion	[lbs]	1.49	1.41	1.16	1.19	11.53	70.93
Ratio - moving wgt/body wgt	[%]	3.31	3.13	2.64	2.38	6.07	13.38
Correlation:							
% Wgt in Motion (wim)	[%]	2.79	2.79	2.768	2.9	5.98	13.46
Est wim	[lbs]	1.2555	1.2555	1.21792	1.45	11.362	71.338
Error	[%]	15.8	11.0	-4.7	-21.8	1.4	-0.6

Natural Frequency

$$Spring\ Cons.:$$

$$k \quad = \quad C_1 \frac{P_{set}\ A_{noz}}{Lift}; \qquad C_1 = 1.34$$

$$Wgt\ in\ motion:$$

$$wim \quad = \quad (1.8 + 0.022\ x\ BW)\ x\ BW\ /\ 100$$

$$Natural\ Frequency:$$

$$f_{nat} \quad = \quad \frac{1}{2\pi}\sqrt{\frac{k}{wim}}$$

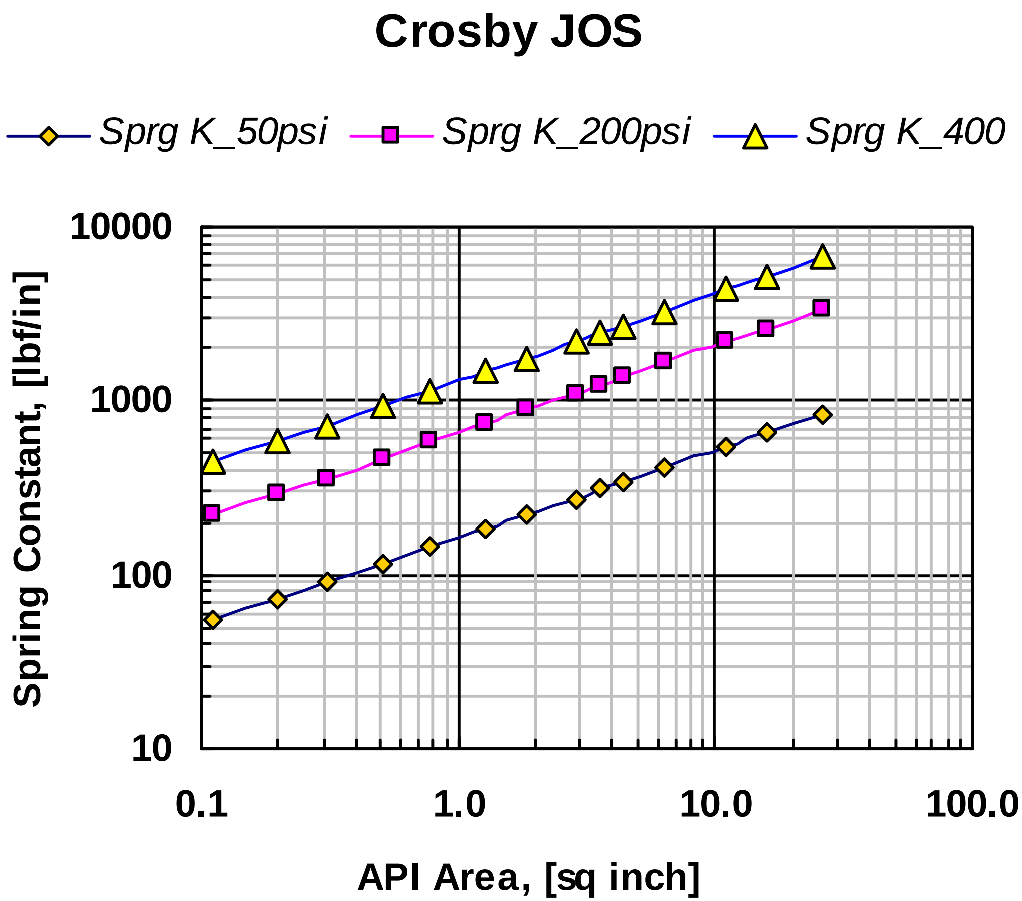
Natural Frequency

Vendor		C-D	C-D	Farris	Farris	Farris	C-D
Model		1906Fc	1905Fc	26FA10	26GA10L	26PB10-120	1910-30-Qc
Bellows:		No	No	No	No	Yes	Yes
Type		11/2 F 2	11/2 F 2	11/2 F 2	11/2 G 21/2	4P6	6Q8
P_set	[psig]	100	120	180	100	50	185
Catalog Weight 150# FLG	[lbs]	45	45	44	50	190	530
Total wgt. In motion (measured)	[g]	676.8	640.3	528.0	540.3	5234.1	32204.3
Total wgt. In motion	[lbs]	1.49	1.41	1.16	1.19	11.53	70.93
Ratio - moving wgt/body wgt	[%]	3.31	3.13	2.64	2.38	6.07	13.38
Est wim	[lbs]	1.26	1.26	1.22	1.45	11.36	71.34
Measured Spring k	[lbf/inch]	229.3	384.8	453.9	242.6	572.1	2365.0
Estimated Spring Constant	[lbf/inch]	275.5	330.6	433.5	229.3	525.9	2916.5
Natural Frequency f_nat							
Measured f_nat	[Hz]	38.8	51.6	61.7	44.6	22.0	18.0
Estimated Natural Frequency	[Hz]	46.3	50.7	59.0	39.3	21.3	20.0
Error	[%]	-19.4	1.8	4.5	11.9	3.4	-10.7

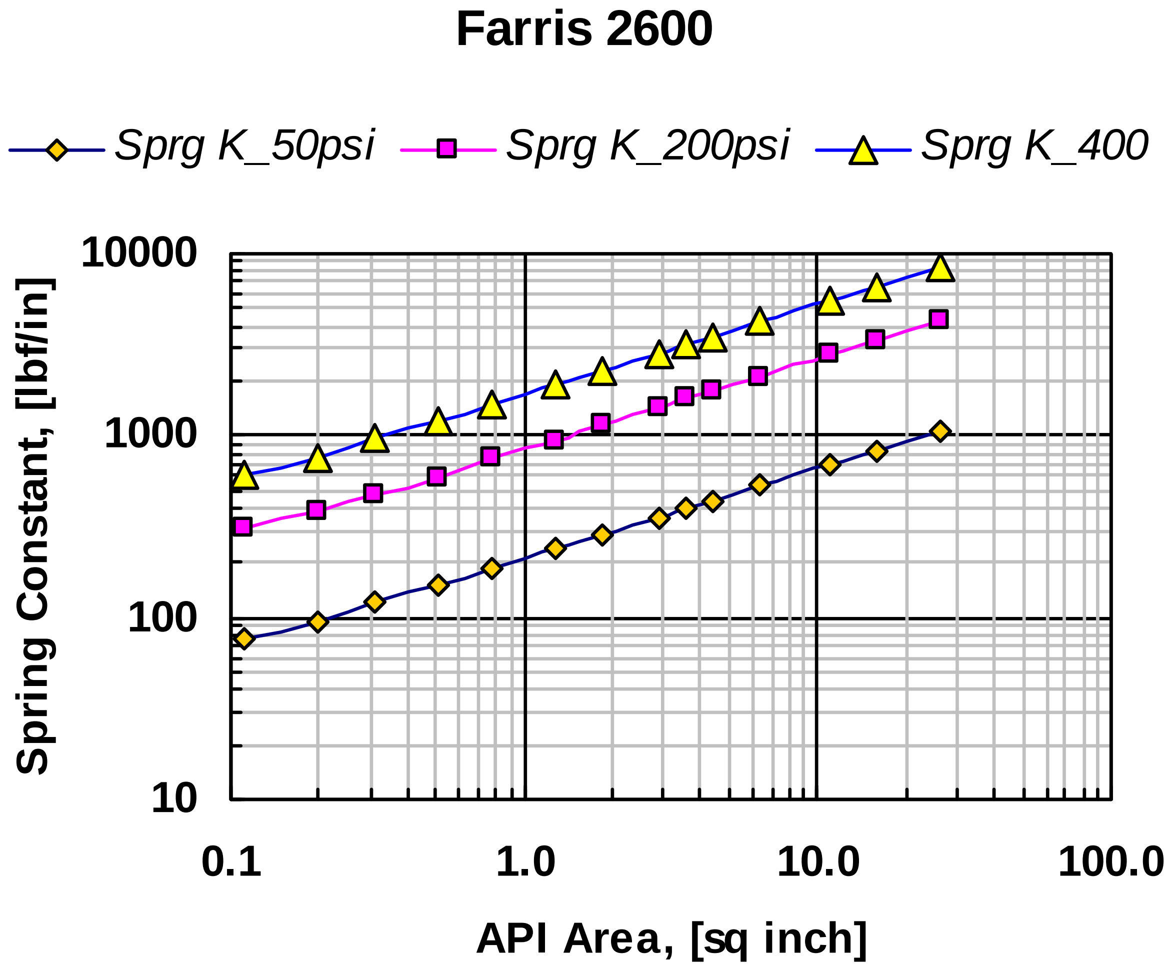
Extrapolate to the Valve Catalogs

Farris
Crosby
Consolidated

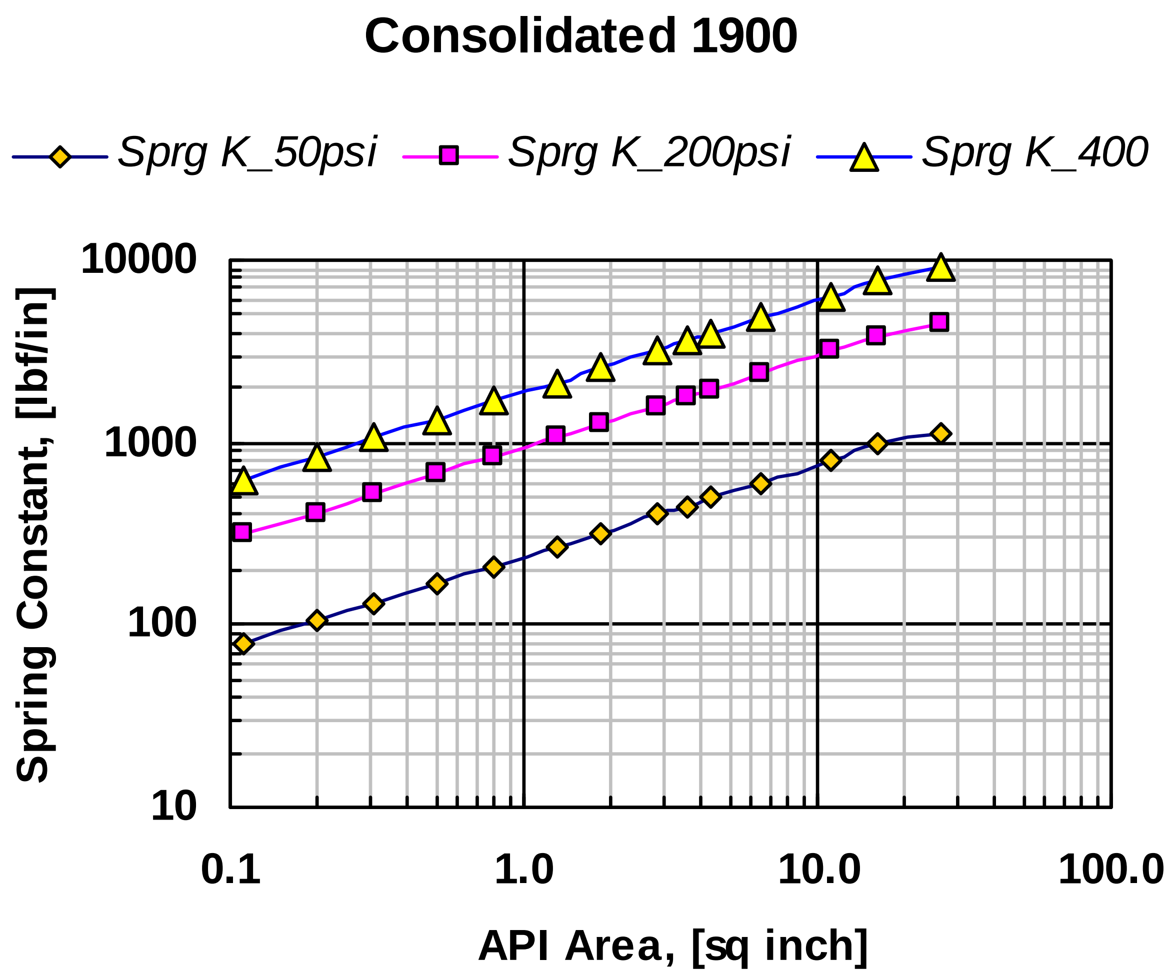
Relief Valve Spring Constant Estimate



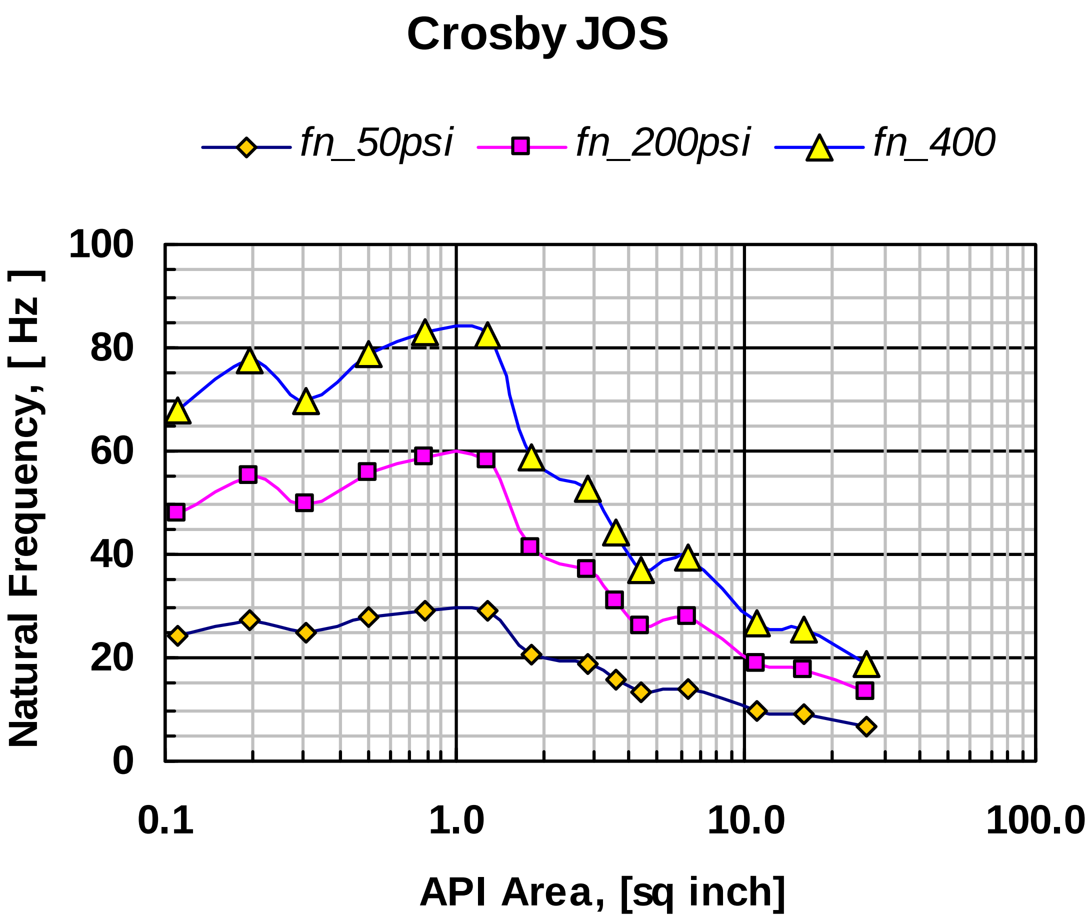
Relief Valve Spring Constant Estimate



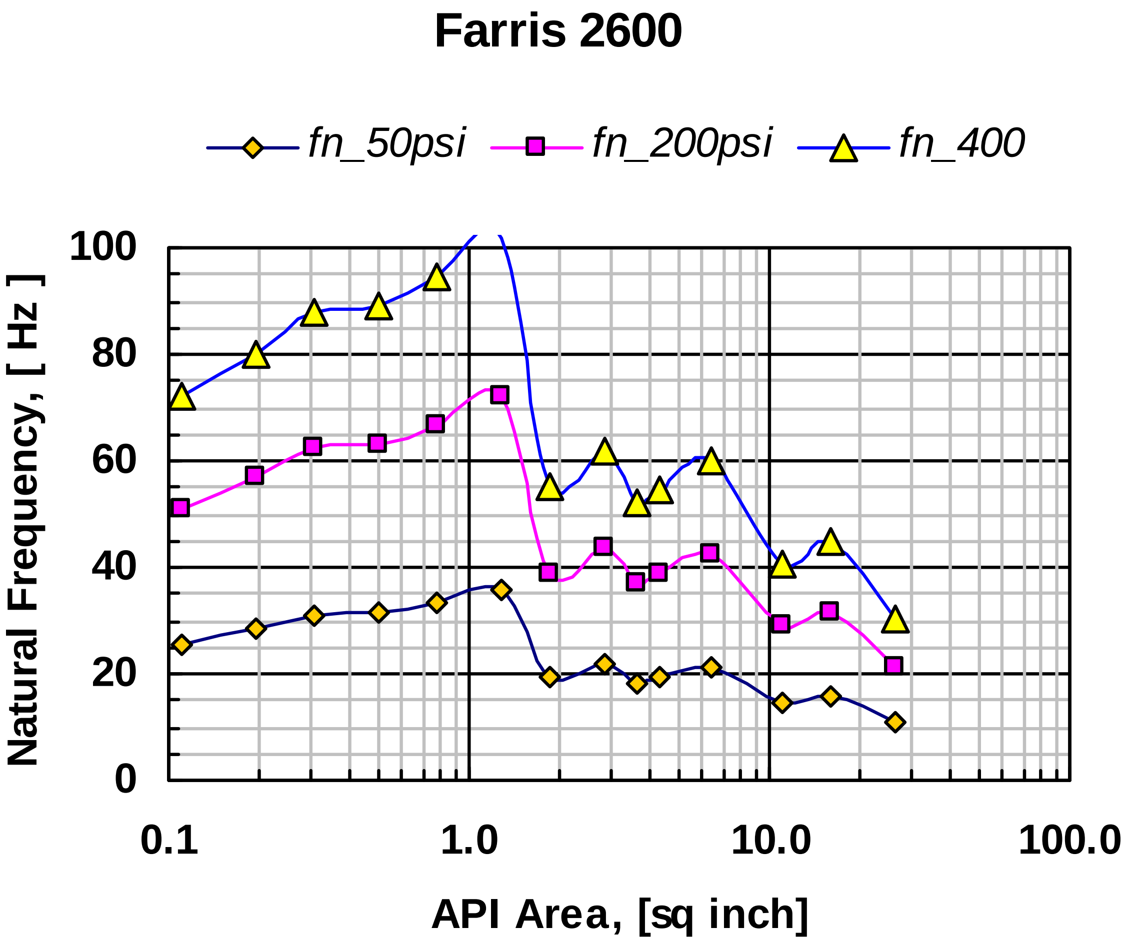
Relief Valve Spring Constant Estimate



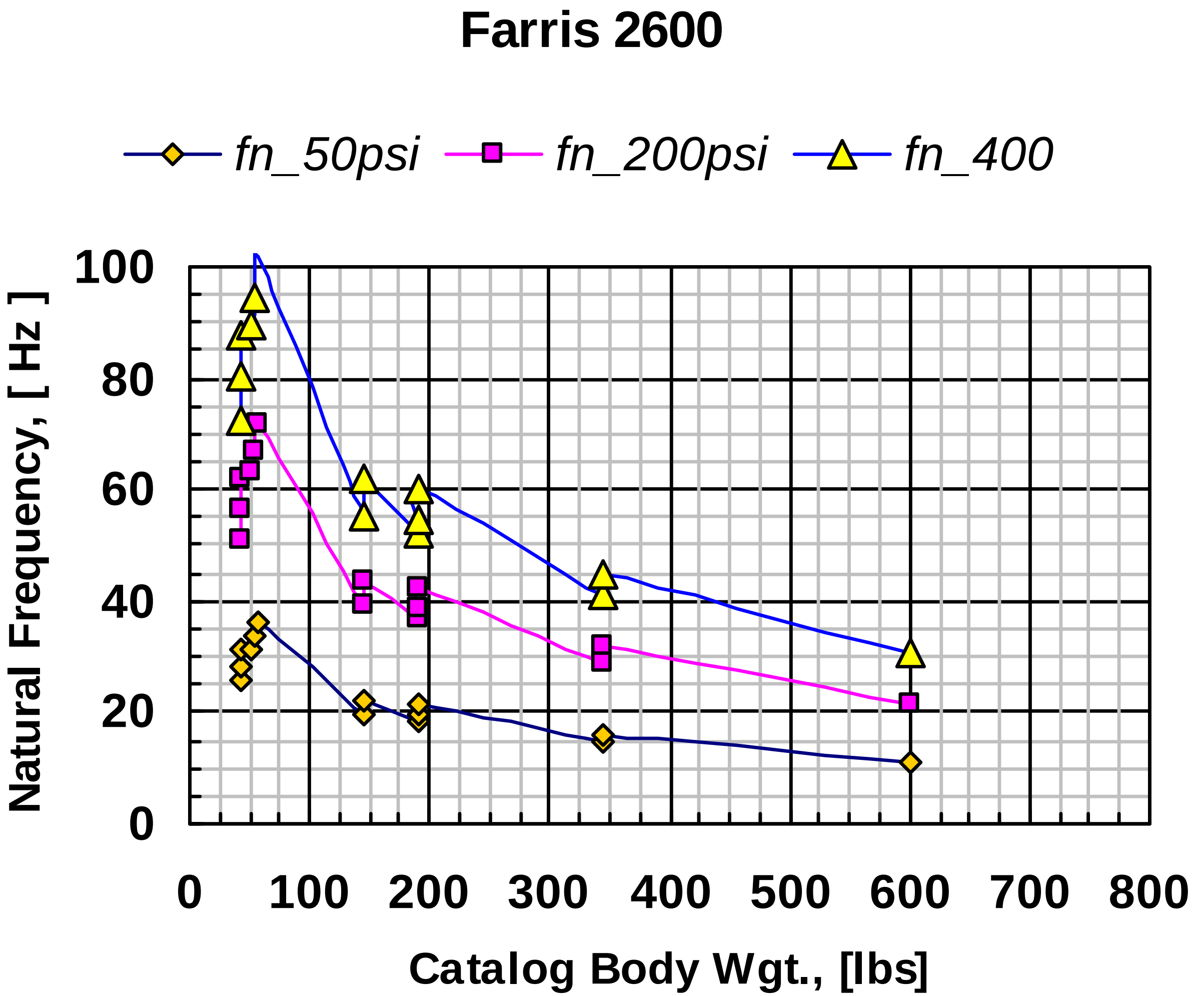
Relief Valve Natural Frequency Estimate



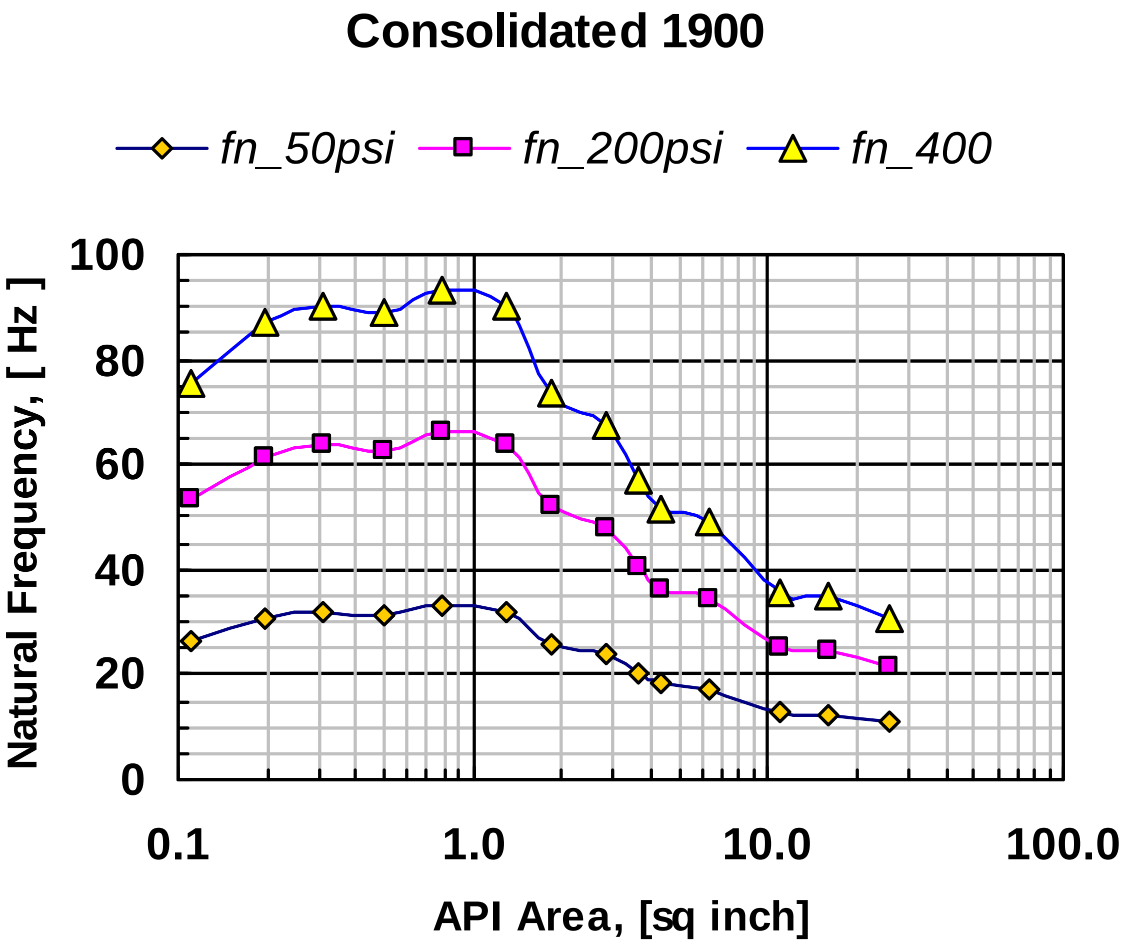
Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Valve Opening Time

We have a derivation that leads to the result:

$$t_{open} = \frac{\sqrt{\dfrac{2}{\left(\dfrac{A_{pop}}{A_{noz}} - 1\right)}}}{2\,\pi\, f_n}$$

Example:

A_pop / A_noz = 1.3

f_n = 50 sec-1

t_open = 8.5 ms

Valve Failure Modes

Seat or Cap Destruction due to repeated cyclic operation – chattering at f_n
Also possibility for damage at guide stops

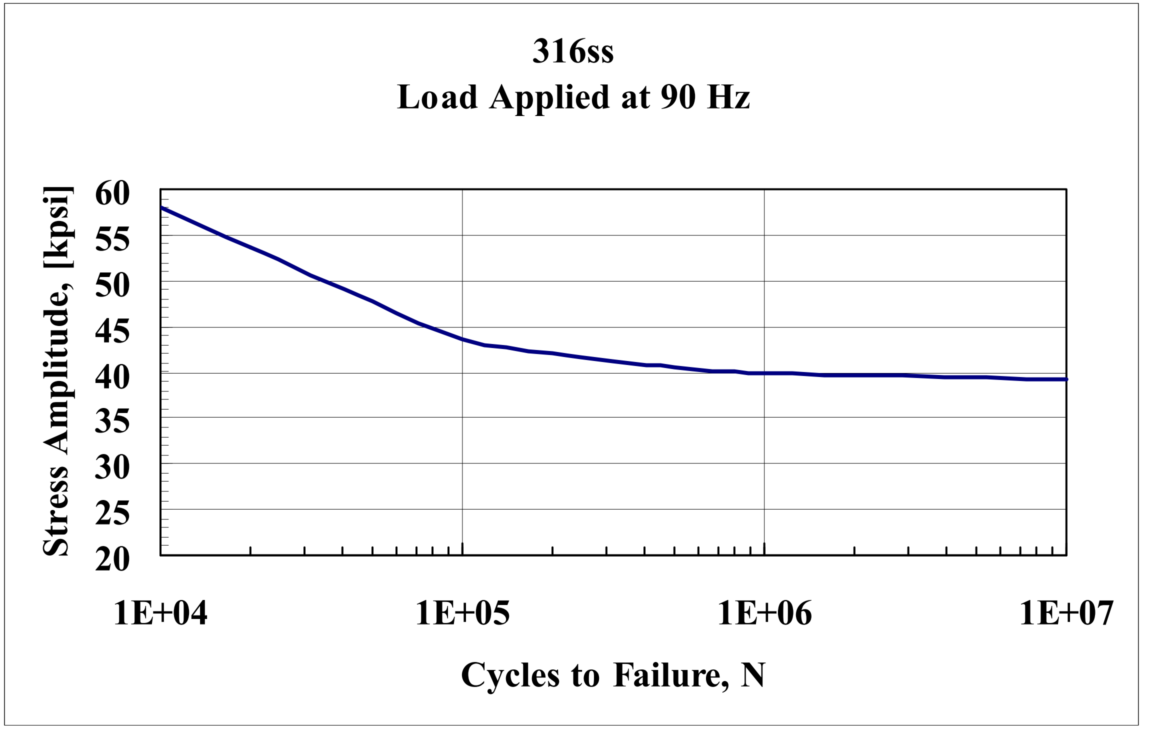
Seizing at or on the Spring Rod Guide

Bellows Failure in repeated cyclic operation – chattering at f_n

Spring Failure

Cycles to Failure

This is a well-established concept for fatigue failure under cyclic stresses.



Cycles to Failure (*pressure stress*)

Applied Stress on the Valve seat

$$s_{[psi]} = C_1 P_{set} \frac{A_{noz}}{A_{impact}}$$

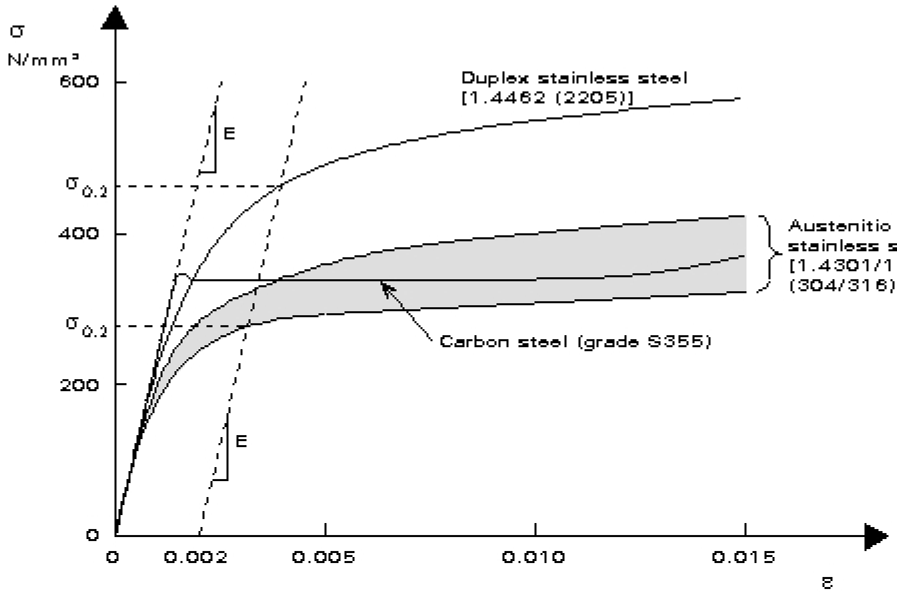
Vendor		**C-D**	**C-D**	**Farris**	**Farris**	**Farris**	**C-D**
Model		**1906Fc**	**1905Fc**	**26FA10**	**26GA10L**	**26PB10-120**	**1910-30-Qc**
Bellows:		**No**	**No**	**No**	**No**	**Yes**	**Yes**
Type		**11/2 F 2**	**11/2 F 2**	**11/2 F 2**	**11/2 G 21/2**	**4P6**	**6Q8**
P_set	**[psig]**	**100**	**120**	**180**	**100**	**50**	**185**
A_noz / A_impact	**[-]**	**2.219**	**2.219**	**4.947**	**2.781**	**6.328**	**4.784**
Impact Stress	**[psi]**	**297**	**357**	**1193**	**373**	**424**	**1186**

Cycles to Failure (*impact energy*)

Impact Energy:

Kinetic Energy = ½ wim (2 Lift f_n)^2

Energy to Yield = ½ σ ε $_{\text{\_yield}}$

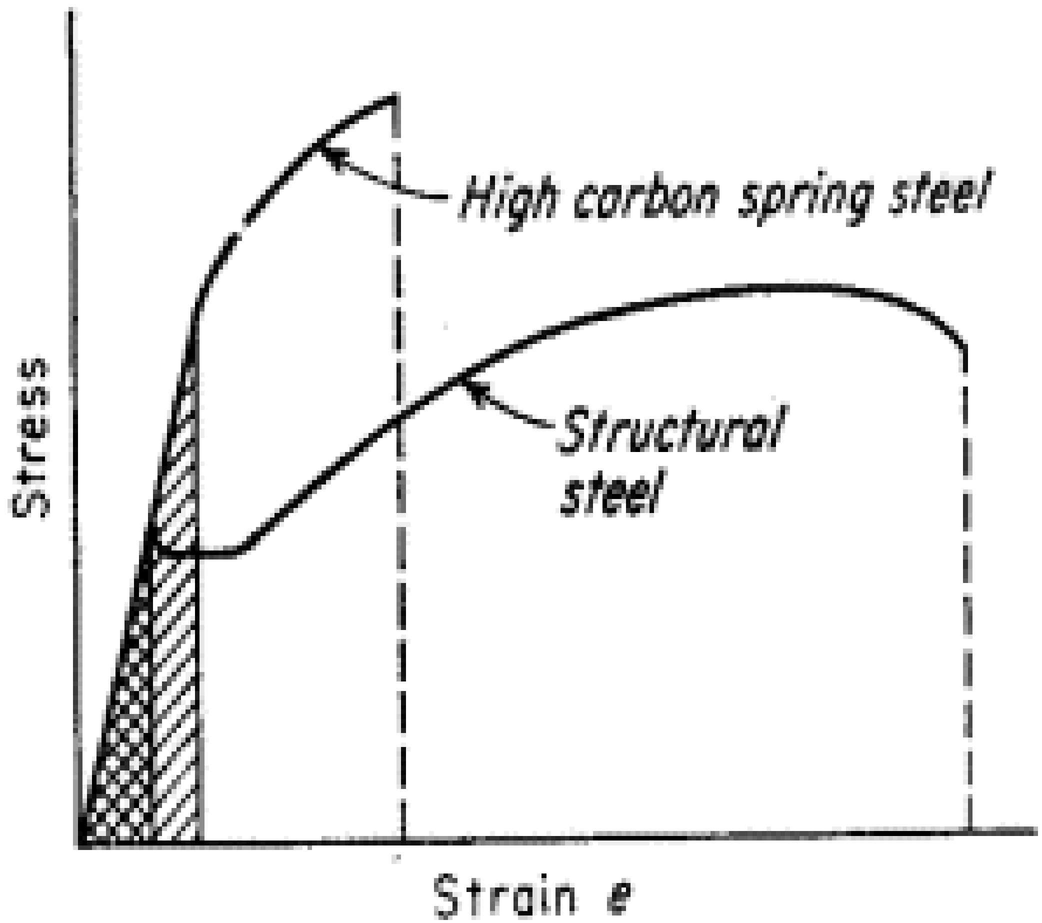


Typical stress-strain curves for stainless steel and carbon steel

($\sigma_{0.2}$ is the 0.2% proof strength, E is Young's modulus)

Note: These values are typical experimental values and should not be used in design. For grades 1.4301 (304) and 1.4401 (316) steels, the two curves shown indicate the extreme values from a series of tests and thus they represent a scatter band.

Stress Strain Diagram



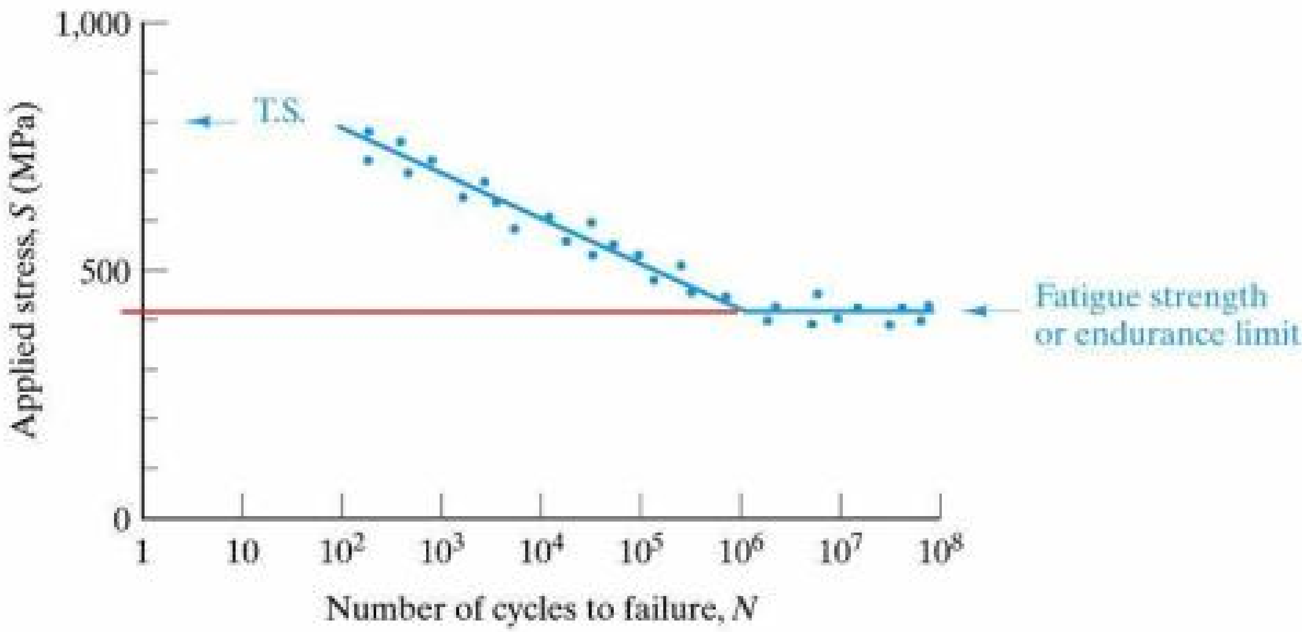
Energy to Yield

For Stainless Steel
About 400 kJ / m^3 or About 0.05 J/g

Now Consider the valve nozzle impact ring:
The energy to yield is approximately
400 x 2.5 x A_noz / (1/(A_noz/A_impact))

Impact Energy to Yield

Vendor		C-D	C-D	Farris	Farris	Farris	C-D
Model		1906Fc	1905Fc	26FA10	26GA10L	26PB10-120	1910-30-Qc
Bellows:		No	No	No	No	Yes	Yes
Type		11/2 F 2	11/2 F 2	11/2 F 2	11/2 G 21/2	4P6	6Q8
P_set	[psig]	100	120	180	100	50	185
Wgt in Motion	[kg]	0.67683	0.64033	0.528	0.54033333	5.234	32.2
Nat Freq by Formula	[Hz]	38.7654	51.6288	61.7487	44.6267915	22.02	18.05
A_noz / A_impact	[-]	2.219	2.219	4.947	2.781	6.328	4.784
Kinetic Energy in Motion	[J]	0.04347	0.07295	0.11024	0.1475658	2.658	16.08
Impact Surface: Energy to yield	[J]	0.10903	0.10903	0.04839	0.12967716	0.723	1.733



Typical fatigue curve. (Note that a log scale is required for the horizontal axis.)

Cycles to Failure

The previous result suggests that seat impact stresses are low at low set pressure and for small relief valves. The relief valve should have a long lifetime in cyclic operation. (greater than 10^7 cycles – this is equivalent to 55 hrs of continuous cyclic operation at 50 Hz)

Cycles to Failure

However, for large relief valves, one can clearly point to a mechanism for nozzle seat damage such that failure to reseat to leak tightness after a chattering event could well be expected.

This would hold for any point of contact for which the impact kinetic energy could be focused – perhaps enhanced by any misalignment.

Cycles to Failure Bellows

One finds similar results for bellows: However, we do not well understand how the bellows assembly is designed.
For example if the cyclic stress from full closed to full open is much less than the bellows assembly yield stress limit, then 10^6 cycles or greater should be expected

Failure by Seizing

We postulate a seizing index as the product of f_n x A_guide
Where A_guide is the spring rod guide surface area

Vendor		C-D	C-D	Farris	Farris	Farris	C-D
Model		1906Fc	1905Fc	26FA10	26GA10L	26PB10-120	1910-30-Qc
Bellows:		No	No	No	No	Yes	Yes
Type		11/2 F 2	11/2 F 2	11/2 F 2	11/2 G 21/2	4P6	6Q8
P_set	[psig]	100	120	180	100	50	185
Nat Freq by Formula	[Hz]	38.8	51.6	61.7	44.6	22.0	18.0
Guide Surface Area	[sq inch]	5.8	5.8	2.4	2.0	16.3	49.1
Seize Index							
f_n x A_guide	[sq in/sec]	224	298	147	89	358	887

Failure by Seizing

One observes that this index tends to increase with increasing valve size and set pressure.

Many other factors can effect seizing including lubrication and cleanliness or the presence of foreign articles in the guide path.

We conjecture that seizing failures will be predominantly “fail-open” type failures for reasons that might be obvious, but will be explained.

Overall Failure in Cyclic Operation

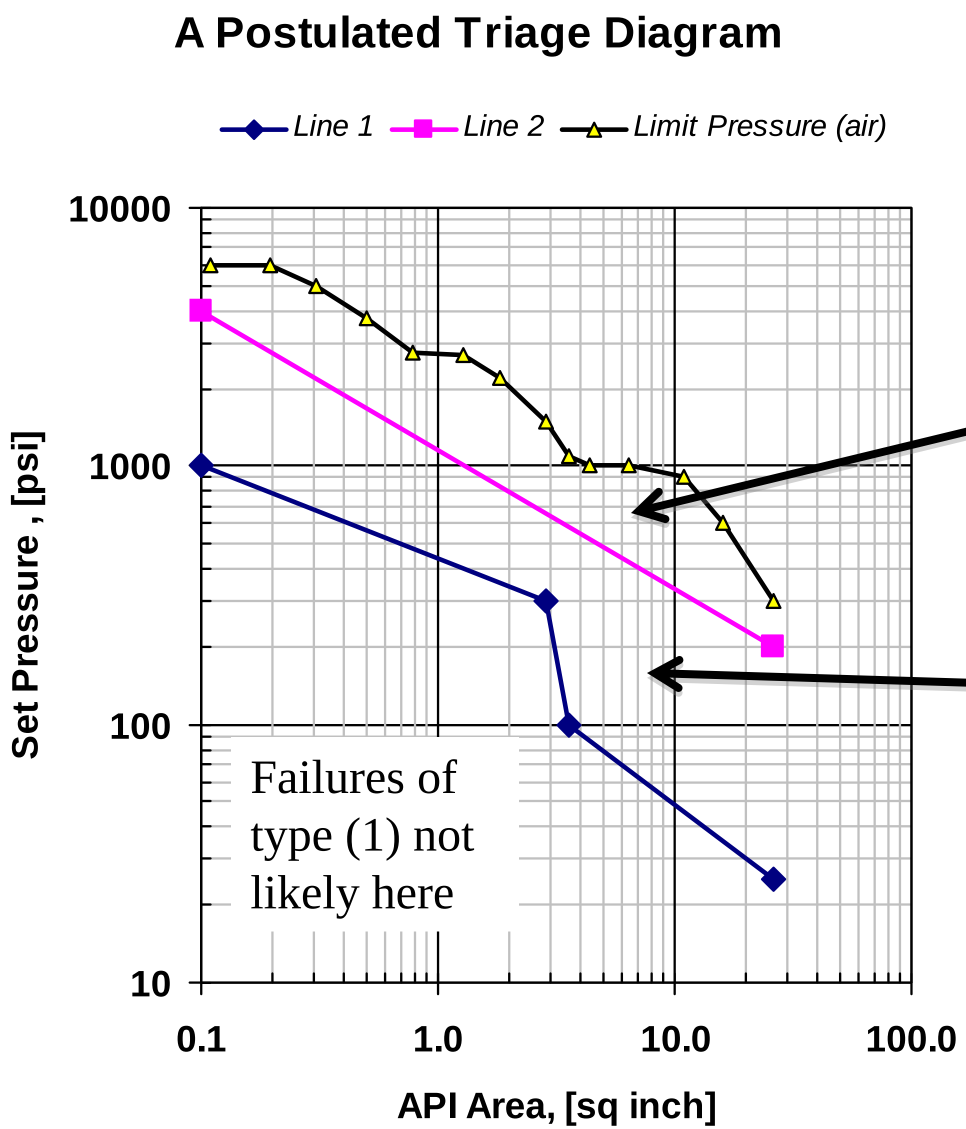
Let one define three modes of failure that might be attributed to cyclic operation

(1) Failure severe enough such that there is a likely failure to pass the required relief flow for a single incident - excessive nozzle damage; possible for bellows failure, possible for seizing failure.

(2) Failure that is not a threat for a single incident, but may affect valve closure or future demand actuations

(3) Failure that may affect valve leak tightness.

Overall Failure in Cyclic Operation



This is a diagram calling for failure data to provide a calibration guide

Type (1) failures possible here

Failures of type (2) and (3) should be considered here

Overall Failure in Cyclic Operation

What is learned is that relief valves are mostly tolerant to cyclic operation.

We can see how this tolerance is significantly degraded for large relief valve and for high set pressures.

We also know that relief valves fail, but details regarding failure are hard to come by.

Overall Failure in Cyclic Operation

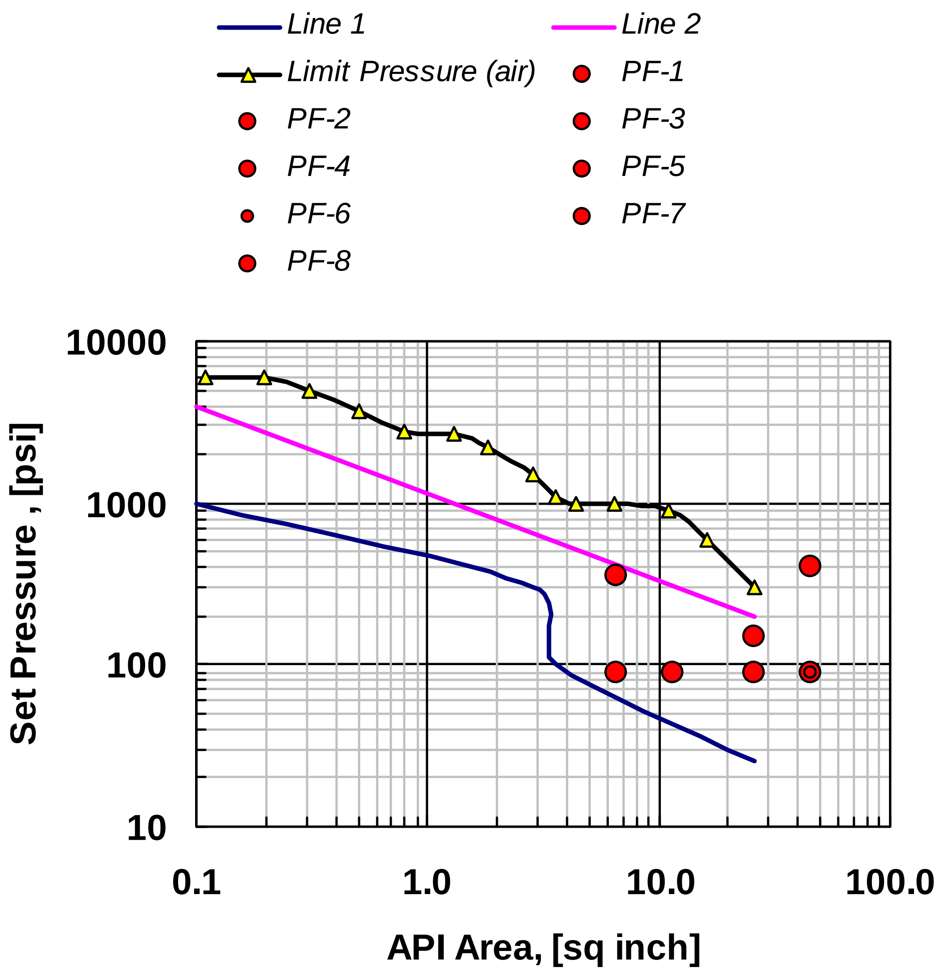
We need good reporting on valve failure modes and damage effects.

It would be a step forward to be able to put such cases on a diagram such as the preceding.

The following is an imperfect illustration, but a start. (Information by courtesy of H.G. Fisher)

Overall Failure in Cyclic Operation

A Postulated Triage Diagram



Red Data Points represent reported failures of large pilot operated relief valves due to excess inlet length. The chattering was attributed mostly to acoustic phenomena

The failures are real. The valve internals are quite different from API 526 Type PSVs

Things one would like to see from Relief Valve Manufacturers

(1) Valve spring constants and natural frequency data – if not in the general catalog at least with the purchased valve certification sheets

(2) Better relief valve Lift and Flow curves.

(3) Better data and guidance on relief valve blowdown

(4) Mechanical data on the number of on/off cycles that can be safely tolerated. Is this number 10 or 10^6 ; It makes a profound difference

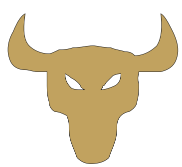
(5) For anyone – better reporting of actual failures.

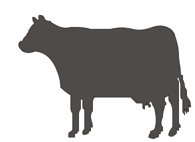
Important Things to Remember

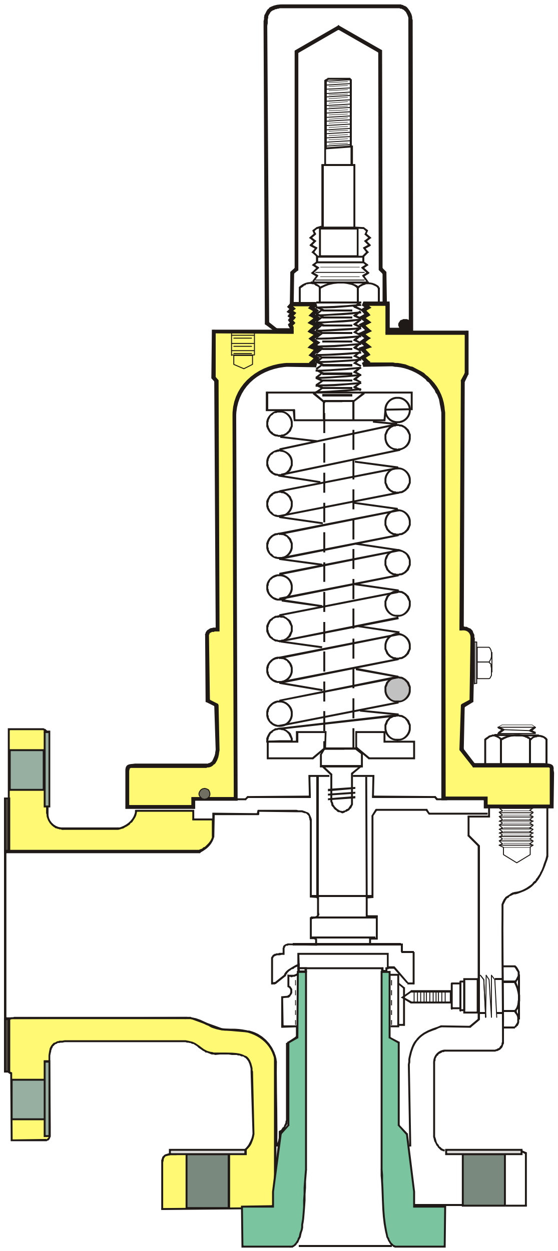
- With all of the preceeding we would like to leave you with an important point to remember

Important Things to Remember

Always draw the drinking water upstream from the herd







There is still much more to this story, but we are getting closer - for now

That's All Folks!!

Centaurus Technology, Inc.

ioMosaic

Business Confidential Document

Pressure Relief Valve Stability with SuperChems

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

DIERS Users Group Meeting

October 2011

Chicago

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

Business Confidential Document

Pressure Relief Valve Stability with SuperChems

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

API Fall Meeting

November 14, 2011

Los Angeles

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

Since 2003, the DIERS users group has been developing models and tools for the assessment of PRV stability

- Advanced tools are useful for a good engineering analysis
  - Gain more insight into PRV stability
  - Gain more insight into system dynamics and performance
- We need to better understand overall system dynamics
  - The majority of practical scenarios deal with finite reservoirs
  - We need to better understand what happens to the system on the way to 10 % overpressure for example
- The DIERS users group will make these models available in SuperChems for DIERS starting with version 6.4mp

ioMosaic

To accurately assess whether a "relief system" will operate in a stable manner (chatter free) you must consider the following important system time constants and how they interact

- Valve time constant
  - How fast does a pressure relief device close and open?
- Vessel or pressure source time constant
  - How fast does a vessel de-pressure and re-pressure after a pressure relief device opens and re-seats?
- Inlet line time constant
  - How long does it take for a pressure wave to propagate upstream from a pressure relief device to the pressure source and back?
- Outlet line time constant
  - Acoustic barriers may be established due to "body bowl choking"
  - Note that acoustic barriers, such as the presence of control valves, change in diameter, etc. can cause standing waves that can lead to acoustic coupling or resonance with relief systems components

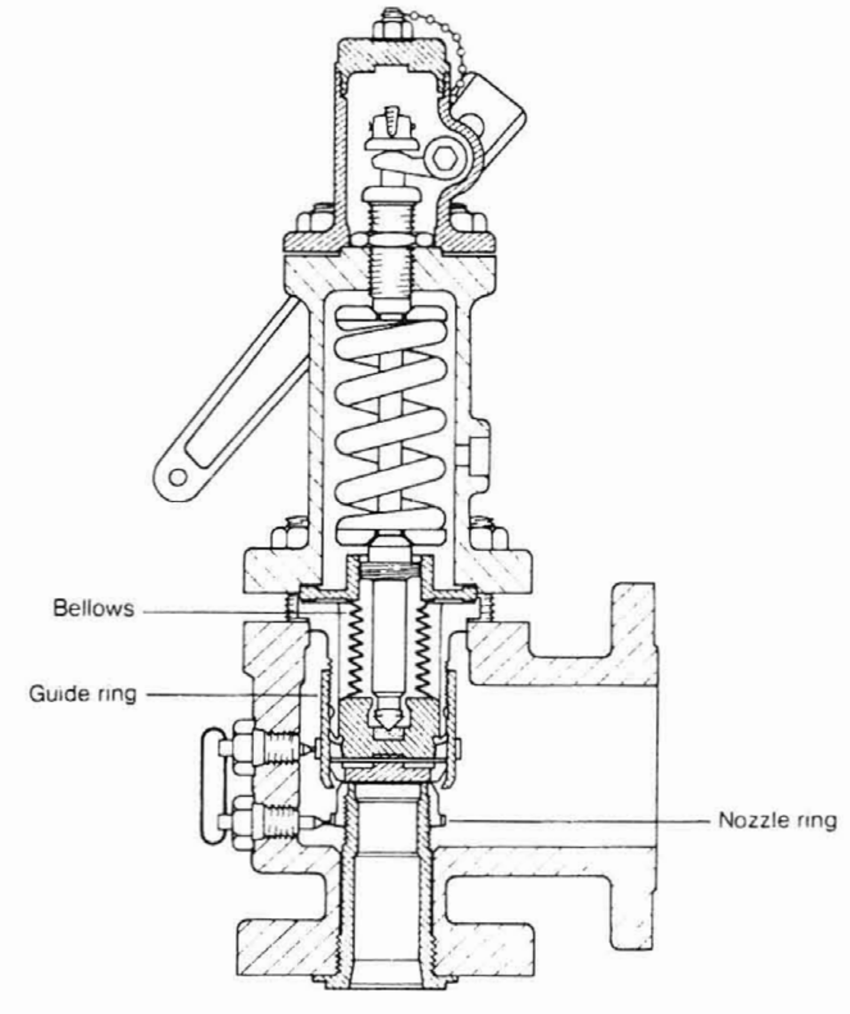
ioMosaic

What can one do without the use of a detailed dynamics model to establish if PRV stability is an issue for a specific installation?

- The key is to decouple PRV frequency from the piping and other important system frequencies or to slow down the valve response (closure) time
  - For example, it has been shown that very long inlet lines will result in stable PRV operation. Why?
  - It has also been shown that inlet lines with expanders and reducers will result in stable PRV operation. Why?
- Since liquid flow is not typically choked, there are less uncertainties for liquid flow solutions
  - The fluid dynamics equations are much easier to solve using 1D or 2D partial differential equation solutions for all liquid flow

The SuperChems PRV stability model consists of four important components

- Dynamic vessel models for single and multiphase reacting flows
- Steady state and dynamic piping models for single and multiphase reacting flows
- Shock discontinuity and multiple chokes methods for flow through pipes of single and multiphase reacting flows
- Dynamic single degree of freedom (SDOF) PRV model



ioMosaic

Detailed SuperChems two-phase flow dynamics for vessels – internal energy balance

$$\left[N_{\text{metal}} C_{v_{\text{metal}}} + \sum_i^C (N_i + n_i) C_{v_i} + N_T \frac{\partial \Delta \underline{U}_v}{\partial T} + n_T \frac{\partial \Delta \underline{U}_l}{\partial T}\right] \frac{dT}{dt} \quad +$$

$$\left[N_T \frac{\partial \Delta \underline{U}_v}{\partial P} + n_T \frac{\partial \Delta \underline{U}_l}{\partial P}\right] \frac{dP}{dt} \quad +$$

$$\sum_i^C \left(\underline{U}_{i,ref} + \int_{T_{ref}}^{T} C_{v_i} dT + \Delta \underline{U}_v + N_T \frac{\partial \Delta \underline{U}_v}{\partial N_i}\right) \frac{dN_i}{dt} \quad +$$

$$\sum_i^C \left(\underline{U}_{i,ref} + \int_{T_{ref}}^{T} C_{v_j} dT + \Delta \underline{U}_l + n_T \frac{\partial \Delta \underline{U}_l}{\partial n_i}\right) \frac{dn_i}{dt} \quad =$$

$$\dot{Q} - \dot{H}_{l,out} - \dot{E}_{l,out} - \dot{H}_{v,out} - \dot{E}_{v,out}$$

ioMosaic

Detailed SuperChems two-phase flow dynamics for vessels – volume, mass, and phase equilibrium balances

Volume

$$(V_v\beta_v + V_l\beta_l)\frac{dT}{dt} - (V_v\kappa_v + V_l\kappa_l)\frac{dP}{dt} + \sum_i \overline{V}_{v_i}\frac{dN_i}{dt} + \sum_i \overline{V}_{l_i}\frac{dn_i}{dt} = 0$$

Mass

$$\frac{dn_i}{dt} + \frac{dN_i}{dt} = -\dot{n}_{i,out} - \dot{N}_{i,out} + \dot{R}_i \quad (\text{for } i = 1, \cdots, C)$$

Equilibrium

$$\frac{N_i}{N_T} - \frac{n_i}{n_T}\frac{\hat{\phi}_{l,i}}{\hat{\phi}_{v,i}} = 0 \quad (\text{for } i = 1, \cdots, C)$$

Detailed SuperChems flow dynamics for reacting two-phase flow in pipes – mass and volume

Vapor Mass

$$\dot{M}_T = \sum^{C} \dot{N}_j M_{w_j} \quad = \quad \rho_v u_v \alpha A$$

Liquid Mass

$$\dot{m}_T = \sum_{j}^{C} \dot{n}_j M_{w_j} \quad = \quad \rho_l u_l (1-\alpha) A$$

Total Mass

$$\frac{d\dot{n}_i}{dz} + \frac{d\dot{N}_i}{dz} = A\dot{R}_i \quad (\text{for } i = 1, \cdots, C)$$

Vapor Volume

$$\frac{d\rho_v}{dz} = \underbrace{\rho_v \kappa_v}_{R_1} \frac{dP}{dz} \underbrace{- \rho_v \beta_v}_{R_0} \frac{dT}{dz} + \underbrace{\frac{\rho_v}{\dot{M}_T} \sum_{j}^{C} \left[M_{w_j} - \rho_v \overline{V}_{v_j} \right] \frac{d\dot{N}_j}{dz}}_{R_n}$$

Liquid Volume

$$\frac{d\rho_l}{dz} = \rho_l \kappa_l \frac{dP}{dz} - \rho_l \beta_l \frac{dT}{dz} + \frac{\rho_l}{\dot{m}_T} \sum_{j}^{C} \left[M_{w_j} - \rho_l \overline{V}_{l_j} \right] \frac{d\dot{n}_j}{dz}$$

Mixture Volume

$$\rho_m = \alpha \rho_v + (1-\alpha) \rho_l$$

ioMosaic

Detailed SuperChems flow dynamics for reacting two-phase flow in pipes – momentum, energy, and phase equilibrium

Momentum

$$\frac{dP}{dz} + \frac{1}{A}\frac{d}{dz}\left[\dot{M}_T u_v + \dot{m}_T u_l\right] = -\left[\frac{dP}{dz}\right]_F - g\rho_m \sin\theta$$

Energy

$$\frac{d}{dz}\left[\dot{H}_v + \dot{H}_l + \frac{\dot{m}_T u_l^2}{2} + \frac{\dot{M}_T u_v^2}{2}\right] + (\dot{m}_T + \dot{M}_T)g\sin\theta = \frac{d\dot{Q}}{dz} - \frac{d\dot{W}}{dz}$$

Equilibrium

$$\left[\frac{\Phi_i^v \dot{N}_i}{\dot{N}_T}\frac{\partial \ln \Phi_i^v}{\partial T} - \frac{\Phi_i^l \dot{n}_i}{\dot{n}_T}\frac{\partial \ln \Phi_i^l}{\partial T}\right]\frac{dT}{dz} + \left[\frac{\Phi_i^v \dot{N}_i}{\dot{N}_T}\frac{\partial \ln \Phi_i^v}{\partial P} - \frac{\Phi_i^l \dot{n}_i}{\dot{n}_T}\frac{\partial \ln \Phi_i^l}{\partial P}\right]\frac{dP}{dz} +$$

$$\sum_j^C \left[-\frac{\Phi_i^v \dot{N}_i}{\dot{N}_T^2} + \frac{\Phi_i^v \dot{N}_i}{\dot{N}_T}\frac{\partial \ln \Phi_i^v}{\partial \dot{N}_j}\right]\frac{d\dot{N}_j}{dz} + \frac{\Phi_i^v}{\dot{N}_T}\frac{d\dot{N}_i}{dz} + \sum_j^C \left[\frac{\Phi_i^l \dot{n}_i}{\dot{n}_T^2} - \frac{\Phi_i^l \dot{n}_i}{\dot{n}_T}\frac{\partial \ln \Phi_i^l}{\partial \dot{n}_j}\right]\frac{d\dot{n}_j}{dz} -$$

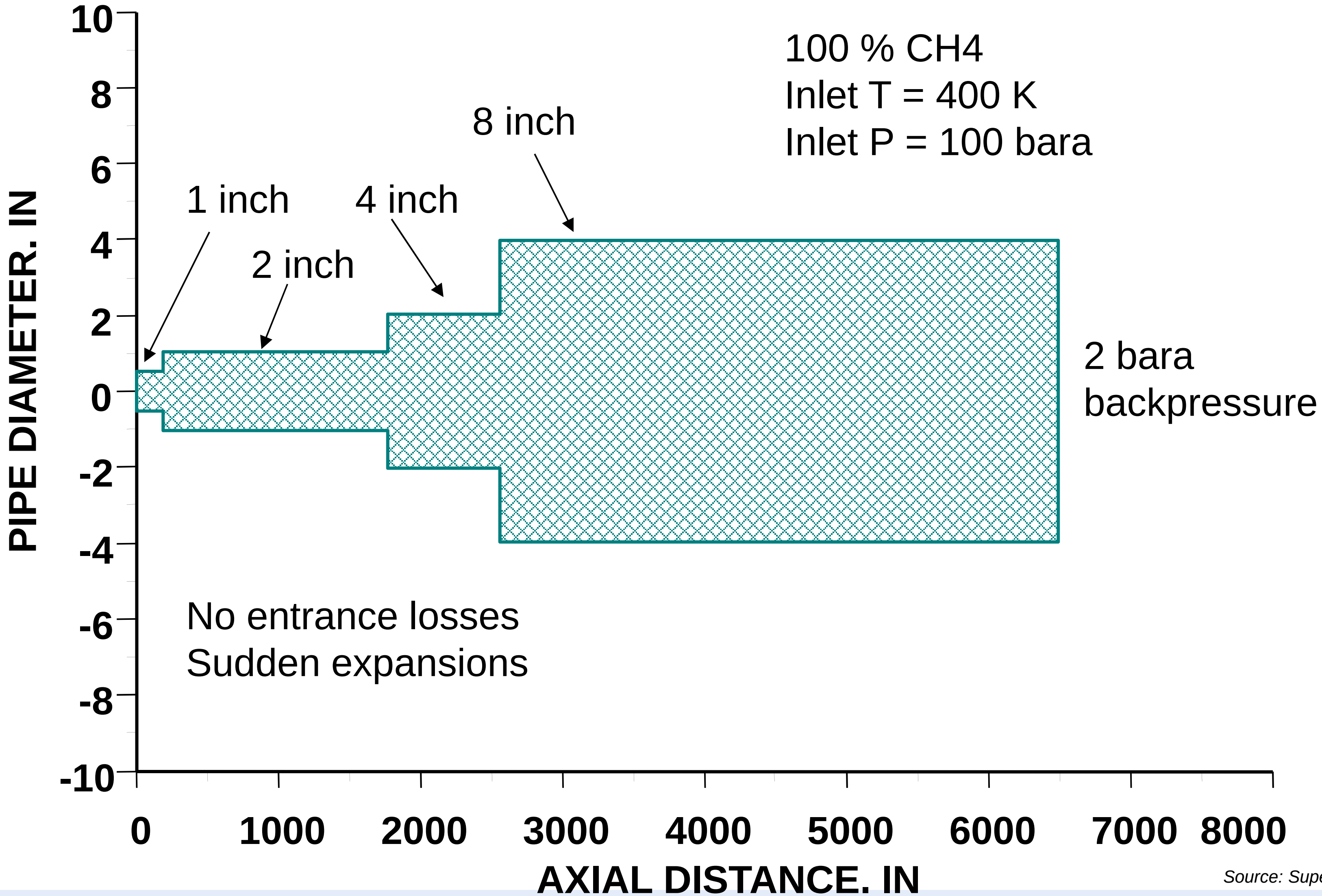
$$\frac{\Phi_i^l}{\dot{n}_T}\frac{d\dot{n}_i}{dz} = 0 \quad (i = 1, \cdots, C)$$

ioMosaic

How do we find multiple chokes?

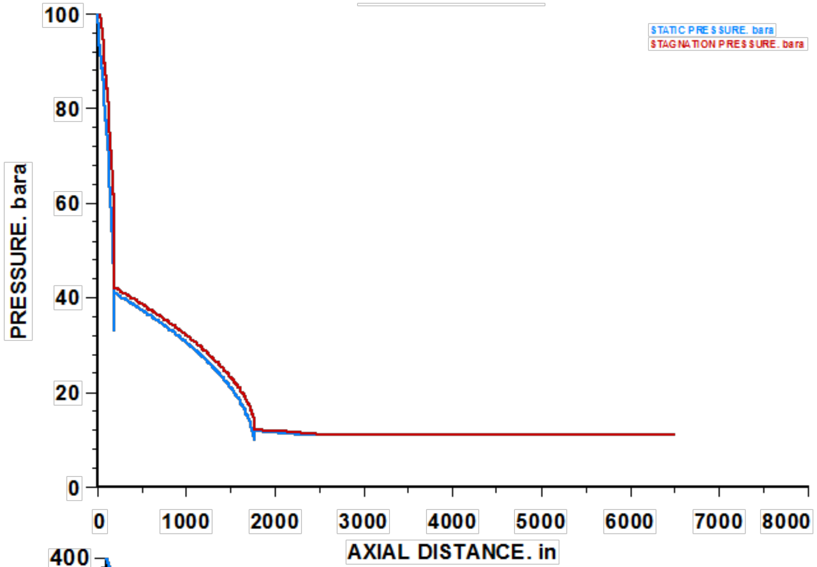
1. Specify number of piping segments (N) and set starting piping segment location to 0
2. Specify composition, starting temperature, and pressure
3. Guess mass flow rate
4. Solve pipe flow differential balances with respect to pressure
5. If normalized travel distance (z) is greater or equal to actual pipe length, increase mass flow and go back to step 4
6. If the choke point is at segment N, stop.
7. Set starting pipe index to choke point location
8. Guess an irreversible energy loss [reduce pressure and resolve temperature], and go back to step 4
9. If exit pressure > back pressure and traveled distance (z) is greater or equal to remaining pipe length, increase energy loss and go back to step 4
10. Go to step 6

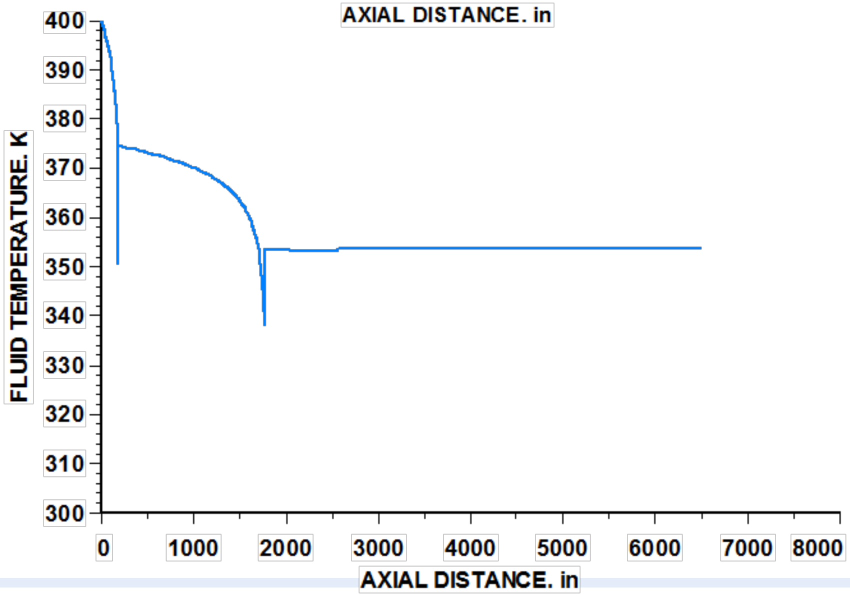
How do we find multiple chokes – All vapor flow example

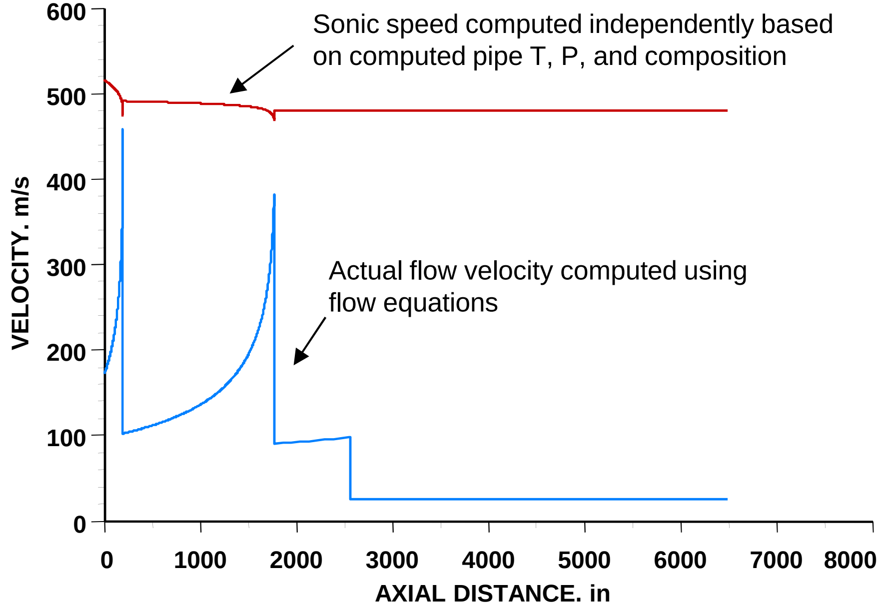


Source: SuperChems Expert

First, locate the first choke point solution





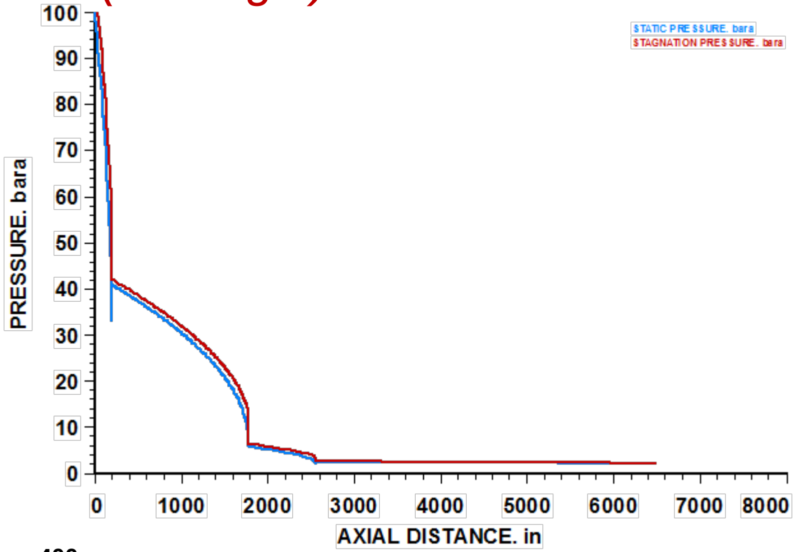


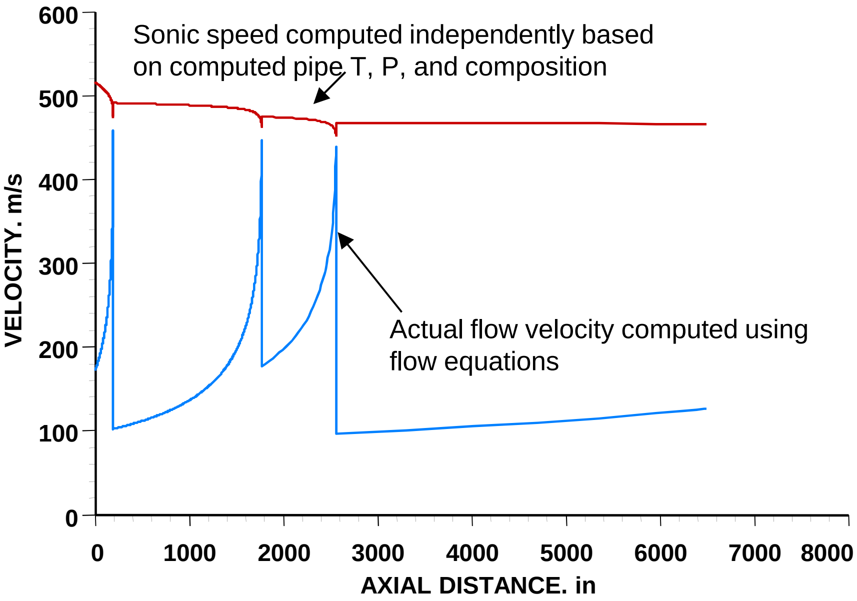
Flow chokes at the exit of the 1 inch line with a critical flow rate of 4.82 kg/s

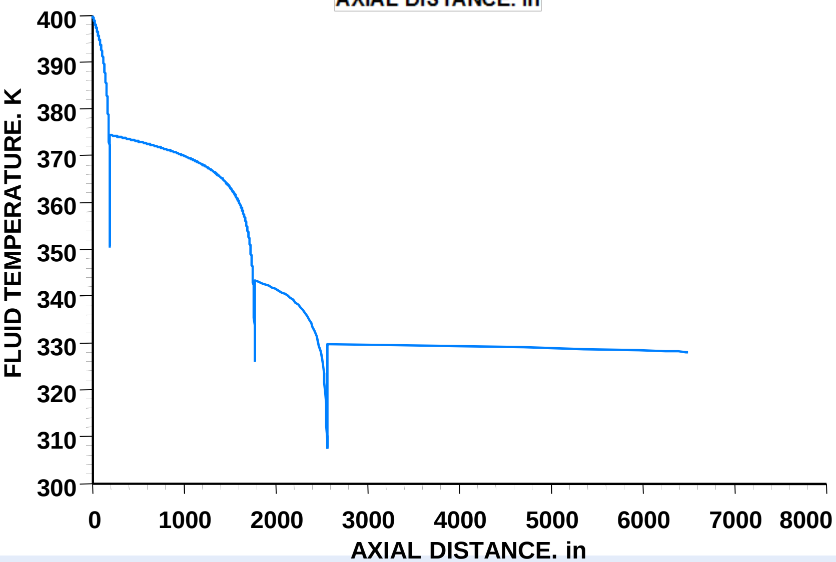
Source: SuperChems Expert

ioMosaic

Then locate the remaining choke points at the flow rate regulated by the first choke point (4.82 kg/s)





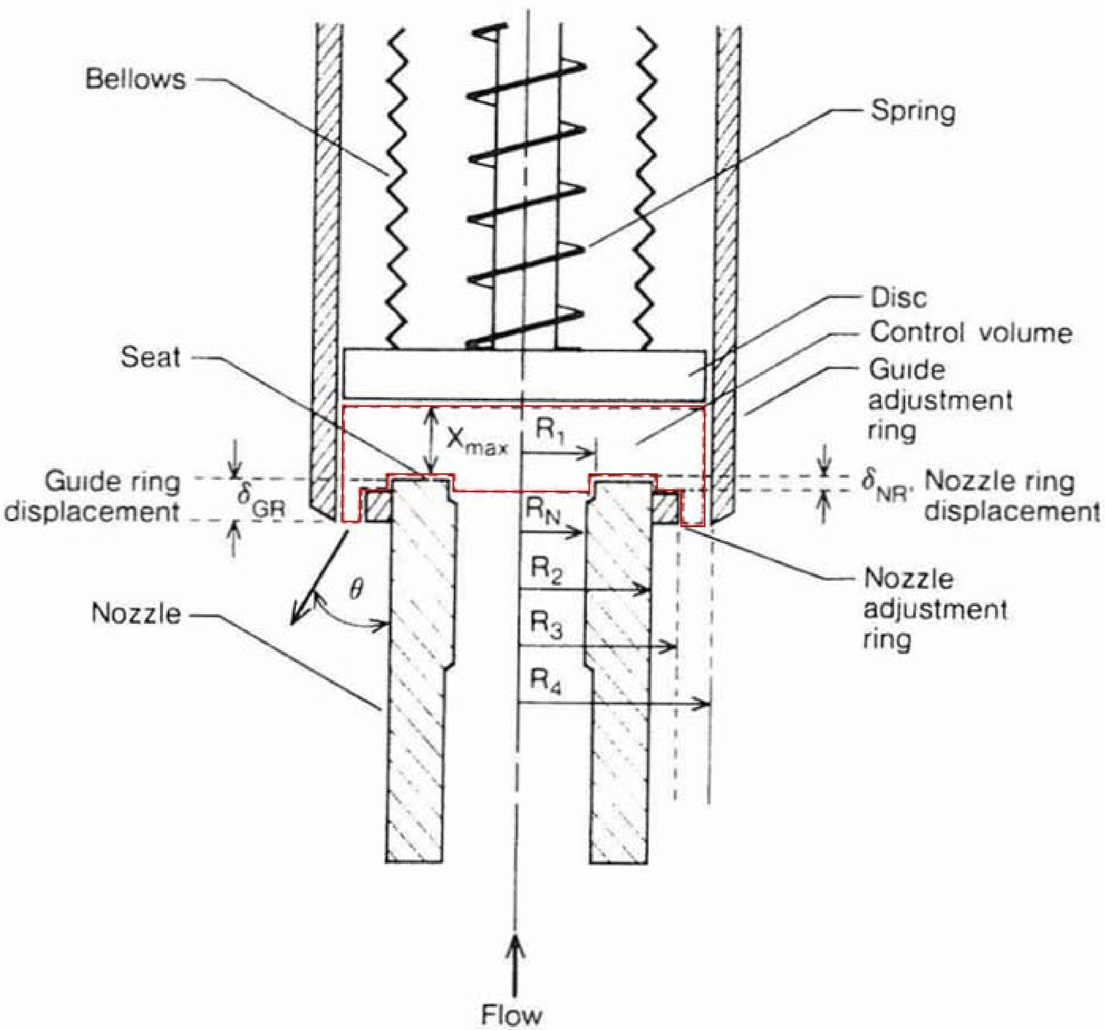


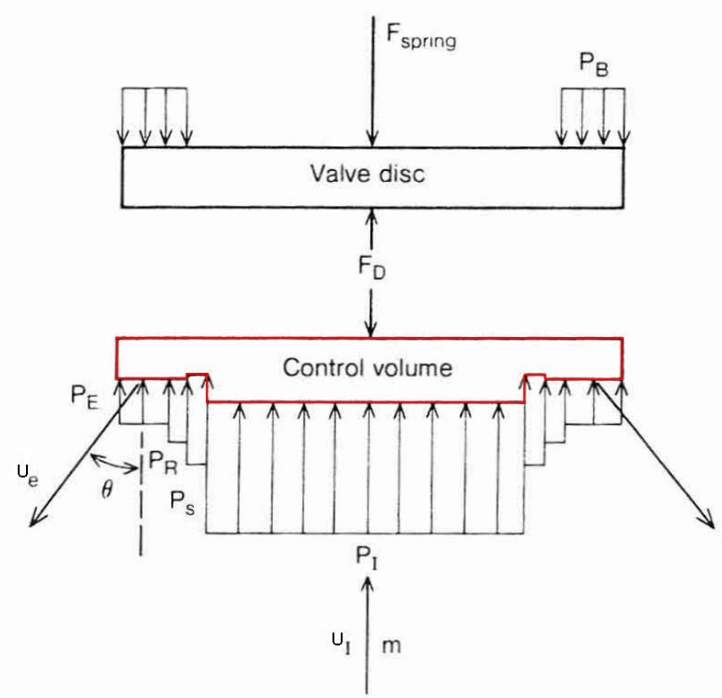
Multiple chokes found in 2 inch and 4 inch segments

Source: SuperChems Expert

ioMosaic

The SuperChems PRV dynamic model is adapted from Melhem/Fisher [2003] and Singh and Shak [1983]



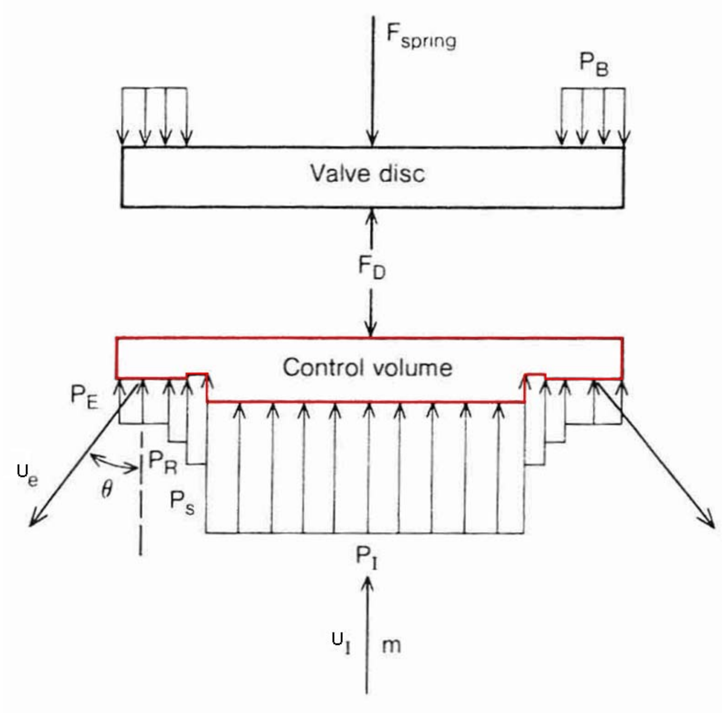


The movement of the PRV disk depends on a force balance between the upward fluid force and downward spring and backpressure forces

$$
\begin{aligned}
F_{Dn} &= P_B A_B + K_s\left(x_o + x\right) + P_{atm} A_{bel} + m_D g \\
&= P_B\left(A_D - A_{bel}\right) + K_s\left(x_o + x\right) + P_{atm} A_{bel} + m_D g \\
&= P_B A_D + K_s\left(x_o + x\right) + \left(P_{atm} - P_B\right) A_{bel} + m_D g \\
A_D &= \pi R_D^2 \simeq \pi R_4^2
\end{aligned}
$$

$$
\begin{aligned}
F_{Up} &= P_I A_I + P_S A_S + P_R A_R + P_E A_E + \dot{m}\, u_e \cos\theta + \frac{\dot{m}^2}{\rho_I A_I} \\
\theta &= 90 + \left(\theta_o - 90\right)\sqrt{\frac{x}{x_{max}}}
\end{aligned}
$$

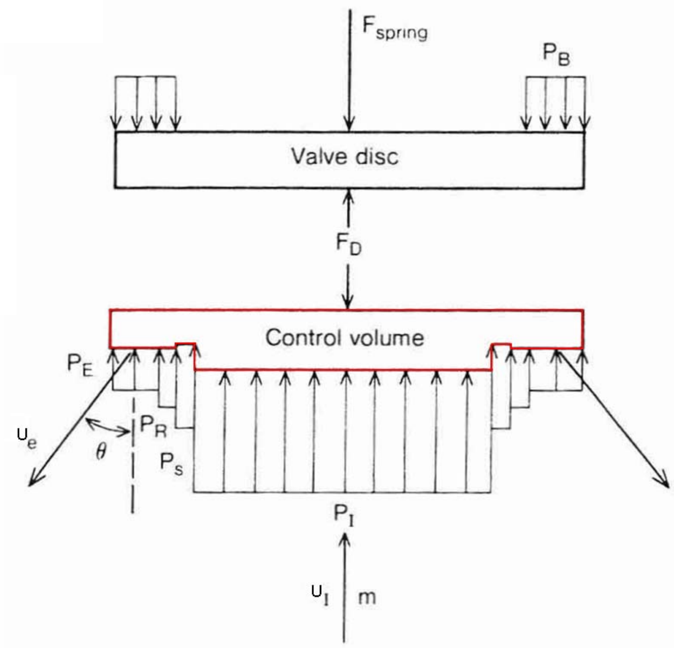
$$
K_s x_o + m_D g = \left(P_{set} - P_{atm}\right) A_I
$$



The net force on the valve disk can be calculated as the difference between the upward and downward forces

$$
\begin{aligned}
F_{NET} &= F_{Up} - F_{Dn} \\
&= (P_I - P_{set} + P_{atm})\, A_I + P_S A_S + P_R A_R + P_E A_E + \dot{m}\, u_e \cos\theta + \frac{\dot{m}^2}{\rho_I A_I} \\
&\quad - P_B A_D - K_s x - (P_{atm} - P_B)\, A_{bel}
\end{aligned}
$$

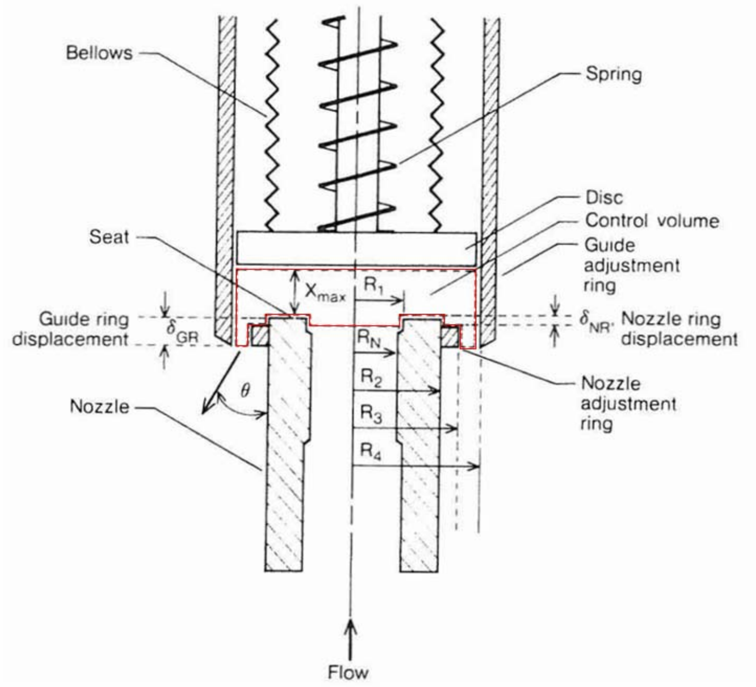
$$
\begin{aligned}
F_{NET} &= F_{Up} - F_{Dn} \\
&= (P_I - P_{set} + P_{atm})\, A_I + \eta P_* (A_D - A_I) + \dot{m}^2 \left[\frac{\cos\theta}{2\pi R_1 x C_d \rho_e} + \frac{1}{\rho_I A_I}\right] \\
&\quad - P_B A_D - K_s x - (P_{atm} - P_B)\, A_{bel}
\end{aligned}
$$



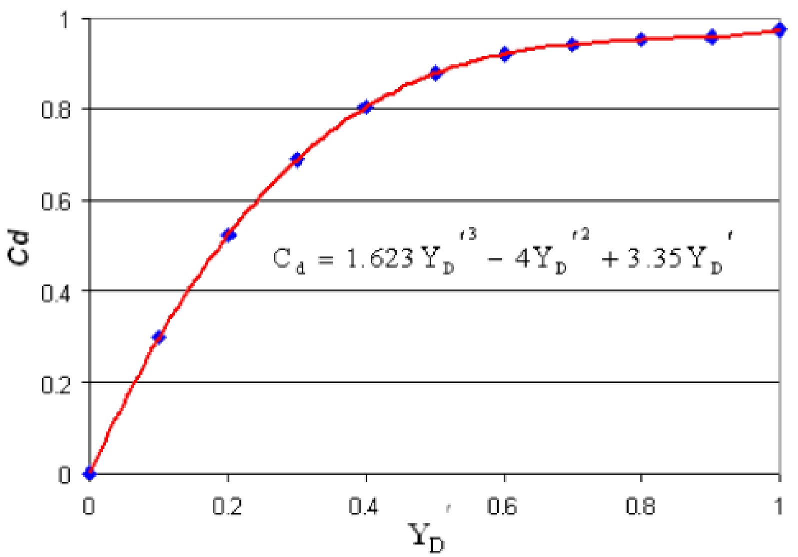
![ioMosaic]

As the valve seat starts to lift, the curtain area will initially be smaller than the nozzle flow area and will regulate the flow

$$x_c = \frac{A_N}{2\pi R_N} = \frac{\pi R_N^2}{2\pi R_N} = \frac{R_N}{2}$$



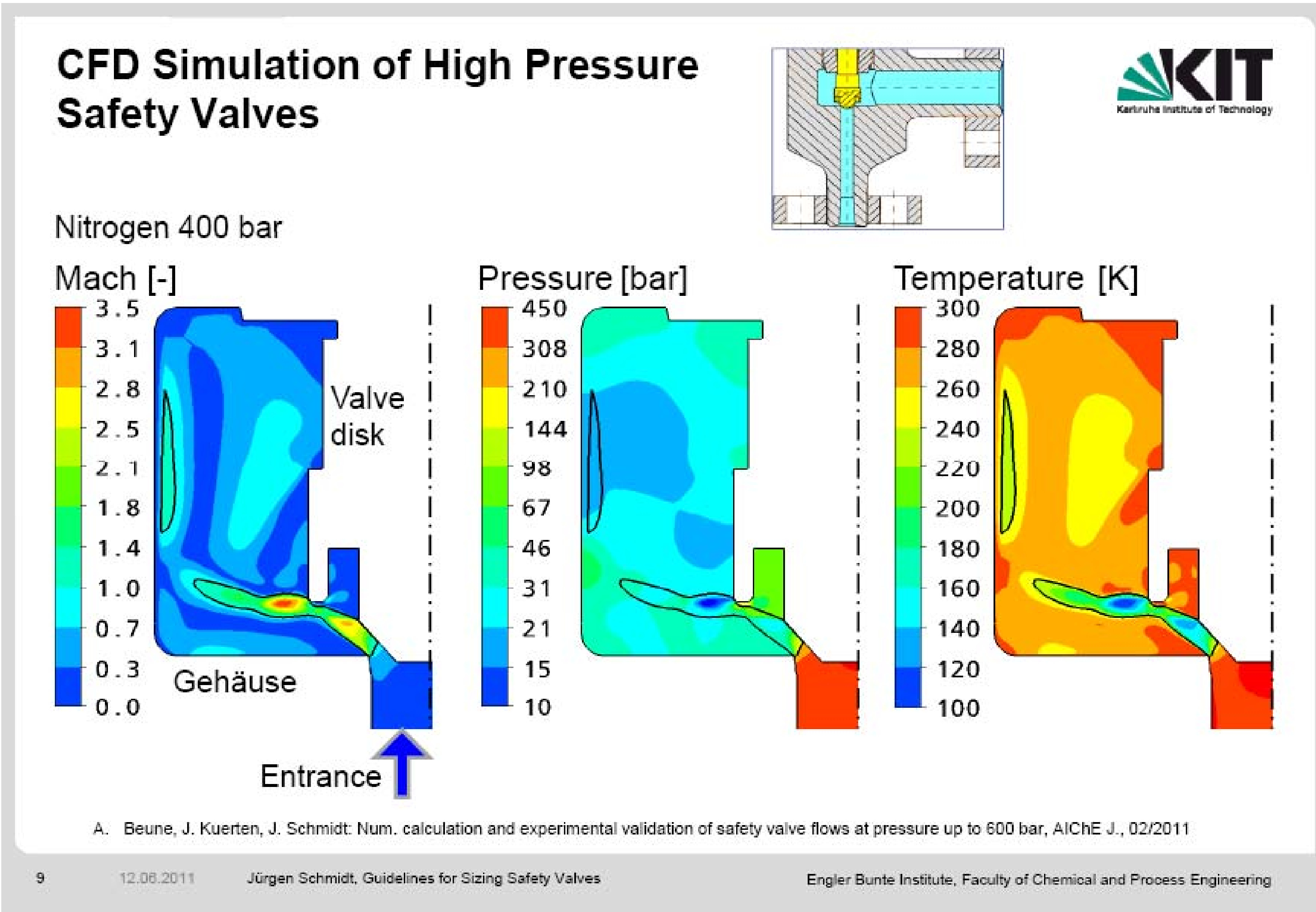
Also note that the discharge coefficient will change with disk lift



$$C_d = C_{d,max} \left[\frac{x}{x_{max}} \right]^{\psi}$$

ioMosaic

Recent CFD work in Europe shows a complex pressure profile!

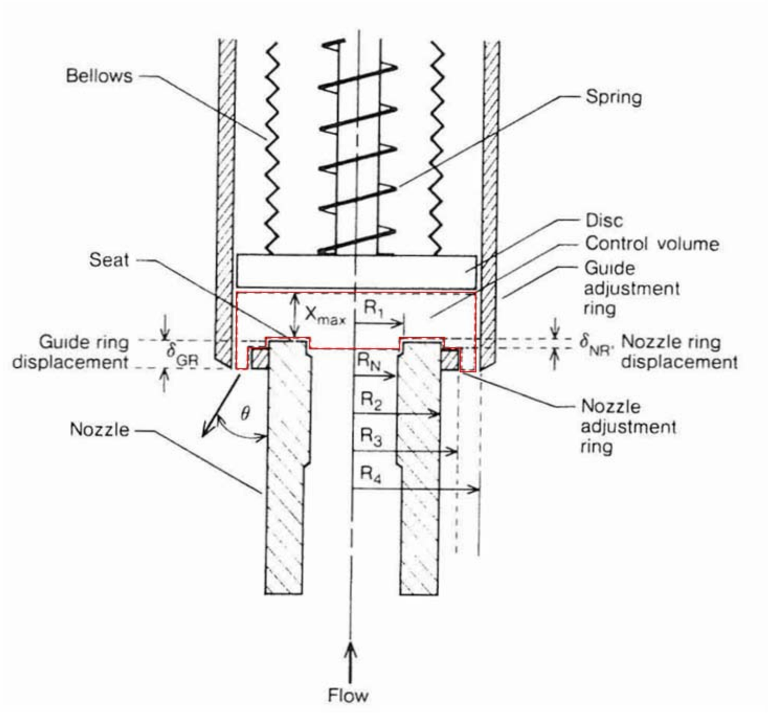


ioMosaic

Using a single degree of freedom analysis, we can describe the motion of the valve disk

$$
\begin{aligned}
F_{NET} &= F_{Up} - F_{Dn} \\
&= \left(P_I - P_{set} + P_{atm}\right) A_I + \eta P_* \left(A_D - A_I\right) + \dot{m}^2 \left[\frac{\cos\theta}{2\pi R_1 x C_d \rho_e} + \frac{1}{\rho_I A_I}\right] \\
&\quad - P_B A_D - K_s x - \left(P_{atm} - P_B\right) A_{bel}
\end{aligned}
$$

$$
\begin{aligned}
\frac{dx}{dt} &= u_d \\
m_D \frac{du_d}{dt} + C u_d &= F_{NET}
\end{aligned}
$$



Using a single degree of freedom analysis, we can describe the motion of the valve disk

$$C = \zeta C_{cr}$$

$$C_{cr} = 2m_D\omega_n = \frac{2K_s}{\omega_n} = 2\sqrt{K_s m_D}$$

$$\omega_n = \sqrt{\frac{K_s}{m_D}}$$

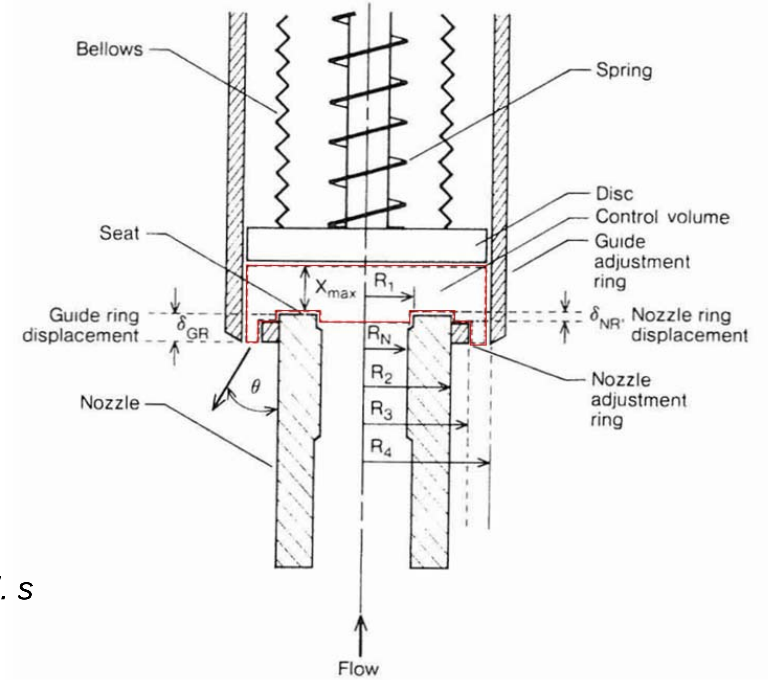
Undamped circular natural frequency. Radians/s

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_s}}$$

Undamped natural period. s

$$f_n = \frac{1}{\tau_n} = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{K_s}{m_D}}$$

Undamped natural frequency. s

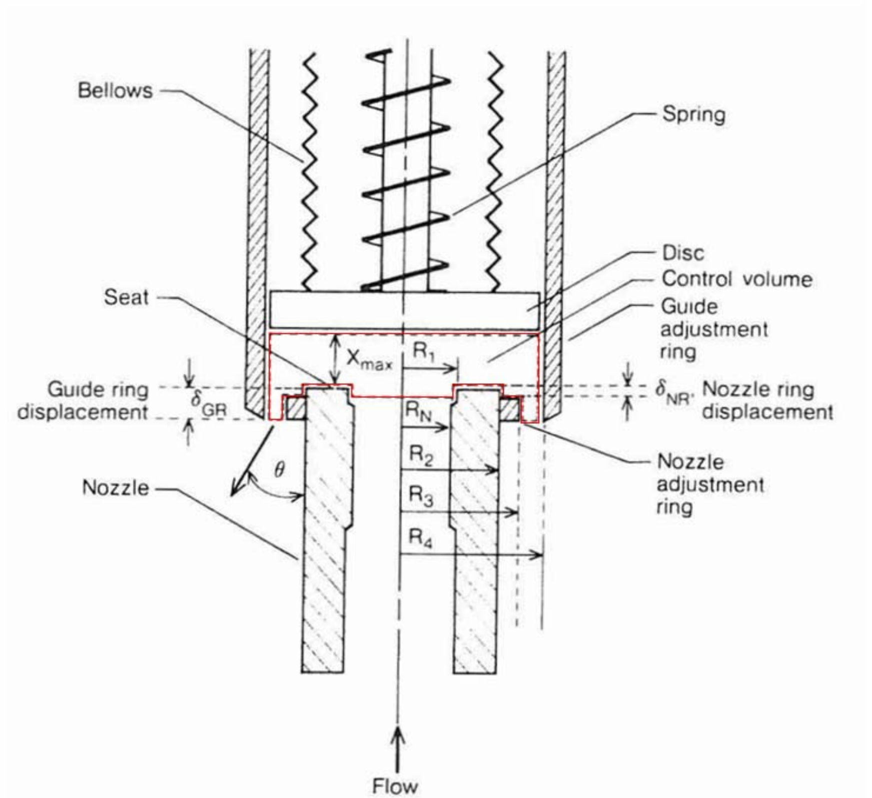


When the valve disk is on the seat or the upper stop, a coefficient of restitution is used to reverse the spindle direction

$$\frac{dx}{dt} = u_d$$

$$\frac{du_d}{dt} = \frac{1}{m_D}\left[F_{NET} - \zeta C_{cr} u_d\right]$$

$$\frac{dx'}{dt} = -\beta \frac{dx}{dt}$$



ioMosaic

The dynamics of the PRV alone are not sufficient to describe practical applications. We must couple the vessel dynamics, and associated piping with the solution

1. Set t = 0. $u_d = 0$ and $x = 0$. Set P_B to ambient pressure or user superimposed back pressure. Set P_I and ρ_I to vessel conditions P_o and ρ_o. Set $\dot{m}$ to 0.
2. Assume that P_B, ρ_I, and $\dot{m}$ persist for one time step, dt
3. Integrate Equations 50.11, 50.16, 50.17, 50.19, 50.20, 50.28 and 50.29 for one time step dt.
4. Set the PRV flow area to the smaller value of either A_N or $2\pi R_1 x$. Set the discharge coefficient for the PRV from Equations 50.19 and 50.20.
5. Solve the coupled vessel-piping dynamics using the existing detailed and complex SuperChems Models for one time step dt.
6. Set P_I and ρ_I to the values calculated at the inlet of the PRV. Set the value of P_B to the value calculate at the discharge of the PRV. Set $\dot{m}$ to the value calculated by the vessel dynamics.
7. Increment time by dt.
8. Go to Step 2.

ioMosaic

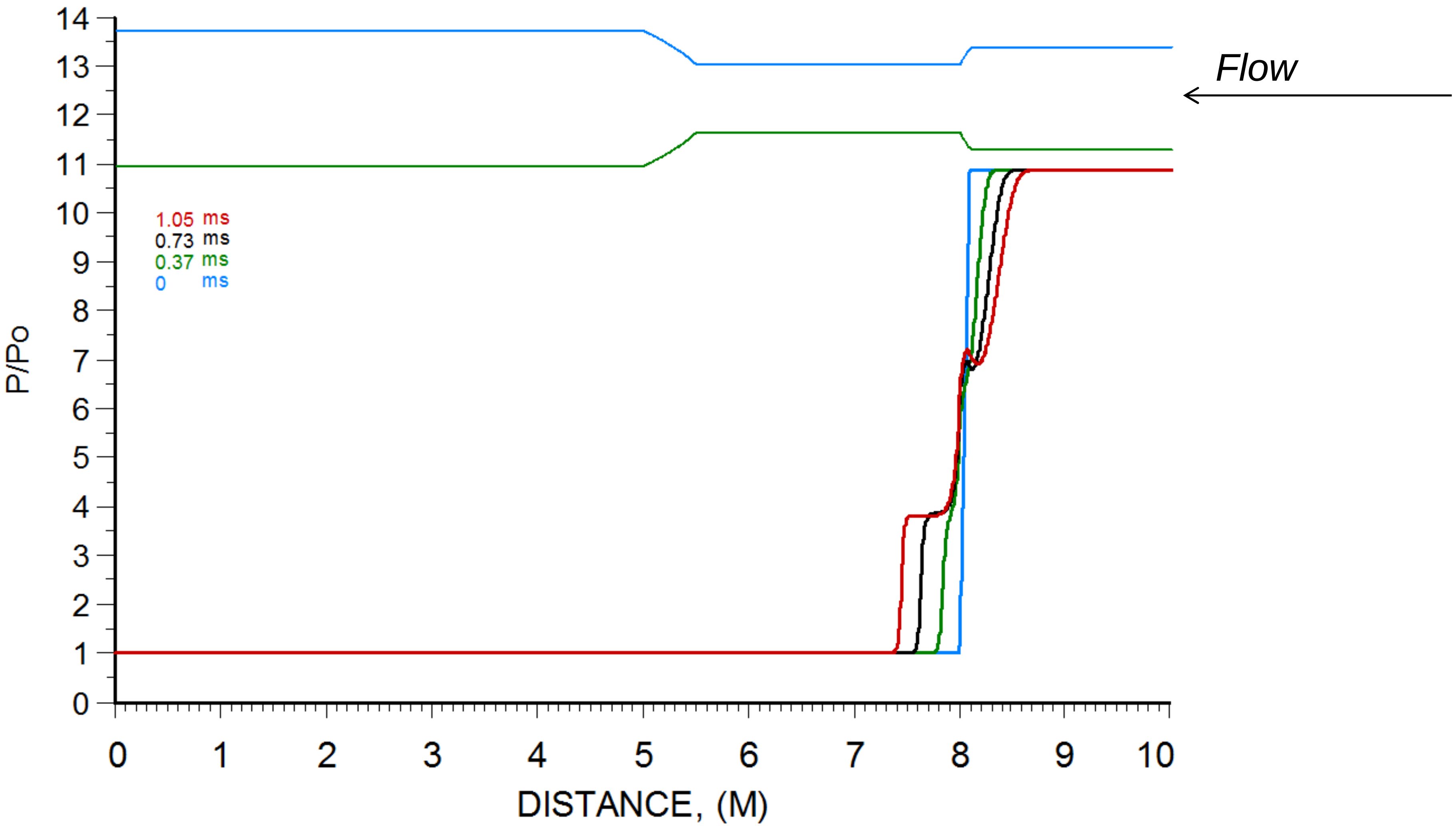
PRV stability is heavily influenced by the configuration of inlet and discharge piping. For simple configurations, the upstream pressure drop or rise can be estimated from the following equations

$$
\begin{aligned}
t_{wave} &= \frac{2L_p}{c_0} \\
\tau &= \min\left(\frac{t_{wave}}{t_{valve}}, 1\right) \\
\Delta P_{wave} &= \tau \frac{c_0 \dot{M}}{A_p} + \tau^2 \frac{\dot{M}^2}{2\rho_0 A_p^2} \quad 0 \le \tau \le 1 \\
\Delta P_{wave} &= \tau \rho u c_0 \left[1 + \frac{\tau \rho u}{2\rho_0 c_0}\right] \quad 0 \le \tau \le 1
\end{aligned}
$$

$$\Delta P_{f,wave} = \tau^2 \Delta P_f = \tau^2 \frac{\dot{M}^2 \left(K + \frac{4fL_p}{D_p}\right)}{2\rho_0 A_p^2} \quad 0 \le \tau \le 1$$

ioMosaic

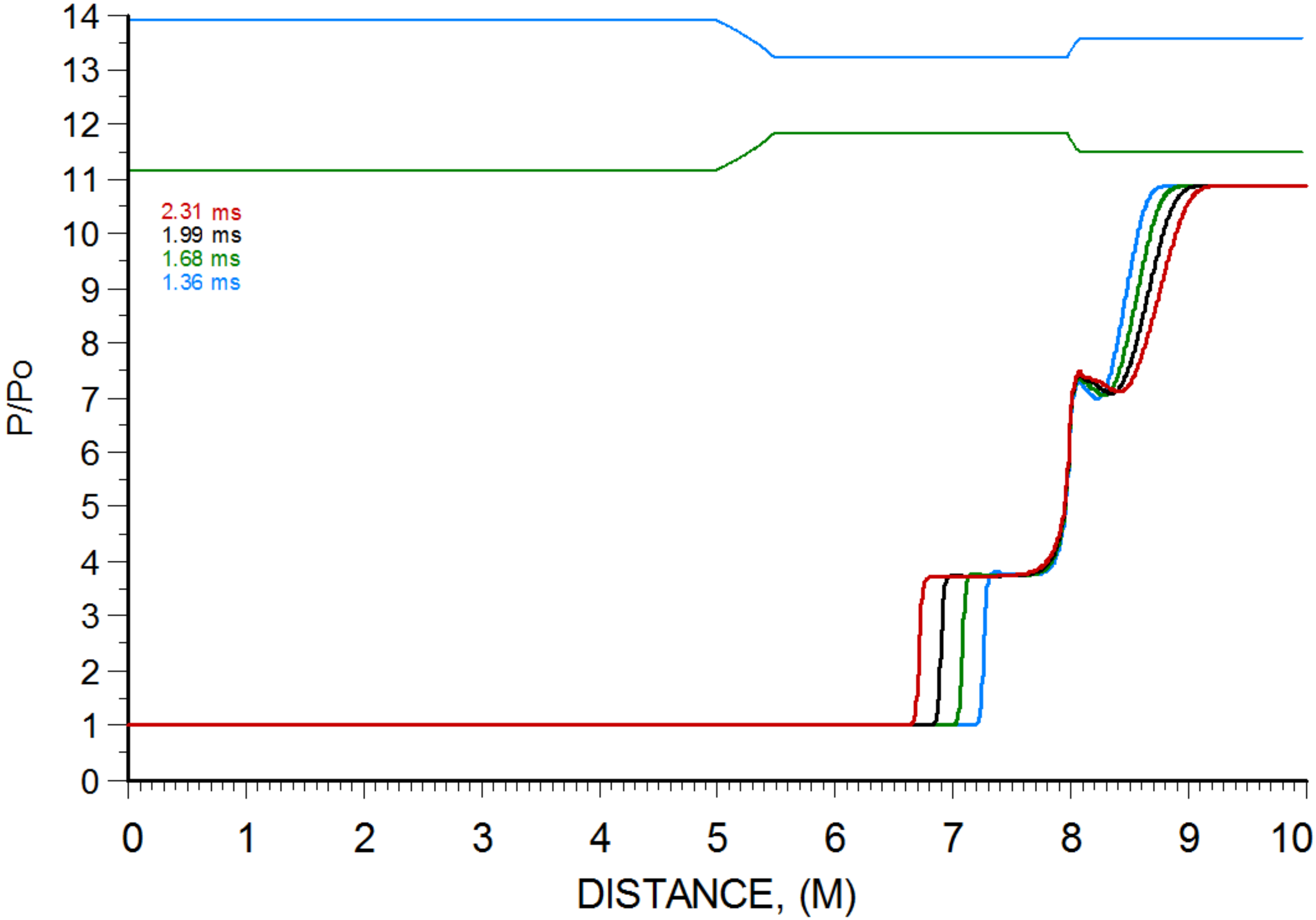
Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

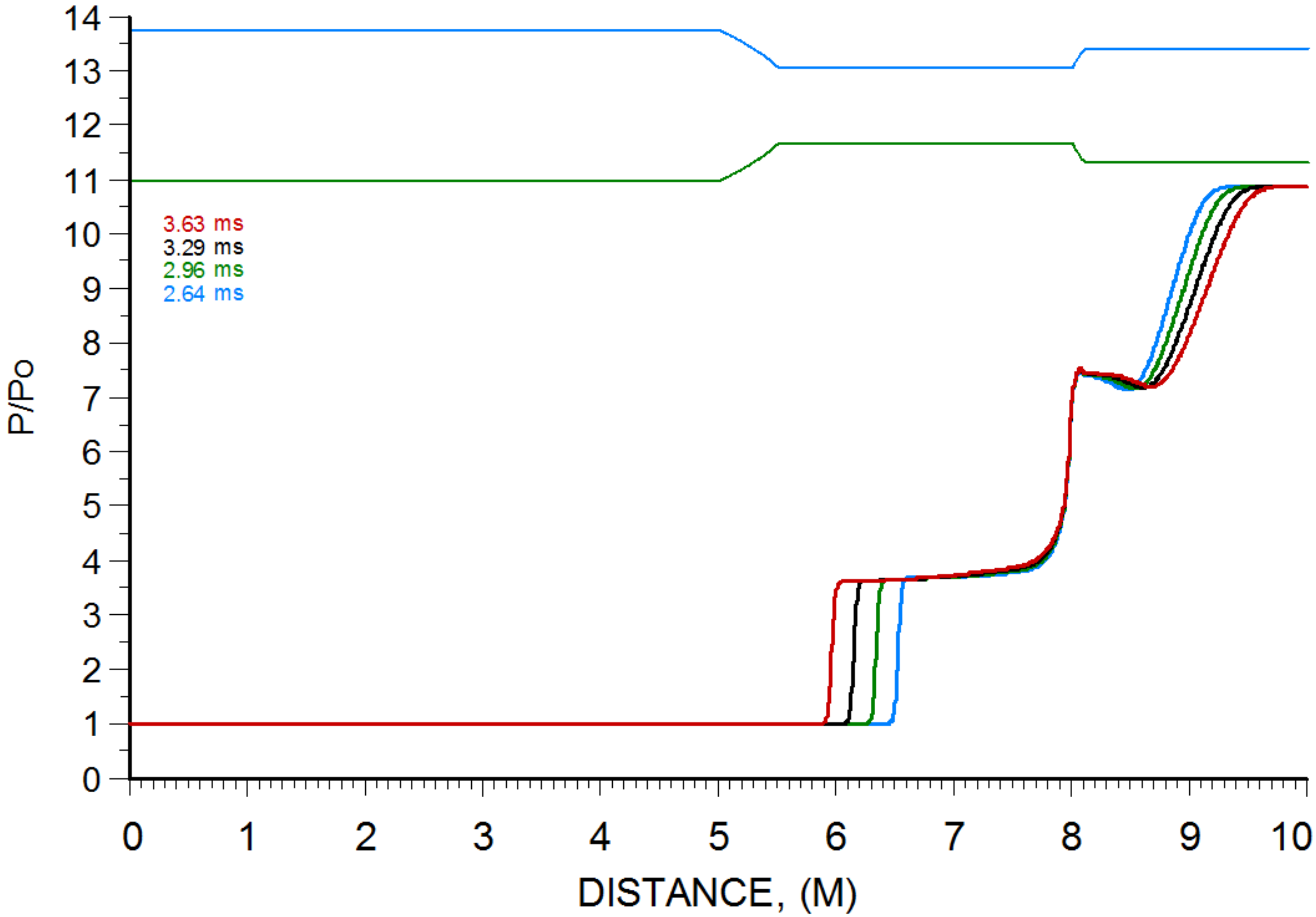
ioMosaic

Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

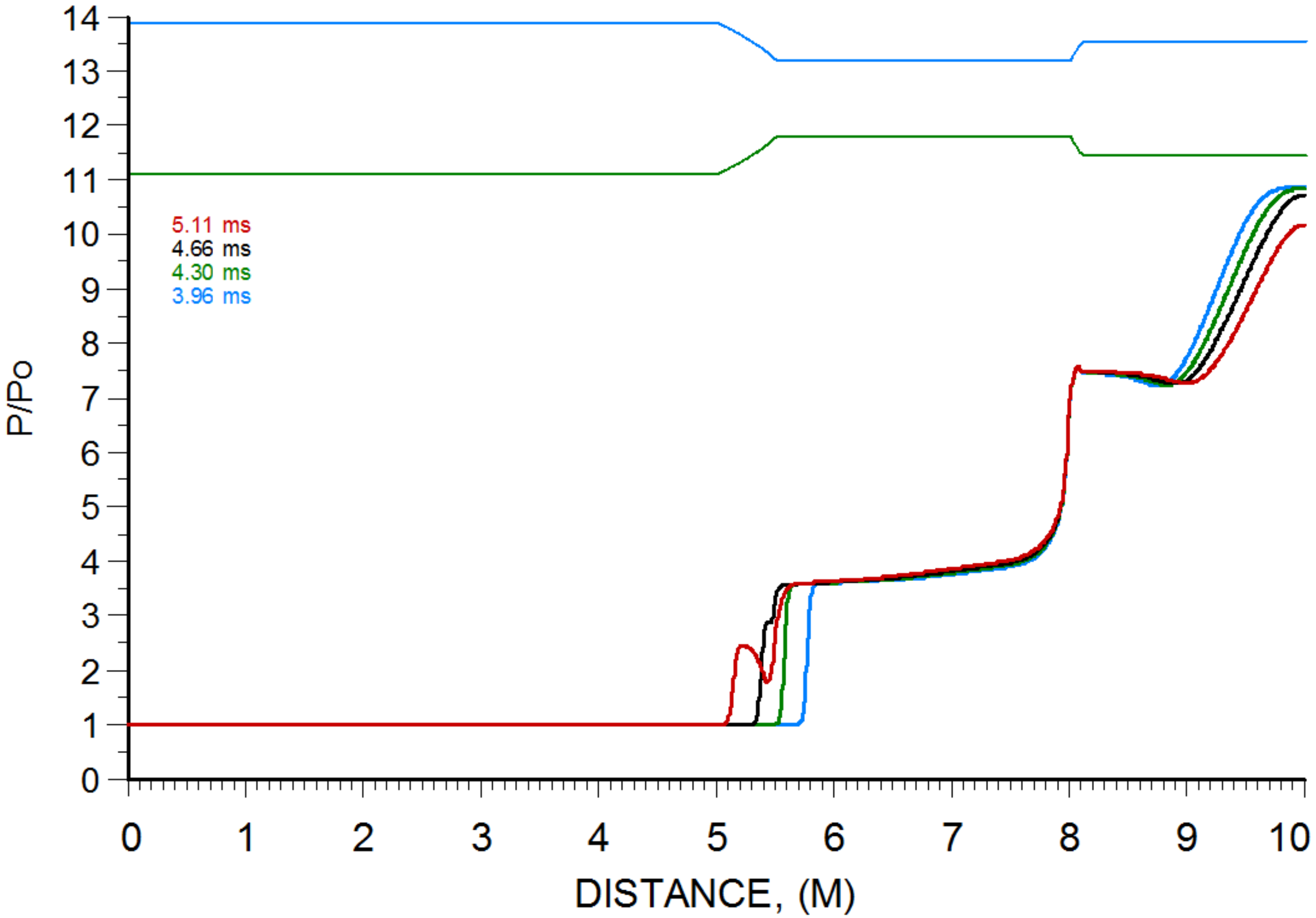
Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

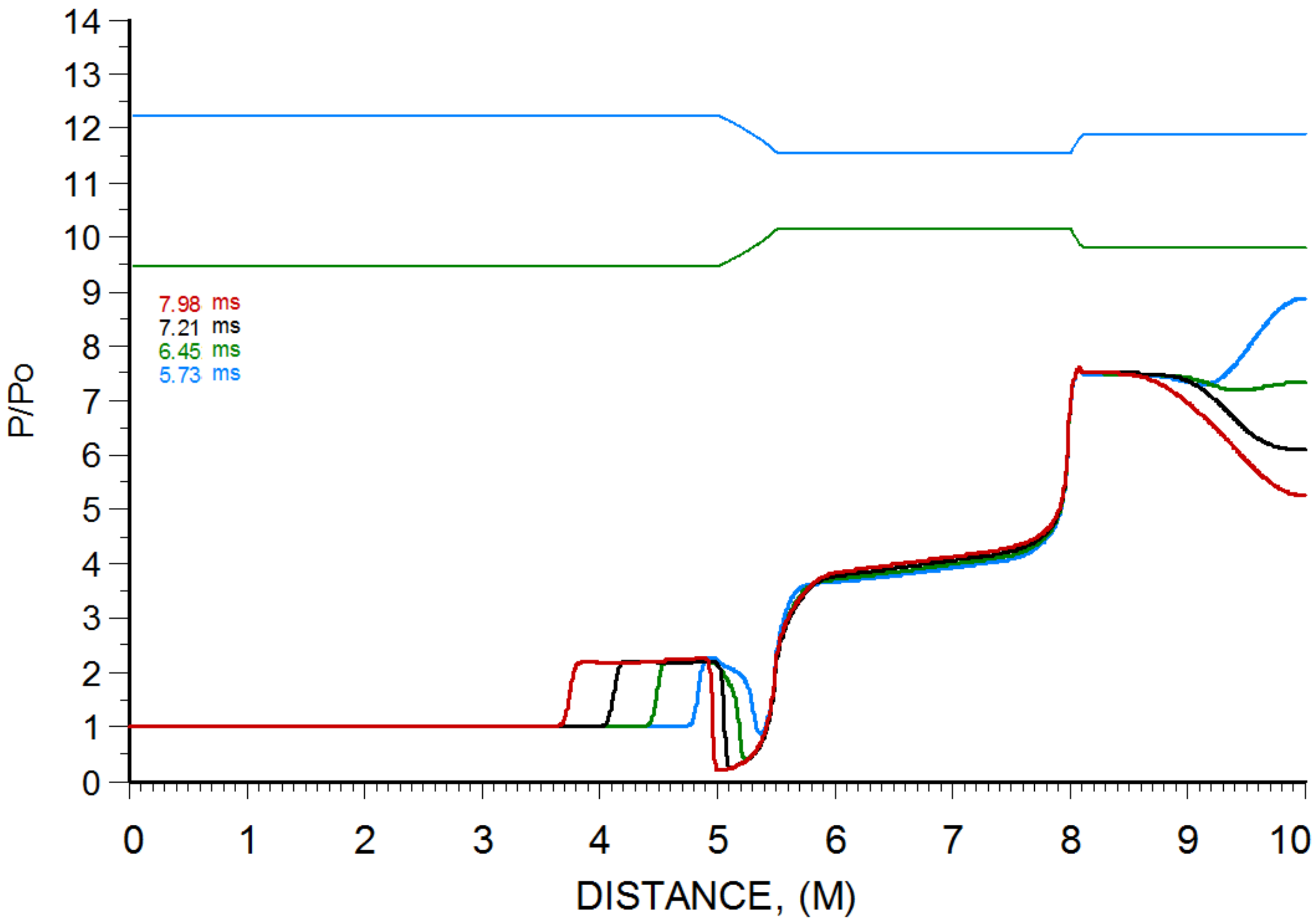
ioMosaic

Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

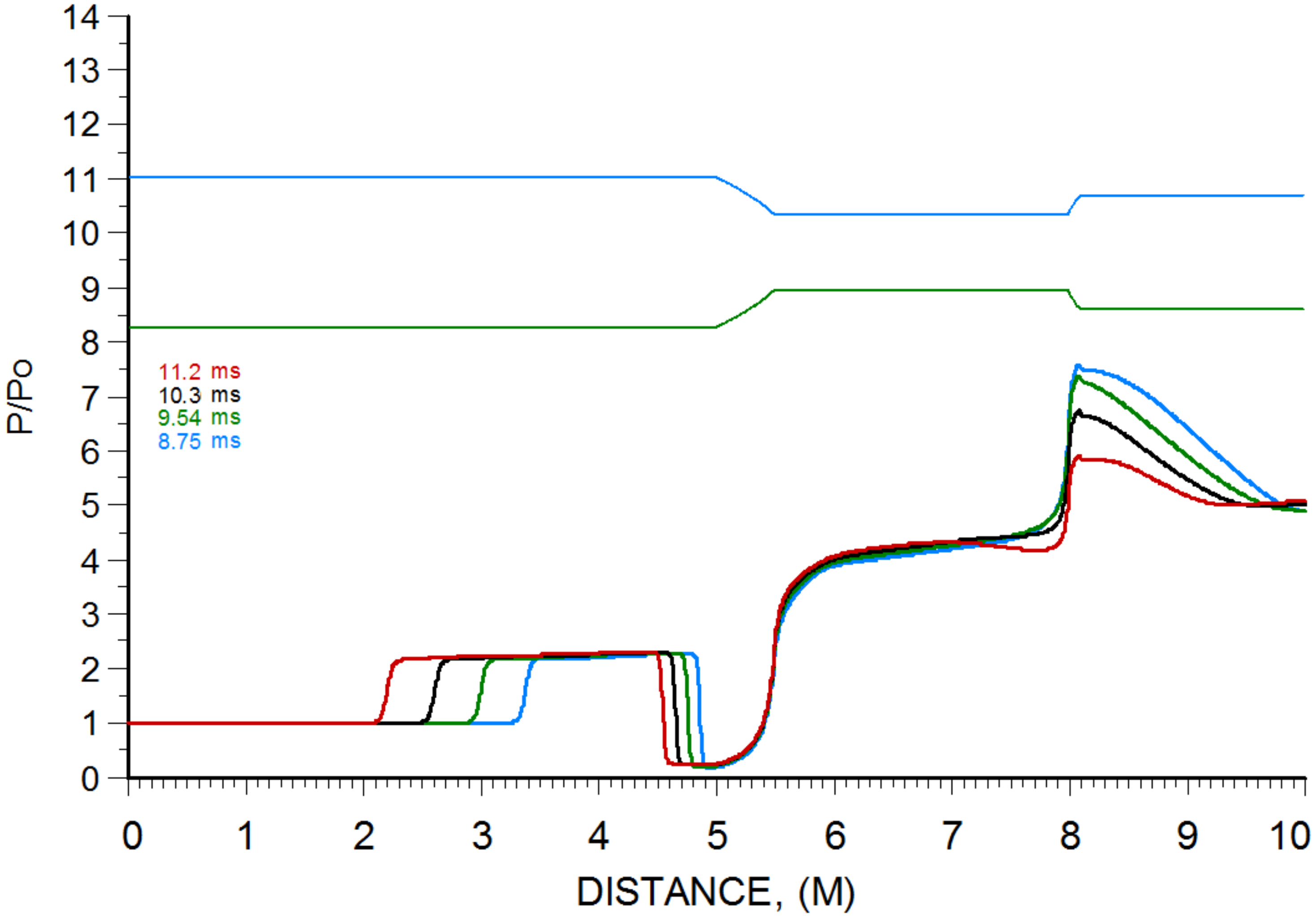
Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

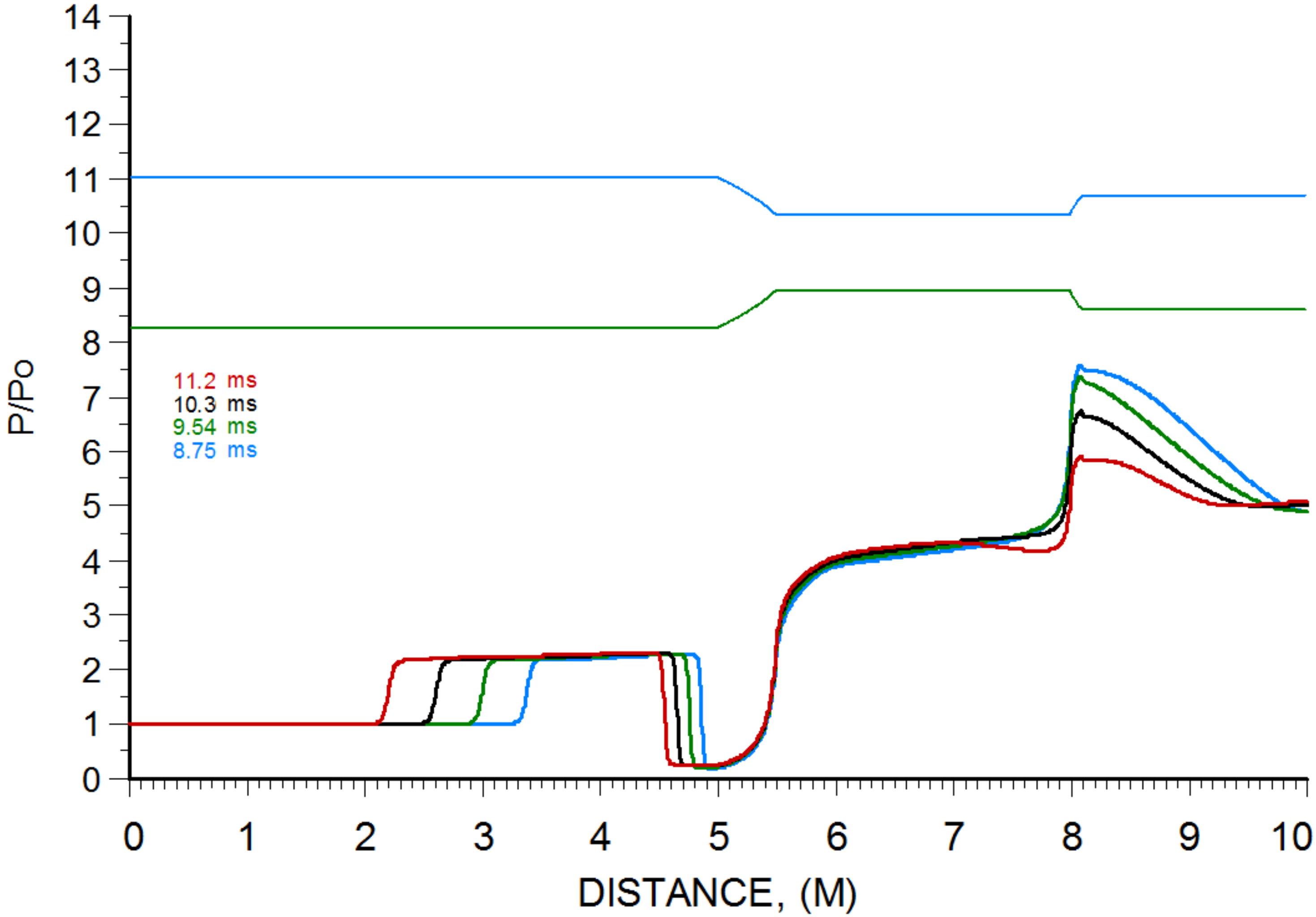
ioMosaic

Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

Complex piping configurations will require the solution of 1D and 2D fluid dynamics equations



Source: SuperChems Expert

ioMosaic

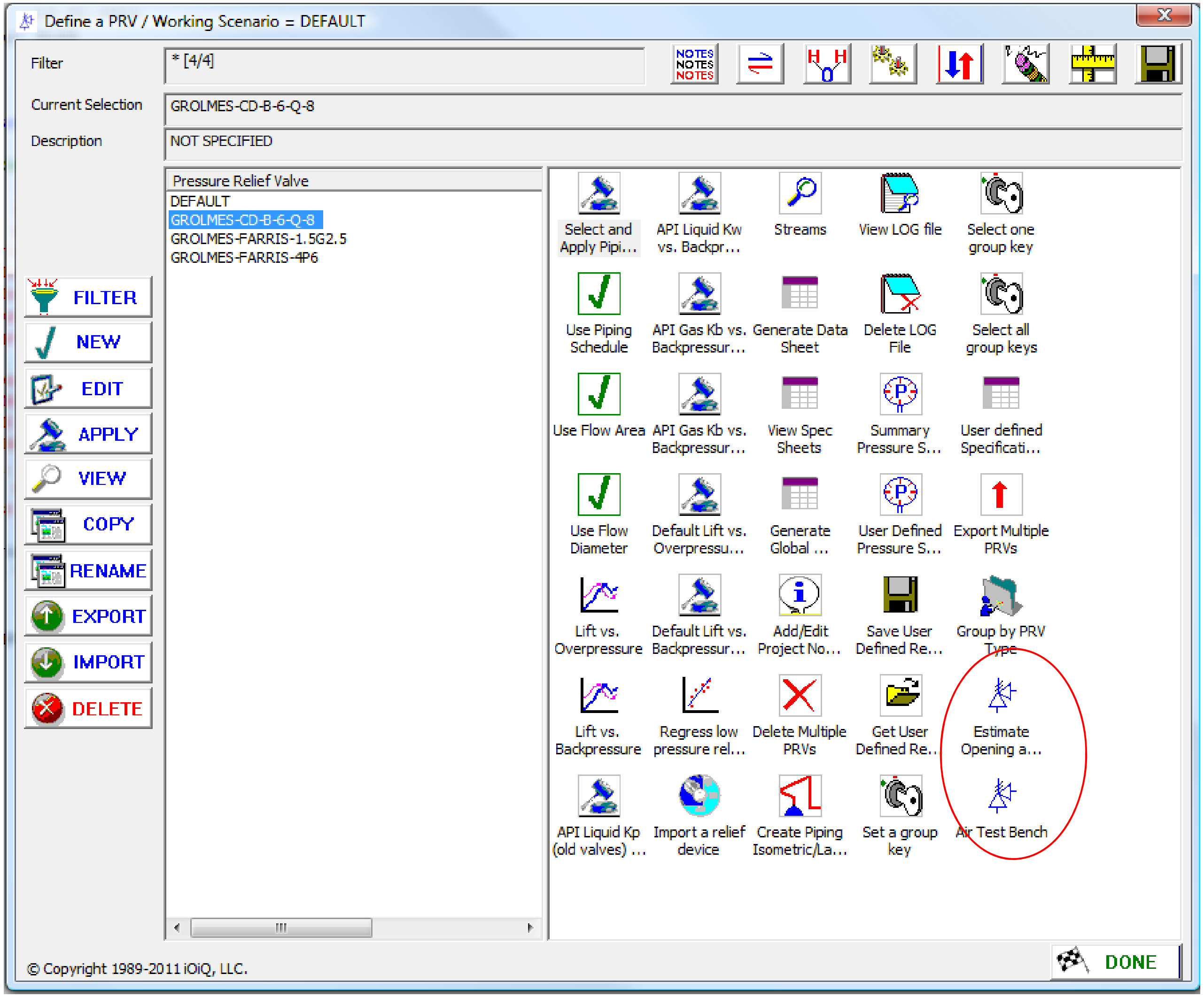
Acoustic barriers may also be established because of body bowl choking

- How do we establish where the acoustic barriers are without having to solve the complete fluid dynamics equations in 1 or 2D?

- Just because we cannot easily solve the 1D or 2D pipe flow dynamics for two-phase flow in complex piping, we can use the single phase solutions to identify the “effective acoustic” length of the piping to use simpler formulas

ioMosaic

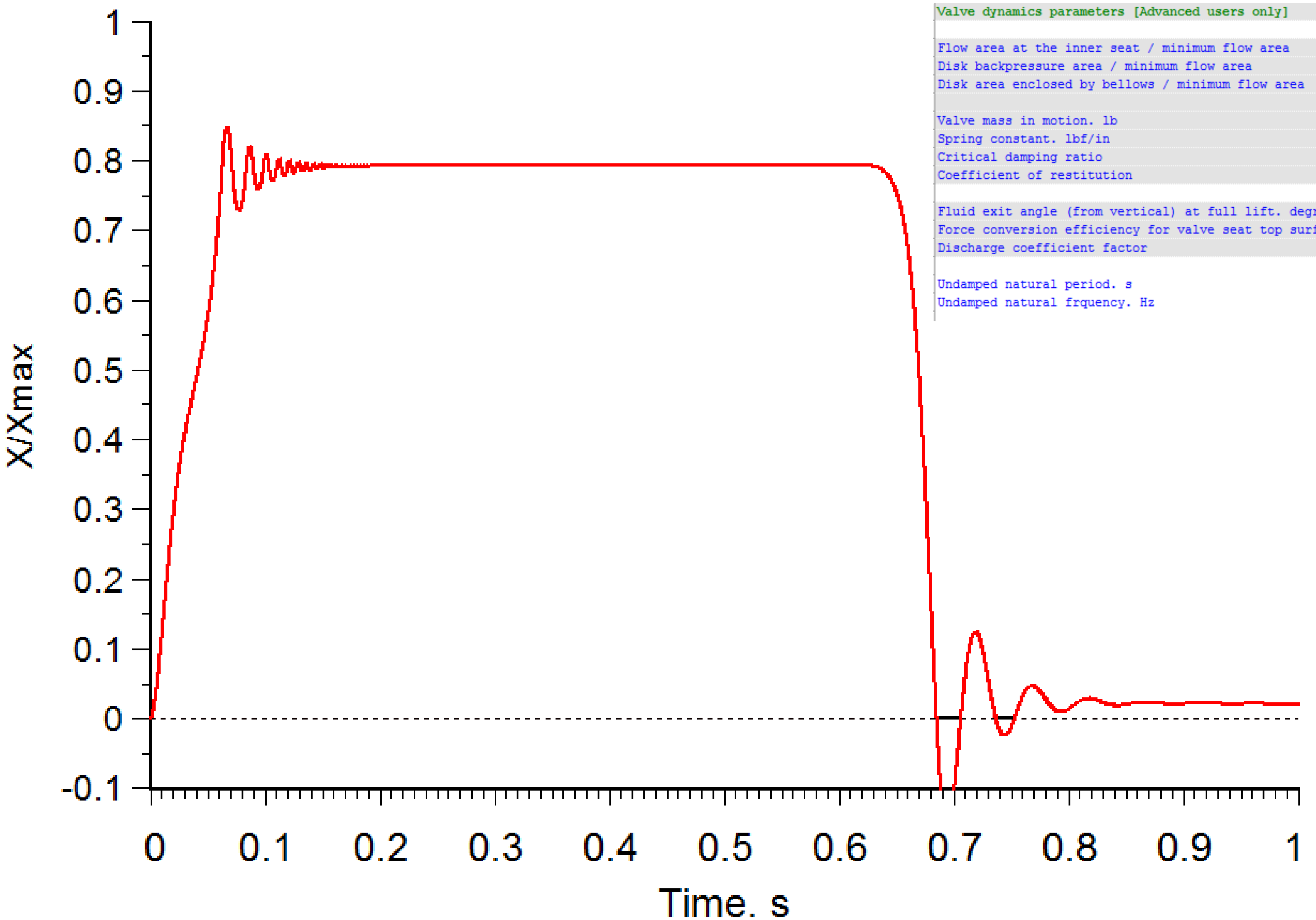
SuperChems version 6.4mp introduces PRV dynamics functionality to the PRV object



Source: SuperChems Expert

ioMosaic

Let's examine the dynamics of a 5 m^3 vessel full of air at 25 C and 55 psig using the Farris 4P6 PRV considered by Grolmes earlier

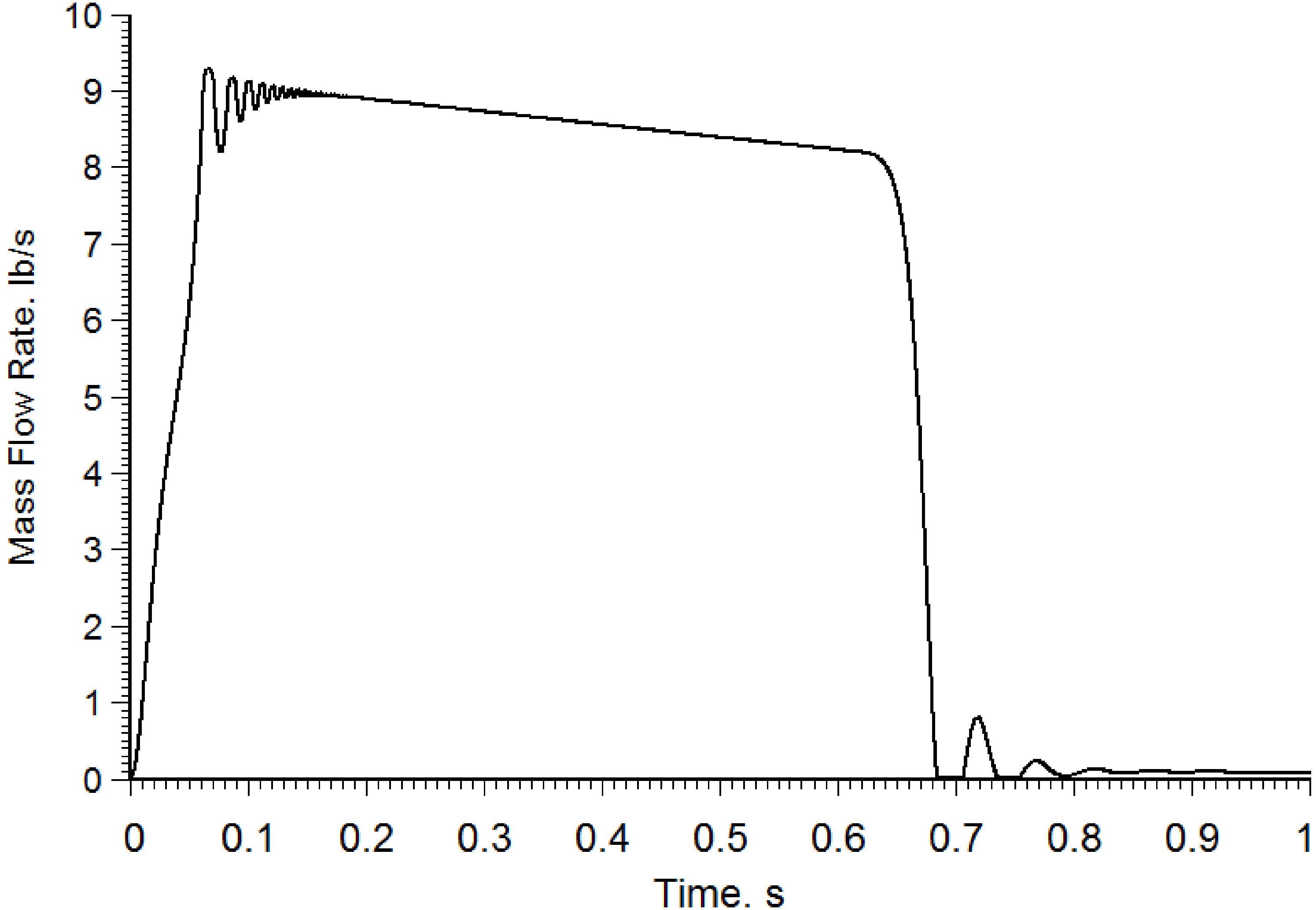


Valve dynamics parameters [Advanced users only]	
Flow area at the inner seat / minimum flow area	1.000
Disk backpressure area / minimum flow area	1.300
Disk area enclosed by bellows / minimum flow area	0.000
Valve mass in motion. lb	11.54
Spring constant. lbf/in	571.96
Critical damping ratio	0.200
Coefficient of restitution	0.010
Fluid exit angle (from vertical) at full lift. degrees	10.00
Force conversion efficiency for valve seat top surface pressure	0.60
Discharge coefficient factor	0.25
Undamped natural period. s	0.0454
Undamped natural frquency. Hz	22.0173

Source: SuperChems Expert

ioMosaic

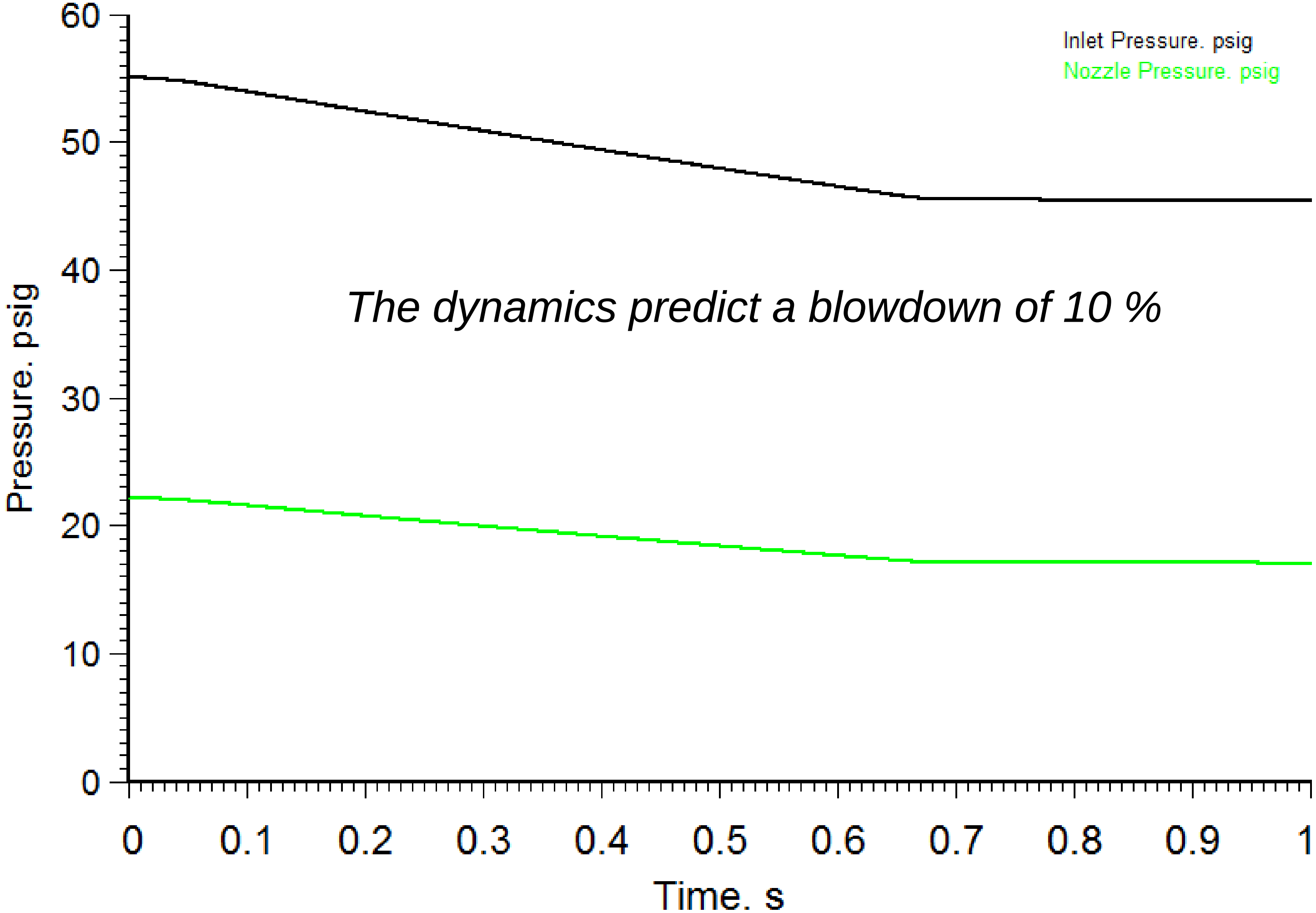
Let's examine the dynamics of a 5 m^3 vessel full of air at 25 C and 55 psig using the Farris 4P6 PRV considered by Grolmes earlier



Source: SuperChems Expert

ioMosaic

Let's examine the dynamics of a 5 m³ vessel full of air at 25 C and 55 psig using the Farris 4P6 PRV considered by Grolmes earlier



The dynamics predict a blowdown of 10 %

Source: SuperChems Expert

ioMosaic

In order to use the dynamics, some parameters have to be either calibrated from test data or set to conservative values

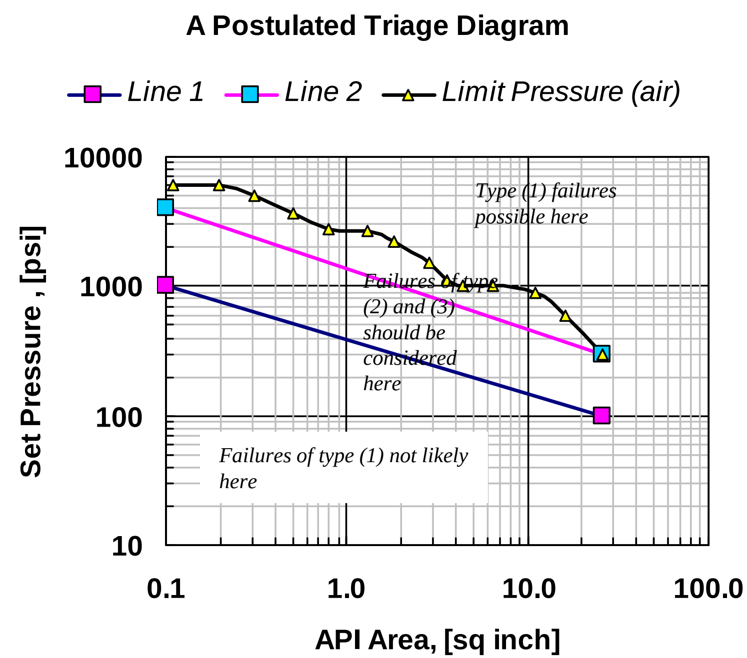
- Accurate valve geometry – A_D, A_I, A_N and A_{bel}
  - ✓ This information should be easy to obtain – Provide as ratios to A_N
- Mass of moving parts and spring constant
  - ✓ Obtain from manufacturer or use Mike Grolmes estimation method
- Maximum lift, and Discharge Coefficient at maximum lift
  - ✓ Obtain from valve manufacturer or Red book
- Obtain from manufacturer or calibrate using PRV dynamic model from actual test data for specific classes of PRVs
  - ❑ Fluid exit angle at full lift
  - ❑ Discharge coefficient vs. lift parameter
  - ❑ Momentum transfer to disk efficiency factor

How many cycles can be allowed before PRV and/or PRV components made from typical steel fail?

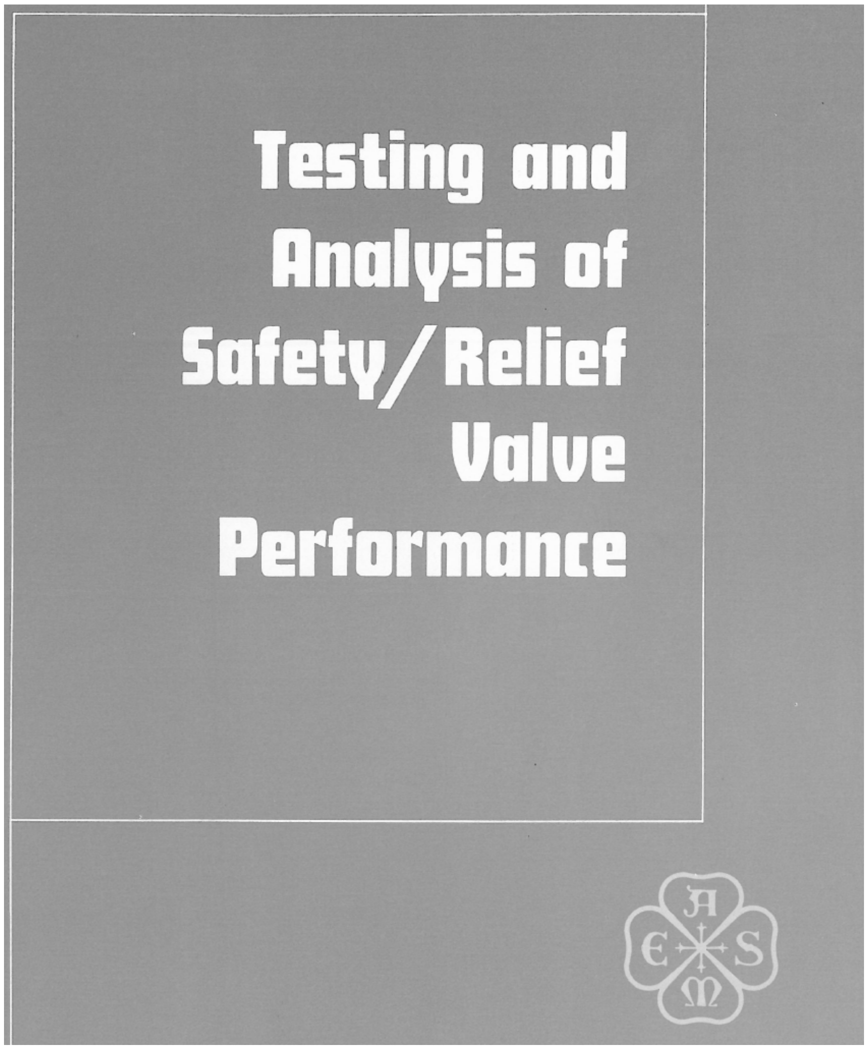
Applied Cyclic Stress - psi	Cycles to Failure
300,000	23
200,000	90
100,000	550
50,000	6,700
30,000	38,000
20,000	100,000
10,000	> 100,000

Source: Peng and Peng, "Pipe Stress Engineering", ASME, 2009

See Grolmes Presentation at DIERS 2011 Fall meeting regarding types of failure and how they relate to size of valve and set points



Key References



CONTENTS

Related Presentations

- Grolmes, M., “PRV Stability Odds and Ends”, Parts I, II, III, and IV. Multiple DIERS Users Group Meetings.
- G. A. Melhem and Harold G. Fisher, “Guidelines for dealing with excessive pressure drop in relief systems”, DIERS Users Group, 2003
- G. A. Melhem, “Deal with controversial topics in pressure relief systems”, DIERS Users Group, Spring 2009, Orlando, Florida.
- G. A. Melhem, “Estimate acoustically induced vibration risk in relief and flare piping”, Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany
- G. A. Melhem, “Pressure Relief Valve Stability”, Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany

Acknowledgements

- The author would like to acknowledge the critique and numerous contributions of Harold Fisher and Mike Grolmes

SuperChems PRV Stability Model Descriptions

This model description is in DRAFT form and is provided for discussion purposes only.

The DIERS users group is continuing to improve and develop the model, especially those aspects dealing with damage to system and valve components

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

SAFETY RELIEF VALVE STABILITY

PILOT OPERATED PRESSURE RELIEF VALVES

RESONANT (CHATTERING) PHENOMENA

PRESENTATION TO THE DIERS USERS GROUP MEETING

by

HAROLD G. FISHER

Burr Ridge, IL

OCTOBER 10, 2011

ANDERSON GREENWOOD

PILOT OPERATED SAFETY RELIEF VALVE

MODEL 273

2 – SERIES 200 (GAS TRANSMISSION)

7 - FULL LIFT, FULL BORE ORIFICE

(NOZZLE DIAMETER EQUALS INLET PIPE CONNECTION DIAMETER)

3 – SOFT SEAT / SEAL

BODY BOWL DUAL OUTLET NOZZLES

3 – 5 PERCENT BLOWDOWN

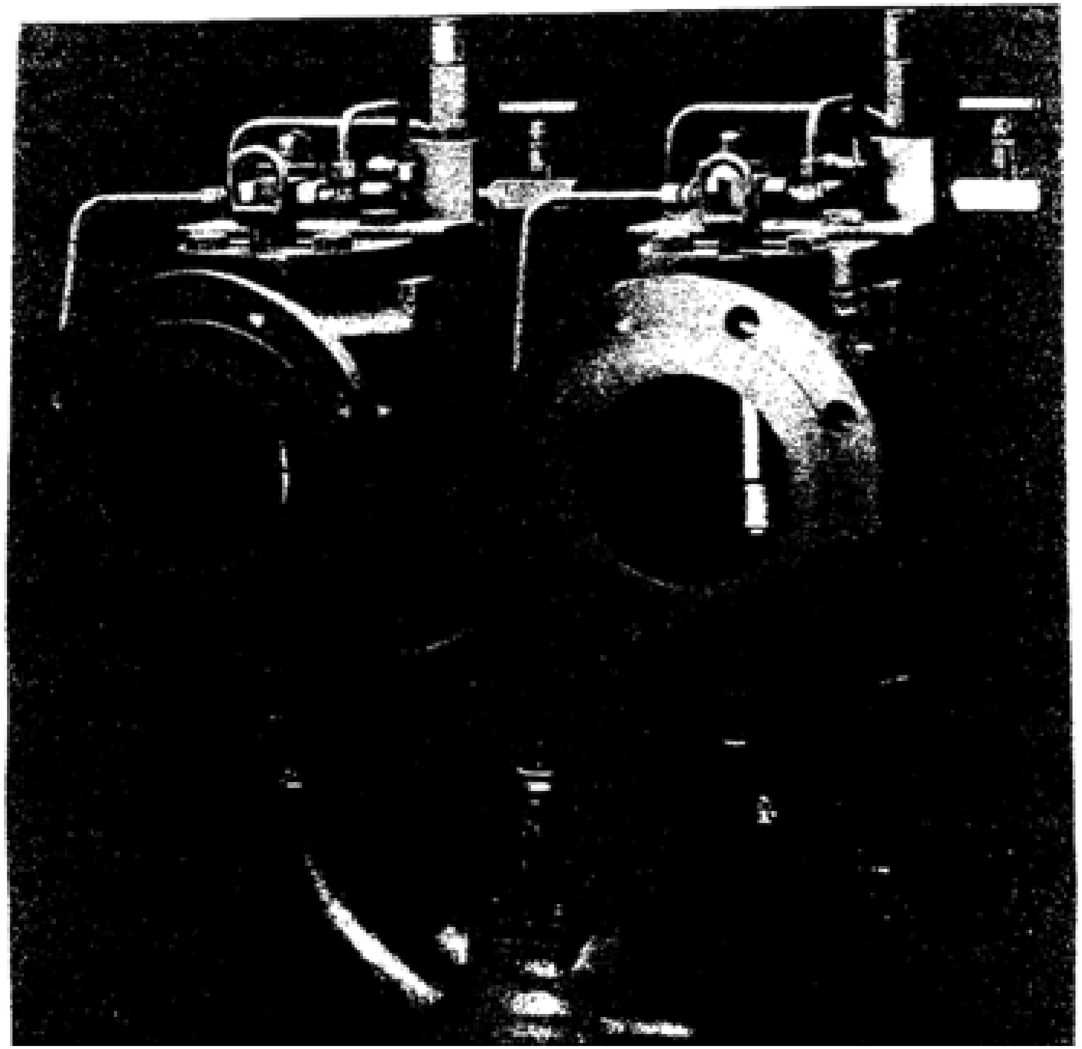
POP ACTION (FLOW AT 1.0 PSET)

HIGHER CAPACITY WITH THE SAME PIPE SIZE THAN DIRECT SPRING VALVES

ANDERSON GREENWOOD

PILOT OPERATED SAFETY RELIEF VALVE – MODEL 273

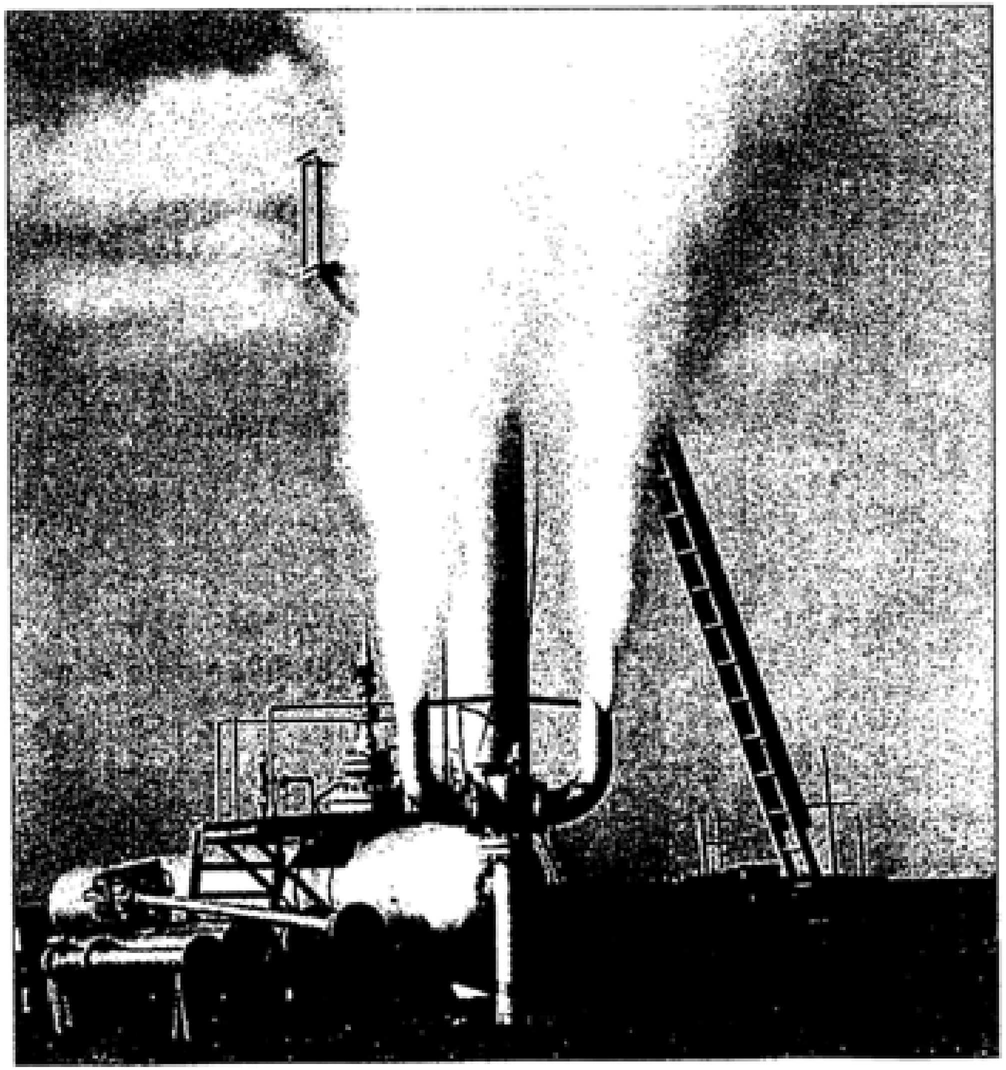
BODY BOWL DUAL OUTLET NOZZLES



ANDERSON GREENWOOD

PILOT OPERATED SAFETY RELIEF VALVE – MODEL 273

BODY BOWL DUAL OUTLET NOZZLES



ANDERSON GREENWOOD

PILOT OPERATED SAFETY RELIEF VALVE – MODEL 273

BODY BOWL DUAL OUTLET NOZZLES

ANDERSON, GREENWOOD & CO.

REPORT NUMBER
2-0175-51

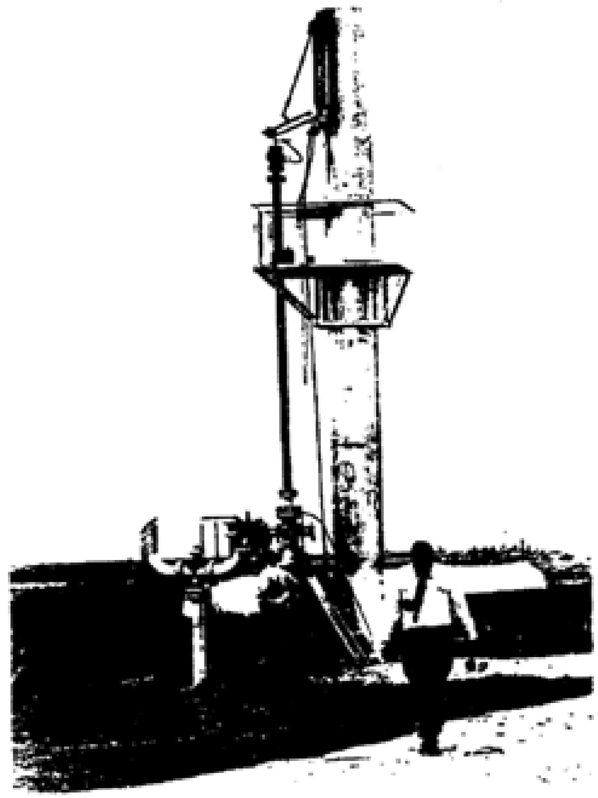


FIGURE II

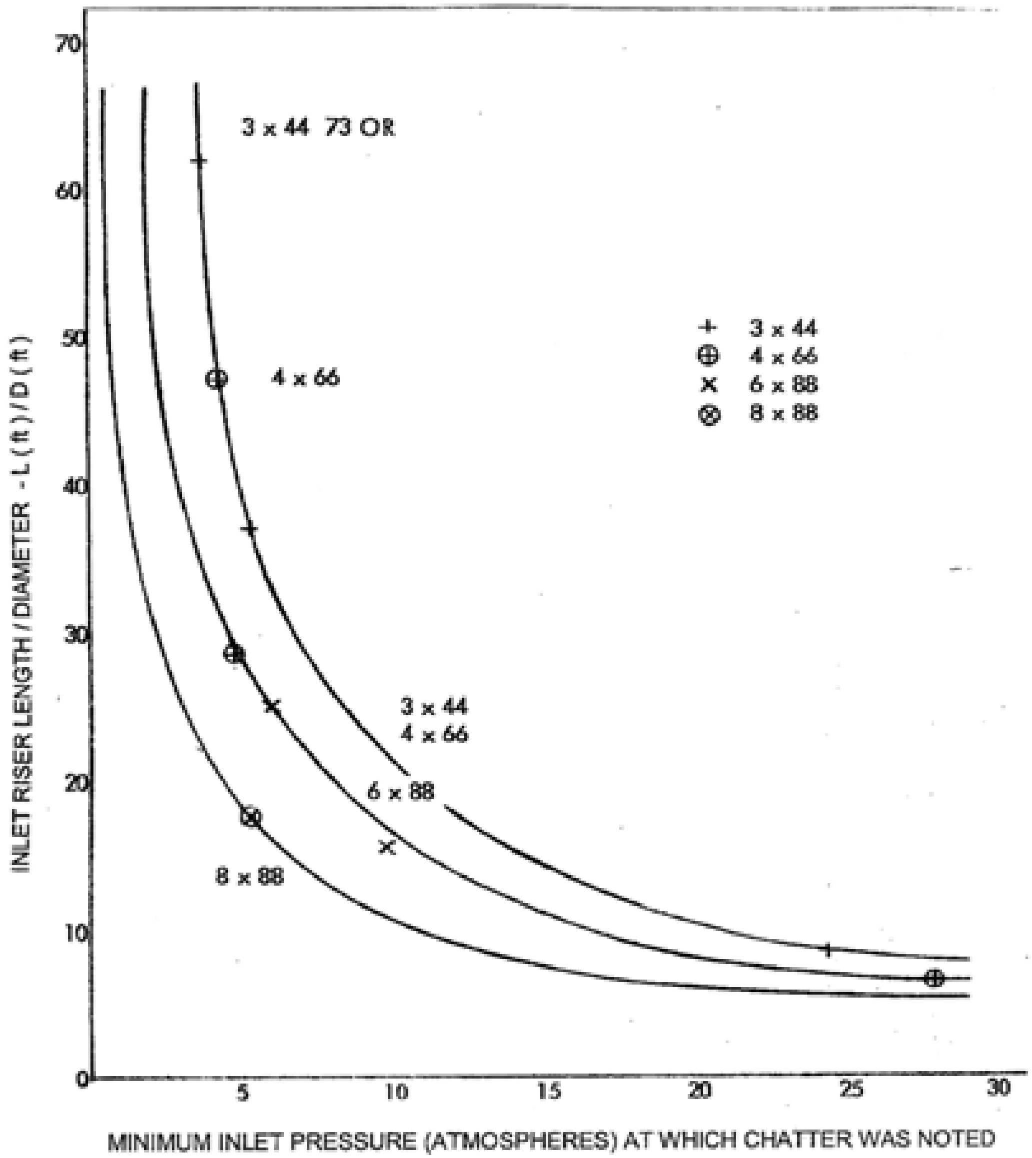
3 x 44 TYPE 273 ON 3 INCH RISER 15 FT. LONG

TEST RISER DIMENSIONS

Inlet Pipe (1) Size (inches)	RISER LENGTH AND L / D								
	Feet	L/D	Feet	L/D	Feet	L/D	Feet	L/D	
3	.5	2.07	2.09	8.66	9	37.2	15	62	
4	.5	1.57	2.09	6.55	9	28.2	15	47	
6	.5	1.04	3	6.25	7.33	15.3	12	25	
8	.5	.79	3.83	6.04	11.09	17.45			

(1) All pipe used Sched. 80

CALC			REV	DATE	FIGURE III MINIMUM CHATTER PRESSURE VS. INLET RISER LENGTH	
CHECK						
APR					ANDERSON, GREENWOOD & CO. HOUSTON, TEXAS	PAGE
APR						



VALVE STABILITY

TOTAL STABILITY OF THE TEST FACILITY WITH THE RISER DIAMETER EQUAL TO THE VALVE NOZZLE DIAMETER WAS ACHIEVED BY LIMITING THE INLET RISER LENGTH TO DIAMETER RATIO TO FIVE

TOTAL STABILITY COULD ALSO BE ACHIEVED THROUGH THE USE A RISER WITH ONE PIPE DIAMETER LARGER THAN THE INLET DIAMETER OF THE SAFETY RELIEF VALVE PIPE NOZZLE

VALVE DESTRUCTION

VALVES HAVE BEEN DESTROYED BY CHATTER WHERE A BLOCK VALVE IS THE ONLY PIPING BETWEEN THE TANK AND THE VALVE

VALVES HAVE ALSO BEEN DESTROYED WHERE A SINGLE ELBOW OR EQUIVALENT RESTRICTION IS USED BETWEEN THE VALVE AND THE TANK

NOTE: THE RISER (PIPE LENGTHS) FOR THE ANDERSON GREENWOOD STUDY WERE CHOSEN TO PROMOTE RESONANT CHATTER

RESONANT (CHATTERING) PHENOMENA

ACOUSTIC STUDY – RESONANT PIPE FREQUENCY AFFECTED BY

1. INLET PIPE GEOMETRY
2. RATIO OF VALVE NOZZLE TO PIPE INSIDE DIAMETER
3. SIZE OF RESERVOIR FEEDING THE PIPE
4. OTHER

NOTE: FUNDAMENTAL FREQUENCY IS INDEPENDENT OF THE PIPE DIAMETER

COUPLING OF THE FUNDAMENTAL AND / OR MULTIPLES OF THE PIPE FREQUENCY CAN RESULT IN HARMONIC (CHATTER) DESTRUCTION OF THE VALVE

CLOSURE - ANDERSON GREENWOOD STABILITY FIX

A PATENTED PRESSURE ACTUATED SEAL DRAG* WAS DEVELOPED AND APPLIED TO THE PISTONS IN PILOT OPERATED SAFETY RELIEF VALVES AS A METHOD TO PROVIDE STABILITY OF THE MAIN VALVE IN LESS THAN IDEAL INSTALLATIONS

* PLASTIC WEDGE RING ACTUATED BY THE PISTON “O” RING WHICH EFFECTIVELY DAMPENS THE PISTON

ENGINEERING ANALYSIS FINDINGS

1. PISTON GEOMETRY DID NOT SIGNIFICANTLY CHANGE THE MINIMUM CHATTER PRESSURE

2. WEIGHT REDUCTION OF THE PISTON DID SIGNIFICANTLY AFFECT THE MINIMUM CHATTER POINT

3. REACTION TIME OF THE MAIN VALVE WAS NOT A FACTOR IN THE MINIMUM CHATTER PRESSURE

4. MODULATION DID NOT INDUCE CHATTER EARLIER PRESSUREWISE THAN SNAP ACTION

5. LOCATION OF THE PILOT PRESSURE PICKUP (INTERNAL OR REMOTE) DID NOT AFFECT THE MINIMUM CHATTER PRESSURE

6. MINIMUM CHATTER PRESSURE IS A DIRECT FUNCTION OF THE EQUIVALENT RISER LENGTH

7. BACKPRESSURE (EITHER SUPERIMPOSED OR DEVELOPED) DOES NOT AFFECT STABILITY. THE PILOT, BEING UNBALANCED, MUST BE VENTED TO THE ATMOSPHERE WHEN HIGH BACKPRESSURE IS ENCOUNTERED

```
RUNAWAY REACTION EMERGENCY RELIEF SIZING PROGRAM (RRERSP)

DATE AND TIME: 10/08/11 13:34:25

PHYSICAL PROPERTY FILE NAME IS: soliddagas

CASE DATA FILE NAME IS: user273chatter3sc

ENTER: REACTIVE SUBROUTINE NAME

     sbclds

ENTER: WT IN TK(LBS), TK TEMP(DEG C), WT FRACTION REACTANT AND INITIAL
       CONVERSION. NOTE: THE COMPOSITION DIFFERENCE IS PRODUCT

      1000.0      15.56  1.0000  0.0000

ENTER: FLOW EVALUATION (1 = RELIEF DEVICE ONLY, 2 = INCLUDE VESSEL
       DISENGAGEMENT)

     1

ENTER: VESSEL VOID FRACTION, MAWP(PSIG), VESSEL DIAMETER(FT), TOTAL
P(PSIA),
       N2 P(PSI), MIN TOTAL P(PSIA), SURROUNDINGS P(PSIA) AND FLOWING
       QUALITY

     0.900   1440.0   5.00      0.0    505.1      0.0   14.696  1.0000

ENTER: TYPE VESSEL (ASME (1 = VERTICAL, 2 = HORIZONTAL), FIELD (3 = API
       650, 4 = API 620), LABORATORY (5 = ARC/VSP, 6 = VENTED VSP/RRSI,
       7 = VENTED TGA, 8 = VENTED DSC), SPECIAL (9 = SADT), DOT (10 =
       TANK TRUCK), AAR (11 = RAIL TANK CAR))

     1

ENTER: VESSEL HEAD TYPE (1 = ASME 2:1 ELLIPTICAL, 2 = FLAT), COMPLEX
       (MULTIPLE DIAMETER) VESSEL (0 = NO, 1 = USER SPECIFIED) AND
       INCLUDE VESSEL WETTED WALL HEAT CAPACITY (0 = NO, 1 = YES)

     1    0    0

ENTER: TYPE RELIEF DEVICE (1 = NO RELIEF, 2 = SAFETY VALVE, 3 = RUPTURE
       DISK, 4 = SV AND RD IN PARALLEL, 5 = BREATHER VENT, 6 = SV/W RD IN
       SERIES, 7 = BV AND SV IN PARALLEL, 8 = BV AND RD IN PARALLEL,
       9 = DUMP SYSTEM, 10 = OPEN PIPE), NUMBER OF RELIEF IDENTIAL RELIEF
       DEVICES, RELIEF LOCATION (1 = TOP OF VESSEL, 2 = BOTTOM OF VESSEL)
       AND VAPOR VENT ON TOP OF VESSEL (0 = NO, 1 = YES)

     2    1    1    0

ENTER: TYPE V-L ONSET/DISENGAGEMENT (1 = PROGRAM CHOICE, 2 = VAPOR, 3 =
       LIQUID, 4 = FLASHING LIQUID, 5 = CHURN-TURBULENT, 6 = BUBBLY,
       7 = HOMOGENEOUS), ONSET/DISENGAGEMENT MODEL (1 = DIERS ORIGINAL,
       2 = INTEGRATED) AND CO VALUE (1 = DIERS STANDARD (CT-1.5, B-1.2,
       H-1.01), 2 = USER SPECIFIED)

     1    1    1

ENTER: SV TWO-PHASE FLOW MODEL (1 = HEQ/HFZ/FL, 2 = HFZ, 3 = HNE), SV SLIP
       FLOW (0 = NO, 1 = YES) AND SV TWO-PHASE VISCOSITY MODEL (1 = VOLUME
       AVERAGE, 2 = BANKOFF W/SIMPSON MODIFICATION, 3 = BEATTIE AND
WHALLEY,
       4 = LIQUID, 5 = USER SPECIFIED CONSTANT VISCOSITY)

     1    0    1

ENTER: SV VISCOSITY BASIS (1 = PRESSURE AVERAGED CONDITION, 2 = THROAT
       (CHOKED) CONDITION)

     1

ENTER: SV AREA(IN2), SV INLET FLANGE RATING(PSIG), SV SET P(PSIA), TYPE
       SV CODE (1 = CONVENTIONAL, 2 = BELLOWS, 3 = PILOT OPERATED, 4 =
       LABORATORY, 5 = USER SPECIFIED), SV DISCHARGE PIPE (0 = NO, 1 =
YES),
       SV INLET PIPE (0 = NO, 1 = YES), SV BLOWDOWN (0 = IDEAL (NO
BLOWDOWN),
       1 = REAL (BLOWDOWN)) AND SV KV FACTOR (1 = DARBY-MOLAVI, 2 = API
       STANDARD)

     6.6510    600.0    505.0    3    0    1    1    1

ENTER: TYPE SV CODE (1 = CONVENTIONAL, 2 = BELLOWS, 3 = PILOT OPERATED),
       VAPOR SVCD (0-1.0), LIQUID SVCD (0-1.0), ASME SV FLOW DERATING
       FACTOR (0.9-1.0), SV OVERPRESSURE FLOW FACTOR (1.0/1.1), SV BLOWDOWN
       FACTOR (0.93(VAP/2PF)/0.7-0.9(LIQ)), SV INLET NOZZLE DIAMETER(IN)
       AND SV DISCHARGE NOZZLE DIAMETER(IN)

     3   0.899   0.899   1.000   1.000   0.930    2.906    3.826

ENTER: PILOT OPERATED SV PRESSURE TAP LOCATION (0 = LOCAL, 1 = REMOTE)

     1

CAUTION: AN ASME SV FLOW DERATING FACTOR GT 0.9 AFFECTS INLET/DISCHARGE
PIPE PRESSURE DROP CALCULATIONS

ENTER: SV INLET PIPE: ID(IN), ELEVATION CHANGE(FT), PIPE LENGTH(FT) AND
       FITTING VELOCITY HEADS - INCLUDE FITTINGS, VALVES, ETC. DO NOT
       INCLUDE PIPE ENTRANCE MOMENTUM VELOCITY HEADS

     2.906   15.00    15.00    0.000

ENTER: PIPE ROUGHNESS (1 = COMMERCIAL STEEL PIPE, 2 = USER SPECIFIED)
       AND PIPE CONSTRUCTION (1 = FLANGED/WELDED, 2 = SCREWED/TUBING,
       3 = USER SPECIFIED CONSTANT FRICTION FACTOR)

     1    3

ENTER: USER SPECIFIED PIPE CONSTANT FRICTION FACTOR
```

```
     0.0067500

ENTER: PIPE FITTING LOSSES (1 = VARIABLE (TURBULENT/LAMINAR FLOW), 2 =
       CONSTANT (TURBULENT FLOW ONLY)), PIPE VISCOSITY (1 = TURBULENT,
       2 = LAMINAR) AND PIPE VISCOSITY MODEL (1 = VOLUME AVERAGE, 2 =
       BANKOFF W/SIMPSON MODIFICATION, 3 = BEATTIE AND WHALLEY,
       4 = LIQUID, 5 = USER SPECIFIED CONSTANT VISCOSITY)

     1    1    1

ENTER: NUMBER OF PIPE FITTINGS: 1) STANDARD RADIUS 90 ELBOWS, 2) STANDARD
       RADIUS 45 ELBOWS, 3) TEES WITH BRANCH FLOW, 4) TEES WITH CROSS FLOW,
       5) FULL-PORT GATE VALVES AND 6) PIPE REDUCTION

     0    0    0    0    0    0

ENTER: VESSEL NOZZLE TYPE (1 = SQUARE-CUT, 2 = IDEAL, 3 = CHAMFER-CUT,
       4 = PENETRATING, 5 = USER SPECIFIED (TURBULENT FLOW ONLY))

     1

*** CAUTION: THE EXPECTED RESEAT PRESSURE OF THE SPECIFIED SV IS  455.5
PSIG. THE NORMAL OPERATING PRESSURE MUST BE LESS THAN THIS VALUE PLUS
A TOLERANCE FOR THE SV TO CLOSE ***

INITIAL PRESS IS  505.10 PSI NITROGEN,   0.00 PSI NON-CONDENSABLE GAS,
     0.00 PSI VOLATILE LIQUID AND    0.00 PSI HYDROSTATIC HEAD. TOTAL
PRESSURE IS  505.10 PSIA

CHOKED VAPOR FLOW IN SV NOZZLE
TURBULENT VAPOR FLOW IN SV NOZZLE
UNCHOKED VAPOR FLOW IN SV INLET PIPE
TURBULENT VAPOR FLOW IN SV INLET PIPE
27.61 PERCENT INLET PRESSURE LOSS GE 25.0 PERCENT ALLOWABLE IN SV INLET
PIPE DUE TO VAPOR FLOW FROM A PILOT OPERATED SV WITH A REMOTE PRESSURE
TAP. SV VAPOR FLOW IS  128783. PPH
*** WARNING: CRYOGENIC CONDITION AT THE SV NOZZLE DISCHARGE TEMPERATURE
OF  -23.0 C DUE TO VAPOR FLOW ***

SV MAX VAPOR FLOW IS   128783. PPH. VESSEL TEMP, PRESS AND QUALITY AT MAX
FLOW ARE     15.6 C,   505.1 PSIA AND 1.0000 WT FRACTION VAPOR. PRESS AND
TEMP AT THE SV NOZZLE EXIT ARE   260.2 PSIG AND  -23.0 C. SV MAX FLOW IS
  36287. SCFM AIR AT 60 F AND 14.7 PSIA

MAX INLET PRESSURE LOSS AT SV IS  135.4 PSI OR  27.6 PERCENT OF SV SET
PRESSURE AT A PRESSURE OF   505.1 PSIA AND A  VAPOR FLOW OF   128783. PPH
```

CALCULATION TABLE (PILOT OPERATED SAFETY RELIEF VALVE)

S80 PIPE DIAMETER	FRICTION FACTOR	VESSEL NOZZLE	% IPD
3	0.00675	SQUARE	27.6
3	0.00675	IDEAL	23.5
3	0.00500	IDEAL	17.1
4	0.00675	IDEAL	8.1

COMMON:	ENGINEERING CONSIDERATIONS
FLOW AT PSET	CHAMFER CUT INSPECTION
L / D = 62	FRICTION FACTOR
REMOTE TAP	0.9 ASME FACTOR

SAFETY RELIEF VALVE STABILITY
(JUSTIFICATION FOR SRV CALCULATION UPDATES)

PRESENTED TO THE

DIERS 2011 JOINT MEETING

JUNE 14, 2011

BY

HAROLD G. FISHER

FISHERINC

(CONSULTANT TO FAUSKE & ASSOCIATES, LLC)

U.S. OCCUPATIONAL SAFETY AND HEALTH ADMINISTRATION (OSHA)

"UNDER THE OCCUPATIONAL SAFETY AND HEALTH ACT OF 1970, OHSA'S ROLE IS TO PROMOTE SAFE AND HEALTHFUL WORKING CONDITIONS FOR AMERICA'S WORKING MEN AND WOMEN BY SETTING AND ENFORCING STANDARDS." [1] OSHA STANDARDS ARE DESIGNED TO SAVE WORKERS' LIVES. THE OSHA PSM STANDARD [2] REQUIRES EMPLOYERS TO REDUCE THE RISKS OF FATAL OR CATASTROPHIC INCIDENTS. EMPLOYER SAFETY MANAGEMENT SYSTEMS ARE TO BE DEVELOPED AND FULLY IMPLEMENTED TO PROTECT WORKERS FROM SERIOUS INCIDENTS.

OSHA PSM STANDARD - PROCESS SAFETY MANAGEMENT OF HIGHLY HAZARDOUS CHEMICALS (29 CFR 1910.119)

THE U.S. OCCUPATIONAL SAFETY AND HEALTH ADMINISTRATION'S (OSHA) PROCESS SAFETY MANAGEMENT (PSM) STANDARD [2], ISSUED ON FEBRUARY 24, 1992, SETS REQUIREMENTS FOR THE MANAGEMENT OF HAZARDS ASSOCIATED WITH PROCESSES THAT USE HIGHLY HAZARDOUS CHEMICALS (HHC) - E.G. CHEMICALS THAT ARE TOXIC, REACTIVE, FLAMMABLE OR EXPLOSIVE. REQUIREMENTS FOR PREVENTING OR MINIMIZING THE CONSEQUENCES OF CATASTROPHIC RELEASES OF TOXIC, REACTIVE, FLAMMABLE OR EXPLOSIVE CHEMICALS ARE CONTAINED IN THE STANDARD.

EMPLOYERS MUST COMPLETE A COMPILATION OF WRITTEN PROCESS SAFETY INFORMATION (PSI). "INFORMATION PERTAINING TO THE EQUIPMENT IN THE PROCESS SHALL INCLUDE THE "RELIEF SYSTEM DESIGN AND DESIGN BASIS." [2](29 CFR 1910.119(D)(3)(D)) "THE EMPLOYER SHALL DOCUMENT THAT THE EQUIPMENT COMPLIES WITH RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICES" (RAGAGEPS)." [2](29 CFR 1910.119(D)(3)(II))

OSHA PETROLEUM REFINERY PROCESS SAFETY MANAGEMENT NATIONAL EMPHASIS PROGRAM (NEP)

AS PART OF ITS OVERALL ENFORCEMENT ACTIVITIES, OSHA USES NATIONAL EMPHASIS PROGRAMS (NEPS) TO TARGET ESTABLISHMENTS OR INDUSTRIES THAT HAVE KNOWN OR SUSPECTED HAZARDOUS CONDITIONS. THE PETROLEUM REFINERY PROCESS SAFETY MANAGEMENT NATIONAL EMPHASIS PROGRAM [3] BECAME EFFECTIVE ON JUNE 7, 2007. THE INSTRUCTION DESCRIBES POLICIES AND PROCEDURES FOR IMPLEMENTING A NATIONAL EMPHASIS PROGRAM (NEP) TO REDUCE OR ELIMINATE THE WORKPLACE HAZARDS ASSOCIATED WITH CATASTROPHIC RELEASES OF HIGHLY HAZARDOUS CHEMICALS AT PETROLEUM REFINERIES BY VERIFYING THE EMPLOYER'S COMPLIANCE WITH OSHA'S PROCESS SAFETY MANAGEMENT (PSM) STANDARD.

THE NEP DIRECTS OSHA COMPLIANCE SAFETY AND HEALTH OFFICERS (CSHOS) TO REQUEST AND REVIEW DOCUMENTS INCLUDING COPIES OF THE RELIEF DESIGN AND DESIGN BASIS FOR THE RELIEF VALVES INCLUDED IN THE PROCESS SAFETY INFORMATION (PSI) OF A PSM-COVERED PROCESS. EXAMPLES OF EQUIPMENT / SYSTEM DEFICIENCIES INCLUDE "EQUIPMENT OR SYSTEMS THAT ARE NOT DESIGNED, FABRICATED, CONSTRUCTED, OR INSTALLED PER RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICES (RAGAGEP. E.G., DEFICIENCIES THAT DO NOT MEET RAGAGEP INCLUDE THE DESIGN PRESSURE DROP AT THE INLET OF A RELIEF DEVICE THAT EXCEEDS LIMITS SPECIFIED IN RAGAGEP SUCH AS THE BPVC AND API 520, PART II. [3](NEP, IX(A)(3)(1)

RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICE (RAGAGEP)

"THE EMPLOYER SHALL DOCUMENT THAT THE EQUIPMENT COMPLIES WITH RECOGNIZED AND GENERALLY ACCEPTED GOOD ENGINEERING PRACTICES." [2] "RAGAGEPS ARE ENGINEERING, OPERATIONAL, OR MAINTENANCE ACTIVITIES BASED ON ESTABLISHED CODES, STANDARDS, PUBLISHED TECHNICAL REPORTS, RECOMMENDED PRACTICES (RP), OR A SIMILAR DOCUMENT. RAGAGEPS DETAIL GENERALLY APPROVED WAYS TO PERFORM SPECIFIC ENGINEERING, INSPECTION OR MECHANICAL INTEGRITY ACTIVITIES, SUCH AS FABRICATING A VESSEL, INSPECTING A STORAGE TANK, OR SERVICING A RELIEF DEVICE." [3](NEP IX(A)(8))

ASME BOILER AND PRESSURE VESSEL CODE (BPVC)

VARIOUS SECTIONS OF THE BPVC HAVE BEEN ADOPTED INTO LAW IN ALL 50 U.S. STATES AND IN ALL THE CANADIAN PROVINCES. INTERNATIONALLY, THE BPVC IS ACCEPTED FOR USE IN MORE THAT 100 COUNTRIES. SPECIFIC SECTIONS OF THE BPVC ADDRESS THE DESIGN, INSTALLATION, AND PERFORMANCE OF SAFETY RELIEF VALVES.

NONMANDATORY APPENDIX M - INSTALLATION AND OPERATION

INLET PRESSURE DROP FOR HIGH LIFT, TOP GUIDED, SAFETY, SAFETY RELIEF, AND PILOT OPERATED PRESSURE RELIEF VALVES IN COMPRESSIBLE FLOW

"THE NOMINAL PIPE SIZE OF ALL PIPING, VALVES, AND FITTINGS, AND PRESSURE COMPONENTS BETWEEN A PRESSURE VESSEL AND ITS SAFETY, SAFETY RELIEF, OR PILOT OPERATED PRESSURE RELIEF VALVES SHALL BE AT LEAST AS LARGE AS THE NOMINAL SIZE OF THE DEVICE INLET, AND THE FLOW CHARACTERISTICS OF THE UPSTREAM SYSTEM SHALL BE SUCH THAT THE CUMULATIVE TOTAL OF ALL NONRECOVERABLE[1] INLET LOSSES SHALL NOT EXCEED 3% OF THE VALVE SET PRESSURE. THE INLET PRESSURE LOSSES WILL BE BASED ON THE VALVE NAMEPLATE CAPACITY CORRECTED FOR THE CHARACTERISTICS OF THE FLOWING FLUID." (M-6(A))

API RECOMMENDED PRACTICE 520, SIZING, SELECTION, AND INSTALLATION OF PRESSURE-RELIEVING DEVICES IN REFINERIES, PART II – INSTALLATION

"WHEN A PRESSURE-RELIEF VALVE IS INSTALLED ON A LINE DIRECTLY CONNECTED TO A VESSEL, THE TOTAL NON-RECOVERABLE PRESSURE LOSS BETWEEN THE PROTECTED EQUIPMENT AND THE PRESSURE-RELIEF VALVE SHOULD NOT EXCEED 3 PERCENT OF THE SET PRESSURE OF THE VALVE EXCEPT AS PERMITTED IN 4.2.3 FOR PILOT-OPERATED PRESSURE RELIEF VALVES. WHEN A PRESSURE-RELIEF VALVE IS INSTALLED ON A PROCESS LINE, THE 3 PERCENT LIMIT SHOULD BE APPLIED TO THE SUM OF THE LOSS IN THE NORMALLY NON-FLOWING PRESSURE-RELIEF VALVE INLET PIPING AND THE INCREMENTAL PRESSURE LOSS IN THE PROCESS LINE CAUSED BY FLOW THROUGH THE PRESSURE-RELIEF VALVE. THE PRESSURE LOSS SHOULD BE CALCULATED USING THE RATED CAPACITY OF THE PRESSURE RELIEF VALVE." (4.2.2)

"KEEPING THE PRESSURE LOSS BELOW 3 PERCENT BECOMES PROGRESSIVELY MORE DIFFICULT AT LOW PRESSURES AS THE ORIFICE SIZE OF A PRESSURE-RELIEF VALVE INCREASES. AN ENGINEERING ANALYSIS OF THE VALVE PERFORMANCE AT HIGHER INLET LOSSES MAY PERMIT INCREASING THE ALLOWABLE PRESSURE LOSS ABOVE 3 PERCENT." (4.2.2)

STABILITY OF SAFETY RELIEF VALVES

A REVIEW AND EVALUATION OF PRIOR WORK

by

Ron Darby
Professor of Chemical Engineering
Texas A&M University

and

Berwanger, Inc.
Houston, Texas

Presented to

The American Petroleum Institute

July 3, 2001

TABLE OF CONTENTS

Urata, E., "Thrust of Popper Valve" (1969)

Johnson, B.L. and D.E. Wandling, "Actual Popping Pressure of a Relief Valve with a Real Helical Spring Under Dynamic Load", (1969)

Kasai, K. "On The Stability of a Poppet Valve With an Elastic Support" (1968)

Jennett, E., "Components of Pressure-Relieving Systems, Part II" (1963)

AD 2000-MERKBLATT, SAFETY DEVICES AGAINST EXCESS PRESSURE - SAFETY VALVES, A2

THE AD 2000 BODY OF REGULATIONS CAN BE APPLIED TO SATISFY THE BASIC REQUIREMENTS OF THE PRESSURE EQUIPMENT DIRECTIVE IN GERMANY. THIS AD 2000-MERKBLATT APPLIES TO SAFETY ACCESSORIES GUARDING AGAINST EXCESS PRESSURE FOR PRESSURE VESSELS IN WHICH UNACCEPTABLE PRESSURE IS PREVENTED BY THE OPENING OF SAFETY VALVES OR CLOSING OF SAFETY SHUT-OFF VALVES.

"THE PRESSURE LOSS IN THE SUPPLY LINE SHALL NOT EXCEED 3% OF THE DIFFERENCE IN THE PRESSURE BETWEEN THE RESPONSE PRESSURE AND THE EXTRANEOUS BACK PRESSURE IN THE CASE OF THE MAXIMUM FLOW DISCHARGED. A PRECONDITION FOR PROPER FUNCTIONING IN THE EVENT OF SUCH PRESSURE LOSS IS THAT THE DIFFERENCE IN CLOSING PRESSURE OF THE FITTED SAFETY VALVE SHALL BE AT LEAST 5 %. WITH A DIFFERENCE IN CLOSING PRESSURE OF LESS THAN 5 % THE DIFFERENCE BETWEEN THE PRESSURE LOSS AND THE DIFFERENCE IN CLOSING PRESSURE SHALL BE AT LEAST 2 %." (6.2.2)

ISO 4126-3, SAFETY DEVICES FOR PROTECTION AGAINST EXCESSIVE PRESSURE, PART 3, SAFETY VALVES AND BURSTING DISK SAFETY DEVICES IN COMBINATION [13]

"THE CONNECTION FROM THE PROTECTED EQUIPMENT TO THE SAFETY VALVE INLET SHOULD BE AS SHORT AS PRACTICABLE AND DESIGNED SO THAT THE TOTAL PRESSURE DROP TO THE SAFETY VALVE …, INCLUDING THE EFFECT OF THE BURSTING DISC SAFETY DEVICE … SHALL NOT EXCEED 3 % OF THE SET PRESSURE OF THE SAFETY VALVE…"

"NOTE: THE 3% PRESSURE DROP IS DETERMINED FROM FLOW THROUGH THE COMBINATION AT MAXIMUM RELIEVING PRESSURE OF THE SAFETY VALVE …" (6.2)

ISO 4126-9, SAFETY DEVICES FOR PROTECTION AGAINST EXCESSIVE PRESSURE, PART 9, APPLICATION AND INSTALLATION OF SAFETY DEVICES EXCLUDING STAND-ALONE BURSTING DISC SAFETY DEVICES

"UNLESS OTHERWISE SPECIFIED BY NATIONAL CODES OR REGULATIONS, THE INLET LINE SHALL BE DESIGNED THAT THE TOTAL PRESSURE DROP TO THE VALVE DOES NOT EXCEED 3% OF THE SET PRESSURE OF THE SAFETY DEVICE, OR ONE THIRD OF THE BLOWDOWN, WHICH EVER IS LESS." (6.2)

"NOTE: THIS IS BASED ON THE CORRELATION OF THE SETTLED PRESSURE LOSS IN THE INLET LINE OF 3% RELATIVE TO THE STANDARD BLOWDOWN OF 10%, WHICH IS APPROXIMATELY ONE THIRD." (6.2)

ANNEX C - SIZING OF INLET LINES

"THIS ANNEX PRESENTS A METHOD FOR SIZING INLET PIPING SYSTEMS OF SAFETY DEVICES TO OBTAIN ACCEPTABLE PRESSURE LOSSES. IT IS APPLICABLE TO STEAM, GAS, AND LIQUID." (C.1)

"… THE PRESSURE DROP IN THE INLET PIPE SHALL NOT EXCEED 3%." (C.3)

AICHE / CCPS GUIDELINE FOR PRESSURE RELIEF AND EFFLUENT HANDLING SYSTEMS

"THE 3% RULE (ASME BPVC, APPENDIX M) IS CURRENTLY ACCEPTED AS THE CRITERION FOR THE UPPER LIMIT IN INLET LOSSES TO SAFETY RELIEF VALVES. THIS RULE REQUIRES THAT THE NONRECOVERABLE (FRICTION) LOSSES BE LESS THAN 3% OF THE SET PRESSURE WHEN THE VALVE IS OPERATING AT THE NAMEPLATE CAPACITY, CORRECTED FOR THE PROPERTIES OF THE FLUID... THIS FLOW CAPACITY IS THE "RELIEVING CAPACITY" OF THE VALVE AT 10% OVERPRESSURE. SOME DESIGNERS AND VALVE MANUFACTURERS FOLLOW THE MORE CONSERVATIVE PRACTICE OF USING THE BEST ESTIMATE FLOW RATE AT 10% OVERPRESSURE FOR THE LOSS CALCULATION. THIS FLOW IS ABOUT 10% HIGHER THAN THE RELIEVING CAPACITY... NOTE THAT STANDARDS FOR COUNTRIES OTHER THAN THE USA MAY SPECIFY THIS HIGHER FLOW RATE BASIS..." (2.4.2.2.1)

OSHA NEWS RELEASES

"ON MARCH 23, 2005, AN EXPLOSION AND FIRE IN THE ISOMERIZATION UNIT OF THE BP TEXAS CITY REFINERY (BPTCR) RESULTED IN THE DEATH OF 15 CONTRACTOR EMPLOYEES AND INJURY OF AT LEASE 170 OTHER BP EMPLOYEES AND CONTRACTORS." [4]

SUMMARY OF ALLEGED VIOLATIONS AND PENALTIES

(1) INDIVIDUAL RELIEF DEVICE DEFICIENCIES

411 INSTANCES (NEW VIOLATIONS)

"STANDARD VIOLATED: §29 CFR 1910.119(D)(3)(I) AND §29 CFR 1910.119(D)(3)(II) GROUPED WITH § 29 CFR 1910.119(J)(5)."

PROPOSED PENALTIES: $28,770,000 ($70,000 PER VALVE)

(2) FAILURE TO PERFORM RELIEF DEVICE STUDIES

28 INSTANCES

DID NOT COMPLY WITH SETTLEMENT AGREEMENT

PROPOSED PENALTIES: $5,880,000 ($210,000 PER INSTANCE)

Occupational Safety & Health Administration

Home Workers Regulations Enforcement Data & Statistics Training Publications Newsroom Small Business OSHA

Fact Sheet on BP 2009 Monitoring Inspection

The U.S. Department of Labor's Occupational Safety and Health Administration (OSHA) has proposed $87,430,000 in propose penalties against BP Products North America, Inc. for 709 alleged failures to comply with the 2005 settlement agreement and citations, and violations of safety and health standards identified during the agency's inspection of the corporation's refinery in Texas City, TX (BPTCR). The inspection of the refinery was conducted from May through October 2009.

Prior History at BPTCR:

- On March 23, 2005, an explosion and fire in the Isomerization Unit of the **BP Texas City Refinery** (BPTCR) resulted in the death of 15 contractor employees and injury of at least 170 other BP employees and contractors.

Current 2009 Monitoring Inspection

- Initiated on May 4, 2009 as a comprehensive monitoring inspection
- Resulted in alleged violations and proposed penalties because BPTCR did not comply with the September 2005 agreement to correct the previous citations (270 instances) and other new violations (439 instances) which were cited.

Summary of Alleged Violations and Penalties

439 willful, per-instance citations with total penalties of $30,730,000 will also be issued for new violations of the PSM standard. A willful violation exists under the Act where an employer has demonstrated either an intentional disregard for the requirements of the Act or plain indifference to employee safety and health.

- **Individual Relief Device deficiencies**

  - 411 instances (New violations)

  - Standard Violated: §1910.119(d)(3)(i) and §1910.119(d)(3)(ii) grouped with §1910.119(j)(5)

  - Classified as Willful Egregious

  - Proposed penalties: **$28,770,000**

- **Failure to perform relief device studies**

  - 28 instances

  - Did not comply with Settlement Agreement

  - Classified as Failure to Abate (FTA)

  - Proposed penalties: **$5,880,000**

"THE U. S. DEPARTMENT OF LABOR'S OCCUPATIONAL SAFETY AND HEALTH ADMINISTRATION HAS CITED BP NORTH AMERICA INC. AND BP-HUSKY REFINING LLC'S REFINERY IN OREGON, OHIO, WITH 42 ALLEGED WILLFUL VIOLATIONS, INCLUDING 39 ON A PER-INSTANCE BASIS, AND 20 ALLEGED SERIOUS VIOLATIONS FOR EXPOSING WORKERS TO A VARIETY OF HAZARDS INCLUDING FAILURE TO PROVIDE ADEQUATE PRESSURE RELIEF FOR PROCESS UNITS. PROPOSED PENALTIES TOTAL $3,042,000." [6]

TOLEDO REFINING CITATIONS AND PROPOSED PENALTIES

"TWENTY-SIX INSTANCES ALLEGE DEFICIENT PRESSURE RELIEF, A VIOLATION OF 29 CFR PARTS 1910.119(D)(3) AND 1910.119(J)(5), WITH TOTAL PENALTIES OF $1,820,000." ($70,000 PER VALVE)

News Release

U.S. Department of Labor

Release Number: 10-234-CHI
March 8, 2010
Contact: Brad Mitchell or Scott Allen
Phone: 312-353-6976

US Labor Department's OSHA proposes more than $3 million in fines to BP-Husky refinery near Toledo, Ohio

OREGON, Ohio -- The U.S. Department of Labor's Occupational Safety and Health Administration has cited BP North American Inc. and BP-Husky Refining LLC's refinery in Oregon, Ohio, with 42 alleged willful violations, including 39 on a per-instance basis, and 20 alleged serious violations for exposing workers to a variety of hazards including failure to provide adequate pressure relief for process units. Proposed penalties total $3,042,000.

Under the Occupational Safety and Health Act of 1970, employers are responsible for providing safe and healthful workplaces for their employees. OSHA's role is to assure these conditions for America's working men and women by setting and enforcing standards, and providing training, education and assistance. For more information, visit http://www.osha.gov/index.html.

Toledo Refinery Citations and Proposed Penalties

Forty-two willful citations with proposed penalties totaling $2,940,000 are proposed as follows:

1. Thirty-eight (38) per-instance, willful citations with penalties totaling $2,660,000 allege as follows:
   a. Twenty-six instances allege deficient pressure relief, a violation of 29 CFR parts 1910.119(d)(3) and 1910.119(j)(5), with total penalties of $1,820,000;

REFERENCES

1. "U.S. LABOR DEPARTMENT'S OSHA ISSUES LETTERS TO OIL REFINERIES STRESSING COMPLIANCE WITH PROCESS SAFETY MANAGEMENT STANDARD", OSHA STATEMENT, RELEASE NUMBER: 09-0648-NAT (JUNE 10, 2009).

2. "PROCESS SAFETY MANAGEMENT OF HIGHLY HAZARDOUS CHEMICALS STANDARD" TITLE 29, CODE OF FEDERAL REGULATIONS (CFR) PART 1910.119 (FR 57 (36): 6356 - 6417 (FEBRUARY 24, 1992).

3. "PETROLEUM REFINERY PROCESS SAFETY MANAGEMENT NATIONAL EMPHASIS PROGRAM", DIRECTIVE CPL 03-00-004, U. S. DEPARTMENT OF LABOR (OSHA) (JUNE 7, 2007).

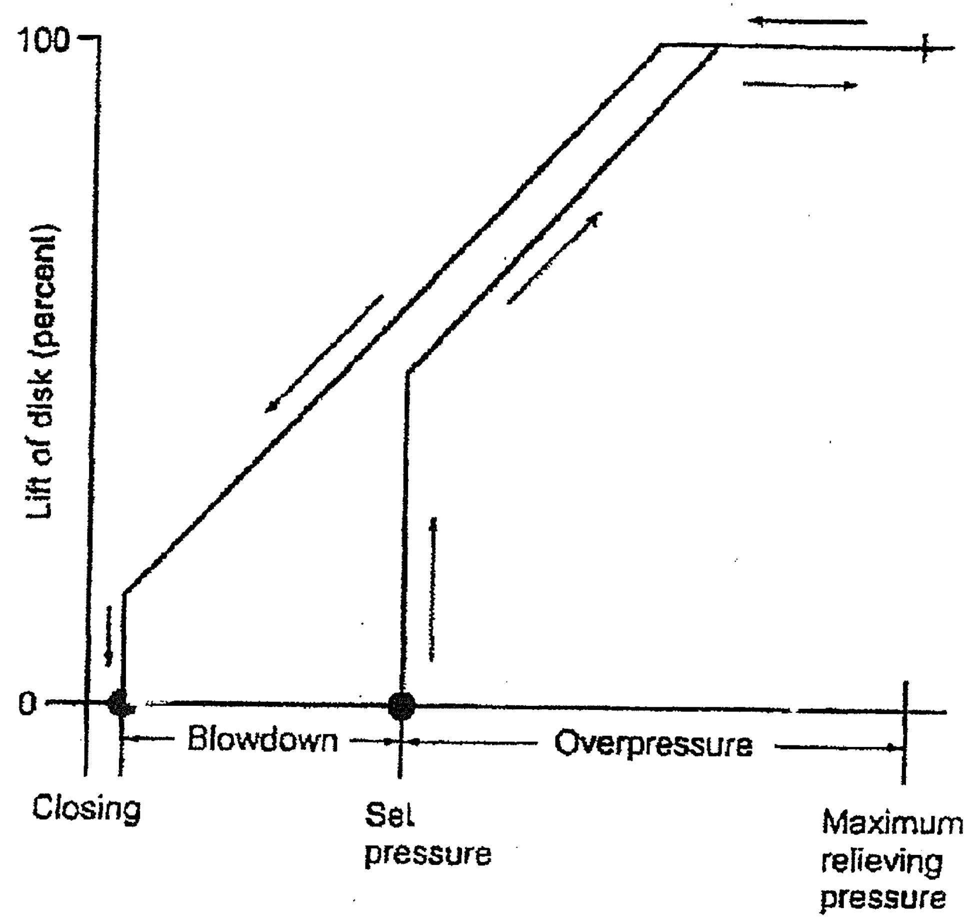
4. "OSHA FINES BP PRODUCTS NORTH AMERICA MORE THAN $21 MILLION FOLLOWING TEXAS CITY EXPLOSION", NATIONAL NEWS RELEASE, U.S. DOL, OSHA (SEPTEMBER 22, 2005). (HTTP://WWW.OSHA.GOV/PLS/OSHAWEB/OWADISP.SHOW_DOCUMENT?P_ID=11589&P_TABLE=NEWS_RELEASES)

5. FACT SHEET ON BP 2009 MONITORING INSPECTION", U.S. DOL, OSHA (OCTOBER 30, 2009). (HTTP://WWW.OSHA.GOV/DEP/BP/FACT_SHEET-BP_2009_MONITORING_INSPECTION.HTML)

6. "US LABOR DEPARTMENT'S OSHA PROPOSES MORE THAN $ 3 MILLION IN FINES TO BP – HUSKY REFINERY NEAR TOLEDO, OHIO", RELEASE NUMBER: 10-234-CHI (MARCH 8, 2010). (HTTP://WWW.OSHA.GOV/PLS/OSHAWEB/OWADISP.SHOW_DOCUMENT?P_TABLE=NEWS_RELEASES&P_ID=17223)

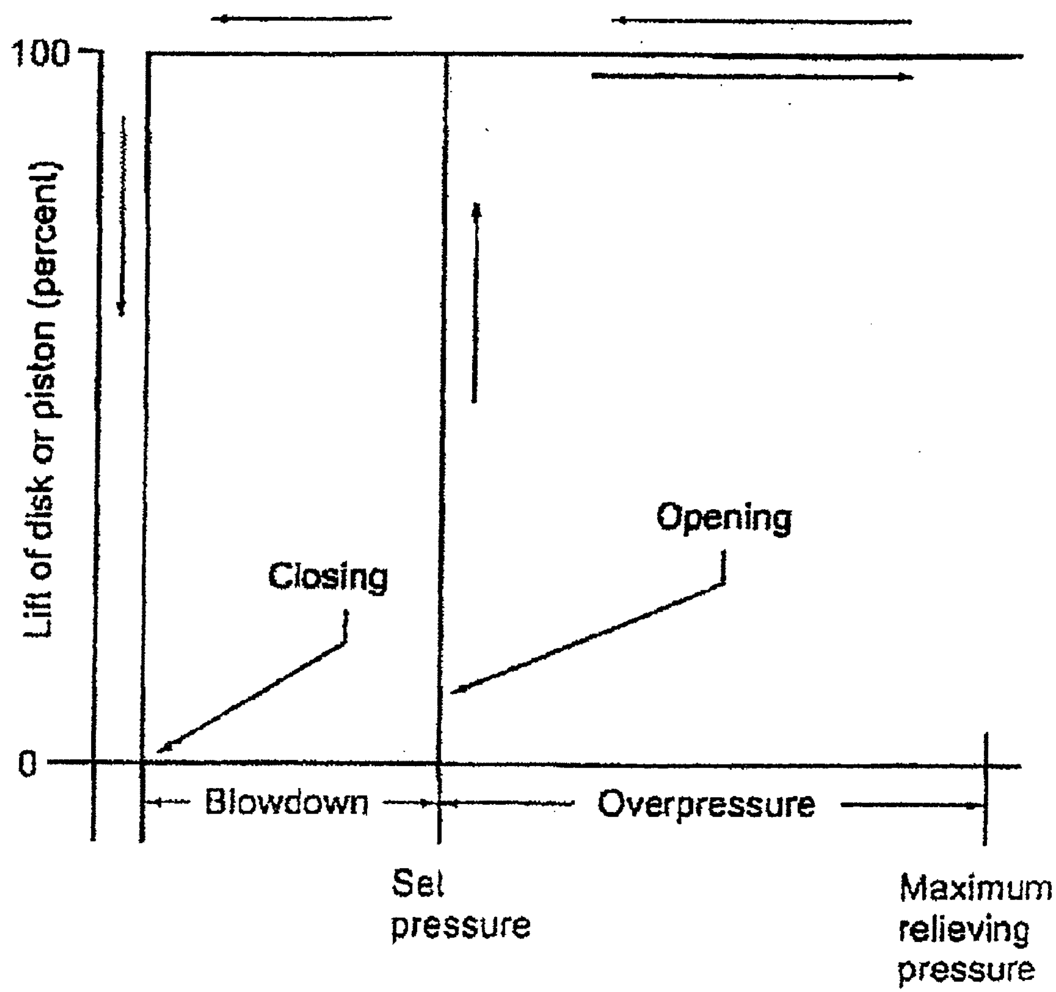
TYPICAL RELATIONSHIP BETWEEN LIFT OF DISK IN A PRV AND VESSEL PRESSURE

API STANDARD 520, PART I (8th ED) (DECEMBER 2008)



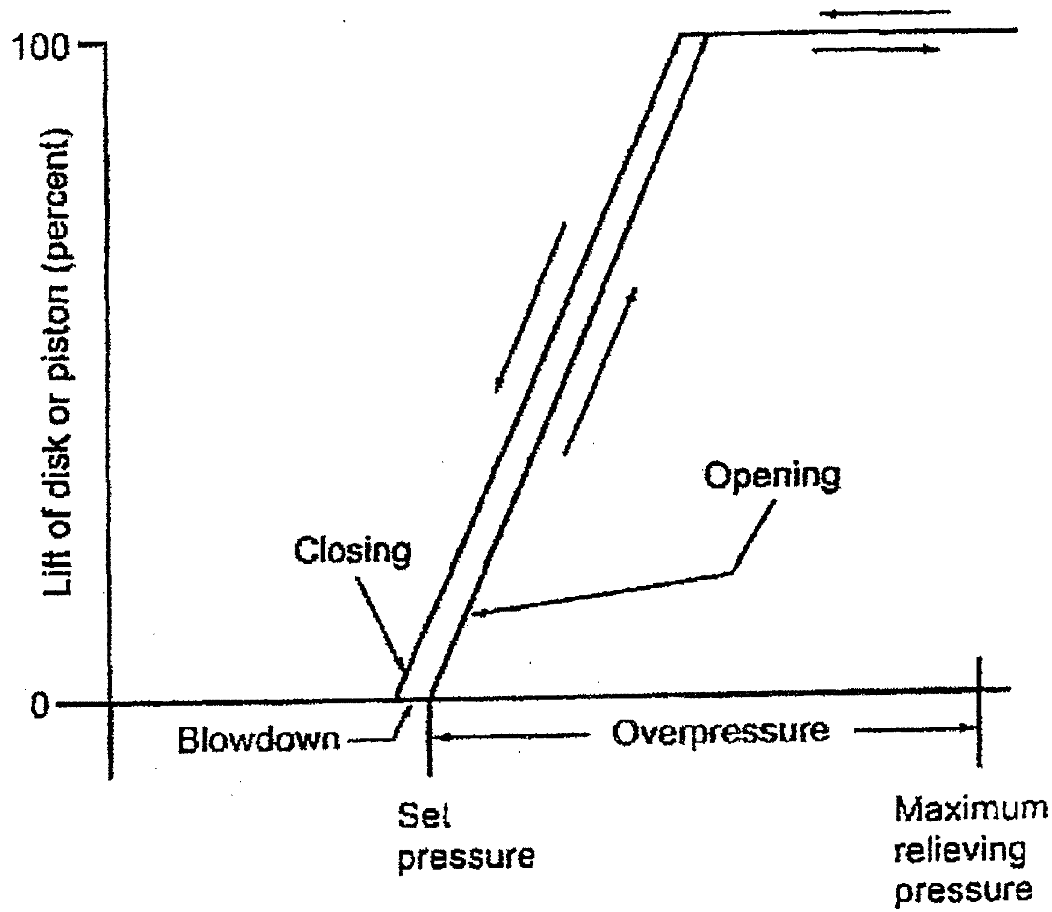
TYPICAL RELATIONSHIP BETWEEN LIFT OF DISK OR PISTON AND VESSEL PRESSURE IN A POP-ACTION PILOT-OPERATED PRV

API STANDARD 520, PART I (8th ED) (DECEMBER 2008)



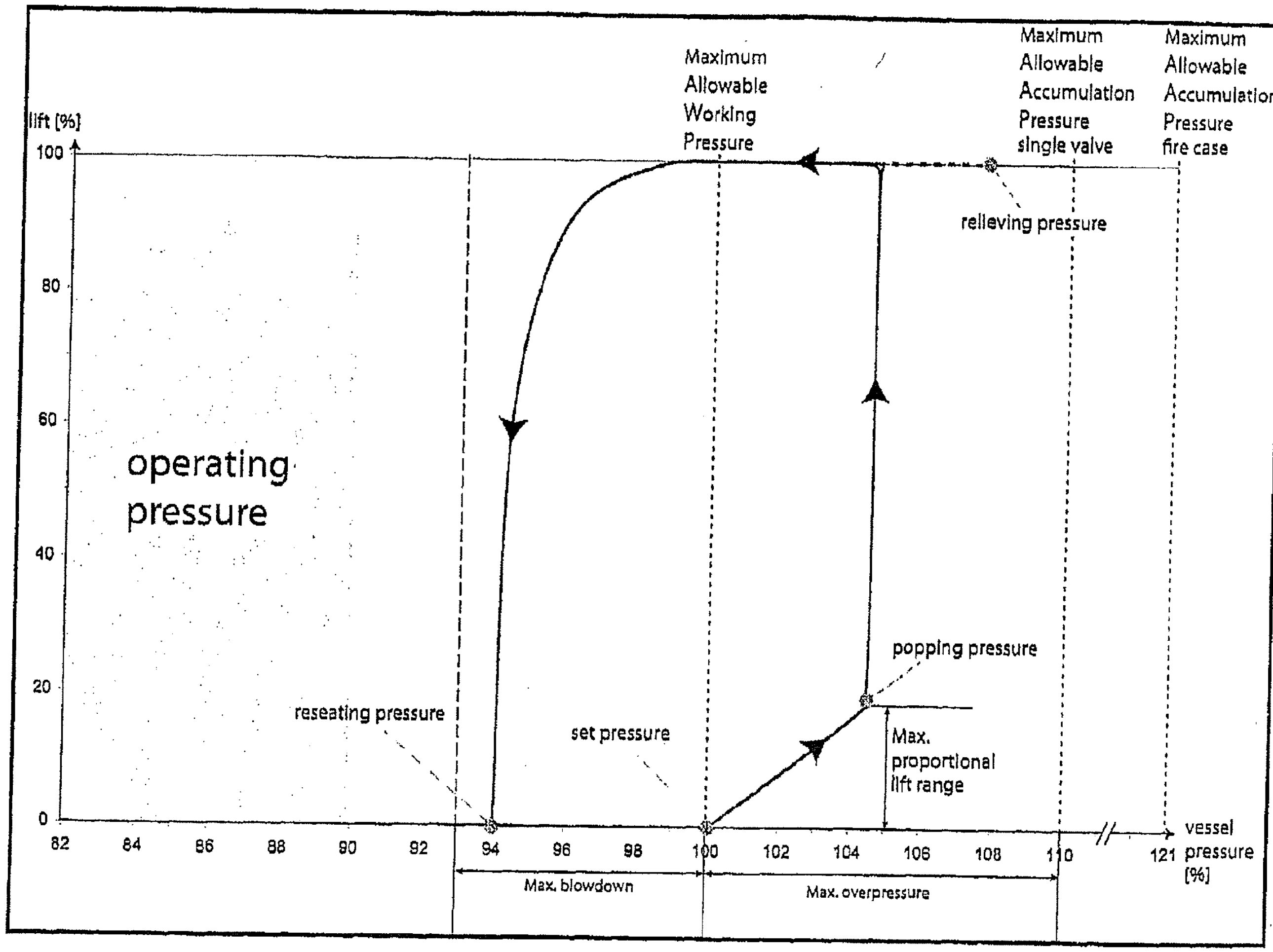
TYPICAL RELATIONSHIP BETWEEN LIFT OF DISK OR PISTON AND VESSEL PRESSURE IN A MODULATION-ACTION PILOT-OPERATED PRV

API STANDARD 520, PART I (8th ED) (DECEMBER 2008)

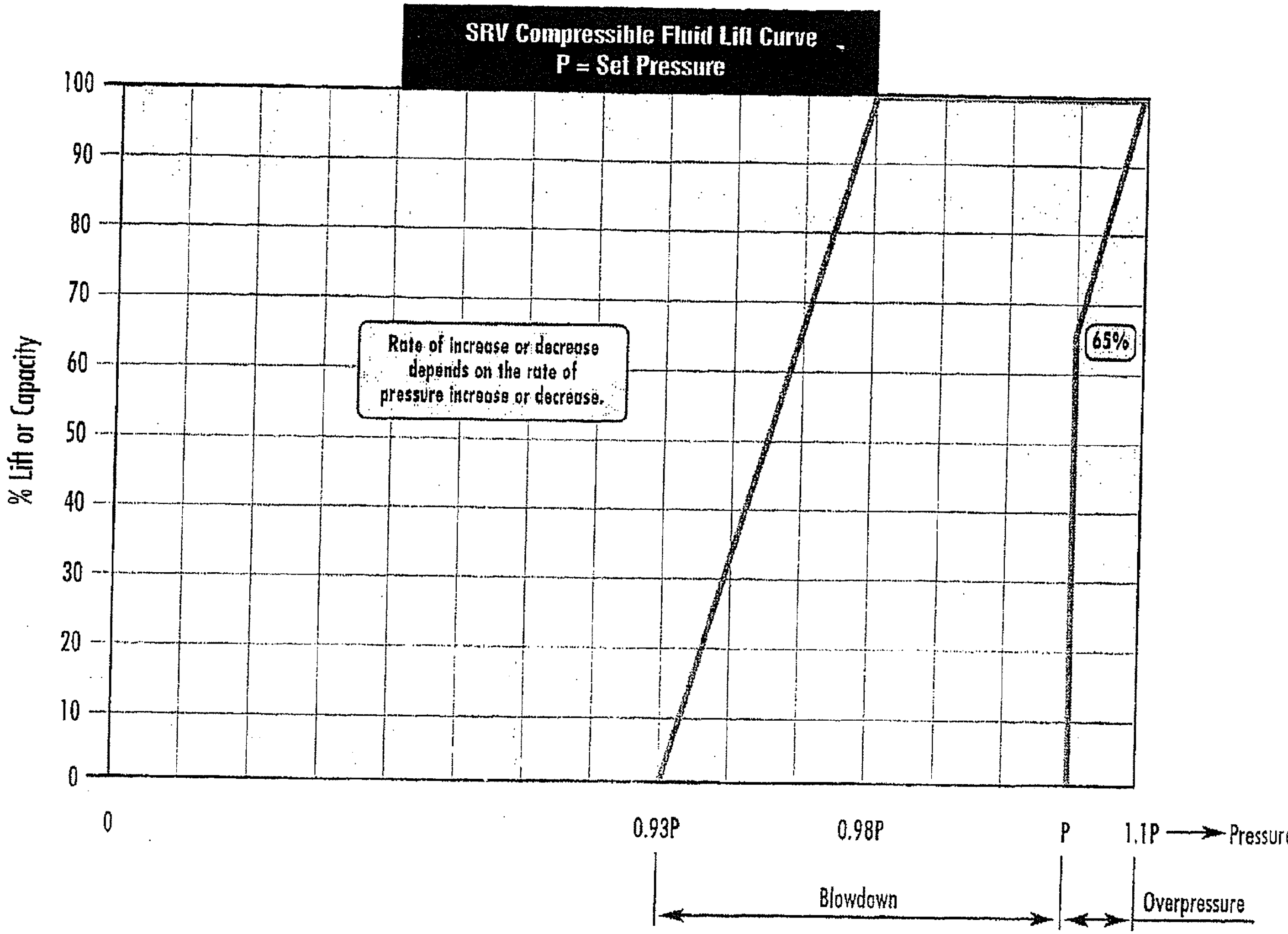


LESER –TYPICAL FUNCTION OF AN API 526 SPRING LOADED SRV

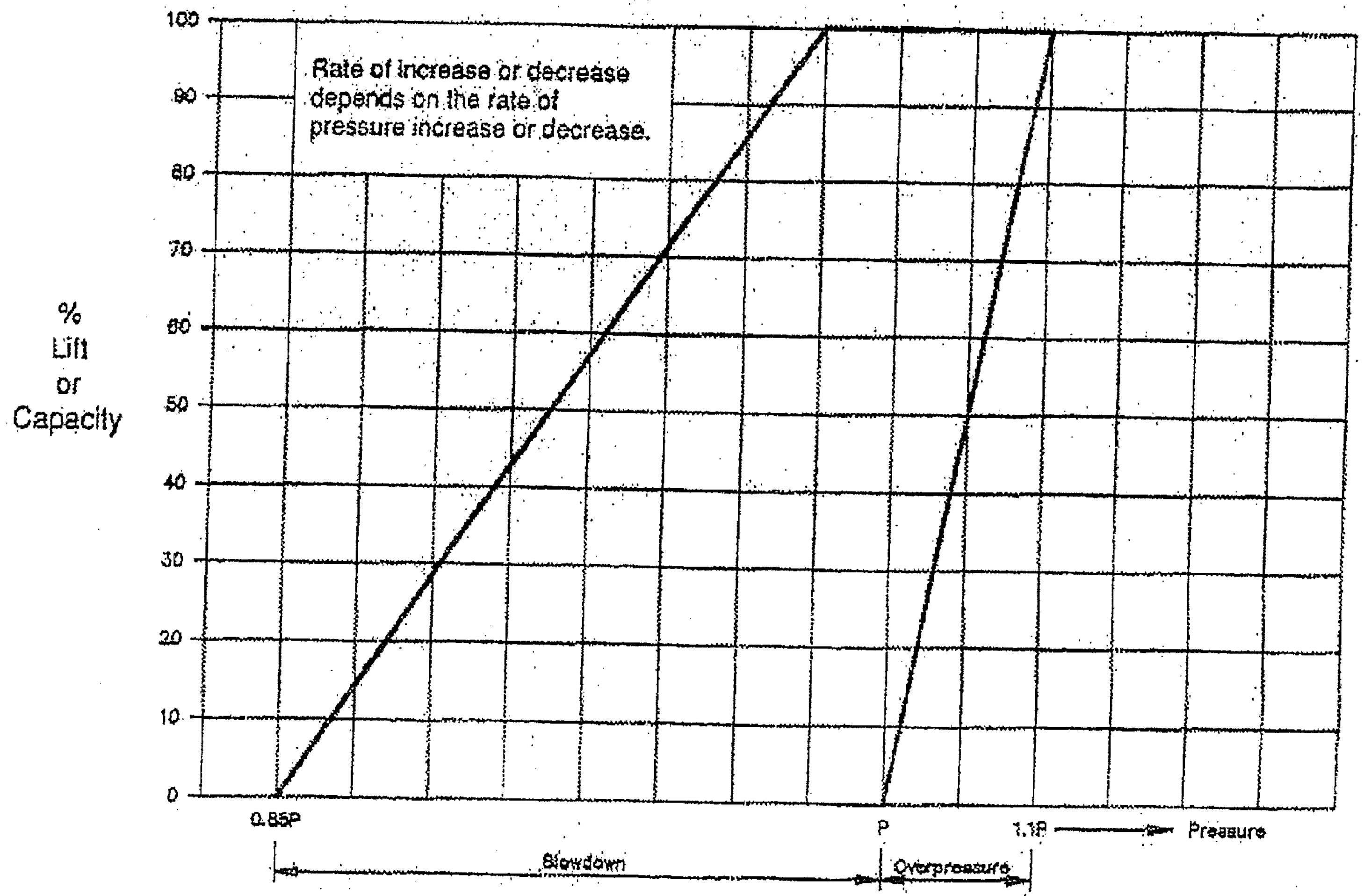
WITH ADJUSTING BLOWDOWN RING



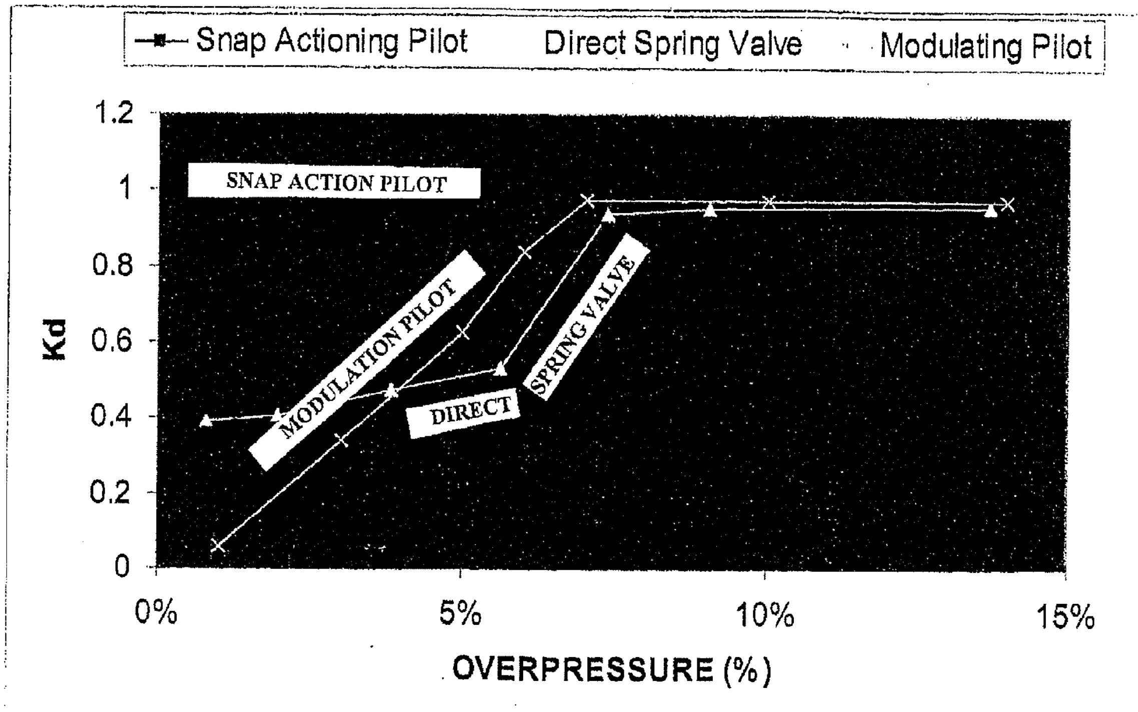
CONSOLIDATED – CAPACITY, LIFT AND CLOSING CURVES FOR SAFETY RELIEF VALVES



SRV Incompressible Fluid Lift Curve
(Based on ASME requirements for full valve lift at 10% overpressure, Valves Supplied with Liquid Trim

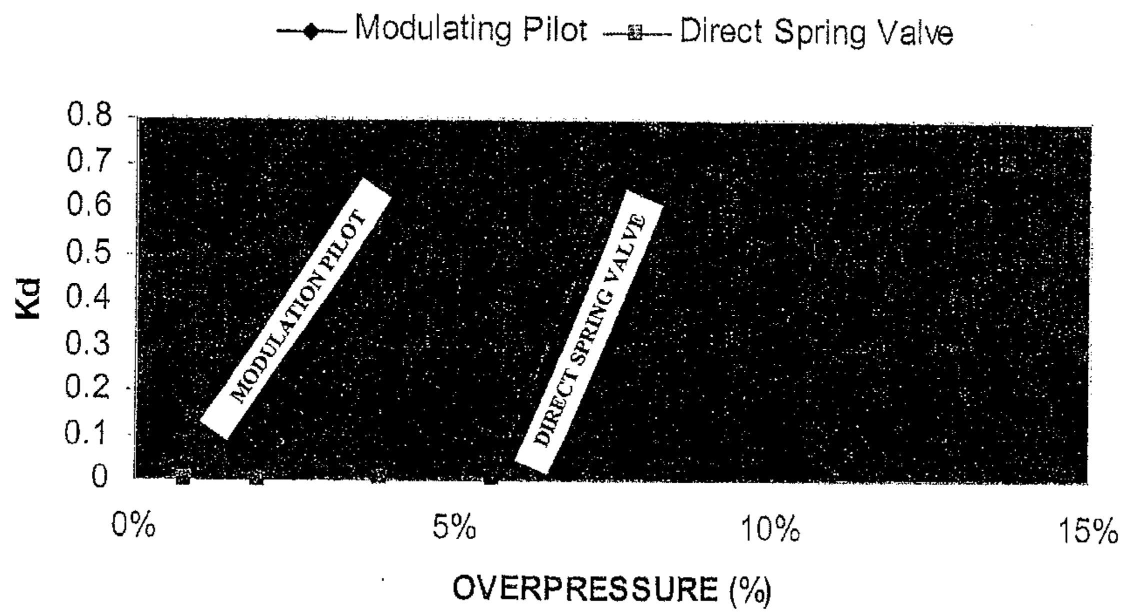


Gas Opening Test Results



2J3 (250 psig set)

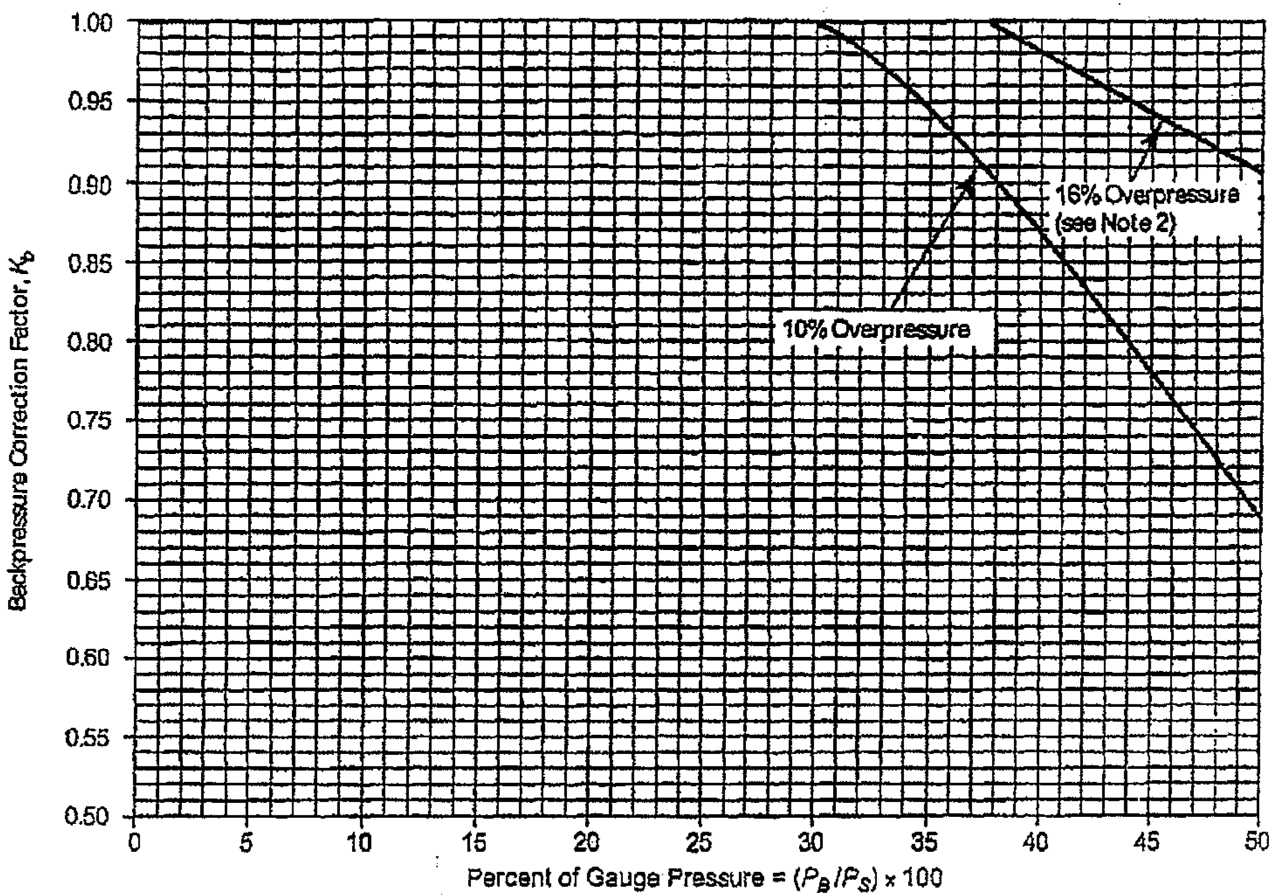
Liquid Opening Test Results



2J3 (250 psig set)

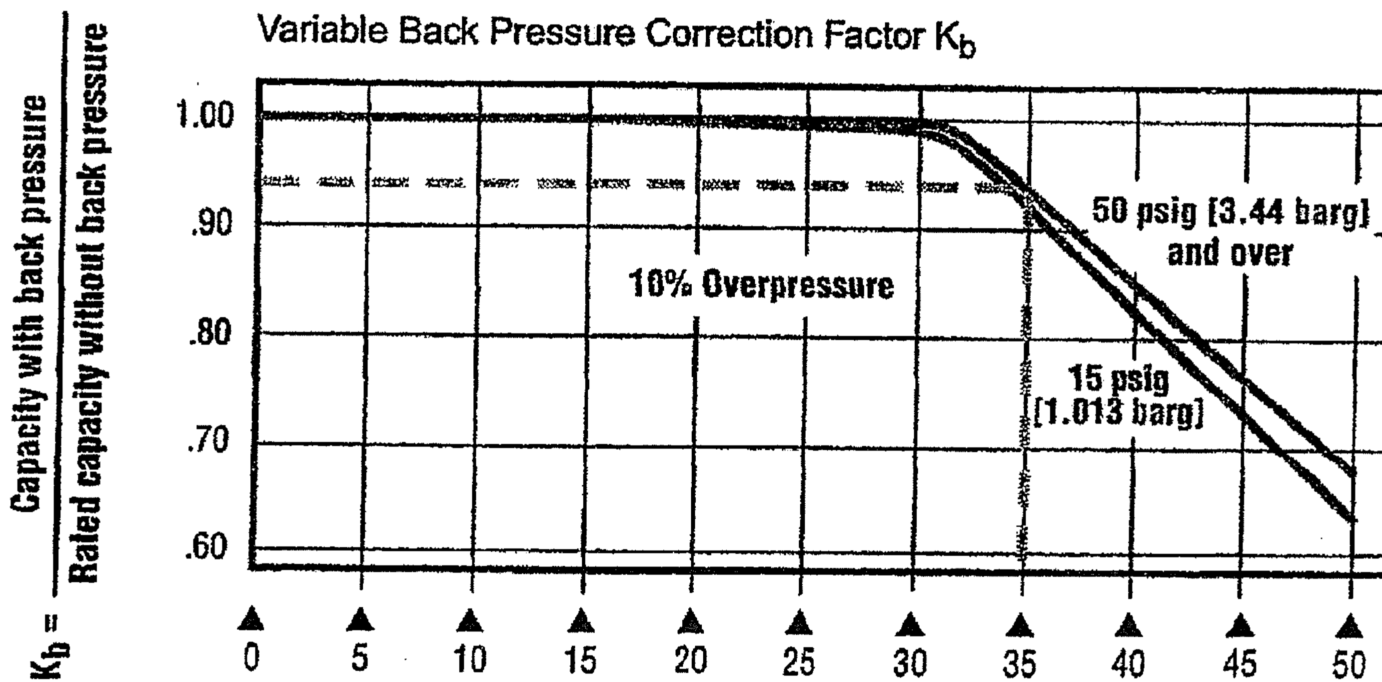
BACK PRESSURE SIZING FACTOR Kb

API STANDARD 520, PART I (8th ED) (DECEMBER 2008)

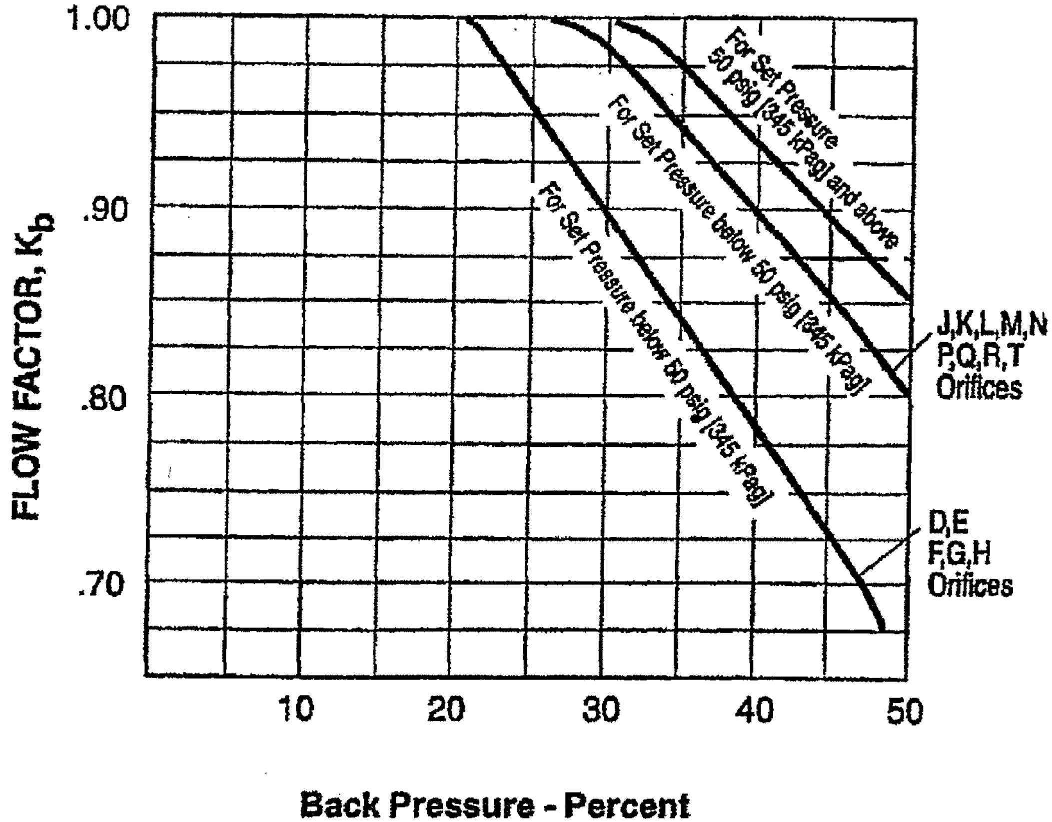


P_B = back pressure, in psig.
P_S = set pressure, in psig.

ANDERSON GREENWOOD - BACK PRESSURE SIZING FACTOR Kb

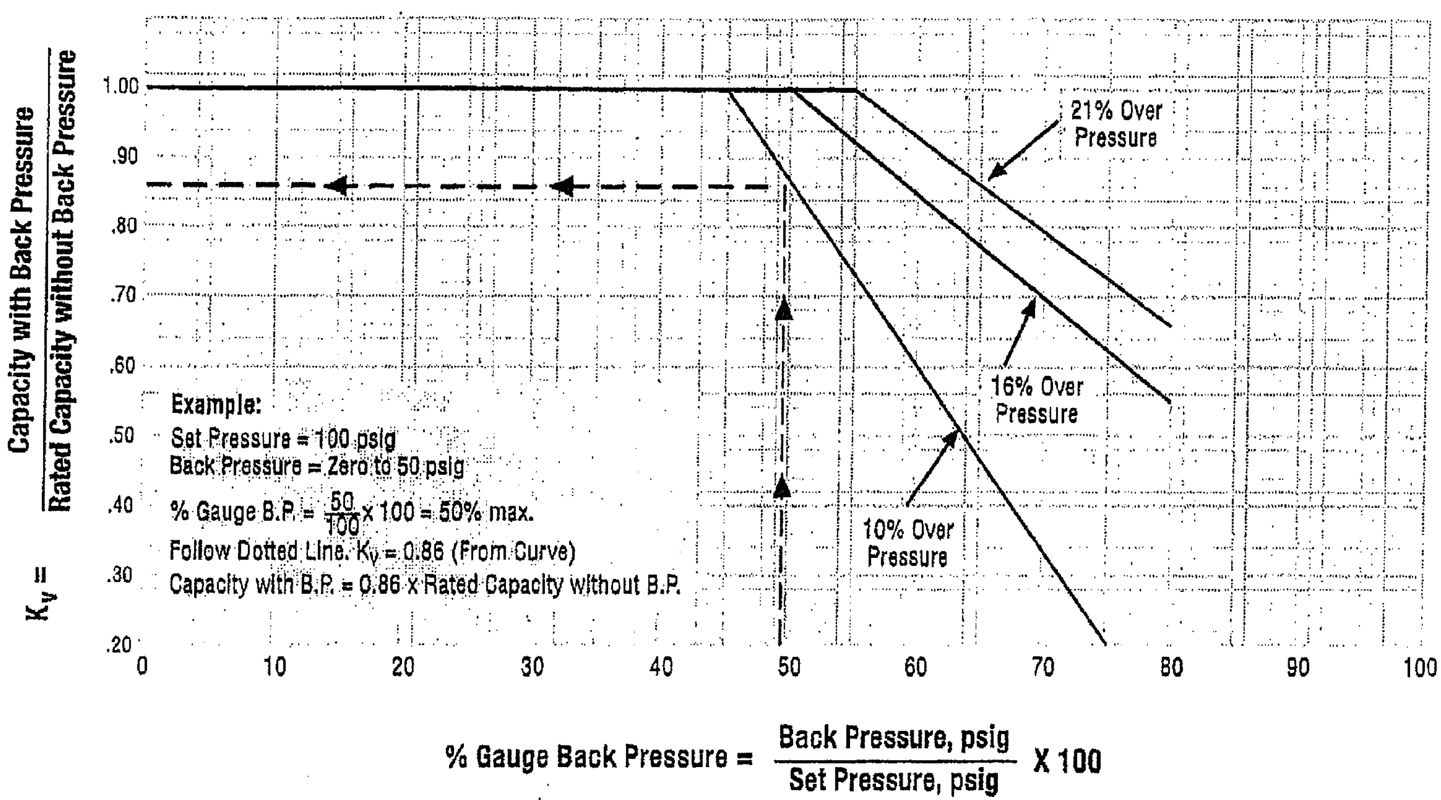


CROSBY JBS - BACK PRESSURE SIZING FACTOR Kb

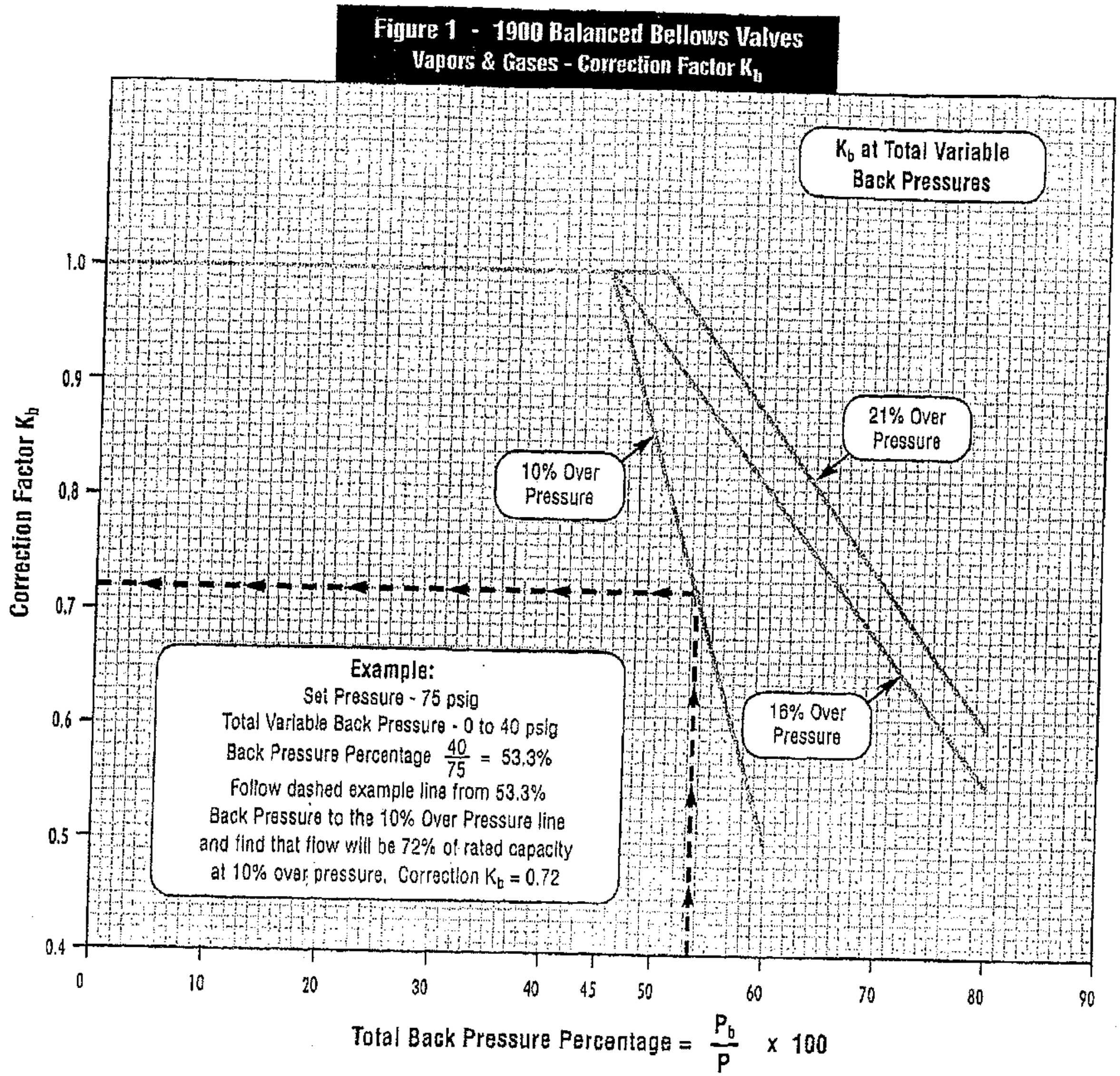


$$\frac{\text{Back Pressure (gage)}}{\text{Set Pressure (gage)}} \times 100$$

FARRIS - BACK PRESSURE SIZING FACTOR Kb

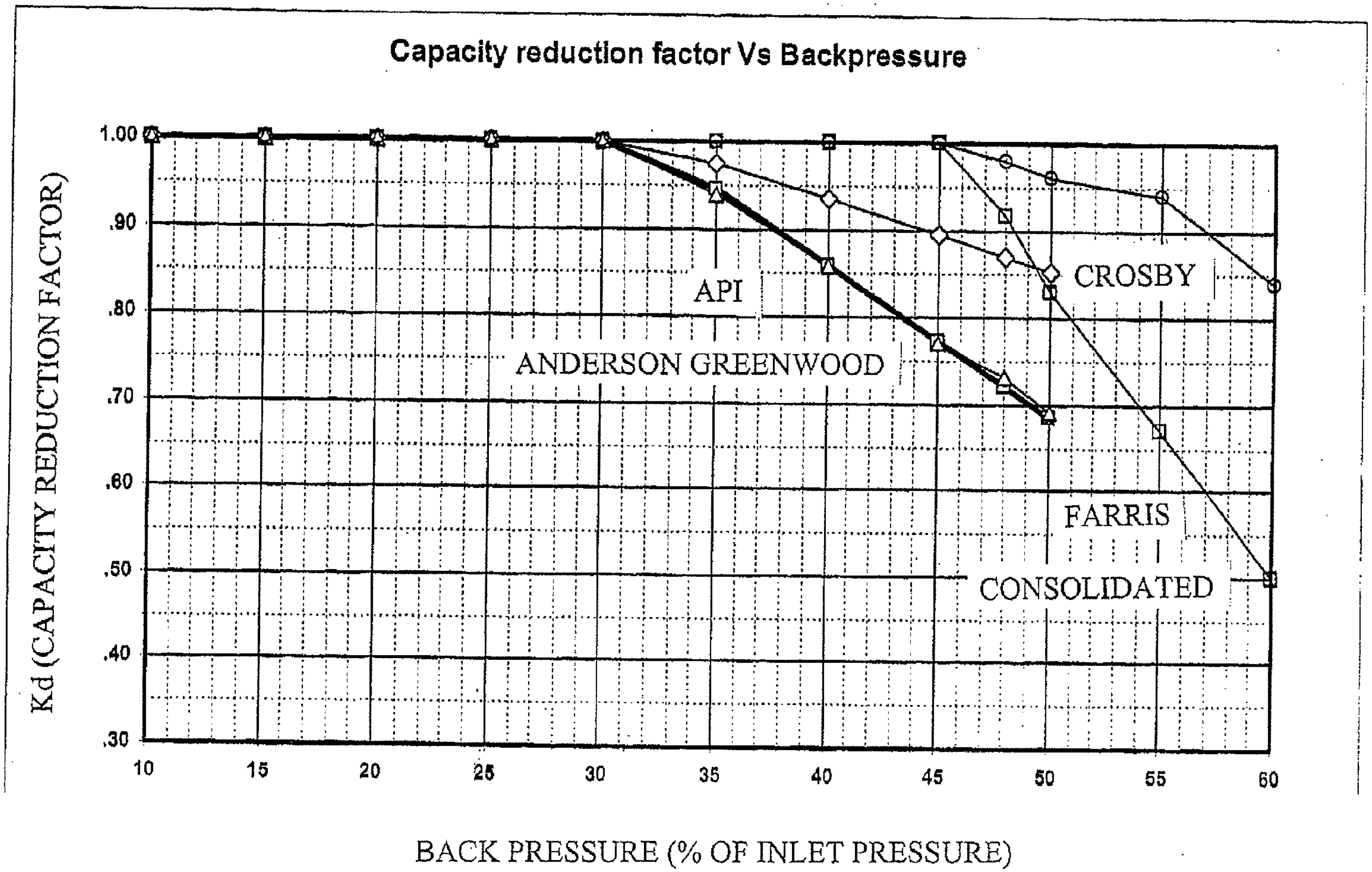


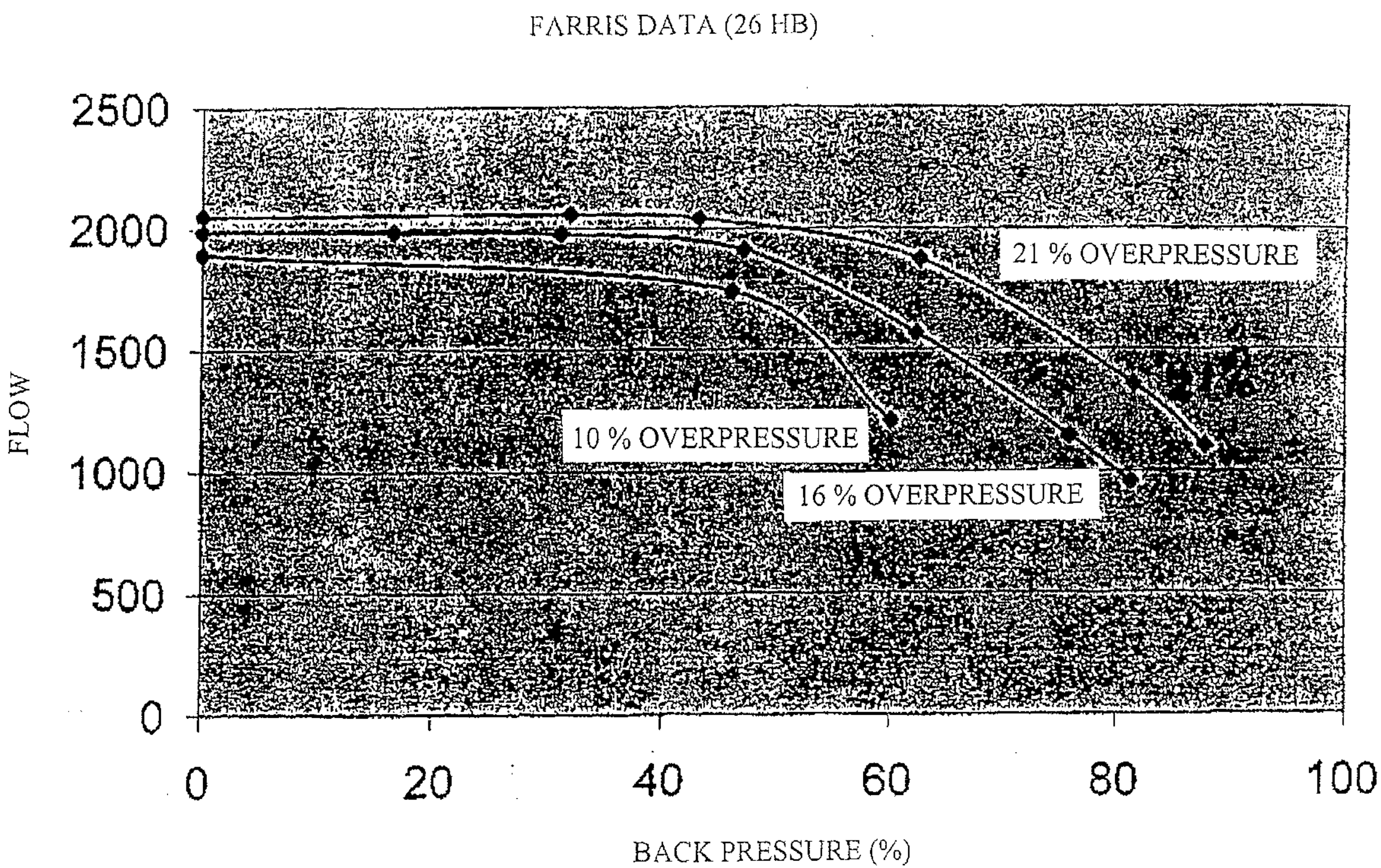
CONSOLIDATED - BACK PRESSURE SIZING FACTOR Kb



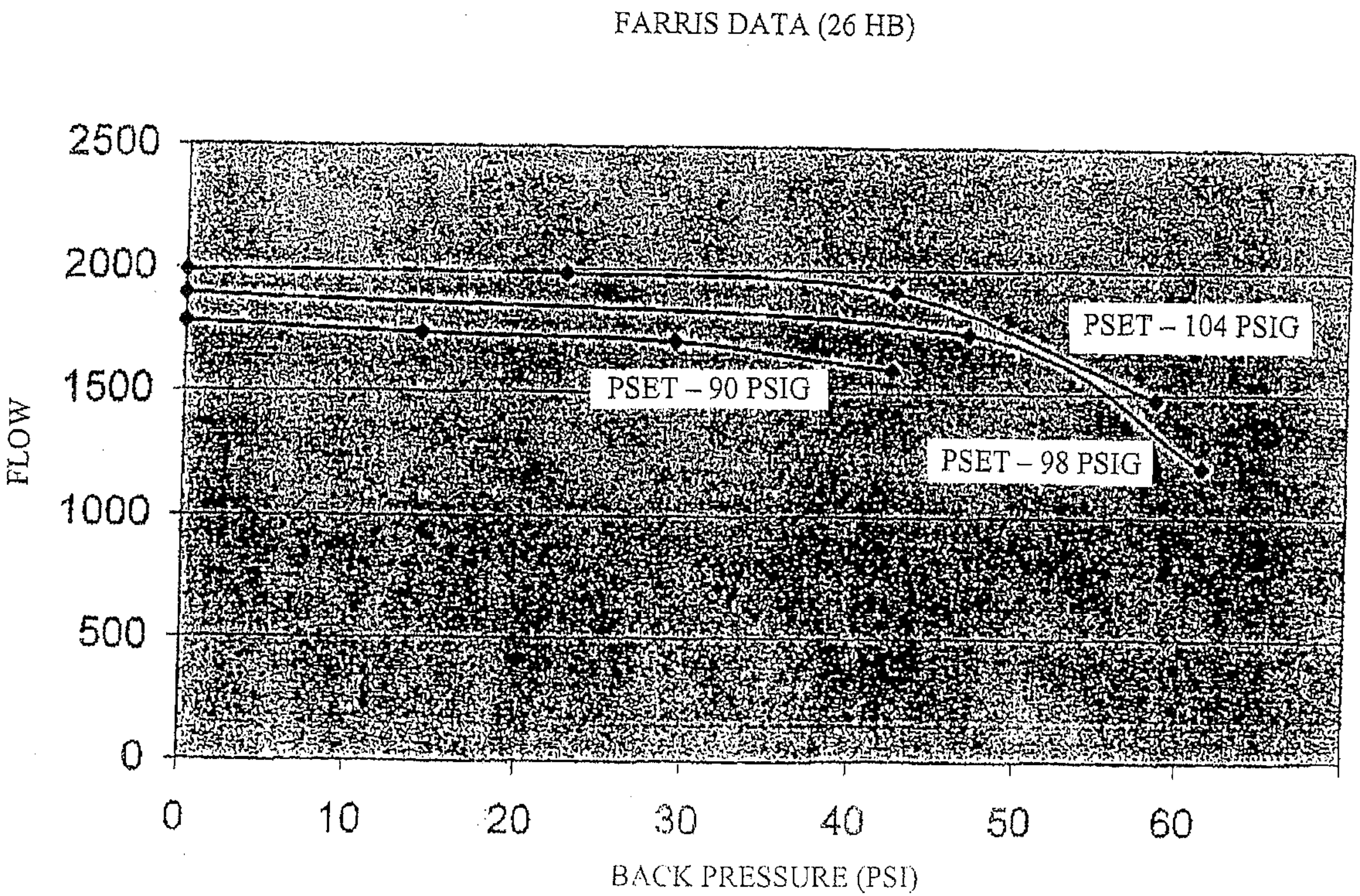
10 % OVERPRESSURE – 50 PSIG PSET

Manufacturers Kb curve Vs API Kb





FARRIS DATA (26 HB)
2500
2000
1500
1000
500
0
FLOW
0
20
40
60
80
100
BACK PRESSURE (%)
21 % OVERPRESSURE
16 % OVERPRESSURE
10 % OVERPRESSURE


FARRIS DATA (26 HB)
2500
2000
1500
1000
500
0
FLOW
0
10
20
30
40
50
60
BACK PRESSURE (PSI)
PSET – 104 PSIG
PSET – 90 PSIG
PSET – 98 PSIG

Summary

- Valve Tested Demonstrate Manufacturer's Selection for:
  - Flow at Backpressure Criteria or,
  - 7% Blowdown Maximum or,
  - "Constant" Set Point
- API 526 Spring loaded bellows PRV's do not meet all three criteria for a broad range of backpressure

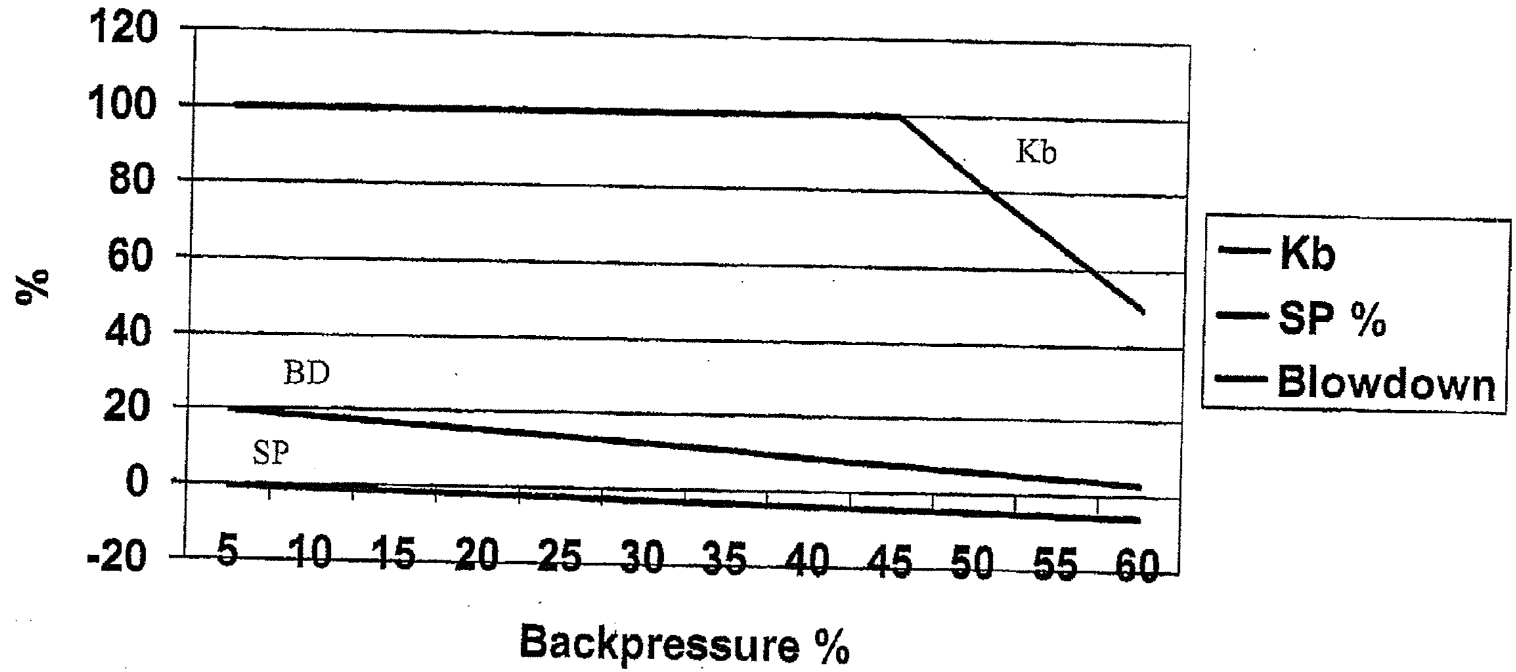
Bellows Valve Info

- Bellows Spring Rate
- Effective Area * Backpressure
  - Net Upward Force
- Effective Area +- 7% (Manufacturing)
- Design Effective Area
  - > 7% to prevent set point increase with increase in backpressure
- Backpressure reduces flow force on disc holder
- Body bowl pressure varies with valve size
  - Nozzle flow area vs. Outlet Area
  - Spring Rate Change vs Backpressure

BELLOWS SAFETY RELIEF VALVE – EFFECT OF BACK PRESSURE

ROGER DANZY DISCUSSED THE EFFECT OF VARYING BACK PRESSURE ON INTER-RELATED PARAMETERS OF SET PRESSURE, FLOW CAPACITY, AND BLOWDOWN FOR A BALANCED-BELLOWS PSV. HE NOTED THAT COMPRESSION OF THE BELLOWS DUE TO THE PRESENCE OF BACK PRESSURE ON THE VALVE TENDS TO GENERATE ADDITIONAL LIFT – REQUIRING SOME COMPENSATION OF THE SPRING SETTING. A CENTRAL THEME OF THIS PRESENTATION WAS THAT FOR A BELLOWS-TYPE VALVE, THE SET PRESSURE, BLOWDOWN, AND BACK PRESSURE CAPACITY CORRECTION ARE INTERDEPENDENT AND THAT EACH VARIES WITH THE BACK PRESSURE. ROGER NOTED THAT BECAUSE OF THIS INTERDEPENDENCE, OPTIMIZATION OF PERFORMANCE WITH RESPECT TO ONE CHARACTERISTIC (E.G., CAPACITY VARIATION WITH BACK PRESSURE) REQUIRES SACRIFICES WITH RESPECT TO ANOTHER CHARACTERISTIC (E.G., BLOWDOWN OR SET PRESSURE). ACCORDINGLY, HE EMPHASIZED THE IMPORTANCE OF PRECISE, THOROUGH COMMUNICATION BETWEEN THE USER AND THE MANUFACTURER OF PERFORMANCE REQUIREMENTS FOR BALANCED BELLOWS PRVS.

Bellows Performance - Built Up



Spring Rate for 60% Backpressure
Bellows Effective Area 10- 24% > Seat Area

Q. USING THE 1ST GRAPH (IN THIS CASE THE PRV CAPACITY IS MORE IMPORTANT THAN BLOWDOWN OR SET PRESSURE). IF I HAD A PRV PROTECTING A VESSEL WITH AN MAWP OF 100 PSIG AND I WANT THE PRV TO OPEN AT THE MAWP DURING THE DESIGN CASE AND THE PRV BUILT UP BACK PRESSURE DURING THE DESIGN CASE IS 60%, THEN AT ZERO BACK PRESSURE (BENCH TEST) WHAT WOULD MY SET POINT BE?

A. THE OPENING PRESSURE AT 0 BACK PRESSURE WOULD BE 7 -10% HIGHER THAN THE OPENING PRESSURE AT 60% BACK PRESSURE.

Q. AND WHAT WOULD THE BLOWDOWN BE?

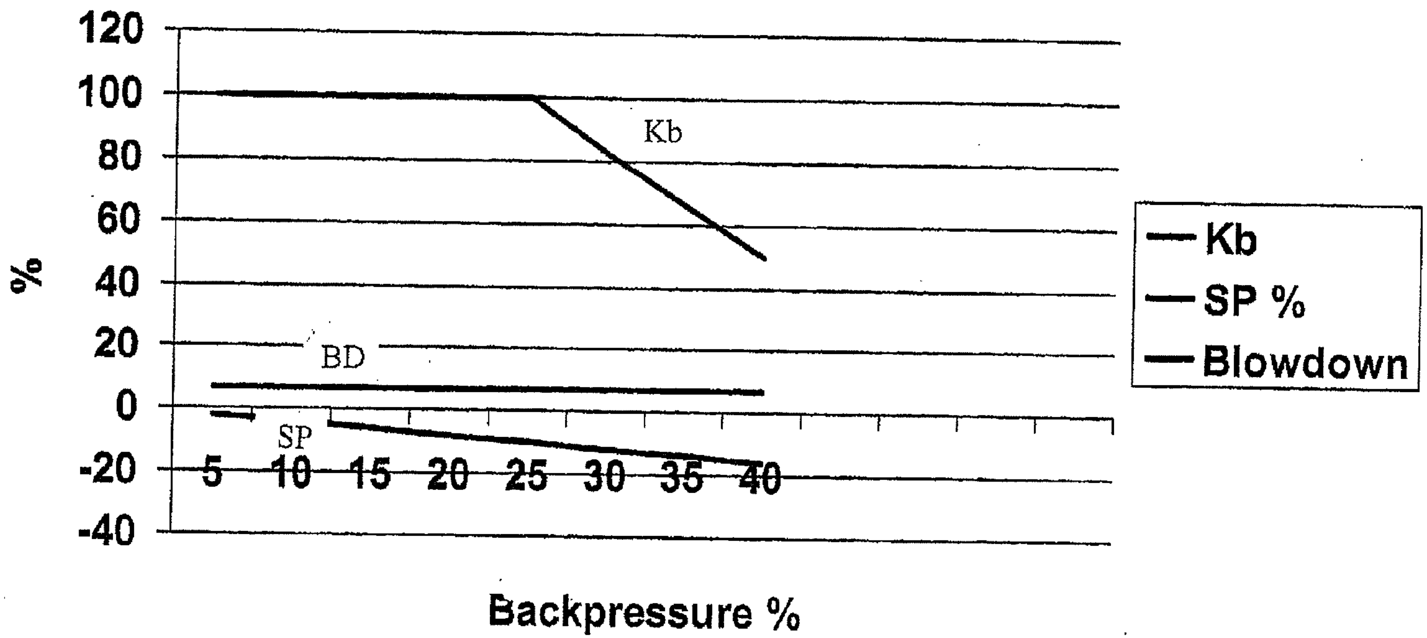
A. THE BLOWDOWN AT 0 BACK PRESSURE WOULD BE ABOUT 15% GREATER THAN THE BLOWDOWN AT 60% BACK PRESSURE. ASSUMING 7% BLOWDOWN AT 60% BACK PRESSURE, THE BLOWDOWN AT 0 BACKPRESSURE WOULD BE 22%.

Q. IF DURING OPERATIONS THE RELIEF DEVICE EXPERIENCES 60% BUILT UP BACKPRESSURE, WHAT WOULD THE VALVE BLOWDOWN BE AND AT WHAT PRESSURE WOULD THE PRV LIFT?

A. THIS IS OPPOSITE OF THE ABOVE EXPLANATION, ASSUMING THAT THE VALVE WAS BENCH SET WITH 0 BACK PRESSURE AND 7% BLOWDOWN, THE SET POINT WOULD BE 7 - 10% LESS AND THE BLOWDOWN WOULD BE ESSENTIALLY ZERO.

Bellows Performance - Constant Blowdown

"Theoretical"



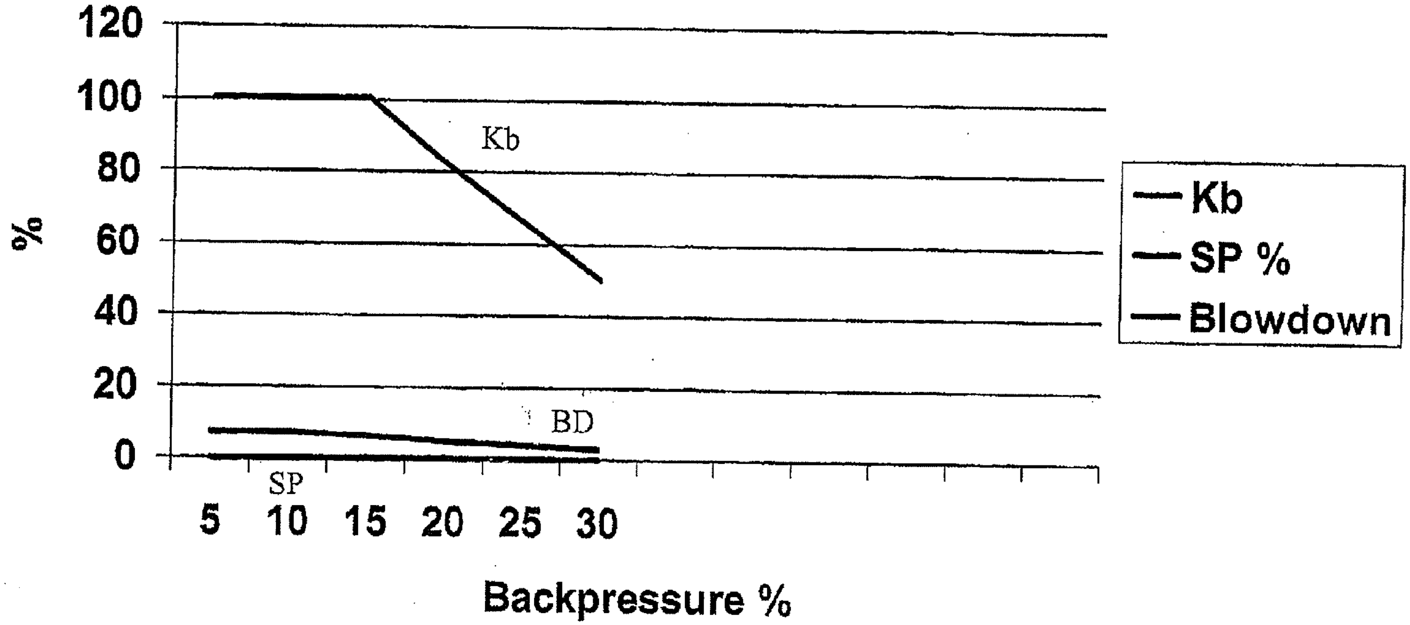
Spring Rate for 10% Backpressure
Bellows Effective Area for 40% Backpressure

Q. USING THE 2ND GRAPH (IN THIS CASE THE BLOWDOWN IS MORE IMPORTANT THAN SET PRESSURE OR PRV CAPACITY). IF I HAD A PRV PROTECTING A VESSEL WITH AN MAWP OF 100 PSIG AND THE PRV BUILT UP BACK PRESSURE IS 40%, THEN AT ZERO BACK PRESSURE (BENCH TEST) WHAT WOULD MY SET POINT BE? IF DURING OPERATIONS THE RELIEF DEVICE EXPERIENCES 40% BUILT UP BACK PRESSURE, AT WHAT PRESSURE WOULD THE PRV LIFT?

A. ASSUMING THAT THE VALVE OPENS AT 100 PSIG WITH 40% BACK PRESSURE, THE OPENING PRESSURE AT 0 BACK PRESSURE WOULD BE ESTIMATED TO BE 20 - 30 % HIGHER THAN AT 0 BACK PRESSURE.

Bellows Performance - "near" Constant Setpoint

"Theoretical"



Spring Rate for 10% Backpressure
Bellows Effective Area "equal to Seat Area"

Q. USING THE 3RD GRAPH (IN THIS CASE THE SET PRESSURE IS MORE IMPORTANT THAN BLOWDOWN OR PRV CAPACITY). IF I HAD A PRV PROTECTING A VESSEL WITH AN MAWP OF 100 PSIG AND THE PRV BUILT UP BACK PRESSURE IS 30%, THEN AT ZERO BACK PRESSURE (BENCH TEST) WHAT WOULD MY BLOWDOWN BE?

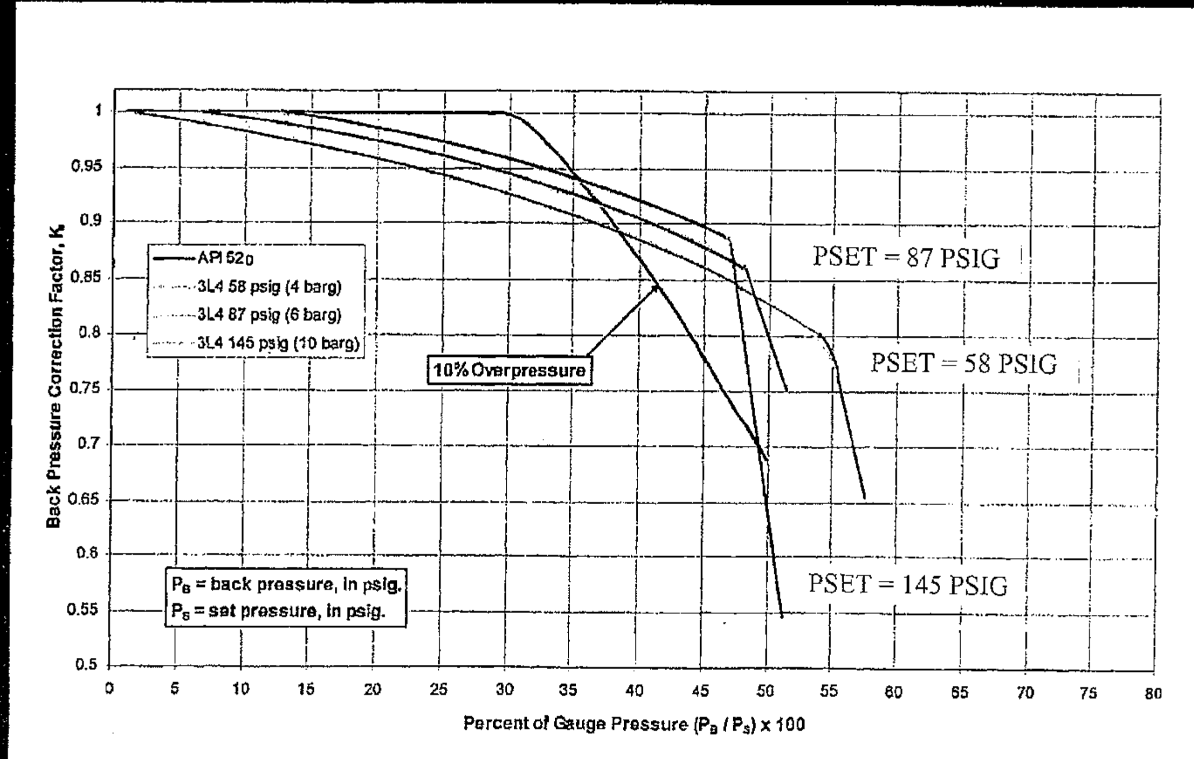
A. ASSUMING THAT THE VALVE HAS A BLOWDOWN OF 7% AT 30% BACK PRESSURE, THE BLOWDOWN AT 0 BACK PRESSURE WOULD BE IN THE 20 - 30% RANGE.

Q. IF DURING OPERATIONS THE RELIEF DEVICE EXPERIENCES 30% BUILT UP BACKPRESSURE, WHAT WOULD THE VALVE'S BLOWDOWN BE AND AT WHAT PRESSURE WOULD THE PRV LIFT?

A. THE BLOWDOWN WOULD BE 20 - 30% LESS THAN THE 0 BACK PRESSURE VALUE. THE VALVE WOULD LIFT AT ESSENTIALLY THE 0 BACK PRESSURE VALUE AS THE BELLOWS IN THEORY WOULD BE BALANCING THE EFFECTS OF BACK PRESSURE ON THE VALVE OPENING PRESSURE.

Back Pressure Correction Factor, K_b

for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures, κ = 1.4, Test results LESER 526 3L4



LESER

SRV GENERALIZED INLET AND DISCHARGE PIPING RULES

- **Enlarged inlet pipe diameter is almost always required for:**
  - » 4P6, 6R8, 6R10, and 8T10
  - » All safety valves used in series with rupture disks
  - » 1.5H3, 2J3, 3L4, and 6Q8 with shutoff valve and L/D=5
- **Enlarged outlet piping is almost always required for:**
  - » 6R8 safety valves
  - » Conventional safety valves 3L4, 4P6, and 8T10 with a set pressure > 100 psig and discharge pipe length more than 10 ft

SAFETY RELIEF VALVES

- ASME SECTION VIII
- API STANDARD SERIES 526 (ALL MFGS, MODELS, SIZES, FLOWS)
- GENERAL CHEMICAL SERVICE
- COMPRESSIBLE FLOW
  - VAPOR – Kd – REDBOOK (Kd = Kd' / 0.9)
    - IDEAL GAS
    - SUPERCRITICAL FLUID
  - TWO-PHASE FLOW
    - MODELS
      - ERM
      - HEQ
      - HFZ
      - SEQ
      - HNEQ
      - SUBCOOLED
    - Kd MODELS
      - FAUSKE
      - LEUNG
      - DARBY

SAFETY RELIEF VALVES

- BALANCED BELLOWS
- 7 % BLOWDOWN (STANDARD) DOSSENA DATA (CERTIFIED: 7 %)
  (MEASURED: 7.3, 6.7, 5.6, 5.3 %)
- NO FLOW MODULATION
- FULL LIFT
- OPENING CHARACTERISTIC (CAPACITY / LIFT vs. PRESSURE)
  - VALVE STABILITY OPENING
    CLOSING
- BACK PRESSURE (Kd – PSET GT 50 PSIG)
- ENGINEERING ANALYSIS

Dossena Performance Requirements

- API 526 2J3 - 300 x 150
- Set pressure 10 barg (145 psig)
- Overpressure 10%
- Blowdown < 7%
- Superimposed back pressure 0.2 - 3.5 barg (2.9 - 50.75 psig) (2 - 35%)
- Built-Up back pressure (max) 4.5 barg (65.25 psig) (45%)

API STANDARD 520, PART 2 – INSTALLATION

1. CONSIDER THE COMPLEXITY ASSOCIATED WITH MODIFYING THE 3 % RULE
2. RECOGNIZE AND ANALYZE SRV MULTIPLE STABILITY ISSUES
3. OBTAIN RAGAGEP / "CERTIFIED" DATA FROM SRV MANUFACTURERS
4. CONDUCT VALID ENGINEERING ANALYSES FOR SRV MODELS / FLOWS
5. PROPOSE "UNIVERSAL" / "LIMITED" MODIFICATIONS TO THE 3 % RULE

ioMosaic

Business Confidential Document

Pressure Relief Valve Stability

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

Joint US and European DIERS Users Group Meeting

June 2011

Hamburg, Germany

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

In 2003, Melhem and Fisher [1] published a detailed valve model to look at valve chatter dynamically and recommended that inlet pressure loss can be tolerated at blowdown – 2 % [2]

$$\frac{dx_v}{dt} = y$$

$$\frac{dy}{dt} = \left(\frac{1}{m}\right)\left(P_V A - k_s(x_0 + x_V) - P_B A_D + P_B(A_D - A) \pm F_r - b\frac{dx_V}{dt}\right)$$

$$\frac{dP_V}{dt} = \text{function of vessel and venting dynamics}$$

$$A = A_s + \left|\frac{x_v}{x_{v,\max}}\right|(A_D - A_s)$$

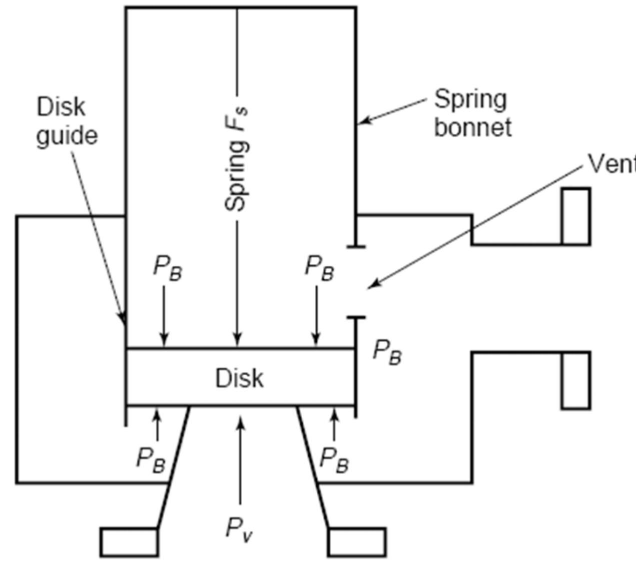
We do not need to couple this solution with vessel dynamics in order to screen for relief system stability

[1] G. A. Melhem and Harold G. Fisher, "Guidelines for dealing with excessive pressure drop in relief systems", DIERS Users Group, 2003

[2] Conditions apply including but not limited to: Vessel maximum pressure has to be adequate despite reduced flow capacity. Can tolerate excessive loss of product. Blowdown setting can be confirmed to +/- 2 %.

These equations can be used to estimate the opening/closing time of the relief device and can be integrated with vessel balances used to establish P_v as a function of time

- m is the spool mass plus 1/3 mass of spring
- b is the viscous damping coefficient
- The above equation can be solved numerically if a suitable value for dA_N/dt is provided for valve opening and closing
- This is difficult because the pressure profile on the disk surface is non-uniform



$$P_V A = k_s(x_0 + x_V) + P_B A_D - P_B(A_D - A) \pm F_r + m\frac{d}{dt}\left(\frac{dx_V}{dt}\right) + b\frac{dx_V}{dt}$$

$$\frac{dA}{dt} = f(x_V, P_V, P_B, \ldots) \text{ at } \mathrm{t} = 0, A = A_N, \text{at } \mathrm{t} = \mathrm{t}_{\text{fully open}}, A = A_D$$

To accurately assess whether a "relief system" will operate in a stable manner (chatter free) you must consider the following important system time constants and how they interact

- Valve time constant
  - How fast does a pressure relief device close and open?
- Vessel or pressure source time constant
  - How fast does a vessel de-pressure and re-pressure after a pressure relief device opens and re-seats?
- Inlet line time constant
  - How long does it take for a pressure wave to propagate upstream from a pressure relief device to the pressure source and back?
- Outlet line time constant
  - Acoustic barriers may be established due to "body bowl choking"
  - Note that acoustic barriers, such as the presence of control valves, change in diameter, etc. can cause standing waves that can lead to acoustic coupling or resonance with relief systems components

Why is chatter important ?

- Can cause severe damage to upstream equipment
  - Especially important for liquid flow
- Can cause severe damage to valve components due to large forces
  - Especially important for large valves
  - Especially important for high set pressures
- Can ultimately cause loss of containment leading to fire, explosion, and toxicity risks

Any mechanism that causes the valve to close rapidly can lead to chatter

- Excessive inlet pressure loss, well in excess of blowdown
- Excessive backpressure
- Oversized valve

ioMosaic

Chatter causes damage in relief systems components primarily due to a "water hammer" effect caused by the rapid closure of the pressure relief device

- Flow is suddenly interrupted causing a large pressure spike to occur at the valve
- If the pressure spike is not large enough to rupture a relief system component, the compression wave created will reverse and travel back towards the pressure source (vessel or another acoustic barrier)
- It will then reverse again and travel back to the pressure relief device which caused the pressure spike in the first place
- The wave will continue to reverberate until something breaks or until its energy is absorbed by the relief system damping
- Even if nothing fails initially, the relief system will be subjected to repeated stresses that can cause fatigue failure
- This is most dangerous for liquid flow in rigid piping – if the liquid was truly incompressible the pressure spike would be infinite!

ioMosaic

The magnitude of the pressure spike generated can be calculated using the following equation

$$P_{\max} = P_o + \rho_o u_{\max}\left[u_o - u_c\right]$$

Where:

P_{max} = *Maximum pressure spike*

P_o = *Initial pressure*

u_{max} = *Acoustic velocity (depends on fluid phase and piping support properties)*

u_o = *Initial flow velocity*

u_c = *Flow velocity after valve closure (0 for complete closure)*

ioMosaic

The pressure spike caused by rapid valve closure due to chatter is much more dangerous for liquid flow and least dangerous for flashing liquid flow

$$P_{\max} = P_o + \rho_o u_{\max}\left[u_o - u_c\right]$$

Example – Liquid, water like flow at 20 m/s

$P_{max} = P_o + 1000*2000*[20-0] = P_o + 400$ *bars!*

Example – Gas, air like flow

$P_{max} = P_o + 5*350*[350-0] = P_o + 6$ *bars*

Example – Two phase, water like, high void fraction, flashing flow

$P_{max} = P_o + 5*30*[30-0] = P_o + 0.045$ *bars*

ioMosaic

The magnitude of the peak pressure depends strongly on the value of acoustic velocity

- Note that the value of the acoustic velocity also determines the length of time that corresponds to an "*instantaneous*" change in valve position

- This is because as long as it takes the valve less time to close than it takes the pressure wave to travel upstream to the source and then back to the valve, the impact on the system is exactly the same as if the valve closed instantaneously

$$t_c < 2\frac{L}{u_{\max}}$$

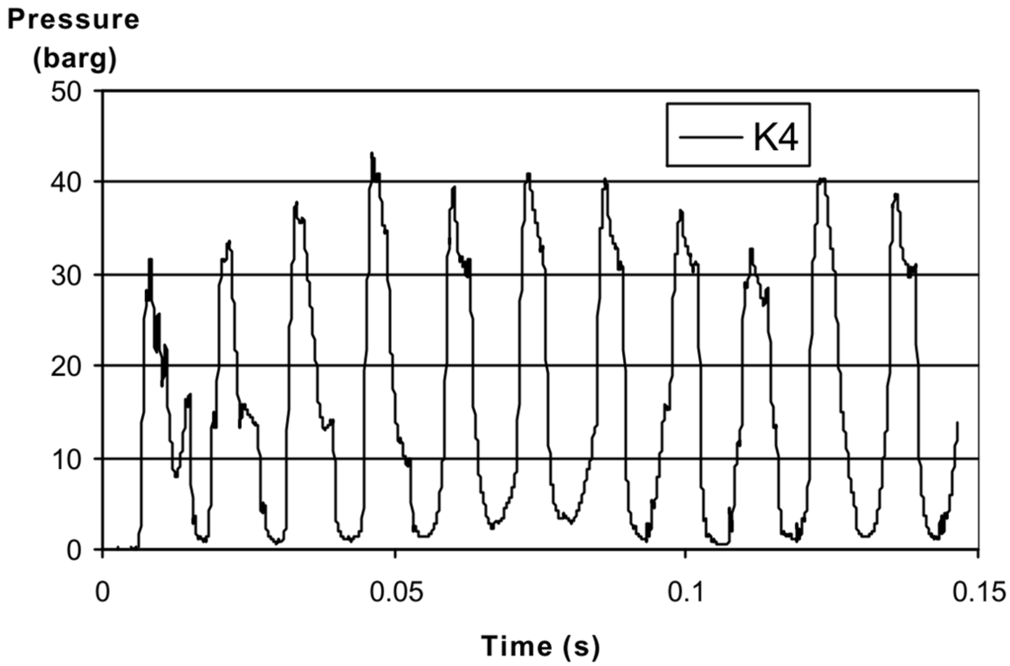
- As long as the valve closes completely in less than t_c, the impact on the relief system will be the same regardless of how quickly the valve actually closes

- Note that $f = u_{max} / 2L$ is essentially the vortex shedding frequency of the relief device

ioMosaic

In order to avoid the "water hammer" type pressures, especially for liquid flow, the valve closure time must be larger than $2L/u_{max}$, or the inlet line length must be short enough

- In a recent API 521 committee meeting, Shell presented data that also confirms based on their experience that the predominant PRV instability risk is with liquid reliefs (see Brad Otis presentation) [1]



Liquid PRV showing chatter behavior [2]

Opening time	*= 4 ms*
Closing time	*= 8.25 ms*
P_{set}	*= 15 barg*

[1] "A Review of PRV Chatter Incidents", API 520 Subcommittee meeting, Brad Otis, May 18, 2011

[2] "Examination of the effect of relief device opening times on the transient pressures developed within liquid filled shells", HSE Offshore Technology Report 2000/130

ioMosaic

The time constant of the pressure source is also very important in determining whether valve chatter or valve cycling is occurring

- To simplify the discussion, let's assume the pressure relief device inlet line length is negligible. Let's assume the cause of valve chatter is excessive backpressure or oversized valve

- We can calculate using dynamic simulation, t_{rep}, the time required for pressure to rise in a vessel (small or large) from the relief device P_{reset} to P_{set} and then to $1.1*P_{set}$

- If t_{rep} is much less than the valve closure time, then the valve will remain partially or fully open

- If t_{rep} is much longer than the valve closure time, then the valve will just cycle and not chatter
  - We should be able to calculate if the sudden valve closure peak pressures can cause damage to the relief system, i.e. when $t_c < 2L/u_{max}$

- If t_{rep} is equal to the valve closure time, chatter can occur and possibly resonance. This is a condition to be avoided

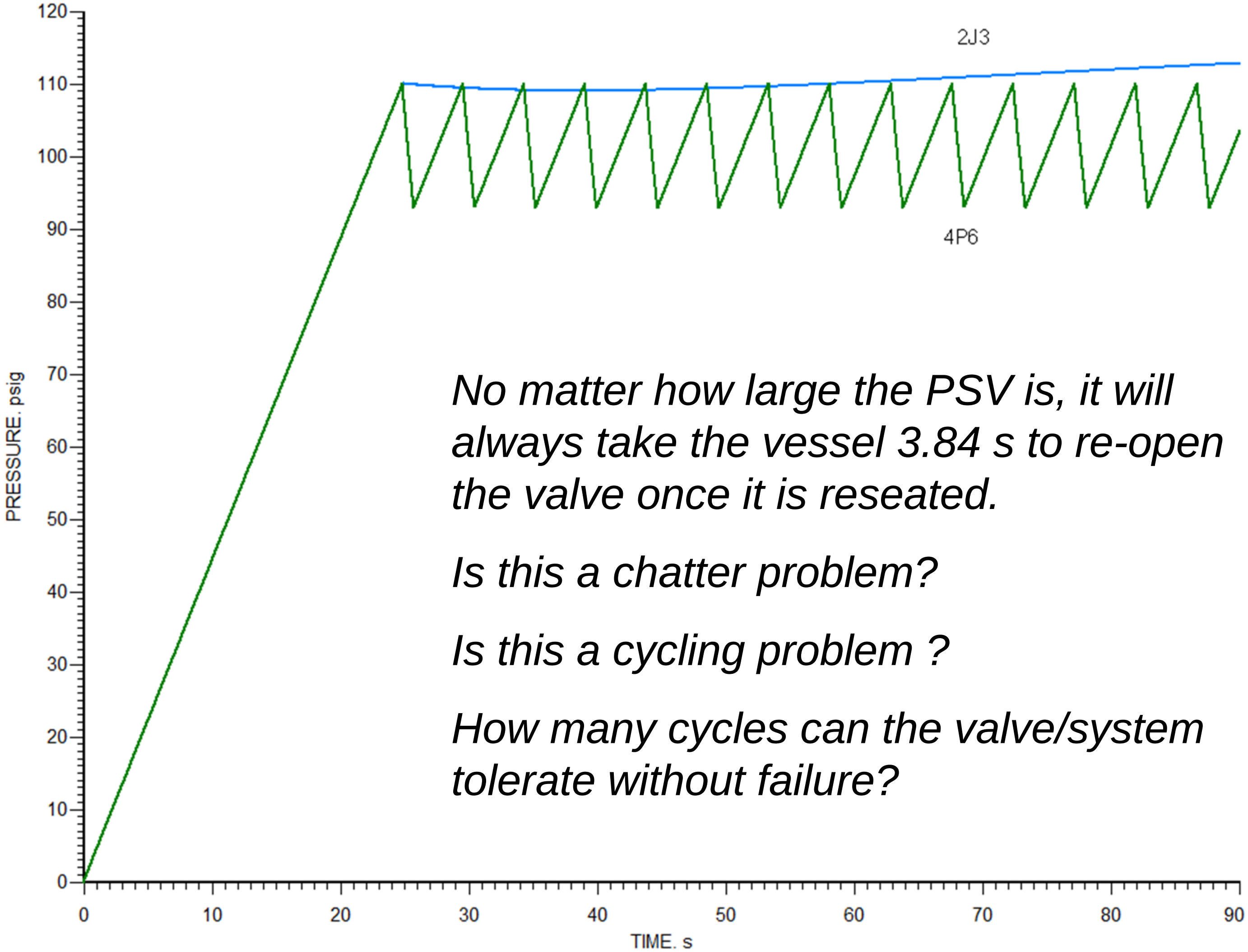
ioMosaic

Let's consider a gas dynamics of a finite pressure reservoir to illustrate the "time constant" concepts

- 10 m^3 vessel equipped with a PSV. Initially at 25 C and 0 psig
- Assume inlet line length L is very short, i.e. L << u_{max}*t_c/2 and does not have acoustic barriers
- Assume discharge line is properly sized, i.e. backpressure is < 10 % of P_{set}
- Nitrogen source is connected to vessel via a control valve (1 inch), Cd = 0.62, 300 psig Nitrogen, at 25 C
- Maximum PSV closing time is 10 ms
- Set pressure of PSV is 100 psig and reset pressure is 93 psig
- Does a 2J3 chatter ?
- Does a 4P6 chatter ?

ioMosaic

Both valves are operating stably and without chatter. Note the time to re-pressure is 3.84 seconds which is much longer than 10 ms. Is the 4P6 cycling a problem?



No matter how large the PSV is, it will always take the vessel 3.84 s to re-open the valve once it is reseated.

Is this a chatter problem?

Is this a cycling problem ?

How many cycles can the valve/system tolerate without failure?

Source: SuperChems Expert v6.3mp

ioMosaic

Let's further extend the previous simulations to take a more detailed at the 2J3 relief line setup

- Assume inlet line is vertical and has a ½ inlet velocity head loss but no other fittings
- The dynamics show that the valve will be first fully open at 24.85 seconds and 110 psig with an inlet temperature of 296.44 C
- The discharge line is a 1 m, 3 inch SCHD 40, no fittings
- Assume the inlet line is rigid
- Note, the value of u_{max} depends strongly on the fluid properties, how the pipe is supported, and should be calculated using a detailed equation of state to account for real fluid effects
- Let's vary the inlet line length and then calculate the associated inlet pressure loss and $2L/u_{max}$

ioMosaic

The % inlet pressure loss required for a stable 2J3 operation for our example can be as high as 4% for a valve with a closing time of 10 ms and 6 % for a valve with a closure time of 16.5 ms

Inlet Line Length, m	u_{max}, m/s	$t_c=2L/u_{max}$, ms	% inlet pressure loss	% Back pressure	Flow rate, kg/s
0.5	485	2.06	2.15	1.84	1.009
1.0	485	4.12	2.86	1.80	1.004
1.5	485	6.18	3.26	1.79	1.000
2.0	485	8.24	3.81	1.78	0.995
3.0	485	12.3	4.88	1.76	0.987
4.0	485	16.49	5.92	1.67	0.978

Source: SuperChems Expert v6.3mp

ioMosaic

In 1980, Cox and Weirick published a simple correlation for calculation of inlet pressure loss (as long as the pressure loss is reasonable small, say < 10 %) in Chemical Engineering Progress

$$\Delta P(\%) = 1.73\times 10^{-5}\,\frac{\left[0.5 + K_F + 4\dfrac{fL}{D}\right]\left(\dfrac{F}{0.9}\right)^2}{A_N^2 \rho P_s}$$

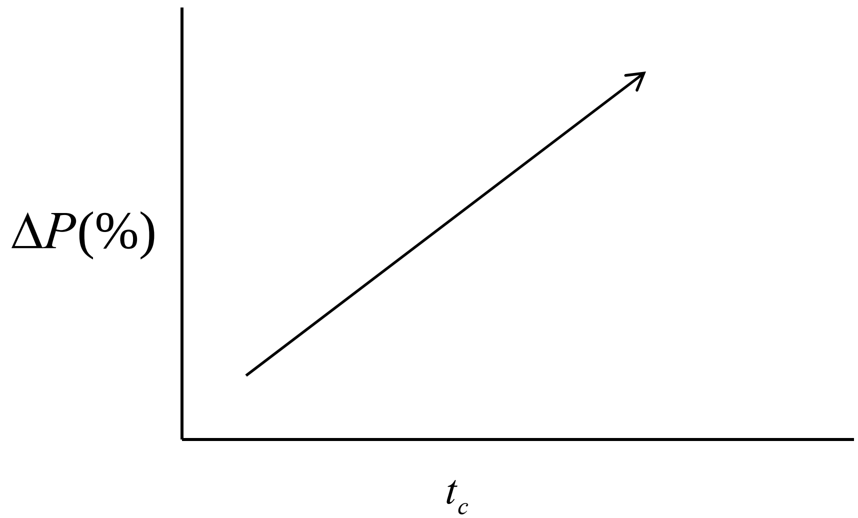
$$L_{\max} = \frac{u_{\max} t_c}{2}$$

$$\Delta P(\%) = 1.73\times 10^{-5}\,\frac{\left[0.5 + K_F + 2\dfrac{f u_{\max} t_c}{D}\right]\left(\dfrac{F}{0.9}\right)^2}{A_N^2 \rho P_s}$$

A_N is the PRV flow area in in^2
F is the rated capacity in lbs/hr
ρ is the flowing density in lb/ft^3
L is the inlet pipe length
D is the inlet pipe diameter
f is the Fanning friction factor
K_F is the overall velocity head loss
0.5 velocity head loss to account for inlet pipe nozzle
P_s is the set pressure in psig

This clearly shows that a linear relationship should exist between t_c and allowable % inlet pressure loss for stable valve operation

Slower valves should tolerate longer inlet lines and more inlet pressure loss



ioMosaic

Let's consider the other limiting case of an infinite pressure reservoir with a relief device attached to it

- Assume time to re-pressure the inlet line is infinitely small since the reservoir is infinitely large compared to the flow capacity of the relief system

- If inlet line length L is < $u_{max}*t_c/2$, the relief valve is expected to behave in a stable manner no matter what the cause of valve closure is

- If inlet line length L is > $u_{max}*t_c/2$, the relief valve is expected to behave in an unstable manner and corrective action is required
  - Excessive Inlet Pressure Loss – Restrict loss to blowdown – 2 %
  - Excessive Backpressure – Enlarge piping to meet manufacturer recommended backpressure limits, change valve type, etc.
  - Oversized valve – Use a modulating relief device, use multiple valves, restrict valve lift, etc.

As long as the inlet line length, $L < u_{max} * t_c / 2$, and the dynamic simulation time step is less then the valve closing time, dynamic simulations should be able to show if the relief system will chatter or not

- In case of rapid cycling one should confirm that the valve components can take the closing force and if the piping supports can take the reaction forces
- The relief systems piping should also be checked for vibration risk (see previous presentation)
- Always check for resonance potential
- Note that two-phase flashing flow requires shorter inlet line lengths than gas flow and than liquid flow (for the same valve closing time) for stable valve operation

A valve will not chatter if stable valve opening is supported by the returning compression wave produced by the reflection of the initial expansion wave

$$L_{\max} < \frac{u_{\max} t_c}{2} \text{ where } u_{\max} = \frac{u_{sonic}}{\sqrt{1 + \frac{K}{E}\psi}}$$

Example – Liquid, water like flow with t_c = 10 ms

L_{max} = 2000 * 0.01 / 2 = 10 m

Example – Gas, air like flow with t_c = 10 ms

L_{max} = 350 * 0.01 / 2 = 1.75 m

Example – Two phase, water like, high void fraction, flashing flow, with t_c = 10 ms

L_{max} = 35 * 0.01 / 2 = 0.175 m

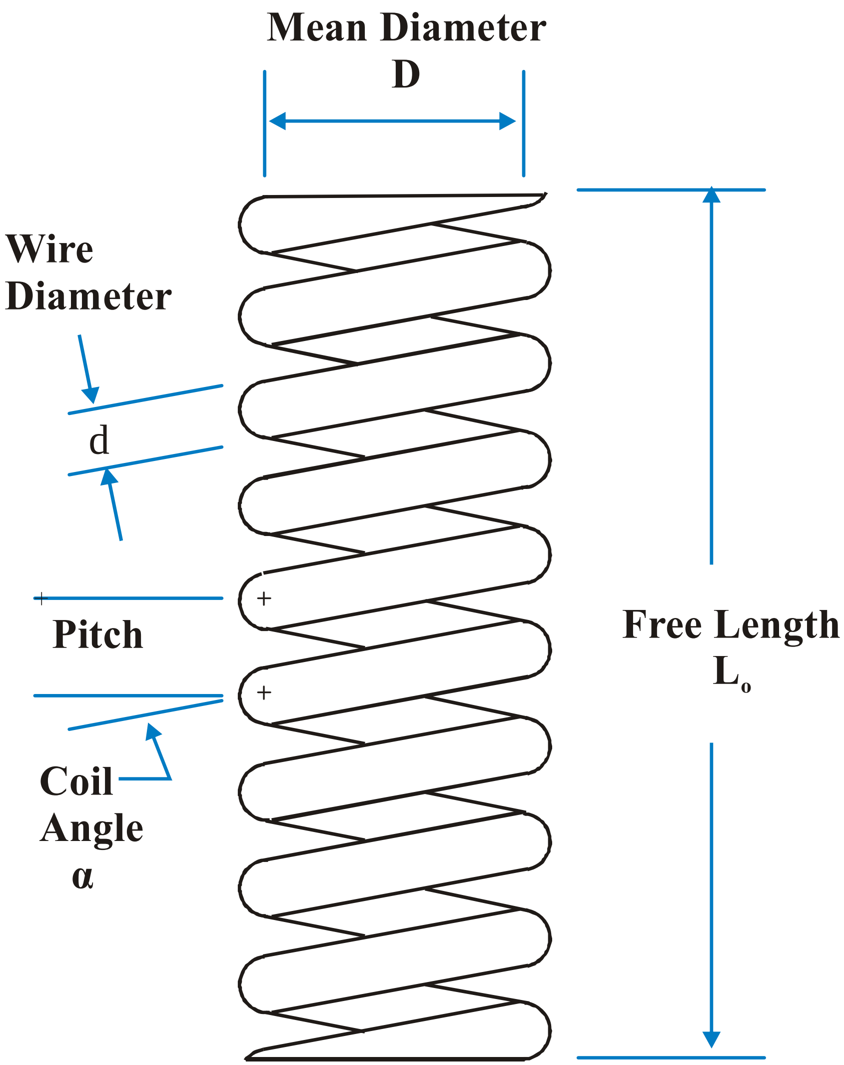
ioMosaic

How do we obtain the valve frequency, opening, and closing time?

- Provided by manufacturer (that would be nice!)
- Measure it
- Use a detailed mathematical model such as the one published by Melhem and Fisher
  - Requires spring constant and mass of moving internals to be provided by manufacturers
- As outlined in the next few slides, use the models developed by Mike Grolmes (several presentations to the DIERS Users Group) to calculate valve frequency, f_n, and closing time $t_c = 1 / 2 f_n$

ioMosaic

The spring constant can be calculated for a helical spring using the following equation



$$k = \frac{G\,d}{8\,C^3\,N_a}$$

N_a is the number of active coils

G is the modulus of torsion or rigidity

C is the diameter modulus; C = D/d

D is the mean diameter

d is the coil diameter

ioMosaic

Material properties are available in the literature

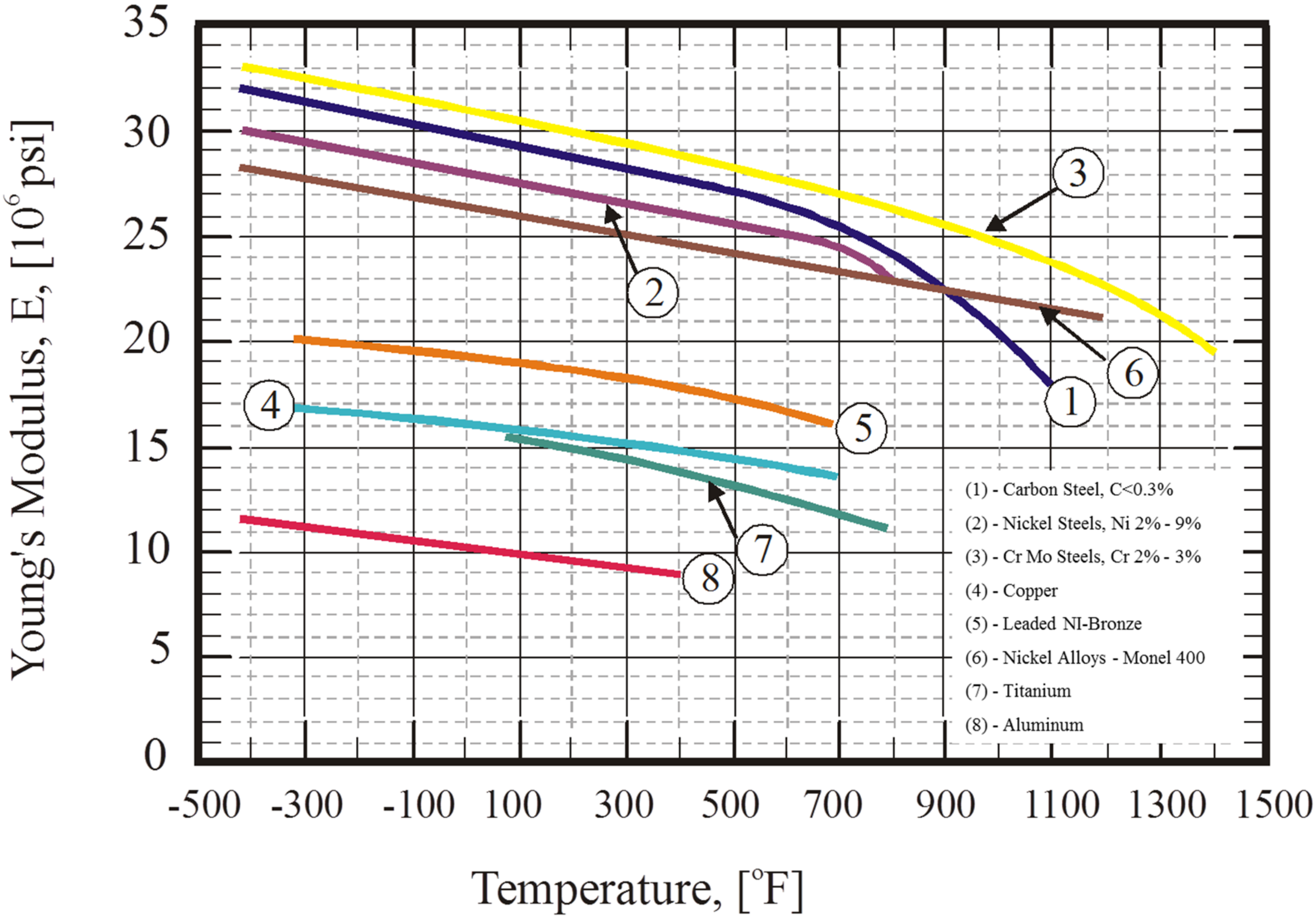
Material	E (GPa)	Poisson's Ratio ν	$K = \frac{1}{\kappa}$ (GPa)	ρ in kg/m3
Aluminum	69	0.33		
Brass	78-110	0.36		
Carbon steel	202	0.303		
Cast iron	90-160	0.25		
Concrete	20-30	0.15		
Copper	117	0.36		
Ductile iron	172	0.30		
Fibre cement	24	0.17		
High carbon steel	210	0.295		
Inconel	214	0.29		
Mild steel	200-212	0.27		
Nickel steel	213	0.31		
Plastic / Perspex	6.0	0.33		
Plastic / Polyethylene	0.8	0.46		
Plastic / PVC rigid	2.4-2.75			
Stainless steel 18-8	201	0.30		
Water - fresh			2.19	999 at 20 C
Water - sea			2.27	1025 at 15 C

E is typically referred to as Young's modulus of elasticity

G is typically referred to as modulus of torsion, $G = \frac{1}{2}\frac{E}{1+\nu}$

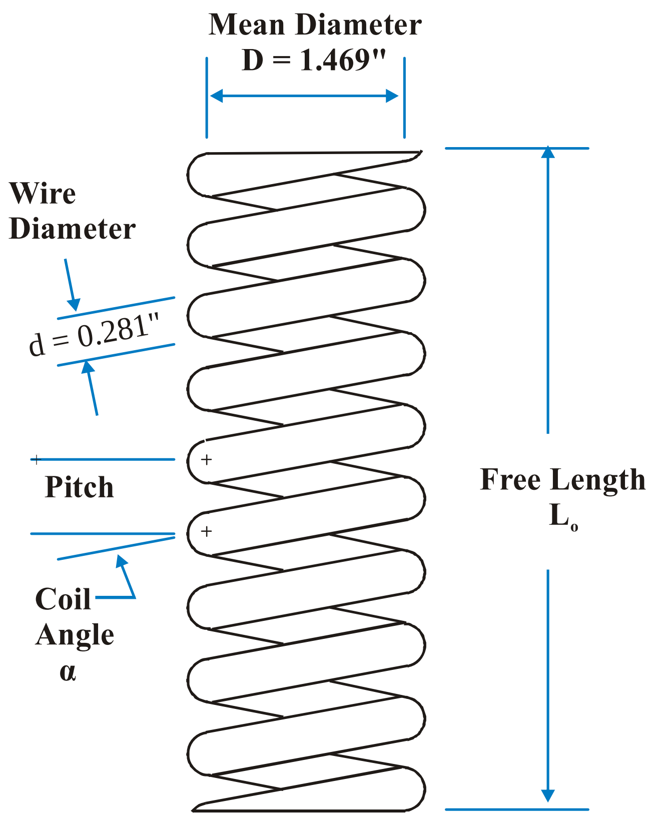
ioMosaic

Note that the valve material properties change with temperature



ioMosaic

An example from Mike Grolmes DIERS presentations



Spring from
Farris 26 FA 10 - 120
180 psig set

$N_a = 8$

$C = D/d = 5.288$

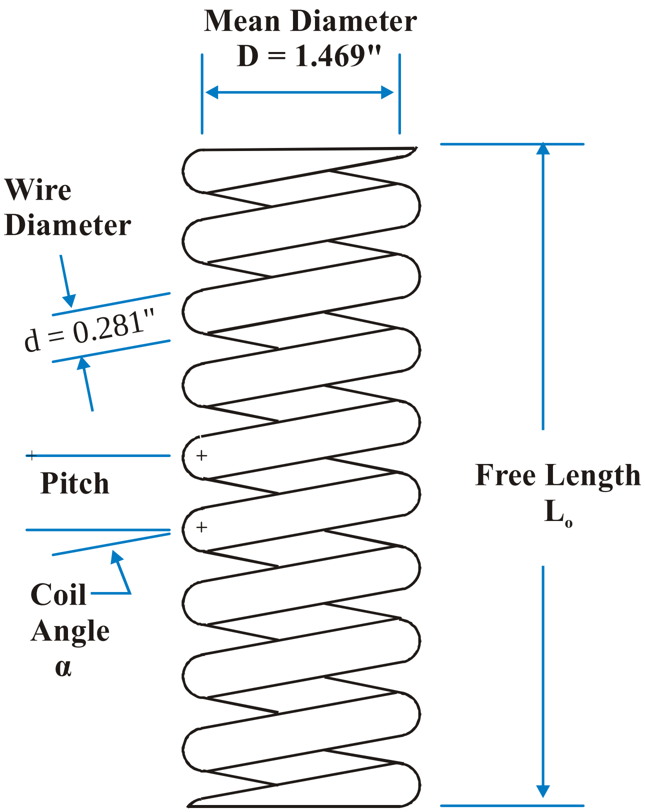
Assume:

$E = 205$ GPa or 29.7×10^6 psi

$\nu = 0.31$

$G = 78.2$ GPa or 11.35×10^6 psi

An example from Mike Grolmes DIERS presentations (continued)



Spring from
Farris 26 FA 10 - 120
180 psig set

Wgt. of Spring = 360 gm
Wgt. of other
parts in motion = 435 gm
Assume mass in motion is
435gm + 360 / 3 = 555 gm
or approximately 1.22 lb

ioMosaic

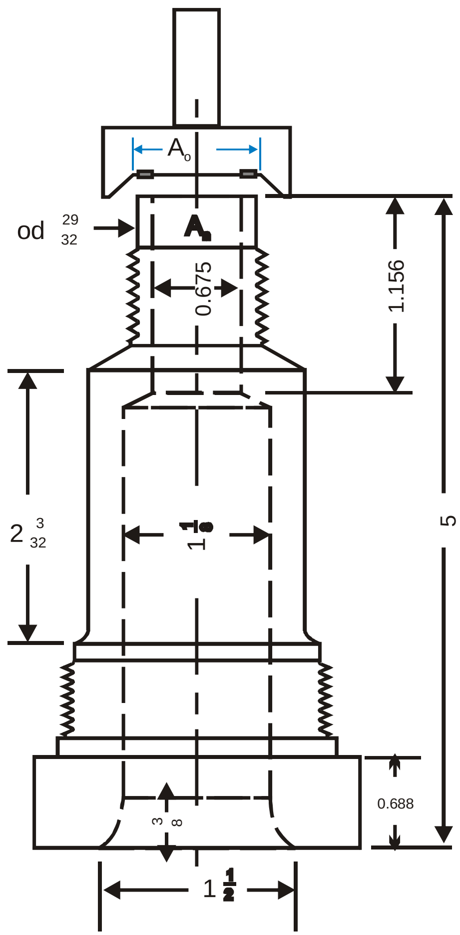
The fastest closing time can then be calculated to be 9.43 ms

$$k = \frac{11.35E6\,x\,0.281}{8\,x\,5.288^3\,x\,8} = 349\,\frac{lbf}{in}\,\text{or } 61294\ \text{N/m}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{61294}{0.555}} = 53\,\text{Hz}$$

$$t_c = \frac{1}{2f_n} = 9.43\ \text{ms}$$

Grolmes also provided a simple correlation to calculate k and m from available valve data based on actual measurements he has conducted on numerous real valves up to 6Q8



$$k = 1.08 \frac{P_{set} \; A_{redbook}}{y_{lift}}$$

$$m = 0.025 \, M_{valve}$$

$$t_c \approx \frac{1}{2 f_n} = \pi \sqrt{\frac{m}{k}} = 0.478 \sqrt{\frac{M_{valve} y_{lift}}{P_{set} A_{redbook}}}$$

M_{valve} $\uparrow$ ➔ $t_c \uparrow$

y_{lift} $\uparrow$ ➔ $t_c \uparrow$

P_{set} $\downarrow$ ➔ $t_c \uparrow$

$A_{redbook}$ $\downarrow$ ➔ $t_c \uparrow$

Use consistent units – Pa gauge, s, kg, m, and m²

ioMosaic

The most stable relief device is one that is "slightly undersized"

- Even with gas or two-phase chatter, a system might actually survive, especially if the inlet line length is < $2L/u_{max}$, regardless of how much pressure drop or what caused the valve to close

- Inlet pressure loss ONLY is not a sufficient criterion to guarantee valve stability

- Relief dynamics are required to determine relief system stability for oversized valves

- Inlet line length can be restricted to ensure stable valve operation even when the inlet pressure drop is excessive

- Inlet line length limits required for stable valve operation (Shortest to Longest)
  - Flashing Two Phase → Gas → Liquid

- Relief systems damage risk from chatter (High to Low)
  - Liquid → Gas → Flashing Two Phase

ioMosaic

If relief device manufacturers can supply additional information, we can better resolve the issue of chatter and relief systems performance

- Valve opening pressure and valve reset pressure (accurate blowdown +/- 2 %)

- Opening and closing time or valve natural frequency or time constant of the spring mass part of the system

- Cycles to failure for all valves, including but not limited to:
  - Seizing either fully open or fully closed
  - Performance failure, fails to deliver rated flow
  - Deformation such that the valve is no longer able to be considered leak tight after the actuation event causing the damage

- One would expect that small valves and/or valves with low set pressures might tolerate several million on-off cycles before failure while larger valves or valves with high set pressures might only tolerate tens of thousands cycles before failure[1]

[1] Personal communication with Mike Grolmes

ioMosaic

The following conclusions can be made based on our current understanding of important and relief systems time constants for an infinite pressure reservoir

System Characteristic	Criteria	Cause of Chatter	Potential Hazards and Possible Mitigation Strategies
Infinite Pressure Reservoir Capacity of reservoir is much larger than the capacity of the relief system	$L < u_{max} * t_c / 2$	Excessive Inlet Pressure Loss	Valve is expected to operate in a stable manner
		Excessive Backpressure	Valve is expected to operate in a stable manner
		Oversized Relief Device	Valve is expected to operate in a stable manner
	$L > u_{max} * t_c / 2$	Excessive Inlet Pressure Loss	Restrict inlet pressure loss to blowdown – 2 % Remove acoustic barriers
		Excessive Backpressure	Restrict backpressure or enlarge discharge piping Change valve type
		Oversized Relief Device	Use a modulating relief device Restrict valve lift Dampen valve, slow down closure Etc.

iоMosaic

The following conclusions can be made based on our current understanding of important and relief systems time constants for a finite pressure reservoir

System Characteristic	Criteria	Cause of Chatter	Potential Hazards and Possible Mitigation Strategies
Finite Pressure Reservoir System capacity is small enough for blowdown to occur within a reasonable time frame	$L < u_{max} * t_c / 2$	Excessive Inlet Pressure Loss	Check cycling frequency using dynamic simulation to ensure stable system behavior
		Excessive Backpressure	Check cycling frequency using dynamic simulation to ensure stable system behavior
		Oversized Relief Device	Check cycling frequency using dynamic simulation to ensure stable system behavior
	$L > u_{max} * t_c / 2$	Excessive Inlet Pressure Loss	Restrict inlet pressure loss to blowdown – 2 % and check cycling frequency using dynamic simulation
		Excessive Backpressure	Restrict backpressure or enlarge discharge piping, change valve type, etc.
		Oversized Relief Device	Check cycling frequency using dynamic simulation, use multiple valves, etc.

ioMosaic

Related Presentations

- G. A. Melhem and Harold G. Fisher, “Guidelines for dealing with excessive pressure drop in relief systems”, DIERS Users Group, 2003
- G. A. Melhem, “Deal with controversial topics in pressure relief systems”, DIERS Users Group, Spring 2009, Orlando, Florida.
- G. A. Melhem, “Estimate acoustically induced vibration risk in relief and flare piping”, Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

LESER
Safety Relief Valve Stability

Experimental and Numerical Research on the Influence of Back Pressure

B. Jörgensen, D. Moncalvo

The-Safety-Valve.com

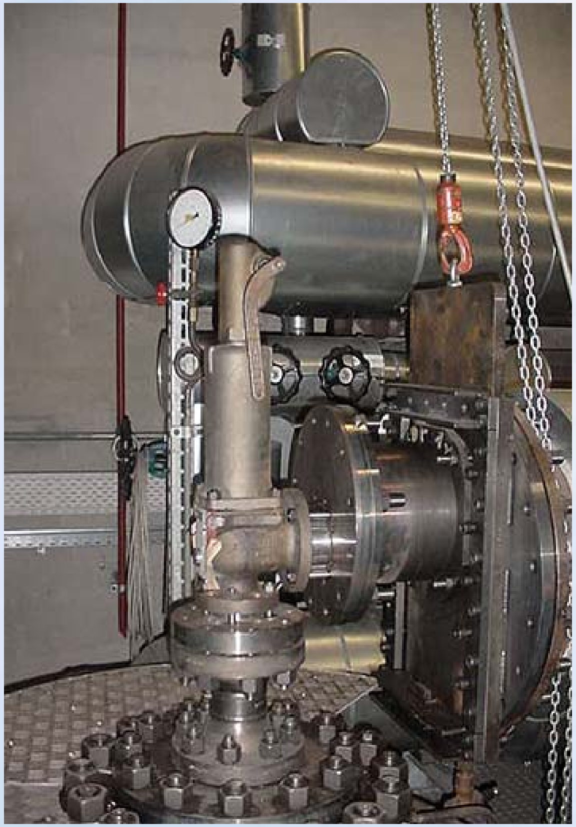
Objectives of this Presentation

Knowledge to learn

- Experimental research at LESER
  - – Test facility
  - – Experimental results
- Numerical research / CFD
  - – CFD in the calculation of mass flow
  - – Fluid Force
  - – Opening of SRV
- Pros and Cons of Experimental and CFD Methods

ASME and TÜV certified Test Facility at LESER

ASME (pending) and TÜV certified Steam Test Facility of LESER

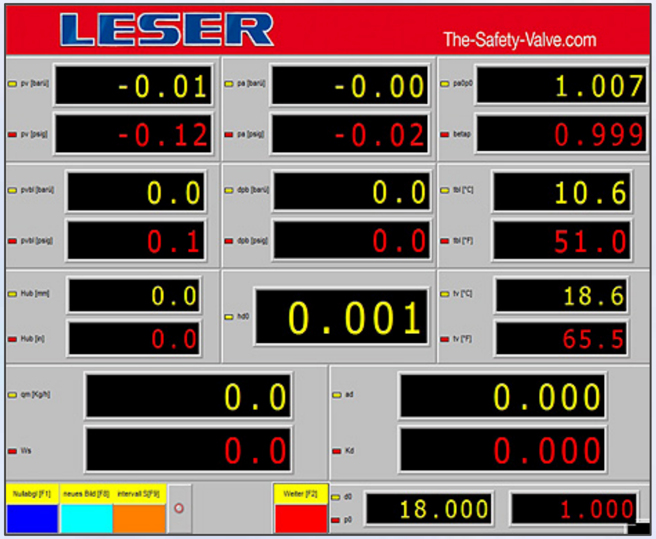


Objectives
ASME / TÜV certified Test Facility
Data Acquisition
Test Lab Capacities and Pressure Ranges
Product Data and Operating Conditions
Tests
Numerical Tools
Calculation of Mass Flow Rates
Fluid Force acting on disk
Transient Calculation of Flow
Opening with atmospheric Back Pressure
Experimental and Numerical Estimation

Data Acquisition

Data Acquisition:

- 24 channel system (Mass flow, Pressures, Temperatures, Lift)
- 50 Hz to 1 KHz Frequency of Recording
- Data recorded on local PC and copied in server
- Predetermined and freely programmable measurement routines



Test Lab Capacities and Pressure Ranges

Media	Pressure	Flow Capacity
Air	0 – 100 barg *0 – 1450 psig*	10 – 70000 kg/h *4,8 – 33600 scfm* (Performance Test) 10 – 52000 kg/h *4,8 – 24964 scfm* (Capacity Test)
Water	0 – 42 barg *0 – 610 psig*	2 – 240 m^3/h *8,8 – 1057 gpm*
Steam	0 – 42 barg *0 – 610 psig*	5 – 24000 kg/h *11 – 52910 lb/h*

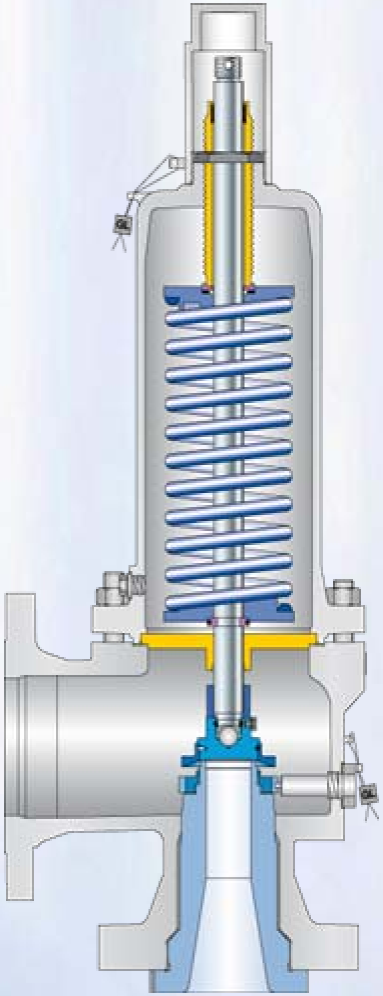
Capacities and Pressures from Test Lab at National Board

Air	0 - 500 psig	0 – 13000 scfm
Water	0 – 500 psig	0 – 550 gpm
Steam	0 – 500 psig	0 – 16000 lbs/h

Product Data of LESER SV Type 526 2H3 and Operating Conditions of Stability Test on Air

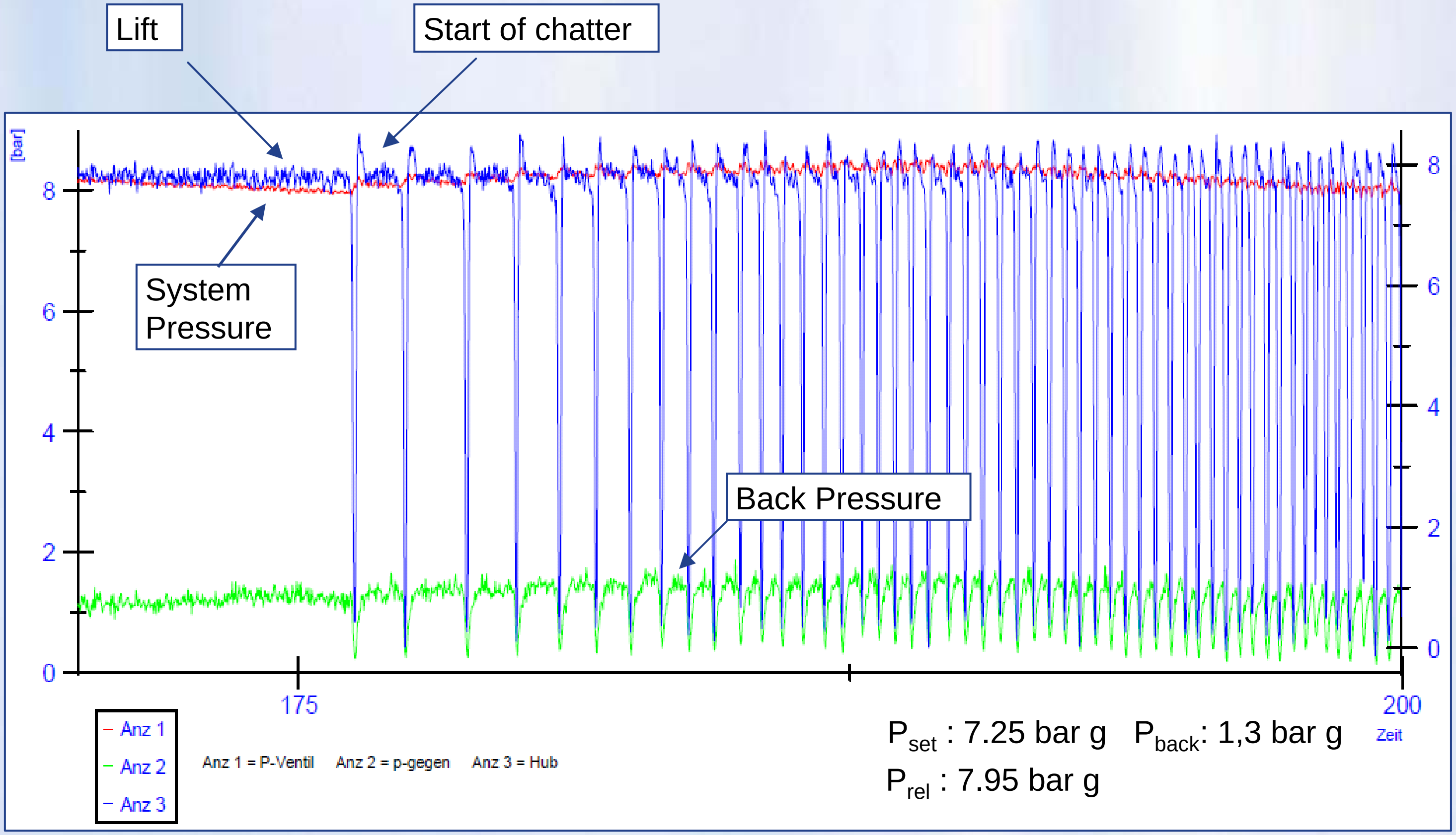
Type: 526 2H3 Conventional design

- **Inlet:** 2“ (50.8 mm); Outlet: 3“ (76.2 mm)
- **Flow Area:** 625 mm² (0.975 sq.in.)
- **Lift:** 7,9 mm (0,31”)
- **Test Pressure:** 7.25 bar g (5 psig)
- **Built up Back Pressure:** up to 1,4 bar g



Test Set Up for Back Pressure Tests on Air

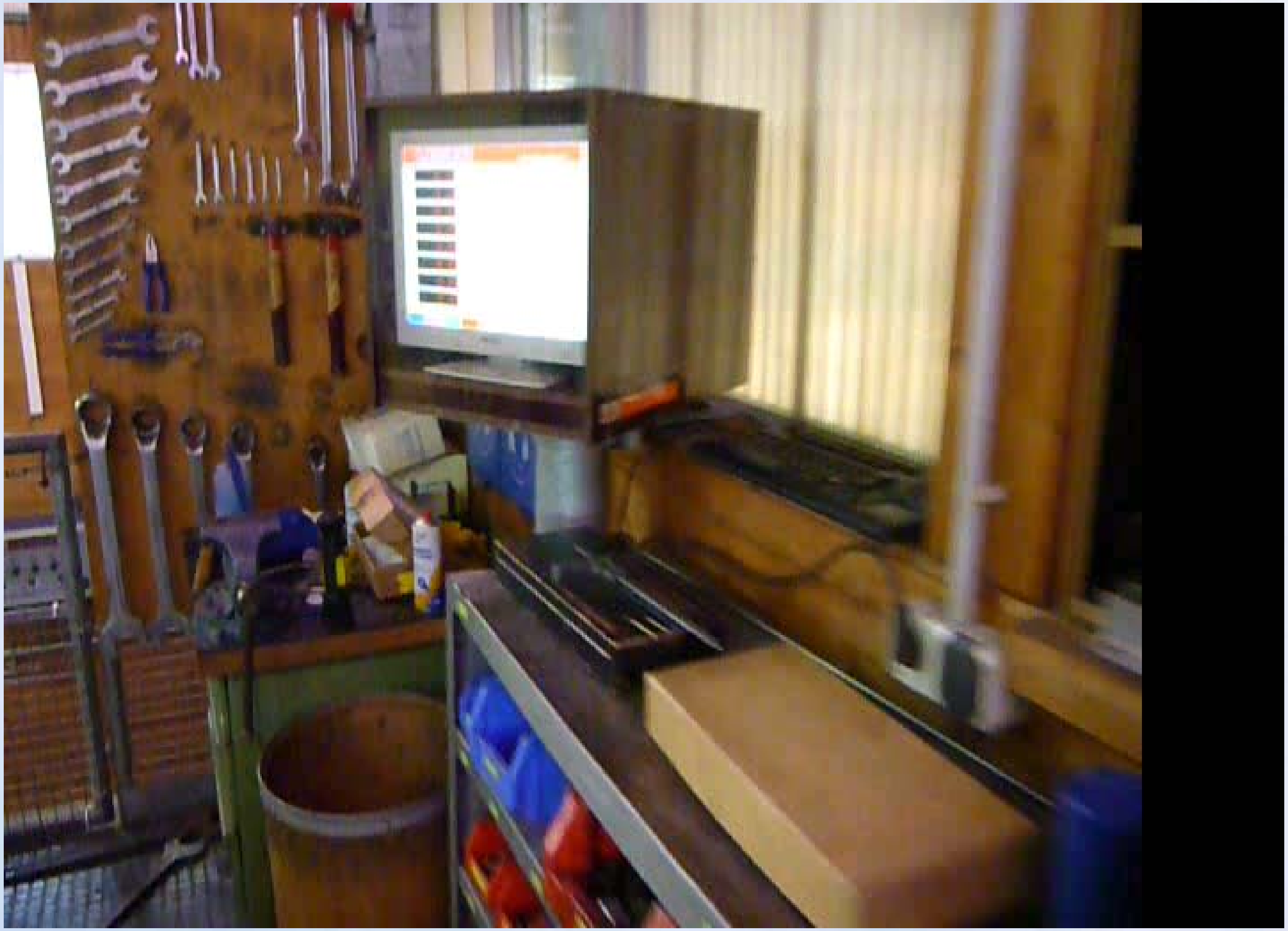
Test on 2H3 API Valve on LESER Test Bench



- Stable function up to 1,3 bar g back Pressure

Test on 2H3 API Valve on LESER test Bench

Video



Numerical Tools

The outcome of CFD calculations

Physical quantities which can be calculated using CFD

- Velocity
- Pressure
- Density
- Others: temperature, shear rates …

Integral quantities

- Mass flow rate
- Fluid force acting on valve disc

Objectives

ASME / TÜV certified Test Facility

Data Acquisition

Test Lab Capacities and Pressure Ranges

Product Data and Operating Conditions

Tests

Numerical Tools

Calculation of Mass Flow Rates

Fluid Force acting on disk

Transient Calculation of Flow

Opening with atmospheric Back Pressure

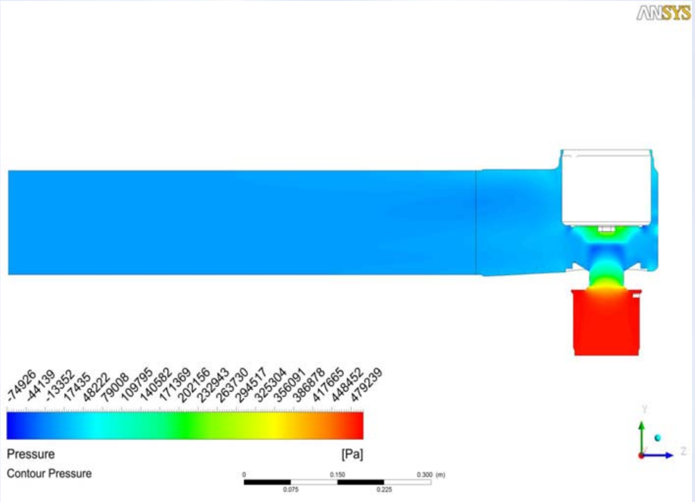
Experimental and Numerical Estimation

Calculation of Mass Flow Rates in Safety Valves using CFD

Software: ANSYS Fluent Version 13

Meshing details:

- Tetreader cells within safety valve body
- Hexaeder cells in inlet and outlet pipes
- Total number of cells: 1 – 4 mio.



→ **Difference between measured and calculated (CFD) steady-state air mass flow rates usually between 3 - 5%**

Fluid Force acting on Disc

Steady state Conditions

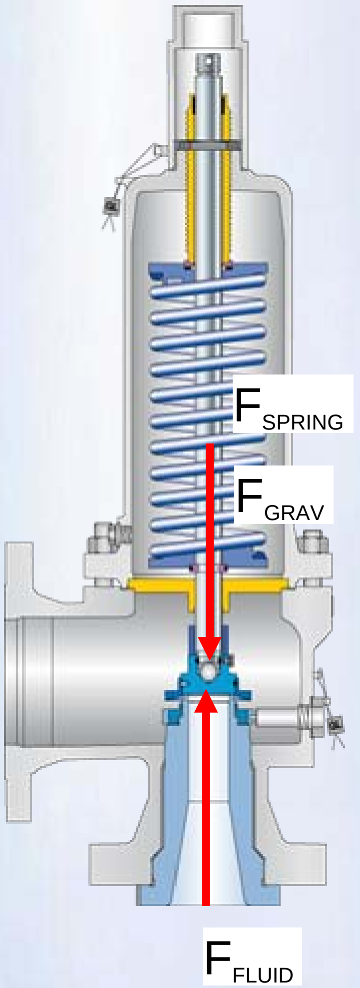
Force balance to maintain the disk at respective lift:

$$F_{FLUID}\ (x) - F_{SPRING}\ (x) - M_{MP}\, g \geq 0$$

Moving parts are:
disk, spindle and lower spring plate

FLUID force is calculated by the CFD program

SPRING force is linearly varying with the lift

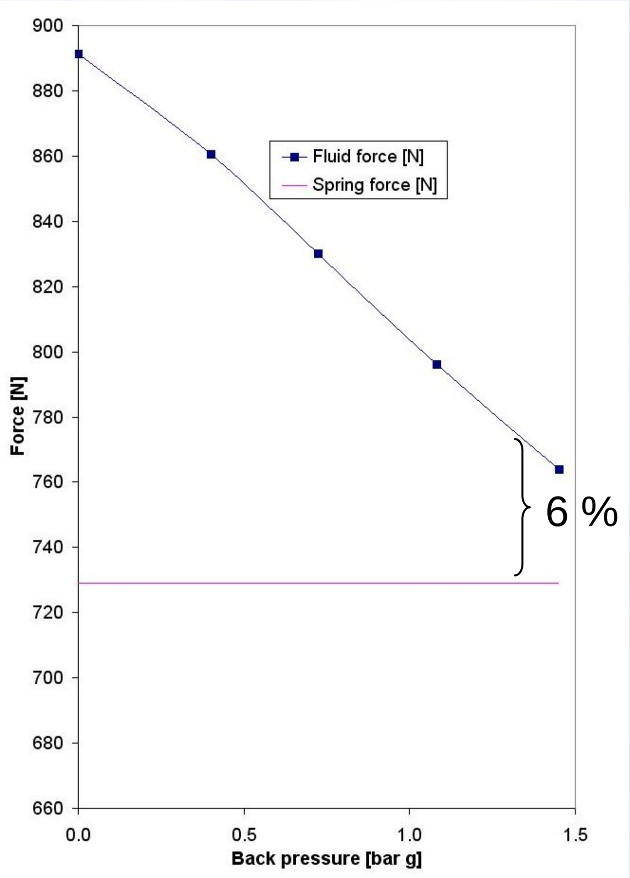


Fluid Forces on Disc of LESER Type 526 2H3 Valve in Function of the Back Pressure

Steady state flow

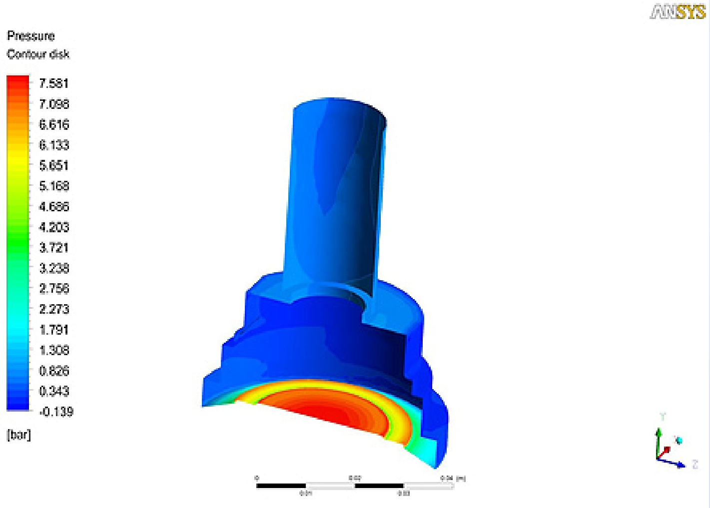
- **Lift:** 7.9 mm
- **Rel. Pressure:** 7.95 bar g
- **Set Pressure:** 7.25 bar g
- **Back pressure:** 0 – 1,4 bar g

Validation of Force Calculation pending

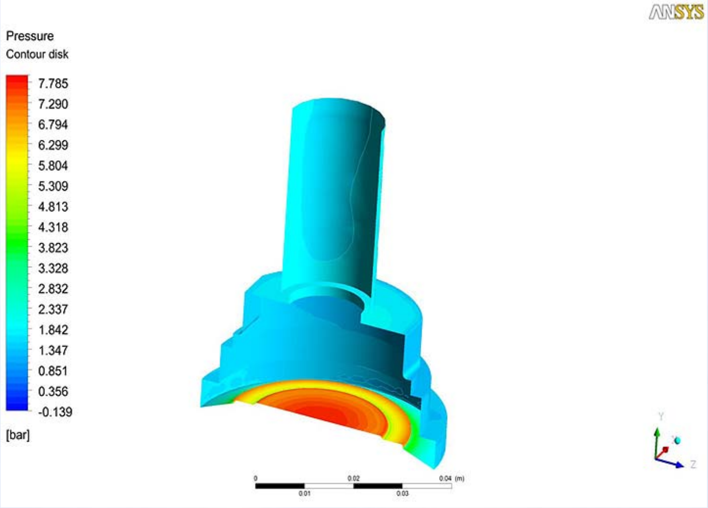


Fluid Forces on Disc of LESER Type 526 2H3 Valve in Function of the Back Pressure

Atmospheric back pressure



Back pressure ratio 20%



Transient Calculation of Flow in a Safety Valve

Opening of a safety valve with constant superimposed back pressure

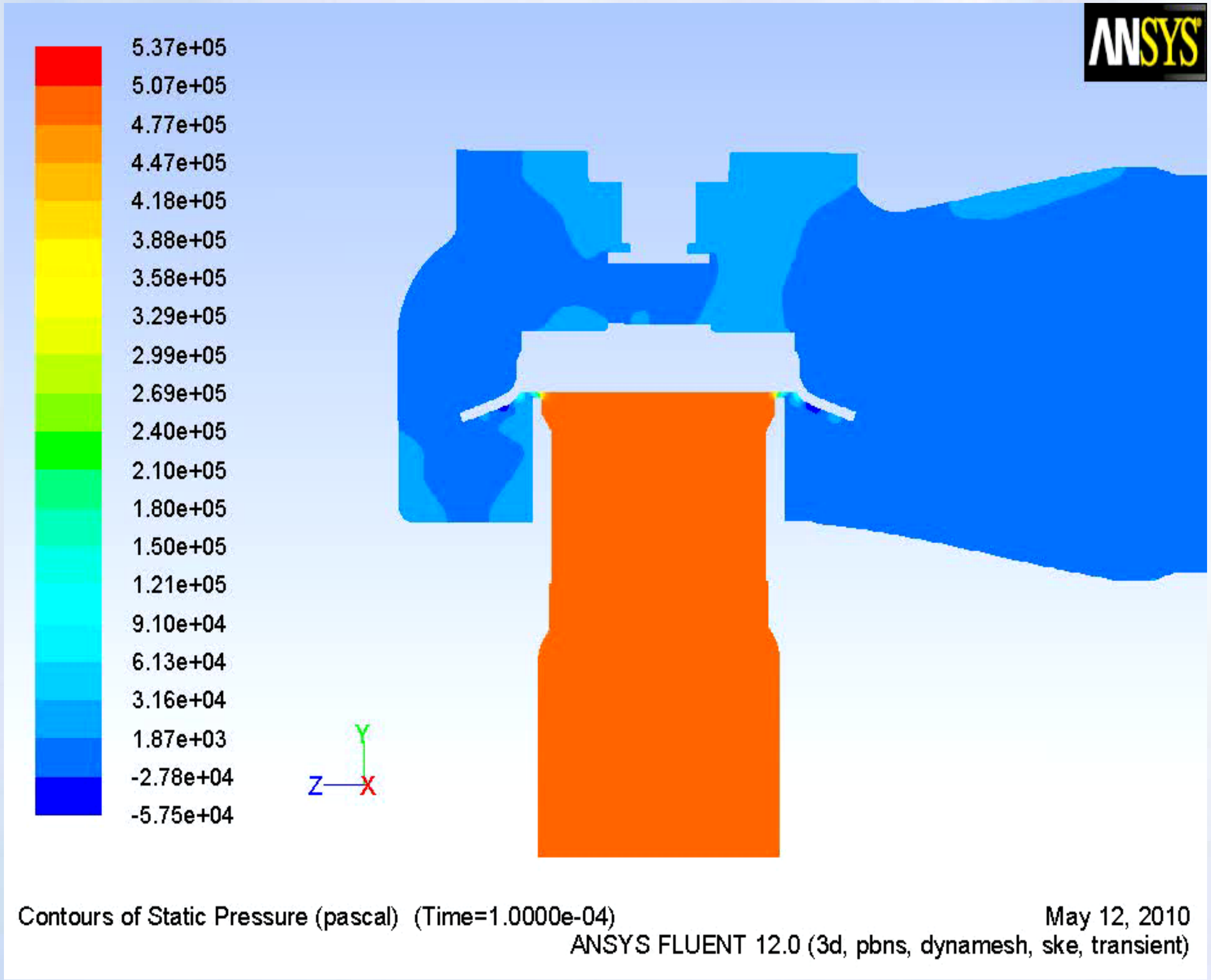
Instantaneous position of the disk (lift)

Calculation of the fluid force on the disk using CFD

Calculation of the new disk velocity solving

$$F_{FLUID}\ (x) - F_{SPRING}\ (x) - F_{FRIC}(\dot{x}) - F_{GRAV} = M_{MP}\frac{d}{dt}\dot{x}$$

Opening of a Safety Valve Typ 441 DN 50/80 with atmospheric Back Pressure



Pros and Cons of experimental and numerical estimation on the influence of Back Pressure

Measurements at test facility:

- Immediate response at which back pressure ratios chattering begins
- no information on the pressure and velocity distribution in the valve

CFD calculations:

- Mass flow calculation within 3 – 5% accuracy (steady state)
- Force calculation on disc possible, validation pending
- Visualization of pressure and velocity distribution possible
- Reduction of prototype testing and time-to-market period

Thank you very much for your attention.

LESER GmbH & Co. KG
20537 Hamburg, Wendenstr. 133 - 135
20506 Hamburg, P.O. Box 26 16 51

Phone +49 (40) 251 65 - 100
Fax +49 (40) 251 65 - 500

E-mail sales@leser.com
www.leser.com

S. EGAN

HAMBURG - 14th June 2011

Feedback on the 3 % rule for the inlet pressure drop at the entry to a relief valve

Contents

- **Background**
- **Definitions**
- **Illustration of the effect of inlet pressure drop on relief valve stability**
- **Case 1 : HCl reactor**
- **Case 2 : Steam boiler**
- **Dynamic stability criteria according to J. Cremers, L. Friedel & B. Pallaks**
- **Conclusions**

Background

- **Legislation and standards now include the 3 % rule:**
  - North America - ASME pressure vessel code & API 520 include the 3 % limit
  - Europe – Pressure vessels directive + EN 764-7 mentions 3 % limit
  - Germany - AD 2000-Merkblatt gives method to check pressure drop ≤ 3 % limit
  - ISO 4126-9 gives gives method to check pressure drop ≤ 3 % limit
- **Rhodia wishes to update the calculation of existing pressure relief valves:**
  - To be carried out during 5 yearly safety reviews of existing units
  - Identify need for each pressure relief valve
  - Identify sizing cases for each pressure relief valve
  - Check valve size required against valve size installed
  - Check inlet and outlet pressure drop
  - If size is wrong etc. evaluate levels of severity, probability and risk
  - Consider installation of new pressure relief valve
- **Problem encountered:**
  - For many relief valves the inlet pressure drop was not calculated before installation
  - In cases the 3 % rule was applied but to the flow rate of the risk scenario
  - So now many relief valves are found not to comply

Definition of the 3 % rule

• **Definition of pressure terms**	**Example**
• Opening pressure – relief valve fully open	22 bar gauge (set + 10 %)
• Set pressure – relief valve starts to open	20 bar gauge
• Reseat pressure – relief valve closes	19 bar gauge (set – 5 %)
• Blowdown = set minus reseat	1 bar gauge

- **The inlet pressure loss calculation**
  - Includes the effect of bends, reducers, rupture discs etc
  - Is based on the rated capacity of the relief valve
  - Flow rate with relief valve fully open
  - Using rated discharge coefficient divided by 0.9
  - Flow capacity typically 2 X requirement
  - Pressure loss proportional to square of flow so 12 % instead of 3 % !

• **The 3 % rule is stated in three ways:**	**Maximum pressure loss**
• Inlet pressure loss ≤ 3 % of set pressure	0.6 bar (3 %)
• Inlet pressure loss ≤ one third of blowdown	0.33 bar (1,7 %)
• Blowdown - inlet pressure loss ≤ 2 % of set pressure	0.6 bar (3 %)

- **Under ISO 4126-9 the smallest limit applies**

Illustration of the effect of inlet pressure drop

- **Iteration with inlet pressure drop equal to 12 % of set pressure**

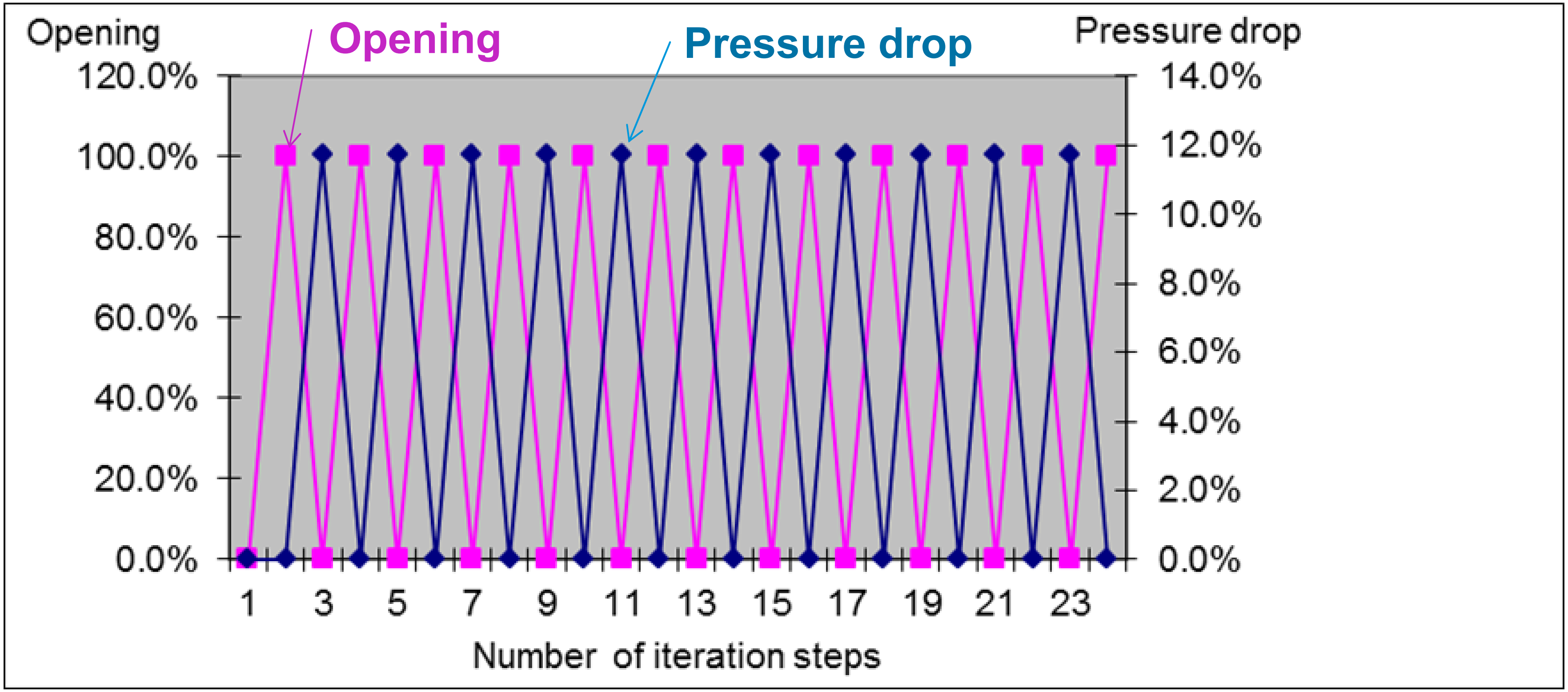


Illustration of the effect of inlet pressure drop

- **Frommann & Friedel, J. Loss Prevention, 11 (1998) 279-290**
  - **Measurements on DN25/DN40 relief valve with 23 mm nozzle**
  - **Inlet line 1 m long DN25**
  - **Inlet pressure loss 9 % of set pressure**
  - **Open-close cycle with periodicity of 0.01 s**

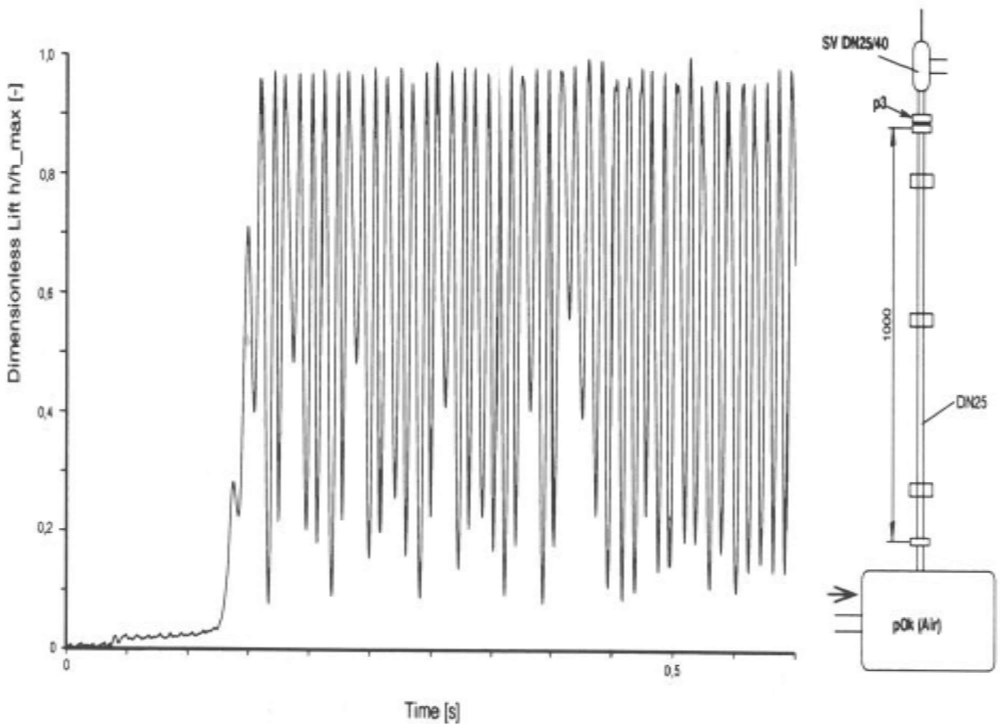


Fig. 7. Measured related disc lift in dependence on time in case of a vertical inlet line DN25 with an average length of about 1 m.

Illustration of the effect of inlet pressure drop

- Iteration with inlet pressure drop equal to 5.5 % of set pressure

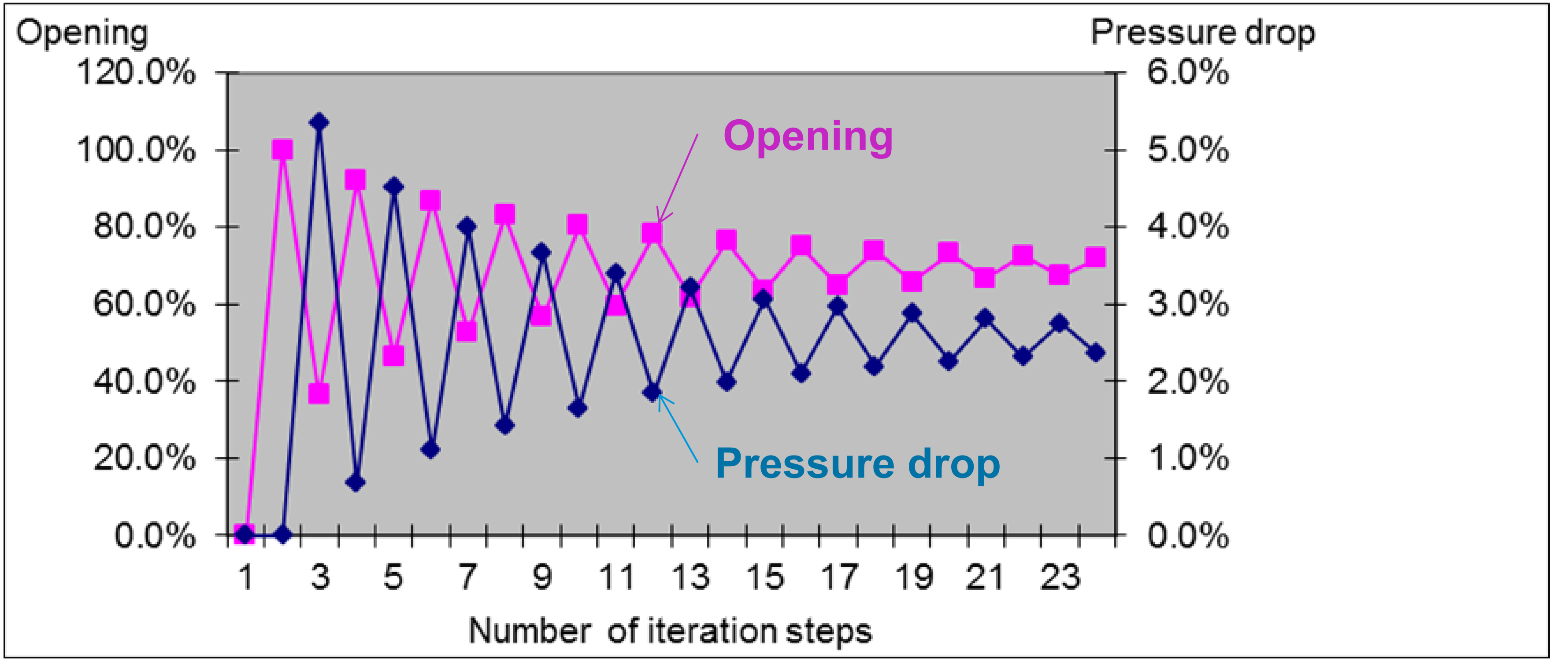
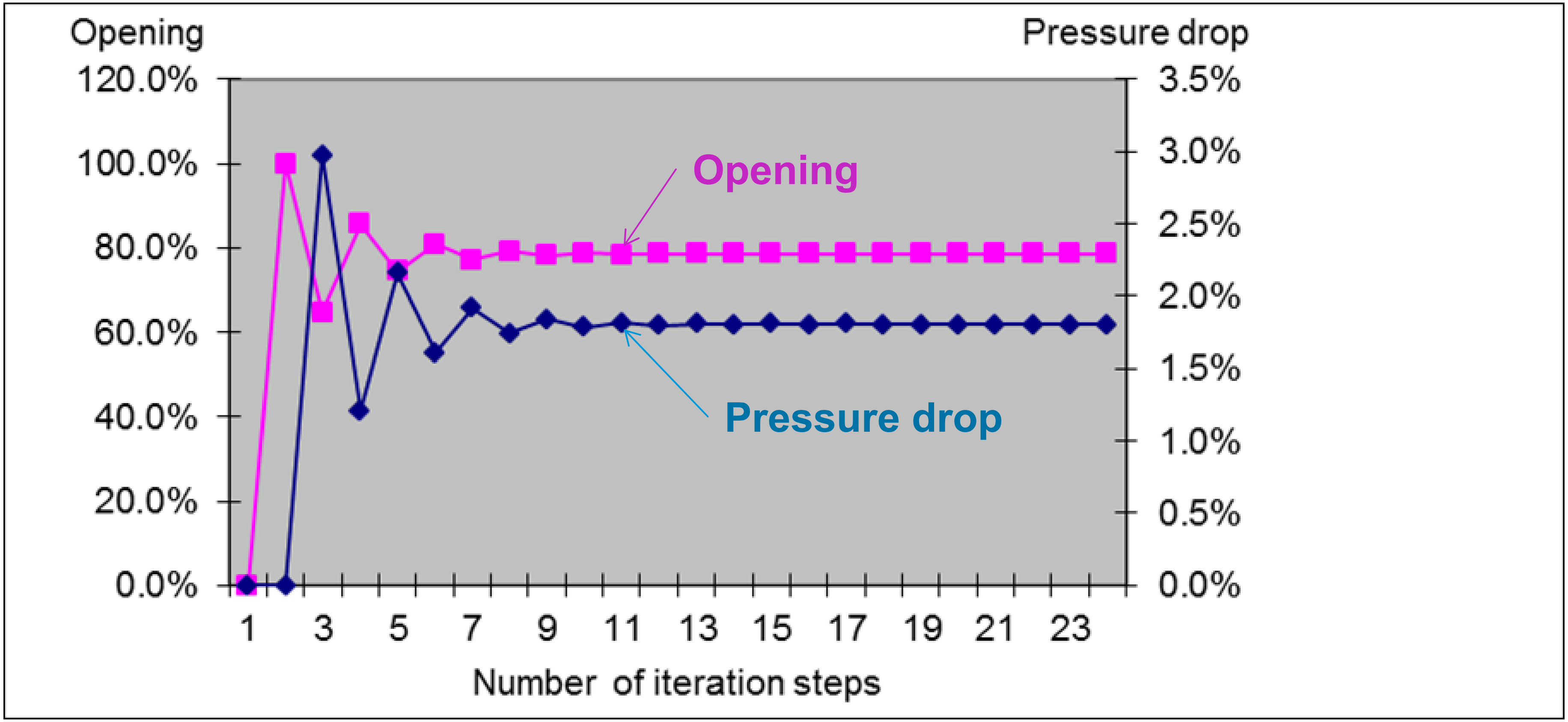


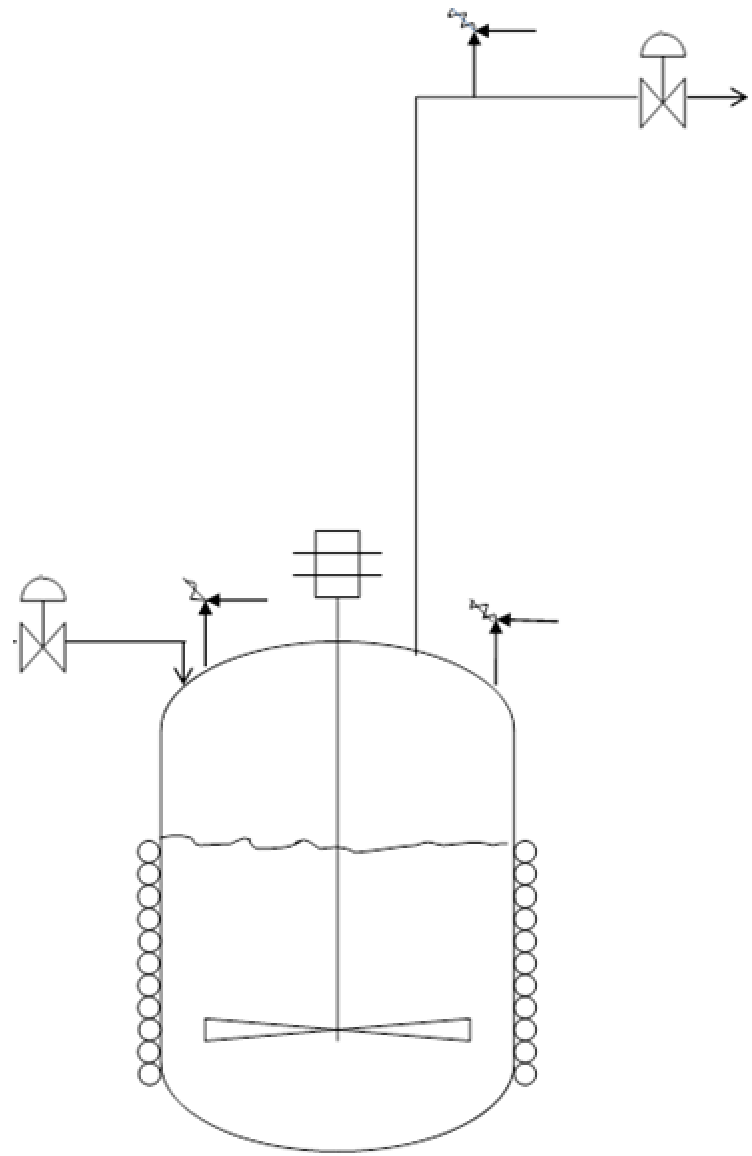
Illustration of the effect of inlet pressure drop

- Iteration with inlet pressure drop equal to 3 % of set pressure



Case 1 : HCl reactor – process & scenario

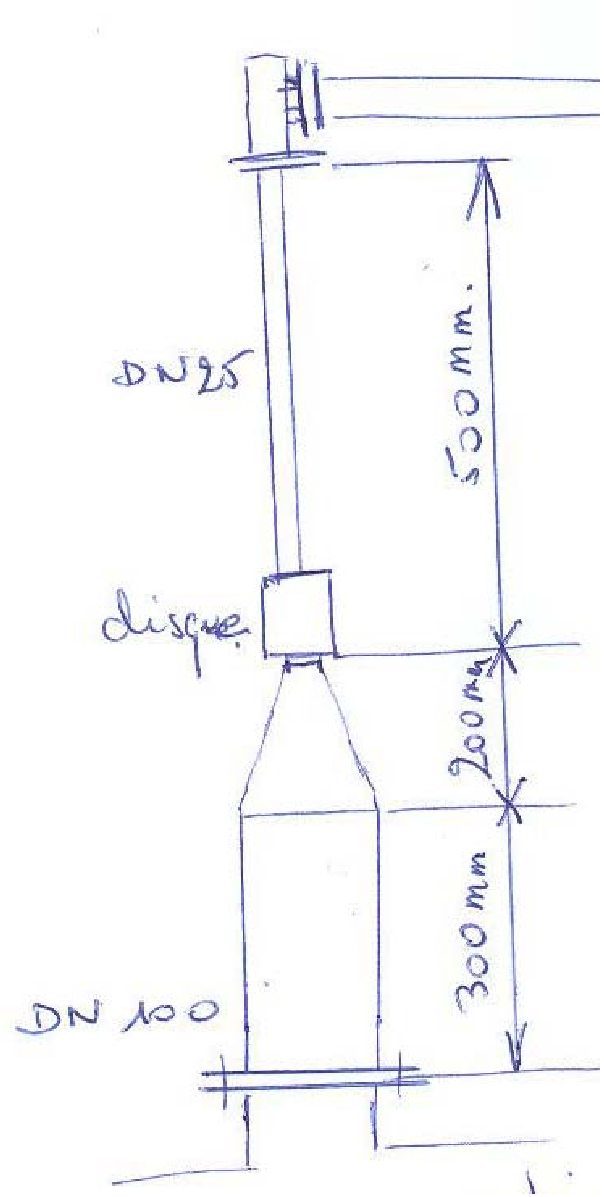
- **Process outline:**
  - 5 m3 reactor
  - Semi batch type operation
  - Controlling reactant added over two hours
  - Reaction produces HCl at 800 kg/hour
- **Scenario identified:**
  - Type 1 according to ISO 23252 / API 521
  - Control valve closes on line to HCl absorber
- **Hardwire trips:**
  - 2 sensors to detect overpressure
  - 2 block valves to stop flow of controlling reactant
  - Full test 1 per year
  - Rated SIL 2 under IEC 61511
- **Pressure relief valves**
  - 3 totally independent relief valves
  - Weir valves STARFLOW model 1E2
  - Each valve rated to 3300 kg/hour HCl



Case 1 : HCl reactor – inlet resistance

- **Calculation of inlet resistance coefficient**

Item	ξ
0,5 m line DN25	0,52
Rupture disc	1,5
Reduction	0,1
0,1 m line DN100	0,05
Entry nozzle (square cut)	<u>0,25</u>
Total	2,42



- **ISO 4216-9 $\Sigma\xi_{max}$ = 1.63 to comply with 3 % rule**
- **None of the existing valves comply**
- **Entry (0.25) + disc (1.5) = 1.75 !**

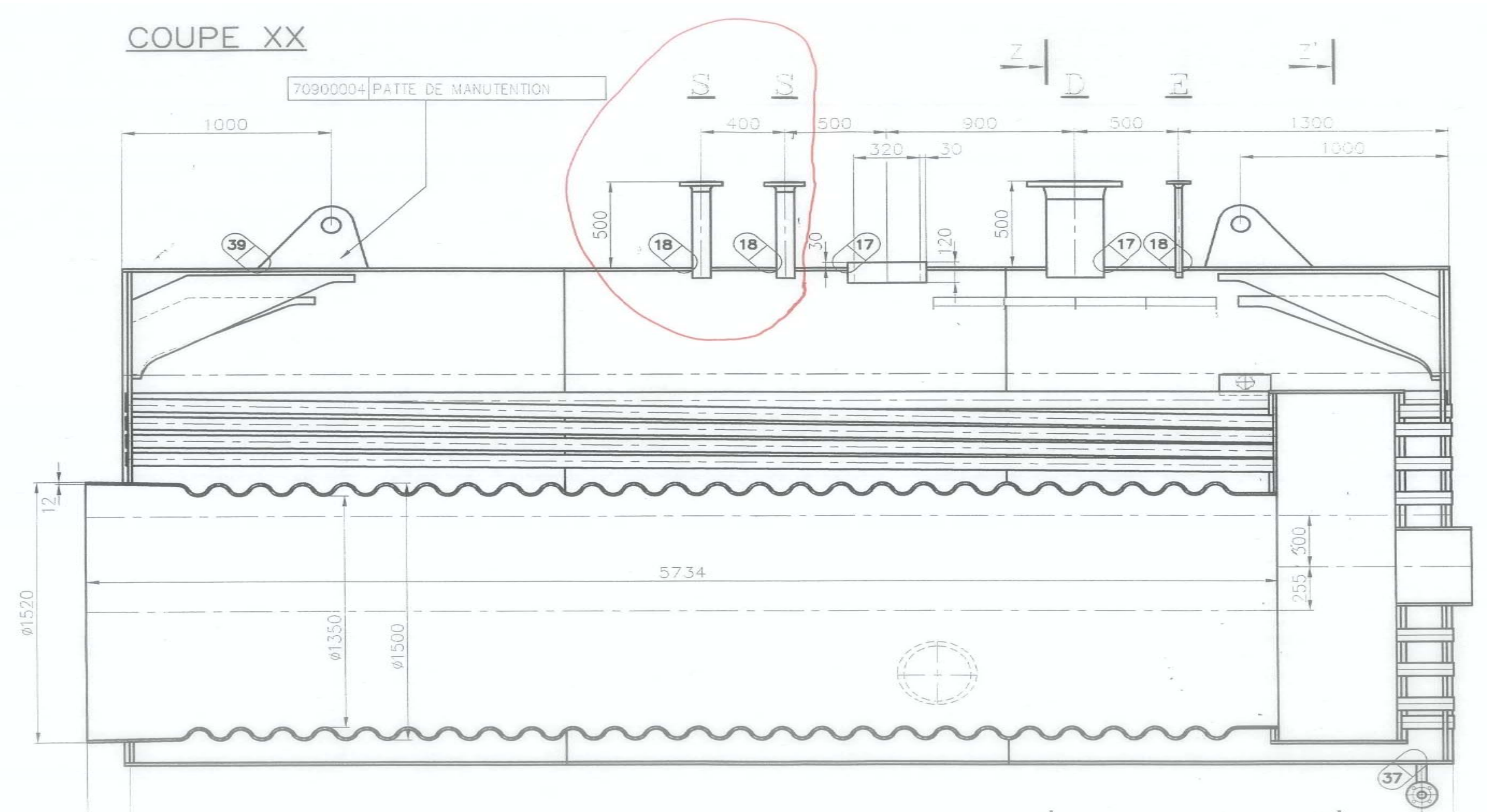
Case 1 : HCl reactor – options

- **Pressure ratings**
  - Reactor design pressure = 29 bar gauge
  - Relief valve full open pressure = 24.2 bar gauge (10 % accumulation)
  - Relief valve set pressure = 22.0 bar gauge
  - Relief valve reseat pressure < 22.0 bar gauge
  - Normal pressure during reaction = 6.0 bar gauge
- **Option 1: do nothing**
  - Inlet pressure loss = 1.3 bar gauge (6 % of set pressure)
  - Nothing can happen unless the pressure exceeds 29 bar gauge
  - At any pressure above 23.3 bar gauge the valves are open anyway
  - But scenario can last 2 hours which could be say 1 million open close cycles
- **Option 2: reduce the resistance at the entry to the valve**
  - Main contribution is from rupture disc : 1.5 with total limit of 1.63
- **Option 3: adjust the blowdown and reseat pressures**
  - ISO 4126-9 Blowdown ≥ 3 X Inlet pressure loss
  - Target for reseat pressure = 20.3 bar gauge
- **Opion 4: put rupture disc in DN100 section (increases limit on ξ)**

Case 2 : Steam boiler – process & scenario

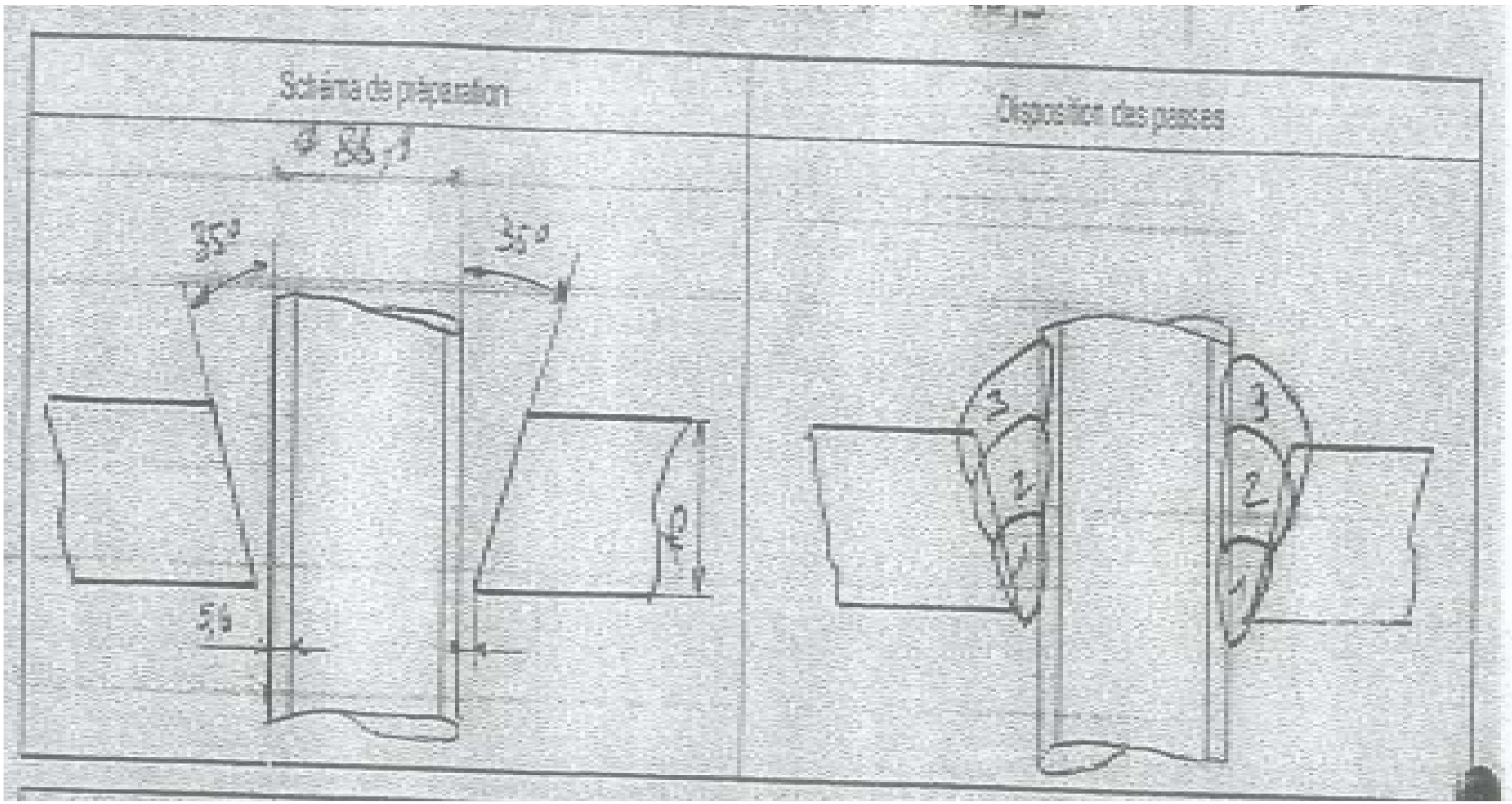
- **Process**
  - Gas fired boiler generates steam at 15000 kg/hour for site utilities
  - Steam is on shell side
  - Smoke is led through tubes (horizonal)
- **Scenario**
  - Site usage of steam stops abrubtly
  - Steam generation continues
  - Explosion of boiler
  - Blast wave with potential offsite effects
- **Hardwired trip**
  - One pressure sensor
  - Block and bleed arrangment on gas line
  - Fully tested
  - Rated to SIL 1 under IEC 61511
- **Pressure relief valves**
  - 2 totally independent relief valves
  - LESER 4422-4565
  - Each valve is rated to over 21000 kg/h

Case 2 : Steam boiler – diagram of boiler



Case 2 : Steam boiler – close up of inlet nozzles

- **Nozzles used for relief valves are penetrating !**



Case 2 : Steam boiler – inlet resistance

- **Calculation of inlet resistance coefficient**

Item	ξ
0,4 m line diameter 77 mm	0,09
Entry nozzle (penetrating)	0,50
Total	0,59

- **ISO 4216-9 $\Sigma\xi_{max}$ = 0.27 to comply with 3 % rule**
- **Neither of the existing valves comply**

Model of J. Cremers, L. Friedel & B. Pallaks

- **J. Cremers, L. Friedel & B. Pallaks, J. Loss Prevention, 14 (2001) 261-267**
  - Consideration of relief valve dynamics
  - Expansion wave caused by valve opening
  - Comression wave caused by valve closing
- **Step 1: calculate the relief valve opening time**

$$t_{open} = \left(0.015 + 0.02\frac{\sqrt{d_0/0.015m}}{(P_0/P_U)^{2/3}(P_U/P_0)^2}\right)\left(\frac{h}{h_{max}}\right)^{0.7} \text{ seconds}$$

Symbol	Meaning	Case 1	Case 2	Units
t_{open}	Opening time	0.013	0.024	s
d_0	Diameter of orifice	0.0127	0.074	m
h	Valve lift	0.0031	0.0185	m
h_{max}	Maximum valve lift	0.0047	0.0185	m
P_0	Opening pressure	2.52	1.365	MPa
P_U	Atmospheric pressure	0.101	0.101	MPa

Model of J. Cremers, L. Friedel & B. Pallaks

- **Step 2 : calculate the time for a sound wave to travel the length of the inlet line**

$$t_w = \frac{L}{a}$$

Symbol	Meaning	Case 1	Case 2	Units
t_w	Time for sound wave to travel the path length	0.0017	0.0008	s
L	Length of inlet line	0.5	0.4	m
a	Speed of sound in medium	298	494	$m\ s^{-1}$
t_{open}	Opening time	0.013	0.024	s

- **Step 3 : check that the opening time is at least twice as long**

$$t_{open} > 2t_w$$

- **First condition is OK in cases 1 and 2**

Rhodia

Model of J. Cremers, L. Friedel & B. Pallaks

- **Calcluate the maximum allowable length of the inlet line**

$$L_{max} = \frac{0.05}{2\sqrt{2\Psi}} \frac{A_E \alpha_{Ae}}{A_0 \alpha} \frac{1}{\sqrt{\rho}} \frac{P_0 - P_a}{\sqrt{P_0}} t_{open}$$

Symbol	Meaning	Case 1	Case 2	Units
L_{max}	maximum allowable length	0.15	0.325	m
Ψ	Outlflow function (MERKBLATT)	0.484	0.450	-
A_E	Area of inlet line	0.000491	0.004766	m^2
A_0	Area of valve orifice	0.000127	0.004301	m^2
α	Discharge coefficient of valve	0.973	0.699	-
α_{Ae}	Discharge coefficient of line	0.54	0.79	-
ρ	Density of medium	48.7	6.944	$kg\ m^{-3}$
P_0	Opening pressure	2.52	1.365	MPa
P_a	Atmospheric pressure	0.101	0.101	MPa
t_{open}	Opening time	0.013	0.024	s

- **So the real length is too long in case 1 (0.5 m) and in case 2 (0.4 m)**

Conclusions

- **For both cases examined:**
  - the inlet pressure loss is well over 3 %
  - the Inlet line is too long according to the dynamic criteria of Cremers et al
  - we cannot rely on the relief valve as a protection layer
- **Possible solutions:**
  1. Change the inlet piping geometry?
  2. Fit oscillation dampers?
  3. Reduce the value of the reseat pressure?
- **We welcome problems!**

SAFETY RELIEF VALVE STABILITY
(AN ENGINEERING ANALYSIS)

PRESENTED TO THE

DIERS SAFETY RELIERF VALVE STABILITY MEETING

JUNE 17, 2011

BY

HAROLD G. FISHER

FISHERINC

(CONSULTANT TO FAUSKE & ASSOCIATES, LLC)

DIERS Users Group Mtg.
April, 2008 Las Vegas Nevada

"Odds and Ends"
Relief Valve Stability
And
Inlet Pressure Loss—
"Searching for Understanding"

Michael A. Grolmes
Centaurus Technology, Inc.
4590 Webb Road
Simpsonville, KY 40067
502-243-9678
ctimag@att.net

CTI
Centaurus Technology, Inc.

DIERS Users Group Mtg.
October, 2009 Houston TX

"Some Further Considerations Of Upstream Parameter Effects On Relief Valve (in) Stability"

Michael A. Grolmes
Centaurus Technology, Inc.
4590 Webb Road
Simpsonville, KY 40067
502-243-9678
ctimag@att.net

Centaurus Technology, Inc.

Presented by: H. G. Fisher, FisherInc.

DIERS Users Group Mtg.
October, 2010 Reno Nevada

"Odds and Ends"
Relief Valve Stability-Part 3
Is There Life Outside of the
3% Inlet Pressure Loss Rule

Michael A. Grolmes

Centaurus Technology, Inc.

4590 Webb Road

Simpsonville, KY 40067

502-243-9678

ctimag@att.net

Centaurus Technology, Inc.

Old Native American wisdom teaches that when one discovers that one is riding a dead horse, one should dismount and find a fresh horse.
However, modern business wisdom has developed many alternative courses of action. Among these are:
Get a bigger whip.
Change riders.
Appoint a committe to study the dead horse.
Visit other sites to see how they ride dead horses.
Develop new training methods to increase dead horse riding skills.
Harness several dead horses together to improve performance.
Study alternative uses for the dead horse.
Promote the dead horse to a higher management position.
Develope motivational slogans and mission statements such as;

> **Our horse is better, faster, and cheaper dead.**
> **No horse is too dead to whip.**
> **This is the way we have always ridden horses.**

Some Background

- Berwanger et. al,. Proc. Safety Prog.; Vol. 19, No. 3 reported the following:
- 1072 PSVs in a 13049 item sample pool had >3% inlet pressure drop (8.2%)
- 1606 PSVs in the same sample pool had excess outlet pressure drop, (12.3%)
- "----various experts in this area hold very strong yet contradictory opinions on this topic."

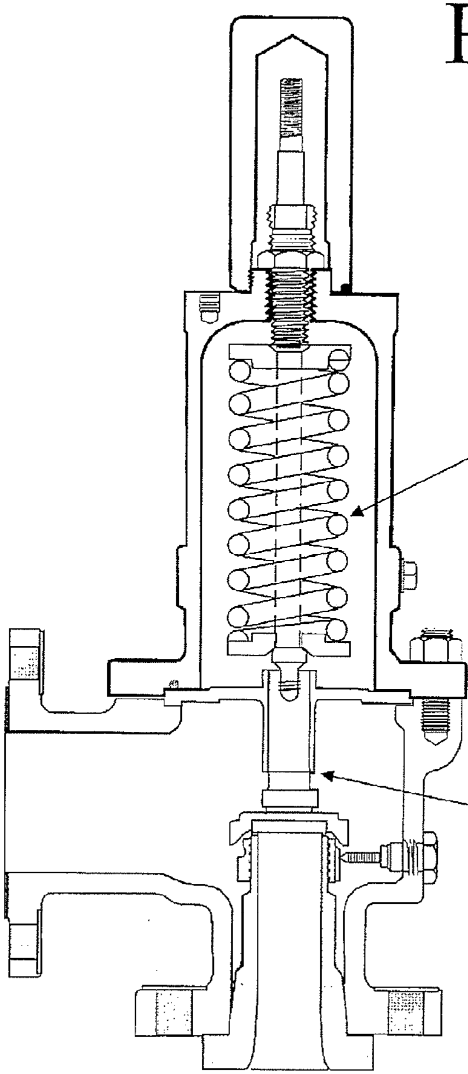
Takaway Observations

(a) Some inlet pipe configurations of the same diameter as the relief valve inlet will not work with a "square cut" vessel inlet or the presence of a rupture disk or both. A larger diameter inlet pipe may be required. The problem increases in severity with increasing PSV size.

(b) All Liquid flow is more restrictive than all gas flow which is more restrictive than "flashing" two phase flow. Therefore evaluation for all gas flow is sufficient for two-phase flow at the same set pressure.

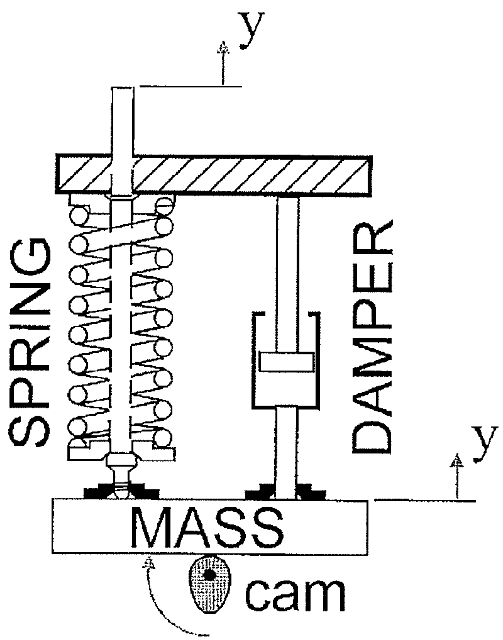
(c) There is a significant difference between allowance for 5% inlet pressure loss and 3%.

Relief Valve Operation as a Spring –Mass System



The mass is there, but not in one lump. There is the valve cap, a slider piece, the spring rod, a restraining button and some part of the spring itself.

Textbook Spring – Mass – Damper System



$$m\frac{d^2y}{dt^2}+c\frac{dy}{dt}+k\,y = F_o\sin(\omega t)$$

has solution

$$y = Y\sin(\omega t-\varphi)$$

where the amplitude factor Y

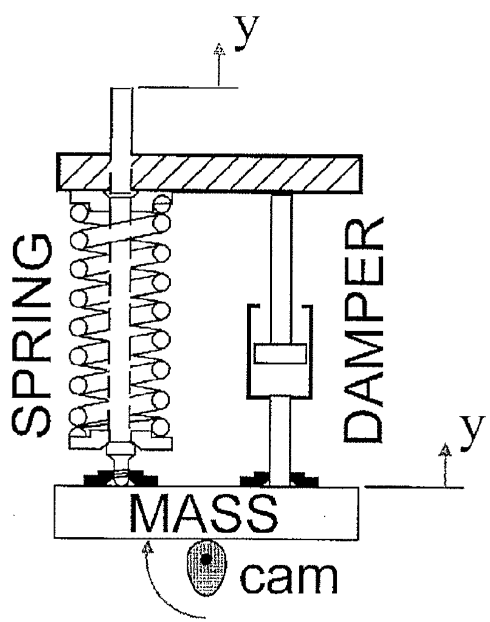
$$Y=\frac{F_o/k}{\left\{\left(1-\frac{m\,\omega^2}{k}\right)^2+\left(\frac{c\,\omega}{k}\right)^2\right\}^{1/2}}$$

m – mass

c – damping coefficient

k – spring constant

F_o - force



m – mass

c – damping coefficient

k – spring constant

F_o - force

The phase shift angle

$$\varphi = \tan^{-1}\left(\frac{c\,\omega}{k - m\,\omega^2}\right)$$

and the natural frequency is

$$\omega_n^2 = \frac{k}{m}$$

The dimensionless *damping factor is*

$$DF = \frac{c}{m\,\omega_n} = \frac{c}{\sqrt{k\,m}}$$

Now let

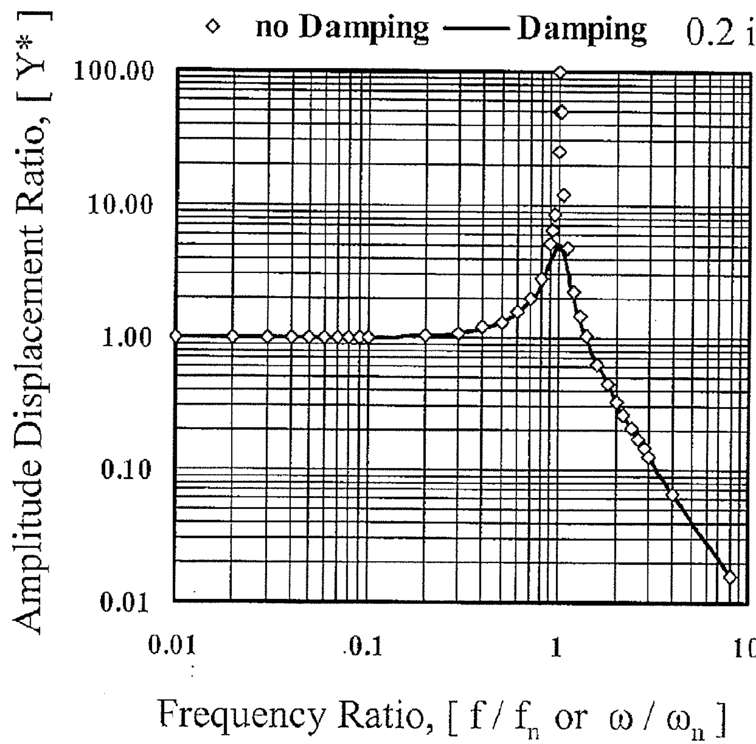
$$Y^* = \frac{Y}{F_o / k}$$

So that

$$Y^* = \frac{1}{\left\{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(DF\,\frac{\omega}{\omega_n}\right)^2\right\}^{1/2}}$$

Amplitude Response for Spring – Mass System with and without damping

Damping Factor = 0.2 in illustration



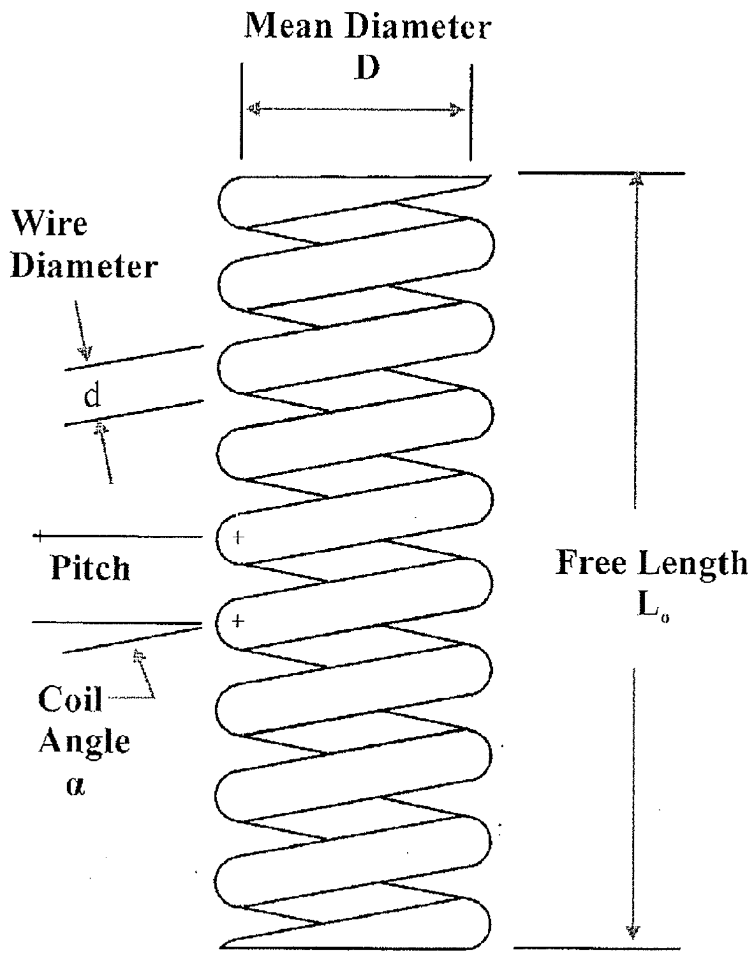
Frequency of the forcing function to the system natural frequency

If the damping factor is 0.2, the phase shift angle is less than 4°

If one wishes to avoid the resonant frequency, one can do so without any further coupling of the spring mass system to the rest of the problem.

However, to avoid resonance, one needs to know the <u>spring constant</u> (k), and the <u>mass</u> (m), in order to identify the <u>natural frequency</u> (ω_n)

Helical Springs

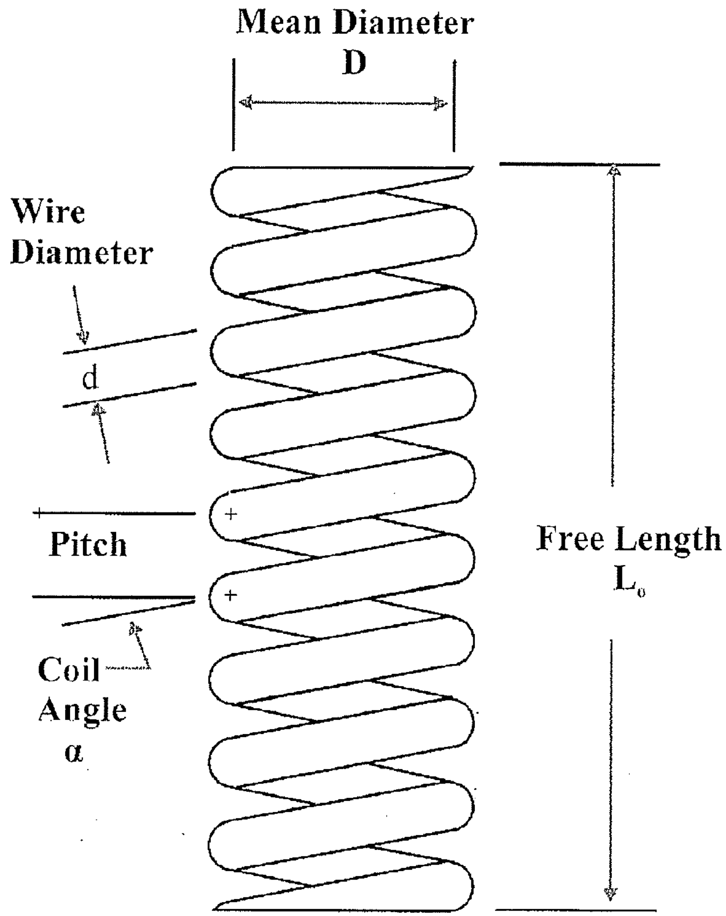


Relief Valves typically

have springs with closed ends ground such that the number of active coils is Na = Nt – 2

Where Nt is the total number of coils

Helical Springs



Nomenclature:

N_a is the number of active coils

G is the modulus of torsion or rigidity

C is the diameter modulus; C= D/d

D is the mean diameter

d is the coil diameter

Spring Constant k

$$k = \frac{G\,d}{8\,C^3\,N_a}$$

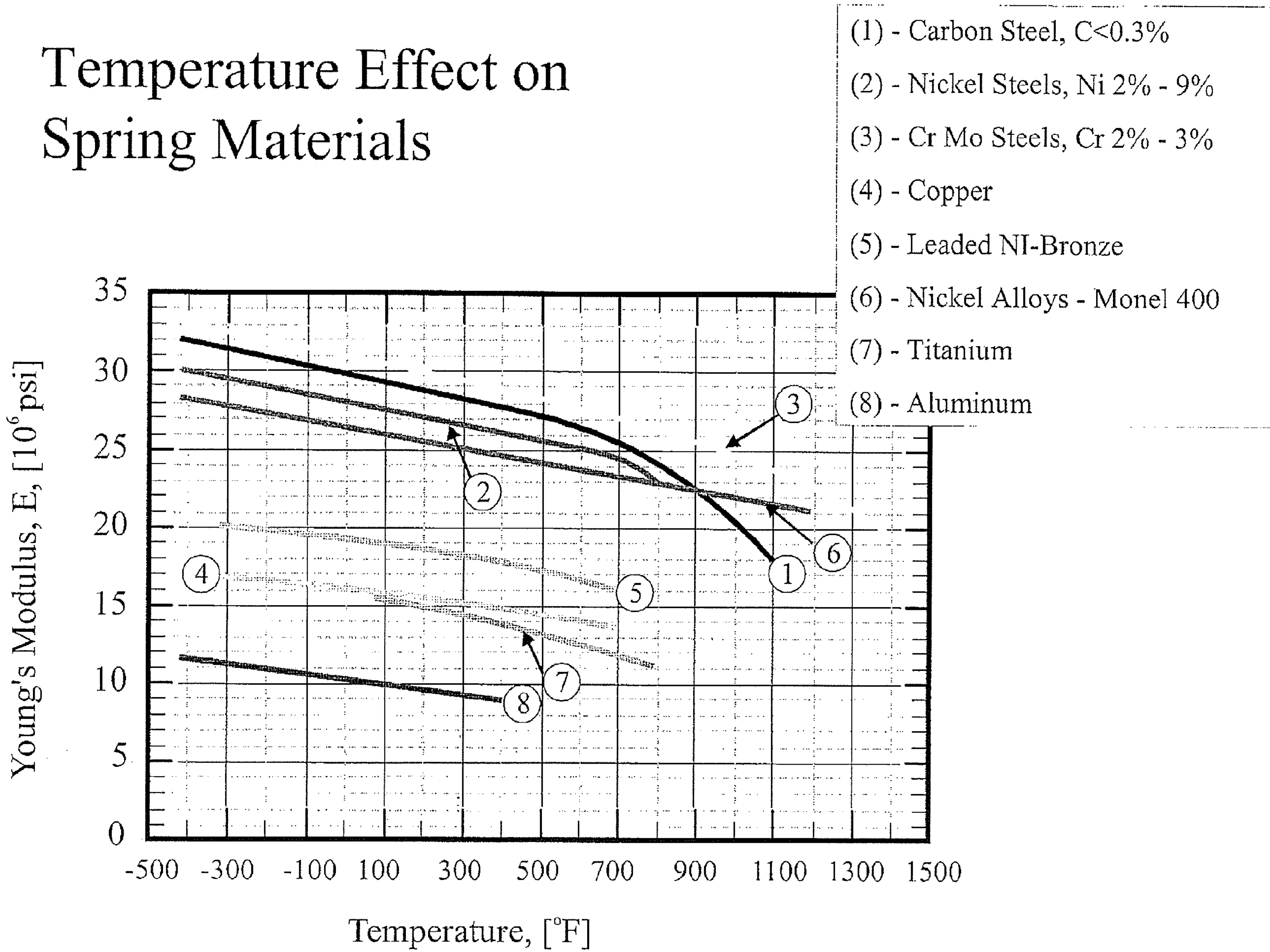
Young's Modulus (E) and Modulus of Torsion (G)

Metal	Poisson's Ratio	E $[10^9$ Pa]	G $[10^9$ Pa]
Aluminum	0.330	69	27
Copper	0.360	117	43
Ni-Steel	0.310	213	76
Stainless Steel 18-8	0.300	201	73
Carbon Steel	0.303	202	79
High Carbon Steel	0.295	210	81
Inconel	0.290	214	79

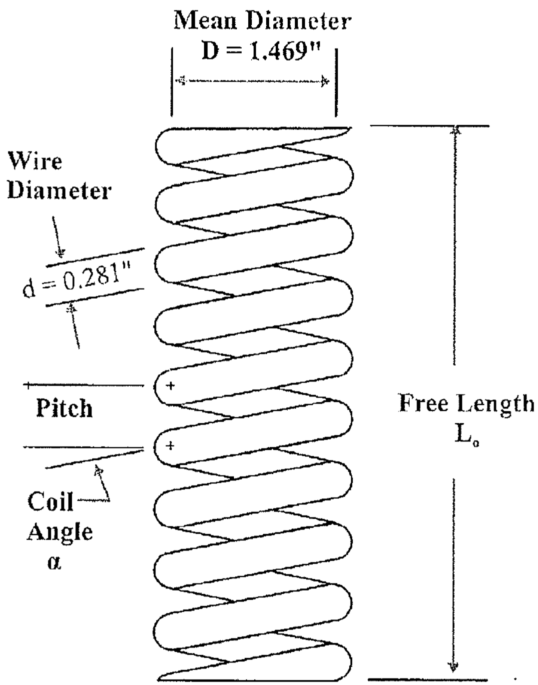
$$G = \frac{E}{2(1+v)}$$

Poisson's Ratio

Temperature Effect on Spring Materials



Real Example:

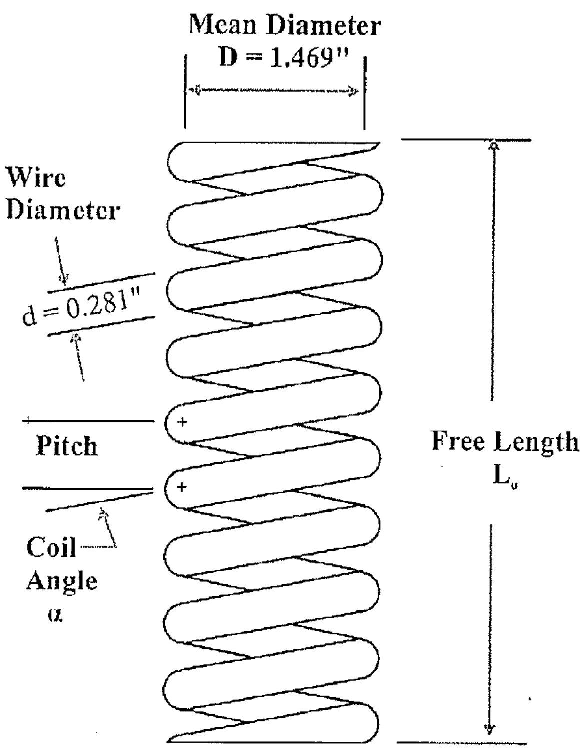


Spring from
Farris 26 FA 10 - 120
180 psig set

$N_a = 8$
$C = D/d = 5.288$
Assume:
$E = 205$ GPa or 29.7×10^6 psi
$\nu = 0.31$
$G = 78.2$ GPa or 11.35×10^6 psi

$$k = \frac{11.35E6 \, x \, 0.281}{8 \, x \, 5.288^3 \, x \, 8} = 349 \, \frac{lbf}{in}$$

Real Example: (continued)



Spring from
Farris 26 FA 10 - 120
180 psig set

Wgt. of Spring = 360 gm
Wgt. of other
parts in motion = 435 gm
Assume mass in motion is
435gm + 360 / 3 = 555 gm
or approximately 1.22 lb

Real Example: (continued)

So now we have

k = 350 lb_f / inch or 61294 N / meter

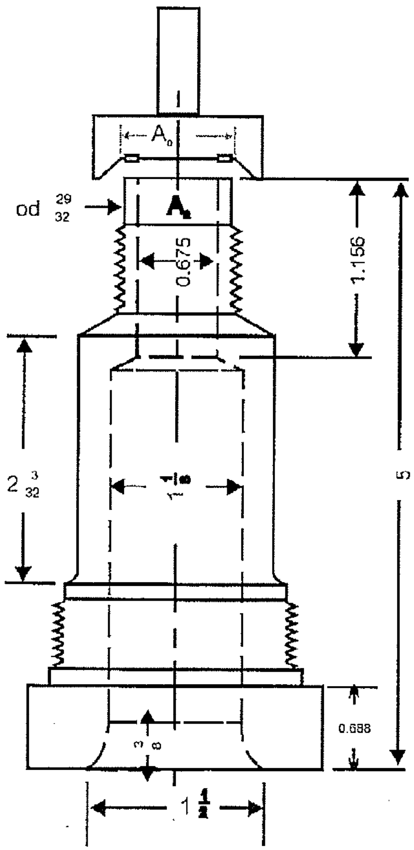
And

m = 0.555 kg

Therefore the natural frequency is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{61294}{0.555}} = 53\,Hz$$

Can we do better ??



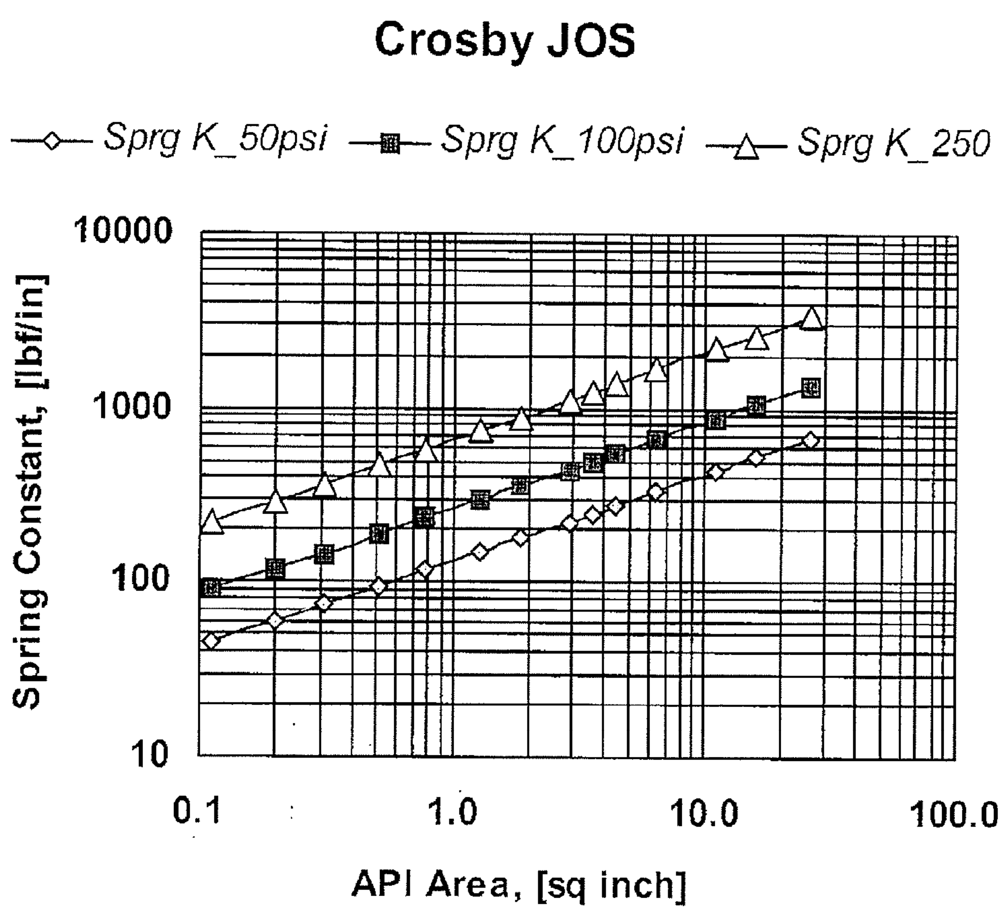
By examining the geometry of the nozzle and cap and doing a little calibration with the known example one can arrive at the following relations:

$$k = 1.08 \frac{P_{set}\ A_{redbook}}{y_{lift}}$$

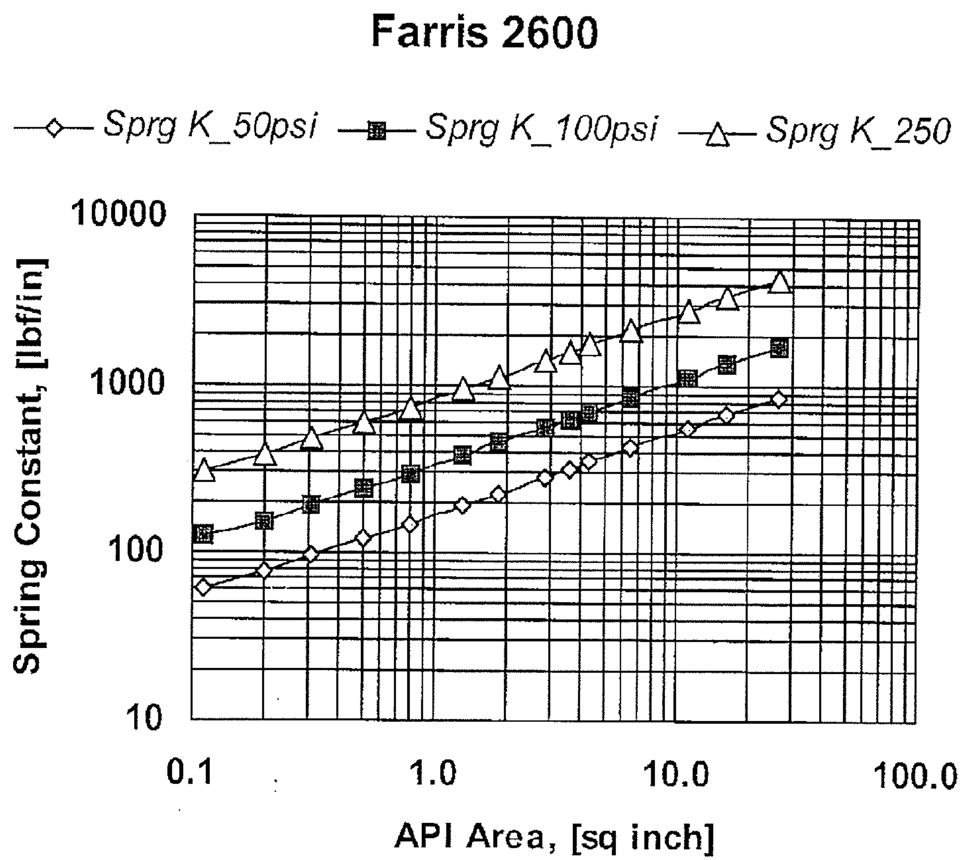
and

$$m = 0.025\, M_{valve}$$

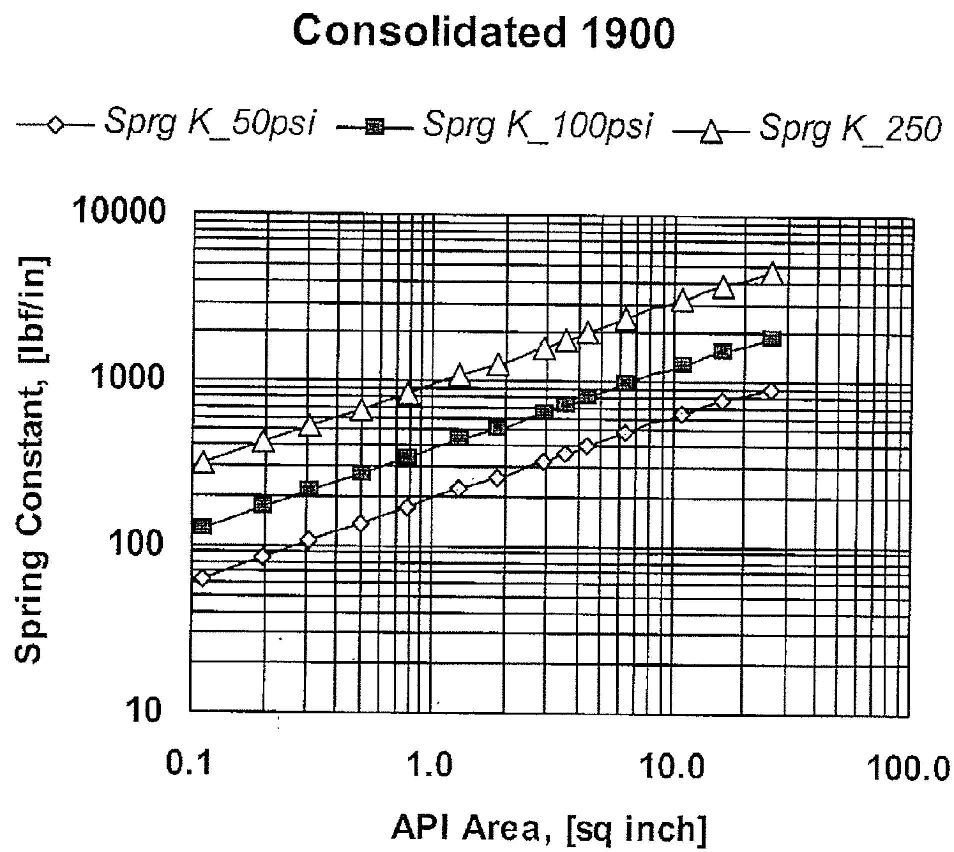
Relief Valve Spring Constant Estimate



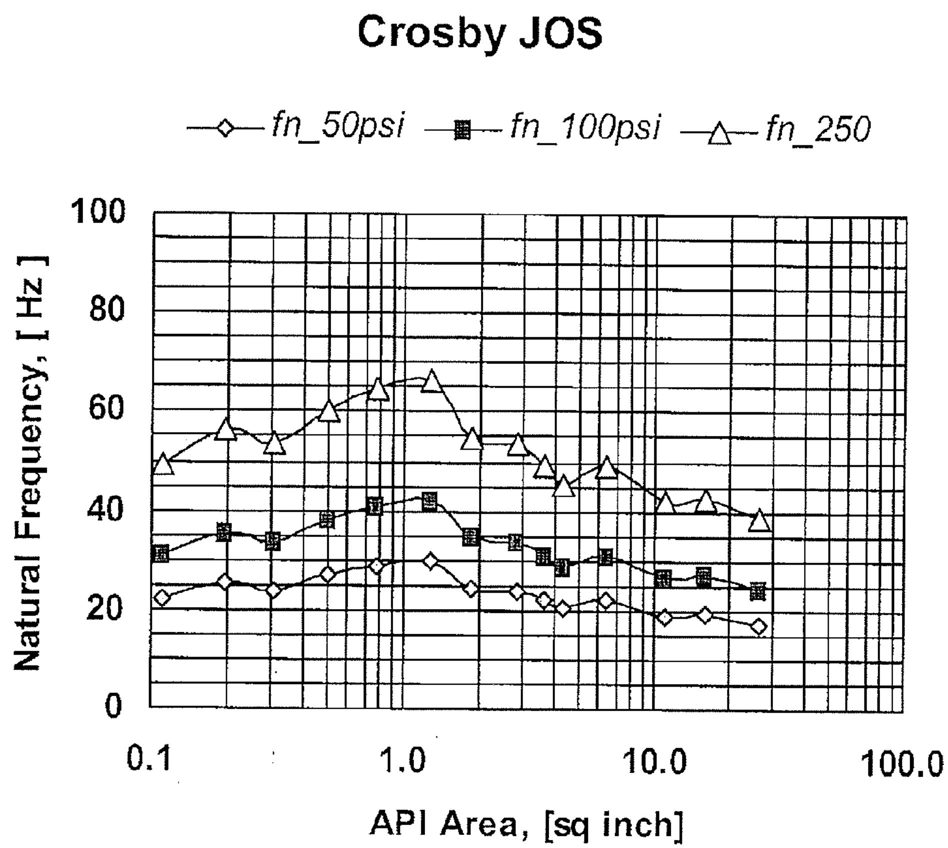
Relief Valve Spring Constant Estimate



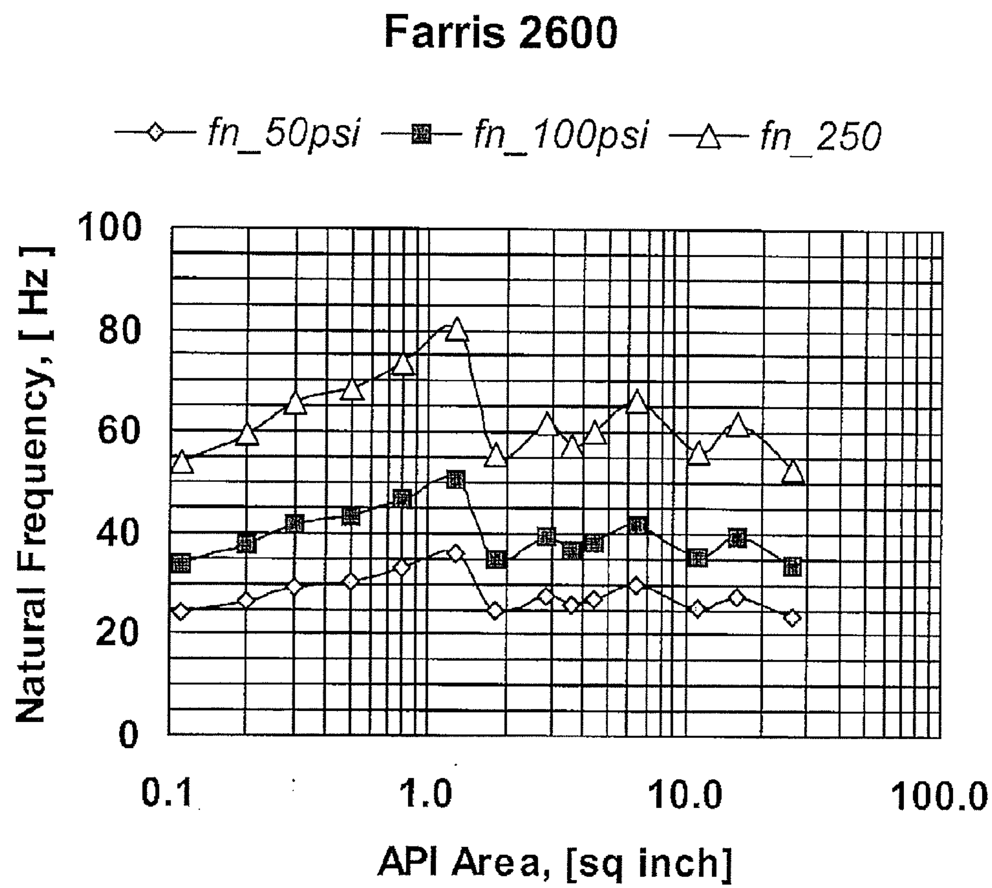
Relief Valve Spring Constant Estimate



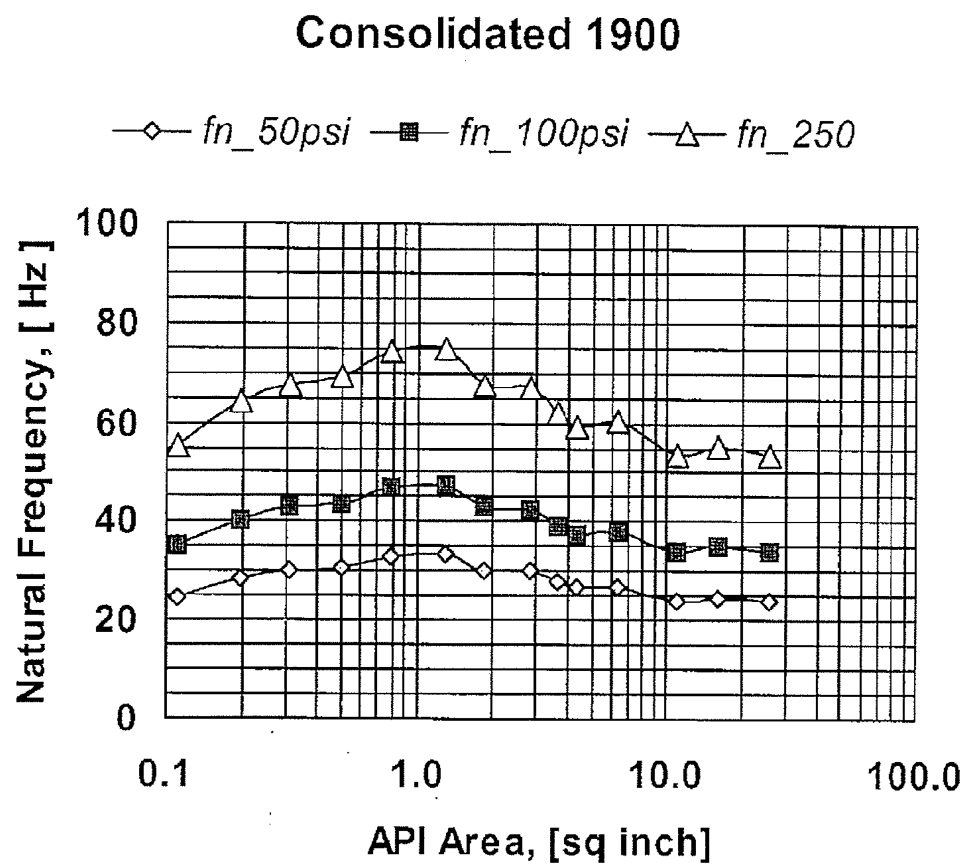
Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Things one would like to see from Relief Valve Manufacturers

(1) Valve spring constants and natural frequency data – if not in the general catalog at least with the purchased valve certification sheets

(2)Better relief valve Lift and Flow curves.

(3)Better data and guidance on relief valve blowdown

(4) Mechanical data on the number of on/off cycles that can be safely tolerated. Is this number 10 or 10^6 ; It makes a profound difference

Some Assumptions & Findings

(6) Fundamentally, I see no reason why Melhem and Fisher are not in excellent position to treat on/off relief valve performance in an effective manner.

Body Bowl Choking in Pressure Safety Valves

Robert D'Alessandro

DUG/EDUG Joint Meeting

Hamburg, Germany

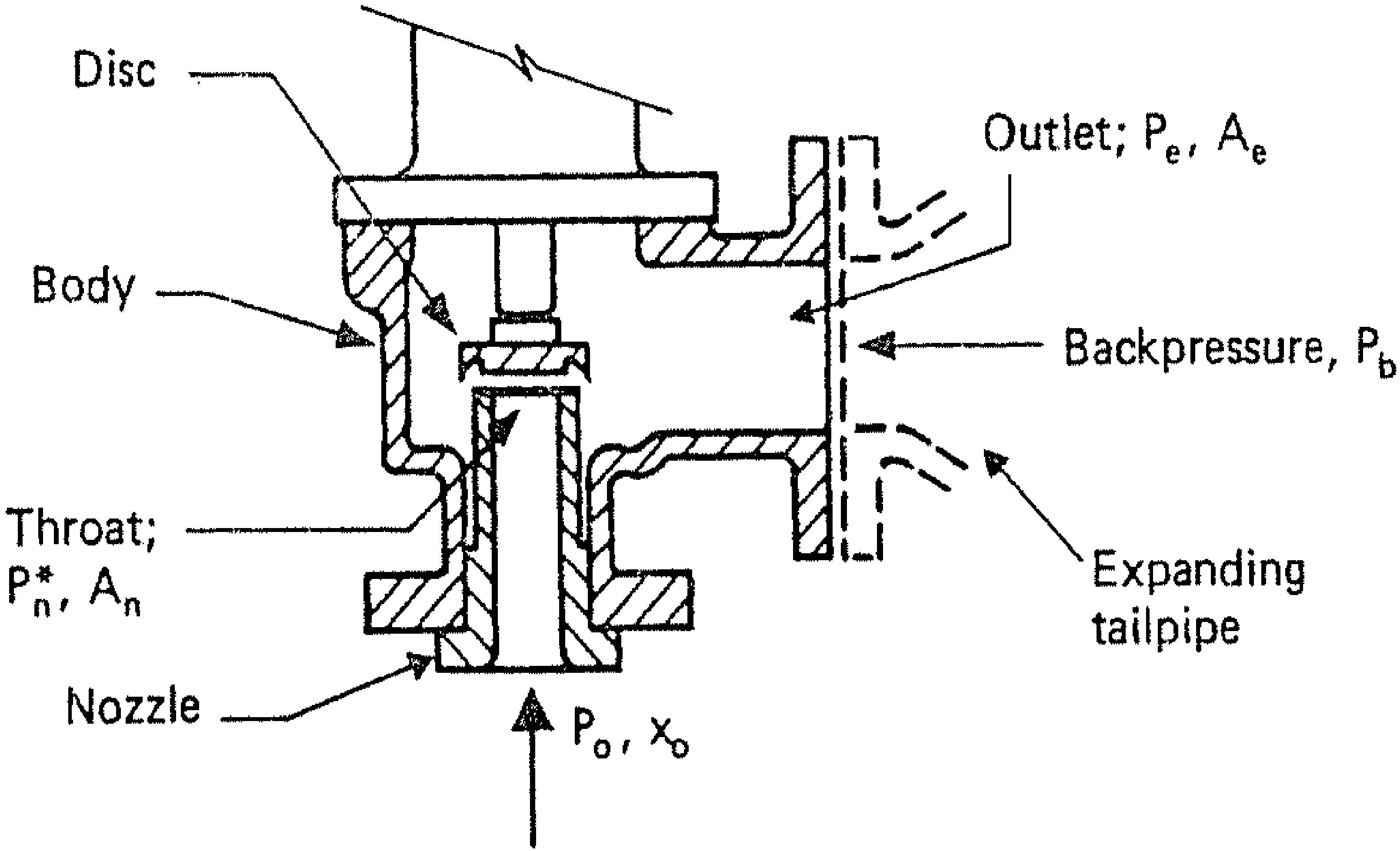
June 17, 2011

Background

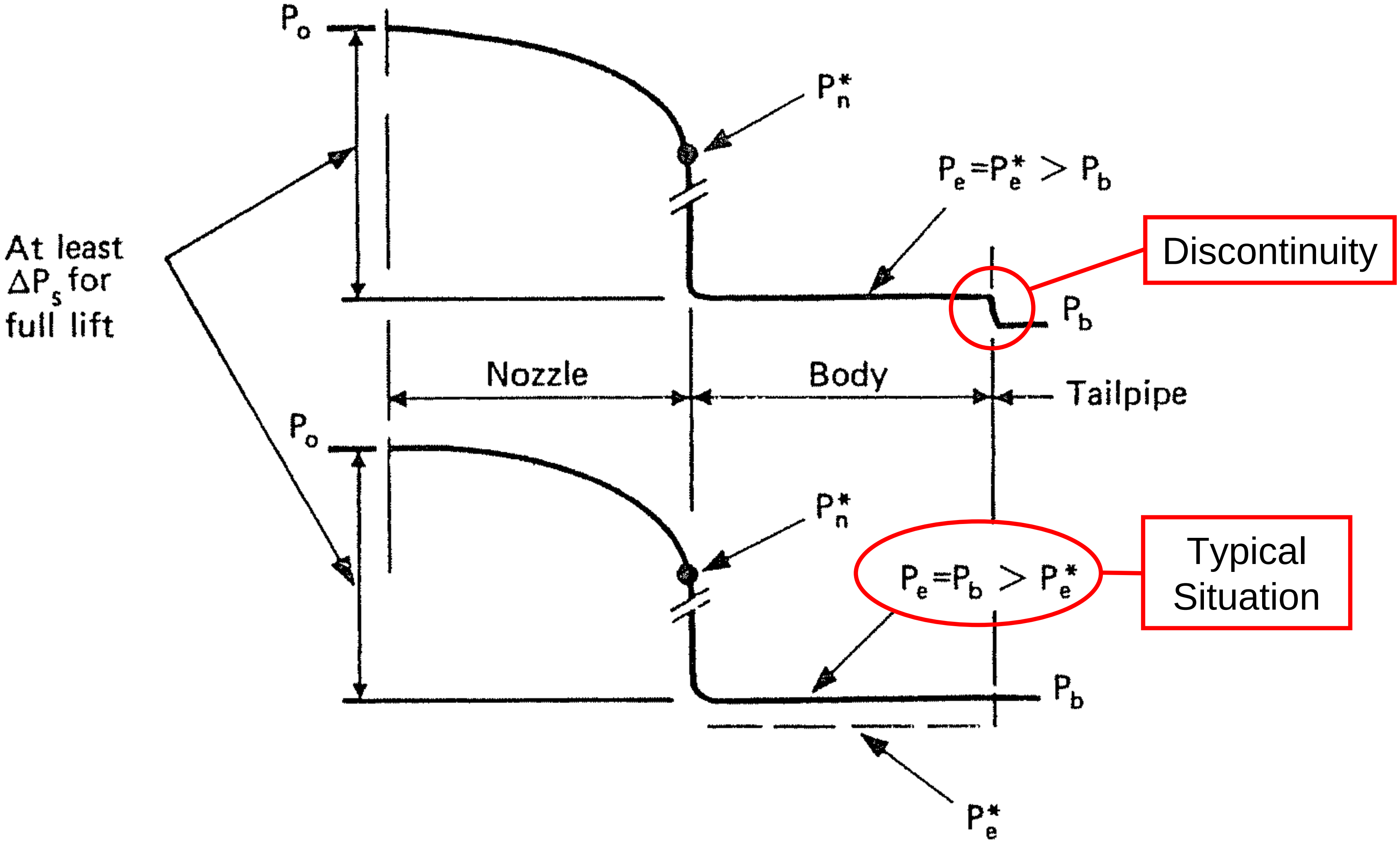
- Body bowl choking occurs when the pressure safety valve body causes a critical flow condition at the valve body outlet.
- This behavior was first noted in 1983 by Huff*
- “A secondary pressure in excess of the external back pressure can develop in the body of safety valves if the maximum flow condition is attained in the body outlet……..the contribution of this choking effect to the true back pressure on the disk of unbalanced valves with closed bonnets is not generally recognized”

* Huff, J., (1983). “Intrinsic Backpressure in Safety Valves”, API Proceedings - 48th Midyear Meeting of the Division of Refining - Los Angeles, California

Valve Schematic



Pressure Profile



Key Concept

The minimum body bowl exit pressure P_e^* is:

- Intrinsic to the valve geometry
- Dependent on the stagnation pressure only
- Independent of the tailpipe
- Not necessarily equal to the back pressure

Definitions

P_s = service set pressure

P_c = superimposed backpressure

ΔP_s = differential set pressure = P_s - P_c

F_0 = fractional overpressure = $\dfrac{P_0 - P_s}{\Delta P_s}$

$$P_0 = (1 + F_0)\Delta P_s + P_c$$

Criteria for Stability

For full lift and stable valve operation: $P_0 - P_e > \Delta P_s$

Therefore: $$P_e < P_c + \frac{F_0\left(P_0 - P_c\right)}{1 + F_0}$$

The valve will begin to close rapidly when the valve body exit pressure becomes equal to or greater than the pressure given by the above equation.

Flow Analysis

Consider an ideal gas with:

- Steady isentropic choked flow from stagnation to the valve nozzle
- Steady adiabatic flow from the valve nozzle to the valve outlet

Steady State Mass Balance

$$G_e A_e = G_n A_n$$

Isentropic Choked Flow to Valve Nozzle

$$G_n^2 = k\rho_n P_n^* \qquad \frac{P_n^*}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Flow Analysis

Ideal Gas

$$\frac{P_n}{\rho_n} = \frac{RT_n}{M} \quad and \quad \frac{P_e}{\rho_e} = \frac{RT_e}{M}$$

Adiabatic Flow from Valve Nozzle to Valve Exit

$$T_n \left\{ 1 + \frac{k-1}{2} M_n^2 \right\} = T_e \left\{ 1 + \frac{k-1}{2} M_e^2 \right\} \quad with \quad M_n = 1.0$$

Definition of Mach Number

$$M_e = \frac{u_e}{u_{se}} = \frac{G_e}{\rho_e u_{se}}$$

Flow Analysis

Combining

$$\left.\begin{aligned}\frac{P_e}{P_0} &= \left\{\left\{\frac{A_n}{A_e}\right\}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}\left\{\frac{1}{M_e}\right\}\left\{\frac{T_e}{T_n}\right\}\right\}^{\frac{1}{2}} \\ \frac{T_e}{T_n} &= \frac{\left\{1+\frac{k-1}{2}\right\}}{\left\{1+\frac{k-1}{2}M_e^2\right\}}\end{aligned}\right\} M_e = 1 \Rightarrow \frac{P_e^*}{P_0} = \left\{\frac{A_n}{A_e}\right\}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Flow Analysis + Stability Criteria

For stable operation stagnation pressure should be less than the value given by the following equation

$$P_0 < \frac{P_c}{(1+F_0)\frac{A_n}{A_e}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - F_0}$$

Example Calculation

Consider a gas with a k of 1.3 flowing to atmosphere through an 8T10 valve with a nozzle to discharge area ratio of 1/3.

$$P_0 < \frac{P_c}{\left(1+F_0\right)\frac{A_n}{A_e}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - F_0}$$

$$P_0 < \frac{14.7}{\left(1+0.1\right)\left(0.333\right)\left\{\frac{2}{1.3+1}\right\}^{\frac{1.3}{1.3-1}} - 0.1} = 146.9\,psia$$

Science For A Better Life

Stability Tests

Steve Kostos, Jung Kim

All trademarks are the property of their respective owners.

Agenda/ Content

- Stability Test for 6R10
  - P_{set} = 26.2 psig with and without inlet piping
  - Calculated Inlet Pressure Losses < Blowdown
- Stability Test for Refrigeration Relief Valves
  - P_{set} = 300 psig with and without selector valve
  - Calculated Inlet Pressure Losses with Selector Valve @ Stamped Capacity > Blowdown

Definitions

Stamped Capacity (Nameplate Capacity)

- The stamped capacity is 90% of the actual capacity at a pressure which does not exceed the set pressure by more than 10% or 3 psi., whichever is greater.

  (ASME Section VIII, div. 1)

Blowdown pressure

- Pressure at which the relief valve closes

#1 Introduction 6R10 tests

Revalidation project identified Farris 6R10 pressure safety valve with excessive pressure drop:

- P_{set} = 26.2 psig, vessel MAWP = 30 psig
- Piping / vessel modifications would have had significant impact on schedule due to special materials of construction

➢ Decision was made to test valve stability as alternative solution to extensive piping / vessel changes

➢ Valve was removed and tested at Anderson Greenwood Test Facility in El Campo, TX

➢ Tested with and without piping stack

Calculated Inlet Pressure Losses for 6R10 + Piping Stack

- Case 1: 0.94 psi or 3.77% (>3%) Component A (Hydrocarbon, Cp/Cv lower value)
- Case 2: 0.95 psi or 3.80% (>3%) Component A
- Case 3: 1.10 psi or 4.39% (>3%) Component B
- Case 4: 1.01 psi or 4.05% (>3%) Component A and C (Hydrocarbon, Cp/Cv lower value)

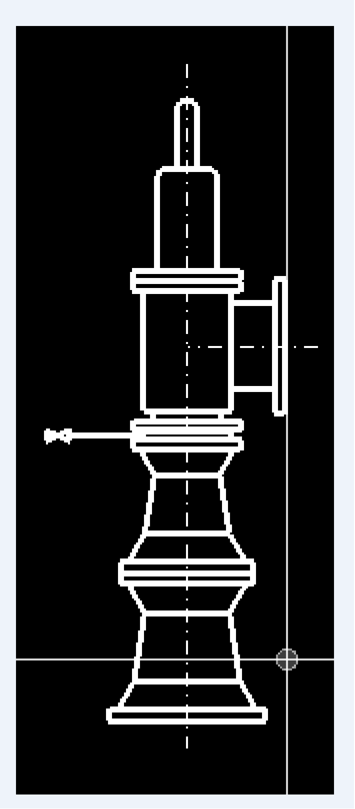
➢ Average inlet pressure loss for four cases is 4.01% (Actual fluid at stamped capacity)

Valve on Test Vessel without Piping Stack

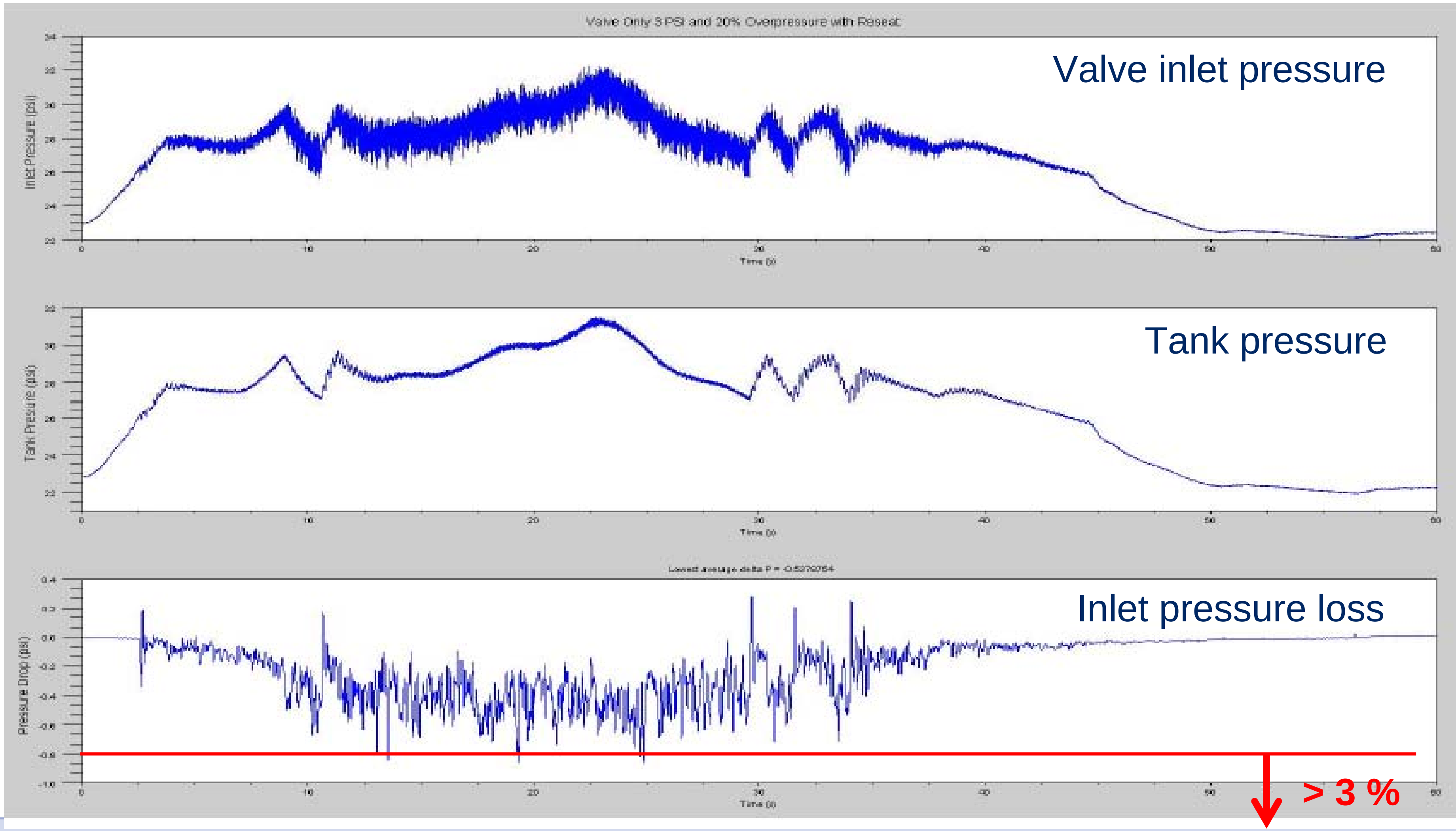
- Valve set pressure: 26.2 psig
- Inlet pressure loss was measured at 3 psi overpressure (29.2 psig) and at 20% overpressure (31.4 psig)

➢ Measured inlet pressure loss is 0.5 psi or 1.9% of set pressure

➢ Measured blowdown pressure is 22.4 psig giving a blowdown pressure percentage of 14.5%

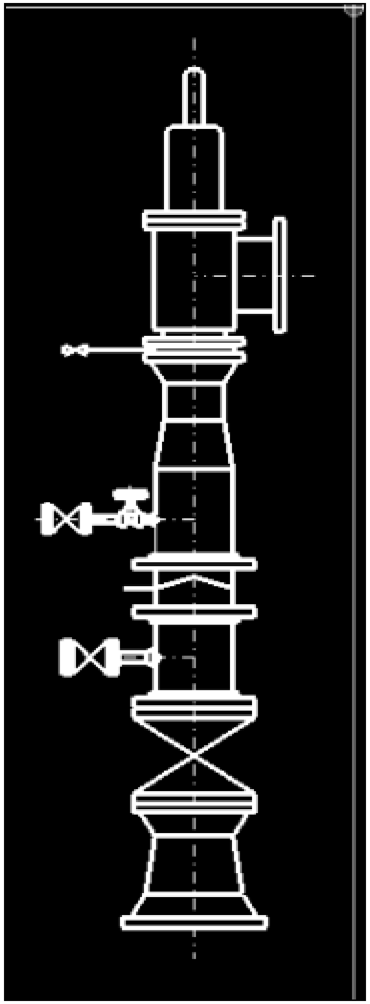


Pressure Readings for Test <u>without</u> Piping Stack



Actual Piping Arrangement

- 6R10 PSV
- Pitot Tube
- Pipe Spool with 6”X8” Reducer
- Rupture Disc
- Pipe Spool 8”
- Gate Valve 8”



There were four overpressure scenarios identified:

- External Fire
- Coolant Failure
- Instrument Failure
- Chemical Reaction.

➢ In all cases the inlet pressure loss was calculated near 4%.

Valve on Test Vessel with Piping Stack

- Valve set pressure: 26.2 psig
- Inlet pressure loss was measured at 20% overpressure (31.4 psig)

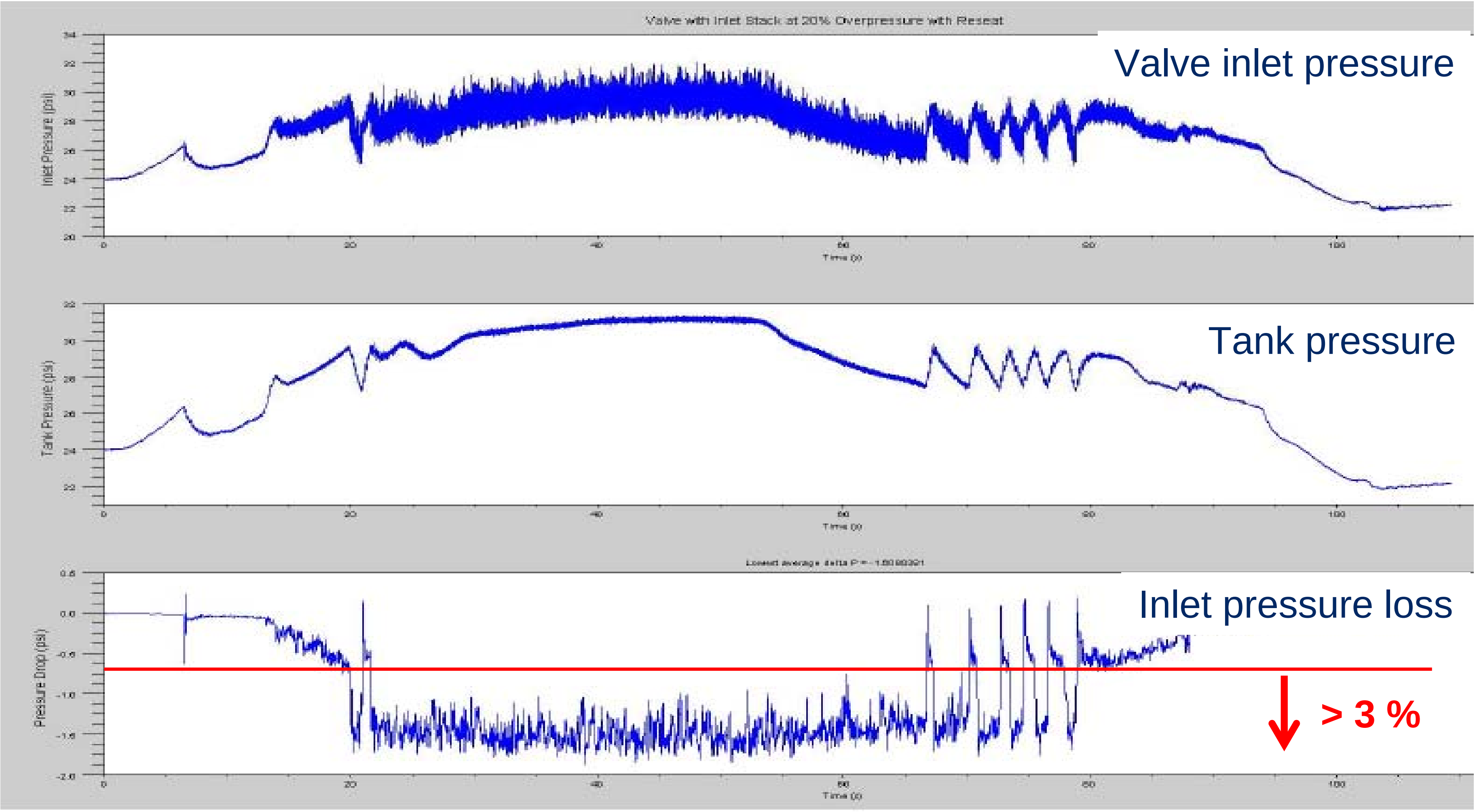
- Measured inlet pressure loss is 1.6 psi or 6.1% of set pressure
- Measured blowdown pressure is 22.4 psig giving a blowdown pressure percentage of 14.5%

Valve on Test Vessel with Piping Stack

- Valve was installed on tank with gate valve, rupture disc and pipe spools.
- Raised inlet pressure.
- Rupture disc burst at 19 psig.
- Stamped capacity 10,600 scfm.
- Flow capacity at 3 psi overpressure (29.2 psig tank pressure) was 7692 scfm.
- At 20% overpressure (31.4 psig tank pressure) the flow rate was 12,878 scfm.

➢ **Valve did not exhibit any instability with or without piping stack.**

Pressure Readings for Test with Piping Stack



Bayer Technology Services

Conclusions for 6R10 Test

- Inlet pressure loss measured during the valve test without the stack was 0.5 psig or 1.9% of set pressure

- Inlet pressure loss measured during the valve test with the stack was 1.6 psig or 6.1% of set pressure

- There was no observed unstable behavior during any of the test protocol.

#2 Introduction Refrigeration Valves

Revalidation project identified Parker H3 relief valves installed on selector valves with unknown Cv. Manufacturer would not certify Cv of their selector valve:

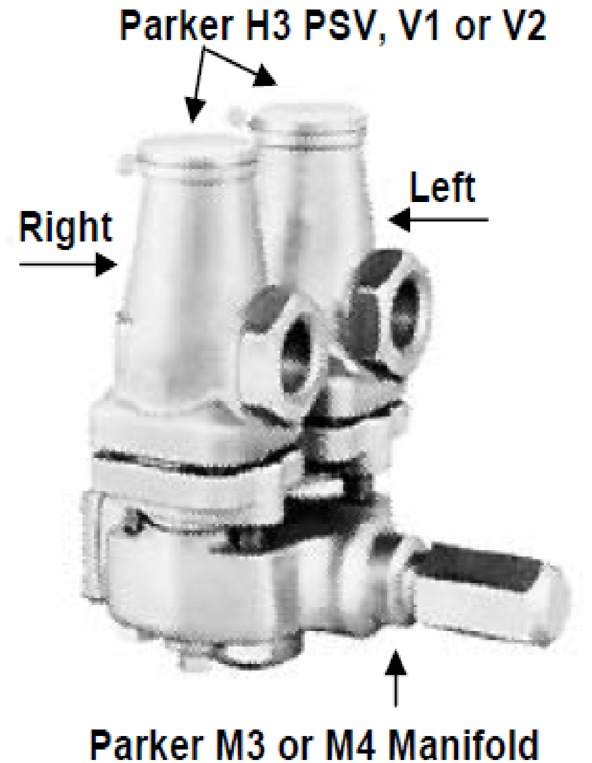
- P_{set} = 300 psig with ±3% tolerance
- Uncertain selector valve Cv's meant the capacities of installed relief valves could not be verified.
- Calculated inlet pressure loss was 53% based on stamped capacity

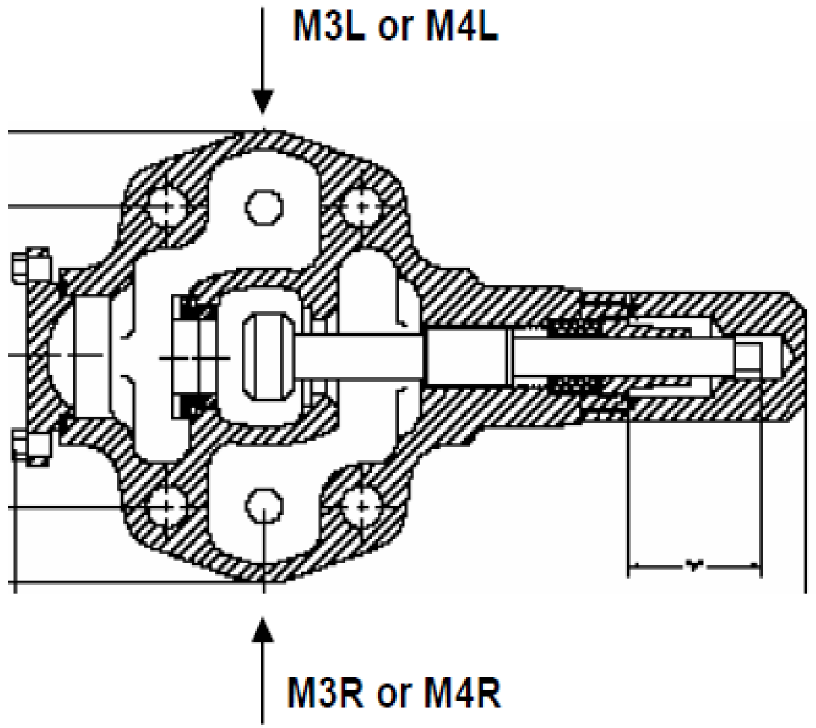
➢ Decision was made to test selector valves with relief valves at the Dresser Consolidated facility in Alexandria, LA.

➢ Two Parker H3 valves were tested with two sizes of selector valves at the Dresser Consolidated facility in Alexandria, LA.

Refrigeration Relief Valves

PSV Capacity Tests for a Parker H3 (3/4” x1.1/4”) on an M3 (1”x1”x1”) and on an M4 (1.1/4”x1.1/4”x1.1/4”) dual stop manifold.





Test Results
Refrigeration Valves

Case	Description	Actual capacity (required capacity = 2408 scfm air)	Actual inlet pressure loss / Set pressure	Blowdown pressure percentage	Stability Behavior
#1	V1	**2841.5**		**38.5%**	stable
#2	V2	**2819.6**		**37.9%**	stable
#3	V1 on M3R	**2099.6**	**31.8%**	**38.5%**	stable
#4	V1 on M3L	**2301.0**		**38.5%**	stable
#5	V2 on M3R	**2078.4**	**31.5%**	**37.9%**	stable
#6	V2 on M3L	**2292.5**		**37.9%**	stable
#7	V1 on M4R	**2491.5**		**38.5%**	stable
#8	V1 on M4L	**2592.5**		**38.5%**	stable
#9	V2 on M4R	**2474.7**		**37.9%**	stable
#10	V2 on M4L	**2561.4**		**37.9%**	stable

- **Excessive pressure drop effected capacity of tested valves but not stability.**

Summary

Farris 6R10 tests

- No instability at an actual 6.1% inlet pressure loss (calculated 4% at stamped capacity)

Parker H3 Refrigeration with and without selector valves:

- No instability at an actual 32% inlet pressure loss (calculated > 53% at stamped capacity) but capacity issues

Outlook discussion

Current results support one methodology discussed in the past:

- Calculate the inlet pressure drop at Stamped Capacity*.
- If inlet pressure drop is less than Blowdown, then the valve is stable.

*The actual relieving capacity at closing conditions is not greater than the stamped capacity.

DIERS Users Group Mtg.
October, 2010 Reno Nevada

"Odds and Ends"
Relief Valve Stability-Part 3
Is There Life Outside of the
3% Inlet Pressure Loss Rule

Michael A. Grolmes

Centaurus Technology, Inc.

4590 Webb Road

Simpsonville, KY 40067

502-243-9678

ctimag@att.net

C T I

Centaurus Technology, Inc.

Old Native American wisdom teaches that when one discovers that one is riding a dead horse, one should dismount and find a fresh horse. However, modern business wisdom has developed many alternative courses of action. Among these are:

Get a bigger whip.

Change riders.

Appoint a committe to study the dead horse.

Visit other sites to see how <u>they</u> ride dead horses.

Develop new training methods to increase dead horse riding skills.

Harness several dead horses together to improve performance.

Study alternative uses for the dead horse.

Promote the dead horse to a higher management position.

Develope motivational slogans and mission statements such as;

> **Our horse is better, faster, and cheaper dead.**
>
> **No horse is too dead to whip.**
>
> **This is the way we have always ridden horses.**

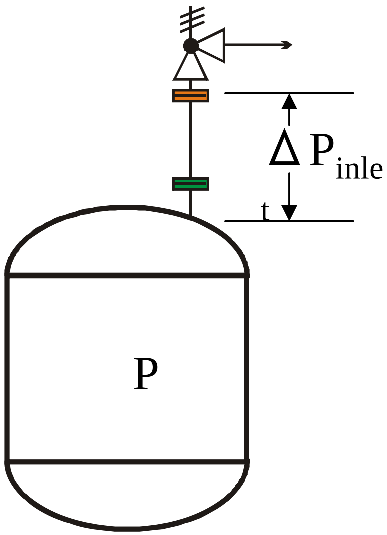
Some Background

- **Berwanger et. al,. Proc. Safety Prog.; Vol. 19, No. 3 reported the following:**
- **1072 PSVs in a 13049 item sample pool had >3% inlet pressure drop (8.2%)**
- **1606 PSVs in the same sample pool had excess outlet pressure drop, (12.3%)**
- **"----various experts in this area hold very strong yet contradictory opinions on this topic."**

Some Background

- My own comments on the above:
- For older plant sites, my guess that only 8% of PSVs have >3% inlet delta P is not typical. I have seen variations from 10% to over 80% in specific instances.
- The existence of contradictory opinions is not good and should be addressed
- The existing 3% API guideline is not sufficiently robust but does leave open the option for "engineering analysis" to justify exceptions to the 3% guideline.
- This effort is an attempt to outline such an analysis

Relief Valve Inlet Pressure Loss



$$LET:$$

$$DelP_{inlet} = x\ P_{set\_g} \quad where \quad 0 \le x \le 0.1$$

$$and$$

$$P_o = 1.1\ P_{set\_g} + P_{atm}$$

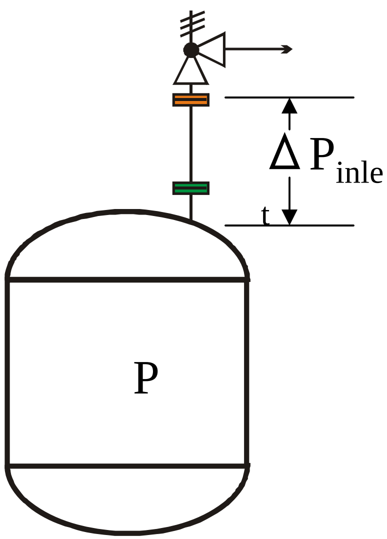
$$Then$$

$$DelP_{inlet} = \frac{1}{2} Rho\, U^2\ L_1^* \quad where \quad L_1^* = K_{ent} + K_{RD} + 4 f\,(L/D)_{eq}$$

$$but$$

$$Rho\ U^2 = \frac{1}{Rho}\left(\frac{C_d A_o G_{noz}}{A_1}\right)^2$$

Relief Valve Inlet Pressure Loss
(continued)



$$Now:$$

$$DelP_{inlet} = \frac{1}{2}\left[\frac{C_d\ A_o}{A_1}\right]^2 \frac{G^2_{noz}}{Rho} L_1^* = x\ P_{set\_g}$$

also define

$$A_1^* = \frac{A_1}{C_d\ A_o}$$

So that

$$\boxed{\frac{L_1^*}{(A_1^*)^2} \le \frac{2\,Rho\ x\ P_{set\_gage}}{G^2_{noz}}}$$

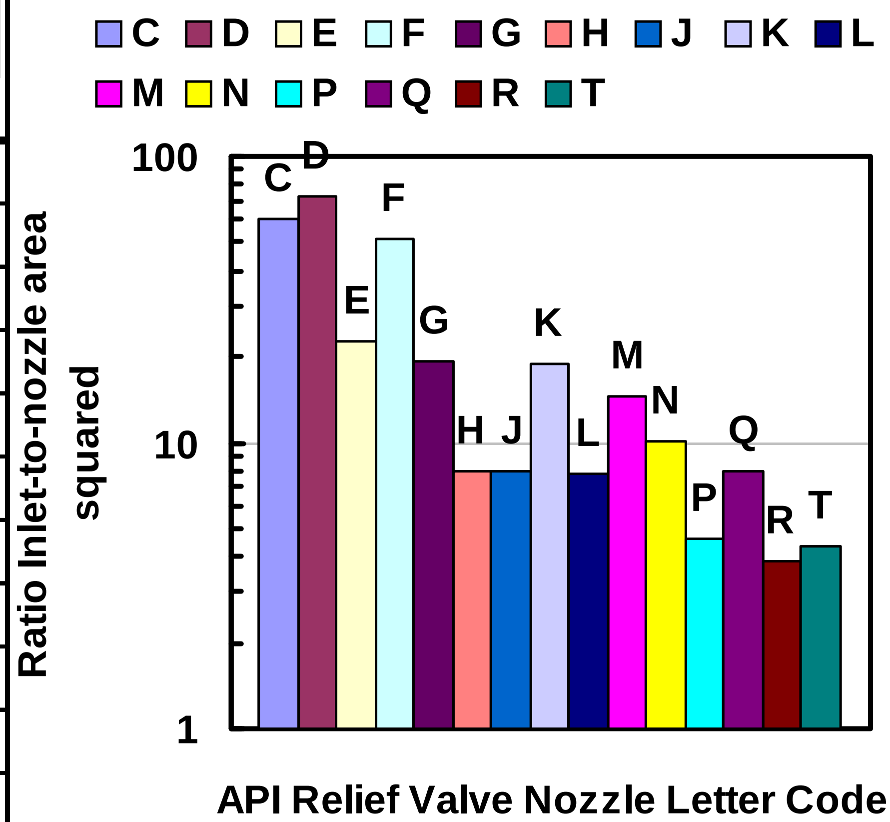
Relief Valve Inlet Pressure Loss

(summary table)

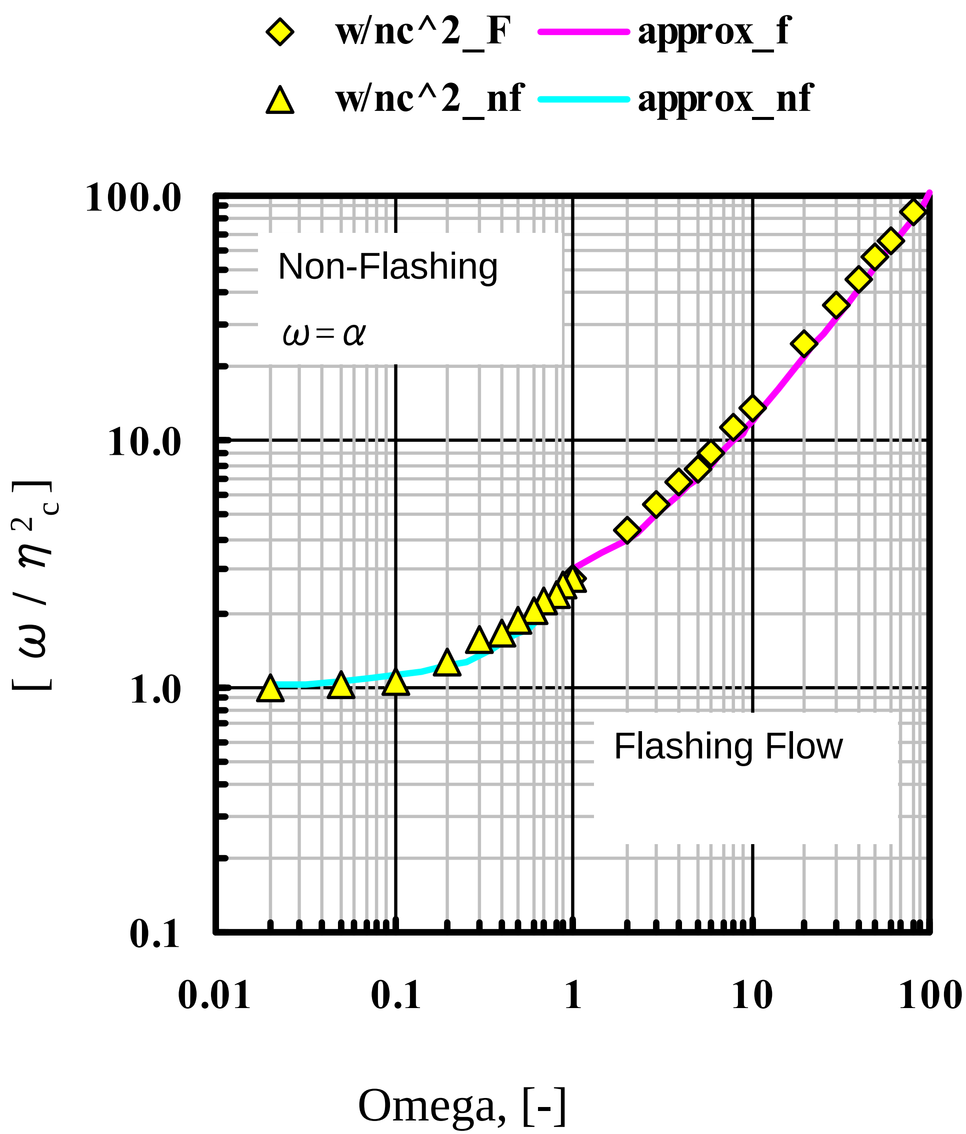
CASE	G_{noz}	$L_1^* / (A_1^*)^2$
LIQUID	$\sqrt{2\,\rho\,1.1\,P_{set\_gage}}$	$\dfrac{x}{1.1}$
GAS	$\dfrac{\varepsilon\, P_o}{\sqrt{z\,R\,T_o\,/\,Mw}}$	$\dfrac{2\,x}{\varepsilon^2\left[1.1+\dfrac{P_{atm}}{P_{set\_gage}}\right]}$
Two-Phase	$\dfrac{\eta_c}{\sqrt{\omega}}\,\sqrt{\rho_{2\varphi}\,P_o}$	$\dfrac{2\,x\left(omega\,/\,eta_c^2\right)}{\left[1.1+\dfrac{P_{atm}}{P_{set\_gage}}\right]}$

Relief Valve Nozzle	Nozzle Area	Inlet Area	Outlet Area	$\left[\frac{A_1}{C_d A_o}\right]^2$	$\left[\frac{A_2}{C_d A_o}\right]^2$
	[sq. inch]	[sq. inch]	[sq. inch]	[-]	[-]
C	0.074	0.533	0.864	57.54	151.14
D	0.110	0.864	3.356	68.40	1031.12
E	0.196	0.864	3.356	21.54	324.77
F	0.307	2.036	3.356	48.73	132.38
G	0.503	2.036	4.909	18.15	105.53
H	0.785	2.036	7.393	7.45	98.27
J	1.287	3.356	7.393	7.53	36.56
K	1.838	7.393	12.730	17.93	53.15
L	2.853	7.393	12.730	7.44	22.06
M	3.600	12.730	28.890	13.86	71.36
N	4.340	12.730	28.890	9.53	49.10
P	6.380	12.730	28.890	4.41	22.72
Q	11.050	28.890	50.027	7.57	22.71
R	16.000	28.890	50.027	3.61	10.83
T	26.000	50.027	78.540	4.10	10.11

Note we use C_d = 0.95



These are A* values for conventional safety relief valves with 150# flanges



Approximation for:

Non-Flashing Flow

$$\frac{\omega}{\eta_c^2} = \exp(\omega)$$

Approximation for:

Flashing Flow

$$\frac{\omega}{\eta_c^2} = \omega + 2$$

Example Illustrations

Inlet Loss Limits Matrix

P_{set} = 40 psig; 3% inlet loss

Square-cut entrance

Ke = 0.5

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	40	0.03	**HEXANE**
1" inlet	Liquid	air	2-phase (f)
L_star	0.49	1.72	16.8
L[inch]	0	87	816
PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	40	0.03	**HEXANE**
4" inlet	Liquid	air	2-phase (f)
L_star	0.12	0.42	4.1
L[inch]	0	0	713

Inlet Loss Limits Matrix

P_{set} = 150 psig; 3% inlet loss

Square-cut entrance

Ke = 0.5

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	150	0.03	**HEXANE**
1" inlet	Liquid	air	2-phase (f)
L_star	0.49	2.11	11
L[inch]	0	115	525

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	150	0.03	**HEXANE**
4" inlet	Liquid	air	2-phase (f)
L_star	0.12	0.51	2.7
L[inch]	0	3	432

Inlet Loss Limits Matrix

P_{set} = 150 psig; 5% inlet loss

Square-cut entrance

Ke = 0.5

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	150	0.05	**HEXANE**
1" inlet	Liquid	air	2-phase (f)
L_star	0.81	3.5	18.4
L[inch]	22	215	892
PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	150	0.05	**HEXANE**
4" inlet	Liquid	air	2-phase (f)
L_star	0.2	0.85	4.4
L[inch]	0	100	786

Inlet Loss Limits Matrix

P_{set} = 40 psig; 5% inlet loss

Square-cut entrance

Ke = 0.5

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	40	0.05	**HEXANE**
1" inlet	Liquid	air	2-phase (f)
L_star	0.81	2.87	28
L[inch]	22	169	1377
PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	40	0.05	**HEXANE**
4" inlet	Liquid	air	2-phase (f)
L_star	0.2	0.69	6.8
L[inch]	0	55	1255

Takaway Observations

(a) Some inlet pipe configurations of the same diameter as the relief valve inlet will not work with a "square cut" vessel inlet or the presence of a rupture disk or both. A larger diameter inlet pipe may be required. The problem increases in severity with increasing PSV size.

(b) All Liquid flow is more restrictive than all gas flow which is more restrictive than "flashing" two phase flow. Therefore evaluation for all gas flow is sufficient for two-phase flow at the same set pressure.

(c) There is a significant difference between allowance for 5% inlet pressure loss and 3%.

Example General Applications

Set-up Parameters

x = 0.03 (3% rule)

ε = 0.65 (useful in most cases)

P_{atm} = 14.7 psia

P_{set_g} = 50 psig

For Flashing Two-Phase Flow:

ω = 20 and η_c = 0.895

(ω / η_c^2) = 25 (compare with ω+2)

For Non-Flashing Two-Phase Flow:
ω = 0.4 and η_c = 0.4
(ω / η_c^2) = 1.25 (compare with e^{ω} = 1.4)

Case	$L_1^*/(A_1^*)^2$ Must be < =
Gas	0.102
Liquid	0.0273
2-P Flashing	1.076
2-P Non-Flashing	0.054

Example Application (continued)

Now, Extend Example to 4P6 or 8T10

Relief valves with

$(A_1^*)^2 = 4.4$ & 4.1

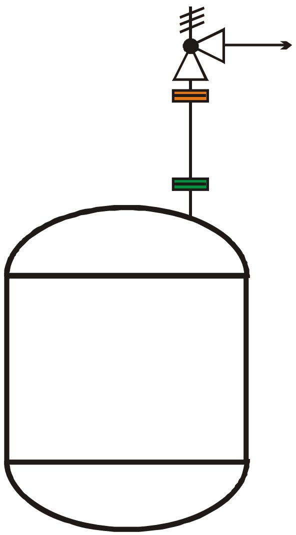
Respectively,

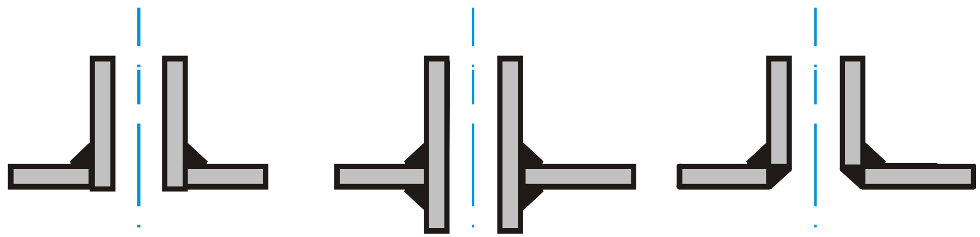
At 50 psig

Set pressure

Case	L* Must be < =	$(L/D)_{eq}$ Must be < =
Gas	≈ 0.43	0 With $K_{ent}=0.5$
Liquid	≈ .12	0 With $K_{ent}=0.5$
2-P Flashing	3.7	106
2-P Non-Flashing	≈ .33	0 With $K_{ent}=0.5$

Entrance Configurations and Entrance Loss Coefficients



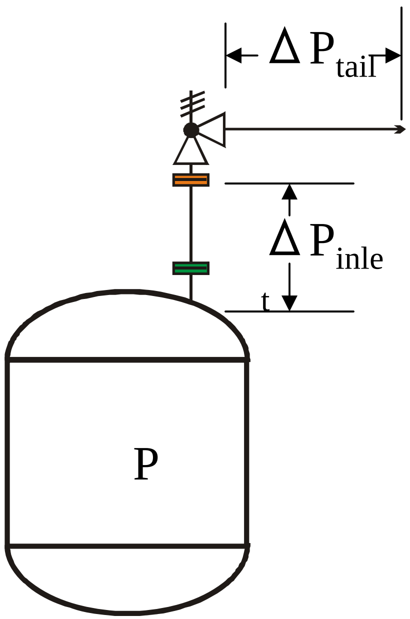


Square Cut Entrance $K_e = 0.5$	Re-entrant Entrance $K_e = 0.8$	Rounded Entrance $K_e = 0.04$

One could be dead in the water with either square-cut or re-entrant configuration.

Further, in older plants one finds many reentrant nozzles on relief valve ports.

Relief Valve Tail Pressure Loss



$$DelP_{tail} = \frac{1}{2}\left[\frac{C_d \; A_o}{A_2}\right]^2 \frac{G_{noz}^2}{Rho} L^* = y \; P_{set\_g}$$

where "y" is typically 0.10

also define

$$A_2^* = \frac{A_2}{C_d \; A_o}$$

So that

$$\boxed{\frac{L_2^*}{(A_2^*)^2} \leq \frac{2\,\overline{Rho}\; y \; P_{set\_gage}}{G_{noz}^2}}$$

Relief Valve Outlet Pressure Loss

(summary table)

CASE	G_{noz}	$L_2^* / (A_2^*)^2$
LIQUID	$\sqrt{2\,\rho\,1.1 P_{set\_gage}}$	$\dfrac{y}{1.1}$
GAS	$\dfrac{\varepsilon\, P_o}{\sqrt{z\,R\,T_o / Mw}}$	$\left(\dfrac{2}{\varepsilon^2}\right)\left\{1+\dfrac{1}{2}\left(\dfrac{y\,P_{set\_g}}{P_{atm}}\right)\right\}\left(\dfrac{y\,P_{set\_g}}{P_{atm}}\right)\left(P_{atm} / P_o\right)^{((k+1)/k)}$
Two-Phase	$\dfrac{\eta_c}{\sqrt{\omega}}\sqrt{\rho_{2\varphi}\,P_o}$	$\dfrac{2\left(\dfrac{\omega}{\eta_c^2}\right)y\left\{1+\dfrac{1}{2}\left(\dfrac{y\,P_{set\_g}}{P_2}\right)\right\}}{\left[1+\omega\left(\left(P_o / P_2\right)-1\right)\right]\left[1.1+\dfrac{P_{atm}}{P_{set\_g}}\right]}$

Note: "y" typically = 0.1, for conventional relief valves.

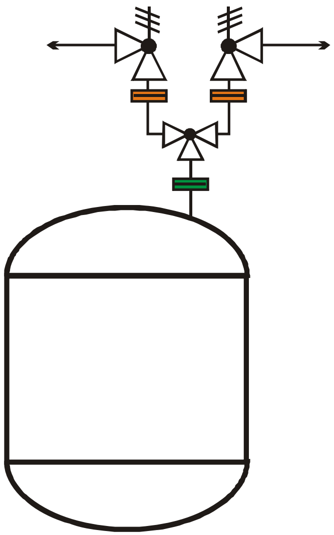
Outlet Loss Limits Matrix
P_{set} = 40 psig; 10% tail loss

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	40	0.1	**HEXANE**
2" outlet	Liquid	air	2-phase (f)
L_star	30	44	14
L[ft]	364	545	121
PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	40	0.1	**HEXANE**
6" outlet	Liquid	air	2-phase (f)
L_star	2	3	1
L[inch]	75	112	25

Outlet Loss Limits Matrix
P_{set} = 150 psig; 10% tail loss

PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"E"	150	0.1	**HEXANE**
2" outlet	Liquid	air	2-phase (f)
L_star	30	32	9
L[ft]	364	399	77
PSV Nozzle	**P_set [psig]**	**Del P / P_set [-]**	
"P"	150	0.1	**HEXANE**
6" outlet	Liquid	air	2-phase (f)
L_star	2	2.3	choked at
L[inch]	75	82	22 psig

Entrance Loss due to 3-Way Valve



Relief Valve Nozzle Letter Size Code	API Nozzle Area A_o	inlet x outlet flange pipe size	Brand#1 3-way valve	Inlet Del P P	Brand#2 3-way valve	Inlet Del P P
	[sq. inch]	150#	Cv	[%]	Cv	[%]
C	0.074	3/4 x 1	7	3.49	14	0.87
D	0.110	1 x 2	20	0.94	22	0.78
E	0.196	1 x 2	20	3.00	22	2.48
F	0.307	1 1/2 x 2	40	1.84	57	0.91
G	0.503	1 1/2 x 2 1/2	40	4.94	57	2.43
H	0.785	1 1/2 x 3	40	12.02	57	5.92
J	1.287	2 x 3	70	10.55	110	4.27
K	1.838	3 x 4	100	10.54	260	1.56
L	2.853	3 x 4	100	25.41	260	3.76
M	3.600	4 x 6	175	13.21	446	2.03
N	4.340	4 x 6	175	19.20	446	2.96
P	6.380	4 x 6	175	41.49	446	6.39
Q	11.050	6 x 8	350	31.11	-	-
R	16.000	6 x 8	350	65.23	-	-
T	26.000	8 x 10	475	93.52	-	-

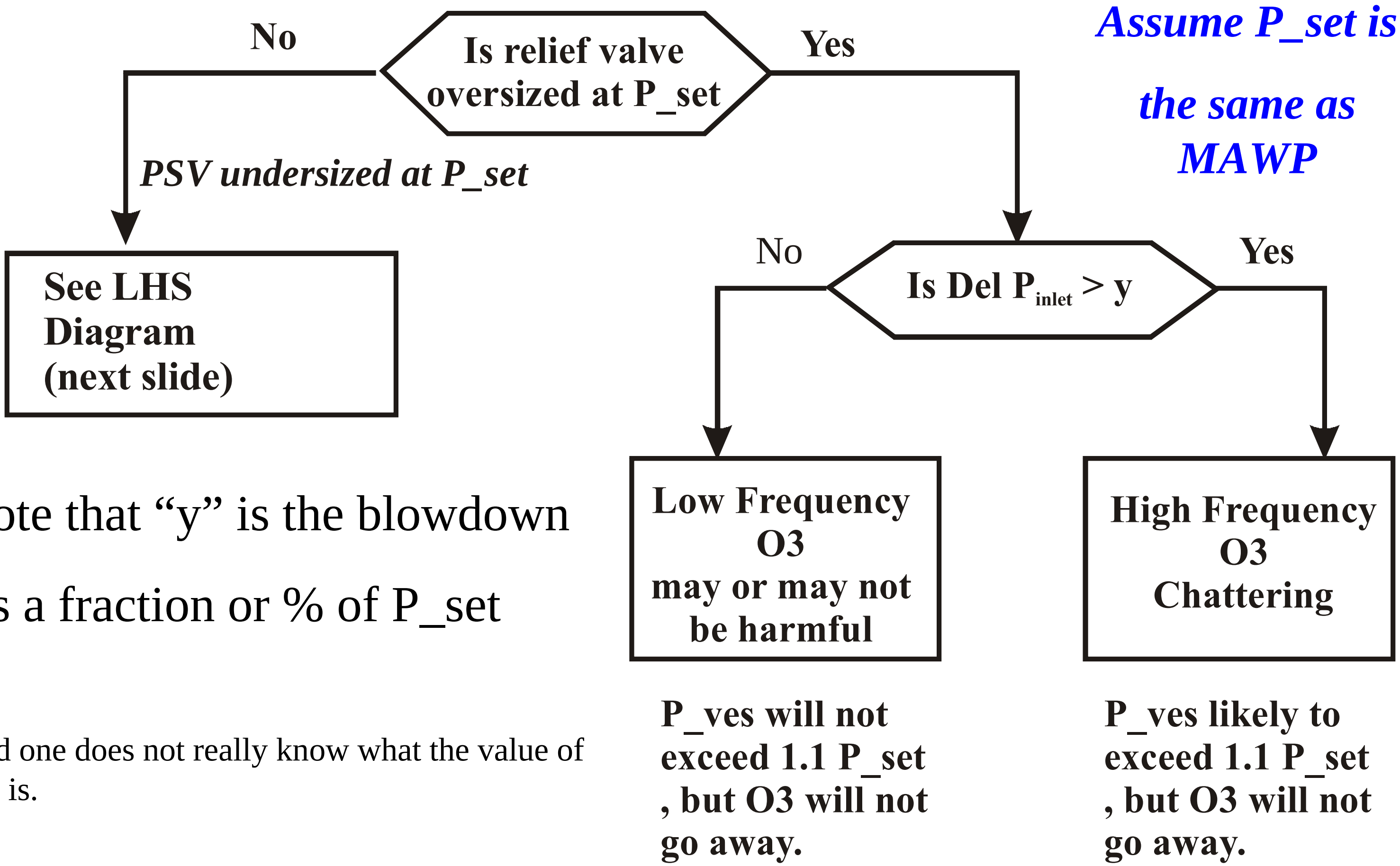
3% Rule Summary:

- Many entrance configurations won’t work with the vessel nozzle size the same as the relief valve inlet size.
- Many 3-way valves won’t work with the 3-way valve size the same as the relief valve inlet size.
- So how important is the 3% rule?
- What has this to do with relief valve on-off instability?

Relief Valve Instability
or
ON-OFF OPERATION (*O3*)

- *O3* can occur if valve is oversized for the demand
- *O3* can occur with excess inlet pressure loss
- It is generally believed that *O3* can be harmful to the relief valve if such leads to cyclic on-off operation at the relief valve natural frequency – or, for extended periods or repeated actuations without maintenance.
- Controlled tests as reported in the available literature are remarkably silent on this point.

Possible Valve Sizing Outcomes



Note that "y" is the blowdown

As a fraction or % of P_set

And one does not really know what the value of "y" is.

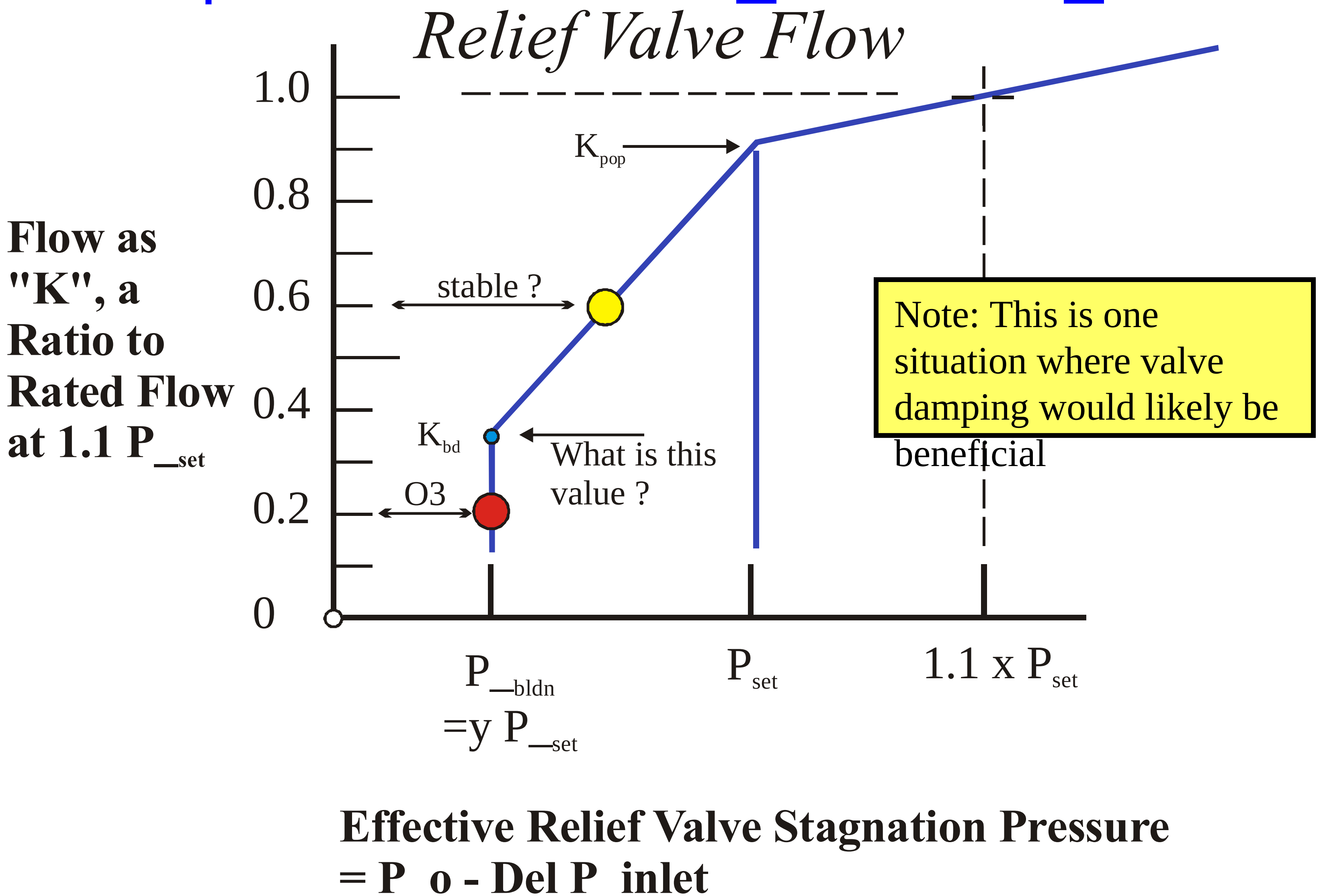
Most pressure relief area sizing

Calculations – that are properly executed

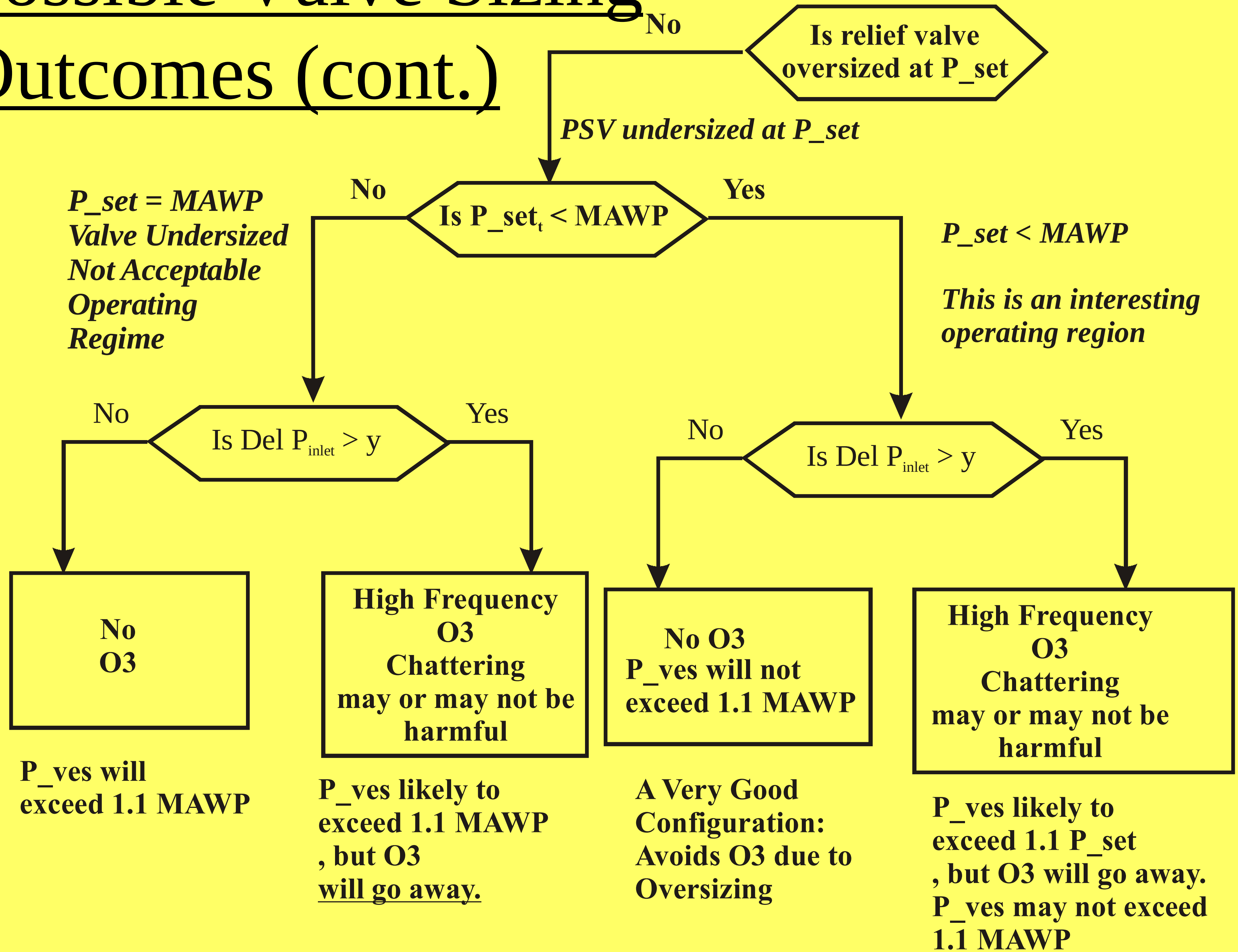
End up with oversized relief devices upon selection. This enhances the opportunity for O3

Relief Valve Nozzle Letter Size Code	API Nozzle Area A_o	Area ratio increase to the next larger size
	[sq. inch]	
C	0.074	
D	0.110	1.486
E	0.196	1.782
F	0.307	1.566
G	0.503	1.638
H	0.785	1.561
J	1.287	1.639
K	1.838	1.428
L	2.853	1.552
M	3.600	1.262
N	4.340	1.206
P	6.380	1.470
Q	11.050	1.732
R	16.000	1.448
T	26.000	1.625

Moderately Oversized Relief Valves are likely to Operate between P_set and P_bd



Possible Valve Sizing Outcomes (cont.)



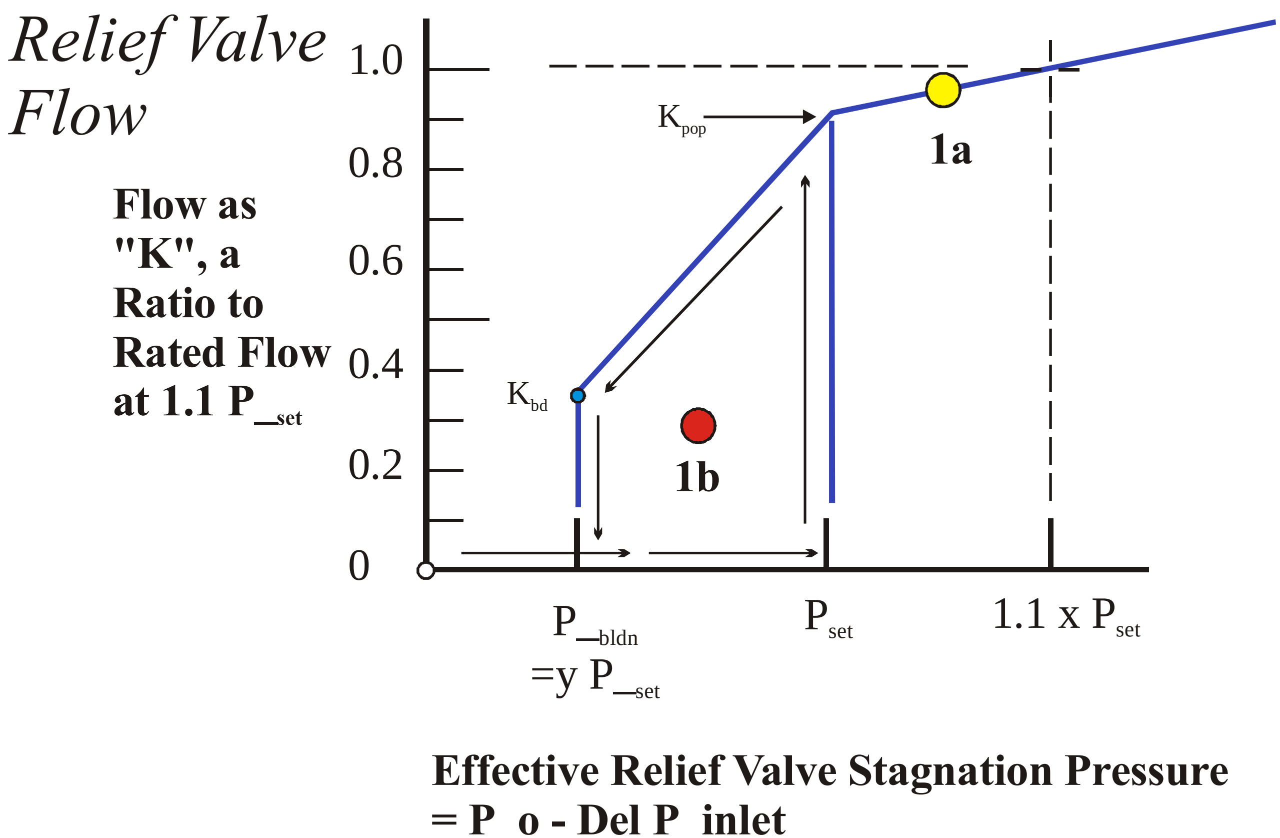
Three Case Illustrations

Case # 1 Relief Valve is just-right sized – with P_set = MAWP

Q_required = Q_rated so that K = 1.0 at P_set

Case 1a: Del P_inlet < y P_set

Case 1b: Del P_inlet > y P_set



Case # 2 Relief Valve is over sized – with P_set = MAWP

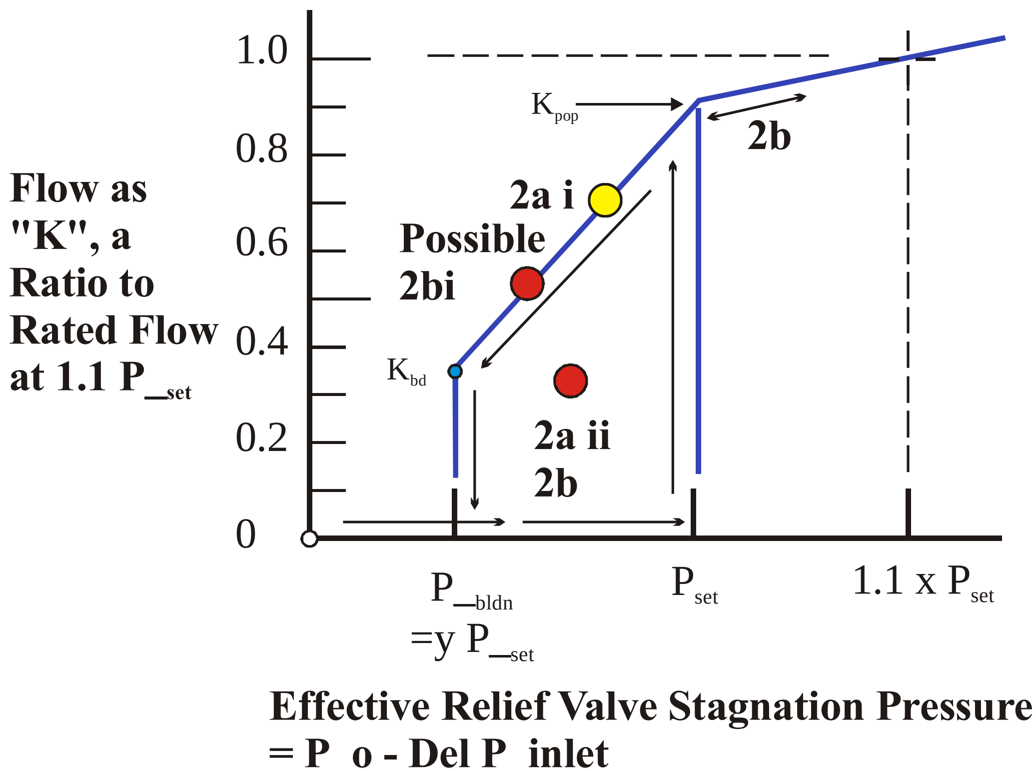
Q_required < Q_rated so that K < 1.0 at P_set

Case 2a: Del P_inlet < y P_set at K=1 Case 2a i: Could be stable if K_required > K_bd   Case 2a ii: Could be cyclic if K_required < K_bd

Case 2b: Del P_inlet > y P_set at K=1

The difference between 2a ii and 2b is one of frequency. There is 2b operating point similar to 2ai if Del P_in at K_required does reach P_bd

Here damping would help



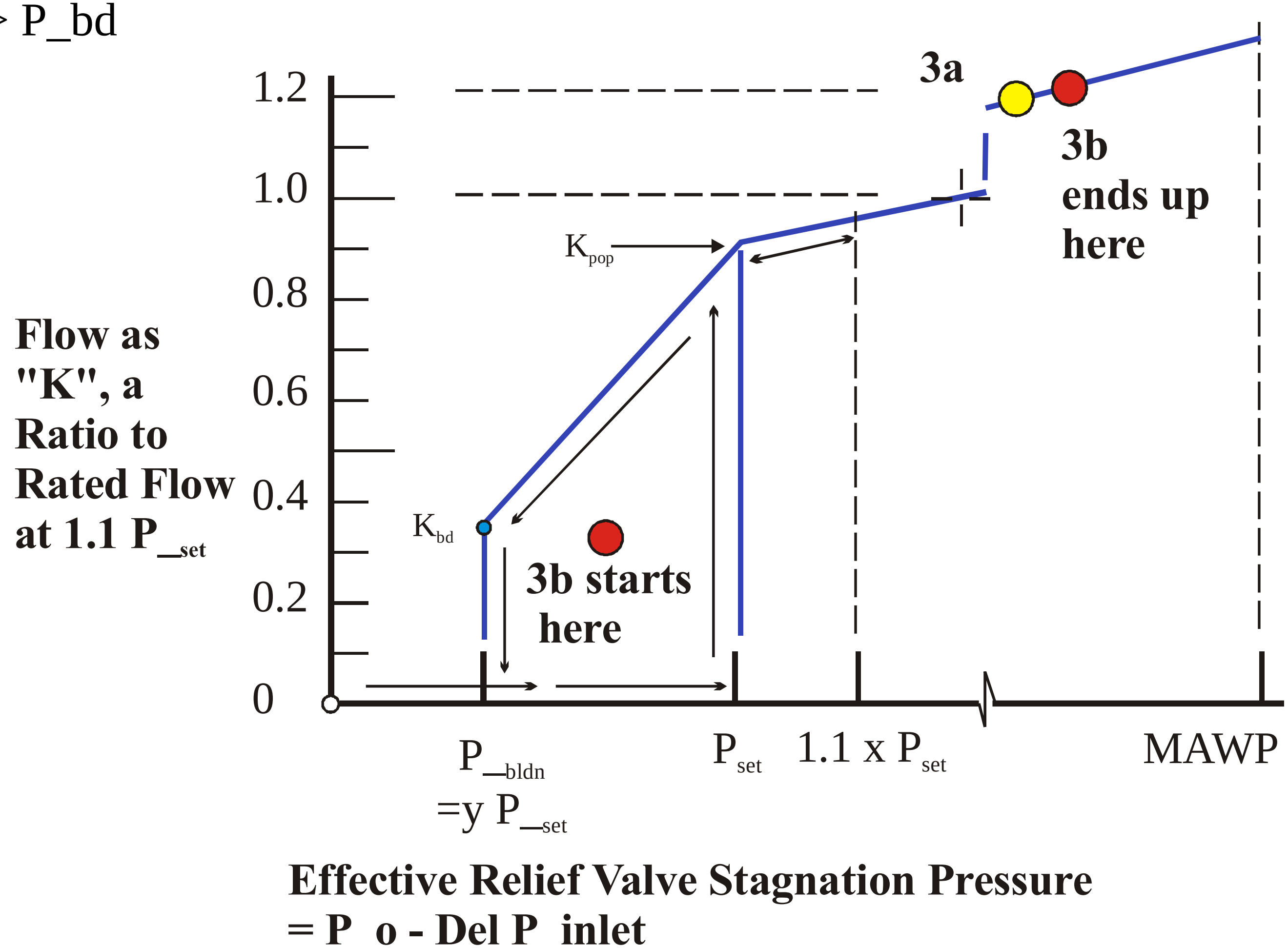
Case # 3 Relief Valve is <u>under sized</u> at P_set. But P_set < MAWP

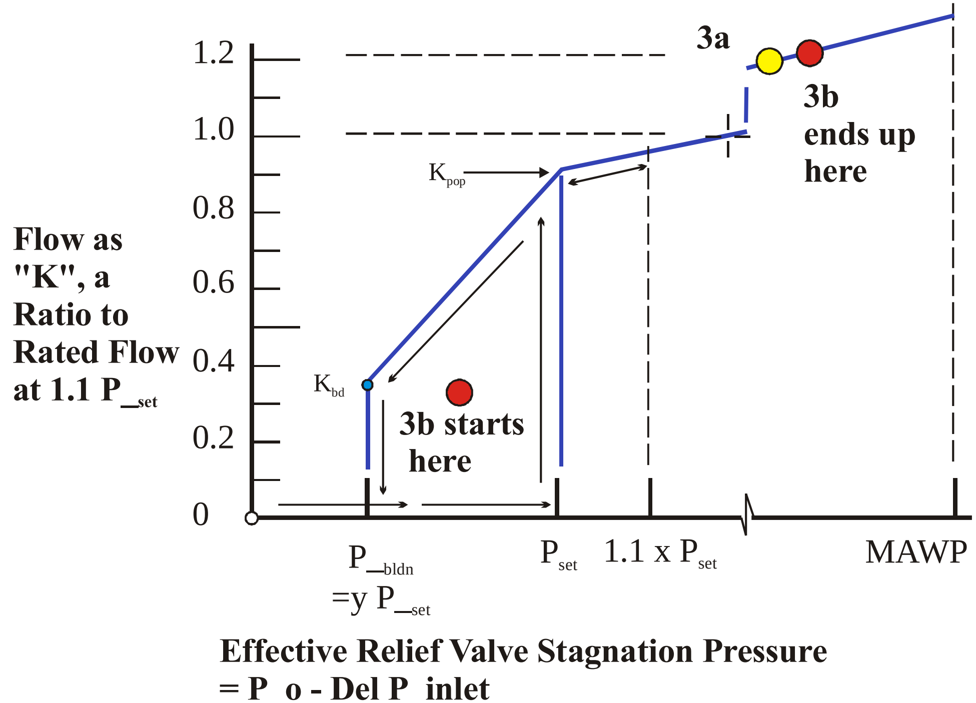
Q_required > Q_rated at P_set so that K > 1.0 at some P, < MAWP

Case 3a: Del P_inlet < y P_set at K=1, but we require K = 1.2

Case 3b: Del P_inlet > y P_set at K=1, but at K=1.2,

$(1.2)^2 * yP\_set > P\_bd$





Case # 3 Example:

MAWP = 100 psig

P_set = 80 psig

y* P_set = 8 psig (10% Del P_in)

P_bd = 74 psig

At 1.2 x 80 psi = 96 psi Del P_in = 1.2^2 x 8 = 11.5 psi

96 -11.5 = 84.5 psig Very comfortably above the 74 psig blowdown pressure.

We have Thus Far Touched on 3 Ways to Address O3

(a) Increase inlet nozzle and pipe size or increase valve body inlet size by design change, so as to force the use of large diameter – low impedance inlet configurations

(b) Employ damping features in the relief valve moving parts

(c) Reduce set pressure relative to MAWP such that valve is undersized at P_set but adequate at MAWP

Some Conjectures

So let us define “truth” as being a correct statement for most circumstances.[(1)] Then we conjecture that the following statements are true:

(a) Del P_{inlet} < 0.03 Pset is a sufficient condition for avoiding high frequency O3 - <u>but not a necessary condition</u>

(b) Del P_{inlet} < 0.03 Pset is neither sufficient or necessary for avoiding O3 for an oversized relief valve.

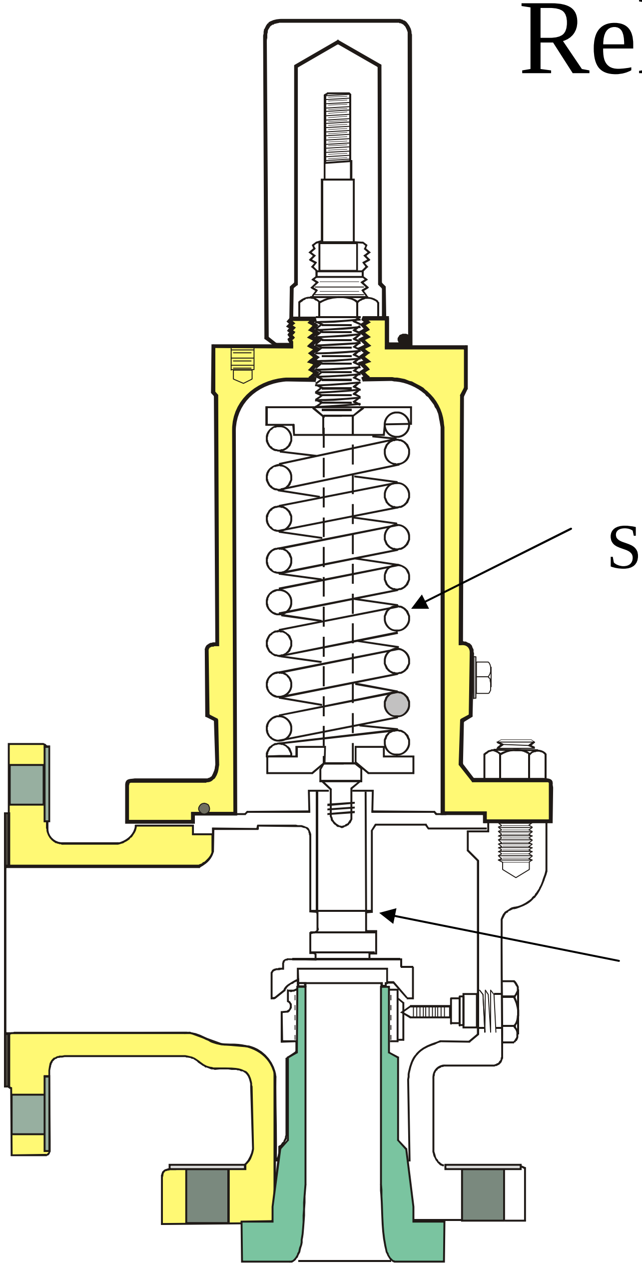
(c) Del P_{inlet} > 0.03 Pset or even > y Pset can be safely accomodated by reucing the set pressure below MAWP with the valve undersized at Pset but right sized at MAWP

(1) Note: this also being an election year, this criterion for “truth” exceeds all threshold levels for truth in current political discourse.

Relief Valve Operation as a Spring –Mass System

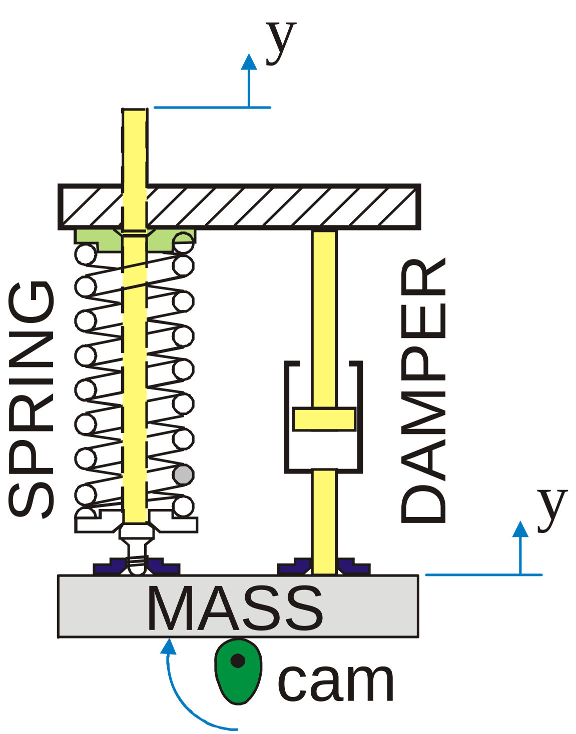
Implicit in the preceding comments is the notion that high frequency O3 is related to resonance with the relief valve natural frequency. <u>And we turn to this issue next.</u>

Relief Valve Operation as a Spring –Mass System



The mass is there, but not in one lump. There is the valve cap, a slider piece, the spring rod, a restraining button and some part of the spring itself.

Textbook Spring – Mass – Damper System



$$m\frac{d^2y}{dt^2} + c\frac{dy}{dt} + k\,y = F_o \sin(a\,t)$$

has solution

$$y = Y\sin(a\,t - b)$$

where the amplitude factor Y

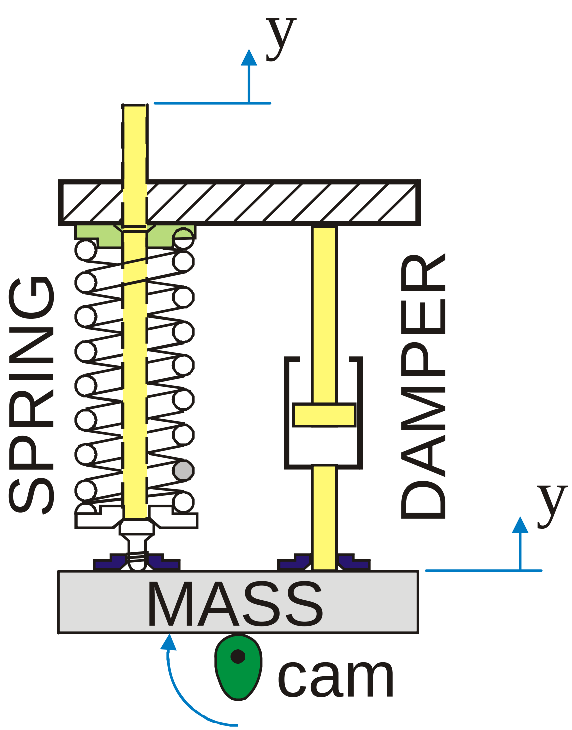
$$Y = \frac{F_o / k}{\left\{\left(1 - \frac{m\,a^2}{k}\right)^2 + \left(\frac{c\,a}{k}\right)^2\right\}^{1/2}}$$

m – mass

c – damping coefficient

k – spring constant

F_o - force



m – mass

c – damping coefficient

k – spring constant

F_o - force

The phase shift angle

$$phi = \tan^{-1}\left(\frac{c\,a}{k - m\,a^2}\right)$$

and the natural frequency is

$$a_n^2 = \frac{k}{m}$$

The dimensionless *damping factor is*

$$DF = \frac{c}{m\,a_n} = \frac{c}{\sqrt{k\,m}}$$

Now let

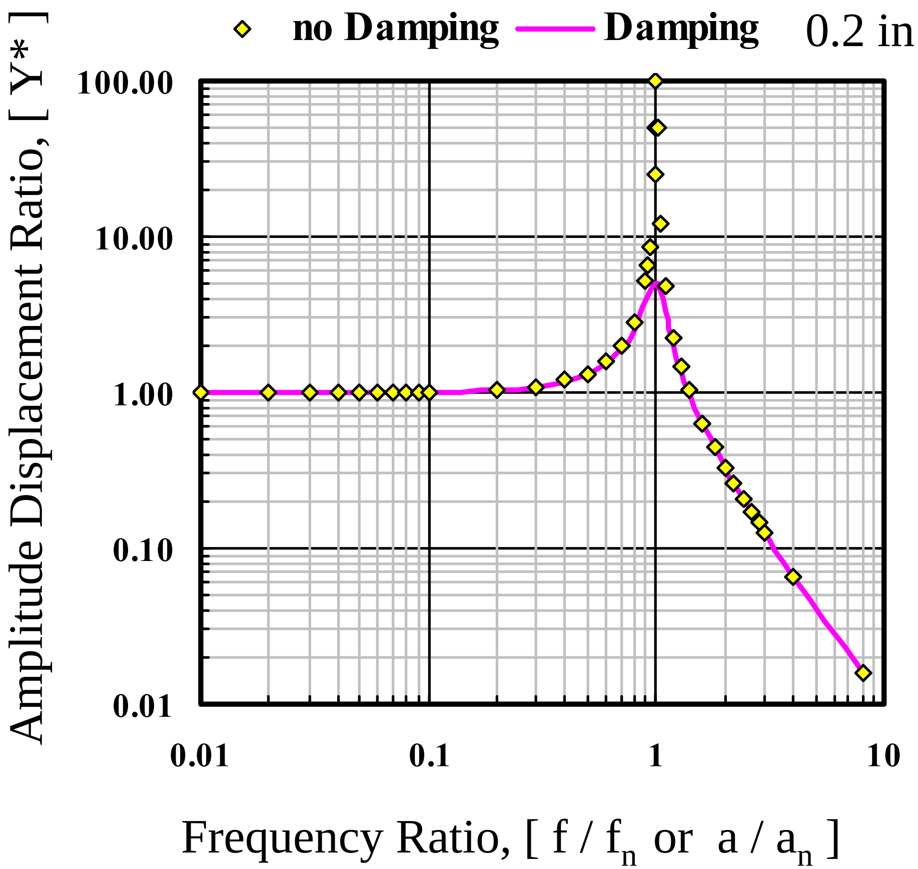
$$Y^* = \frac{Y}{F_o / k}$$

So that

$$Y^* = \frac{1}{\left\{\left(1 - \left(\frac{a}{a_n}\right)^2\right)^2 + \left(DF\,\frac{a}{a_n}\right)^2\right\}^{1/2}}$$

Amplitude Response for Spring – Mass System with and without damping

Damping Factor = 0.2 in illustration



If the damping factor is 0.2, the phase shift angle is less than 4°

Frequency of the forcing function to the system natural frequency

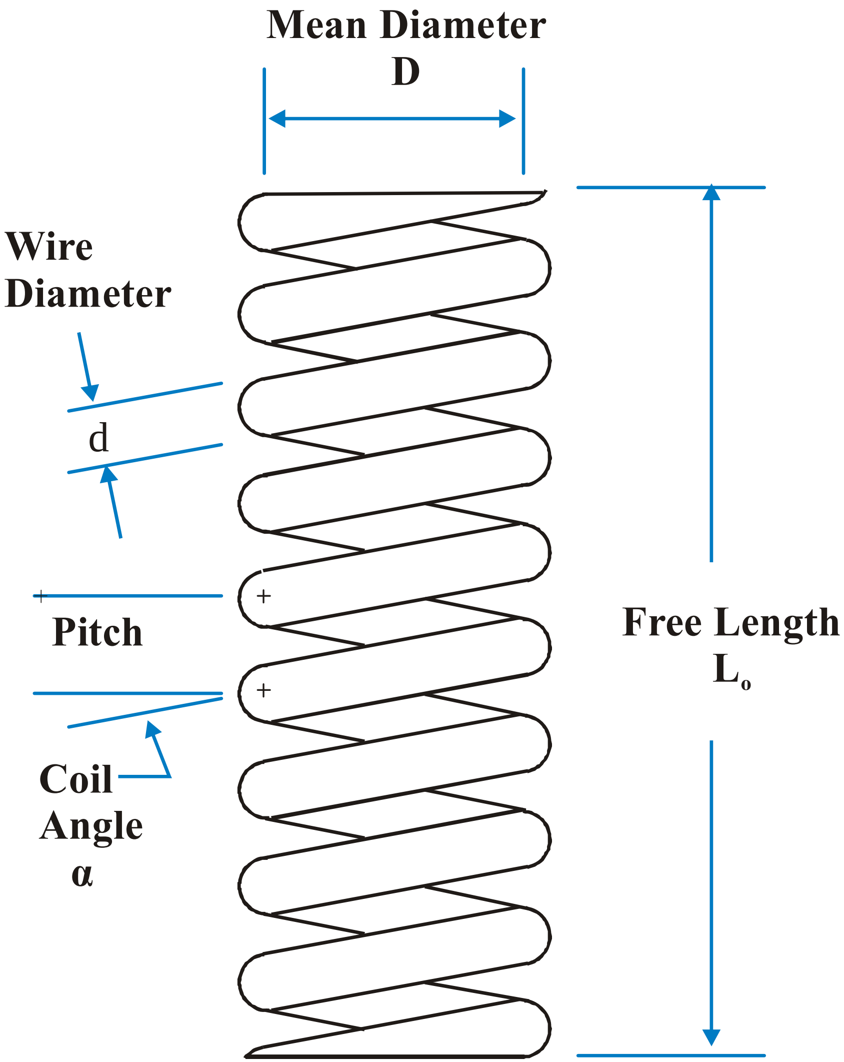
Summary of this Part:

If one wishes to avoid the resonant frequency, or if one wishes to determine whether O3 will be near the resonant frequency, one can do so without any further coupling of the spring mass system to the rest of the problem.

However, one needs to know the spring constant (k), and the mass (m), in order to identify the natural frequency (a_n)

This we take up next.

Helical Springs

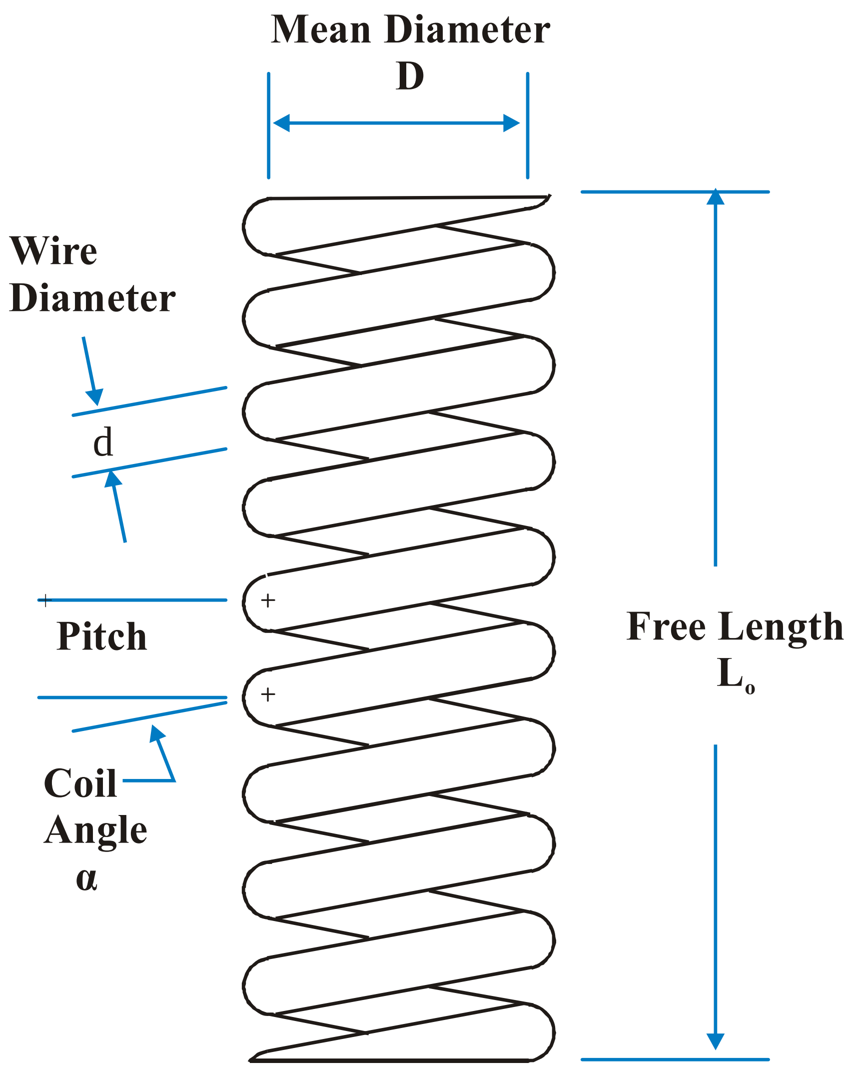


Relief Valves typically

have springs with closed ends ground such that the number of active coils is $Na = Nt - 2$

Where Nt is the total number of coils

Helical Springs



Nomenclature:

N_a is the number of active coils

G is the modulus of torsion or rigidity

C is the diameter modulus; C= D/d

D is the mean diameter

d is the coil diameter

Spring Constant k

$$k = \frac{G\,d}{8\,C^3\,N_a}$$

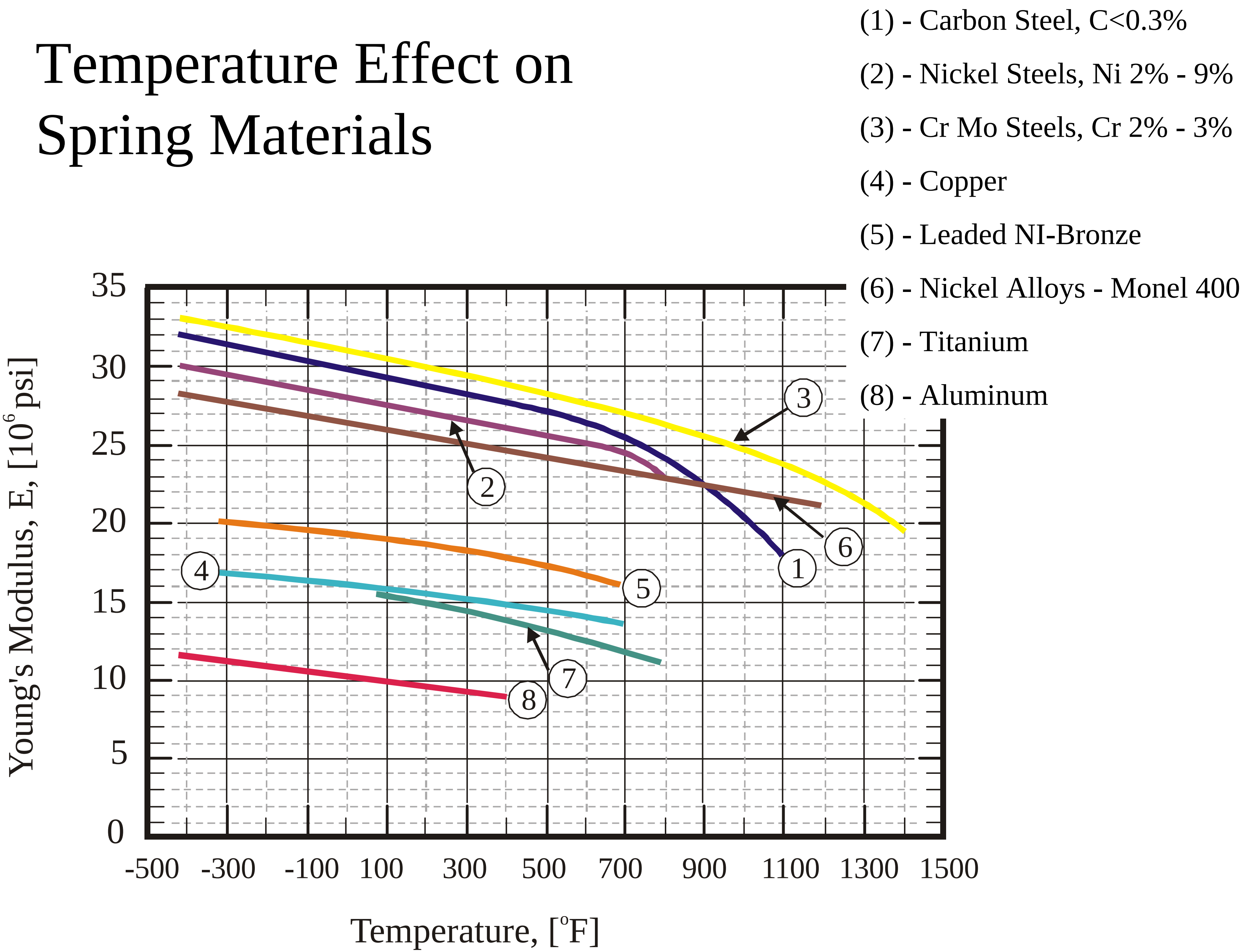
Young's Modulus (E) and Modulus of Torsion (G)

Metal	**Poisson's Ratio**	**E [10^9 Pa]**	**G [10^9 Pa]**
Aluminum	**0.330**	**69**	**27**
Copper	**0.360**	**117**	**43**
Ni-Steel	**0.310**	**213**	**76**
Stainless Steel 18-8	**0.300**	**201**	**73**
Carbon Steel	**0.303**	**202**	**79**
High Carbon Steel	**0.295**	**210**	**81**
Inconel	**0.290**	**214**	**79**

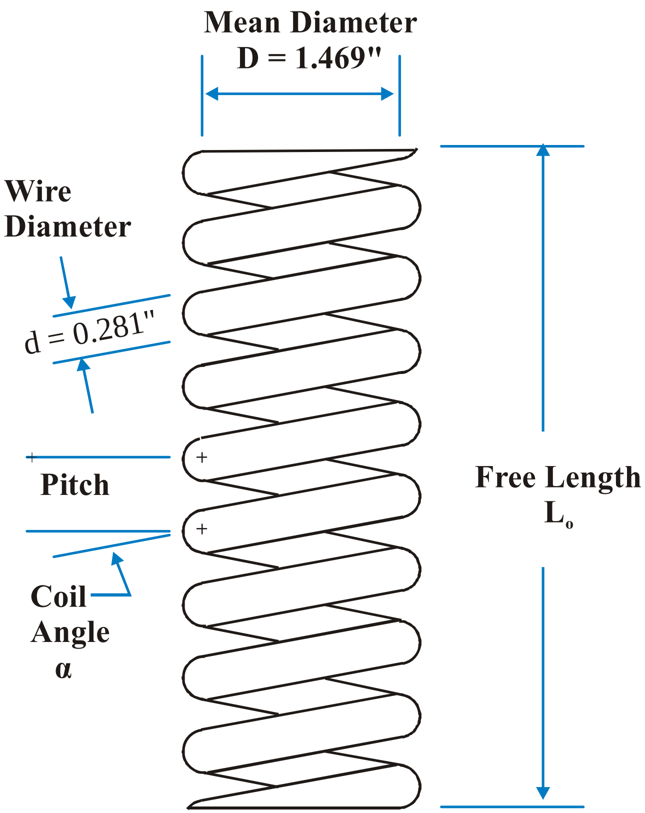
$$G = \frac{E}{2(1+v)}$$

Poisson's Ratio

Temperature Effect on Spring Materials



Real Example:

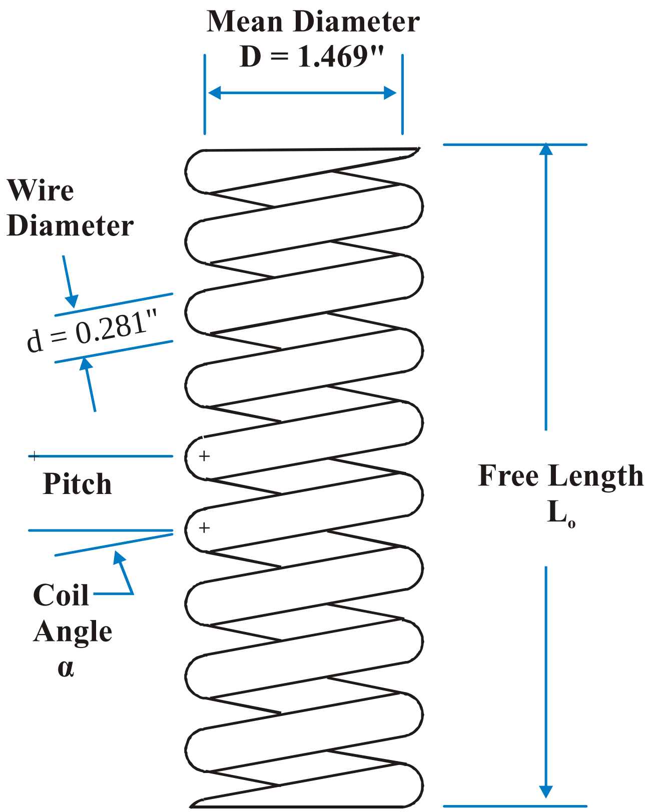


Spring from
Farris 26 FA 10 - 120
180 psig set

$N_a = 8$
$C = D/d = 5.288$
Assume:
$E = 205$ GPa or $29.7x10^6$ psi
$\nu = 0.31$
$G = 78.2$ GPa or $11.35x10^6$ psi

$$k = \frac{11.35E6\,x\,0.281}{8\,x\,5.288^3\,x\,8} = 349\,\frac{lbf}{in}$$

Real Example: (continued)



Spring from
Farris 26 FA 10 - 120
180 psig set

Wgt. of Spring = 360 gm
Wgt. of other
parts in motion = 435 gm
Assume mass in motion is
435gm + 360 / 3 = 555 gm
or approximately 1.22 lb

Real Example: (continued)

So now we have

k = 350 $\mathbf{lb_f}$ / inch or 61294 N / meter

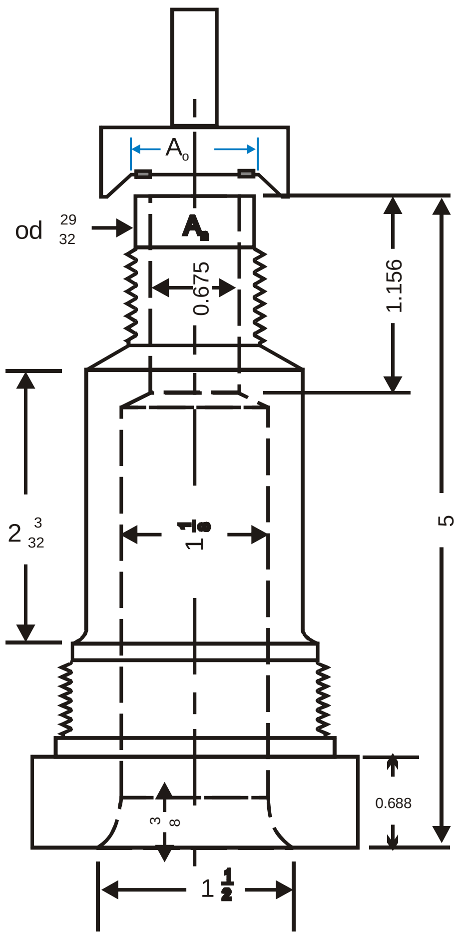
And

m = 0.555 kg

Therefore the natural frequency is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{61294}{0.555}} = 53\,Hz$$

Can we do better ??



By examining the geometry of the nozzle and cap and doing a little calibration with the known example one can arrive at the following relations:

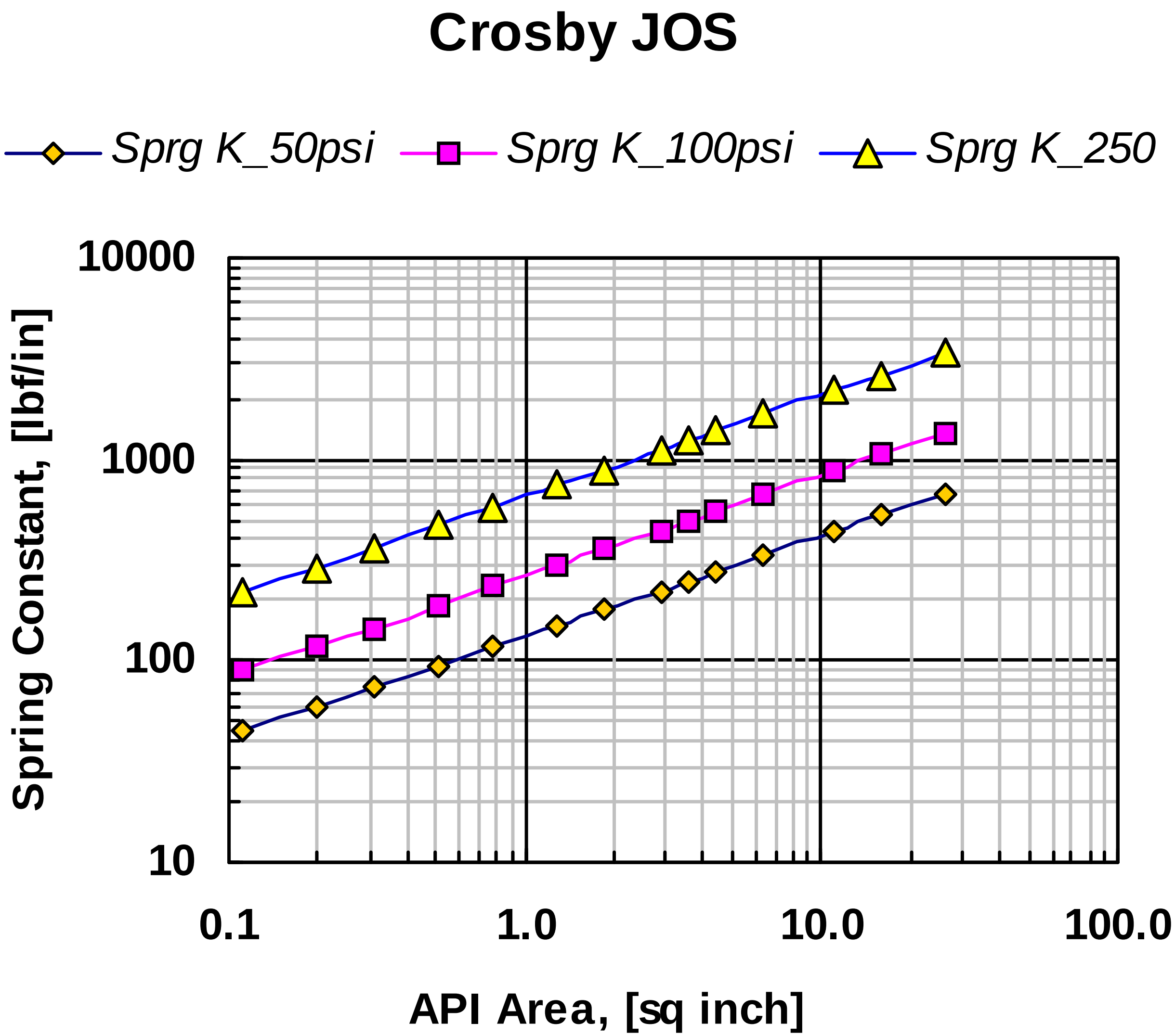
$$k = 1.08 \frac{P_{set} \; A_{redbook}}{y_{lift}}$$

and

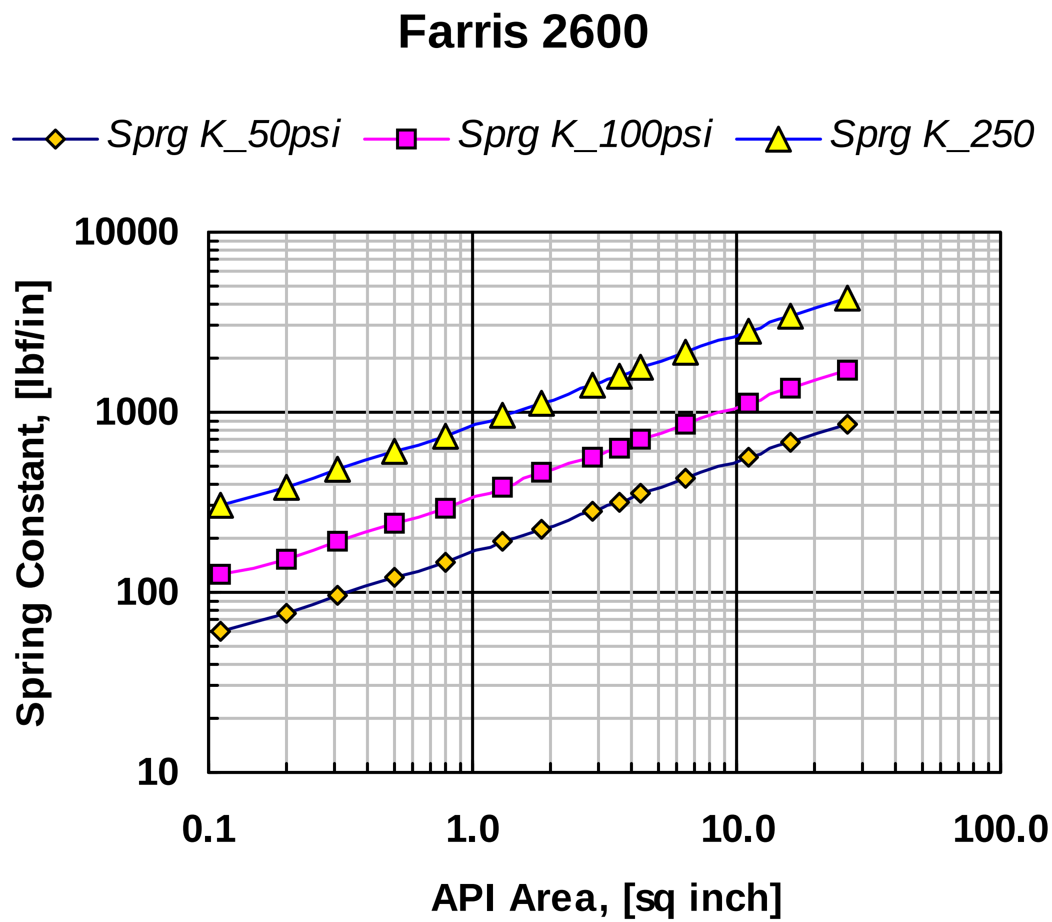
$$m = 0.025 \, M_{valve}$$

See PDF File on desktop

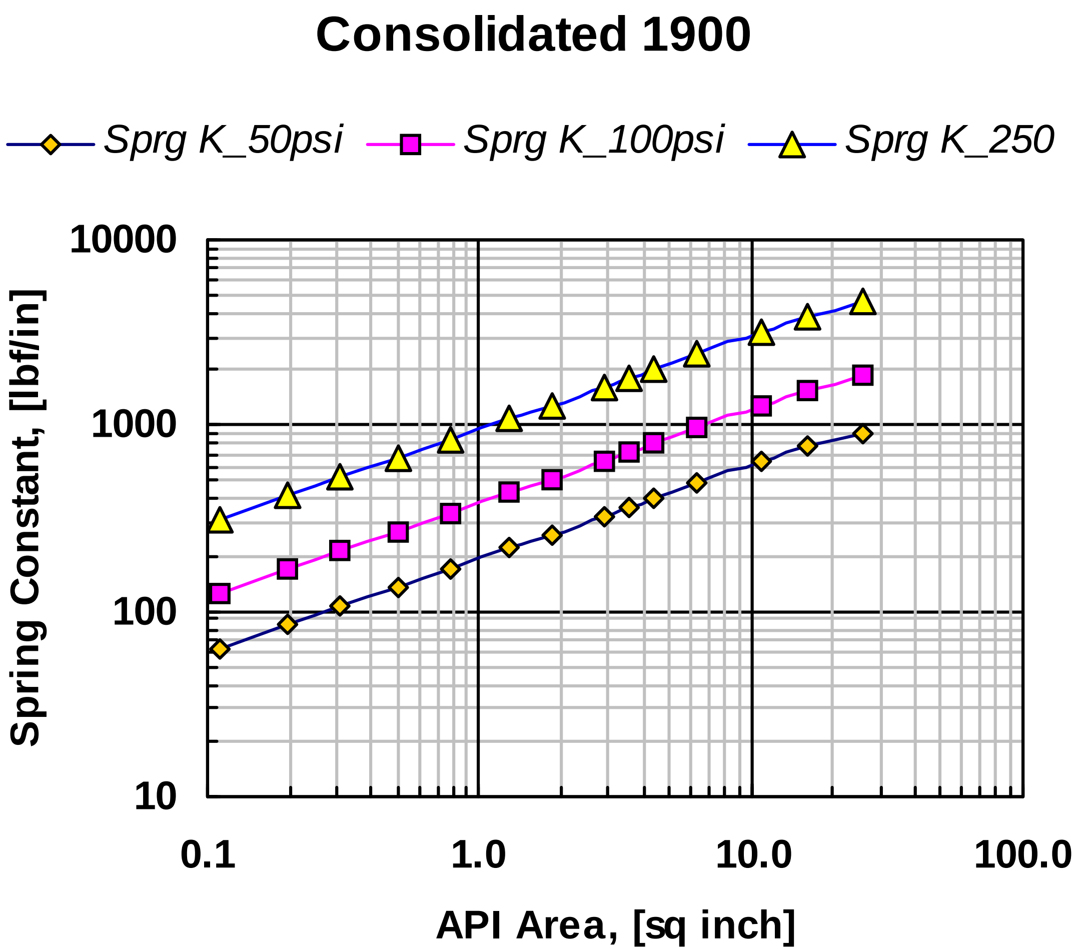
Relief Valve Spring Constant Estimate



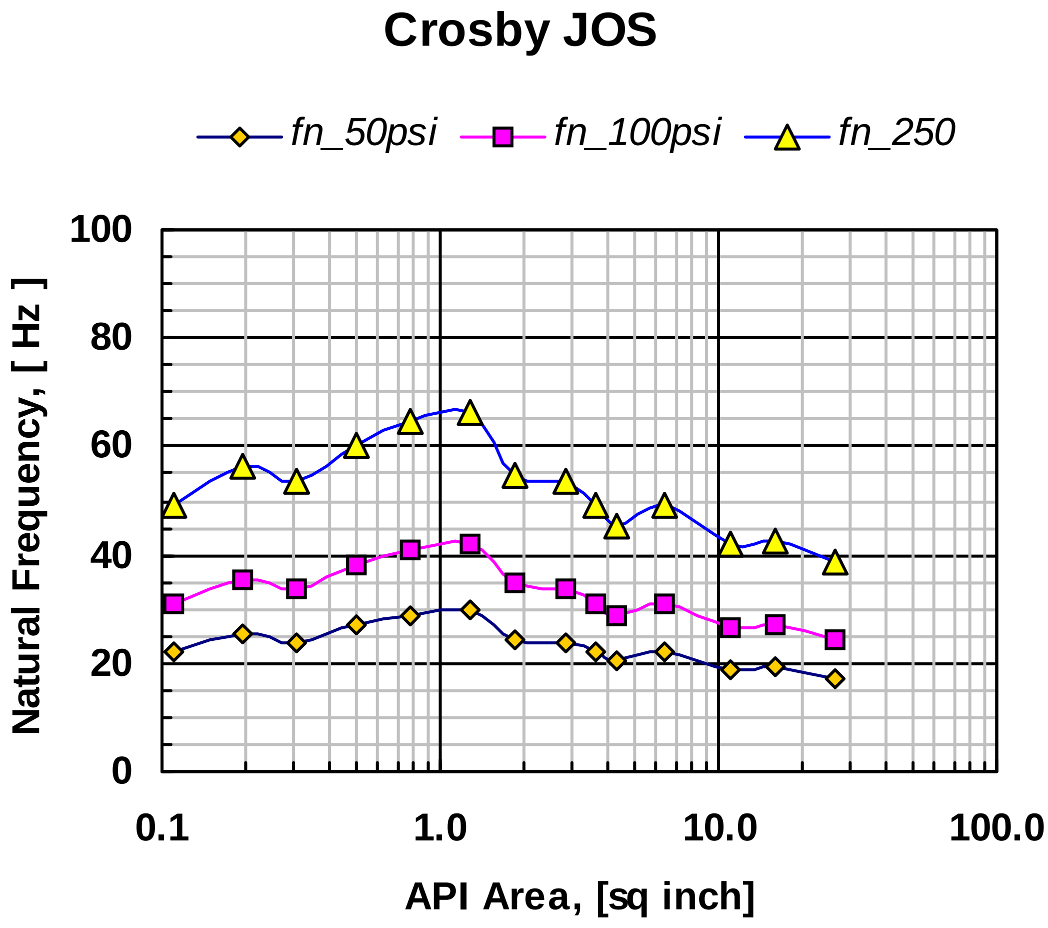
Relief Valve Spring Constant Estimate



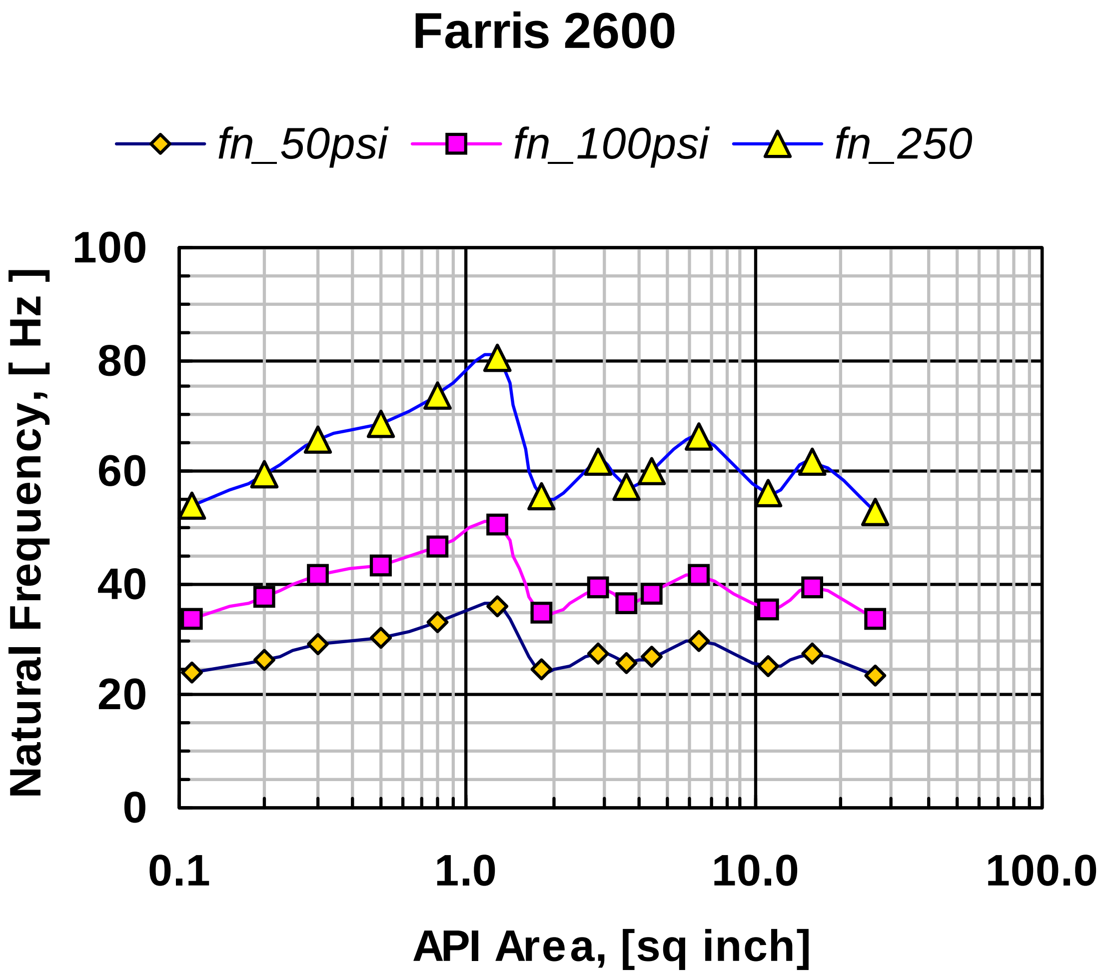
Relief Valve Spring Constant Estimate



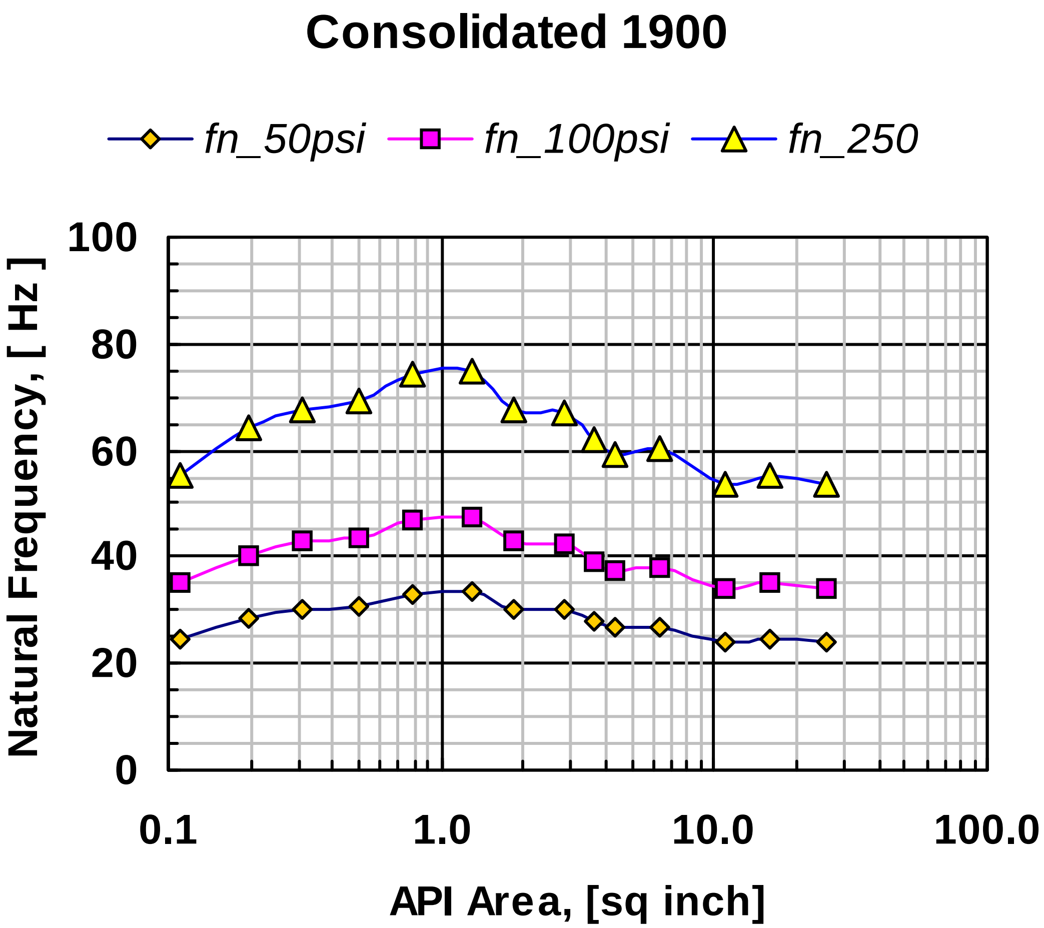
Relief Valve Natural Frequency Estimate



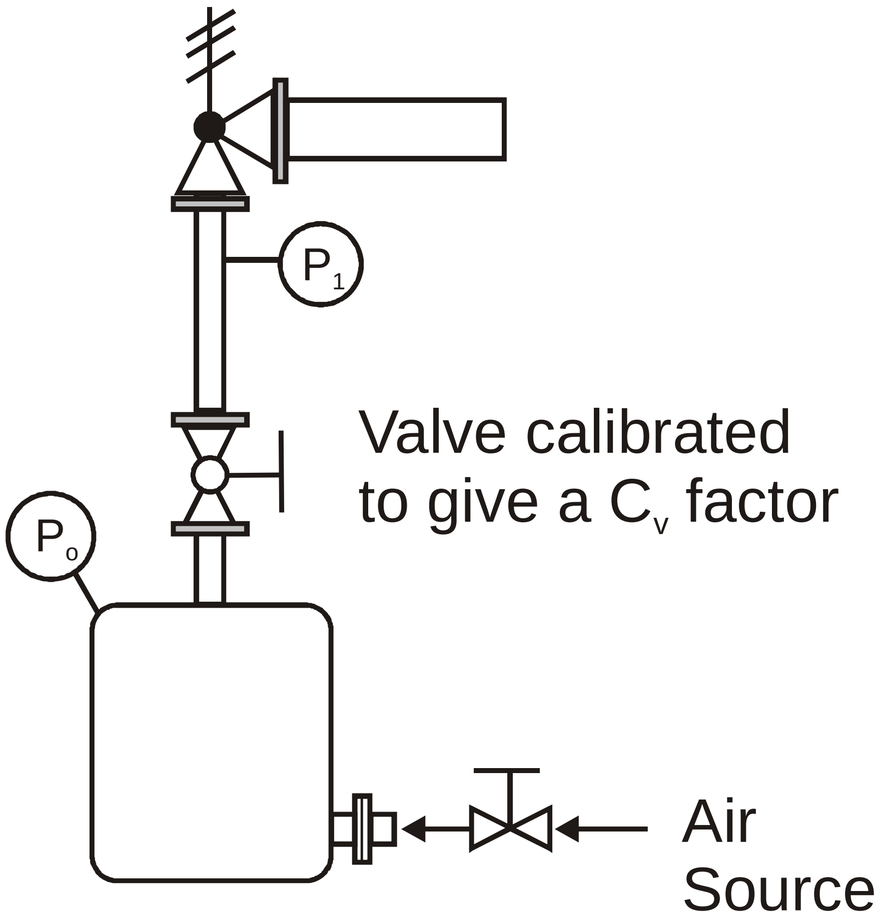
Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Kastor's Tests



Relief Valve:

Consolidated 1905 E (1 E 2)

P_{set} = 50 psig

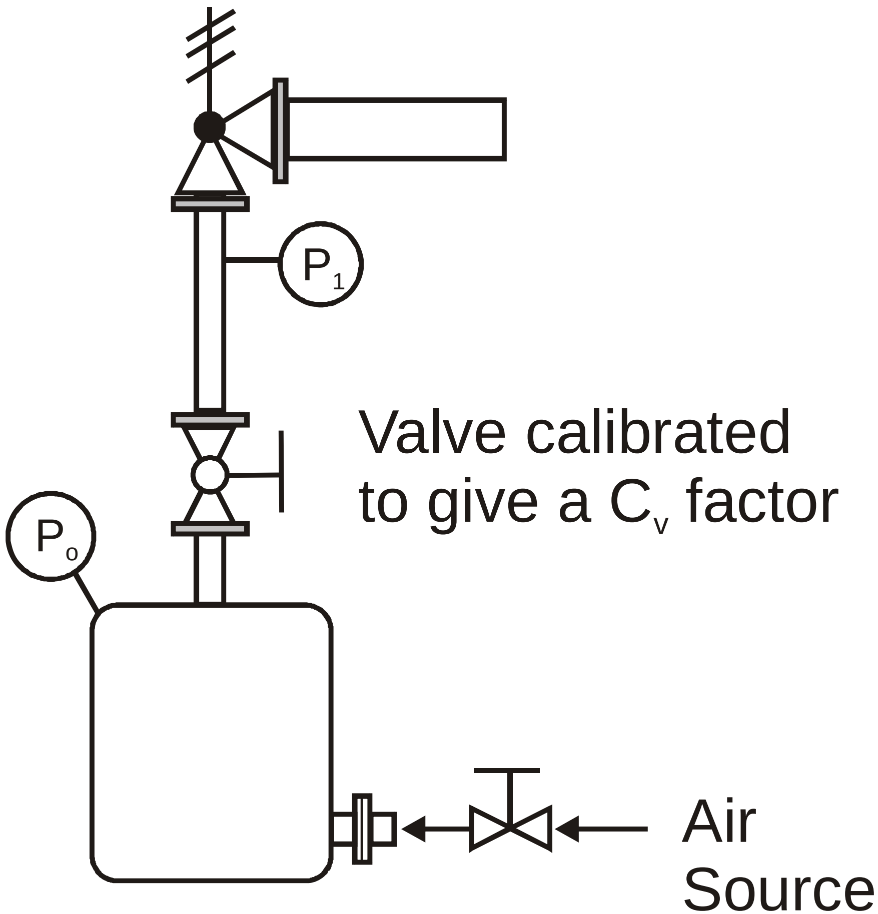
E nozzle: 0.196 sq in API; 0.2279 sq in ASME

Rated Flow at 50 psig (1.1op); 1115 lb/hr air

Reference Test Air Flow:

1200 lb/hr air

Initial Observations :



Relief Valve:

Redbook Valve Lift: 0.147 inch

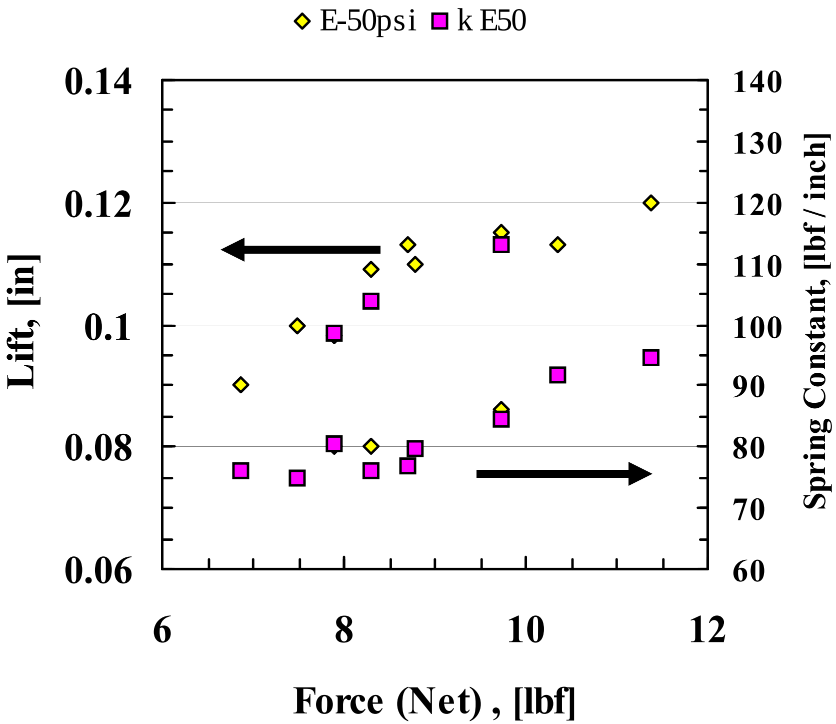
From previous material, we estimate the spring constant and natural frequency as

k = 84 lbf / in

f_n = 29 Hz

Kastor's data show chattering at 30 Hz and his lift vs pressure data show k_{avg} = 87 lbf / in

Observations :



Also Stated:

Avg Opening Pressure 50.6 psig

Avg Reclosing Pressure 44 psig

Damping coefficient 0.2

Note: Blowdown, P_{bd} is over 10% relative to P_{set}

<u>Relief Valve:</u>

Redbook Valve Lift: 0.147 inch

From previous material, we estimate the spring constant and natural frequency as

k = 84 lbf / in

f_n = 29 Hz

Kastor's data show chattering at 30 Hz and his lift vs pressure data show k_{avg} =87 lbf / in

5 calibration data points for

Approx method for "fn" and "k"

Objectives

- To extend the work initiated Kastor and others, and to point out some potential means for anticipating O3 and minimizing the adverse consequences.
- We take it as self-evident that O3 can be provoked by either excessive inlet or tail pipe pressure loss. However, for now, to keep matters simple, we will focus on inlet pressure loss.

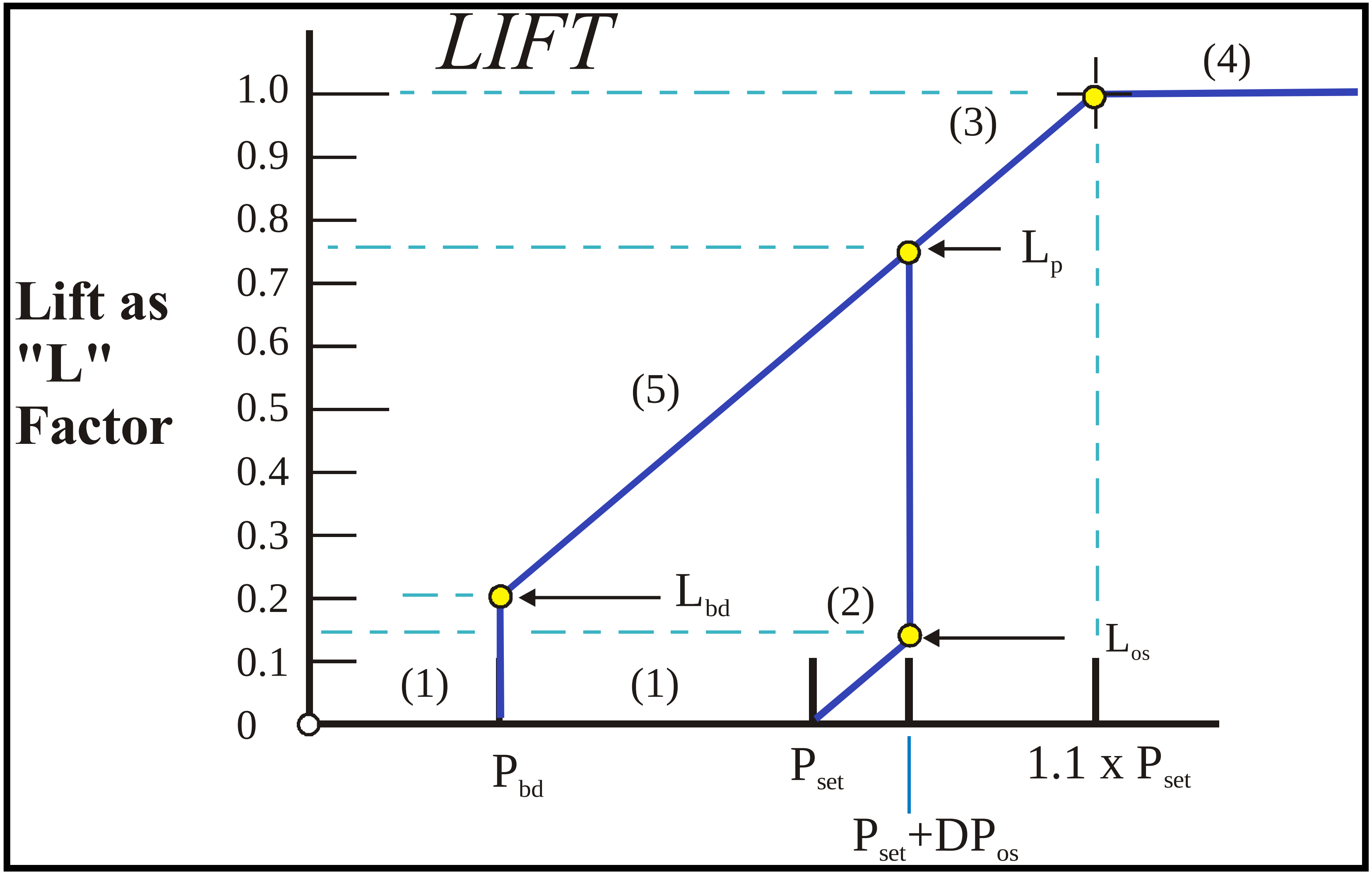
Summary of this Part (i):

- Even close the f_n, the lift amplitude overshoot, is severely limited by the "stops" as defined by the valve nozzle and the cap stem guide.
- Further, even though we do not know the damping factor, evidence suggests that it is currently small, and any resulting phase shift is not of first order significance.

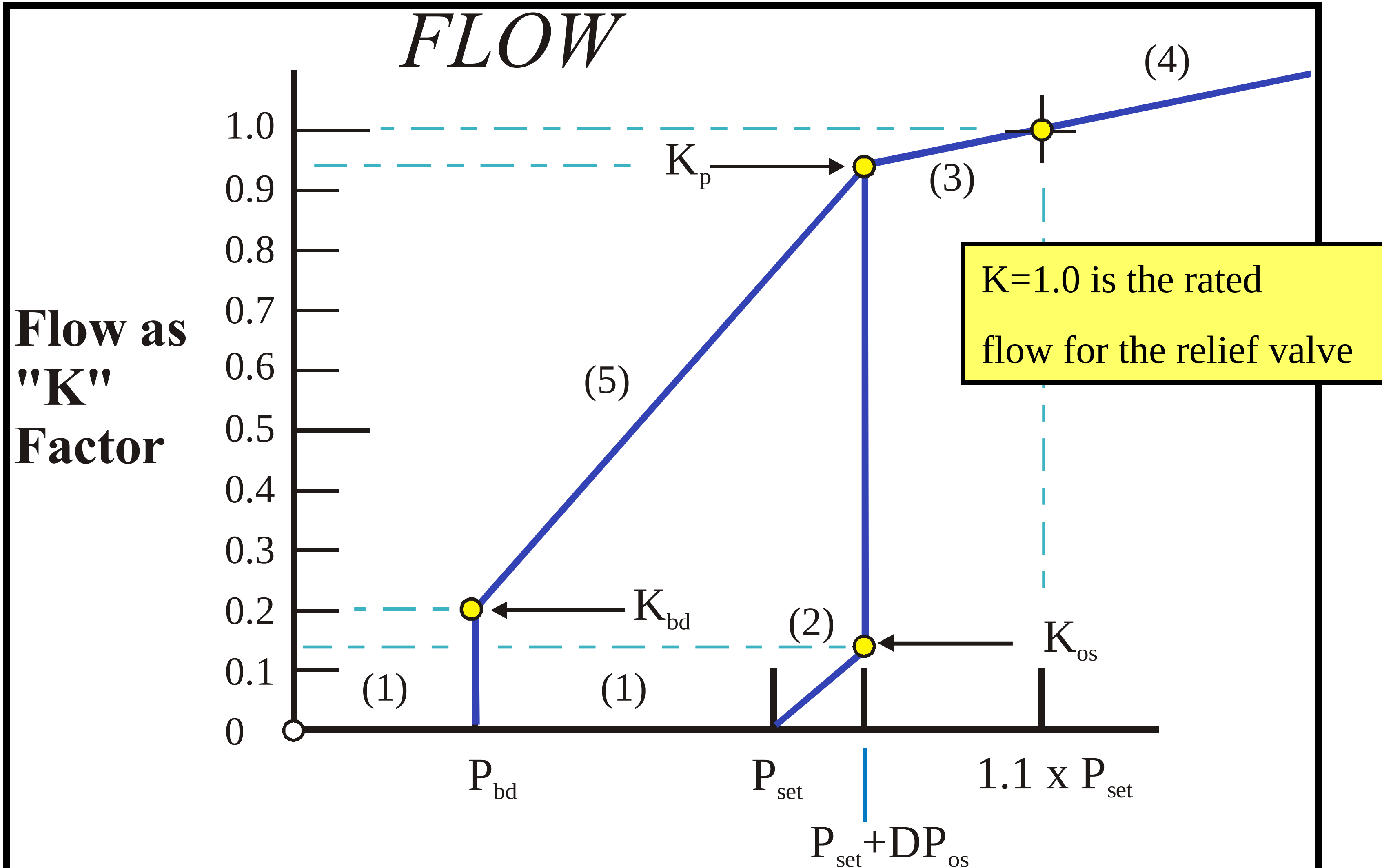
Summary of this Part (ii):

- Therefore if we know the natural frequency of the valve – (not currently a published property) – all we have to do is to know how to stay away from it or to get away from it.
- This leads to the suggestion of replacing the valve SMD equation with the valve steady state response represented by the following figures.

Relief Valve Lift



Relief Valve Flow



We have constructed the relief valve "lift" and "flow" figures to be internally consistent

(1) The valve will start to open at P_{set} and "pop" at P_{set} + DP_{os}, an offset pressure difference , which may be zero

(2) The valve "pops" at P_{set} + DP_{os} to a specified lift, Lp, alleged to be ≈ 65% to 75% of full lift,

(3) The valve reaches full lift and full flow at 1.1 x P_{set}

(4) The valve flows ≈ 90% of rated flow at Lp or after "popping".

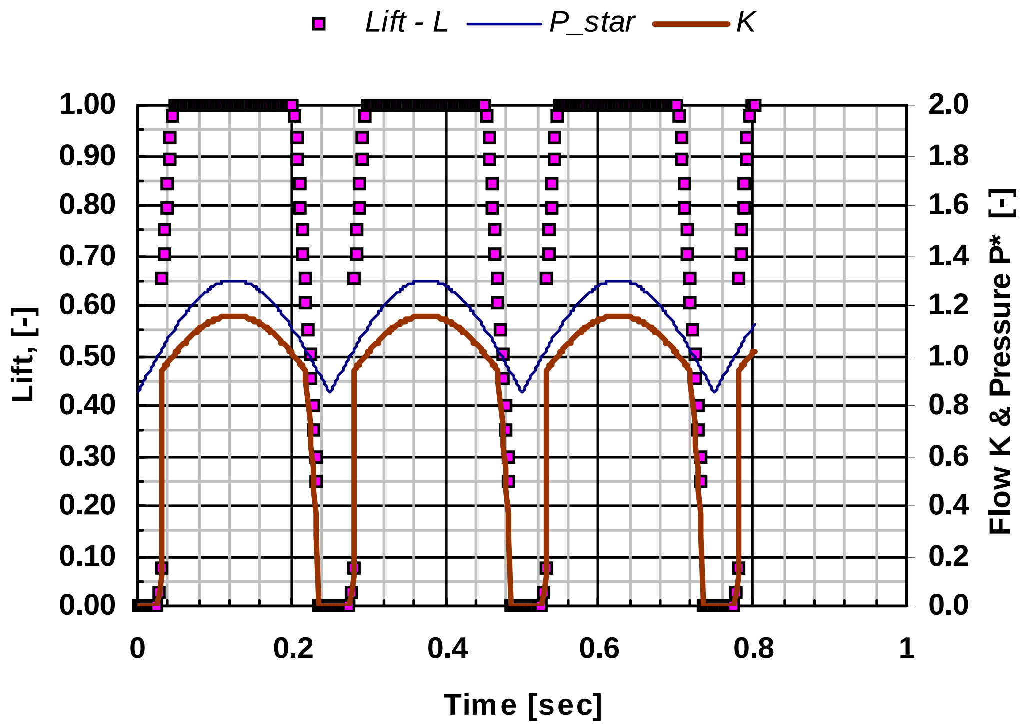
(5) The minimum flow at P_{bd} is estimated to be ≈ 25% of rated flow. P_{bd} is the blowdown pressure

(6)The P_{bd} value is estimated to be 93% of Pset but this can vary.

Operating the Lift and Flow function

(1) We have programmed the valve "Lift" amd "Flow" curve. (Even in Excell).

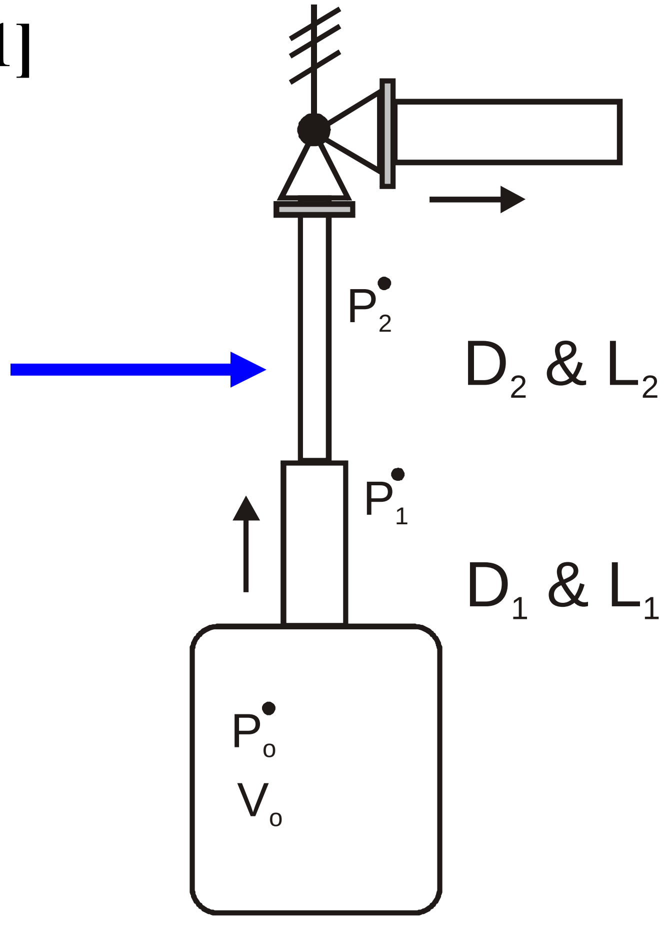
(2) It's response to a forcing function is shown below.



Brief Description of Model

(1) We start with a model schema that appears similar to that which might be similar to the Kastor modeling approach – see Ref [K1]

We lump the segments (1) and (2) into a single equivalent volumes and assign the flow resistance to an overall flow coefficient according to conventional rules.



This leads to an Equation Set:

The Source Flow

Vessel

$$\frac{P_0^{\bullet}}{P_o} = \lambda_0 \left[B - \frac{(1/eps)(C_d\ A)_1}{(C_d\ A)_o} \sqrt{2(P_0 - P_1)/P_0} \right]$$

Inlet Pipe

$$\frac{P_1^{\bullet}}{P_0} = \lambda_1 \left[\frac{(1/eps)(C_d\ A)_1}{(C_d\ A)_o} \sqrt{2(P_0 - P_1)/P_0} - K(P_1) \right]$$

The relief valve Flow

Notes

- $K(P_1)$ is the flow coefficient from the valve lift and flow curves at P_1
- B is a dimensionless source flow term, a multiple of the valve rated flow:

$B < 1$ Valve is over sized: Source flow is less than valve rated flow at P_{set}

$B = 1$ Valve is "just right " sized with source flow = valve rated flow.

$B > 1$ Valve is under sized at P_{set}, source flow is > rated flow at P_{set}

Notes

- The segment volume pressure response time constants are:

$$\lambda_o = \frac{(C_d\ A)_o}{V_o}\sqrt{\square R\,T}$$

$$\lambda_1 = \frac{(C_d\ A)_o}{V_1}\sqrt{\square R\,T}$$

Notes

- The time constants determine the transient response of the vessel and the piping segments
- The system is “stiff” but will do O3 operation and progress to a steady state solution if there is one.
- We can actually learn much from the steady state solution
- The steady state solution is contained in the terms in brackets only with the P* terms set =0

Some Assumptions & Findings

(1) It is assumed that the fastest the relief valve can close is $1/(2f_n)$. Ex.: f_n = 50 Hz; del t $_{close}$ = 10 ms

(2) If B = 0.8; Del P_in =7%; Po = 100 psigand one has a 10 m3 vessel volume; an 8 L inlet volume and put this together we find

(3)The valve can close in 10 ms, the inlet line can depressurize in about 4 ms, but the acoustic response time is ≈ 40 ms. Therefore the acoustic time rules. Then we find that the recovery time is coincidentally ≈ 35 ms , so we have an ≈ 80 ms cycle time or on / off at about 12 hz

Some Assumptions & Findings

(3) Quite coincidentally – in 2004 Melhem and Fisher showed results of their code(s) producing valve oscillations at 10 to 12 Hz.

(4) In the previous case these oscillations go on indefinitely, but if we Change to B=1.2 (valve undersized by 20%, we can estimate the valve oscillation time to get out of O3.

(5)We assume Po must increase 15 psig with a set pressure of 80 to 85 psig. We find the time to get out of chatter at about 10 sec --- or about 120 on/off cycles.

Some Assumptions & Findings

(6) **Fundamentally, I see no reason why Melhem and Fisher are not in excellent position to treat on/off relief valve performance in an effective manner.**

Implications for 3% rule

(1) The present 3% rule does not consider the positive effects of reducing the set pressure relative to the MAWP. Depending on the set pressure reduction, the inlet pressure loss can be much greater than 3% and relief flow will be entirely stable

(2) If the 3% rule is excessively conservative, it should be changed. We have shown that a 5% allowance can make a large difference

Implications for 3% rule

(3) Reducing the relief valve set pressure below the vessel design pressure can also have the same positive effect on allowance for larger inlet pressure loss

(4) A very good situation is to have the relief valve "undersized" at the relief set pressure and properly sized for some defined overpressure. In this case 3% or even 10% inlet pressure loss is irrelevant if the relief valve design properly considers the actual inlet pressure loss in the sizing calculation. Undersized at the relief set pressure is common recommended practice for runaway tempered reactions as a class.

Things one would like to see from Relief Valve Manufacturers

(1) Valve spring constants and natural frequency data – if not in the general catalog at least with the purchased valve certification sheets

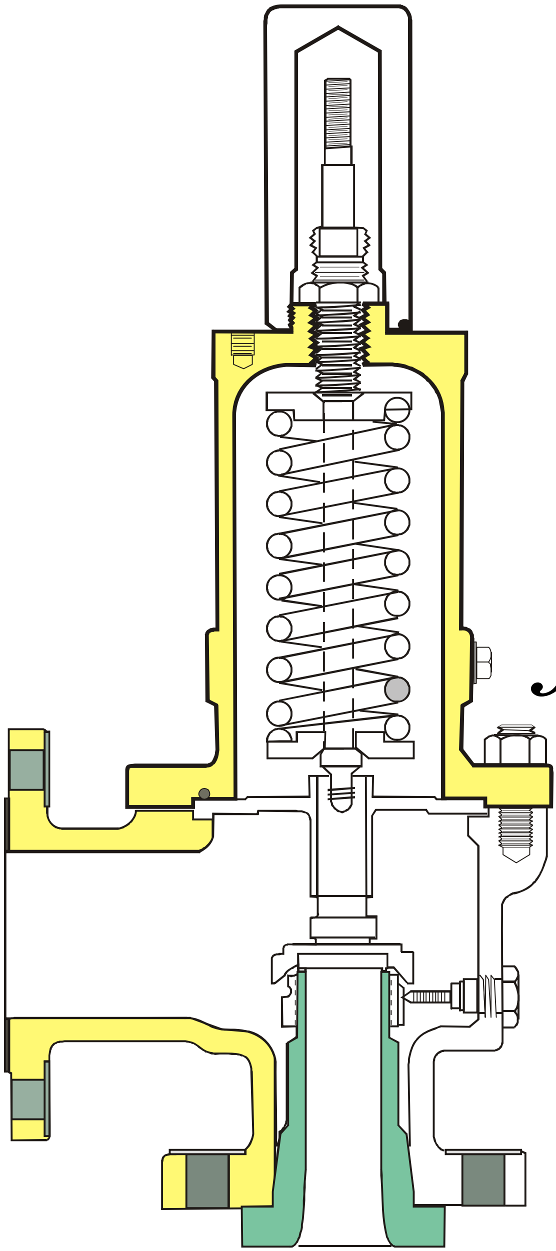
(2)Better relief valve Lift and Flow curves.

(3)Better data and guidance on relief valve blowdown

(4) Mechanical data on the number of on/off cycles that can be safely tolerated. Is this number 10 or 10^6 ; It makes a profound difference

Final Thoughts: The 3% rule should not become a rule without exceptions – we have indicated several possible alternative routes to stable operation with greater than 3% inlet pressure loss.

mag



There is still more to this story, but for now

That's All Folks!!

CTI
Centaurus Technology, Inc.

This effort recognizes the following references as important prior contributions and source material.

- Ref. [D1], Darby, R. and Berwanger, Inc., “Stability of Relief Valves, a Review and Evaluation of Prior Work”, unpublished presentation to API, July 3, 2001 – private communication.
- Ref. [M&F1], Melhem, G. A and Fisher, H. G., “Practical Guidelines for Dealing with Excessive Pressure Drop in Relief Systems” DIERS USERS GROUP Presentation, 2003/4
- Ref. [I1], Izuchi, H., “Chatter of Safety Valve”, Slide presentation, April, 2008, Chiyodaq

References: continued.

- Ref. [CFP1], Cremers, J., Friedel, L. and Pallaks, B., "Validated sizing rule against chatter of relief valves during gas service", J. Loss Prevention, Vol. 14, 2001, pp 261-267
- Ref. [K1], Kastor, K. A., "A Dynamic Stability Model for Predicting Chatter in Safety Relief Valve Installations, Part I – Model Development, Verification and Application", A DuPont Technical Report, Feb. 3, 1986.
- Ref. [K1], Kastor, K. A.,"Relief Valve Chatter Design", DIERS USERS GROUP Presentation, April, 1990

References: continued.

- Ref. [K1], Kastor, K. A.,“Relief Valve Chatter Testing”, PVP – Vol. 237-2, Seismic Engineering, ASME Vol. 2, pp 133-137.
- Ref. [G1], Grolmes, M. A., “Relief Valve Stability and Inlet Pressure Loss”, DIERS USERS GROUP Presentation, April, 2008

DIERS Users Group Mtg.
October, 2009 Houston TX

"Some Further Considerations Of Upstream Parameter Effects On Relief Valve (in) Stability"

Michael A. Grolmes

Centaurus Technology, Inc.

4590 Webb Road

Simpsonville, KY 40067

502-243-9678

ctimag@att.net

Centaurus Technology, Inc.

Presented by: H. G. Fisher, FisherInc.

This effort recognizes the following references as important prior contributions and source material.

- Ref. [D1], Darby, R. and Berwanger, Inc., “Stability of Relief Valves, a Review and Evaluation of Prior Work”, unpublished presentation to API, July 3, 2001 – private communication.
- Ref. [M&F1], Melhem, G. A and Fisher, H. G., “Practical Guidelines for Dealing with Excessive Pressure Drop in Relief Systems”, DIERS USERS GROUP Presentation, 2003/4
- Ref. [I1], Izuchi, H., “Chatter of Safety Valve”, Slide presentation, April, 2008, Chiyoda

References: continued.

- Ref. [CFP1], Cremers, J., Friedel, L. and Pallaks, B., “Validated Sizing Rule Against Chatter of Relief Valves During Gas Service”, J. Loss Prevention, Vol. 14, 2001, pp 261-267
- Ref. [K1], Kastor, K. A., “A Dynamic Stability Model for Predicting Chatter in Safety Relief Valve Installations, Part I – Model Development, Verification and Application”, A DuPont Technical Report, Feb. 3, 1986.
- Ref. [K1], Kastor, K. A., “Relief Valve Chatter Design”, DIERS USERS GROUP Presentation, April, 1990

References: continued.

- Ref. [K1], Kastor, K. A., “Relief Valve Chatter Testing”, PVP – Vol. 237-2, Seismic Engineering, ASME Vol. 2, pp 133-137.
- Ref. [G1], Grolmes, M. A., “Relief Valve Stability and Inlet Pressure Loss”, DIERS USERS GROUP Presentation, April, 2008

Definitions.

- We will define any relief valve ON-OFF-OPERATION, (O3), as a form of relief valve operating instability – regardless of frequency.
- We will consider "Chattering" as a form of higher frequency O3 – that may be closely associated with the relief valve natural frequency, f_n

Background.

- There exists some interesting data for O3 for small relief valves
- Ref [K1] strongly suggests that the 3% inlet pressure loss rule is a poor guide to either the occurrence or avoidance of O3
- It is undoubtedly the case that repeated and extended periods of O3 of any kind, can lead to relief valve damage.

Background.

- However, it is interesting to note that Refs. [K1 and [I1] report extensive tests with small relief valves with no mention of relief valve damage.
- Ref [G1] pointed out reasons that O3 caused relief valve damage may be more likely for large valves at high set pressure with the converse also likely.
- Many prior references have proposed criteria for avoiding relief valve instability with varying degrees of success.

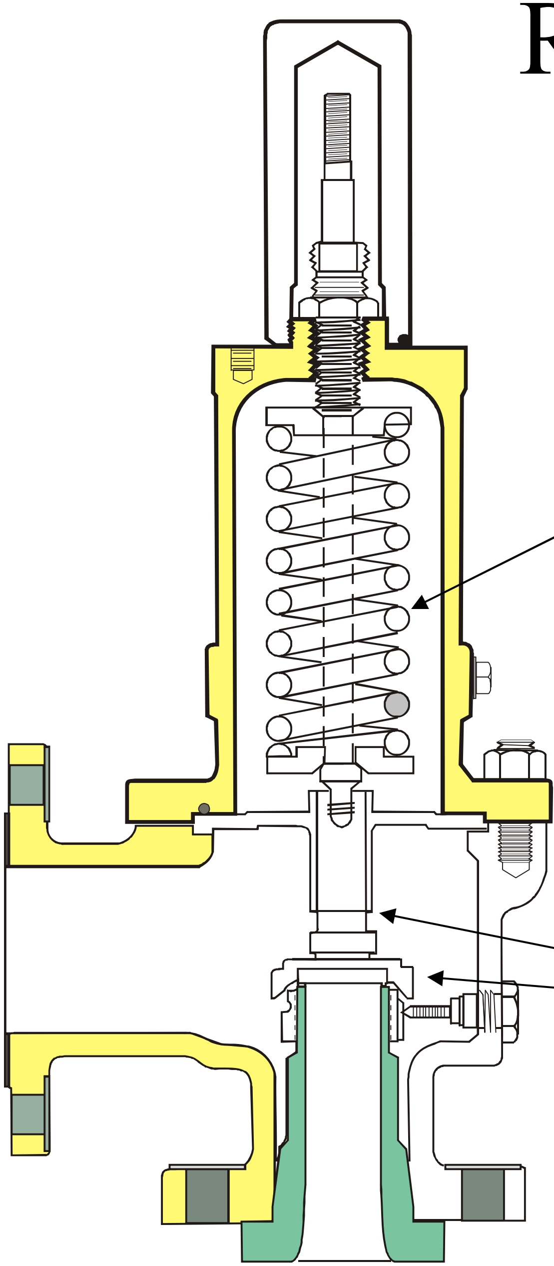
Background.

- This reviewer finds no existing method that is either easy to use at all or easy to use without parameter values that are not readily available.

Objectives

- To extend the work initiated in Ref. [G1] and point out some potential means for anticipating O3 and minimizing the adverse consequences.
- We take it as self-evident that O3 can be provoked by either excessive inlet or tail pipe pressure loss. However, for now, to keep matters simple, we focus on inlet pressure loss.

Relief Valve Operation as a Spring –Mass System

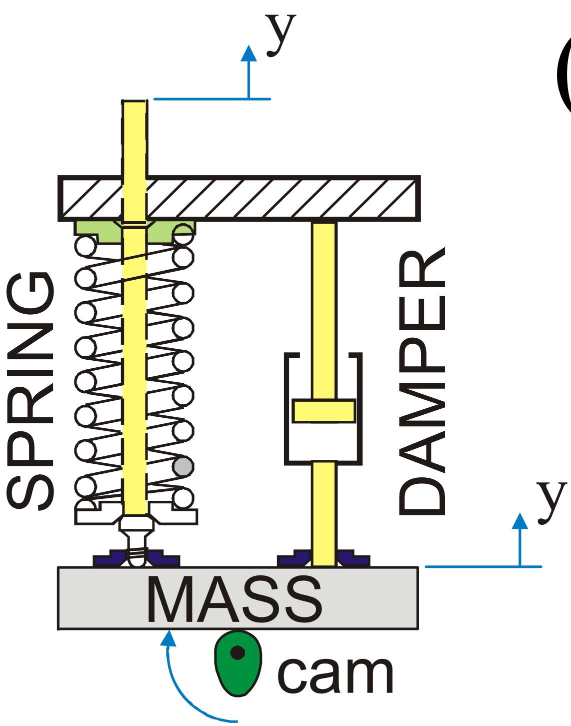


Spring

Note Cap displacement stops at nozzle and stem guide

The mass is there, but not in one lump. There is the valve cap, a slider piece, the spring rod, a restraining button and some part of the spring itself.

Textbook Spring – Mass – Damper (SMD) System



$$m\frac{d^2y}{dt^2}+c\frac{dy}{dt}+k\,y \;=\; F_o\sin(\omega t)$$

has solution

$$y \;=\; Y\sin(\omega t-\varphi)$$

where the amplitude factor Y

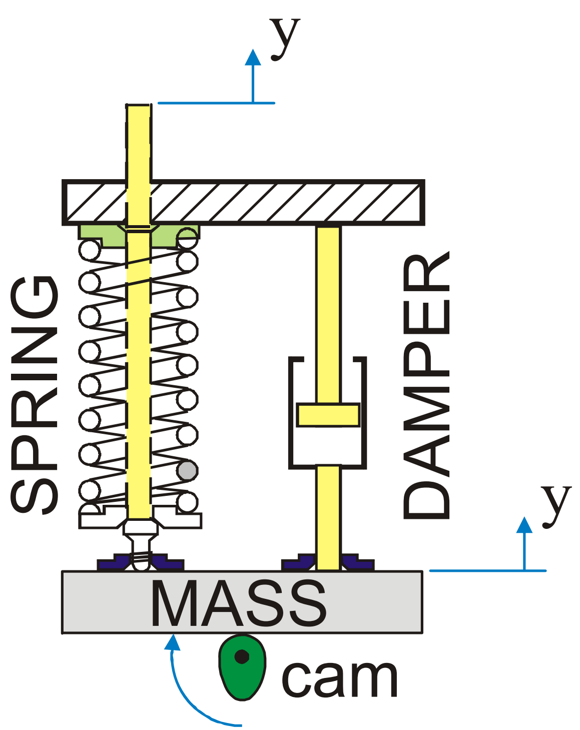
$$Y=\frac{F_o/k}{\left\{\left(1-\dfrac{m\,\omega^2}{k}\right)^2+\left(\dfrac{c\,\omega}{k}\right)^2\right\}^{1/2}}$$

m – mass

c – damping coefficient

k – spring constant

F_o - force



m – mass

c – damping coefficient

k – spring constant

F_o - force

The phase shift angle

$$\varphi = \tan^{-1}\left(\frac{c\,\omega}{k - m\,\omega^2}\right)$$

and the natural frequency is

$$\omega_n^2 = \frac{k}{m}$$

The dimensionless *damping factor is*

$$DF = \frac{c}{m\,\omega_n} = \frac{c}{\sqrt{k\,m}}$$

Now let

$$Y^* = \frac{Y}{F_o / k}$$

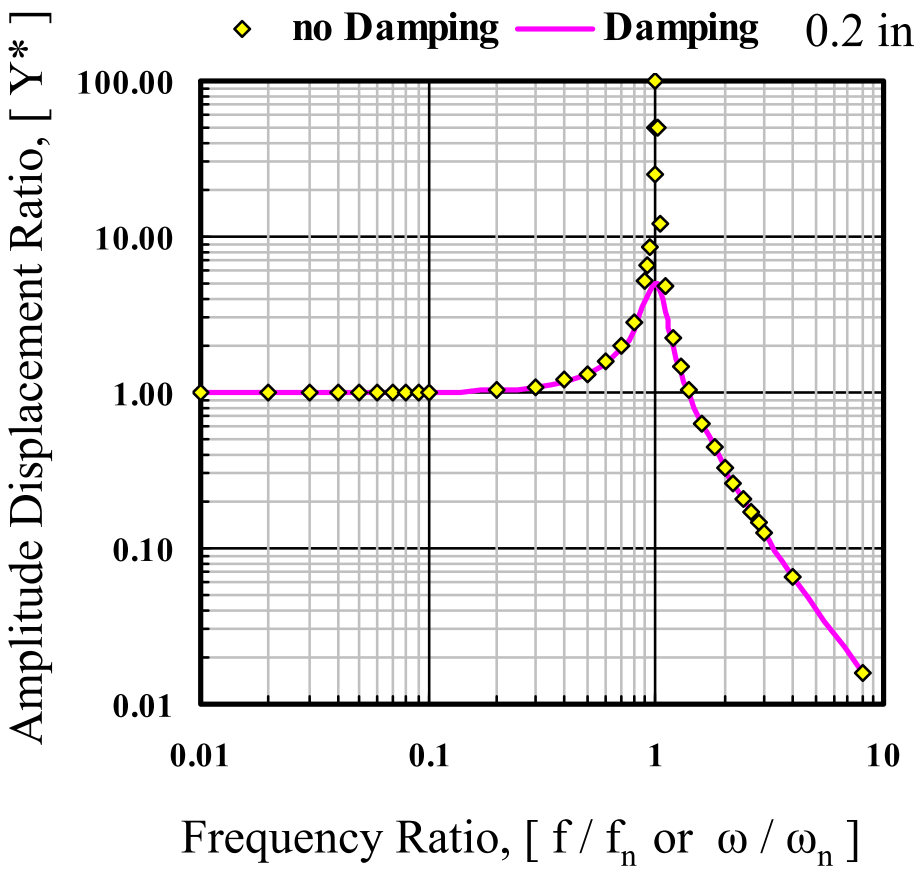
So that

$$Y^* = \frac{1}{\left\{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(DF\,\frac{\omega}{\omega_n}\right)^2\right\}^{1/2}}$$

Amplitude Response for Spring – Mass System with and without damping

Response to a sinusoidal forcing function of frequency "f".

Damping Factor = 0.2 in illustration



If the damping factor is 0.2, the phase shift angle is less than 4°

Frequency of the forcing function to the system natural frequency

If one wishes to avoid the resonant frequency, one can do so without any further coupling of the spring mass system to the rest of the problem.

However, to avoid resonance, one needs to know the <u>spring constant</u> (k), and the <u>mass</u> (m), in order to identify the <u>natural frequency</u> (ω_n)

Summary of this Part (a):

- There is nothing unique regarding the mechanical SMD equation. The solutions are well known.
- The response function for the relief valve, viewed as a SMD system, shows that for excitation frequencies less than 50% of fn (natural frequency), the valve is “passive”, i.e., it responds to the forcing function with no overshoot.

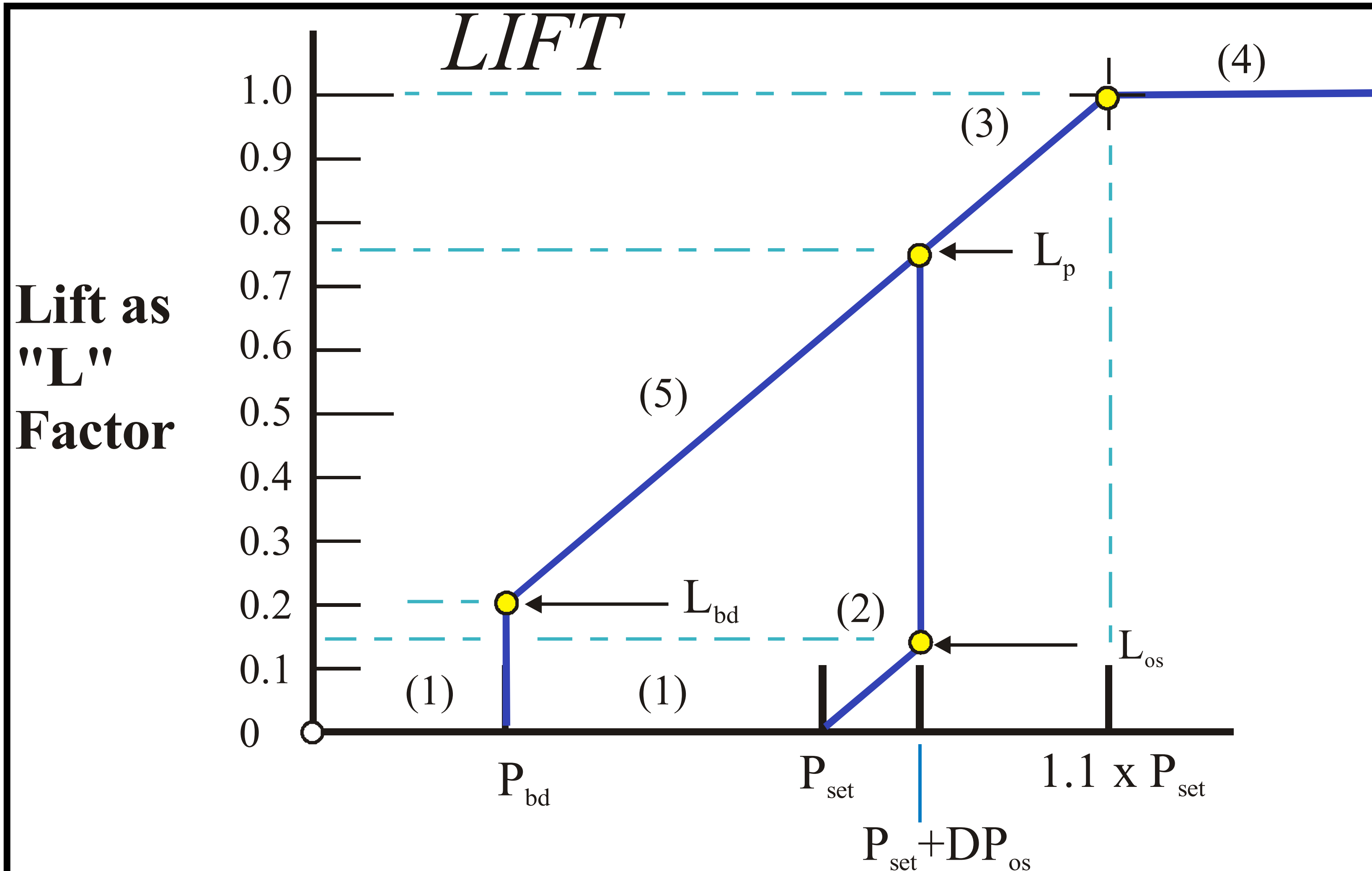
Summary of this Part (b):

- Even close to the f_n, the lift amplitude overshoot is severely limited by the “stops” as defined by the valve nozzle and the cap stem guide.
- Further, even though we do not know the damping factor, evidence suggests that it is small and any resulting phase shift is not of first order significance.

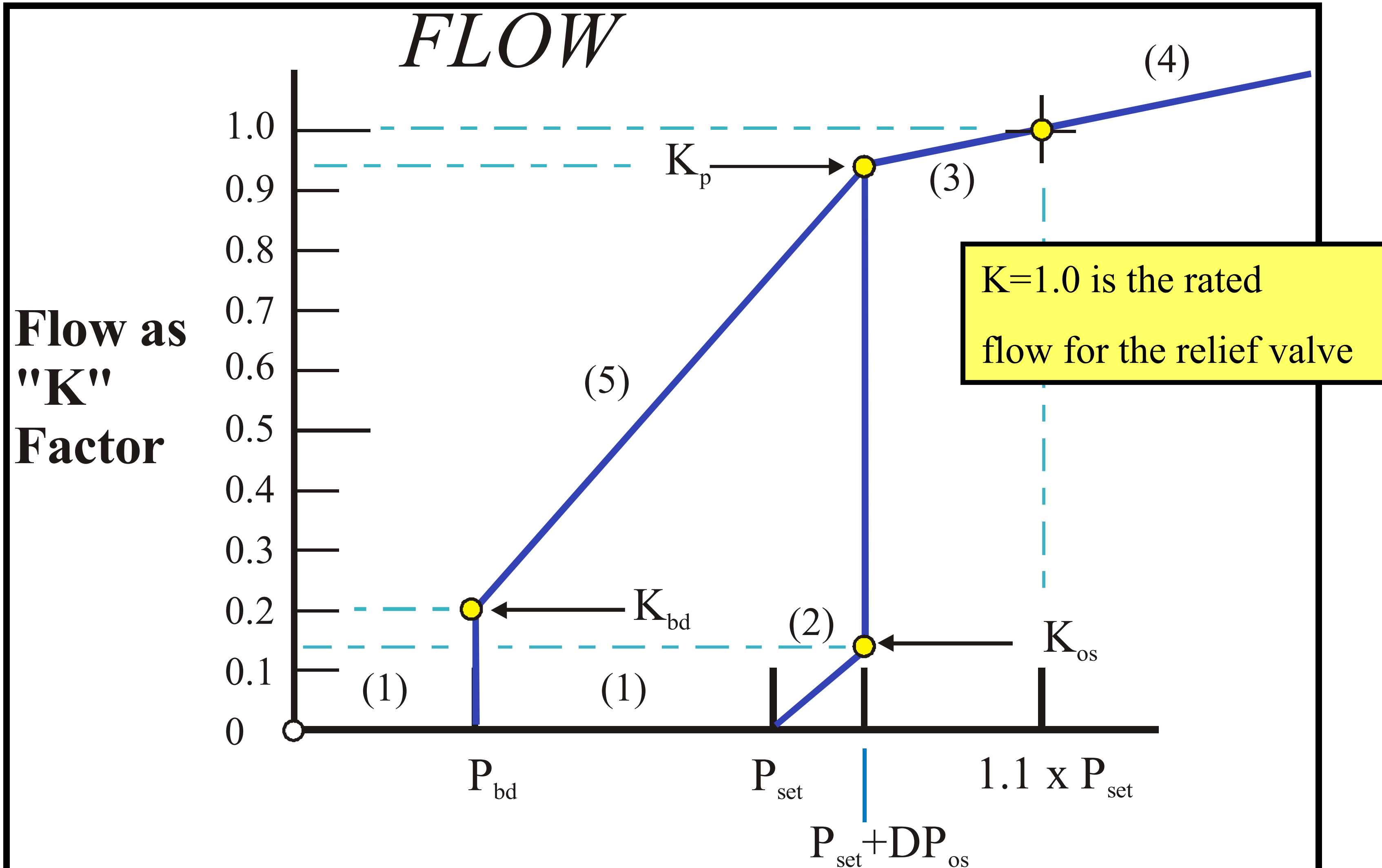
Summary of this Part (c):

- Therefore if we know the natural frequency of the valve – (not currently a published property) – all we have to do is to know how to stay away from it.
- This leads to the suggestion of replacing the valve SMD equation with the valve steady state response represented by the following figures.

Relief Valve Lift



Relief Valve Flow



We have constructed the relief valve "lift" and "flow" figures to be internally consistent

(1) The valve will start to open at P_{set} and "pop" at P_{set} + DP $_{os}$, an offset pressure difference, which may be zero

(2) The valve "pops" at P_{set} + DP $_{os}$ to a specified lift, Lp, alleged to be ≈ 65% to 75% of full lift,

(3) The valve reaches full lift and full flow at 1.1 x P_{set}

(4) The valve flows ≈ 90% of rated flow at Lp or after "popping"

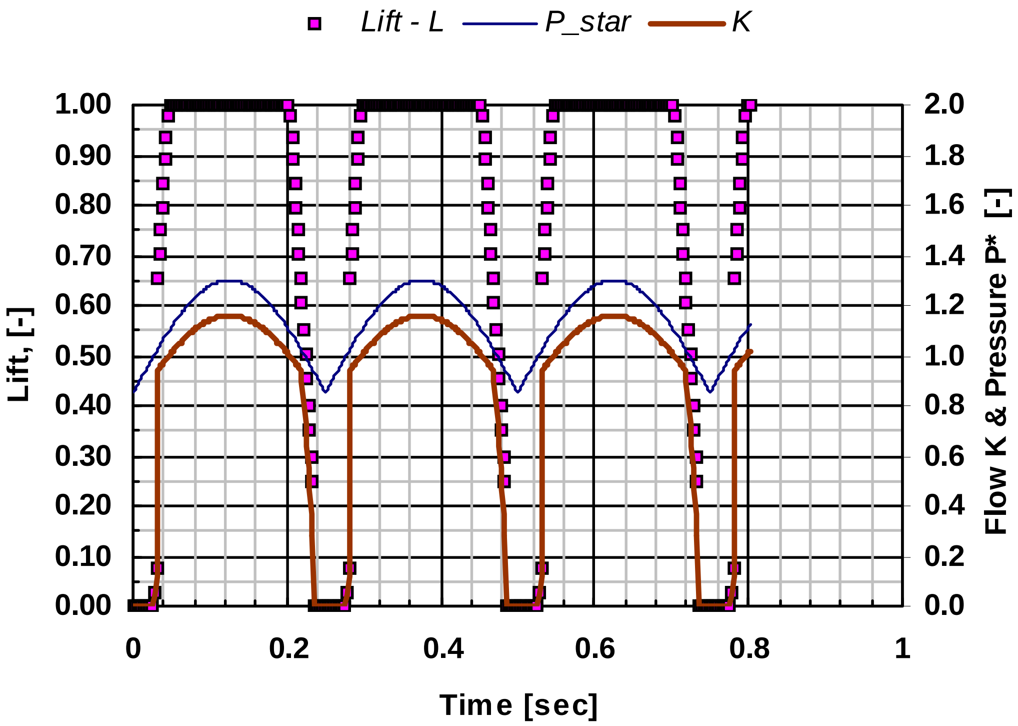
(5) The minimum flow at P_{bd} is estimated to be ≈ 25% of rated flow. P_{bd} is the blowdown pressure

(6) The P_{bd} value is estimated to be 93% of Pset, but this can vary

Operating the Lift and Flow function

(1) We have programmed the valve "Lift" and "Flow" curve (Even in Excel)

(2) It's response to a forcing function is shown below



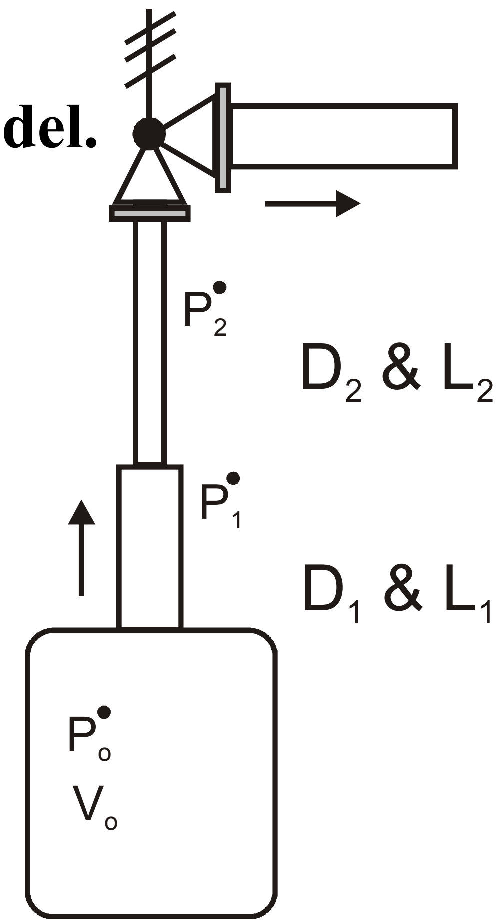
Brief Description of Model

(1) We are intrigued by the Kastor modeling approach

(2) This is roughly described in Ref. [K1]

(3) We attempt to recreate the Kastor model.

We lump the segments (1) and (2) into equivalent volumes and assign the flow resistance to an overall flow coefficient according to conventional rules



This leads to an Equation Set:

The Source Flow

$$\overset{\bullet}{P_0} = \lambda_0 \left[\frac{B\, F(k)\, P_o}{C_d A_1} - F(k,\eta)\, P_o \right]$$

$$\overset{\bullet}{P_1} = \lambda_1 \left[\frac{C_d A_1\, F(k,\eta)\, P_o}{C_d A_2} - F(k,\eta)\, P_1 \right]$$

$$\overset{\bullet}{P_2} = \lambda_2 \left[\frac{C_d A_2\, F(k,\eta)\;\; P_1}{C_d A_v} - K(P2)\, F(k)\, P_{set} \right]$$

The relief valve
Flow

Notes

- $K(P_2)$ is the flow coefficient from the valve lift and flow curves at P_2
- B is a dimensionless source flow term, a multiple of the valve rated flow:

$B < 1$ Valve is over sized: Source flow is less than valve rated flow at P_{set}

$B = 1$ Valve is "just right " sized with source flow = valve rated flow.

$B > 1$ Valve is under sized at P_{set}, source flow is > rated flow at P_{set}

Notes

- $F(k)\,\tilde{P}_{set}$ is a reduced parameter representing the relief valve rated flow.
- The segment volume pressure response time constants are:

$$\lambda_o = \frac{C_d \; A_1}{V_o}\sqrt{\square R\,T}$$

$$\lambda_1 = \frac{C_d \; A_2}{V_1}\sqrt{\square R\,T}$$

$$\lambda_2 = \frac{C_d \; A_v}{V_2}\sqrt{\square R\,T}$$

Notes

- The time constants determine the transient response of the vessel and the piping segments
- The system is “stiff” but will do O3 operation and progress to a steady state solution if there is one
- We can actually learn much from the steady state solution
- The steady state solution is contained in the terms in brackets only with the P* terms set =0

Consider the following cases

Case	B	Inlet Pressure Loss	P_{set}	Operation
1	1.0	$0.03 \times P_{set}$	P_{set}=MAWP	No O3 $P_{set} < Po < 1.1$ MAWP
2	0.8	$0.03 \times P_{set}$	P_{set}=MAWP	No O3 $Po \approx P_{set}$
3	1.0	$0.1 \times P_{set}$	P_{set}=MAWP	See case 3 note
4	1.3	$0.1 \times P_{set}$	P_{set}=0.75 x MAWP	See case 4 note
5	0.5	$0.1 \times P_{set}$	P_{set}=MAWP	See case 5 note

Case Notes

- Cases 1 & 2 are ordinary good 3% Rule compliant cases with no O3
- Cases 3, 4, & 5 are the most interesting
- Case 3: Valve and source flow are right sized, but inlet pressure loss is greater than P_{bd} Vessel pressure will rise until $P_2 > P_{bd}$. Vessel pressure exceeds 1.1 MAWP

Case Notes

- Case 4: We have excess inlet pressure loss, but we reduce the relief valve set pressure so that it is undersized at the set pressure, but adequate at 1.1 MAWP. At this point the 10% inlet pressure loss – fully adjusted for the higher pressure and flow is safely above the valve P_{bd}. The valve performs satisfactory. There is a short period of O3 amounting to a few cycles. This is determined by λo

Case Notes

- Case 5: This is the worst case. An oversized valve with excess inlet pressure loss. Will do O3 indefinitely. Will “chatter” indefinitely with frequency depending on λo

Summary

- These operating condition can be figured out from the system “steady state” solution
- In some cases O3 will lead to vessel pressure increase to greater than 1.1 MAWP – to seek a stable operating condition with $P_2 > P_{bd}$

Summary

- However, when confronted with excess inlet pressure loss, there is a viable “solution” to be obtained by reducing the relief valve set pressure relative to MAWP. The result is that the vessel pressure can rise to MAWP and the excess inlet pressure loss will still be well above P_{bd} with the reduced set pressure
- The written paper will provide additional explanatory details. A copy will be submitted to the meeting minutes. To receive a copy email; ctimag@hughes.net

Implications for 3% Rule

(1) An alternative and technically superior way of avoiding O^3 and chatter is to require inlet pressure loss to be related to P_{bd}. That is, if P_{bd} is 7% of P_{set}, then inlet pressure loss should not get one any closer than 2 percent points to shutoff. ***We need to have better and reliable information for the blowdown or reclosing pressure.***

(2) The present 3% Rule does not consider the positive effects of allowance for overpressure. For example the fire case allowance for 21% overpressure should allow for at least 10% to 15% inlet pressure loss as long as this effect is properly included in the sizing calculations

Implications for 3% Rule

(3) The present 3% Rule does not consider the positive effects of reducing the set pressure relative to the MAWP. Depending on the set pressure reduction, the inlet pressure loss can be much greater than 3% and relief flow will be entirely stable

Things one would like to see from Relief Valve Manufacturers

(1) Valve spring constants and natural frequency data – if not in the general catalog at least with the purchased valve certification sheets

(2) Better relief valve Lift and Flow curves

(3) Better data and guidance on relief valve blowdown

Final Thoughts: The 3% Rule should not become a rule with no exceptions – we have indicated several possible alternative routes to stable operation with greater than 3% inlet pressure loss

Implications for 3% Rule

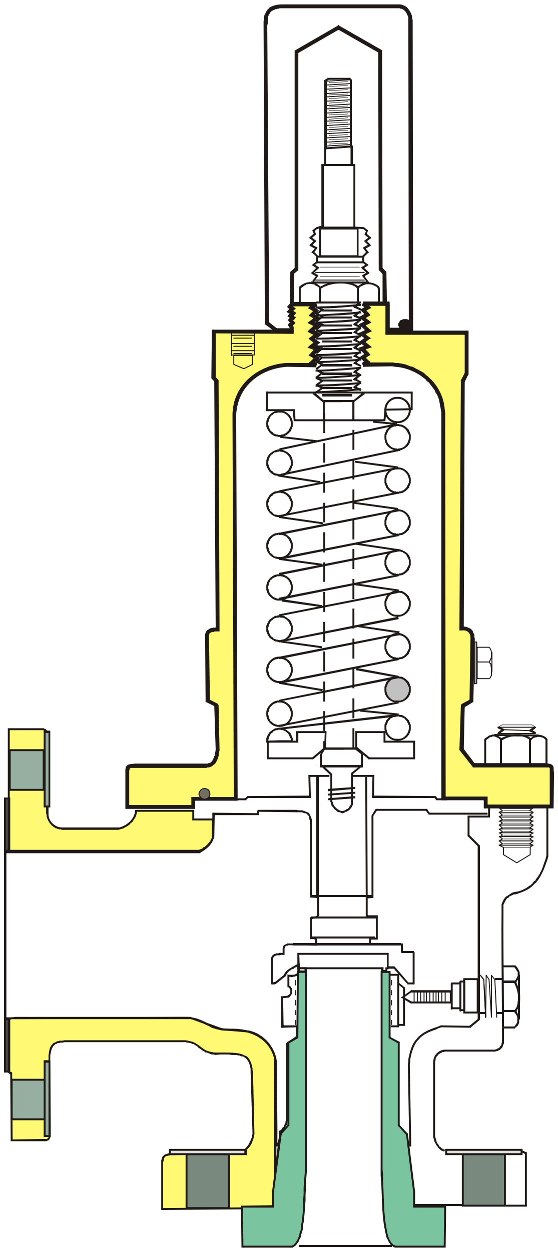
(3) Reducing the relief valve set pressure below the vessel design pressure can also have the same positive effect on allowance for larger inlet pressure loss

(4) A very good situation is to have the relief valve "undersized" at the relief set pressure and properly sized for some defined overpressure. In this case 3% inlet pressure loss is irrelevant if the relief valve design properly considers the actual inlet pressure loss in the sizing calculation. Undersized at the relief set pressure is common practice for runaway tempered reactions as a class

Acknowledgement

- Thanks to Harold Fisher
- Harold has no responsibility for errors in this material
- Harold has provided useful input and comments along the way. His help is much appreciated

mag



There is much more to this story, but for now

That's All Folks!!

Centaurus Technology, Inc.

ioMosaic

Business Confidential Document

Deal with Controversial Topics in Pressure Relief Systems

Paper Presented at the Spring 2009 DIERS Users Group Meeting, Orlando, Florida

By

G. A. Melhem, Ph.D.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

There are several technical issues that are currently the subject of debate and some controversy amongst relief systems experts including but not limited to:

- The use of actual (best estimate) flow vs. required flow for (a) inlet pressure loss, (b) backpressure, and (c) sub-header/header flare hydraulics
- The use of 3 % requirement for inlet pressure loss vs. use of blowdown minus 2 % for new and old installations
- The correct usage of a two phase discharge coefficient
- How to estimate two phase density where slip is involved
- The use of fire flux for dynamic simulations and the issue of decreasing wetted surface area for all gas flow as well as use of total vessel surface area for two-phase flow
- Fire exposure of gas filled vessels
- How much documentation is sufficient to meet the OSHA PSI requirements. Do API Standards apply to chemical facilities?

ioMosaic

What is Recognized & Generally Accepted Good Engineering Practice (RAGAGEP)?

- Simply stated RAGAGEP is a consensus code, standard or guideline that documents the engineering practices for the evaluation, design, fabrication, installation, maintenance, and/or inspection and testing of equipment. RAGAGEP can be (a) Mandatory (i.e., regulatory or contract related), (b) suggested, and/or (c) common practice.

- Moreover RAGAGEP defines the **standard of care** expected of companies by regulatory agencies, government, and society in operating chemical manufacturing (and other) businesses.

- RAGAGEP is specifically referenced in Federal Regulations from Occupational Safety and Health Administration (OSHA) and the Environmental Protection Agency (EPA), and is also mentioned in the American Chemistry Council's (ACC) Responsible Care® Process Safety Code.

Recommended Reading:

G. A. Melhem, "Relief System's Last Line of Defense, Only Line of Defense?", Process Safety Progress, Volume 25, No. 4, December 2006.

ioMosaic

For pressure relief systems design, the definition of RAGAGEP is very broad

- RAGAGEP includes (but is not limited to):
  - Recommended engineering practices such as API-520, API-521, API-2000, NFPA-30,
  - More than fifty CCPS Guidelines,
  - Numerous Responsible Care publications,
  - Design Institute for Emergency Relief Design (DIERS) methods for two-phase flow,
  - ASME Standards,
  - Corporate Engineering Guidelines and internal corporate standards,
  - Contractors engineering standards, training manuals,
  - Guidance documents provided by relief systems manufacturers,
  - Authoritative open literature publications on pressure relief systems evaluations and design,
  - etc.

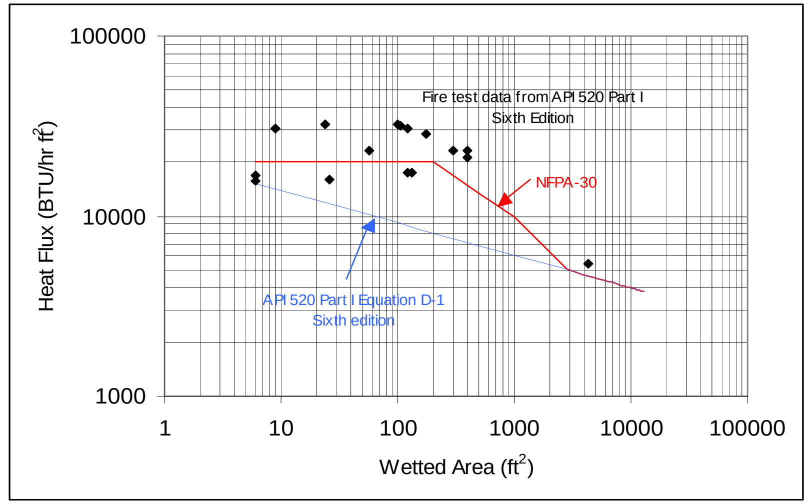
ioMosaic

There are several issues associated with fire exposure that need to be considered

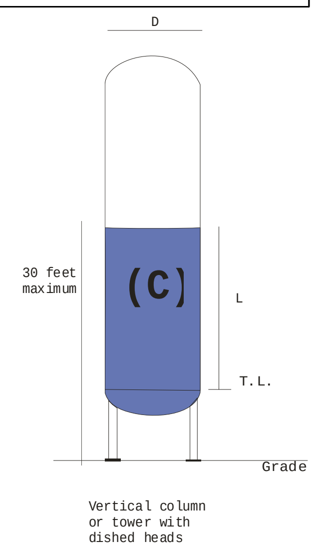
- Wetted surface area – All wetted surface area should be used for multiphase flow
- Flame emissive power – Existing API/NFPA-30 fire flux is not conservative for pool fires of light hydrocarbons (< C6)
- Flame height limitation to 25 or 30 ft is only adequate for very small leaks – typically flame height is between 2 to 4 times the burning pool diameter
- Heat transfer into vessel should be based on actual heat transfer dynamics for those who use computer codes – Heat input decrease with wetted surface area for all gas flow
- Fire duration
- Fire exposure of gas filled vessels
- Fire exposure for vessels containing reactive chemicals

ioMosaic

Existing guidance by NFPA and API regarding fire heat input into vessels may not be conservative for fuels < C6



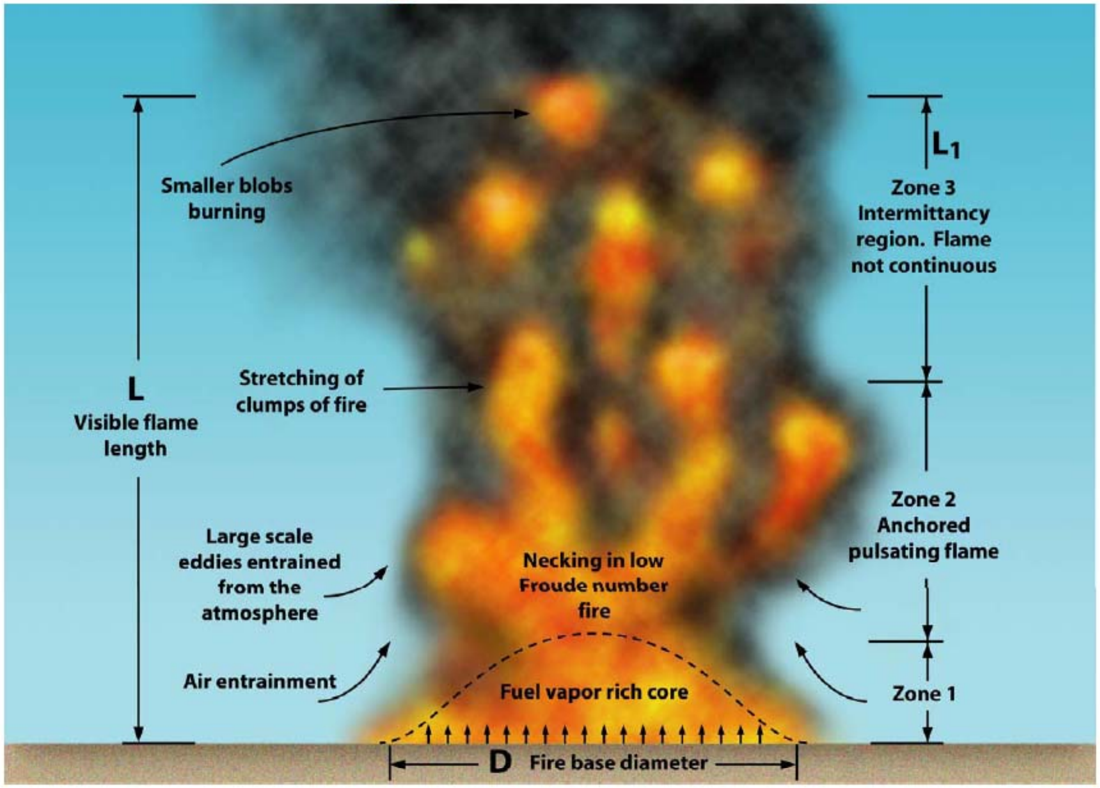
- Flame emissive power
  - Heat input is underestimated for light hydrocarbon fuels
- Wetted surface area
  - The wetted surface area is underestimated when two-phase flow occurs
- Flame height
  - Eliminating heat input for structures elevated above 25 or 30 ft implies a pool fire diameter of less than 10 ft
- Fire Duration
  - The actual fire duration is assumed



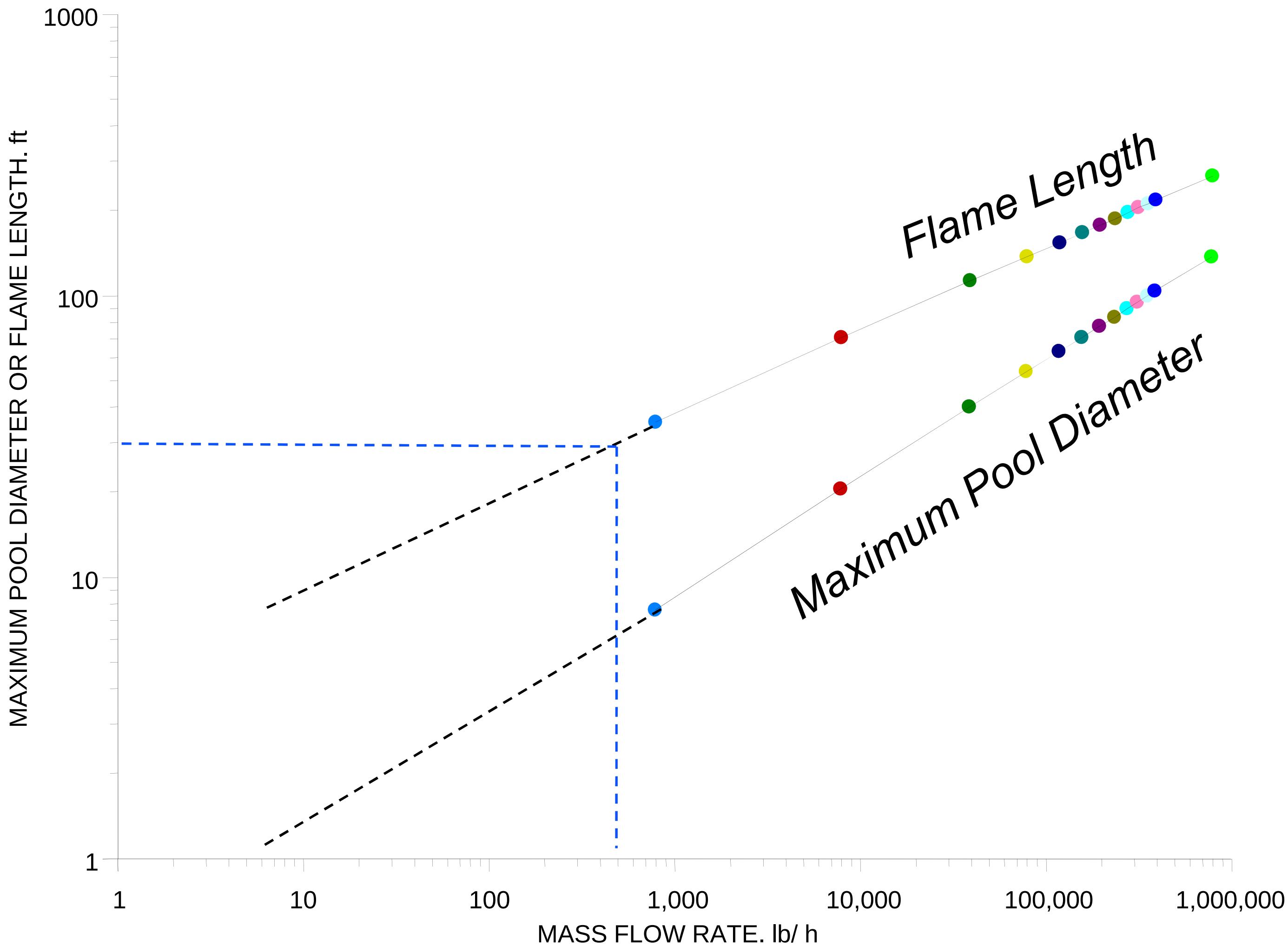
ioMosaic

Let's consider a 1 hour release of liquid cyclohexane on a concrete surface at different flow rates

- Vary flow rate from 0.1 kg/s to 100 kg/s
- Assume ignition occurs 1 minute after the release starts
- Assume the wind speed is 2 m/s and the relative humidity is 70 % at a visual range of 6,600 ft
- Spill surface is concrete

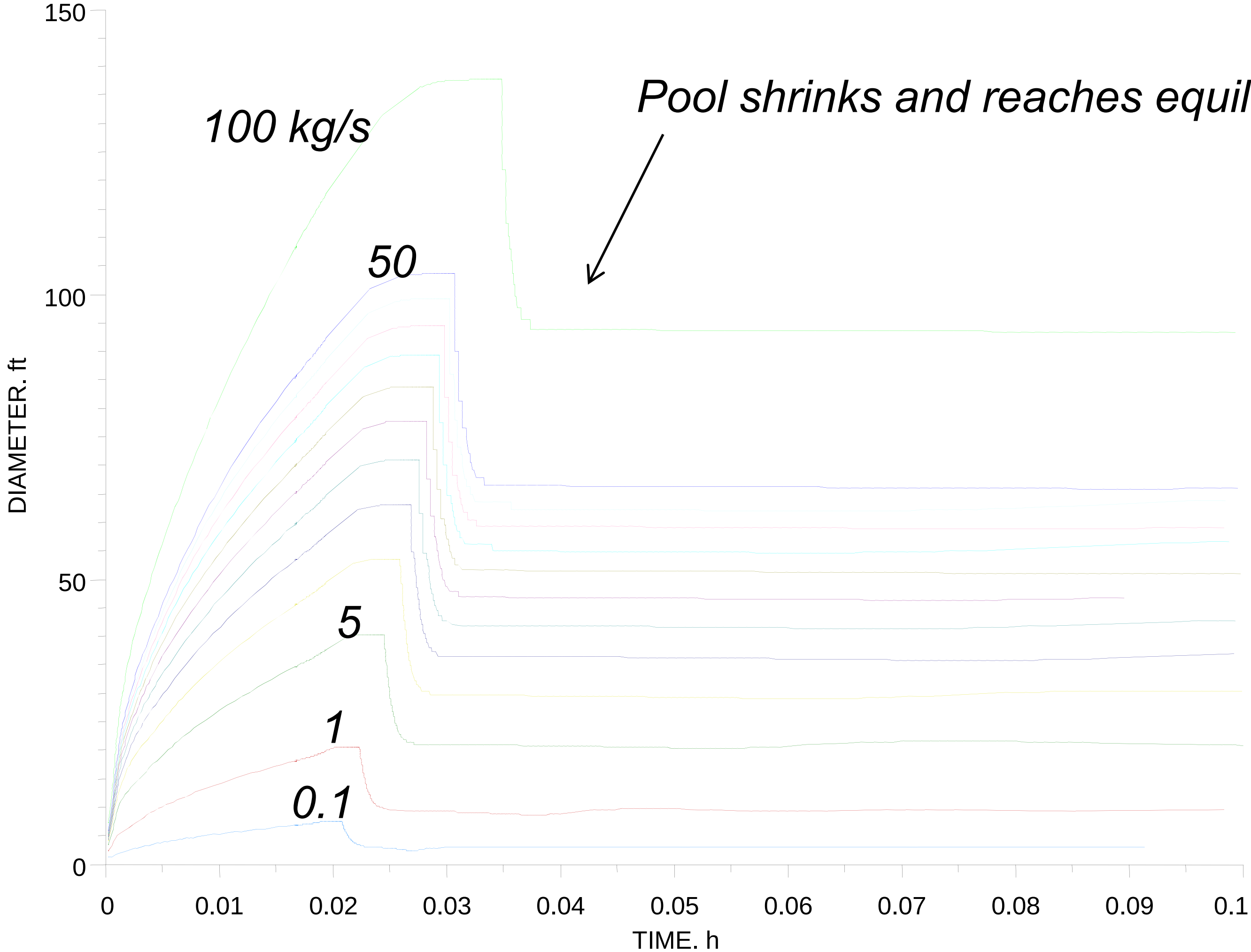


This simple example clearly illustrates that the NFPA/API flame height restriction only applies to small leaks



Source: SuperChems Expert v5.98mp

The maximum pool diameter and flame height depend on the ignition time and fuel burning rate



Source: SuperChems Expert v5.98mp

ioMosaic

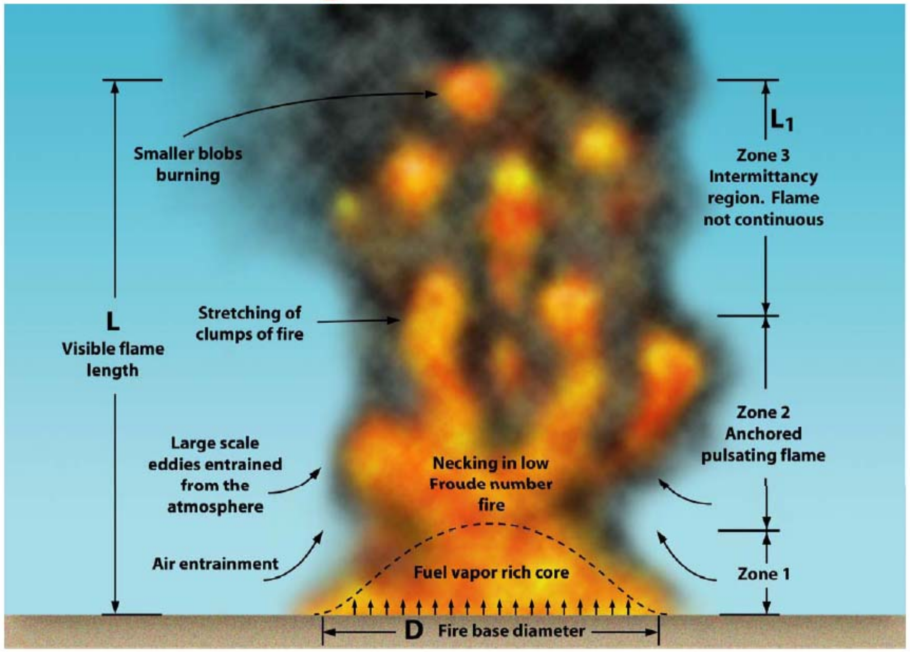
Many useful correlations were published to estimate the flame emissive power for hydrocarbon fuels

$$E_p = 117 - 0.313 T_{NBP}$$

E_p = *Flame Emissive Power (kW/m²) ≥ 20*

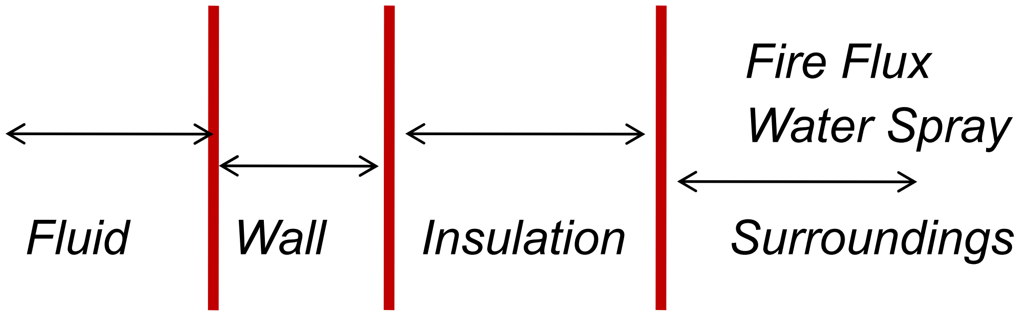
T_{NBP} = *Normal Boiling Point (F)*

FUEL	Carbon Number	Normal Boiling Point (F)	Estimated Flame Emissive Power (kW/m2)
Methane	C1	-258.68	198
Ethane	C2	-127.48	157
Propane	C3	-43.67	131
Butane	C4	31.10	107
Pentane	C5	96.92	87
Hexane	C6	155.71	68
Heptane	C7	209.17	52
Octane	C8	258.22	36
Nonane	C9	303.48	22
Decane	C10	345.48	20



Source: SuperChems Expert v5.98mp

We can do much better than just using NFPA-30 or API fire flux



$$\frac{d}{dt}\left[Ax_{wall}C_{p,wall}T_{wall}\right]=h_{fluid,wall}A\left(T_{fluid}-T_{wall}\right)-\frac{k_{ins}}{x_{ins}}A\left(T_{wall}-T_{ins}\right)$$

$$\frac{1}{2}\frac{d}{dt}\left[Ax_{ins}C_{p,ins}T_{ins}\right]=\frac{k_{ins}}{x_{ins}}A\left(T_{wall}-T_{ins}\right)+h_{ins,surr}A\left(T_{ins}-T_{surr}\right)$$

ioMosaic

We will first examine the origin and implications of the 3 % rule for inlet pressure loss

- ASME
- API
- CCPS
- API is the source, ASME adopted the 3 % rule in the mid 1980's
- **Recommended Reading:**
  - N. E. Sylvander and D. L. Katz, ***"The Design and Construction of Pressure Relieving Systems"***, Engineering Research Bulletin No. 31, University of Michigan Press, Ann Arbor, April 1948.
  - G. A. Melhem and H. G. Fisher***, "Practical Guidelines for Dealing with Excessive Pressure Drop in Relief Systems"***, D-32-160-1, DIERS Users Group Fall Meeting, 2003.
  - Ed Zamjec, ***"Origin of the 3 % Rule",*** Presentation at the Fall 2007 API Refining Meeting, San Antonio, Texas.
  - M. A. Grolmes, ***"Odd and Ends - Relief Valve Stability and Inlet Pressure Loss"***, DIERS Users Group Spring Meeting, 2008.

ioMosaic

A large fraction of existing relief device installations suffer from excessive pressure loss (inlet and/or outlet)

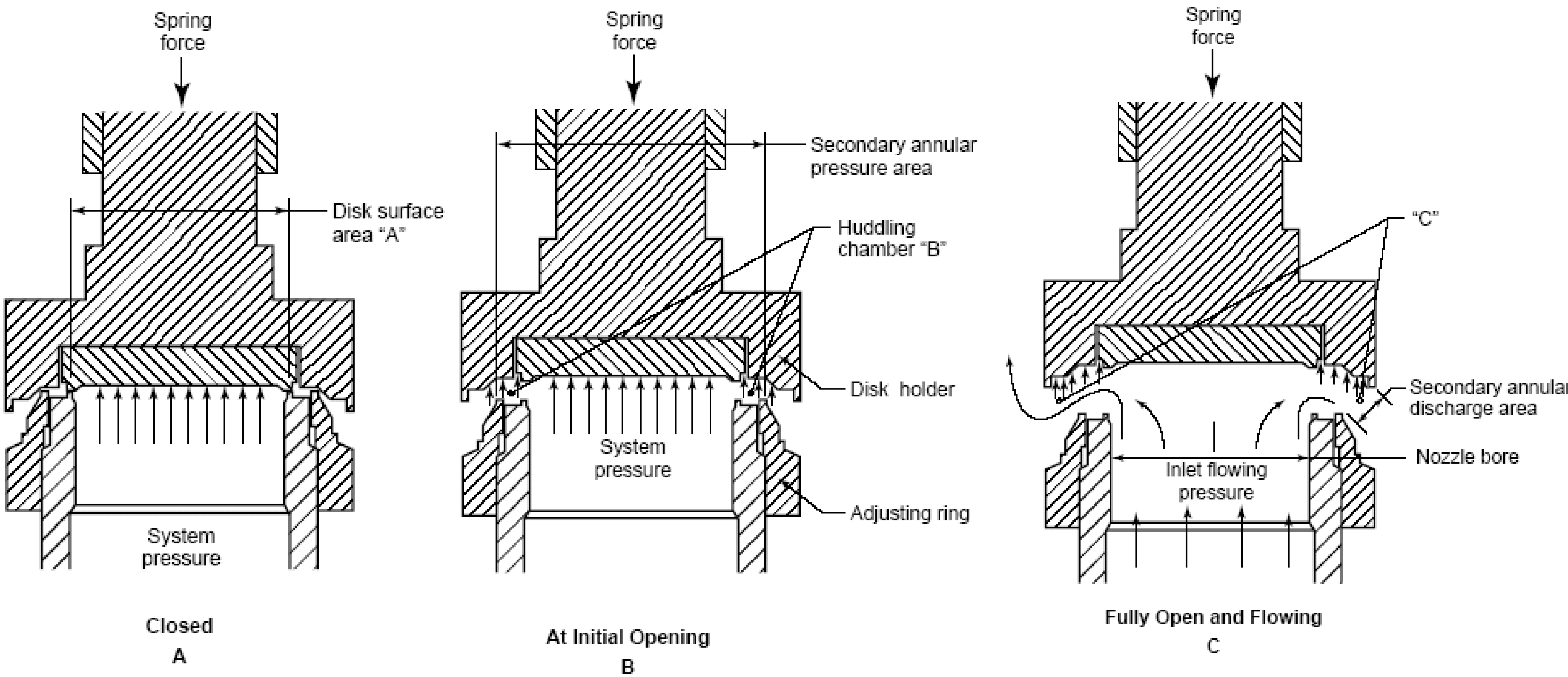
- Excessive pressure loss can lead valve instability and possibly valve failure
- Operating companies are faced with significant upgrade/mitigation costs
- We examine key factors leading to valve instability and provide some practical guidance on mitigation options

ioMosaic

There is a difference between chatter, flutter, and simmering

- Chatter is an abnormal rapid reciprocating motion of the movable parts of a pressure relief valve in which the disc contacts the seat
- The impact on the seat is usually very strong, and therefore can damage the valve. Chatter can occur in either vapor or liquid service
- Flutter is the same reciprocating motion, but the disc does not contact the seat
- Simmering is the audible or visible escape of fluid between the seat and disc at an inlet static pressure below the popping pressure and at no measurable capacity
- Blowdown is the difference between the popping pressure and reseating pressure expressed either as a percentage of the popping pressure or in pressure units

A Typical Relief Valve Model : Force Dynamics During Flow



Pressure relief valve Operation: Vapor/Gas Service. Source API RP 520 iōMosaic

A Typical Relief Valve Model : Force Dynamics During Flow

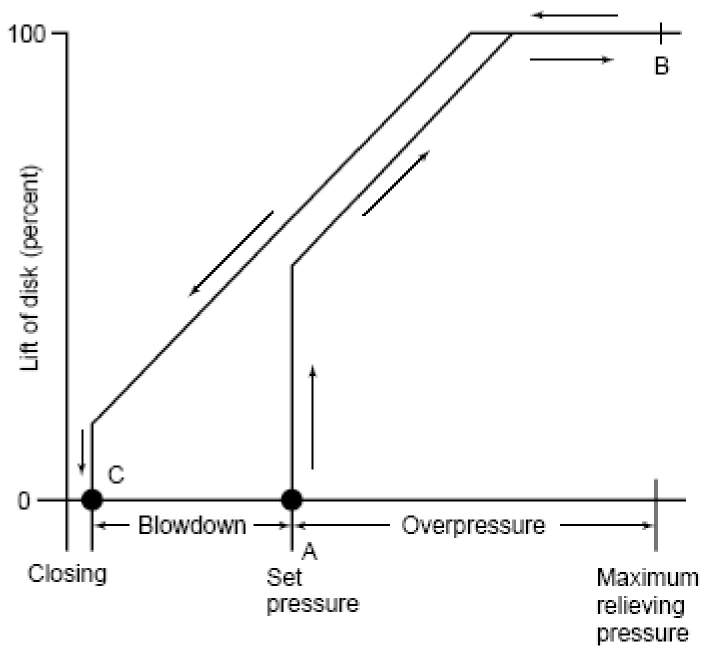
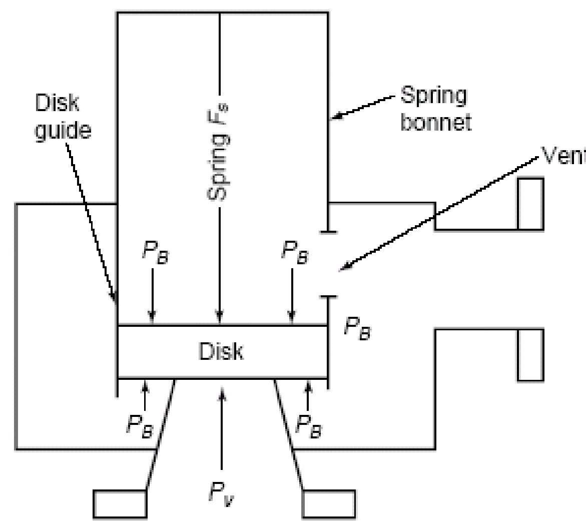


Figure 20—Typical Relationship Between Lift of Disk in a Pressure Relief Valve and Vessel Pressure

Spring Bonnet
Vented to Valve Discharge



Back pressure increases set pressure

$$P_V A_N = F_S + (P_B A_N)$$

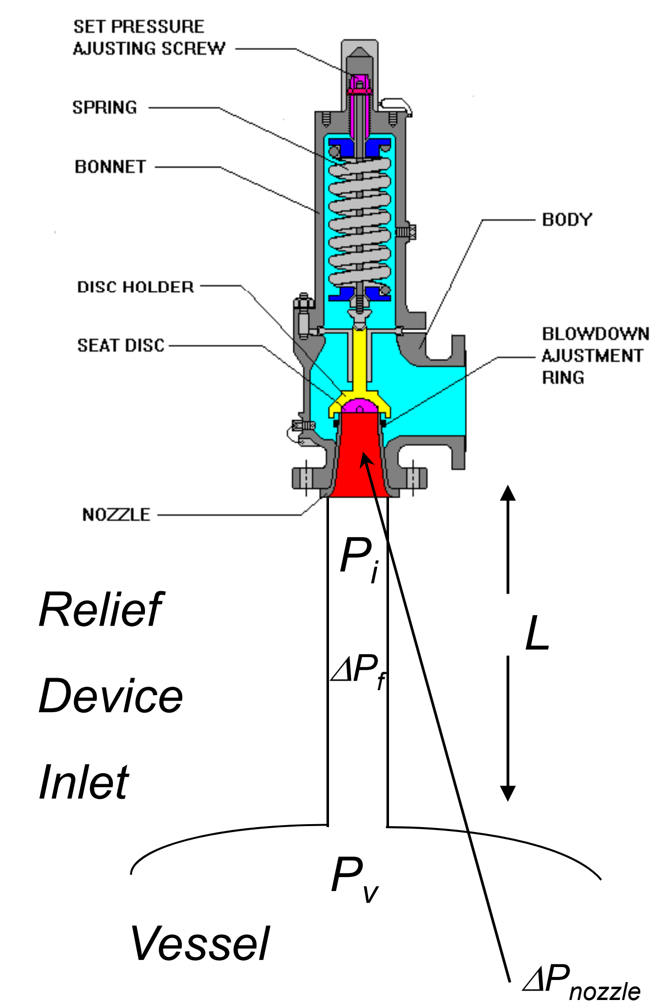
$A_D > A_N$

A_D = disk area,
A_N = nozzle seat area,
F_S = spring force,
P_V = vessel pressure in pounds per square inch gauge,
P_B = superimposed back pressure, in pounds per square inch gauge.

Figure 22—Typical Effects of Superimposed Back Pressure on the Opening Pressure of Conventional Pressure Relief Valves

Source API RP 520

It is not a perfect valve



$$t_{Chatter} \approx 2\frac{L}{c_o}$$

$$\text{Blowdown} \approx \Delta P_f + \Delta P_{\text{nozzle}} + \Delta P_{\text{safety margin}}$$

For short pipe lengths, $t_{chatter}$ may be smaller than the actual valve opening or closing time

The valve can close due to excessive inlet pressure loss or excessive backpressure development

ioMosaic

The momentum exchange between the fluid impinging on disk surface and the disk is an important factor

- If the fluid characteristic chatter time is less than the valve closure time, the valve will flutter and not chatter
- If the fluid characteristic chatter time is more than the valve closure time, the valve will chatter
- If the fluid characteristic chatter time is equal to the valve closure time, the valve will chatter with increased severity due to acoustic coupling
- Note that damage to the valve and piping can occur as the pressure in the vessel increases to 10 % or 21 %

ASME is often the referenced source of the 3% inlet pressure loss rule. ASME Section VIII, Div 1 (2008), non-mandatory Appendix M-6(a) states:

- *"(a) The nominal pipe size of all piping, valves and fittings, and vessel components between a pressure vessel and its safety, safety relief, or pilot operated pressure relief valves shall be at least as large as the nominal size of the device inlet, and the flow characteristics of the upstream system **shall** be such that the cumulative total of all non-recoverable inlet losses shall not exceed 3% of the valve set pressure. The inlet pressure losses will be based on the valve nameplate capacity corrected for the characteristics of the flowing fluid."*

- ASME Section VIII, Div 2 (2008), informative annex 9.A.3 provides similar guidance. Additional guidance is not provided in ASME Section VIII, Div 3 (2008).

- Note the use of the word "shall" above. Also note that Appendix M is a non-mandatory appendix that is for information and, thus, this is a suggested, not mandatory standard. ASME Section VIII, Div 1 (2008) Section U-1 Scope formally defines non-mandatory appendices as:
  - *"The Non-mandatory Appendices provide information and suggested good practices."*

ioMosaic

A review of prior versions of ASME Section VIII, Div 1 indicates that the 3% rule became part of the code in the early to mid 1980's.

- ASME Section VIII, Div. 1, 1986, M-7(a) has the same wording as in the 2008 edition, M-6(a).

- The 1980 edition of ASME Section VIII, Div. 1 Appendix M makes no other mention of the 3% criterion or any other general guidance on inlet piping design.

- Neither inlet piping design guidance nor pressure loss criteria are given in the 1977, 1974, 1968, 1965, or the 1962 editions of ASME Section VIII, Div 1 Appendix M.

ioMosaic

ASME Section VIII, Div. 1 provides mandatory guidance on general inlet piping design without addressing pressure loss. This mandatory guidance has not significantly changed since 1962.

- *"The opening through all pipe and fittings between a pressure vessel and its pressure-relieving device shall have at least the area of the pressure-relieving device inlet, and in all cases shall have sufficient area so as not to unduly restrict the flow to the pressure-relieving device. The opening in the vessel wall shall be designed to provide direct and unobstructed flow between the vessel and its pressure-relieving device."*

- The 3% criterion was introduced into ASME Section VIII, Div. 1 non-mandatory Appendix M in the early to mid 1980's

- ASME does not provide a basis for why the 3 % rule was included.

ioMosaic

ASME Section I (power boilers) does not provide limits on inlet pressure loss.

- ASME Section I (2008), section PG-72.1 states:
  - *"Safety valves and safety relief valves shall be designed and constructed to operate without chattering, with a minimum blowdown of 2 psi (15 kPa) or 2% of the set pressure, whichever is greater, and to attain full lift at a pressure not greater than 3% above their set pressure."*
- ASME Section I (2008), section PG-71.2 states:
  - *"The safety valve or safety relief valve or valves shall be connected to the boiler independent of any other connection, and attached as close as possible to the boiler or the normal steam flow path, without any unnecessary intervening pipe or fitting. Such intervening pipe or fitting shall be not longer than the face-to-face dimension of the corresponding tee fitting of the same diameter and pressure under the applicable ASME Standard listed in PG-42 and shall also comply with PG-8 and PG-39."*
- ASME Section I implies qualitative guidance on minimizing inlet pressure loss rather than limiting the inlet pressure loss to any defined value.

ioMosaic

The current edition of API RP-520 Part 2, 5th Ed (2003) uses the word "should" which suggests rather than mandates the 3% criterion.

- *"When a pressure-relief valve is installed on a line directly connected to a vessel, the total non-recoverable pressure loss between the protected equipment and the pressure-relief valve should not exceed 3 percent of the set pressure of the valve except as permitted in 4.2.3 for pilot-operated pressure relief valves."*

- *"When a pressure-relief valve is installed on a process line, the 3 percent limit should be applied to the sum of the loss in the normally non-flowing pressure-relief valve inlet pipe and the incremental pressure loss in the process line caused by the flow through the pressure-relief valve. The pressure loss should be calculated using the rated capacity of the pressure-relief valve."*

- *"An engineering analysis of the valve performance at higher inlet losses may permit increasing the allowable pressure loss above 3 percent."*

- The type and form of engineering analysis is to be selected by the user.

ioMosaic

The 3 % rule appeared in the API RP 520 2nd edition (1963) section 2.2(b):

- *“For gas, vapor, or flashing-liquid service, it is recommended that the inlet piping between the protected equipment and the inlet flange of the pressure relief valve be designed so that the total pressure loss in the line shall be the sum total of the inlet loss, line loss, and valve loss and shall not exceed 3 percent of the set pressure, in pounds per square inch gage, of the valve. This pressure loss should include any stop valve loss. It should be calculated using the maximum rated flow through the pressure relief valve. Such losses can be reduced materially by rounding the entrance to the inlet piping or by the use of larger inlet piping.”*

- API RP 520 1st edition (1955) does not provide guidance regarding inlet piping.

ioMosaic

API sponsored work at the University of Michigan relating to pressure relief system design in the 1940s

- See N.E. Sylvander and D.L. Katz and entitled “The Design and Construction of Pressure Relieving Systems”, University of Michigan Press Engineering Research Bulletin 31, April 1948, Pages 72-73:
  - *“Pressure drop through inlet piping has a two-fold importance in relief system design. First, flow capacity varies with the pressure drop available. Second, the operating characteristics of many relief devices indicate that improper pressure drop on the inlet side may cause intermittent operation. Conventional spring-loaded relief valves have the rate of flow affected by the inlet pressure drop, since the pressure at the inlet to the valve will then be below the vessel pressure. If an excessive pressure drop occurs through the inlet piping, the valve tends to close prematurely. As the valve closes, pressure within the protected equipment builds up rapidly and the valve is forced open again. This action results in what is commonly referred to as "chattering." Chattering creates serious vibrational disturbances and can result in damage to the relief valve parts and possible failures of piping connections.“*

Sylvander and Katz also provide simple qualitative guidance on minimizing inlet pressure loss:

- *"The design of inlet piping to relief devices need not be complicated. Careful attention to the principles outlined here should result in satisfactory installations. Frequently increased capacity can be obtained by using a well-rounded approach in the nozzle leaving the protected equipment. This well-rounded approach becomes increasingly important when the ratio of the actual area of the relief device to the actual area of the inlet piping approaches unity."*

ioMosaic

The total allowable inlet pressure drop recommended by Sylvander and Katz is 3% of the allowable pressure for capacity relief for a relief valve with a 4 % blowdown

- *"Rupture discs and pilot operated valves are not susceptible to intermittent operation. The rate of flow may be decreased by large pressure drops through the inlet connections. "*
- "*For a relief valve having approximately 4 per cent blowdown (that is, the valve will snap shut when the pressure has decreased to 4 per cent below the opening or set pressure), these recommendations are made:*
  - *The pressure drop due to friction should not exceed 1 per cent of the allowable pressure for capacity relief.*
  - *The pressure drop due to the conversion of pressure to kinetic energy, commonly referred to as velocity head loss, should not exceed 2 per cent of the allowable pressure for capacity relief*."
- The Sylvander and Katz document appears to be the source for the 3% rule.
- This a reasonable conclusion since the Sylvander and Katz document served as a primary source for guidance in the first editions of API 520 and 521.

The recoverable inlet pressure loss is no longer considered

- Sylvander and Katz
    - Inlet pressure loss considers both friction and recoverable loss (dynamic pressure)
- The CCPS “Guidelines for Pressure Relief and Effluent Handling Systems” section 2.4.2.2.1 states:
    - *“Note that the non-recoverable pressure loss from the vessel to the valve is less than the pressure drop, since the drop includes the change in velocity head from vessel to valve. This velocity head is recoverable (part of the lifting force on the disk), and thus is not included in the determination of inlet loss.”*

ioMosaic

Meeting the 3 % rule does not guarantee that a pressure relief valve will always operate in a stable state and without problems associated with the valve or inlet piping.

- Operating close to the set pressure can result in continuous or intermittent leakage across the valve leading to potential intermittent vibration on the piping or valve internals leading to a potential for fatigue failure and loss-of-containment.
- Installation of the pressure relief valve near locations of high turbulence or pulsation can result in fatigue failure of the inlet piping or premature lifting/failing open of the pressure relief valve.
- Relieving at a much lower flow rate than the rated capacity (< 25 %) can cause heavy cycling/chatter of the valve with consequent fatigue failure of the inlet piping or the pressure relief valve failing open.
- Relieving an incompressible liquid through a vapor trim valve, especially at a much lower rate than the capacity of the valve, can cause heavy cycling/chatter of the valve with subsequent fatigue failure of the inlet piping or failing open of the pressure relief valve due to repeated mechanical liquid-hammer.
- Extended relief duration can cause fatigue failure of the inlet or outlet piping if the piping is inadequately supported and/or has a thinner wall diameter.
- Plugging, degradation, corrosion, abnormal wear-and-tear, etc. of the valve or inlet piping can also lead to a multitude of other problems.

ioMosaic

The API Joint Industry Project (JIP) on spring-loaded pressure relief valve stability is still ongoing. The findings of this project will most likely result in less stringent new guidance regarding inlet pressure loss.

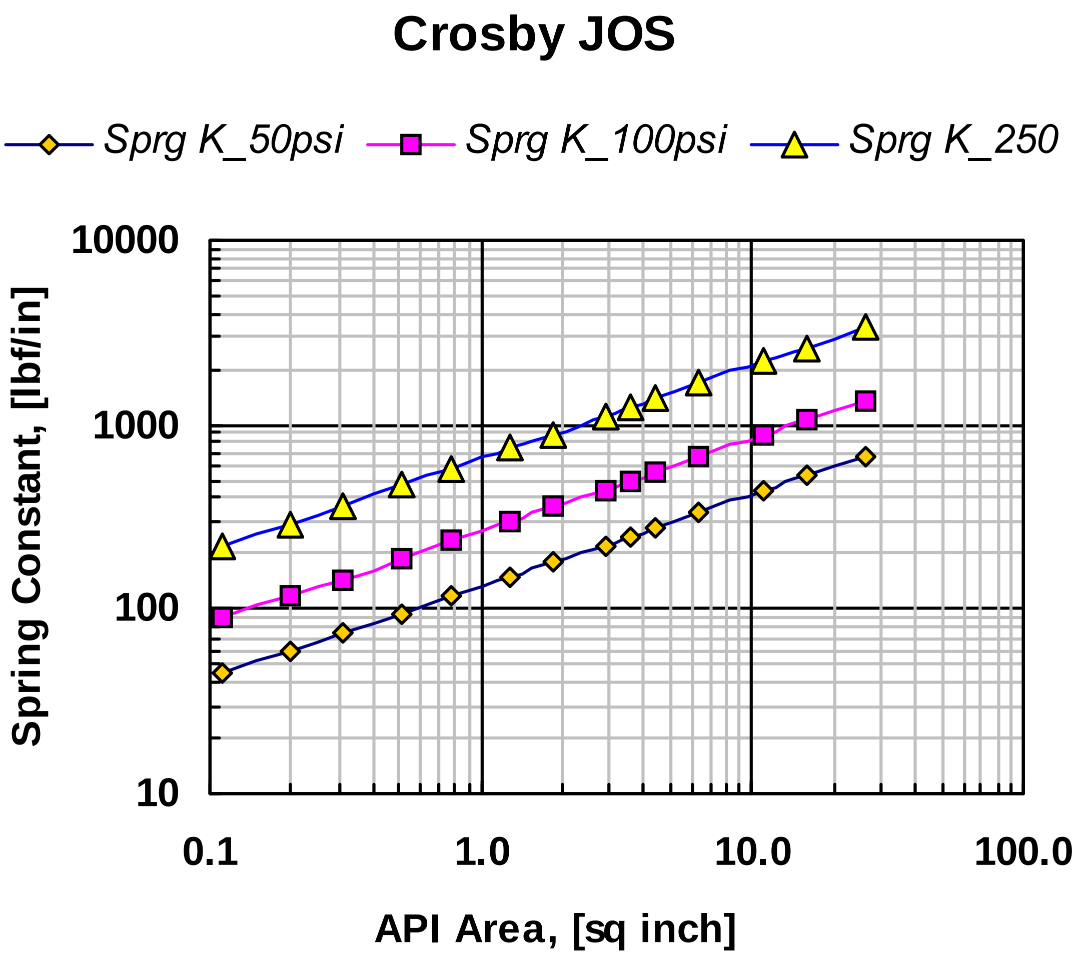
- JIP planning started in 1998
- Goal of providing a thorough technical review and testing of pressure relief valve stability.
  - Phase 1 - Literature Review - Completed
  - Phase 2 - Industry Survey - Completed
  - Phase 3 - Development of a mathematical model - Completed
  - Phase 4 - Measurements of Opening Time - Completed.
  - Phase 5 - Model Validation – Ongoing – Developed by Prof. Ron Darby
    - Model is to predict conditions and piping configurations where chattering will occur
    - Model conditions and piping configurations will be used to perform full-scale tests for validation

Results from the JIP program are currently only available to the project sponsors

- Most of the project documents will be released to the general public some time after the project is completed.
- The results from this project will most likely replace the 3% criteria in both API and ASME with a comprehensive approach to pressure relief valve stability.
- For more information on the JIP program, please contact API

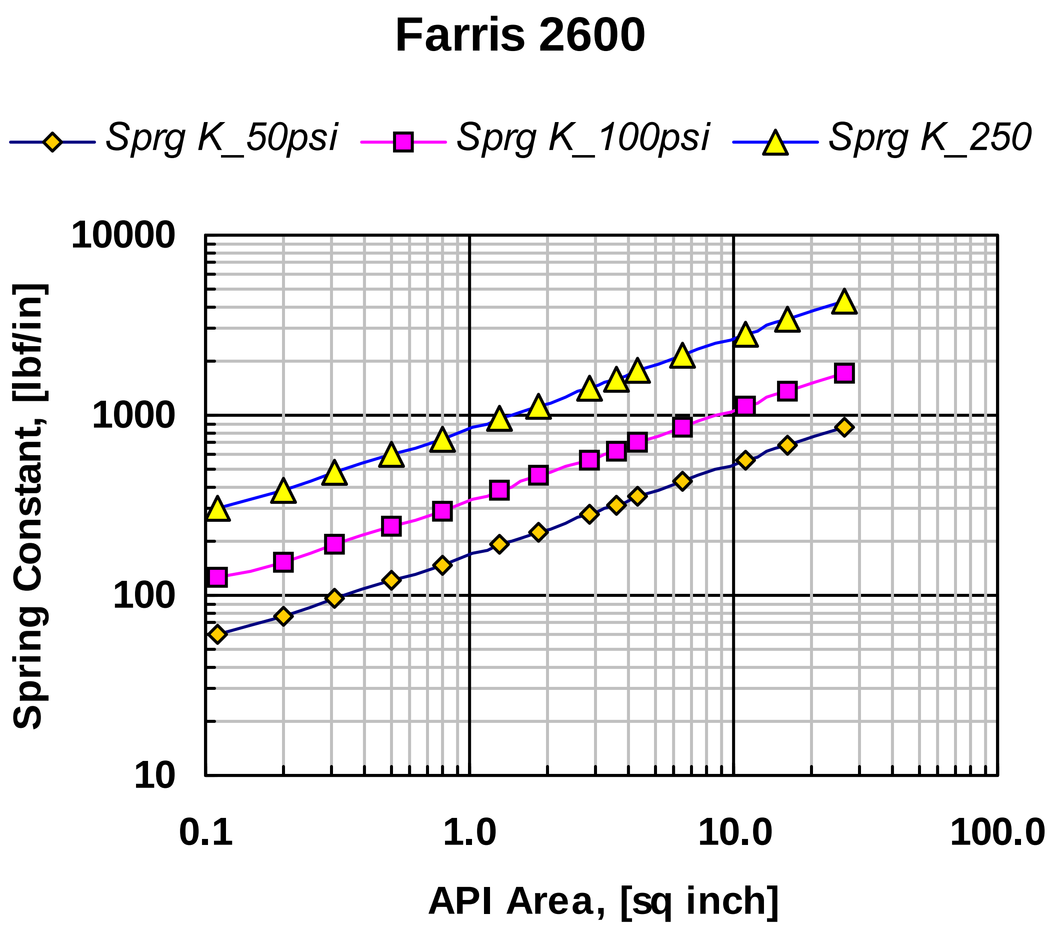
ioMosaic

Relief Valve Spring Constant Estimates – Taken From Grolmes

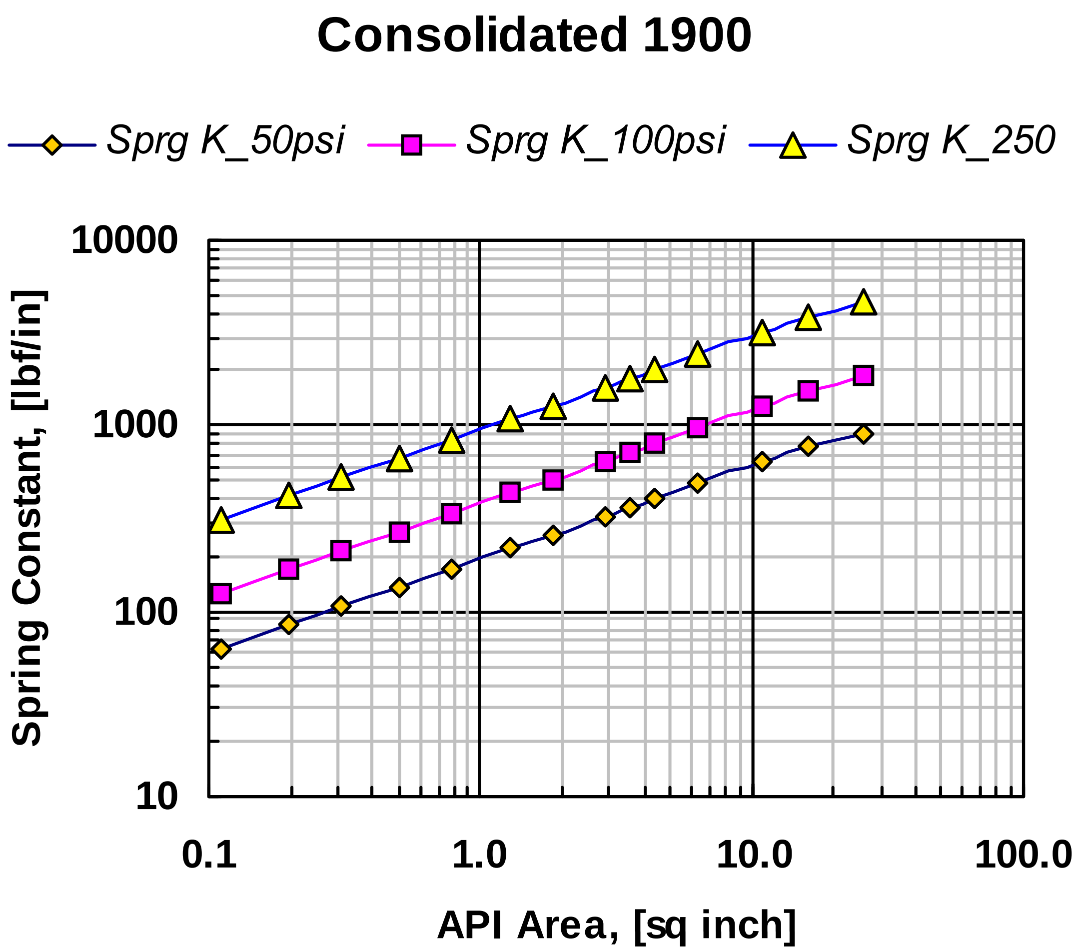


ioMosaic

Relief Valve Spring Constant Estimates – Taken From Grolmes

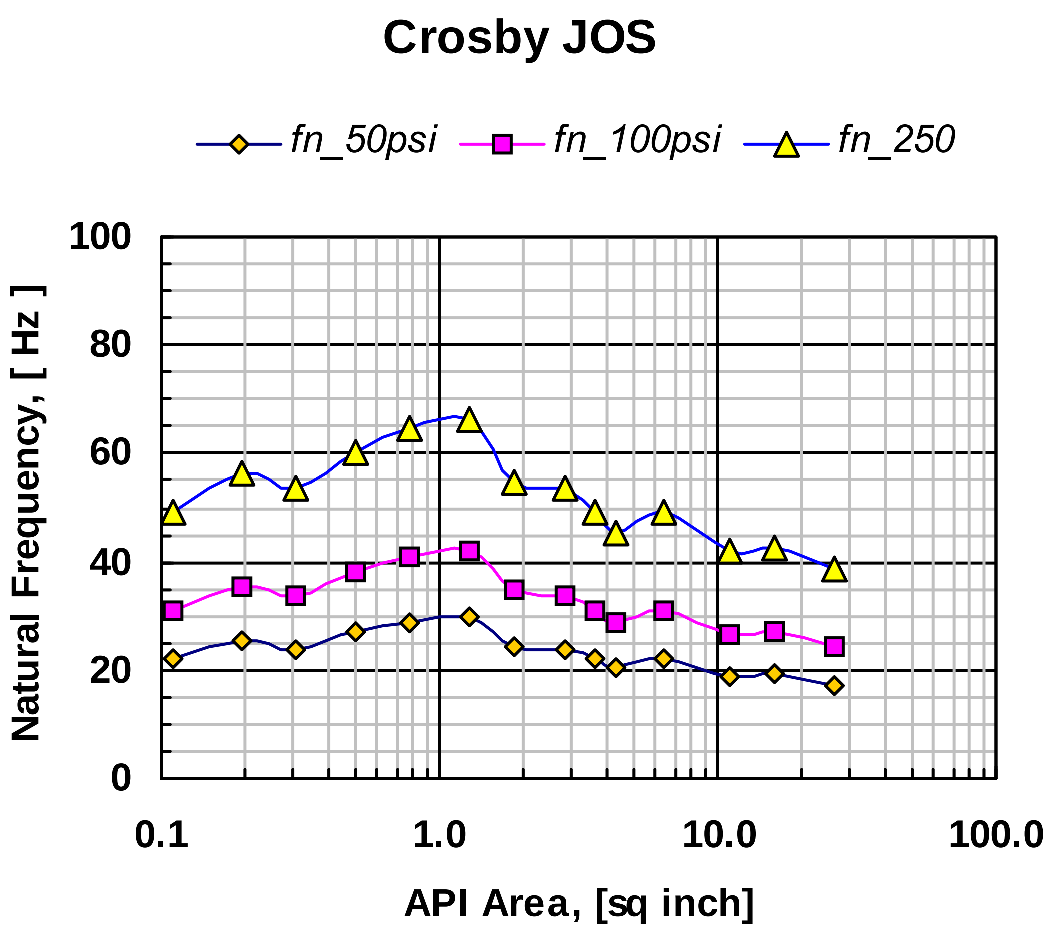


Relief Valve Spring Constant Estimates – Taken From Grolmes



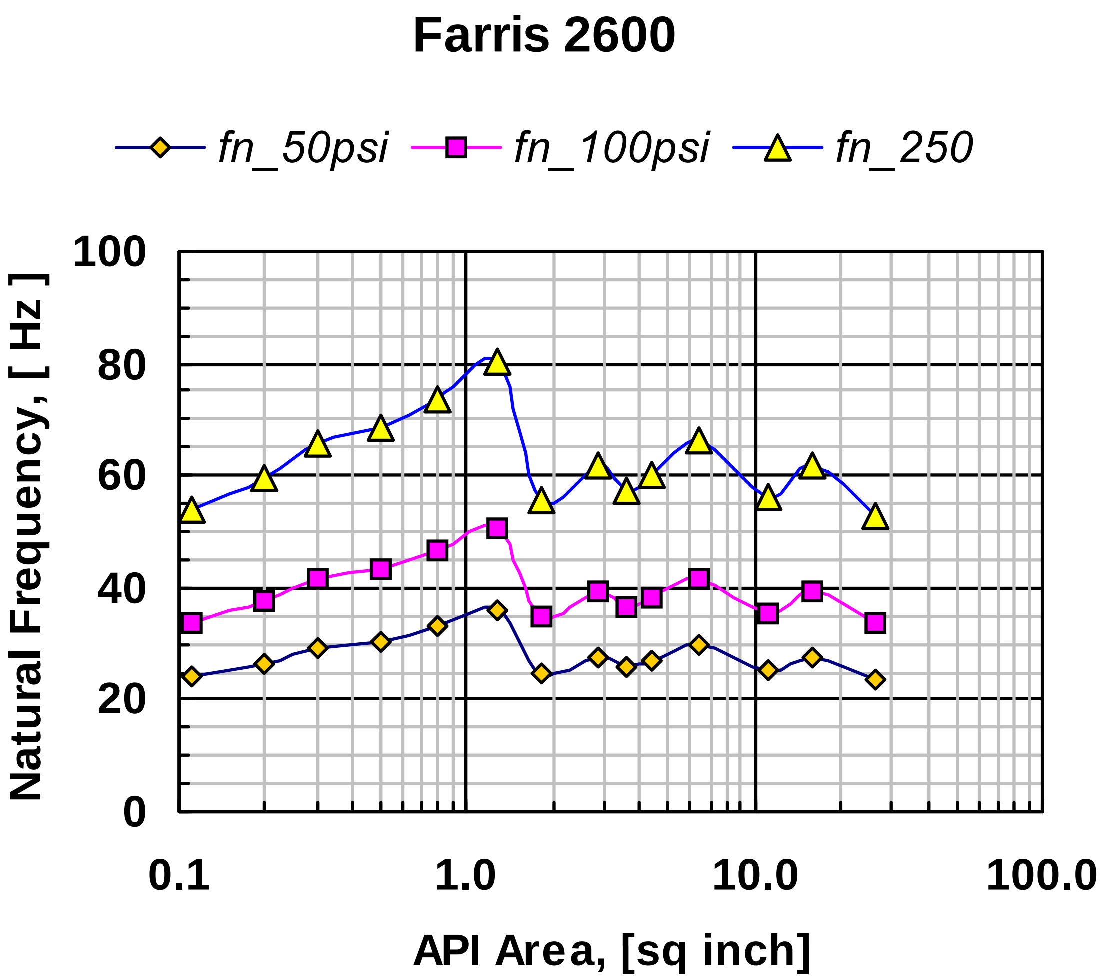
ioMosaic

Relief Valve Natural Frequency Estimates – Taken From Grolmes

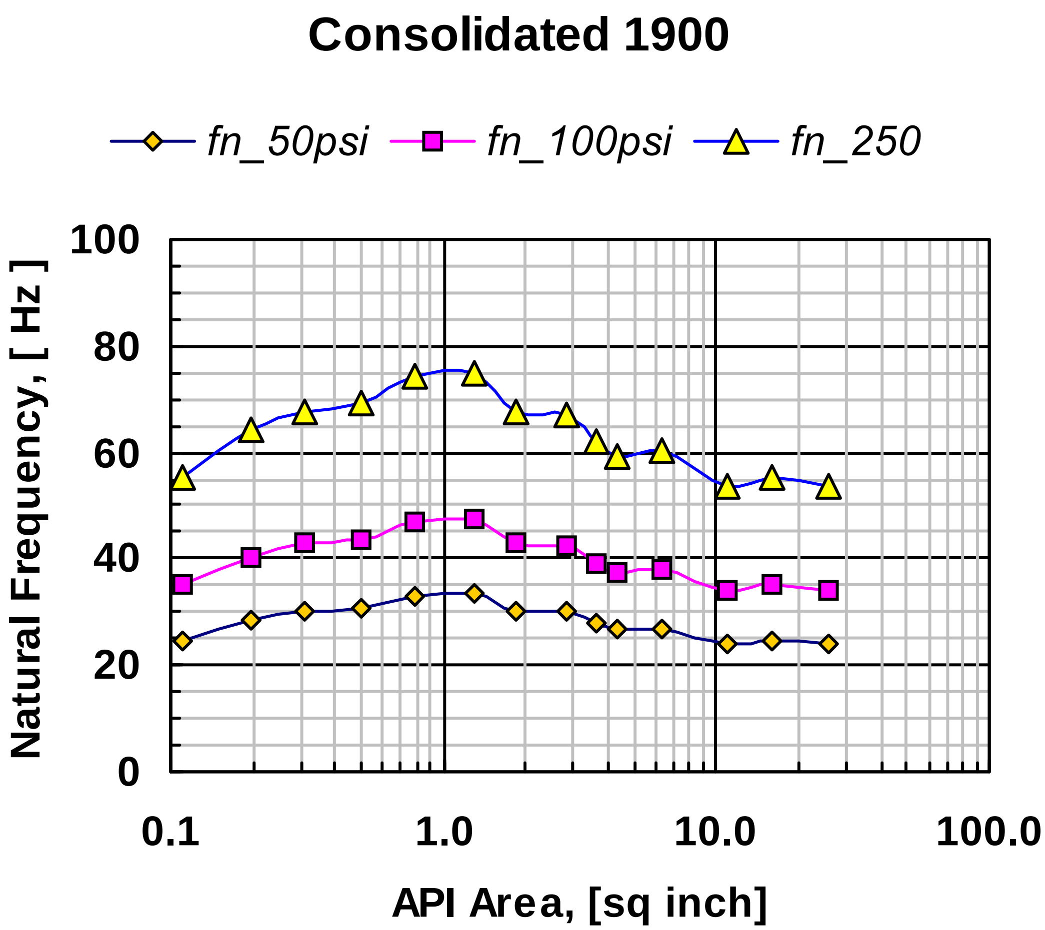


ioMosaic

Relief Valve Natural Frequency Estimates – Taken From Grolmes



Relief Valve Natural Frequency Estimates – Taken From Grolmes



ioMosaic

Typical causes of chatter

- PRV oversized for installation
  - Flow is < 25 % of rated capacity
  - Valve is handling widely different rates
- Inlet piping has excessive length. $P_{inlet} = P_{vessel} - \Delta P_{loss} <= P_{blowdown}$
- Inlet piping is undersized for PRV (Starving PRV)
- Outlet piping has excessive length
- Outlet piping is undersized for PRV
- Upper adjusting ring too high

ioMosaic

Chatter Solutions

- Avoid turns, elbows and sharp area reductions in inlet and outlet lines
- Use long radius elbows
- Use larger piping
- If you cannot change the piping
  - Increase blowdown (for example, 5 % inlet loss can be tolerated if blowdown is set at 7 %)
  - Install a smaller PRV or use a restricted lift valve
  - Install a different type of PRV (for example, a modulating pilot valve)
- Use multiple valves with staggered set pressures when the lowest required contingency rate is less than 25% of highest rate

ioMosaic

Quick Rules for Discharge Pipe

- Enlarged inlet pipe diameter is almost always required for:
  - 4P6, 6R8, 6R10, and 8T10
  - All safety valves used in series with rupture disks
  - 1.5H3, 2J3, 3L4, and 6Q8 with shutoff valve and L/D=5
- Enlarged outlet piping is almost always required for:
  - 6R8 safety valves
  - Conventional safety valves 3L4, 4P6, and 8T10 with a set pressure > 100 psig and discharge pipe length more than 10 ft

Where necessary, and with proper analysis, limit the irreversible inlet pressure loss to blowdown minus 2 %

- For spring loaded relief valves, the recommended irreversible inlet pressure drop maximum limit in ASME VIII (non-mandatory) and API-RP-520 for compressible fluid flow is 3% of the relief valve set pressure.

- For existing installations, verify relief valve history and inspection records to ensure chatter damage has never been present.

- Blowdown settings range from 7-14% of a relief valve set pressure. Often times the blowdown settings are unknown and difficult to specify / set properly.

- It is good practice to allow a 2% safety factor between the calculated irreversible inlet line loss and the blowdown setting (see Darby, Melhem, Fisher, and Grolmes); if the blowdown setting is 7%, limit the inlet pressure drop to 5% of the set pressure.

- There are many publications and flow test data dating to the early 1990s that clearly demonstrate valve stability at high inlet pressure loss and decreased blowdown.

- There are many practical installations where the 3 % rule cannot be met even with a very short inlet line.

ioMosaic

Where necessary, and with proper analysis, limit the irreversible inlet pressure loss to blowdown minus 2 % - Continued

- A safety margin of 2 %, is recommended in order to allow for uncertainties in flow calculations, physical properties, two-phase flow, setting the valve blowdown, and increased surface roughness of piping for older existing installations.
- Valve instability becomes a significant safety concern for large valves due to the amount of force that can be exhibited by valve components and associated piping when the valve closes rapidly.
- It is important to note that higher inlet pressure loss leads to lower flow rates and while this may be a solution to avoid valve stability issues, the reduced flow rate has to be sufficient to protect the vessel/equipment from overpressure.
- Increased blowdown also leads to longer discharge durations and more product loss and may not be desirable if the material discharged in highly toxic or a known carcinogen.
- Adjust relief device blowdown and set points with care using a qualified shop.

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

CONTROVERSIAL TOPICS / UNKNOWNS IN EMERGENCY RELIEF SIZING

Presentation to the DIERS Users Group Meeting

by

HAROLD G. FISHER

FisherInc

ORLANDO, FL

MARCH 23, 2009

SAFETY RELIEF VALVE Kd

SAFETY RELIEF VALVES ARE NOT PRESENTLY REQUIRED TO BE CERTIFIED FOR TWO-PHASE, VAPOR-LIQUID FLOW BY THE NATIONAL BOARD OF BOILER AND PRESSURE INSPECTORS. THE INTERNAL CONSTRUCTION DIFFERENCES BETWEEN THE VALVE MODELS OF VARIOUS MANUFACTURERS CAN RESULT IN A FLOW VARIATION OF A FACTOR OF TWO TO THREE. THE CALCULATION OF PIPE PRESSURE DROPS AND EFFLUENT FLOWS IS HIGHLY UNCERTAIN.

SAFETY RELIEF VALVE Kd

DARBY, R. D.

"On Two-phase frozen and Flashing Flows in Safety Relief Valves. Recommended Calculation Method and the Proper use of the Discharge Coefficient", JLPPI, 17(4), 255 - 260 (July 2004).

FAUSKE, H. K.

"A Practical Approach to Capacity Certification" Chemical Processing, 42 - 50 (February 2003).

LEUNG, J. C.

"A Theory on the Discharge Coefficient for Safety Relief Valves", JLPPI, 17 (5), 301 – 313 (2004).

SAFETY RELIEF VALVE Kb and Kw

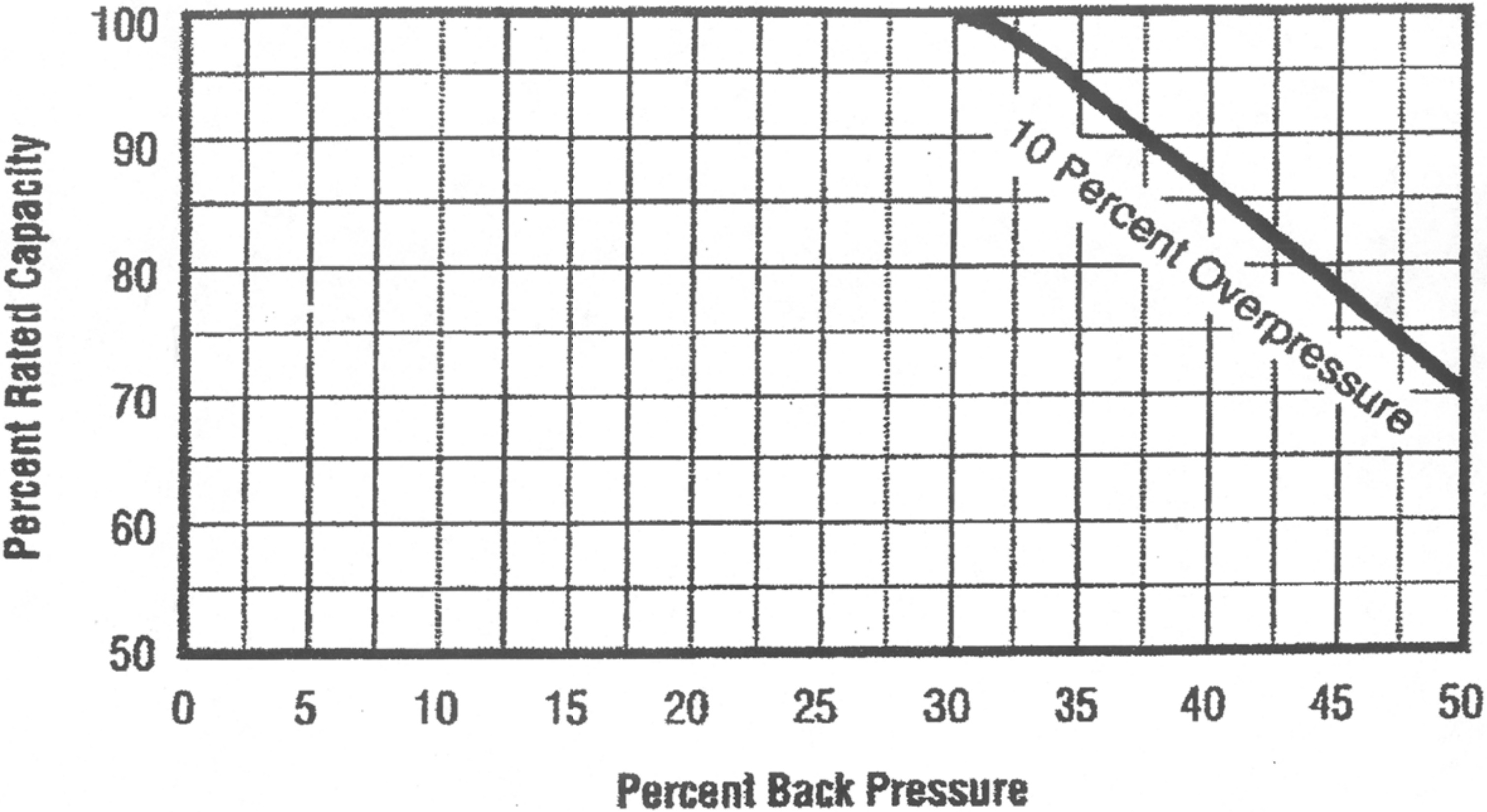
THE Kb AND Kw BACKPRESSURE FLOW REDUCTION CURVES PUBLISHED FOR USE WITH BELLOWS SAFETY RELIEF VALVES BY THE MANUFACTURERS AND THE AMERICAN PETROLEUM INSTITUTE (API) ARE NOT BACKED BY PUBLISHED DATA, ARE TECHNICALLY INCONSISTENT, APPLY TO VALVES NOT LONGER PERMITTED FOR USE BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS (ASME) CODE, AND DO NOT APPLY TO TWO-PHASE, VAPOR-LIQUID FLOW.

SAFETY RELIEF VALVE Kb AND Kw

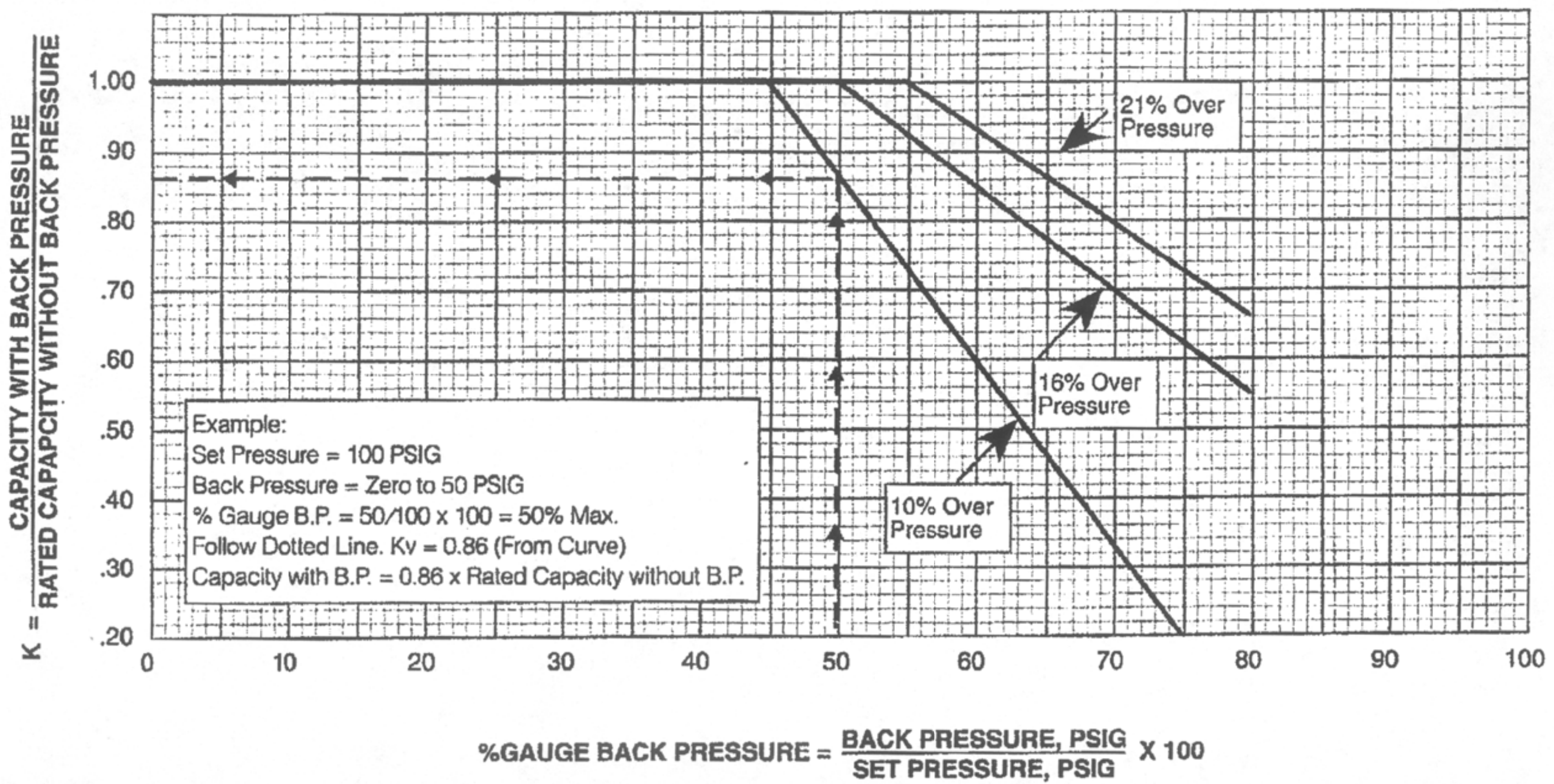
WHEN THE FLOWING AND CONSTANT SUPERIMPOSED BACKPRESSURES ON THE SAFETY RELIEF VALVES EXCEED CERTAIN VALUES, Kb AND Kw FACTORS ARE ALSO APPLIED TO REDUCE THE VAPOR OR LIQUID FLOWS, RESPECTIVELY, TO ACCOUNT FOR THE VALVES GOING OUT OF FULL LIFT. THE Kb AND Kw VALUES IN API RP 520 ARE CONSENSUS VALUES OF VARIOUS MANUFACTURERS. EACH MANUFACTURER ALSO PUBLISHES Kb AND Kw CURVES FOR THEIR OWN VALVES. A SINGLE CURVE IS USED TO REPRESENT THE PERFORMANCE OF ALL MODELS IN A MANUFACTURER'S LINE OF VALVES. THESE VALUES WERE SUPPOSEDLY MEASURED, BUT MOST OF THE DATA ARE NOT NOW AVAILABLE TO JUSTIFY THE PUBLISHED CURVES. THE PUBLISHED VALUES OF THE VARIOUS MANUFACTURES DO NOT AGREE.

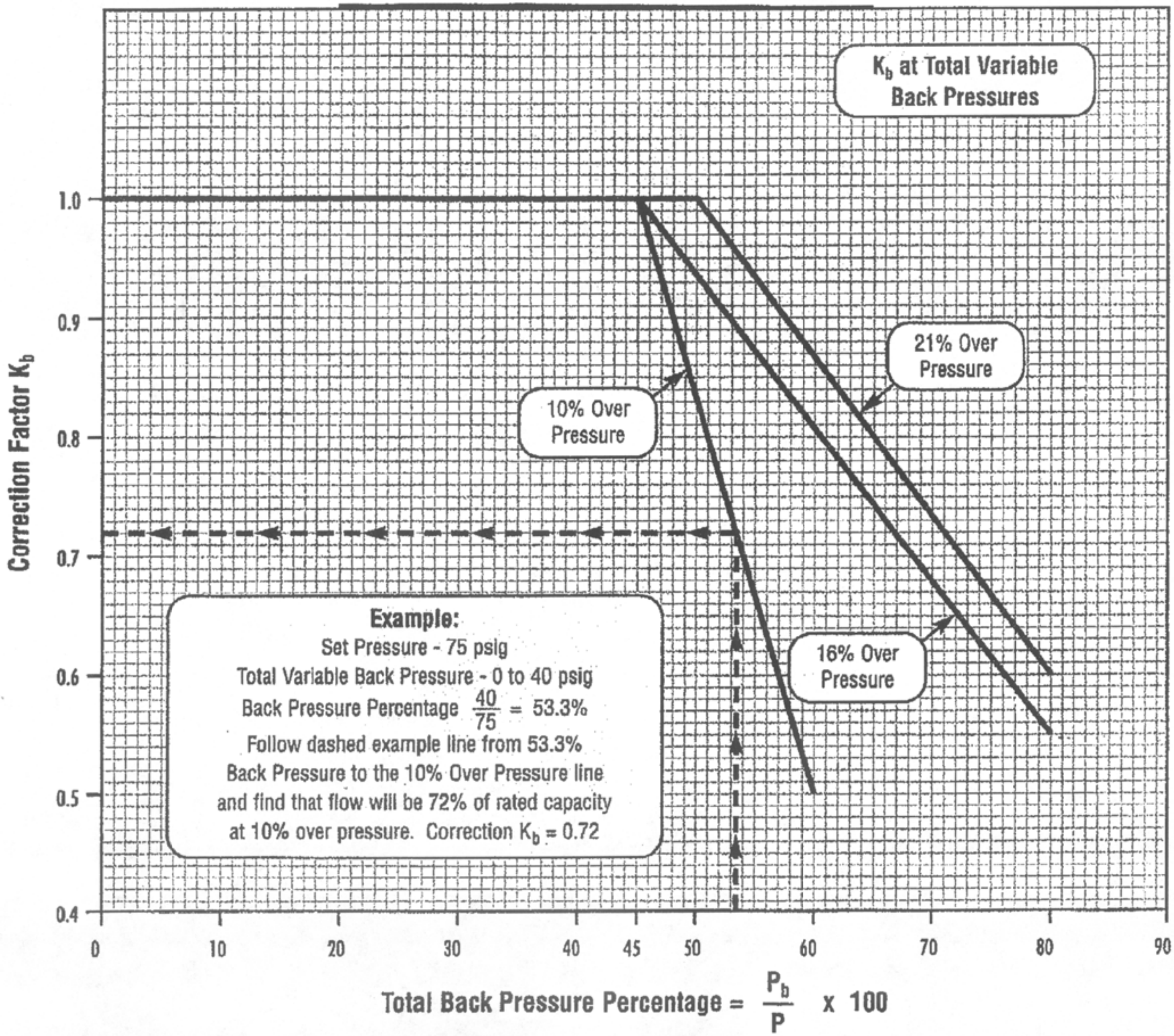
SAFETY RELIEF VALVE Kb AND Kw

SAFETY RELIEF VALUES WITH MODIFIED LIQUID TRIM ARE NOW REQUIRED BY THE ASME CODE TO HAVE THEIR RATED CAPACITY FLOW CERTIFIED AT 1.1 PSET. THE MANUFACTURERS, HOWEVER, HAVE NEVER CHANGED THE Kw CURVES PUBLISHED FOR THEIR OLDER MODEL LIQUID VALVES WITH THE RATED CAPACITY CERTIFIED AT 1.25 PSET.



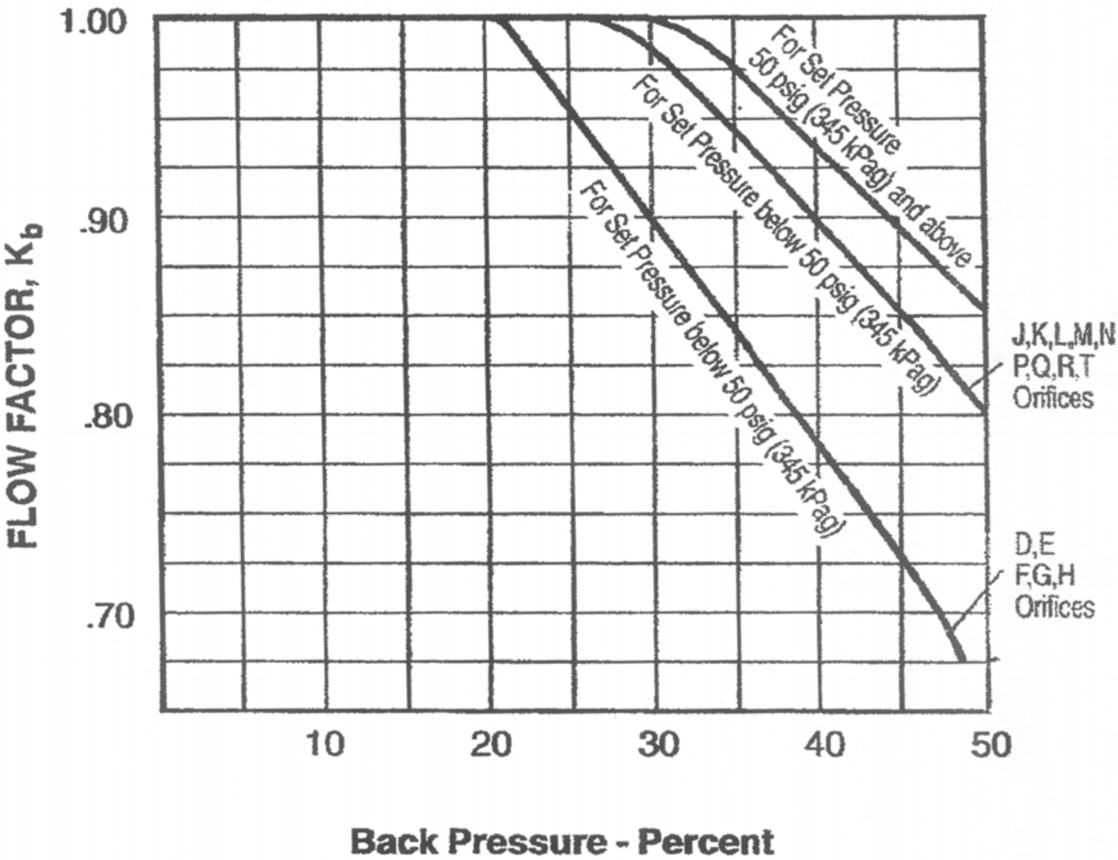
Effects of Back Pressure on Relief Capacity of Typical Balanced Direct Spring PRV on Gas

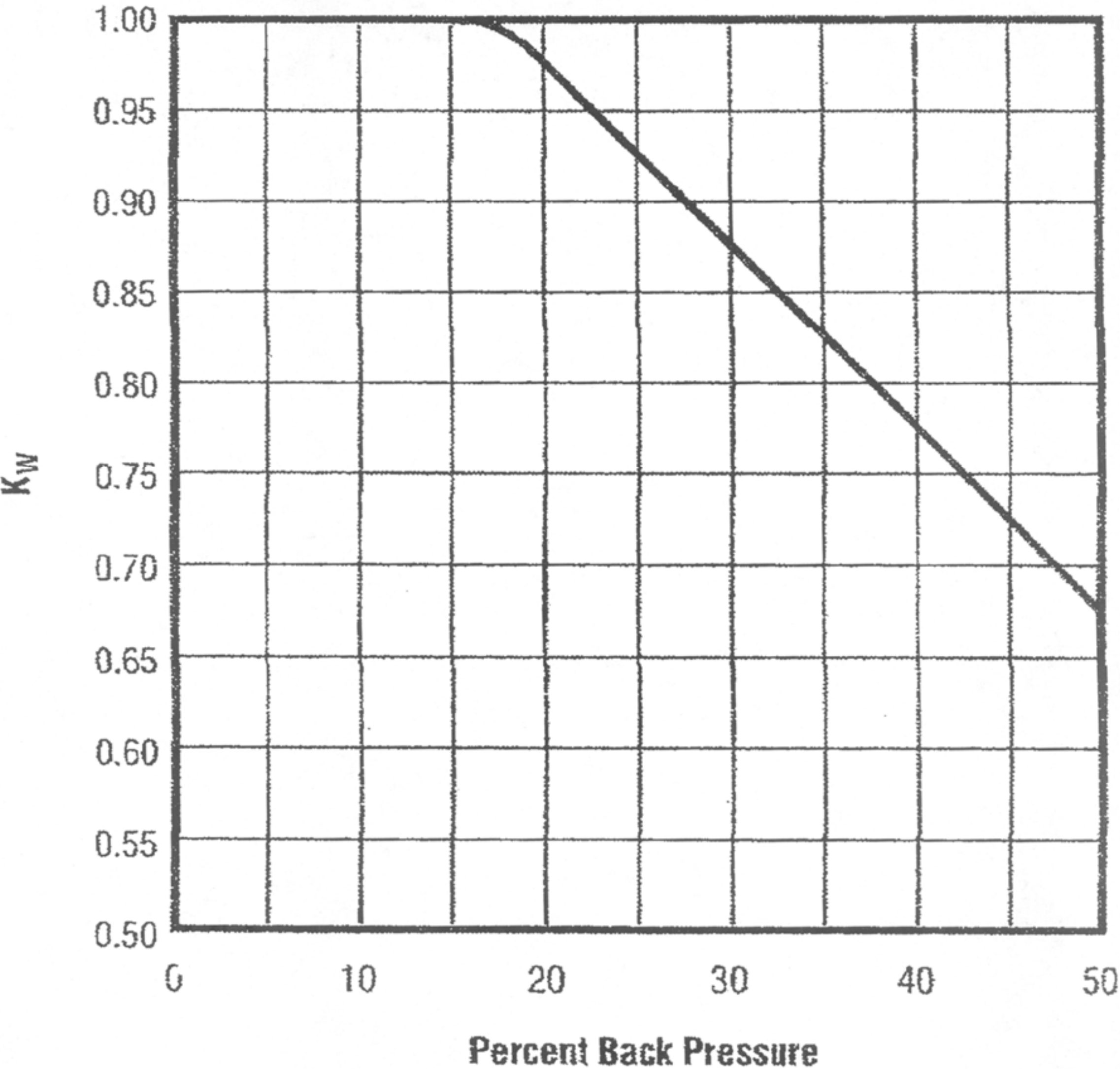

K = CAPACITY WITH BACK PRESSURE / RATED CAPAPCITY WITHOUT BACK PRESSURE
1.00
.90
.80
.70
.60
.50
.40
.30
.20
0
10
20
30
40
50
60
70
80
90
100
21% Over Pressure
16% Over Pressure
10% Over Pressure
Example:
Set Pressure = 100 PSIG
Back Pressure = Zero to 50 PSIG
% Gauge B.P. = 50/100 x 100 = 50% Max.
Follow Dotted Line. Kv = 0.86 (From Curve)
Capacity with B.P. = 0.86 x Rated Capacity without B.P.
%GAUGE BACK PRESSURE = BACK PRESSURE, PSIG / SET PRESSURE, PSIG X 100



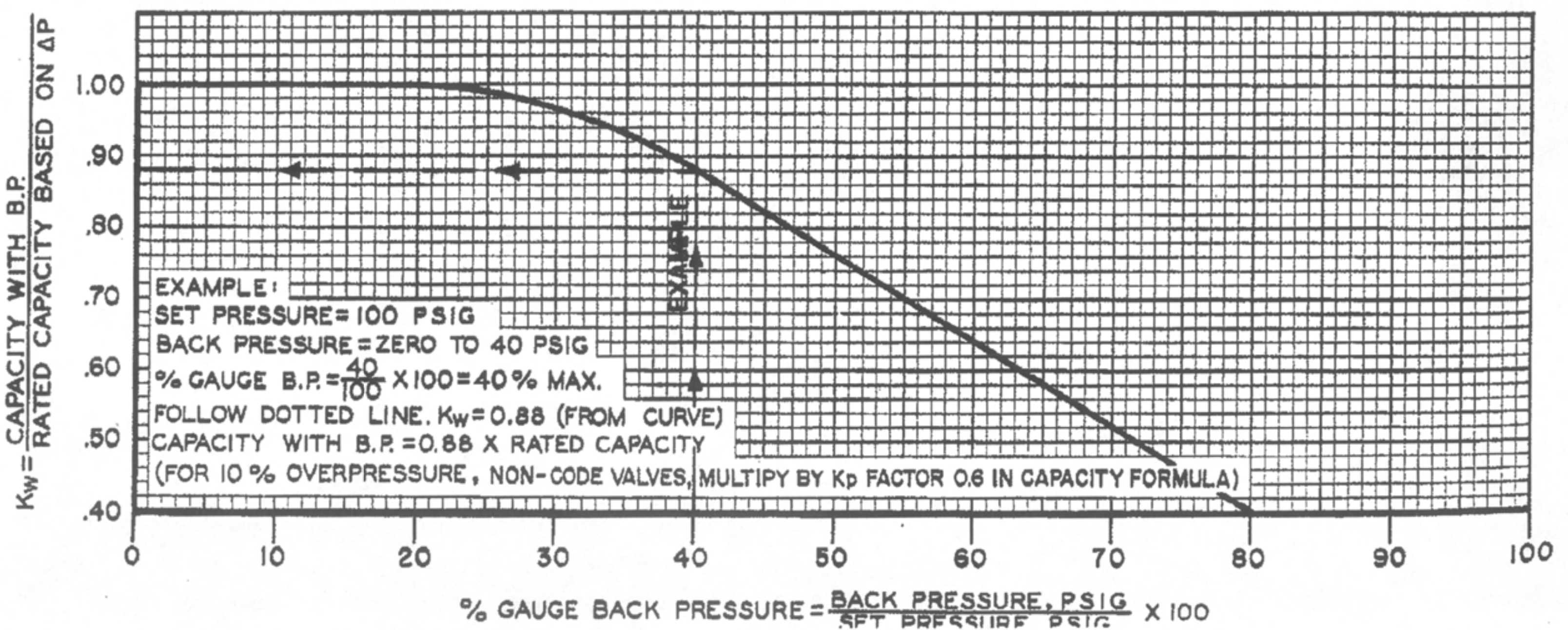
K_b at Total Variable Back Pressures
Correction Factor K_b
1.0
0.9
0.8
0.7
0.6
0.5
0.4
21% Over Pressure
10% Over Pressure
16% Over Pressure
Example:
Set Pressure - 75 psig
Total Variable Back Pressure - 0 to 40 psig
Back Pressure Percentage $\frac{40}{75}$ = 53.3%
Follow dashed example line from 53.3% Back Pressure to the 10% Over Pressure line and find that flow will be 72% of rated capacity at 10% over pressure. Correction K_b = 0.72
0
10
20
30
40
45
50
60
70
80
90
Total Back Pressure Percentage = $\frac{P_b}{P}$ x 100

Correction Factor for Vapors and Gases, K_b for Balanced Bellows Valves - 10% Overpressure

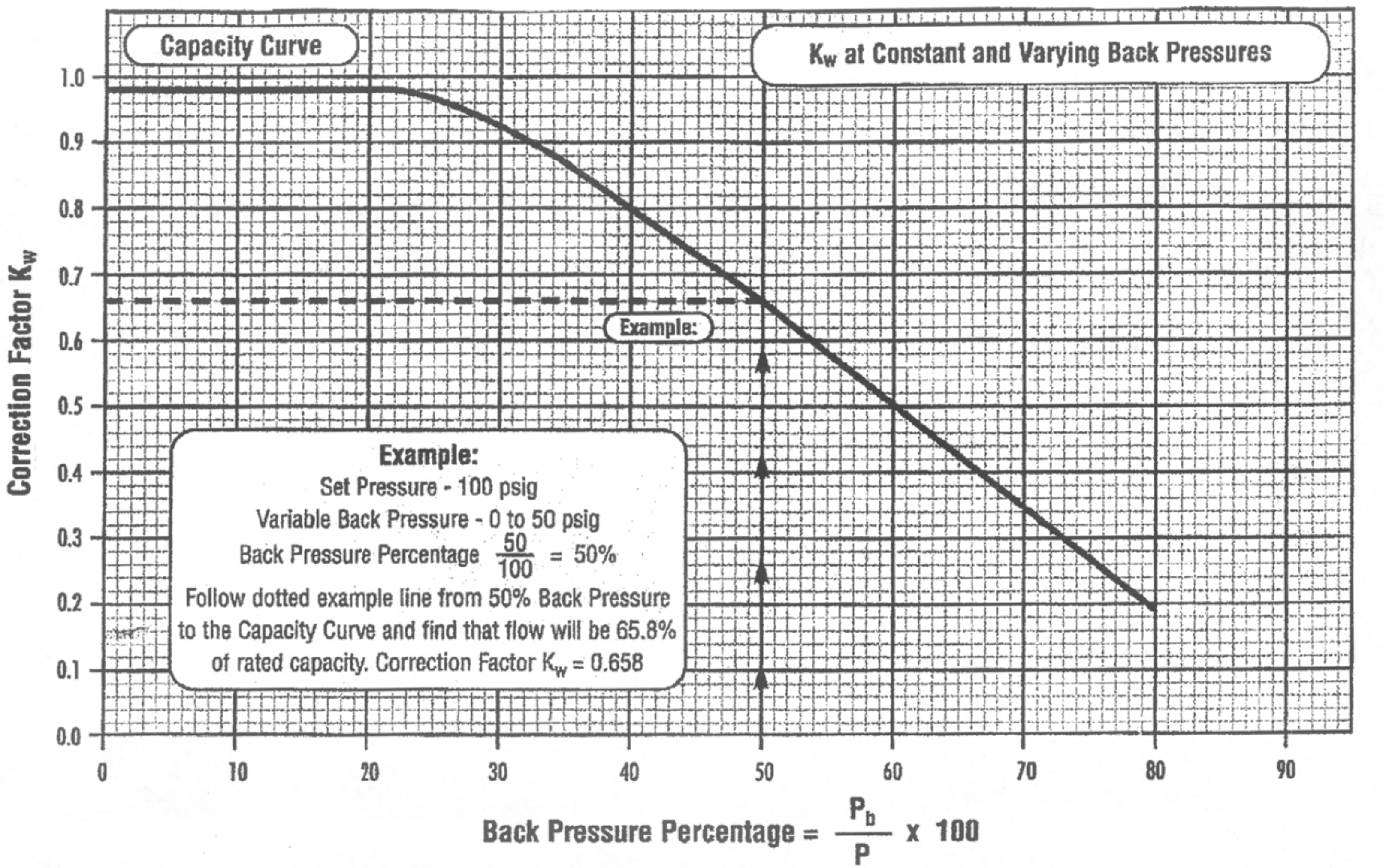




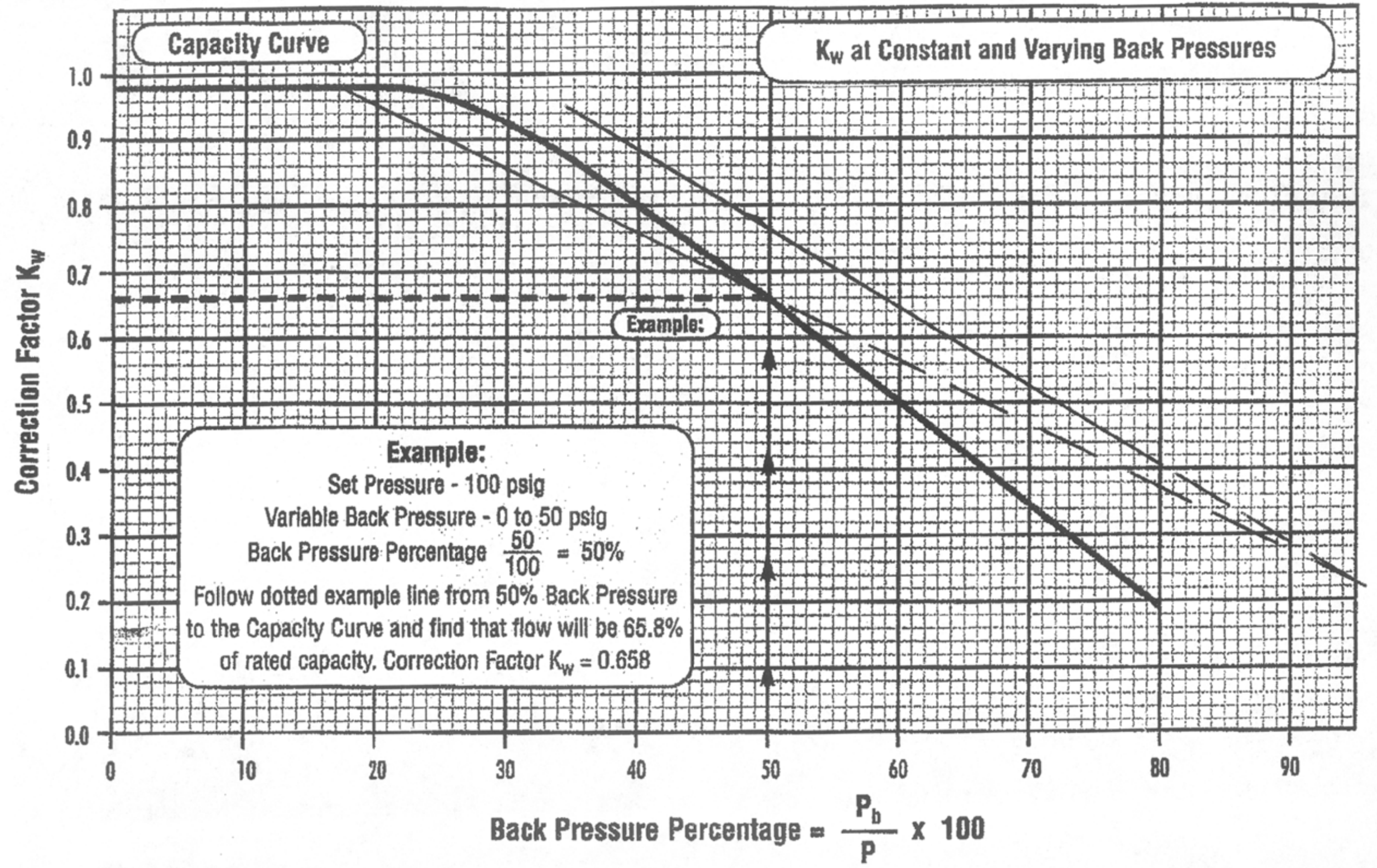
K_W for Balanced Bellows Spring Valves on Liquid



K_W = CAPACITY WITH B.P. / RATED CAPACITY BASED ON ΔP
1.00
.90
.80
.70
.60
.50
.40
EXAMPLE
EXAMPLE:
SET PRESSURE = 100 PSIG
BACK PRESSURE = ZERO TO 40 PSIG
% GAUGE B.P. = 40/100 X 100 = 40% MAX.
FOLLOW DOTTED LINE. K_W = 0.88 (FROM CURVE)
CAPACITY WITH B.P. = 0.88 X RATED CAPACITY
(FOR 10% OVERPRESSURE, NON-CODE VALVES, MULTIPY BY Kp FACTOR 0.6 IN CAPACITY FORMULA)
0
10
20
30
40
50
60
70
80
90
100
% GAUGE BACK PRESSURE = BACK PRESSURE, PSIG / SET PRESSURE, PSIG X 100



Capacity Curve
K_W at Constant and Varying Back Pressures
Correction Factor K_W
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Example:
Example:
Set Pressure - 100 psig
Variable Back Pressure - 0 to 50 psig
Back Pressure Percentage $\frac{50}{100}$ = 50%
Follow dotted example line from 50% Back Pressure
to the Capacity Curve and find that flow will be 65.8%
of rated capacity. Correction Factor K_W = 0.658
0
10
20
30
40
50
60
70
80
90
Back Pressure Percentage = $\frac{P_b}{P}$ x 100



Capacity Curve
K_W at Constant and Varying Back Pressures
Correction Factor K_W
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Example:
Example:
Set Pressure - 100 psig
Variable Back Pressure - 0 to 50 psig
Back Pressure Percentage $\frac{50}{100}$ = 50%
Follow dotted example line from 50% Back Pressure
to the Capacity Curve and find that flow will be 65.8%
of rated capacity. Correction Factor K_W = 0.658
0
10
20
30
40
50
60
70
80
90
Back Pressure Percentage = $\frac{P_b}{P}$ x 100

ioMosaic

Business Confidential Document

Best Practices for Flare Systems Evaluations

Paper Presented at the Spring 2009 DIERS Users Group Meeting, Orlando, Florida

By

G. A. Melhem, Ph.D.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

This presentation will cover three major topics

- Flare hydraulics
- Flare hydraulics checks
- Flare systems risk

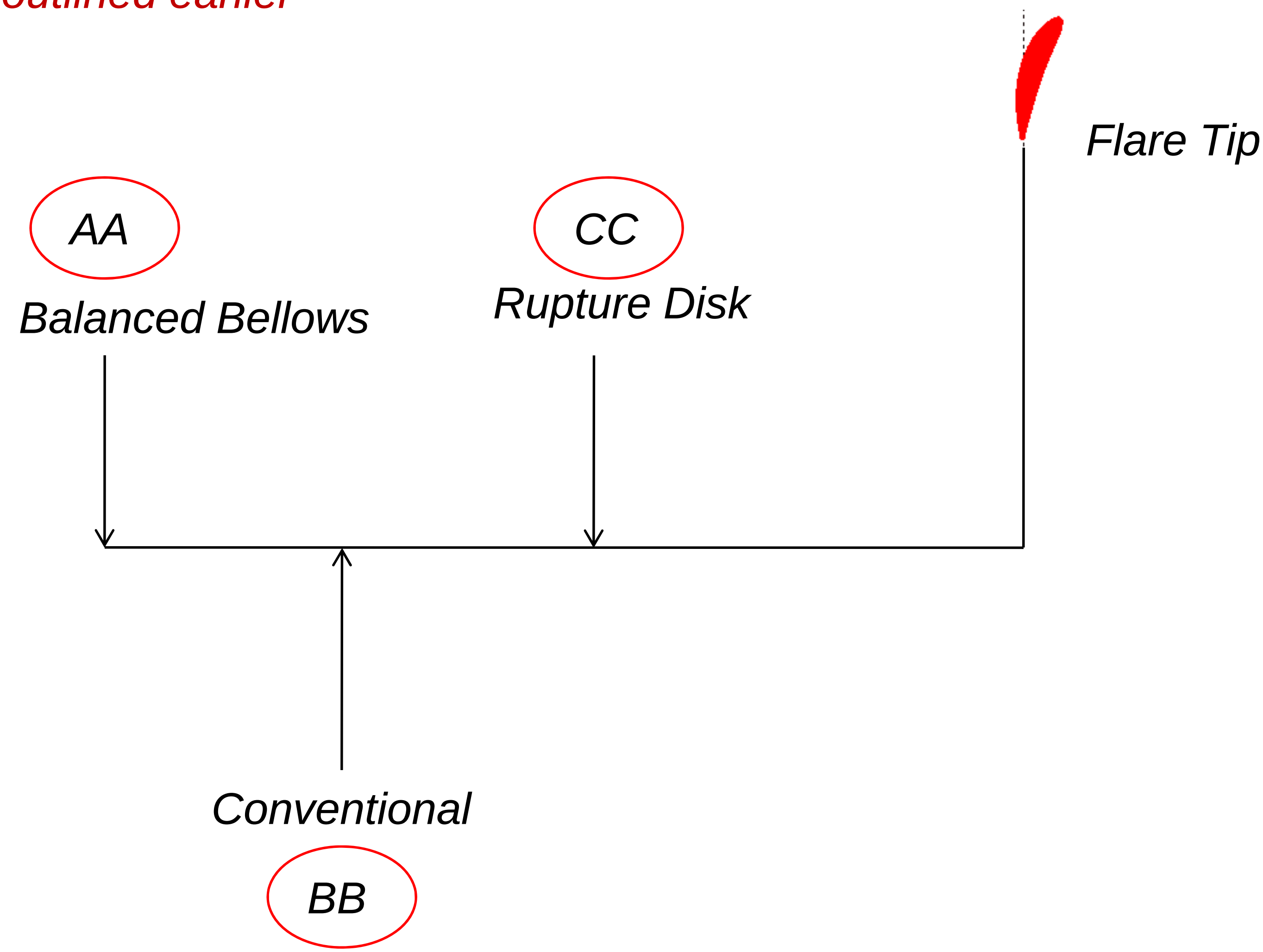
ioMosaic

We solve flare hydraulics problems by first decomposing each flare node in order to guarantee convergence

- A backpressure curve is first calculated for each individual relief device including the inlet line, relief device, and discharge line
- A backpressure curve is also calculated for each sub-header connecting into the main flare header
- The main flare header isometric is defined
  - Nodes are identified for specific piping segments
  - Nodes can be turned on or off from the main flare header model
- The final pressure profile in the main flare header is estimated starting from the flare tip
  - Pressures are then propagated into each connecting node
- Summary tables are then produced to determine the impact of flare header pressure on flow rate for each node

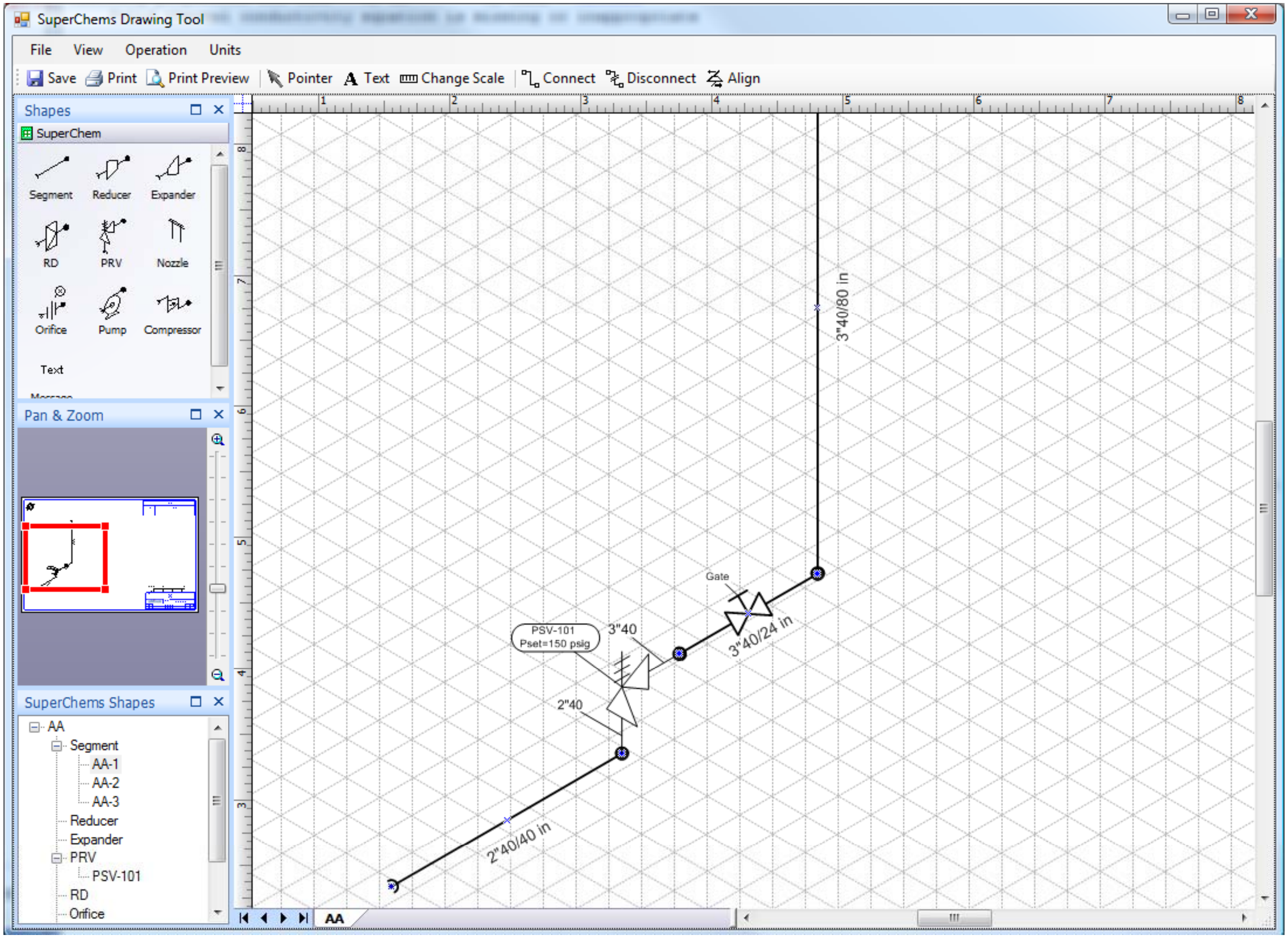
ioMosaic

We consider the following simple example to illustrate the flare hydraulics concepts outlined earlier



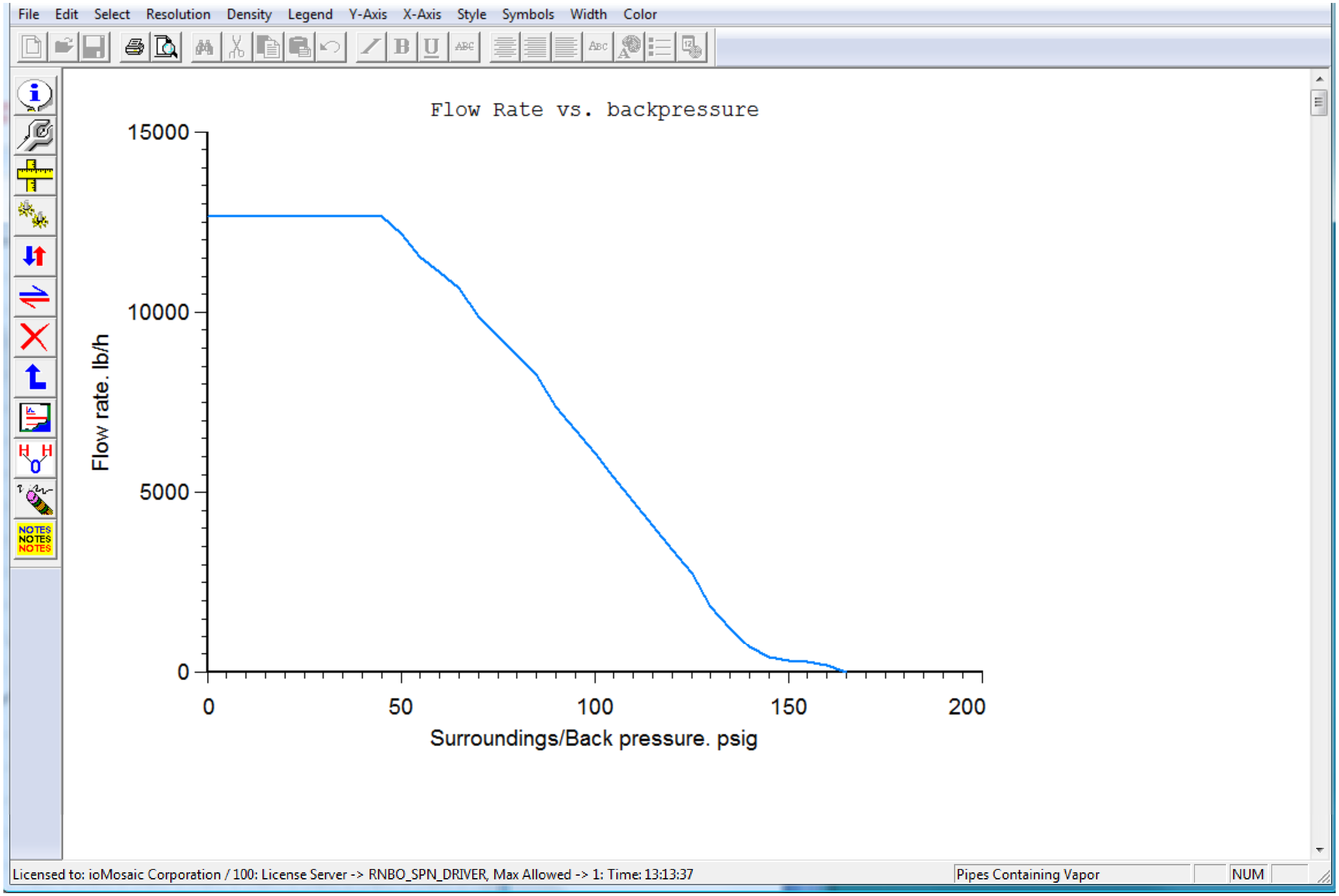
ioMosaic

The AA relief line consisting of a balanced bellows PRV is illustrated below



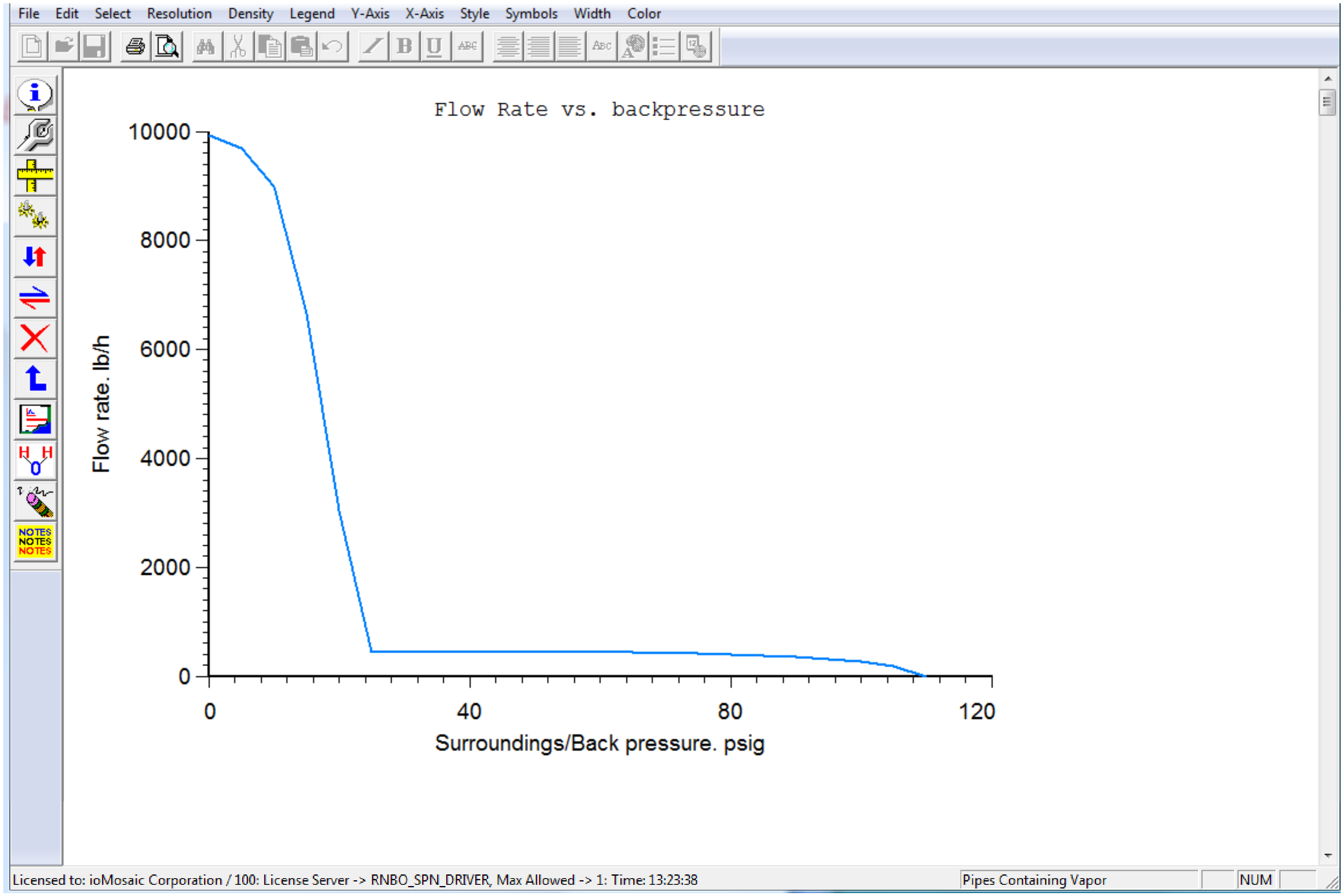
ioMosaic

A backpressure curve is calculated for AA

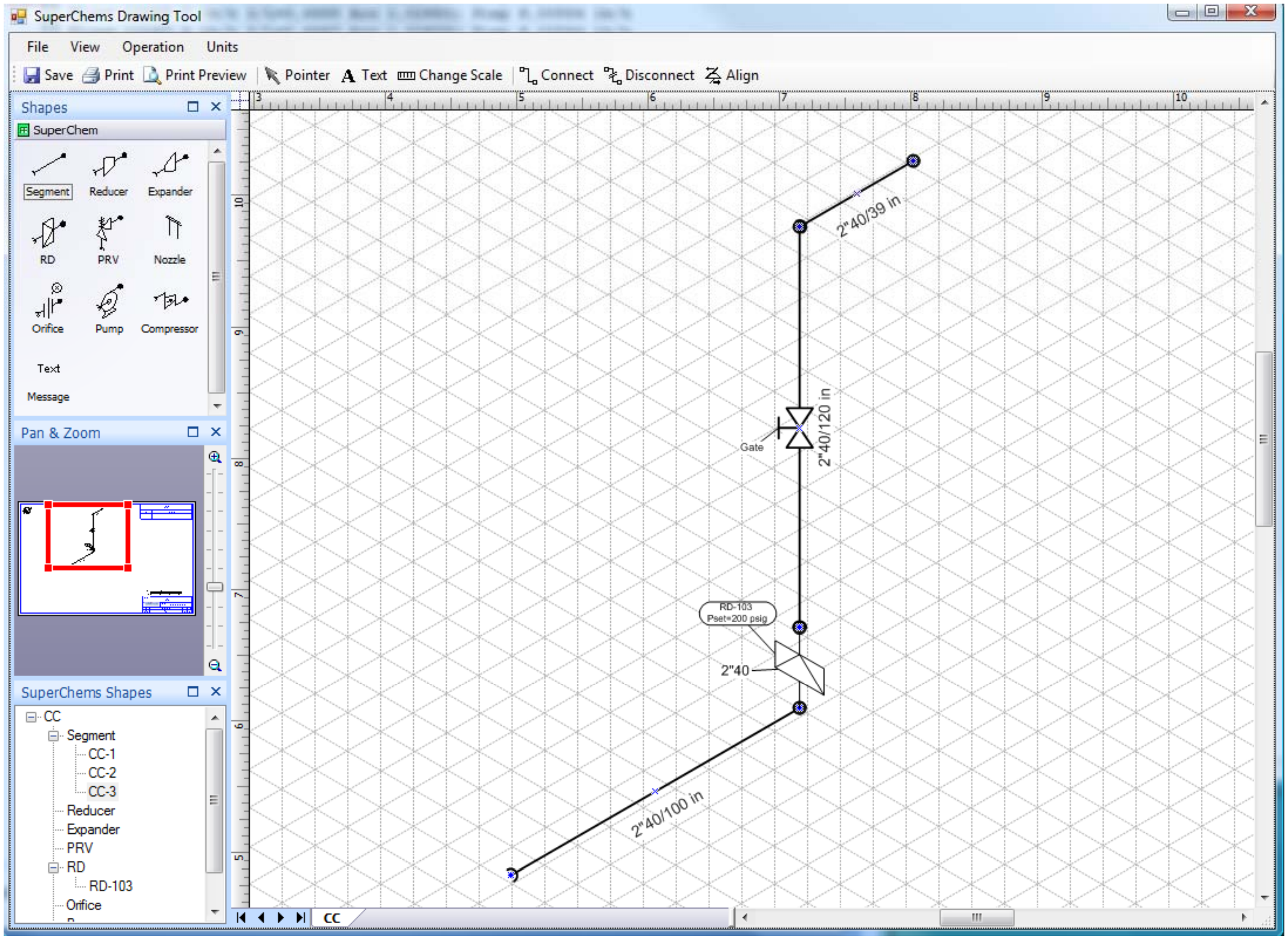


ioMosaic

A backpressure curve is calculated for BB, PRV is set at 100 psig

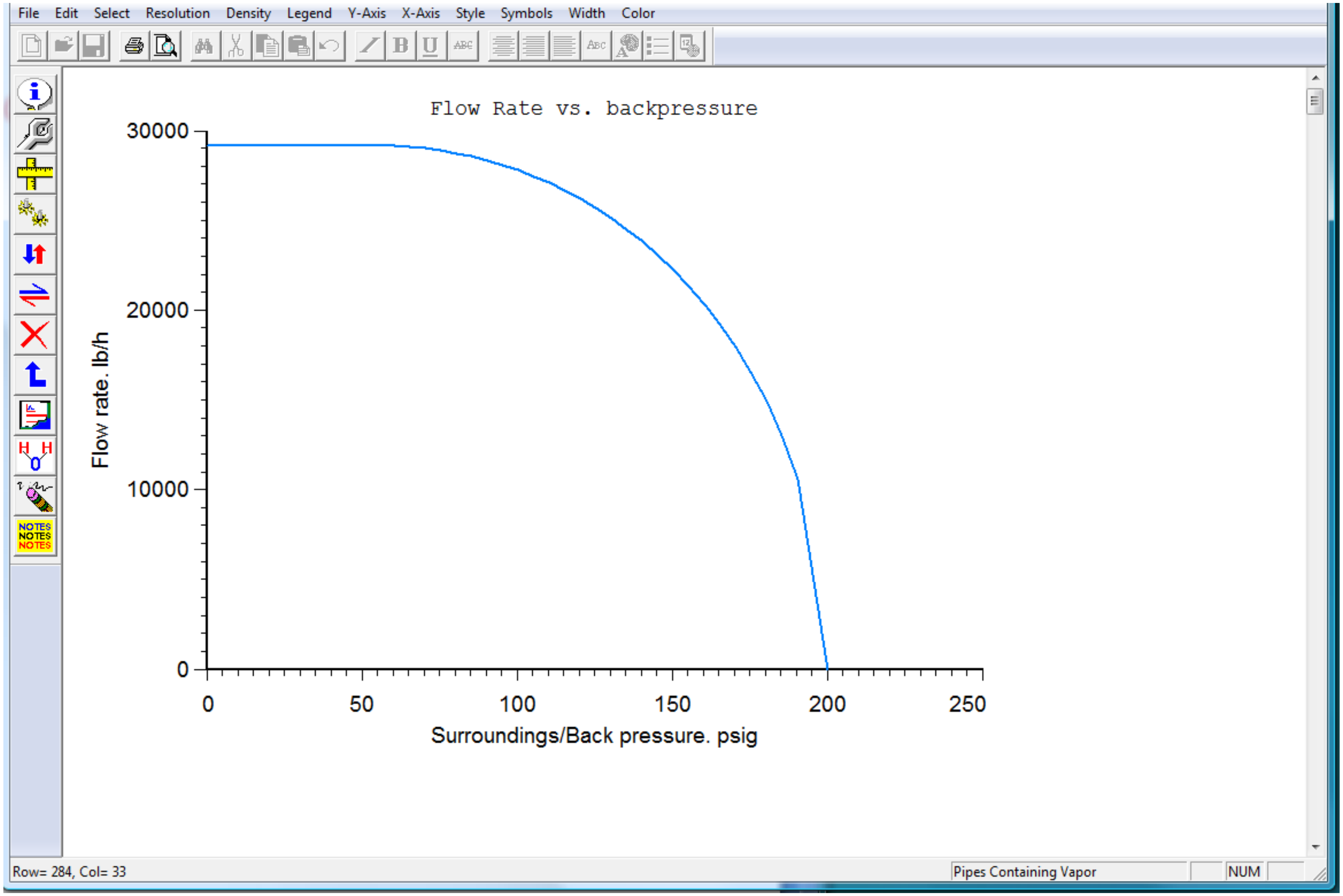


The CC relief line consisting of a rupture disk is illustrated below



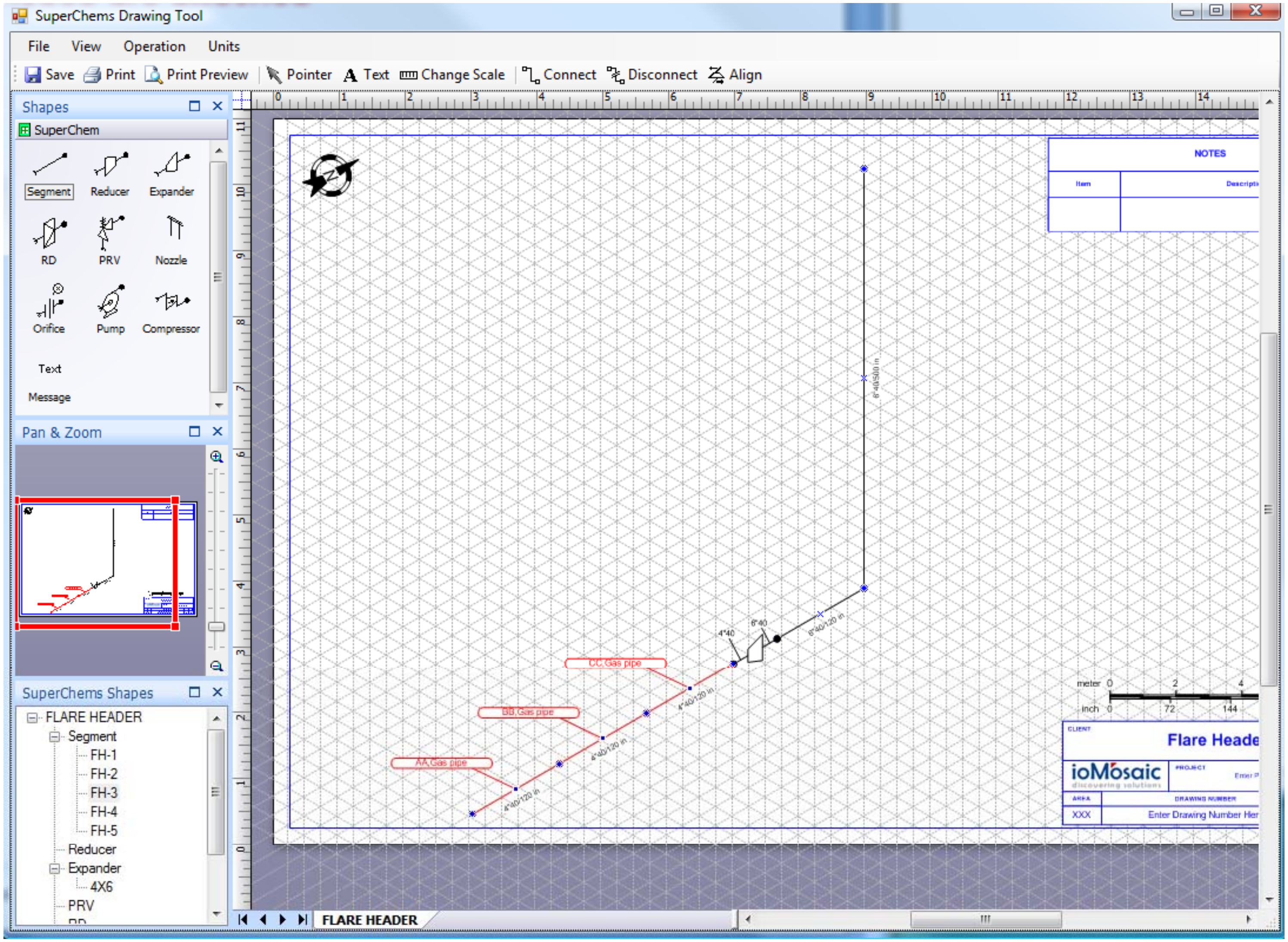
ioMosaic

A backpressure curve is calculated for CC



ioMosaic

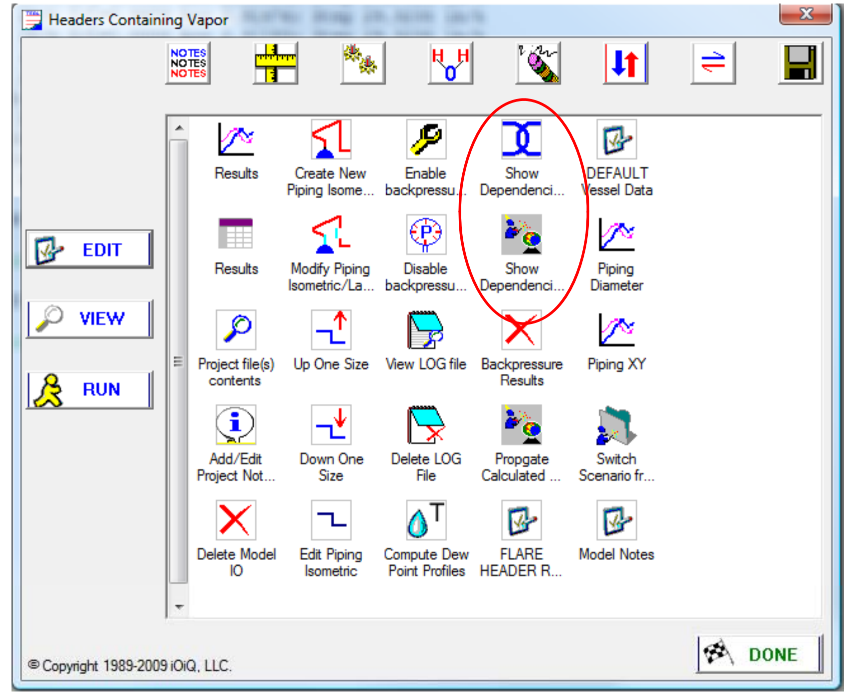
Create the flare header and attach the nodes



ioMosaic

Execute the flare hydraulics using the batch queue with dependencies

- Check summary report
- Check detailed flare hydraulics report
- Check the flare pressure, temperature, and Mach number profiles
- Check reaction forces and vibration risk
- Check flare thermal radiation and noise
- Check dispersion estimates in case of flame out
- Check backpressure on flowing and non-flowing devices



ioMosaic

Check the hydraulics summary report

Header and Subheader Dependencies Tables

	A	B	C	D	I	J	K	L	M	N	O	P	Q	R	S	
1	Tree Level	Piping Isometric	Scenario Name	Flow Type	Max Flow. lb/h	Act Flow. lb/h	Act/Max	Inlet Pres. psig	Backpres. psig	Inlet Temp. C	% Vapor	Mach Number	NRe	15,400*rhoj/rhoa	rhoj*uj^2 . Pa	Device
2	L0	FLARE	FLARE	Gas Header	42512.78	42512.78	1.00	36.86	5.00	71.27	100.00	0.690	3,399,778	14,655	73,211	
3																
4	L1	FLARE														
5	L2	AA	AA	Gas Pipe	12962.19	12957.73	1.00	165.00	36.86	100.00	100.00	0.357	2,135,004	26,308	57,890	PSV-1
6	L2	BB	BB	Gas Pipe	9930.37	432.60	0.04	110.00	36.10	200.00	100.00	0.011	65,775	37,537	45	PSV-1
7	L2	CC	CC	Gas Pipe	29129.99	29129.99	1.00	200.00	35.28	250.00	100.00	1.000	6,783,561	72,819	512,579	RD-10
8																
9																
10																
11																
12																
13																
14																
15																
16																
17																
18																
19																
20																
21																
22																
23																
24																
25																
26																
27																
28																
29																
30																
31																
32																
33																
34																
35																
36																
37																

Format | Print | Save As | Cut | Copy | Paste | Cancel | OK

ioMosaic

Check the flare hydraulics detailed report

REPORT WRITER

Table of Contents | Vapor Flow From Headers and Subheaders | Vapor Flow From Pipes | Ideal Nozzle Flow

Row	B	C	D
21	3.3998E+06		
22			
23	42512.783		
24	539892.867		
25			
26	MW	FLOW IN. lb/h	FLOW OUT. lb/h
27	16.043	12956.231	12956.231
28	30.070	432.430	432.430
29	44.097	20386.885	20386.885
30	58.123	8737.237	8737.237
31			
32			

Row	Flow In. lb/h	Flow Out. lb/h	Exit Diam. in	Exit/Flow Area. in2	Inlet Temp. C	Inlet Pres. psig	Irrev. dP. psia	Total dP. psia	Length. in
34	12956.23	12956.23	4.0260	12.7303	71.2676	36.8619	0.7161	0.7584	120.0000
35	13388.66	13388.66	4.0260	12.7303	73.8764	36.1035	0.7704	0.8203	120.0000
36	42512.78	42512.78	4.0260	12.7303	180.3699	35.2832	6.3573	17.2772	120.0000
37	42512.78	42512.78	6.0650	28.8903	164.9266	15.0803	0.0536	0.0700	6.0000
38	42512.78	42512.78	6.0650	28.8903	164.8970	15.0103	1.0990	1.4585	120.0000
39	42512.78	42512.78	6.0650	28.8903	164.2321	13.5518	5.4249	8.5531	500.0000

40

41 ılation is enabled Last Specified: 01:45:30 PM, Sat Jan 10 2009

42 Last Executed: 03:01:47 PM, Sat Jan 10 2009

43

Row	Scenario Reference	Piping Layout	Source	Source Temp. C	Source Pres. psig	Backpressure. psig	Max. Node Flow. lb/h	Act. Node Flow. lb/h	% of Max. Flow
45	AA	AA	Gas pipe	100.0000	165.0010	36.8619	12962.20	12956.20	99.9540
46	BB	BB	Gas pipe	200.0000	110.0010	36.1035	9930.37	432.4300	4.3546
47	CC	CC	Gas pipe	250.0000	200.0000	35.2832	29130.00	29124.10	99.9799
48						TOTAL FLOW >>>	52022.50	42512.80	81.7199

49

50

51

Format | Print | Save As | Cut | Copy | Paste | Cancel | OK

ioMosaic

Check the hydraulics detailed report for each input

REPORT WRITER

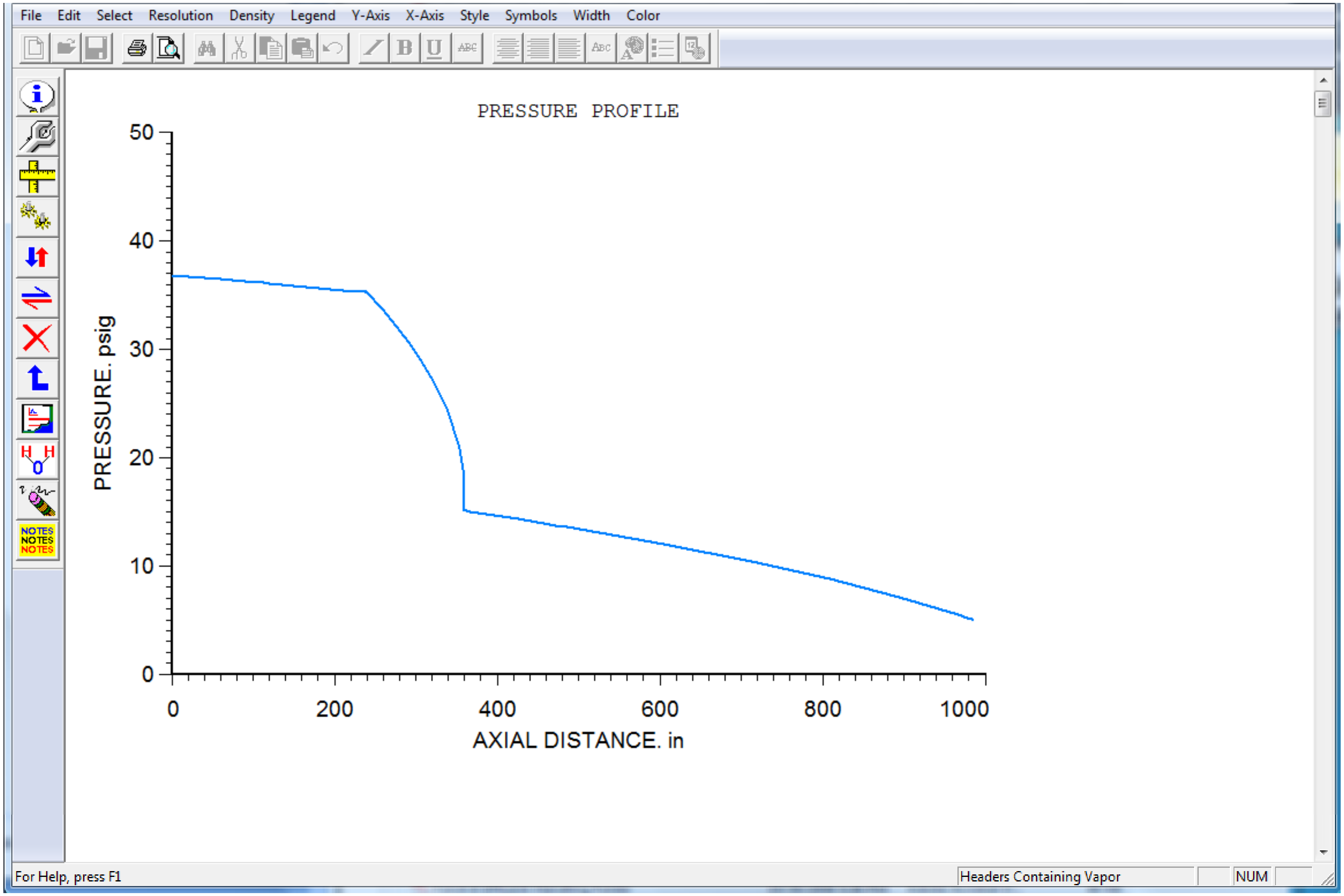
Table of Contents | Vapor Flow From Headers and Subheaders | Vapor Flow From Pipes | Ideal Nozzle Flow

	A	B	C	D	E	F
11	Pressure. psig	165.001				
12						
13	** EXIT CONDITIONS					
14	Temperature. C	67.555				
15	Pressure. psig	36.915				
16	Surroundings/back pressure. psig	36.862				
17	Barometric pressure at site elevation. psig	-3.974E-06	DEFAULT site			
18						
19	Speed of sound. m/s	473.716				
20	Exit Mach Number	0.357				
21	Maximum Mach Number	1.0	** WARNING: Mach Number is > 0.7			
22	Exit rho*u*u in kg/m/s2 or in Pascals	57890.170				
23	Maximum rho*u*u in kg/m/s2 or in Pascals	90825.880				
24	Exit Reynold's Number	2.135E+06				
25						
26	Exit steady state impulse. N	1973.283				
27	Maximum steady state impulse difference. N	1508.794				
28						
29	Mass flow rate. lb/h	12962.188				
30	Volumetric flow rate. SCFH	305815.598				
31						
32	COMPONENT	MW	FLOW IN. lb/h	FLOW OUT. lb/h		
33	METHANE	16.043	12960.689	12960.689		
34						
35	Relief device found at segment 2	PSV-101: V-101. Bellows. Service= Gas and Liquid				
36	Last iteration Kb correction	1.0				
37	Pressure at device inlet. psig	158.927				
38	% Inlet pressure drop relative to actual set point	4.050				
39	% Irrev. inlet pressure drop relative to actual set point	3.757	** WARNING: Inlet pressure drop exceeds 3 %			
40	% back pressure relative to actual set point	28.313				
41						
42						

Format | Print | Save As | Cut | Copy | Paste | Cancel | OK

ioMosaic

Check the flare pressure profile

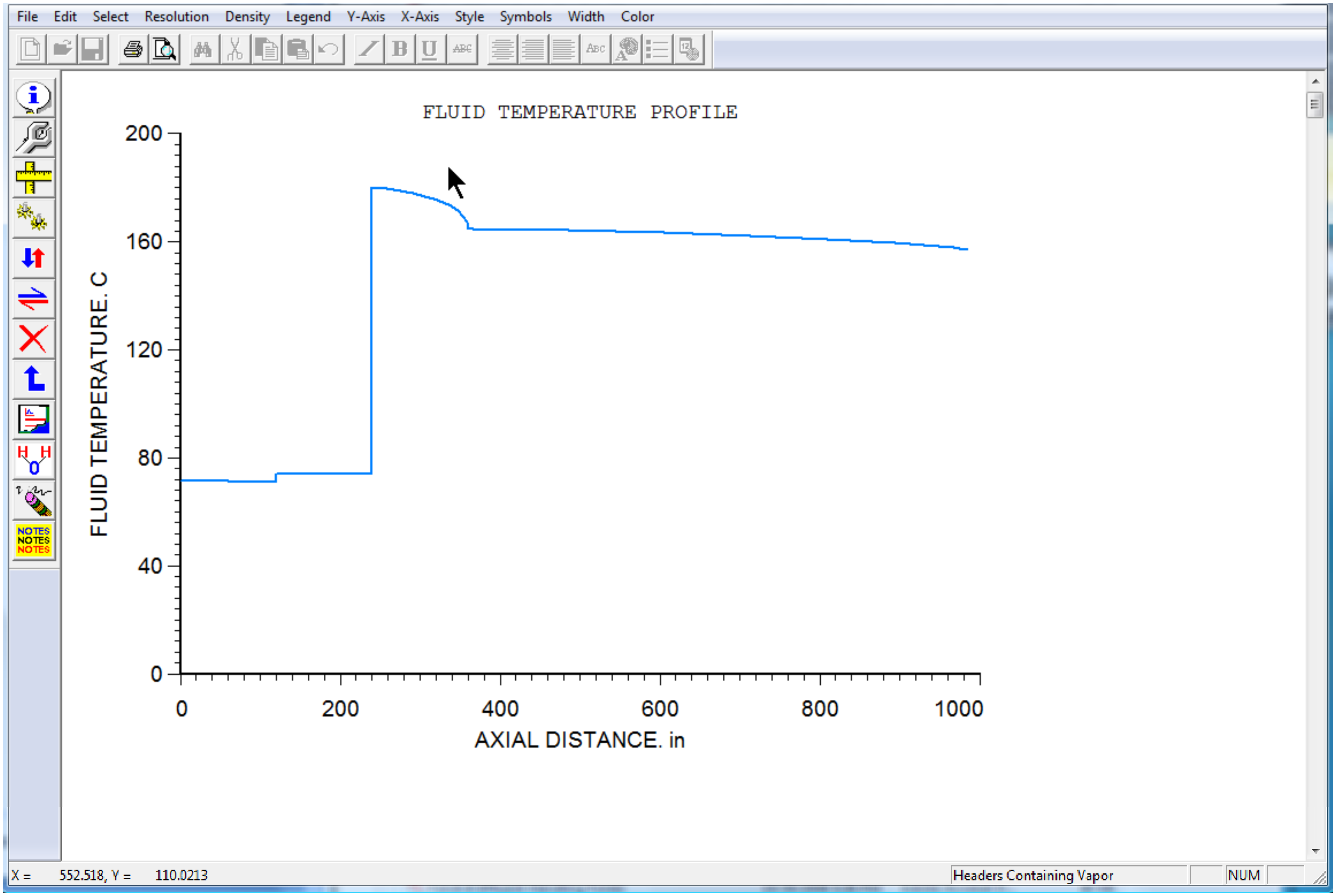


Check header and piping temperatures

- Determine if the temperature in the collection system or effluent handling equipment is lower than the minimum design metal temperature (MDMT) or greater than the design temperature at that location
- Temperatures outside of these limits may challenge the mechanical integrity of the piping or vessels
- Use -20°F as a screening criterion for the MDMT
- Check for condensation along the header and sub-header
- Check hydraulics for cold ambient temperature conditions

ioMosaic

Check the flare temperature profile



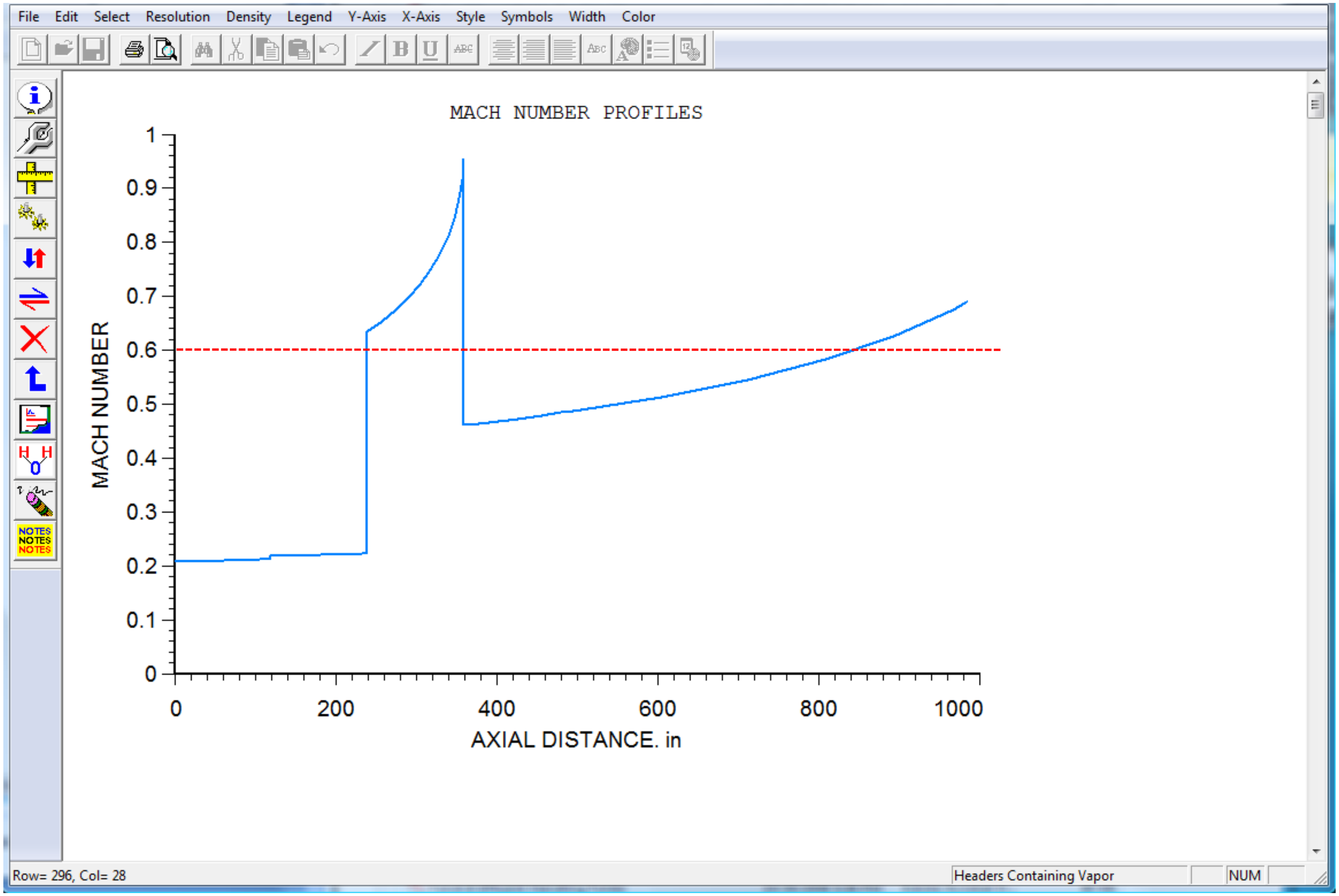
ioMosaic

Check the header and piping flowing velocities

- Determine whether or not the velocities within the collection system exceed your established criteria, typically 60 to 70 % of the Mach Number

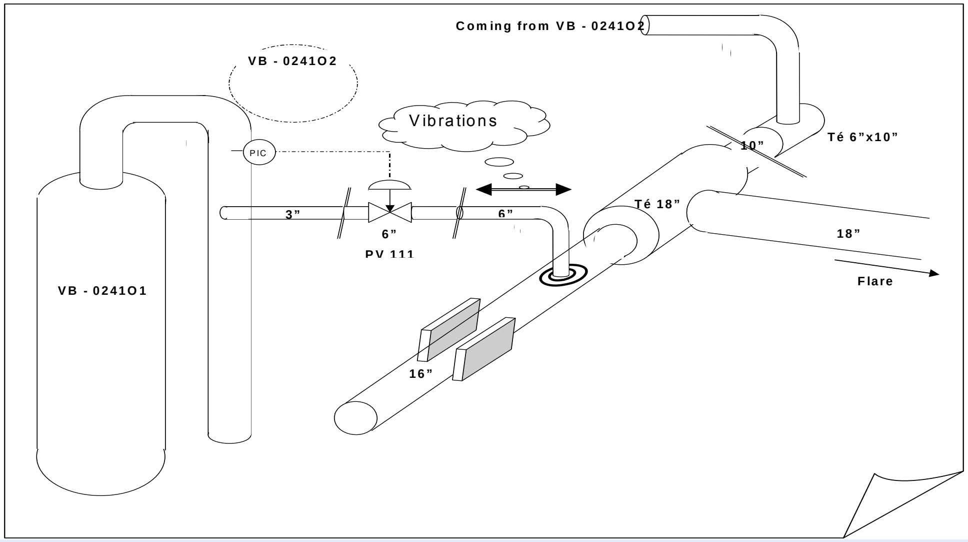
ioMosaic

Check the flare Mach Number profile



ioMosaic

Always check reaction forces and vibration risk for piping, especially for de-pressuring systems



ioMosaic

It is all about backpressure! Check backpressure for flowing devices

- For conventional type relief valves, back pressures greater than 10% result in effective back pressure correction factors of 0.
- For pilot-operated relief valves, back pressures greater than the choked flow pressure ratio (typically 55-65%) will cause subsonic flow and flow reduction.
- For bellows type relief valves, back pressures greater than 30% will result in a back pressure correction factor less than 1, depending on the allowable overpressure and model of relief valve.
  - Back pressure correction factors are not generally reliable above back pressures greater than 50% of the set pressure.
- Compare the calculated flow rate vs. relief requirement to ensure adequate overpressure protection for the scenario under evaluation
  - For multiple relief devices providing overpressure protection as a system, compare the total calculated flow rate to the total required relief rate.

It is all about backpressure! Check backpressure for non-flowing relief devices

- Each pressure relief device is designed to withstand a maximum amount of superimposed backpressure when it is not flowing.
- The API Standard 526 limits for flanged pressure relief valves are typically used as the screening criterion for the allowable superimposed backpressures.
- The manufacturer of the pressure relief device also publishes tested limits, which can be used as the criterion for whether or not the superimposed backpressure is acceptable.

It is all about backpressure! Check backpressure for non-flowing rupture disks

- If the superimposed back pressure in a header approaches the bursting pressure of a rupture disk that is not relieving, the disk may burst

- Identify any superimposed back pressures greater than 75% of the burst pressure of a rupture disks

ioMosaic

It is all about back pressure! Check back pressure for bellows type pressure relief valves

- Significant superimposed back pressures on bellows type relief valves may lower the opening pressure of the relief valve and cause the relief valve to open.

- In some cases where the header pressure exceeds the set pressure of a bellows type relief valve, especially for those relief valves with very low set pressures, the relief valve may open and introduce the header fluid back into the vessel being protected by the relief valve.

- Identify any superimposed back pressures greater than 75% of the set pressure of bellows type relief valves.

ioMosaic

It is all about back pressure! Check back pressure impact on other connected systems

- Check if the back pressure on other interface points will impact the operation of that source

- Determine if the pressure in the collection system or effluent handling equipment exceeds the design pressure at that location

ioMosaic

Check the purge gas requirements

- Vendor specifications
- Requirements to prevent air ingress due to physical effects such as wind, density, and diffusion effects
- Requirements to prevent air ingress due to thermal contraction
- Requirements for minimum heating value
  - Steamout of equipment during maintenance activities
  - Minimum heating value requirements during maintenance modes of operation

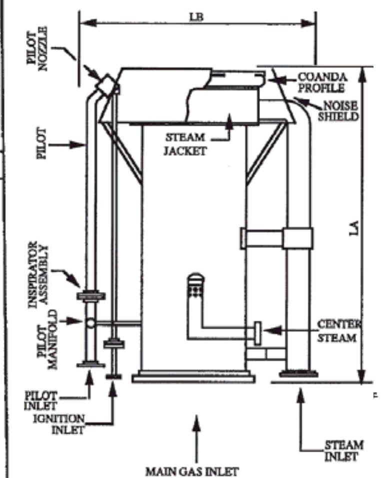
ioMosaic

Consider using a pressure drop for the flare tip of at least 2 psi

- Pressure drop is typically the result of a flame retainer at the flare tip that restricts the flow area from 2 to 10 %
- Represent the flare tip using a piping segment with a user specified pressure drop, or
- Estimate a velocity head loss using flare tip data obtained from the manufacturer

General Information:

Tag No.:	308D-4/5	
Model:	QFSC-48	Type: Steam-Assisted
Length:	10'- 0 "	
Weight:	3475 lbs	
No. of pilots:	4	(By Others)



Design Case:

Governing Case:	Case 1
Molecular weight:	55.7
L. H. V. :	2,882 BTU/SCF
Temperature:	268 deg. F
Available pressure:	5.0 psig
Design flow rate:	1,632,000 lbs/hr
Design smokeless rate:	200,000 lbs/hr
Steam Consumption (@ max. smokeless rate):	43148 lbs/hr
Approximate exit velocity:	391 ft/s
Approx. tip press. drop:	1.73 psig

Construction:

Upper Section:	310 ss	Windshield:	No
Lower Section:	310 ss	Flame retention Ring:	310 ss
Refractory:	None	Refractory Thk:	N/A
Surface Preparation:	SSPC-SP6	Primer:	High Heat Aluminum
Paint (c. s. surfaces):	High Heat Aluminum		

Connections:	Qty.	Size	Type	Material
N1 - Flare Gas Inlet:	1	48 "	Class 175 RF	Carbon Steel
N2 - Upper Steam:	1	6 "	300# RF	Carbon Steel
N3 - Center Steam:	1	3 "	150# RF	Carbon Steel
N4 - Pilot Gas: (By others)				
N5 - Ignition Line: (By others)				

Miscellaneous Notes:

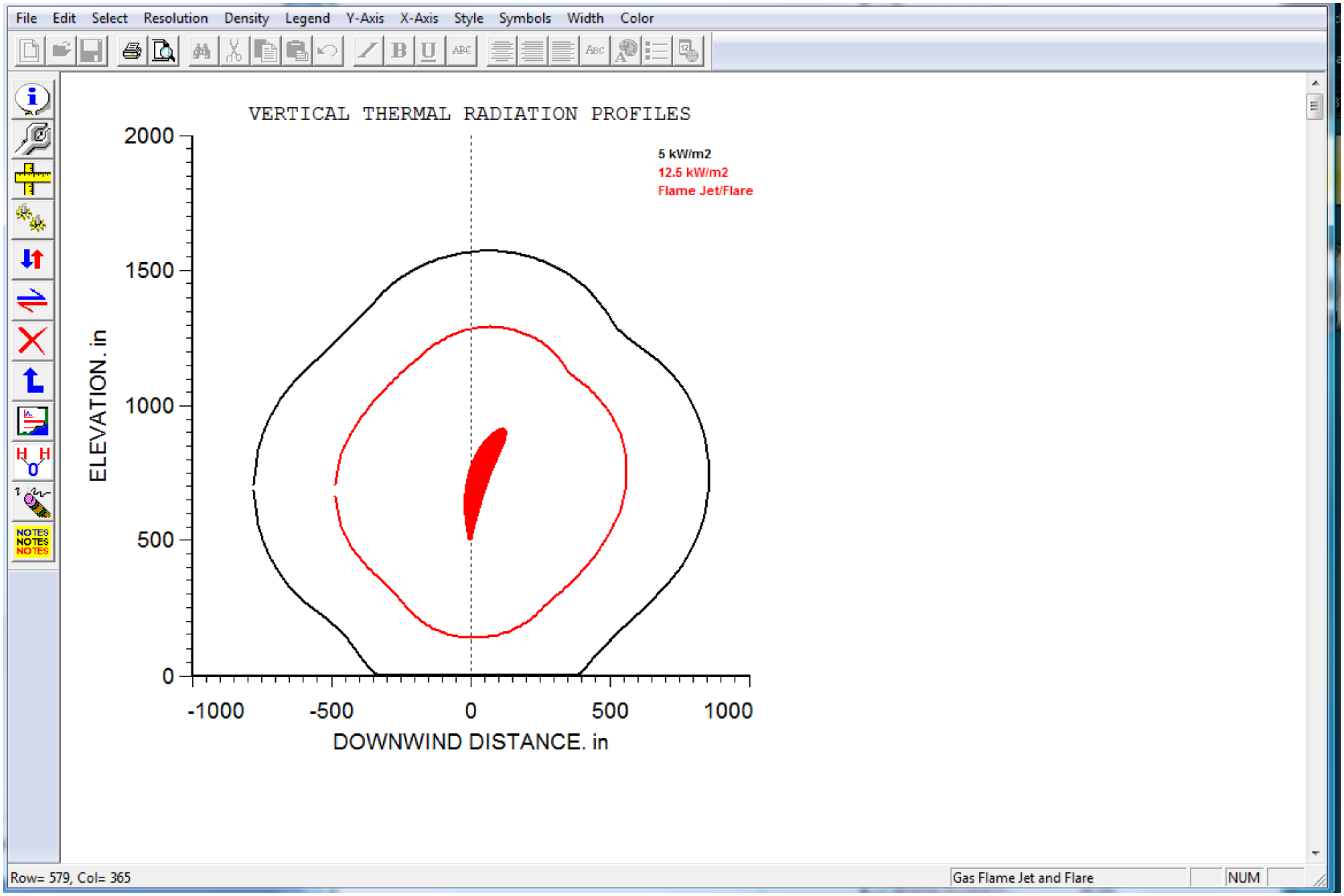
1. Includes Purge Reducing Integral Velocity Seal
2. Required Purge Rate = 1350 SCFH.
3. Existing pilots (4) and ignition system will be used.

Check the flare tip velocity

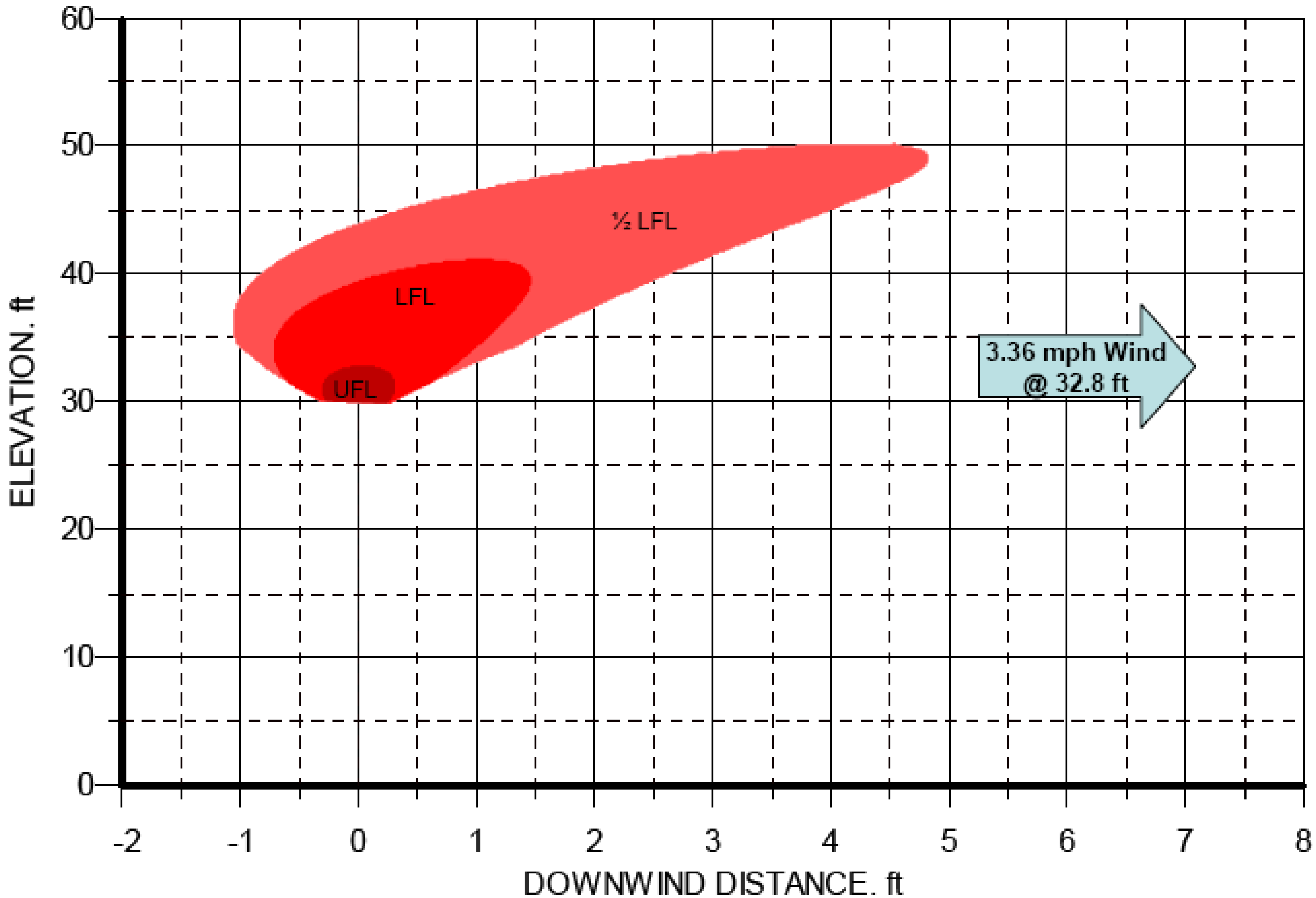
- API recommended limit of ½ Mach for low velocity (subsonic) flares
- EPA criteria from 40 CFR 60.18 based on the net heating value of the fluid being sent to the flare

ioMosaic

Check the thermal radiation impact at ground level and for elevated structures



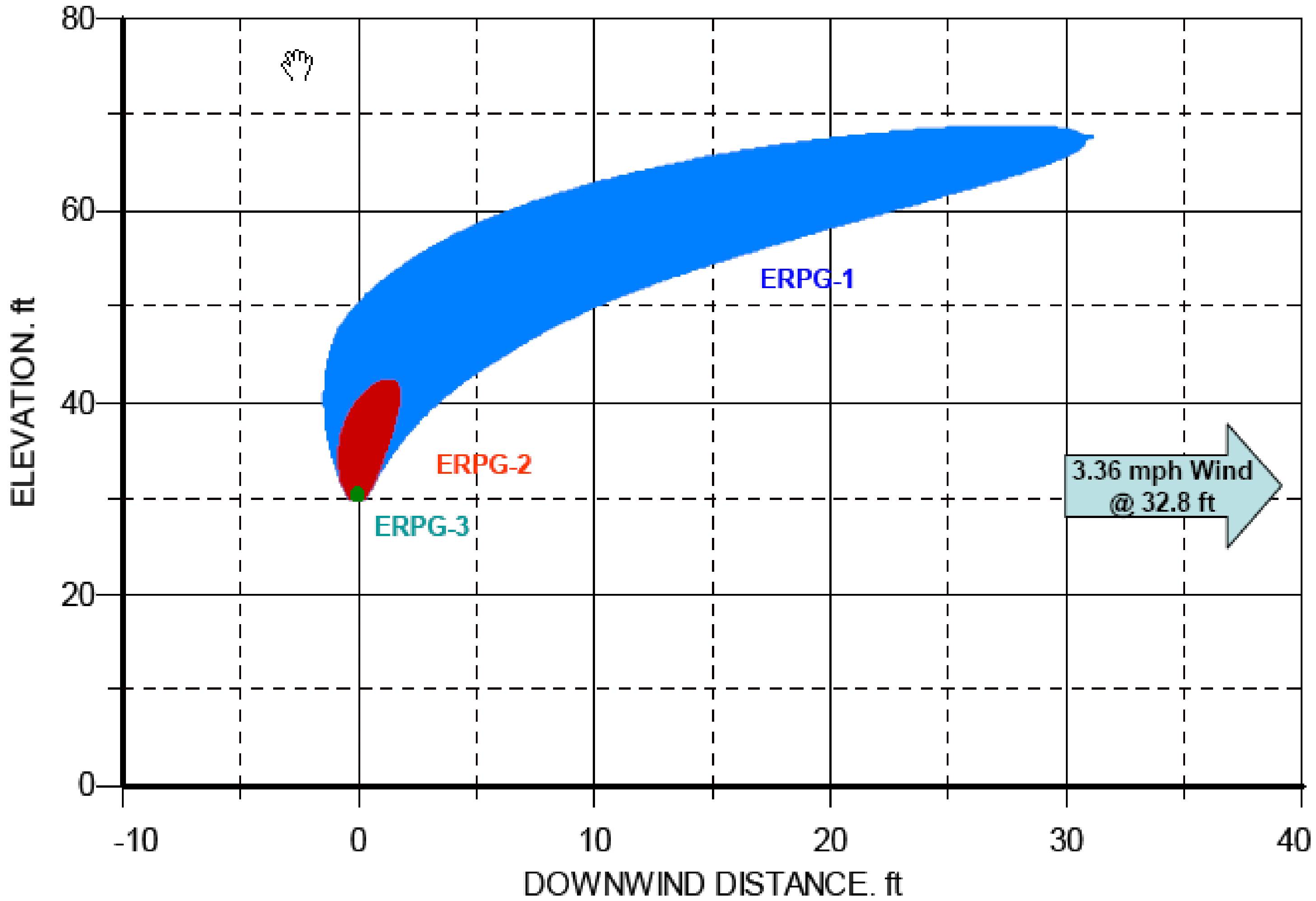
Estimate the distance to ½ LFL at low wind speed and stable atmospheric conditions in case of a flame out



Source: SuperChems Expert

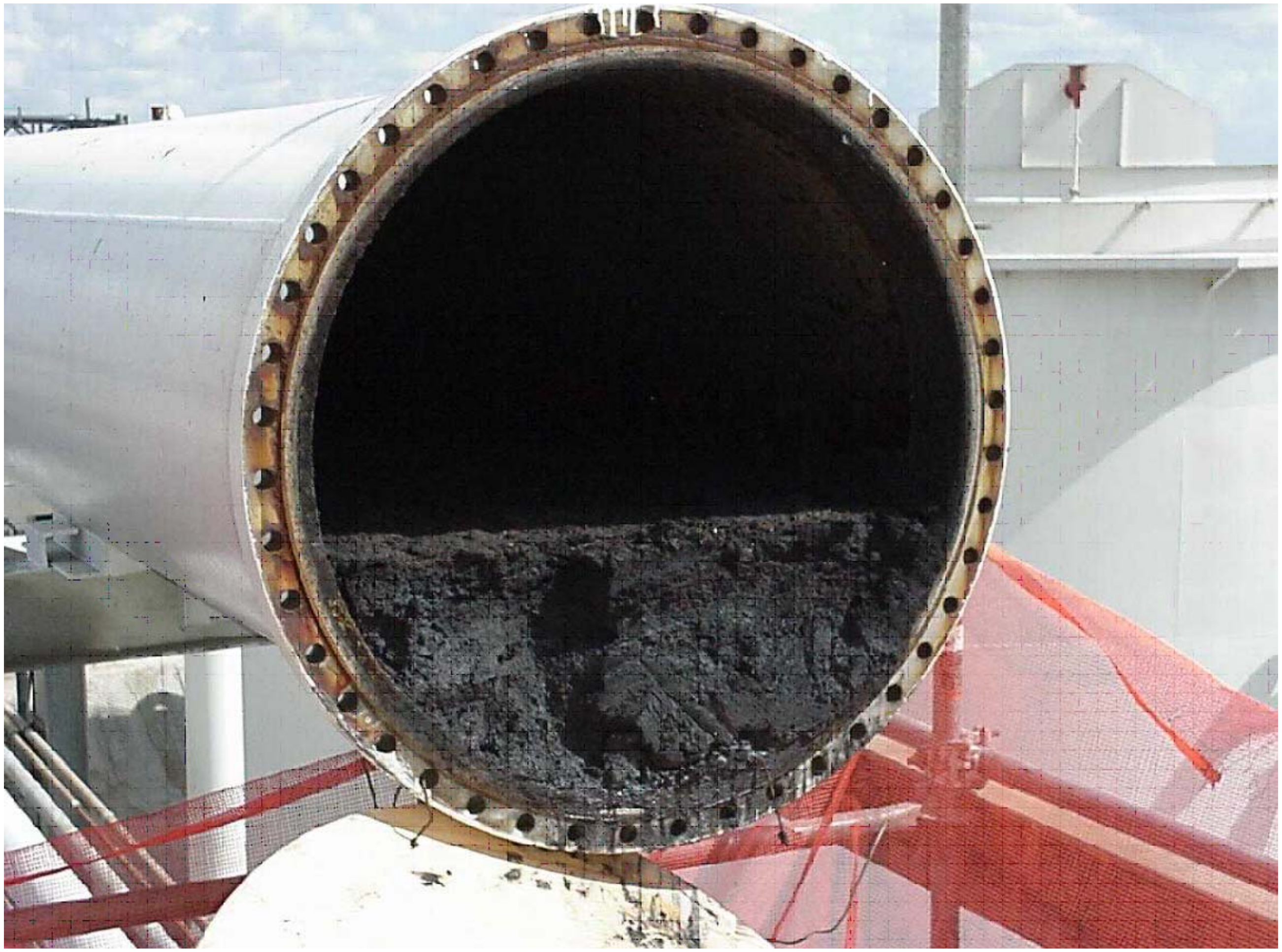
ioMosaic

Estimate the distance to ERPG-1, 2, or 3 at low wind speed and stable atmospheric conditions in case of a flame out



Source: SuperChems Expert

Consider using a piping surface roughness coefficient up to 10 times that of new pipe



ioMosaic

Check the suitability of all knockout drums

- Operating pressure
- Design pressure
- Retention time
- Path length or superficial vapor velocity required to achieve the required separation
- Hot and cold knockout drums

ioMosaic

The ideal gas equation of state is often used in flare hydraulics studies to simplify the calculations (not recommended by ioMosaic for high pressure systems)

Even with high speed computing and the availablity of many useful and accurate equations of state, many engineers still use a constant γ ideal gas equation of state. This is used for example in calculating the hydraulics of flare headers at low pressure. A constant value of γ is used to define the equation of state and all the associated thermodynamic properties:

$$\gamma = \frac{C_p}{C_v} = \frac{C_p}{C_p - R} \tag{B.64}$$

or

$$C_p = R_g \frac{\gamma}{\gamma - 1} \tag{B.65}$$

The pressure, enthalpy, and internal energy can be calculated using the following expressions:

$$P = \rho e (\gamma - 1) = \frac{\rho R_g T}{M_w} \tag{B.66}$$

$$e = \frac{P}{\rho (\gamma - 1)} = \frac{R_g T}{M_w (\gamma - 1)} \tag{B.67}$$

$$h = e + \frac{P}{\rho} = \frac{P}{\rho}\left(\frac{\gamma}{\gamma - 1}\right) = \frac{R_g T}{M_w}\left(\frac{\gamma}{\gamma - 1}\right) \tag{B.68}$$

where e and h are internal energy and enthalpy per unit mass. It can be shown that the speed of sound in a constant flow area pipe under adiabtic conditions is equal to:

$$u_{max}^2 = \frac{\gamma R_g T}{M_w} = \left[\frac{\partial P}{\partial \rho}\right]_s = \frac{1}{\rho \kappa_s} \tag{B.69}$$

A Simple Case Study Using SuperChems Expert

ioMosaic

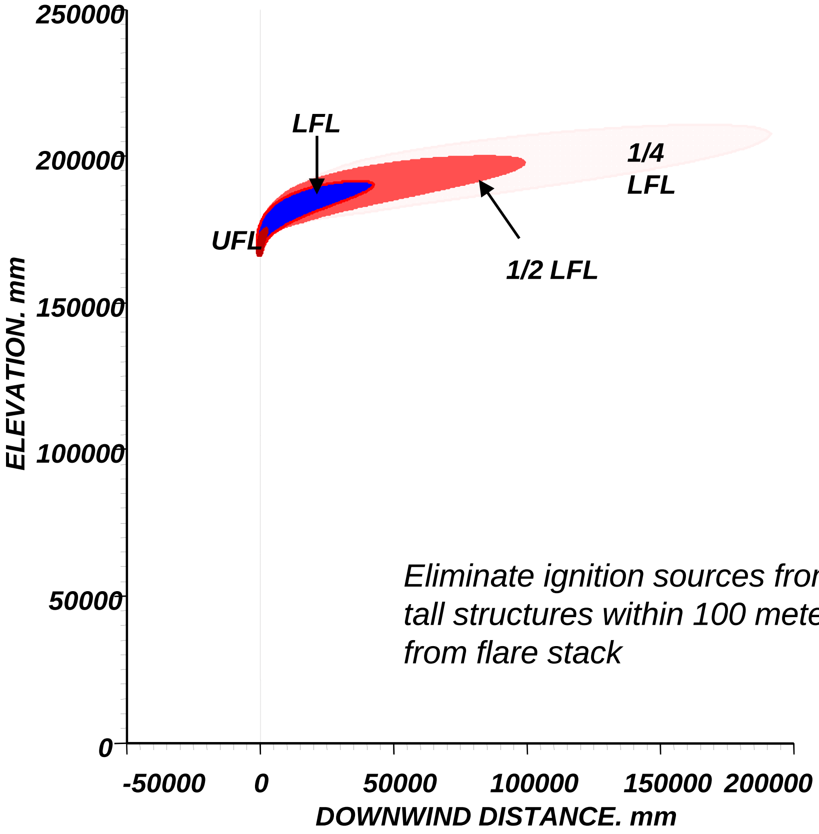
The use of high integrity pressure protection systems (HIPPS) can help you maximize the use of your existing flare structures

- Due to the design vintage of many petroleum refineries and petrochemical plants, existing pressure relief and flare systems may be overloaded
- Prior unit expansions/upgrades have increased the load on the flare for combined flaring scenarios beyond the original design intentions
- The desire to connect atmospheric relief valves to the flare for environmental and safety consideration and to eliminate blow down drums
- The addition of new process units that need access to flaring capacity

ioMosaic

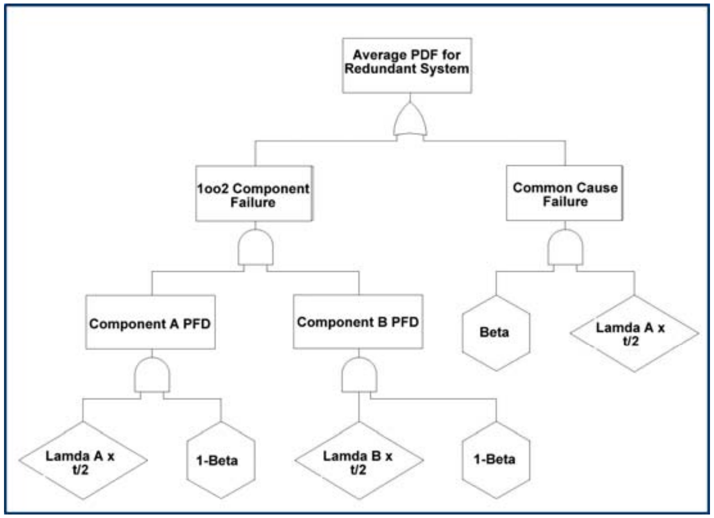
Many companies are connecting atmospheric relief systems to existing flare structures

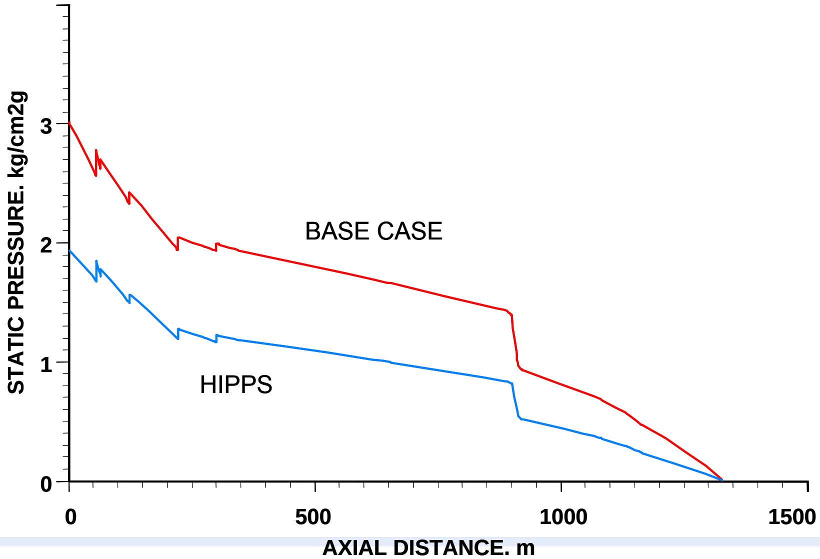
- Which existing atmospheric relief devices present vapor cloud explosion and thermal radiation hazards and need to go to the flare?
- What is the impact of the additional flaring loads on the existing flare header system and individual relief devices during combined flaring events (such as loss of power or cooling)?
- Where and how many High Integrity Pressure Protection Systems (HIPPS) should be employed to reduce the worst case flaring load?
- How should the HIPPS components be configured to achieve the required safety integrity level (SIL)?



How do I maximize the use of existing flare structures using a risk based approach?

- Dynamic simulation of relieving vessels and flare piping networks to identify capacity constraints
- Risk tolerability criteria related to vessel overpressure hazards
- Risk assessment and reliability analysis to properly select and configure the HIPPS





HIPPS, SIS, and SILS, What are they?

- The ISA/ANSI Standard S84.01 96 defines a Safety Instrumented System (SIS) as a system composed of sensors, logic solvers, and final control elements for the purpose of taking the process to a safe state when predetermined conditions are violated

- A SIS acts independent of the basic process control system

- The term *high integrity protective instrumented system* is used in Section 2.2 of API RP 521 Guide to Pressure-Relieving and Depressuring Systems, as an alternative in some scenarios for preventing overpressure and over-temperature conditions

- A HIPPS is a SIS that is designed to provide overpressure and over-temperature protection that is at least equivalent in reliability to a mechanical relief device

ioMosaic

HIPPS, SIS, and SILS, What are they?

- HIPPS have traditionally been used for rapid depressurization of hydrocrackers and acetylene hydrogenators in runaway conditions, to simultaneously reduce pressure and remove heat, where a safety valve is ineffective

- More recently, HIPPS have been employed to remove the heating supply to fractionation columns to avoid activation of the pressure relief device and causing a release to atmosphere or a flare system

- HIPPS can be used as a secondary overpressure protective system for the purpose of optimizing the design of the flare header system

- The Safety Integrity Level (SIL) is the discrete integrity level (SIL 1, SIL 2, SIL 3) of the SIS defined in terms of Probability of Failure on Demand (PFD) as presented in the table below:

Safety Integrity Level	Probability of Failure on Demand Average Range (PFD_{avg})
1	10^{-1} to 10^{-2}
2	10^{-2} to 10^{-3}
3	10^{-3} to 10^{-4}

Most flare systems evaluations require the development of flare hydraulics

- Establish worst credible global overpressure scenarios:
  - ✓ Typically driven by general power or cooling failure
  - ✓ Instrument air failure
  - ✓ Consider cascading failures
  - ✓ Consider basic process control elements
  - ✓ Consider other properly designed safeguarding systems
  - ✓ Generate an inventory of all the individual flare loads pertaining to each global scenario including relief devices, control valves, depressuring valves, etc.
  - ✓ Establish a design flare hydraulics base case

ioMosaic

Most flare systems evaluations require the development of flare hydraulics

- Verify capacity and stability for activated relief devices:
  - ✓ Check inlet and outlet piping
  - ✓ Incorporate valve characteristics for accurate flow representation
  - ✓ Multi-component representation of stream compositions
  - ✓ Multiple choke points
  - ✓ Condensation
  - ✓ Relief device flow and opening characteristics for accurate representation of peak flow
  - ✓ The presence of multiphase, supercritical, high-viscosity, and/or reacting flows

ioMosaic

Most flare systems evaluations require the development of flare hydraulics

- Analyze Flare Systems Hydraulics:
  - ✓ Analyze backpressure, flow reduction, and pressure accumulation (% MAWP) in protected equipment (base case system profile)
  - ✓ Identify devices and sub-headers that are deficient
  - ✓ Exclusion zones for thermal radiation and noise restrictions

ioMosaic

Typical flare systems design constraints include flow velocity and backpressure limits

Design Criteria	**VALUE**	**DESCRIPTION**
Maximum Flow Velocity	Mach ≤ 0.6	Maximum value for header and subheaders design
	$\rho u^2 \leq 100{,}000$ Pa	Maximum value for discharge piping, header, and subheader design for gas flow
	$\rho u^2 \leq 50{,}000$ Pa	Maximum value for discharge piping, header, and subheader design for two-phase flow
	$N_{Re} \geq 15{,}400\ \rho_f/\rho_a$ Mach ≥ 0.2	Minimum exit velocity required for gas flow from vapor stacks to ensure good mixing and dispersion
Flow rate	Rated Capacity	Value for subheaders and relief discharge piping design
	Required Capacity	Value for main header design
Backpressure	≤ 0.1 Pset	Conventional relief valves
	≤ 0.3 Pset	Balanced relief valves. Balanced relief valves may be accepted for backpressures up to 0.5 P_{set} with prior consultation with manufacturer and ioMosaic
	≤ 0.5 Pset	Pilot operated valves. Pilot relief valves will be accepted for backpressures up to 0.7 P_{set} with prior consultation with manufacturer and ioMosaic

ioMosaic

Typical flare systems design constraints include thermal radiation and noise limits

Design Criteria	VALUE		DESCRIPTION
Radiation Intensity Solar radiation component should be added and can be as high as 317 Btu/h ft^2 in some geographical locations like the middle east and south America	500 *BTU/h ft^2*	1.57 *kW/m^2*	Value at any location where personnel with appropriate clothing may be continuously exposed
	630 *BTU/h ft^2*	1.98 *kW/m^2*	Maximum value for pressured storage equipment
	1000 *BTU/h ft^2*	3.15 *kW/m^2*	Maximum value for atmospheric storage equipment
	1500 *BTU/h ft^2*	4.72 *kW/m^2*	Heat intensity in areas where emergency actions lasting several minutes may be required by personnel without shielding but with appropriate clothing. Maximum value for Process equipment.
	2000 *BTU/h ft^2*	6.30 *kW/m^2*	Heat intensity in areas where emergency actions up to 1 minute may be required by personnel without shielding but with appropriate clothing. Maximum value for Knock Out Drum.
	3000 *BTU/h ft^2*	9.45 *kW/m^2*	Heat intensity at any location to which people have access; exposure should be limited to a few seconds, sufficient for escape only.
Emergency Flaring Noise (working areas)	85 dBA		At maximum flaring load
Emergency Flaring Noise (residential areas)	80 dBA		At maximum flaring load
Normal operation Flaring Noise (residential areas)	68 dBA		At maximum flaring load

ioMosaic

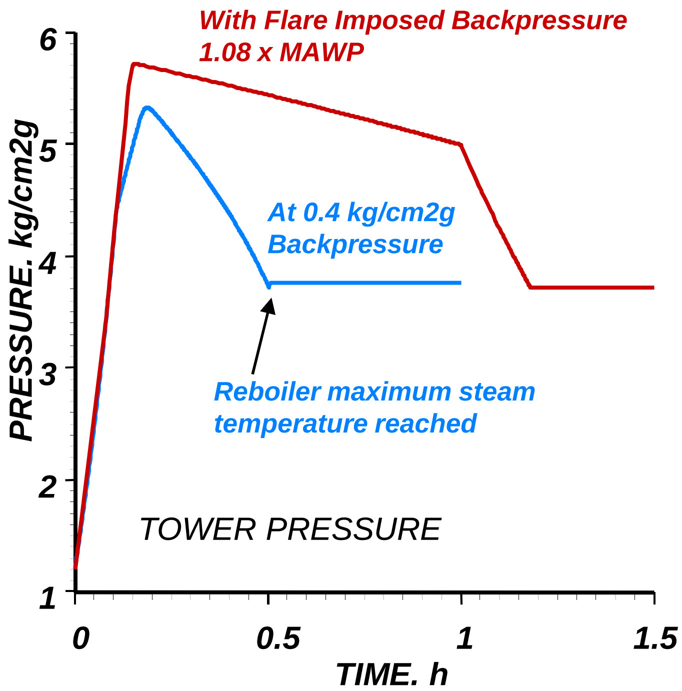
Always use the right combination of relief device area and discharge coefficient

Device Type	Determine Required Relief Capacity	Determine Pressure Drop, Relief Device Stability, and Effluent Handling Systems Performance
PRV alone	ASME Area ASME K_d	ASME Area ASME $K_d/0.9$
	API Area API K_d	API Area/0.9 API K_d
RD alone	Use as a pipe fitting with K_R Multiple the entire system relief capacity by 0.9	Use actual piping without a derating factor
RD/PRV Combination	ASME Area ASME K_d x CCF RD K_R	ASME Area ASME $K_d/(0.9 \times CCF)$ RD K_R
	API Area API K_d x CCF RD K_R Use a CCF of 0.9 if actual value is not available	API Area/0.9 API K_d/CCF RD K_R

ioMosaic

Consider the following simple fixes before using HIPPS

- Automate shutdowns and/or isolation systems currently requiring operator intervention
- If possible, Maximize use of bellows/pilot valves
- If possible, make reasonable header size and relief device adjustments to correct deficiencies
- Review timing of loads (e.g., automated de-pressuring systems)
- Model vessel dynamics and establish actual pressure accumulation based on flare pressure profiles:
  - ✓ Reduced set points less than MAWP
  - ✓ Required flow rate vs. excess relief capacity
- To MAXIMIZE the use of the existing flare collection system, these aspects need to be thoroughly defined and evaluated



ioMosaic

Typically HIPPS are considered for de-bottlenecking existing flare collection systems in order to address one or more of the following conditions

- Flare tip and/or sub-header connection Mach Number > 0.6
- Excessive PSV backpressure
- Excessive vessel accumulation/overpressure
- High flare thermal radiation levels on/off site
- High flare noise levels on/off site
- Adding atmospheric relief devices to the existing flare collection system

ioMosaic

The use of HIPPS for flare systems was first outlined by CCPS and Williams

- Select HIPPS candidates
- Define HIPPS configurations
- Confirm HIPPS design flare loads
- Verify required SIL

Recommended Reading:

AIChE/CCPS, "Guidelines for Safe Automation of Chemical Processes" , AIChE, 1993.

Williams and Donavan, "Reliability-based Approach Reduces Flare Design Relief Load", Oil and Gas Journal on December 15, 1997.

Williams, "Reliability for Safety Instrumented Systems" appeared in Chemical Engineering Progress Magazine on September of 2004.

ioMosaic

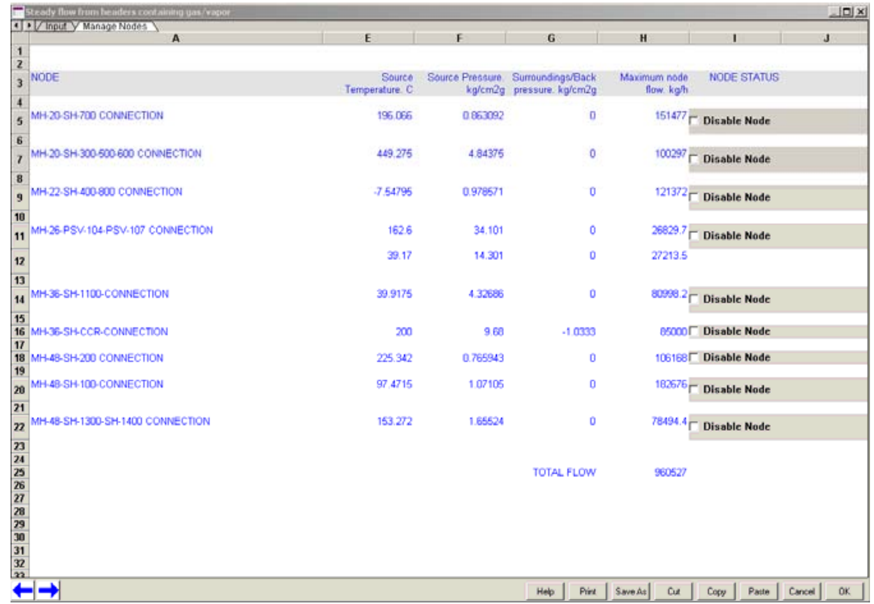
Maximizing the use of existing flare structures using HIPPS is a systematic process

- Evaluate and optimize all individual relief lines
- Evaluate and optimize all individual sub-headers
- Evaluate the flare hydraulics for a base case (no credit for Safeguards, Global Scenario)
- Evaluate the flare hydraulics for a best case assuming all Safeguards work as designed (Global Scenario)
- Evaluate the flare hydraulics for worst possible failure combination of the safeguards yielding the largest flare load back into the flare system (Global Scenario)
- Evaluate and establish the tolerable safeguard failure frequencies based on the concepts developed by Williams and Donavan

ioMosaic

HIPPS Selection Methodology

- Using base case flare loads established earlier:
  - ✓ Identify largest loads for elimination or mitigation
  - ✓ Develop preliminary configuration of HIPPS candidates and SILs for each sub-header
  - ✓ Apply risk-based approach for selecting number of HIPPS and required SILs
  - ✓ Run flare network simulation to check backpressure, overpressure, flow rates
- Perform iterative design of HIPPS configurations and simulations to arrive at most cost effective and <u>tolerable risk</u> solution


Steady flow from headers containing gas/vapor

Input / Manage Nodes

NODE	Source Temperature, C	Source Pressure, kg/cm2g	Surroundings/Back pressure, kg/cm2g	Maximum node flow, kg/h	NODE STATUS
MH-20-SH-700 CONNECTION	196.066	0.863092	0	151477	Disable Node
MH-20-SH-300-500-600 CONNECTION	449.275	4.84375	0	100297	Disable Node
MH-22-SH-400-800 CONNECTION	-7.54795	0.978571	0	121372	Disable Node
MH-26-PSV-104-PSV-107 CONNECTION	162.6	34.101	0	26829.7	Disable Node
	39.17	14.301	0	27213.5	
MH-36-SH-1100-CONNECTION	39.9175	4.32686	0	80998.2	Disable Node
MH-36-SH-CCR-CONNECTION	200	9.68	-1.0333	85000	Disable Node
MH-48-SH-200 CONNECTION	225.342	0.765943	0	106168	Disable Node
MH-48-SH-100-CONNECTION	97.4715	1.07105	0	182676	Disable Node
MH-48-SH-1300-SH-1400 CONNECTION	153.272	1.65524	0	78494.4	Disable Node
			TOTAL FLOW	960527	

Help Print Save As Cut Copy Paste Cancel OK

Largest loads for elimination may include vessels that have adequate relief but are good candidates such as columns using steam reboilers, for example

ioMosaic

HIPPS Selection Methodology

- The risk tolerability of an overpressure condition in a vessel should be assigned based on:
  - ✓ The consequences of the overpressure in terms of vessel integrity
  - ✓ The frequency of occurrence of the undesirable outcome
- Using recognized vessel overpressure effect characteristics, a system of consequence and frequency criteria can be established
  - ✓ 135% accumulation has no effect on steel vessels
  - ✓ 400-500% Accumulation results in rupture, an unacceptable event

Overpressure Event Risk Matrix*

EFFECT OF PRESSURE ACCUMULATION IN CARBON STEEL VESSELS

Accumulation	Effects	Remarks	Frequency Target
<135	None expected	None	10^{-2}/yr
135-165	Potential for slight permanent deformation	This range of pressure corresponds to the tensile limit of the vessel and is both material- and code-dependent. The lower and upper limits correspond to ASME VII, Div.2 and ASME VIII, Div. 1 (1998 edition and earlier) vessels, respectively. ASME VIII, Div. 1 (1998 edition with 1999 addenda) vessels fall in between these values. Therefore, a respresentative value for this range is 150%.	10^{-3}/yr
165-300	Permanent deformation, possible small leak	Valid for remote contingencies, as more frequent overpressuring could weaken the vessel by fatigue.	10^{-4}/yr
300-400	Same as avove, but with a higher likelihood of a large leak or burst	Dangerous overpressuring.	10^{-5}/yr
400-500	Burst	Typical for healthy ASME VIII code vessels.	10^{-6}/yr

**** Based on aligning a vessel rupture with a 1 in a million year tolerable event***

Accumulation	Frequency
< 135	1 in 100 years
135-165	1 in 1000 years
165-200	1 in 10,000 years
200-300	1 in 100,000 years
>300	Not allowed

ioMosaic

Sample: Tolerable Event Frequency Targets Based on Level of Accumulation

Flare Sub-header	Problem PSVs	Overpressure Ratio	Tolerable Event Frequency Target Occurrence/yr	Comments
S-100	PSV-103	1.43 x MAWP	10^{-3}/yr	
S-200	PSV-203	1.69 x MAWP	10^{-4}/yr	
S-300	PSV-302	1.5 x MAWP	10^{-3}/yr	
	PSV-304	1.39 x MAWP	10^{-2}/yr	
S-400	None	$\leq$ 1.1 x MAWP	0	No HIPPS required
S-500	PSV-502	1.47 x MAWP	10^{-3}/yr	

Notes: (1) SuperChems Expert calculates vessel accumulation as part of the header analysis

A Simple HIPPS Risk Analysis Example

- General Power Failure Occurrence is 2/yr based on refinery experience
- Probability of failure on demand (PFD) for SIL:
  - SIL 1 $= 10^{-1}$ /yr
  - SIL 2 $= 10^{-2}$ /yr
  - SIL 3 $= 10^{-3}$ /yr

HIPPS Risk Analysis Example

- Arrange HIPPS candidate loads in decreasing order starting with largest
- Assign SIL levels to loads to minimize the number of large loads that have to fail to meet the target event frequency
- Consider effect of different failure sequences on header design load and converge on a solution

ioMosaic

How Many HIPPS are Enough?

- There are sufficient HIPPS when the following criteria are satisfied:
  - All vessels comply with code requirements for overpressure accumulation if all HIPPS function as designed on demand, <u>and</u>
  - It is not possible for any simultaneous failures of one or more HIPPS to cause code violations at a frequency $\geq$ the established target tolerability frequency
  - Code violations include thermal radiation and noise impacts
- SILs need to be selected to satisfy the above criteria

ioMosaic

Sub-Header S-300 Example: ***Target Tolerability Frequency = 10^{-3}/YR***

Item #	PSV Tag No.	Flow Rate. Kg/hr	Specified SIL	Combined Failure Freq. Occur/yr	Header Load w/Subsequent HIPPS Failures Kg/Hr	Number Failures
					4835	none
1	PSV-304	74231	2	2.00E-02	79066	1
2	PVS-303	72458	2	2.00E-04	151524	2
3	PSV-302	70443	3	2.00E-07	221967	3
4	PSV-306	21326	1	2.00E-08	243293	4
5	PSV-305	13590	1	2.00E-09	256883	5
6	PSV-307	4835				
		256883				

ioMosaic

Sub-Header S-300 Example

- If all HIPPS function on demand, the problem loads from PSV-302 and 304 will be eliminated and code compliance is obtained

- Using the failure sequence of PSV-304 and PSV-303 results in the largest design load on the flare that also achieves the target event frequency:
  - No other failure combination sequence will add back as much load to the flare (151,524 Kg/h)

- By making the PSV-302 HIPPS a SIL 3, the frequency of accumulation exceeding code is $2\text{x}10^{-3}$/year from a single failure (if it fails first)
  - With the other HIPPS loads removed, the pressure is < 1.35xMAWP and the revised target frequency of $1\text{x}10^{-2}$/year is met

ioMosaic

Business Confidential Document

Sample Flare Systems Results

SuperChems Expert v5.91

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

Sample Output

REPORT WRITER

	A	B	C	D
7	Barometric pressure at site elevation. kg/cm2g	0		
8				
9	Liquid Density Method	Equation of state		
10	VLE/PVT Method	Equation of state		
11				
12	Inlet temperature. C	219.854		
13	Inlet pressure. kg/cm2g	1.939		
14	Surroundings/back pressure. kg/cm2g	0		
15				
16	Ignore chemical reaction in piping segments			
17				
18	Exit temperature. C	84.130		
19	Exit pressure. kg/cm2g	3.5469E-05		
20	Speed of sound at exit conditions. m/s	175.659		
21	Mach Number	0.446		
22	Pipe length travelled / total piping length	1.0		
23				
24	Flow rate. kg/h	557486.190		
25	Flow rate. SCFH	1.1089E+06		
26				
27	COMPONENT	MW	FLOW IN. kg/h	FLOW OUT. kg/h
28	HYDROGEN	2.016	21575.717	21575.717
29	METHANE	16.043	105872.082	105872.082
30	ETHANE	30.070	52302.907	52302.907
31	ETHYLENE	28.054	25148.216	25148.216
32	PROPANE	44.097	48628.531	48628.531
33	ISOBUTANE	58.123	8482.018	8482.018
34	n-BUTANE	58.123	34102.489	34102.489
35	ISOPENTANE	72.150	21181.317	21181.317
36	n-PENTANE	72.150	37336.958	37336.958
37	CYCLOPENTANE	70.134	631.468	631.468
38	2,2-DIMETHYLBUTANE	86.177	1849.060	1849.060
39	2,3-DIMETHYLBUTANE	86.177	1799.037	1799.037
40	2-METHYLPENTANE	86.177	8513.772	8513.772
41	3-METHYLPENTANE	86.177	5682.130	5682.130
42	1-HEXENE	84.161	0.000	0.000
43	n-HEXANE	86.177	21960.061	21960.061
44	METHYLCYCLOPENTANE	84.161	2427.561	2427.561
45	BENZENE	78.114	24257.830	24257.830
46	CYCLOHEXANE	84.161	11310.035	11310.035
47	n-HEPTANE	100.204	32673.057	32673.057
48	TOLUENE	92.141	11385.908	11385.908

Table of Contents / Vapor Flow From Headers and Subheaders / Piping Isometrics / Piping Segments

Format | Print | Save As | Cut | Copy | Paste | Cancel | OK

ioMosaic

Sample SuperChems Input

Steady flow from headers containing gas/vapor

Input | Manage Nodes

NODE	Source Temperature. C	Source Pressure. kg/cm2g	Surroundings/Back pressure. kg/cm2g	Maximum node flow. kg/h	NODE STATUS
MH-20-SH-700 CONNECTION	196.066	0.863092	0	151477	Disable Node
MH-20-SH-300-500-600 CONNECTION	449.275	4.84375	0	100297	Disable Node
MH-22-SH-400-800 CONNECTION	-7.54795	0.978571	0	121372	Disable Node
MH-26-PSV-104-PSV-107 CONNECTION	162.6	34.101	0	26829.7	Disable Node
	39.17	14.301	0	27213.5	
MH-36-SH-1100-CONNECTION	39.9175	4.32686	0	80998.2	Disable Node
MH-36-SH-CCR-CONNECTION	200	9.68	-1.0333	85000	Disable Node
MH-48-SH-200 CONNECTION	225.342	0.765943	0	106168	Disable Node
MH-48-SH-100-CONNECTION	97.4715	1.07105	0	182676	Disable Node
MH-48-SH-1300-SH-1400 CONNECTION	153.272	1.65524	0	78494.4	Disable Node
			TOTAL FLOW	960527	

Help | Print | Save As | Cut | Copy | Paste | Cancel | OK

Sample Output

REPORT WRITER

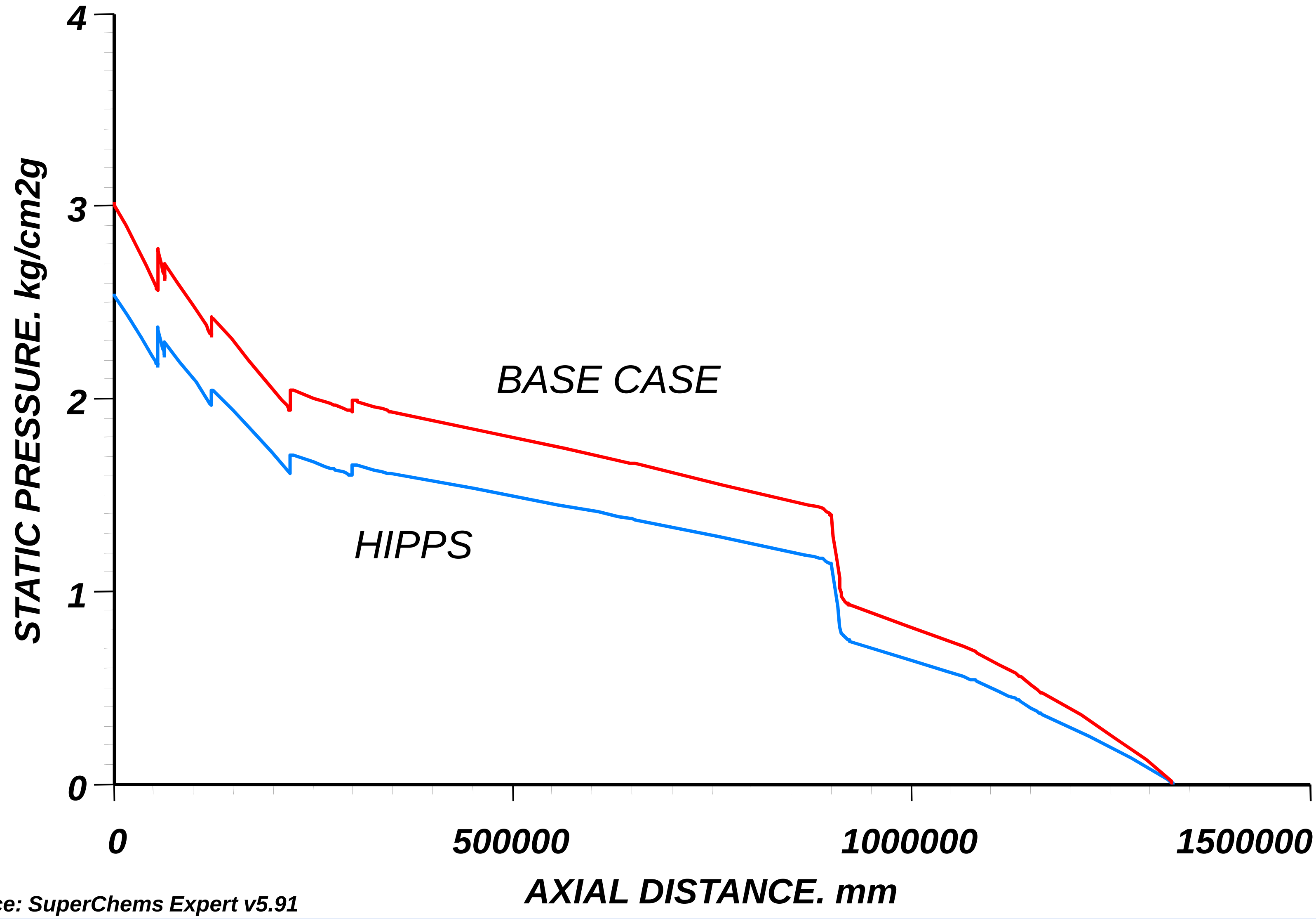
	B	C	D	E	F	G	H	I	J
88	557486.19	557486.19	1206.50	12.3059					
89	557486.19	557486.19	1206.50	12.3059					
90	557486.19	557486.19	1206.50	12.3059					
91	557486.19	557486.19	1206.50	12.3059					
92	557486.19	557486.19	1206.50	12.3059					
93	557486.19	557486.19	1206.50	12.3059					
94	557486.19	557486.19	1206.50	12.3059					
95	557486.19	557486.19	1206.50	12.3059					
96	557486.19	557486.19	1206.50	12.3059					
97									
98									
99									
100	Scenario Reference	Piping Layout	Source	Source Temperature. C	Source Pressure. kg/cm2g	Surroundings/Back pressure. kg/cm2g	Maximum node flow. kg/h	Actual node flow. kg/h	Percent of Max. Flow
101									
102	SH-700	SH-700 DETAILED PIPING	Gas header	217.1020	2.0083	1.9394	79041.80	70024.10	88.5912
103	** WARNING: Node is flowing at =88.5912 Percent of full capacity								
104									
105	SH-300-500-600	SH-300-500-600 DETAILED PIPING	Gas header	449.4190	5.0589	1.9392	85109.20	85033.30	99.9109
106									
107	SH-400-800	SH-400-800 DETAILED PIPING	Gas header	-14.8791	1.9536	1.8415	73103.30	62878.60	86.0133
108	** WARNING: Node is flowing at =86.0133 Percent of full capacity								
109									
110	SH-1100	SH-1100 DETAILED PIPING	Gas header	39.8639	4.3770	1.2721	80998.20	80188.80	99.0007
111	** WARNING: Node is flowing at =99.0007 Percent of full capacity								
112									
113	CCR UOP INPUT	DEFAULT	User Defined	200.0000	9.6800	1.2705	85000.00	85000.00	100.0000
114									
115	SH-100	SH-100 DETAILED PIPING	Gas header	94.9636	1.6434	1.2197	114141.00	112036.00	98.1556
116	** WARNING: Node is flowing at =98.1556 Percent of full capacity								
117									
118	SH-1300-1400	SH-1300-1400 DETAILED PIPING	Gas header	153.5540	1.9314	1.2188	63806.30	60282.50	94.4774
119	** WARNING: Node is flowing at =94.4774 Percent of full capacity								
120									
121									
122						TOTAL FLOW	581200.00	555443.00	95.5683

Table of Contents | Vapor Flow From Headers and Subheaders | Piping Isometrics | Piping Segments

Format | Print | Save As | Cut | Copy | Paste | Cancel | OK

Note that SuperChems estimates what fraction of actual relief device and/or sub-header flow capacity is possible instead of providing a pass/fail answer. This data is used by dynamic models in SuperChems to determine if specific levels of concerns of vessel MAWP/MAWT are exceeded.

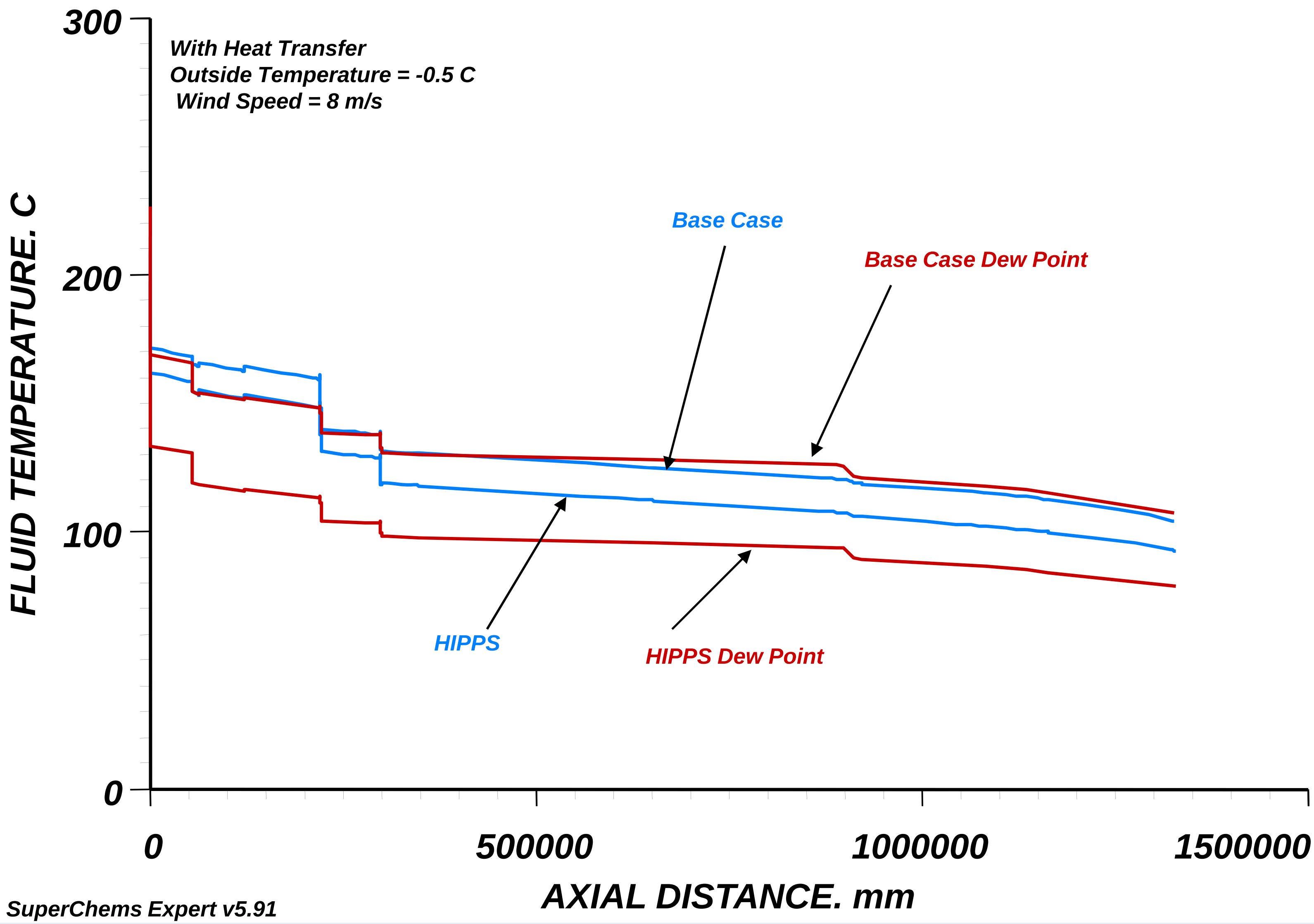
Main Flare Header Pressure Profile



Source: SuperChems Expert v5.91

ioMosaic

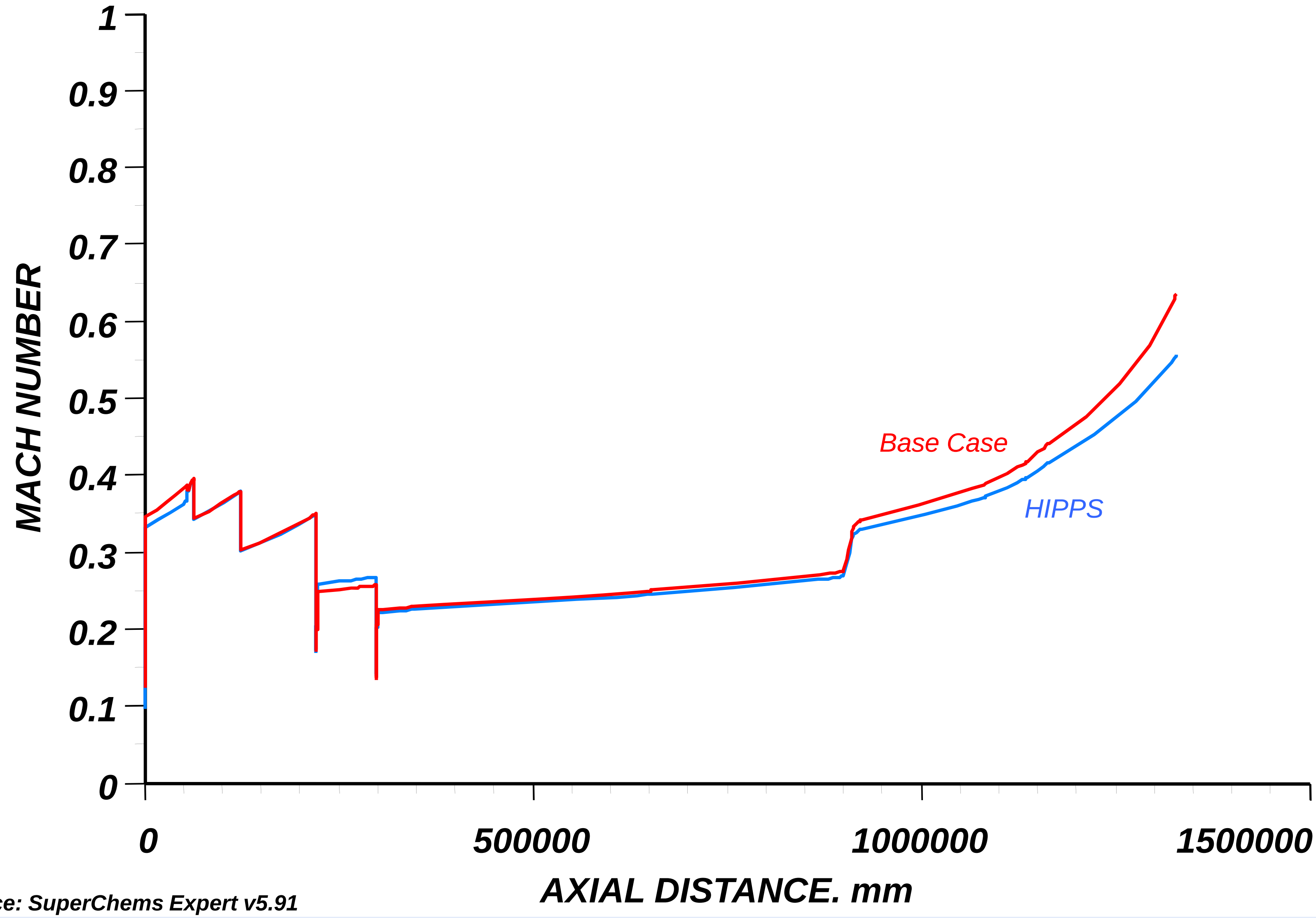
Main Flare Header Temperature Profile



Source: SuperChems Expert v5.91

ioMosaic

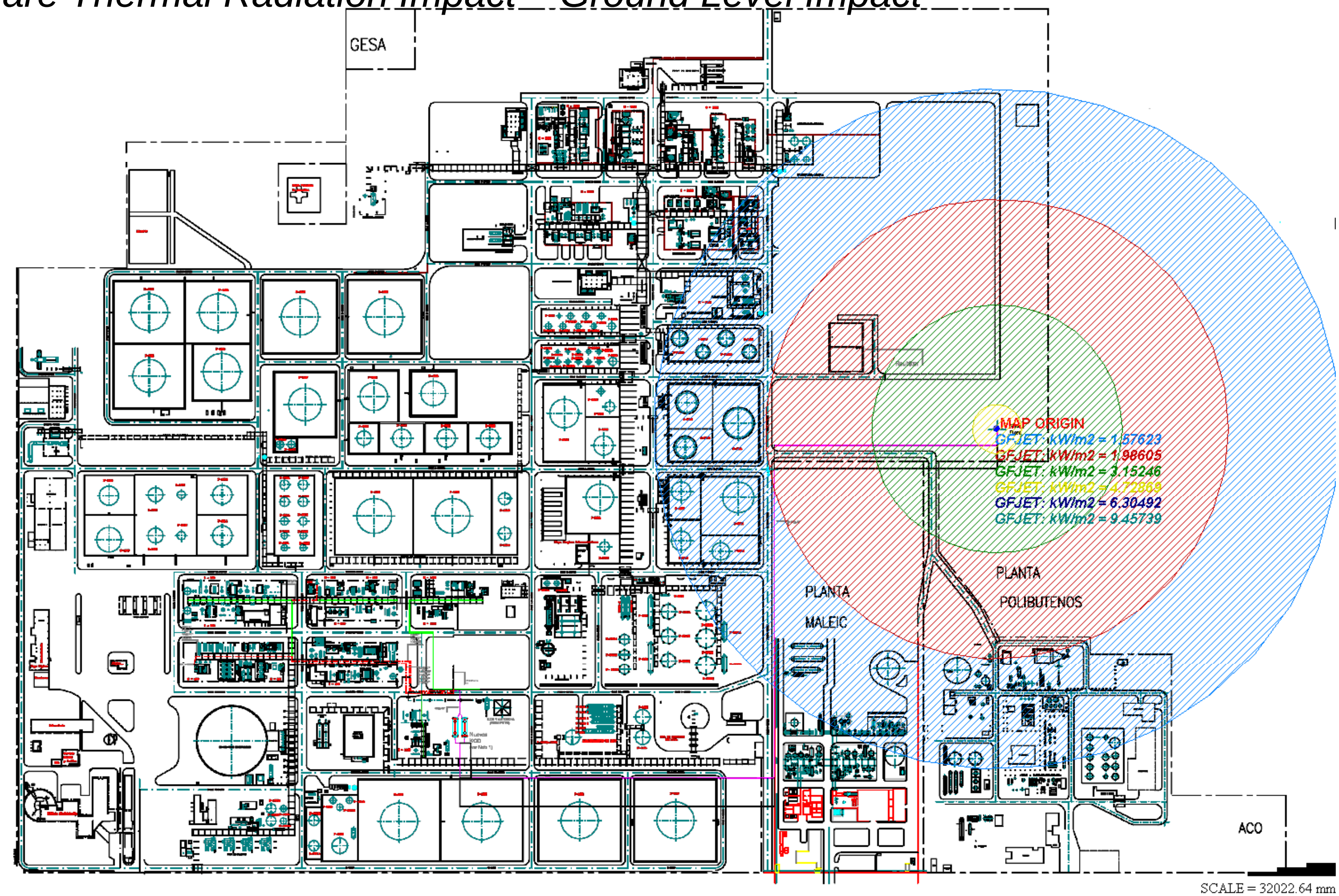
Main Flare Header Velocity Profile



Source: SuperChems Expert v5.91

ioMosaic

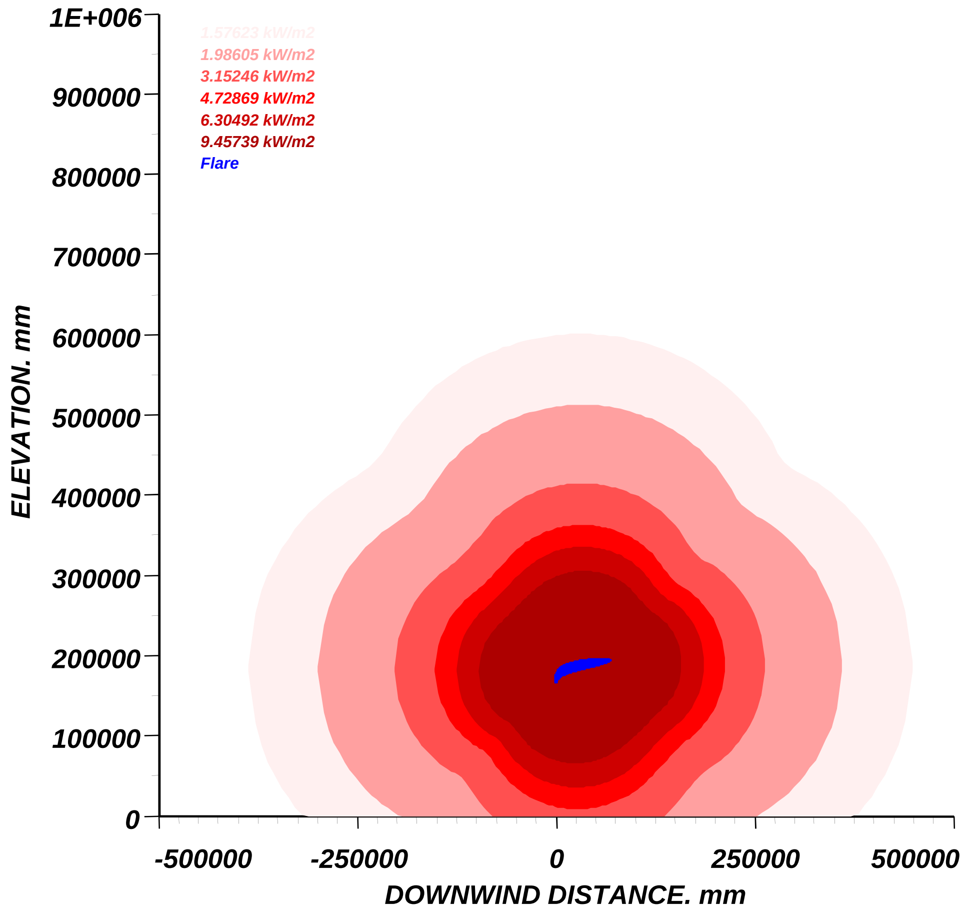
Flare Thermal Radiation Impact – Ground Level Impact



Source: SuperChems Expert v5.91

ioMosaic

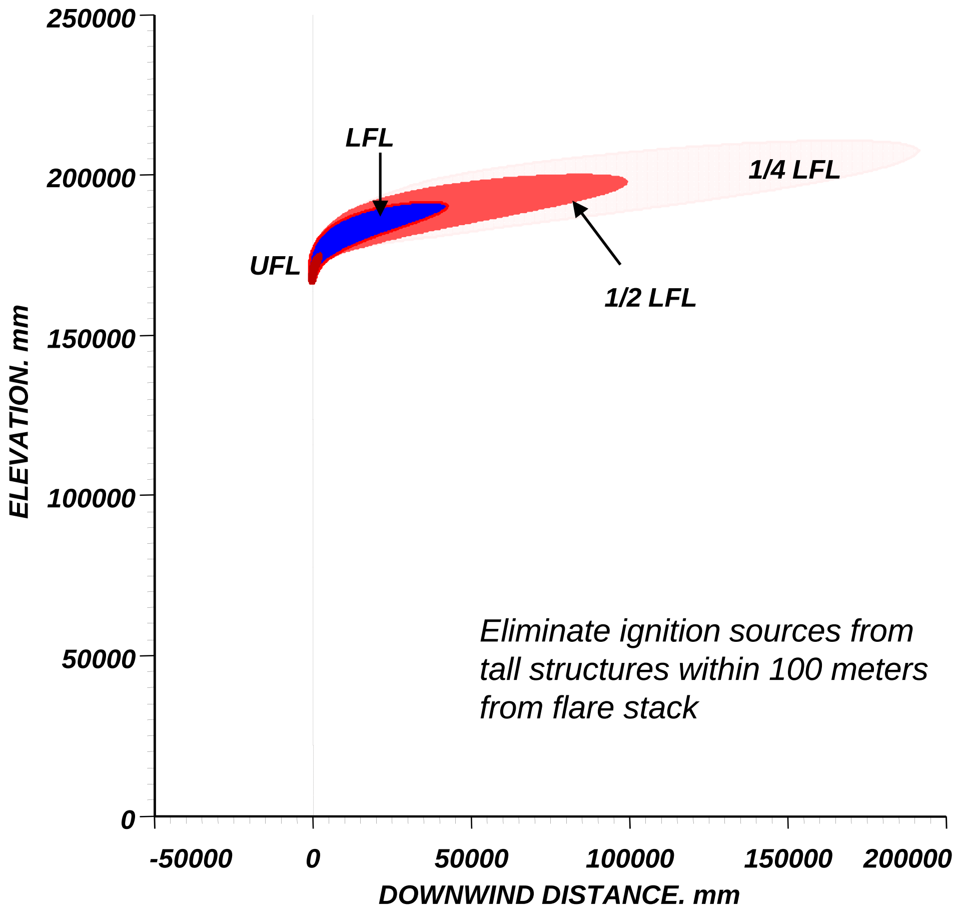
Flare Thermal Radiation – Vertical Profiles



Source: SuperChems Expert v5.91

ioMosaic

Flare Flammability Hazard Zones In Case of Flame Out

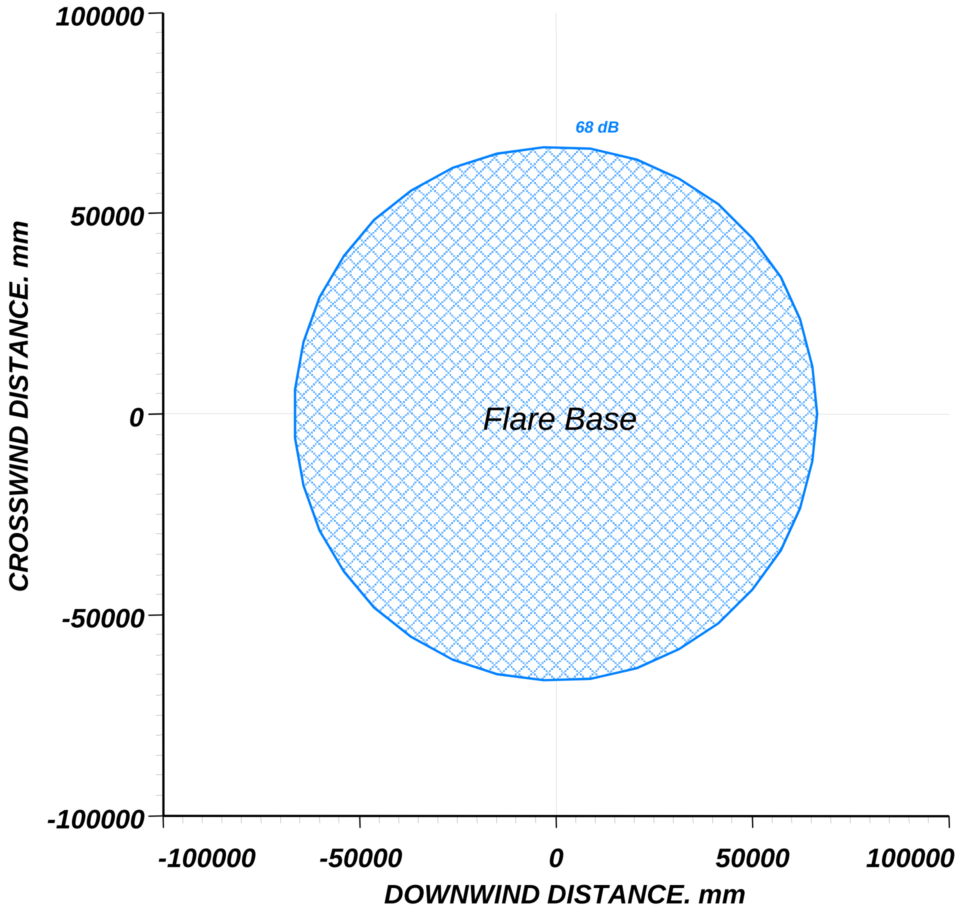


Eliminate ignition sources from tall structures within 100 meters from flare stack

Source: SuperChems Expert v5.91

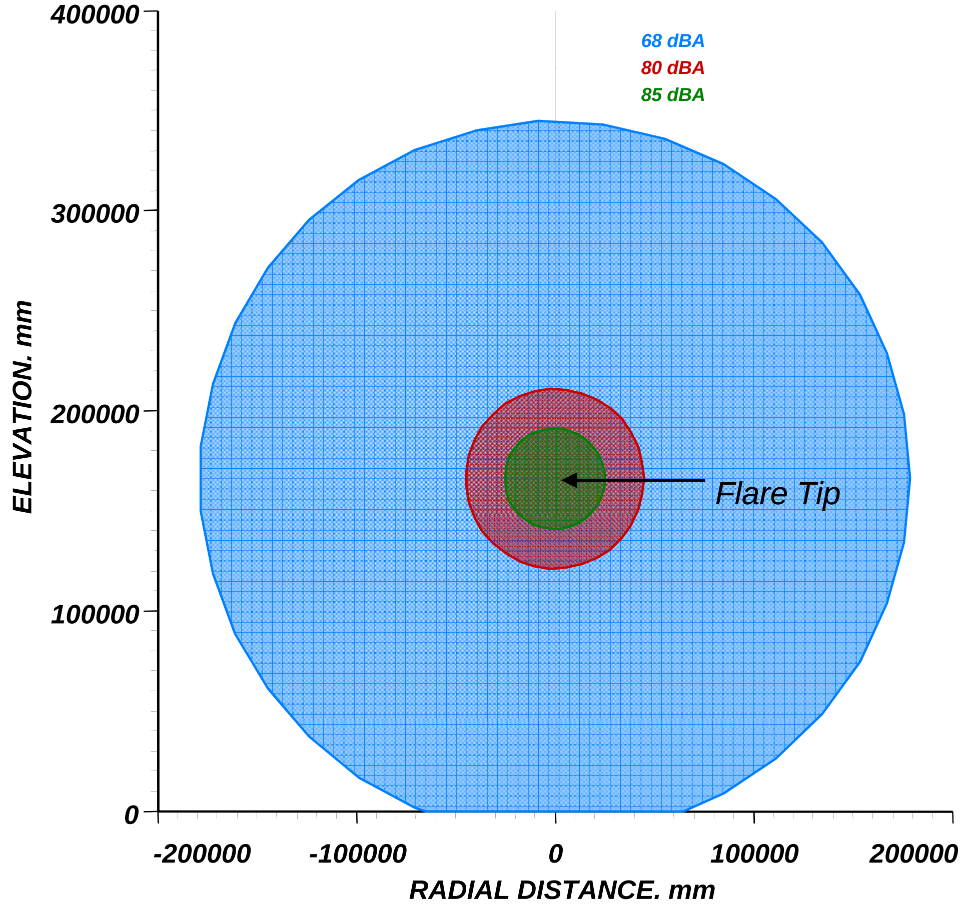
iomosaic

Flare Noise Estimates – Ground Level Impact



Source: SuperChems Expert v5.91

Flare Noise Estimates – Vertical Profiles



Source: SuperChems Expert v5.91

ioMosaic

The ioMosaic risk analysis approach utilizes pragmatic criteria to comply with recognized and generally accepted good engineering practices

- Analyzing flare header constraints typically involves addressing capacity limitations at each sub-header before combining into the overall system:
  - ✓ There are a manageable number of sub-header HIPPS to evaluate
  - ✓ Appropriate solutions can be devised utilizing experience without resorting to “random walk” techniques for determining HIPPS failure sequences
- The ability of the simulation model to calculate vessel dynamics (pressure and temperature conditions) as well as flow capacity is **critical** for an optimized and cost effective solution:
  - ✓ It determines the target frequency before and after the addition of HIPPS
  - ✓ It is the ultimate determinant (not backpressure) of whether a relief device is acceptable or not

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

General Notes on the Use of Control Valve Flow Coefficients

Centaurus Technology, Inc.

Prepared for Presentation
at
2009 Spring DIERS User Group Meeting
Orlando, FL
March 23, 24, & 25, 2009

Vendor Supplied Flow Relations:

Standard for Liquids

$$Q_{[gpm]} = C_v \sqrt{\frac{\Delta P_{[psi]}}{sg}}$$

Vendor Supplied Flow Relations:

Incompressible Gas Flow

$$Q\ [scfm] = 22.67\ C_v\ \sqrt{\frac{P_{[psia]}\ \Delta P_{[psi]}}{sg\ T_{Rankine}}}$$

Limitations not always stated, but Δ P/P should be less than 0.1

Vendor Supplied Flow Relations:

Compressible Gas Flow

Set

$$\Delta P = 1/2 \quad P_{[psia]}$$

In previous equation

This is not a very good approximation and leads to overestimating flow capacity.

Vendor Supplied Flow Relations:

Fisher / Emerson Universal Gas Sizing Relation:

$$Q\ \left[scfm\right] = \frac{\sqrt{520}}{60}\ \frac{C_g\ P}{\sqrt{sg\ T_R}}\ SIN\ \left\{\frac{59.64}{C_1}\ \sqrt{\frac{\Delta P}{P}}\right\}_{rad}$$

Unusual form, but very good if one has Cg and C1 factors.

Vendor Supplied Flow Relations:

Fisher / Emerson Universal Steam Sizing Relation:

$$w_{stm}\ [lb/hr] = \frac{C_s\ P}{1 + 0.00065\, T_{SH}}\ SIN\left[\frac{3417}{C_1}\ \sqrt{\frac{\Delta P}{P}}\right]_{\deg rees}$$

Also very good but same as previous relation with $C_s = C_g / 20$ and unit conversion factors

General Finding

All vendor data and Cv or Cg & C1 factors lead to the same result:

$$C_d \; A \left[in^2 \right] = 0.0262 \; C_v$$

This is an interesting and useful result

General Relation in C_v (only)

The previous result permits a general Cv flow formula

In SI units

$$w\ [kg/s] = \left[\frac{0.0262\ C_v}{1550_{[in^2/m^2]}}\right] G_{[kg/m^2 s]}$$

Can be used for any case. Even Fisher/Emerson lists consistent Cv factors that can be used in this relation

The Appropriate Mass Flux Relations Are:

Liquid Flow

$$G_{[kg/m^2 s]} = \left[2\,\rho_{[kg/m^3]}\, \Delta P_{[N/m^2]} \right]^{1/2}$$

Gas Flow

$$G_{[kg/m^2 s]} = \frac{F(k,\eta)\, P_{[N/m^2]}}{\sqrt{RT/Mw}} \qquad (all\ SI\ units)$$

Two Phase Flow

$$G = \left[\frac{(1 - x_o)}{G^2_{x_o = 0}} + \frac{x_o}{G^2_{x_o = 1.0}} \right]^{-1/2}$$

This is the Fauske

$$G_{(x=0)} = \frac{\eta_c \; L \; P \; Mw}{z \; RT \; \sqrt{CT}}$$

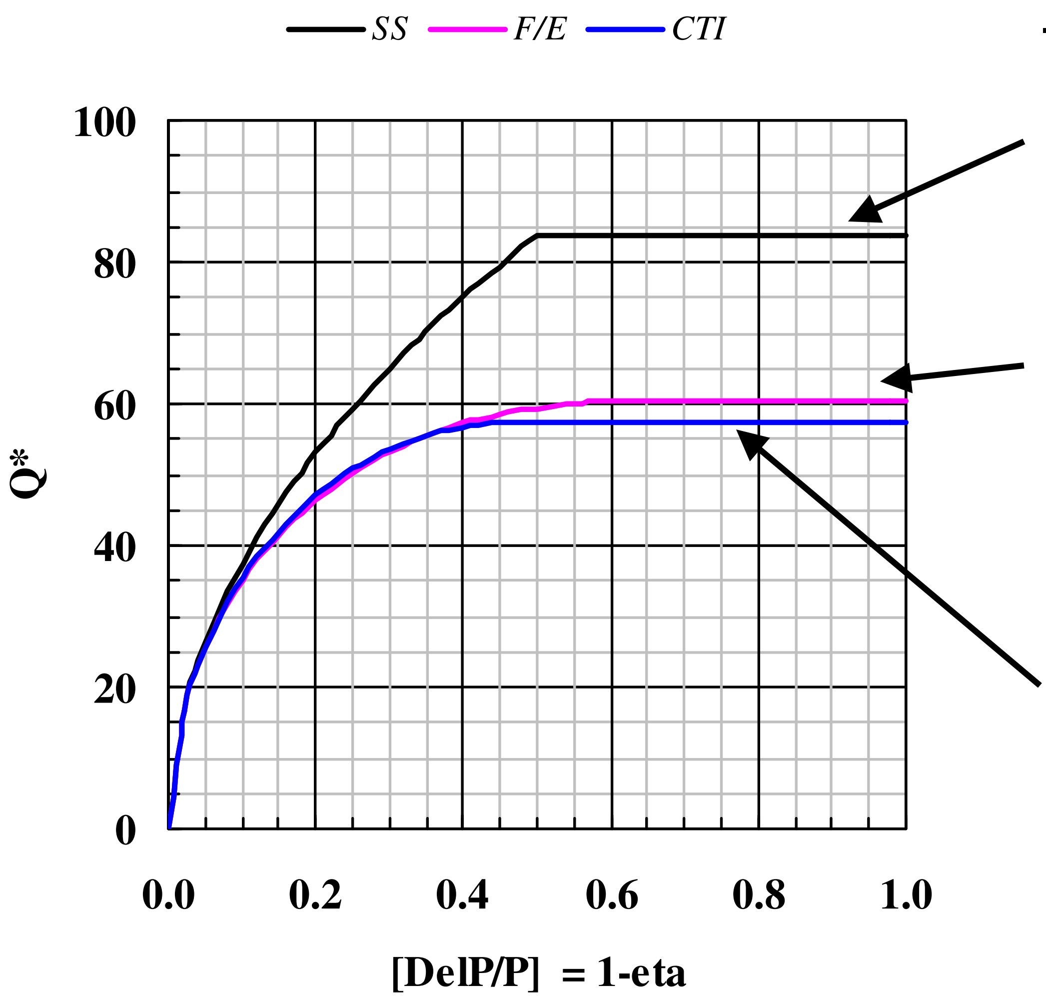
Universal two-phase flow relation

$$G_{[x=1]} = \frac{F(k,\eta) \; P_{[N/m^2]}}{\sqrt{RT / Mw}}$$

See Ref. 8 for details

Some Comparisons

Gas Flow



This is not-good subsonic approximation

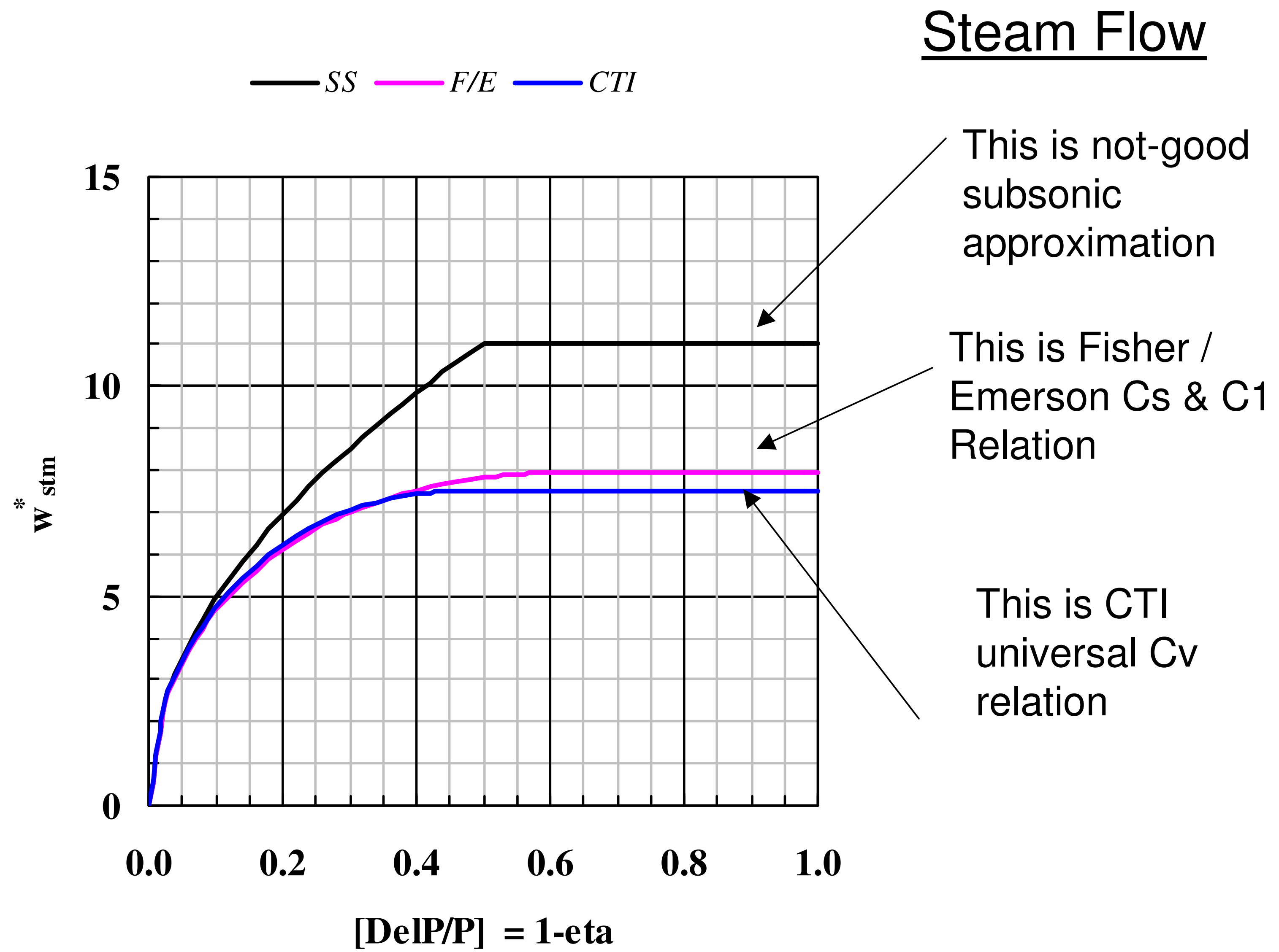
This is Fisher / Emerson Cg & C1 Relation

This is CTI universal Cv relation

Observation

- One can see by comparing the Fisher /Emerson equation and the CTI Cv gas flow relation that there is a good correspondence between the terms,
- $F(k,\eta)$ and SIN $[(59.64/C1)(1-\eta)]^{1/2}_{rad}$.
- The latter is a well-constructed approximation for the former
- See text for exact form of $F(k.\eta)$

Some Comparisons



Warning

- The fact that $C_d \cdot A_{[in2]} = 0.0262$ Cv for both liquid and gas flow is curious.
- This implies that Cd is the same for both liquid and compressible gas flow.
- We know this is not the case for good relief valves
- Further exploration suggests that most if not all gas flow ratings for flow control articles are based on tests in the incompressible flow regime

Summary

- We have a good universal method for providing a general formula for flow control articles using only Cv.
- We do not know enough regarding how Vendors rate flow control devices for gas in the sonic flow regime.
- The details are in the text document for the record
- We hope this is helpful and leads to further clarification
- Feedback would be appreciated.

Acknowledgement

- Thanks to Harold Fisher
- Harold has no responsibility for errors in this material
- Harold is not the Fisher of Fisher/Emerson Control Valves

<u>Technical Note CTI 09-525</u>

General Notes on the Use of Control Valve Flow Coefficients

Centaurus Technology, Inc.

Prepared for Presentation
at
2009 Spring DIERS User Group Meeting
Orlando, FL

March 23, 24, & 25, 2009

4590 Webb Road
Simpsonville, KY 40067
ctimag@att.net

General Notes on the Use
of
Control Valve Flow Coefficients

Centaurus Technology, Inc.

M. A. Grolmes

1.0 INTRODUCTION

The flow vs. pressure drop performance for various flow control devices (ball valves, control valves, pressure regulators, etc.), for which the actual flow area is not easy to specify, is determined by various vendor supplied flow coefficients; Cv – for liquid, Cg – for gas, and Cs - for steam. The objective of this technical note is to clarify the use of C_v, C_g and C_s factors and the various flow relations appropriate to vendor catalog tabulations of these items. We address two issues. The first is that some vendors list only C_v factors for various flow control articles, while others will list a C_v factor for liquid flow, and C_g factor for gas flow and a C_s factor for steam flow. So the question becomes; Can one reliably use a single C_v factor for both incompressible, and compressible fluids? The second, and related issue, is the identification of the appropriate flow relation for use in correctly evaluating the flow vs pressure drop performance for a given device. This is especially relevant for the treatment of compressible single and two phase flows. Without a clear understanding of vendor customs and certain application fundamentals, it is possible to improperly mix flow relations and flow coefficients and produce inaccurate results.

2.0 INCOMPRESSIBLE LIQUID FLOW

2.1 Standard Relation

We start with considerations of incompressible liquid flow. This is the least difficult and least confusing case. All vendors of flow control devices list C_v factors for liquid flow vs pressure drop performance calculations. It is noted that the first use of the flow coefficient referred to as "C_v" can be attributed to Masoneilan Valves in 1944, Ref. [1]. The concept evolved to standard usage. As a measure of flow capacity, $C_{v,}$ typically refers to the full or "wide open" flow capacity of control valves, regulators and similar devices. The C_v coefficient is also frequently given for ball, and 3-way valves. In many cases, Cv is the only device flow coefficient that has been verified by direct testing. The standard formula for liquid flow is:

$$Q_{[gpm]} = C_v \sqrt{\frac{\Delta P_{[psi]}}{sg}} \qquad (1)$$

Q is the liquid flow in [us gpm]: 1 usg = 3.785 L
C_v is the flow coefficient
ΔP is the pressure drop across the flow device (control valve, etc.) in [psi]

sg   is the specific gravity relative to water at 60°F or 15.55°C.

$$sg = \rho_l / \rho_{water,std.} \ where \ \rho_{water,std.} = 62.42 \ lb/ft^3 \ or \ 999.5 \ kg/m^3$$

Obviously C_v is a dimensional flow coefficient, representing the flow rate in usgpm for 1 psig pressure drop referenced to standard water. (Note this topic is a mess of mixed units which will be retained here as representative of standard US vendor catalog usage). Attempts to present flow relations in sufficient generality to address multiple unit conventions, such as can be found in Ref. [1], are often more confusing than helpful. In this note we will use both conventional US catalog units and in some cases SI units. In most instances, units will be indicated in [ ].

It may be assumed that most flow control articles have been calibrated at some low-to-moderate pressure drop with water. Even so, C_v factors are rarely listed to more that two significant figures.

2.2 Relation of C_v to Flow Area

The obvious utility of the C_v factor is that flow control devices do not always have a readily identifiable flow area, or even if this were the case, the internal geometry is such that the flow loss, or discharge coefficient is similarly difficult to identify.

However, it will be instructive to consider an equivalent flow relation based upon an assumed knowledge of the effective device full-flow area, which will be referred to as the product C_d A. For this we use the relation

$$w\left[kg/s\right] = C_d \ A\left[m^2\right] \ G\left[kg \ m^{-2} \ s^{-1}\right] \tag{2}$$

where

w   is the liquid mass flow rate in [kg/s]
C_dA   is the effective device flow area in [m^2]
G   is the liquid mass flux in [kg $m^{-2}s^{-1}$]

The above relation is given in consistent SI units. The liquid mass flux relation is

$$G = \sqrt{2 \ \rho \ \Delta P} \tag{3}$$

where ρ is the liquid density in [kg/m^3] and ΔP is the pressure drop in [N/m^2].

As an intermediate step one can divide by the density ρ to obtain Equation (4a).

$$Q\left[m^3/s\right] = C_d A\left[m^2\right] \sqrt{\frac{2\,\Delta P\left[N/m^2\right]}{\rho\left[kg/m^3\right]}} \tag{4a}$$

Then with unit conversion factors:

1	m^3/s	=	15850	usgpm
1	m^2	=	1550	in^2
1	psi	=	6894.73	N/m^2

And the replacement with the 'sg' factor

$$\rho = \rho_{water\,,\,std}\;\; sg \tag{4b}$$

where

$$\rho_{water,std} = 999.5\; kg/m^3$$

one can combine Equations (4a) and (4b) to yield the relation,

$$Q\left[gpm\right] = 37.98\; C_d\; A\left[in^2\right]\sqrt{\frac{\Delta P\left[psi\right]}{sg}} \tag{5}$$

The important result to be obtained by comparing Equations (1) and (5) is

$$C_d A\;\left[in^2\right] = 0.0263\;\; C_v \tag{6}$$

Equation (6) provides a means of estimating the effective flow area for a liquid flow control device, based on the specified C_v factor.

We will extend the above comparison to gas flow relations.

3.0 GAS FLOW RELATIONS

3.1 Incompressible Gas Flow

Incompressible gas flow is similar to the incompressible liquid flow case considered in Section 2.2. Again if one assumes knowledge of an effective flow area, C_dA, then one can write

$$w\;\left[kg/s\right] = C_d\;A\left[m^2\right]\sqrt{2\;\rho_{\left[kg/m^3\right]}\;\Delta P_{\left[N/m^2\right]}} \tag{7}$$

For gas flow, additional relations required are,

$$\rho = \frac{P\,Mw}{R\,T} \tag{8}$$

where in SI units

ρ is the gas density in [kg/m^3]
P is the pressure in [N/m^2]
Mw is the molecular weight of the gas [kg/kg mol]
R is the gas constant, [8314.47 J/kg mol K]
T is the absolute temperature in [K]

One also uses,

$$\rho_{std} = sg \;\; \rho_{air,std} \tag{9}$$

and

$$sg = Mw_g \,/\, Mw_{air} \tag{10}$$

where Mw_{air} = 28.84

Now, the standard air density is evaluated from Equation (8) at P = 101325 N/m^2 (14.696 psia) and 60^{o}F or 15.55^{o}C or 288.7 K, is

$$\rho_{air,std} = \frac{101325 \;\; 28.84}{8314.47 \;\; 288.7} = 1.217 \;\; kg/m^3 \tag{11}$$

or

$0.076 \;\; lb/ft^3$.

On dividing Equation (7) by ρ_{std}, and using Equation (8), one obtains

$$Q\left[s\, m^3/s\right] = \frac{C_d\, A}{\rho_{std}} \sqrt{\frac{2\, P\, \Delta P \;\; Mw_g}{R \;\; T_K}} \tag{12}$$

Then using Equations (9), (10) and (11) one obtains

$$Q\;\left[scfm\right] = 865.46 \;\; C_d \;\; A_{\left[in^2\right]} \sqrt{\frac{P\,\Delta P}{sg \;\; T_R}} \tag{13}$$

The above Equation (13) gives the volume flow rate in [scfm] of a gas with Mw = Mw_g for an effective flow area C_d A [in^2]. Other terms are

P is the absolute upstream pressure in [psia]
ΔP is the flow control device pressure drop in [psi]
sg is the specific gravity relative to air at standard conditions, sg = Mw_g/28.84

T_R is the absolute upstream gas temperature in [R], $T_R = 1.8\ T_K$.

The above Equation (13) is valid only for incompressible gas flow.

A common formula given in Refs. [1 through 4] and other places, for gas flow is

$$Q\ [scfm] = 22.67\ C_v \sqrt{\frac{P\ \Delta P}{sg\ T_R}} \tag{14}$$

(Note that one often finds Equation (14) with Q given in [scfh] with the constant term 1360. The Equation (14) constant, 22.67 = 1360/60.)

As before, with the liquid flow case, one can compare Equations (13) and (14) and establish

$$C_d A\ \left[in^2\right] = 0.0262\ C_v \tag{15}$$

Equation (15) for incompressible gas, is identical with Equation (6) for liquid. This author has never seen a discussion of the source of the constant 22.67 (or 1360) in Equation (14). The value could arise from direct gas flow calibration, or from an assumption that the underlying device flow area should be the same for either liquid or gas flow. The results of Equations (6) and (15) show that in practice, the latter is confirmed. The former is also likely true. (See Section 5.)

3.2 Compressible Gas Flow

Many flow control device catalogs suggest Equation (14) may be applied to compressible gas flow up to the point where $\Delta P \geq \frac{1}{2}\ P$. After which, the term ΔP is set equal to ½ P.

We recast this discussion by defining the pressure ratio

$$\eta = P_1 / P \tag{16}$$

where

P is the absolute upstream pressure, [psia]
P_1 is the absolute downstream pressure, [psia]

Then one can define

$$\Delta P = P - P_1 = P\ (1 - \eta) \tag{17}$$

Using Equation (17) in Equation (14) one obtains

$$Q_{[scfm]} = 22.67 \; C_v \; (1-\eta)^{1/2} \; \frac{P_{[psia]}}{\sqrt{sg \, T_R}} \tag{18}$$

In Equation (18) subsonic gas flow is assumed to be represented by $0 \le \eta \le 0.5$, with sonic flow defined by $\eta = 0.5$ for any value of $\Delta P > ½ \, P$.

For sonic flow conditions, ($\eta = 0.5$), this would lead to an equation of the form

$$Q \; [scfm] = 16.03 \; C_v \; \frac{P}{\sqrt{sg \;\; T_R}} \tag{19}$$

Ref. [5] , (internet site *Engineering Toolbox*), provides a similar relation for sonic flow as

$$Q \; [scfm] = 11 \; C_v \; \frac{P}{\sqrt{sg \;\; T_R}} \tag{20}$$

We will compare these two forms later.

Exact Formulation – If one again assumes knowledge of an effective device flow area, C_dA, then using conventional compressible gas flow relations, and appropriate unit conversion factors, one can establish an "exact" compressible flow relation as,

$$Q \; [scfm] = 612 \; C_d \;\; A\left[in^2\right] \; \frac{F\,(k,\eta) \;\; P_{[psia]}}{\sqrt{sg \;\; T_R}} \tag{21}$$

In Equation (21) F (k, η) provides the exact treatment for both subsonic, and sonic flow.

For subsonic flow;

$$\eta = P_1 / P \ge \eta_c \tag{22a}$$

where

$$\eta_c = \left[\frac{2}{k+1}\right]^{k/(k-1)} \tag{22b}$$

and

$$F\ (k,\eta)\ =\ \left[\frac{2k}{(k-1)}\ \left(1-\eta^{(k-1)/k}\right)\ \eta^{2/k}\right]^{1/2} \tag{22c}$$

Then for sonic flow, where one would find

$$\eta\ =\ P_1 / P\ \leq\ \eta_c \tag{23a}$$

The term F (k, η) is given by;

$$F\ (k\,,\eta) = F\ (k) = \left\{k\ \left[\frac{2}{k+1}\right]^{(k+1)/(k-1)}\right\}^{1/2} \tag{23b}$$

In the above Equations (23a) and (23b),

k is the gas specific heat ratio
η_c is the critical pressure ratio

Table 1, provides some typical values.

Table 1 Select values of k, η_c and F(k)for common gasses.

Specie	**Mw**	**sg**	**k**	**η_c**	**F(k)**
Air	28.84	1.00	1.400	0.528	0.685
N_2	28.02	0.97	1.400	0.528	0.685
O_2	32.0	1.11	1.400	0.528	0.685
CO_2	44.01	1.526	1.300	0.546	0.667
Iso butane	58.12	2.015	1.110	0.583	0.630
H_2O	18	0.624	1.31	0.544	0.669

Example: Consider as an example, either Air, N_2, O_2, all diatomic gasses for which F (k) = 0.685. With this value, Equation (21) for sonic flow becomes

$$Q\left[scfm\right]\ =\ 419.2\ \ C_d\ \ A\left[in^2\right]\ \frac{P\left[psia\right]}{\sqrt{sg\ \ T_R}} \tag{24}$$

We now compare Equation (24) with previous Equations (19) and (20) for sonic flow, and find the following.

Comparing Equations (24) and (19) one obtains

$$C_d\ \ A\left[in^2\right]\ =\ 0.0382\ \ C_v \tag{25}$$

Comparing Equation (24) and (20) one obtains

$$C_d \;\; A \left[in^2 \right] = 0.0262 \;\; C_v \tag{26}$$

The result in Equation (26) is again identical with previously established relations between C_d A [in^2] and C_v and suggests the Ref. [5] sourced Equation (20) is the more accurate of the simplified approximate relations for sonic flow. (There are of course other fundamental reasons to expect that Equation (19) would overestimate sonic flow by the approximation given.)

3.3 Fisher/Emerson Relations

To this point, we have considered flow control device relations for gas flow utilizing only the vendor supplied C_v factor. By and large, satisfactory results may be obtained using the C_v factor if one employs Equations (24) and (26). However, the arbitrary numerical constant 22.67, in the approximate relation of Equation (14), along with certain other considerations led Fisher/Emerson (F/E) to develop a different relation for gas flow, Refs. [4,5].

In 1963 Fisher Controls introduced a relation which they referred to as their "Universal Gas Sizing Equation", given below as:

$$Q \; [scfm] = \frac{\sqrt{520}}{60} \; \frac{C_g \; P}{\sqrt{sg \; T_R}} \; SIN \left\{ \frac{59.64}{C_1} \sqrt{\frac{\Delta P}{P}} \right\}_{rad} \tag{27a}$$

Equation (27a) addresses two issues perceived to be significant by Fisher Controls. The first is the introduction of C_g a gas flow coefficient different from C_v. In this way the numerical coefficient $\sqrt{520}/60 = 0.381$ is related to defined numerical values for, 520 R, the standard 60°F gas temperature, and the unit conversion 60 min/hr. The second issue of better representation of gas compressibility is achieved by the SIN ( ) term. This term accounts for the compressibility of the gas throughout the subsonic range. This will be illustrated shortly. The additional constant C_1 is defined by Fisher as the ratio of C_g to C_v, that is, $C_1 = C_g/C_v$. Values of C_1 are such that $16 \le C_1 \le 37$. The Fisher Catalog, Vol. 10, Ref. [6], provides a complete tabulation of C_v, C_g, C_1 and C_s for all Fisher flow control devices.

If we proceed further by using Equation (17), one finds that Equation (27a) takes the form

$$Q \left[scfm \right] = 0.3801 \; \frac{C_g \; P}{\sqrt{sg \;\; T_R}} \; SIN \left[\frac{59.64}{C_1} \sqrt{1-\eta} \right]_{rad} \tag{27b}$$

where again $\eta = P_1/P$. The argument of the SIN term in [ ] is to be taken as radians. If the SIN function is to be evaluated in degrees then one must multiply the argument by $180/\pi$ and obtain

$$Q\left[scfm\right] = 0.3801 \ \frac{C_g \ P}{\sqrt{sg \ T_R}} \ SIN\left[\frac{3417}{C_1}\sqrt{1-\eta}\right]_{\deg rees} \tag{27c}$$

To consider an alternative form of an exact compressible flow formulation, let us combine Equations (21) and (26) to obtain

$$Q \ \left[scfm\right] = 16.03 \ \frac{C_v \ P}{\sqrt{sg \ T_R}} \ F\left(k,\eta\right) \tag{28}$$

One can see by comparing Equations (28) and (27b) that there is a correspondence between the terms, F (k,η) and SIN$[(59.64/C_1)\ \sqrt{1-\eta}\ ]_{rad}$. Now recall the incompressible relation for gas flow as first given in Equation (14) and restated here as

$$Q \ \left[scfm\right] = 22.67 \ \frac{C_v \ P}{\sqrt{sg \ T_R}} \ \left(1-\eta\right)^{1/2} \tag{29}$$

Comparison – We now have three possible relations for gas flow through a control device. Let us select a set of parameters from Ref. [6 – Cat 10 – pg 1-134.8.3], for a V150, 1 inch ball valve, at full open conditions.

The parameters are:

C_v	=	5.23	C_s	=	7.95
C_g	=	159	C_1	=	30.4

So as to illustrate the compressible nature of the three relations we further define a Q^* term as

$$Q^* = \frac{Q_{[scfm]} \ \sqrt{sg \ T_R}}{P_{[psia]}} \tag{30}$$

We now summarize the forms of the Q^* term as given in Table 2.

Table 2 Q^* term for gas flow.

Source Equation	Name	Function
(29)	Subsonic, Ref. [1] (SS)	$Q^* = 22.67\ C_v\ \sqrt{(1-\eta)}$
(28)	Exact, This work (CTI)	$Q^* = 16.03\ C_v\ F\ (k,\eta)$
(27b)	Fisher/Emerson Universal Ref. [4] (F/E)	$Q^* = 0.3801\ C_g\ \mathrm{SIN}\left[\frac{59.64}{C_1}\ \sqrt{(1-\eta)}\right]$

Figure 1 shows results obtained with the above equations (27b), (28) and (29), as a function of the pressure ratio term $(1 - \eta) = \Delta P/P$. Note the close agreement between the (CTI) and (F/E) solution forms.

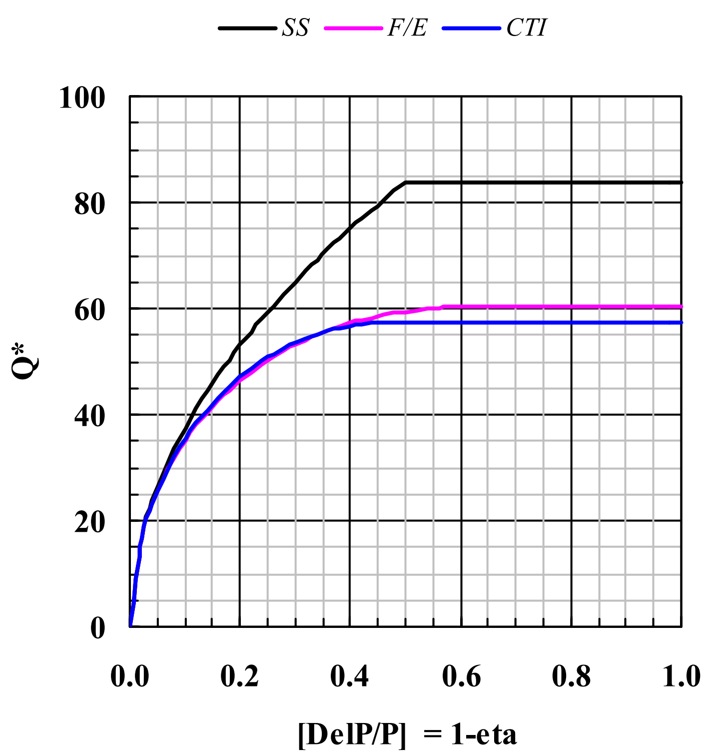


Fig. 1 Generalized gas flow relations for control valves, (per Table 1).

For sonic or choked flow conditions, the rules for each of the various equations are given in Table 3.

Table 3 Rules for sonic conditions – gas flow.

Source Equation	**Name**	**Sonic Condition**	**Choked Flow Limiting Condition**
(29)	SS	$\eta = 0.5$	$Q^* = 16.03\ C_v$
(28)	CTI	$\eta = \eta_c$	$Q^* = 10.98\ C_v$
(27b)	F/E	SIN [ ] = 1.0	$Q^* = 0.3801\ C_g$

The example for Equation (28) is based on air for which k = 1.4 and $\eta_c = 0.528$, and F (k, η_c) = 0.6847. One should also note that for the example illustration with Equation (27b) $C_g = C_1 \times C_v$ or $C_g = 30.4\ C_v$, so that for Equation (27b) in Table 2, it would also be the case that $Q^* = 11.55\ C_v$ being equivalent to the indicated 0.3801 C_g.

Now, recall Fig. 1 and observe that results using Equations (28) and (27b) are virtually identical. Equation (28) reaches the sonic or choked flow condition at ΔP/P = 0.472. Equation (27b) reaches the same condition at SIN [ ] = 1.0, or ΔP/P ≈ 0.41.

The Fisher/Emerson literature would suggest that this difference better reflects device specific pressure recovery captured by the additional C_1 parameter. It is also evident that the SIN [ ] function does an excellent job of capturing the exact F (k,η) behavior in the subsonic ΔP/P range. Further, in Fig. 1, it is also apparent that extension of the incompressible gas flow Equation (29) into the compressible range, over predicts the device flow. One could observe that Equation (29) over predicts flow for any ΔP/P > 0.1.

4.0 STEAM FLOW RELATIONS

Flow control devices are ubiquitous in utility power plants where steam is the working fluid for electricity generation and process heat. While steam is in reality just another gas, general practice has resulted in special formulae for steam flow in control devices.

4.1 Incompressible Steam Flow

Ref. [2] also gives the following relation for steam flow.

$$w_{stm}\ [lb/hr] = 63\ C_v \sqrt{\frac{\Delta P_{[psi]}}{v_{[ft^3/lb]}}} \tag{30}$$

For Equation [39];

w_{stm} is the steam flow rate in [lb/hr]

C_v is the same flow coefficient used for liquid and gas when the vendor provides only one value – under the label "C_v"

ΔP is the flow control device pressure drop in [psig]

v is the steam specific volume at the upstream absolute pressure P[psia], with units [ft^3/lb].

As before, when ΔP exceeds P/2, ΔP is taken as P/2. We will illustrate that similar to the counterpart of Equation (30) for gas, Equation [30] is accurate only for the subsonic flow case with $\Delta P/P \leq 0.1$. For larger ΔP up to P/2, Equation (30) will over estimate the device flow.

For further use, we convert Equation (30) to a more convenient form by employing the previously used relation

$$\Delta P \;=\; P\left(1-\eta\right) \tag{31}$$

and

$$v\;\left[\,ft^{3}/lb\,\right] \;=\; \frac{447.8}{P_{[psia]}} \tag{32}$$

Equation (32) provides an acceptably accurate correlation of the specific volume for saturated steam as a function of absolute pressure. With Equations (31) and (32), Equation (29) becomes

$$w_{stm}\;\;[lb/hr] \;=\; 2.98\;\;C_v\;\;P\;\sqrt{1-\eta} \tag{33}$$

With η = 0.5, the sonic, or choked flow form of Equation (33) is

$$w_{stm}\;\;[lb/hr] \;=\; 2.1\;\;C_v\;P \tag{34}$$

4.2 Compressible Steam Flow Relations

For Fisher/Emerson products Ref. [3] provides the following relation

$$w_{stm}\;\;[lb/hr] \;=\; \frac{C_s\;P}{1+0.00065\,T_{SH}}\;\;SIN\left[\frac{3417}{C_1}\;\sqrt{\frac{\Delta P}{P}}\right]_{\deg rees} \tag{35}$$

The term in the denominator on the RHS of Equation (35), ($0.00065\ T_{SH}$), is related to the degrees of superheat in ^{o}F, and can be safely ignored in all but cases of exceptionally large superheat. Now, with previously defined substitutions, Equation (35) can be written.

$$w_{stm}\;\;[lb/hr] \;=\; C_s\;P\;\;SIN\left[\frac{3417}{C_1}\;\;\sqrt{1-\eta}\right]_{\deg rees} \tag{36}$$

Note that Fisher/Emerson arbitrarily define $C_s = C_g/20$, so that C_s is not a new or independent flow coefficient. (Note this is consistent with physical principles.) Choked flow conditions are again defined as SIN [ ] = 1.0, so that Equation (36) for choked flow becomes

$$w_{stm}\ [lb/hr] = C_s\ P_{[psia]} = \frac{C_g}{20}\ P_{[psia]} \tag{37}$$

Recall that for Fisher/Emerson products $C_g \approx 30\ C_v$, and the leading flow coefficient in Equation (37) is ≈ 1.5 C_v. (Compare this observation with Equation (34).)

As has been done previously, an "exact" flow relation of the form

$$w_{stm}\ \left[kg/s\right] = C_d\ A_{\left[m^2\right]} \frac{F(k,\eta)\ P_{\left[N/m^2\right]}}{\sqrt{\frac{8314}{18}\ T_R}} \tag{38}$$

Again with unit conversion factors used previously Equation (38) becomes

$$w_{stm}\ \left[lb/hr\right] = 2203.8\ C_d\ A_{\left[in^2\right]}\ F(k,\eta)\ \frac{P_{[psia]}}{\sqrt{T_R}} \tag{39}$$

Further simplification can be obtained by (a) noting that for steam, $\sqrt{T_R} \approx 27$, and (b) assuming that the relation $C_d\ A_{\left[in^2\right]} = 0.0262\ C_v$, also holds for the steam case. These considerations lead to

$$w_{stm}\ \left[lb/hr\right] = 2.1385\ C_v\ F_{(k,\eta)}\ P_{[psia]} \tag{40}$$

The sonic, or choked flow case for steam is defined by; k = 1.31, and $F_{(k,\eta)} = F_{(k)} = 0.685$. Therefore, for sonic conditions Equation (40) becomes

$$w_{stm}\ \left[lb/hr\right] = 1.46\ C_v\ P_{[psia]} \tag{41}$$

Now, compare this result again with Equation (34), and also with the discussion under Equation (37).

<u>Comparison</u> - Using the same set of example flow coefficients:

C_v	=	5.23	C_s	=	7.95
C_g	=	159	C_1	=	30.4

and restating Equations (33), (36) and (40) in the form of $w_{stm}/P_{[psia]}$ and referring to this term as w^*_{stm} we have forms given in Table 4.

Table 4 Reduced w^*_{stm} terms for steam flow.

Source Equation	Name	Function
(33)	Subsonic (SS)	$w^*_{stm} = 2.98\ C_v\ \sqrt{1-\eta}$
(40)	Exact (CTI)	$w^*_{stm} = 2.139\ C_v\ F(k,\eta)$
(36)	Fisher/Emerson (F/E)	$w^*_{stm} = C_s\ SIN\left[\frac{3417}{C_1}\ \sqrt{1-\eta}\right]$

Figure 3 shows results obtained with the above equations as a function of the pressure ratio term (1 – η) = ΔP/P. For sonic or choked flow conditions the rules for the various equations are given in Table5.

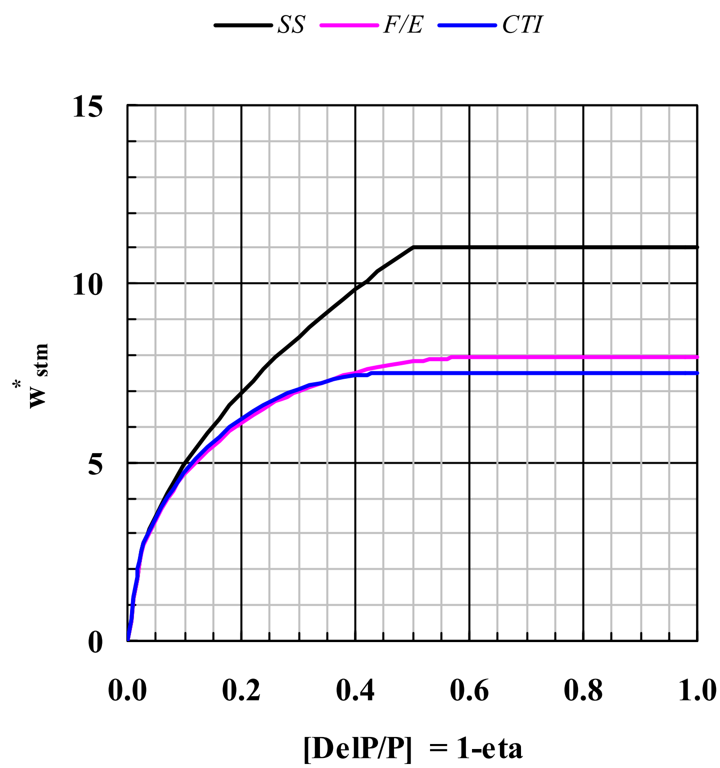


Fig. 3 Generalized steam flow relations for control valves, (per Table 3).

Table 5 Rules for choked steam flow.

Source Equation	Name	Choked Flow Limiting Conditions
(33)	SS	$w^*_{stm} = 2.1\ C_v$
(40)	(CTI)	$w^*_{stm} = 1.46\ C_v$
(36)	(F/E)	$w^*_{stm} = C_s = 0.05\ C_g \approx 1.5 C_v$

Figure 3 shows the same qualitative results as Fig. 2. The subsonic steam flow relation is not accurate beyond ΔP/P ≈ 0.1. The "CTI" and the "F/E" relations are in very close agreement, and may be used interchangeably.

5.0 THE RELATION $C_d\ A\left[in^2\right] = 0.0262\ C_v$

Thus far it has been shown that most good US vendor data and flow vs. pressure drop relations for single phase incompressible, and compressible flow lead to the same relation between the flow coefficient C_v and the effective full flow area in [in^2] given by

$$C_d\ A\left[in^2\right] = 0.0262\ C_v \tag{42}$$

This is surely a useful relation and it has been shown that it can be used reliably with only the C_v factor, even for Fisher/Emerson products.

There is one troublesome unresolved detail that is uncovered by this finding. Consider is safe to assume that the full flow area for any device characterized by a C_v factor is the same regardless of whether the flow is incompressible or compressible. The finding summarized by the result expressed by Equation (42) would suggest that the effective discharge coefficient is the same for both compressible and incompressible flow. This implication is suspicious.

One knows from relief valve literature that the relief valve device discharge coefficient is different for the same relief valve in compressible versus incompressible flow. For a well configured relief valve a factor $C_d = 0.95$ would be recommended for sonic gas flow and a factor $C_d = 0.9$ might be recommended for incompressible liquid flow. For sonic gas flow, choking at the nozzle minimum area eliminates the flow resistance of the remainder of the valve body flow path. One might be inclined to expect similar considerations for control valves and other flow regulating devices.

This concern is further developed by noting that US flow testing standards for control valves such as ANSI/NFPA T3.21.3-1990 permit measuring the C_v factor for gas flow at low pressure drops (not greater than 2 psi), thus assuring incompressible flow. Under this testing protocol, the result expressed in Equation (42) is not surprising. Ref. [7] points out the Japanese test standard as one exception which calls for gas flow testing at sonic conditions. Ref. [7] notes that different results are obtained.

For now, all one has is vendor data leading to the results described in Sections 1 through 4 of this document. Until one has vendor specific data for compressible flow performance, one can only proceed with the results obtained thus far. However, it would be prudent to anticipate somewhat higher (of the order of 10%) flow for sonic compressible flow. This point however, remains as an issue of uncertainty.

6.0 FLASHING TWO-PHASE FLOW

No vendor literature treats two-phase flashing flow either at all, or with appropriate consideration of modern techniques. The extension to two-phase flashing flow can be accomplished directly from material presented in this document. We will assume that Equation (42) is applicable and on knowing the effective flow area all relevant two-phase flow relations are applicable. Recall Equation (2), as restated below;

$$w\left[kg/s\right] = C_d \;\; A\left[m^2\right] \; G\left[kg \;\; m^{-2}\; s^{-1}\right] \tag{43}$$

Since there are no precedent conventions, the use of SI units will be retained. An expression for the mass flux for two-phase flashing flow from a saturated liquid source condition is given by

$$G = \frac{\eta_c \; L \; P \; Mw}{z \; RT \; \sqrt{CT}} \tag{44}$$

In Equation (43),

η_c	is the critical pressure ratio for two-phase flashing flow [-]
L	is the latent heat in [J/kg]
P	is the upstream absolute pressure in [N/m^2]
Mw	is the molecular weight [kg/kg mol]
z	is a compressibility factor on the saturation line [-]
R	is the gas constant [8314 J/kg mol K]
T	is the upstream absolute temperature [K]
C	is the liquid heat capacity [J/kg K]

One may assume with the expectation of +/- 10% accuracy that $\eta_c/z \approx 0.9$.

Now if one invokes the previously established relation $C_d \;\; A_{[in^2]} = 0.0262 \; C_v$ the direct counter part is $C_d \;\; A\left[m^2\right] = \dfrac{0.0262}{1550} \; C_v \;=\; 1.69 \, x \, 10^{-5} \; C_v$.

Now combining terms into Equation (43) one obtains

$$w_{[kg/s]} \; = \; 1.69 \; x \, 10^{-5} \; \frac{C_v \; L \; P \; Mw}{R T \; \sqrt{C \, T}} \tag{45}$$

The above result provides the necessary relation for using a C_v factor to determine a flashing two-phase flow with saturated liquid source conditions.

Now, two-phase flow through a control valve can have any of the following characteristics:

(a) Flashing flow with saturated liquid source conditions.
(b) Flashing flow with inlet sub cooling.
(c) Flashing flow with net steam quality at the source.
(d) Geometry imposed non-equilibrium two-phase flashing flow.
(e) Two component gas and liquid nonflashing flow.

Given the difficulty of making the correct selection from among the possibilities listed above, plus the uncertainty in discharge coefficient discussed in Section 5, the general purpose two-phase sonic flow correlation method of Fauske, Ref. [8] would be recommended and is ideally suited for rapid calculations with . In this correlation the mass flux term G for use in Equation (43) would be,

$$G = \left[\frac{\left(1 - x_o\right)}{G^2_{x_o = 0}} \; + \; \frac{x_o}{G^2_{x_o = 1.0}} \right]^{-1/2} \tag{45}$$

For cases (a) and (c) above, $G_{x=0}$ would be given by Equation (44) and $G_{x=1.0}$ would be given by

$$G_{x=1.0} \; = \; \frac{F\left(k\right) \; P}{\sqrt{RT / Mw}} \tag{46}$$

For other cases, see Ref. [8].

7.0 REFERENCES

1. Masoneilan Control Valve Sizing Handbook, Masoneilan Bulletin OZ 1000, Masoneilan-Dresser Valve Division.

2. Badger Meter, Inc. Research Control Valves, Product Bulletin 940493–4/94–10M. (Also see other Badger Control Valve Catalogs).

3. Selection, Sizing, and Operation of Control Valves for Gasses and Liquids, Fishers Controls Training Lecture No. 6110, Fisher Controls Int. 205 South Center St., Marshalltown, Iowa 50158.

4. Valve Sizing & Selection Technical Reference, Warren Controls Bulletin VSSTR-9105.

5. Eng. Tool Box – Internet Site, Google "Engineering Tool Box" Search for Control Valve Flow.

6. Fisher Catalog 10, Control Valve Spec. Sheets with Flow Coefficients.

7. Fleishner, H., Forster, K, and Franson, D. C., "Clearing the air on pneumatic valve ratings". Machine Design, Oct. 5, 2000.

8. Fauske, Hans K., "Determine Two-Phase Flows During Releases", Chemical Engineering Progress, February, 1999.

SOME SAFETY RELIEF VALVE PERFORMANCE FINDINGS FROM THE BP INCIDENT INVESTIGATION

Presentation to the DIERS Users Group Meeting

by

HAROLD G. FISHER

FisherInc

LAS VEGAS, NV

APRIL 28, 2008

U.S. CHEMICAL SAFETY AND HAZARD INVESTIGATION BOARD

INVESTIGATION REPORT

ISP and Subject Matter Expert Review

REFINERY EXPLOSION AND FIRE

(15 Killed, 180 Injured)

BP

TEXAS CITY, TEXAS

MARCH 23, 2005

KEY ISSUES:

- SAFETY CULTURE
- REGULATORY OVERSIGHT
- PROCESS SAFETY METRICS
- HUMAN FACTORS

REPORT NO. 2005-04-I-TX

MARCH 2007

FAI / 05-95

AN EVALUATION of the EXPLOSION and FIRE on MARCH 23, 2005 at the BP PRODUCTS NORTH AMERICA, INC. REFINERY in TEXAS CITY, TEXAS (Emergency Relief System Evaluation)

Submitted To:

U.S. Chemical Safety and Hazard Investigation Board

Washington, DC 20037

Prepared By:

Fauske & Associates, LLC

16W070 West 83rd Street

Burr Ridge, Illinois 60527

TEL: (630) 323-8750

FAX: (630) 986-5481

February 2007

PROCESS / EQUIPMENT DESCRIPTION

RAFFINATE SPLITTER TOWER

THE RAFFINATE SPLITTER TOWER IS A VERTICAL ASME DISTILLATION COLUMN WITH 70 TRAYS, A MAWP OF 40 PSIG, AN INSIDE DIAMETER OF 12.5 FEET, AND AN OVERALL HEIGHT OF 170.75 FEET INCLUDING TWO ASME 2:1 ELLIPTICAL HEADS. THE APPROXIMATE LIQUID-FULL VOLUME IS 154,845 GALLONS.

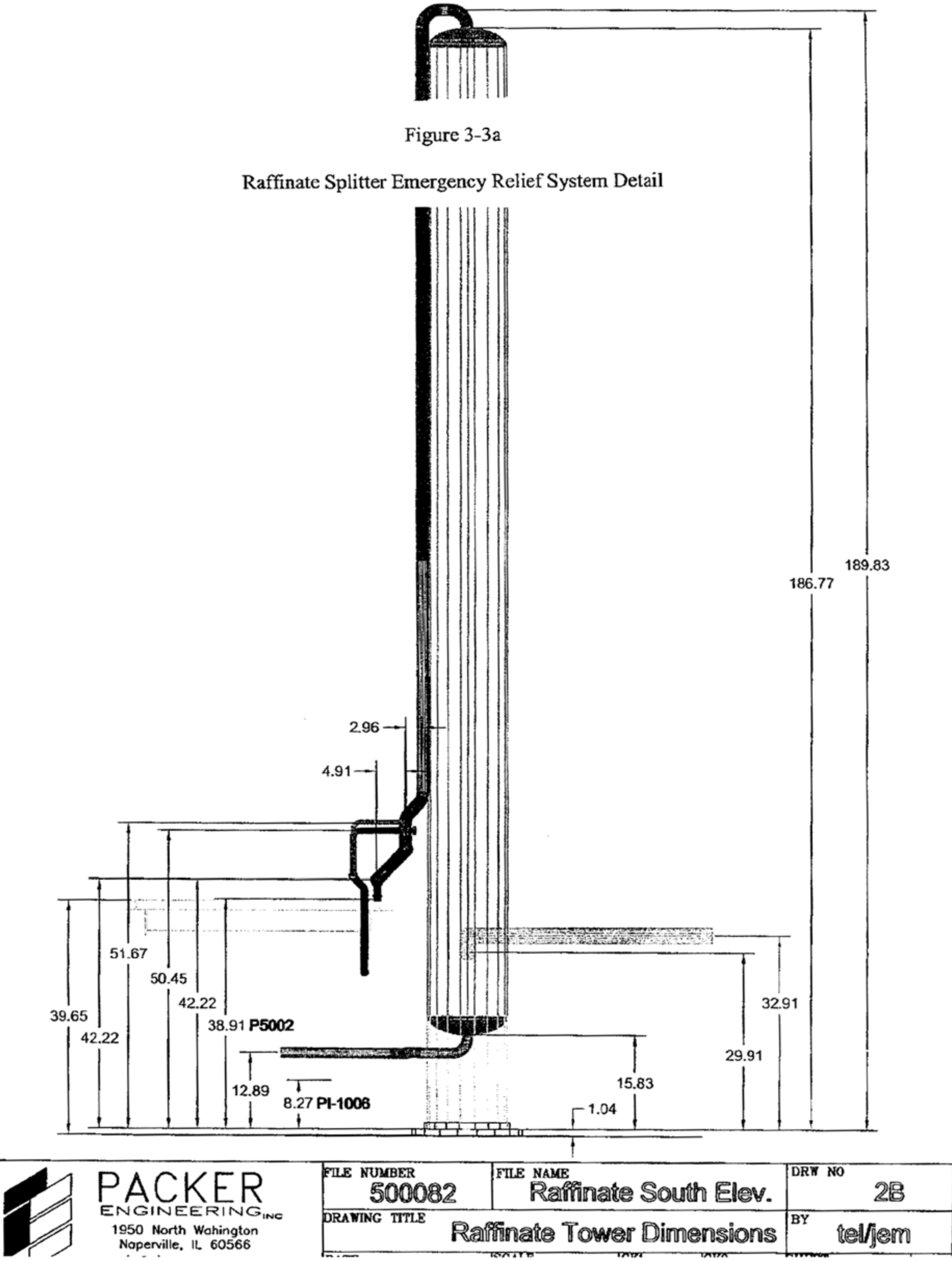
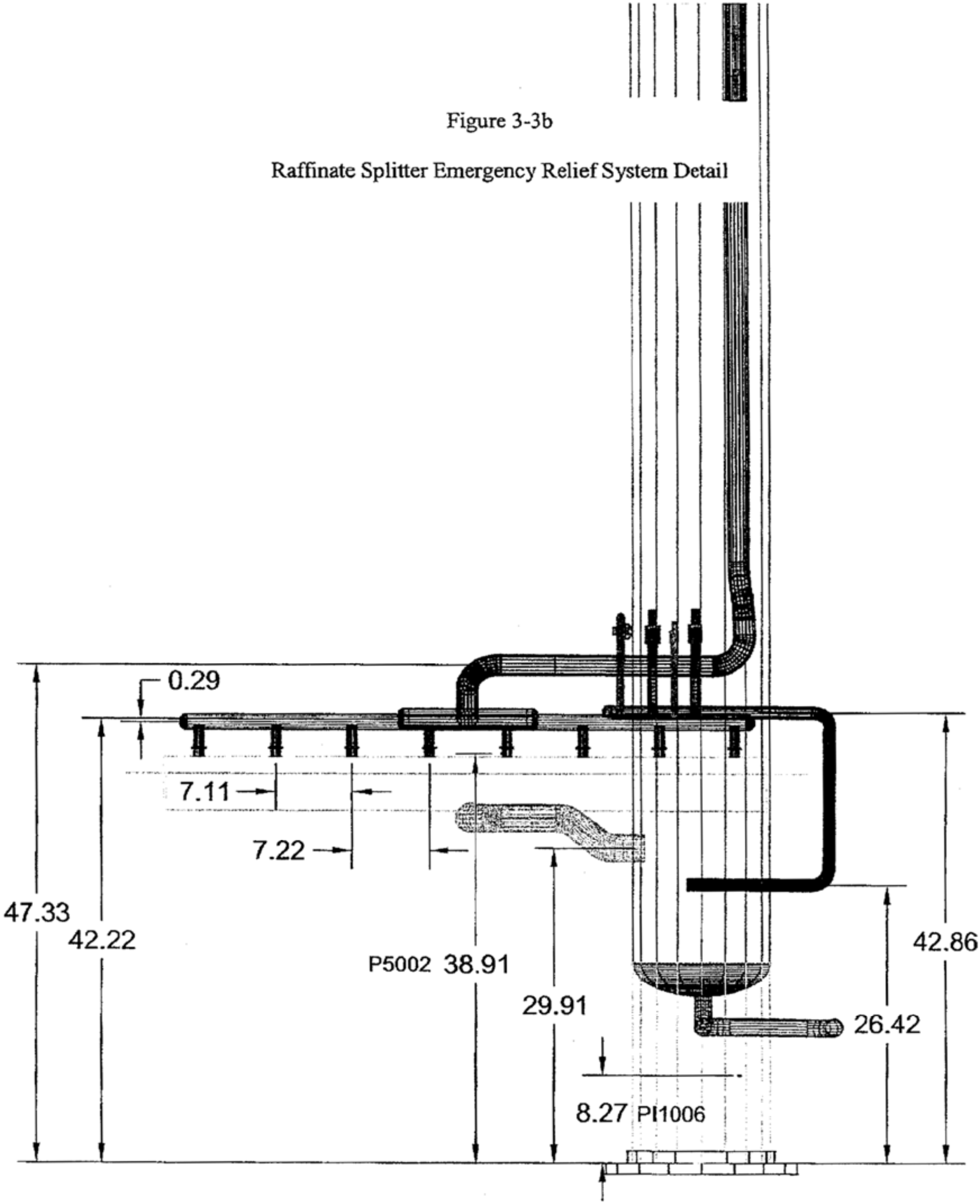


Figure 3-3a

Raffinate Splitter Emergency Relief System Detail

Figure 3-3b

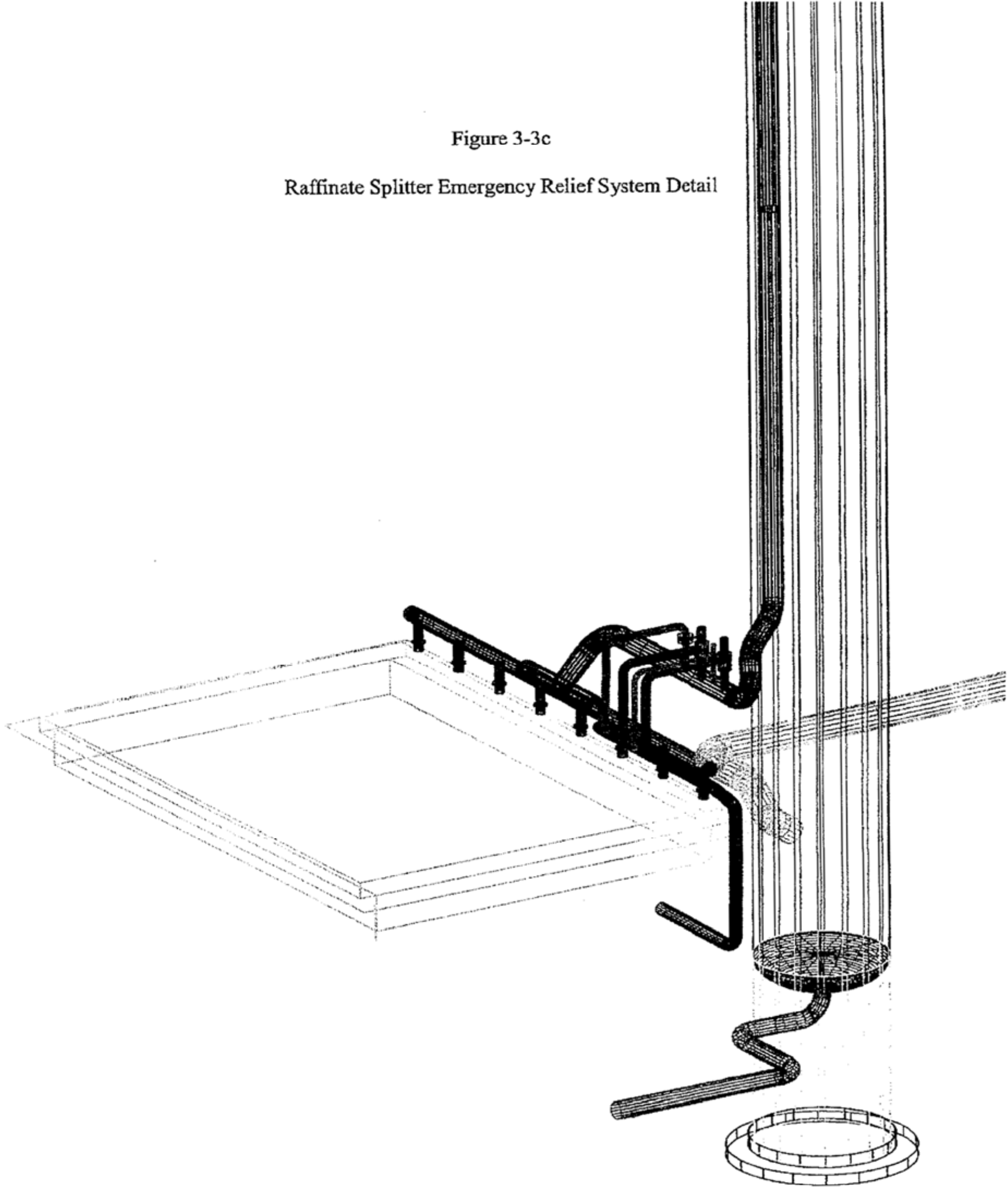
Raffinate Splitter Emergency Relief System Detail



PACKER ENGINEERING INC 1950 North Wahington Naperville, IL 60566	FILE NUMBER 500082	FILE NAME Raffinate WEST Elev.	DRW NO 2C
	DRAWING TITLE Raffinate Tower Dimensions		BY tel/jem

Figure 3-3c

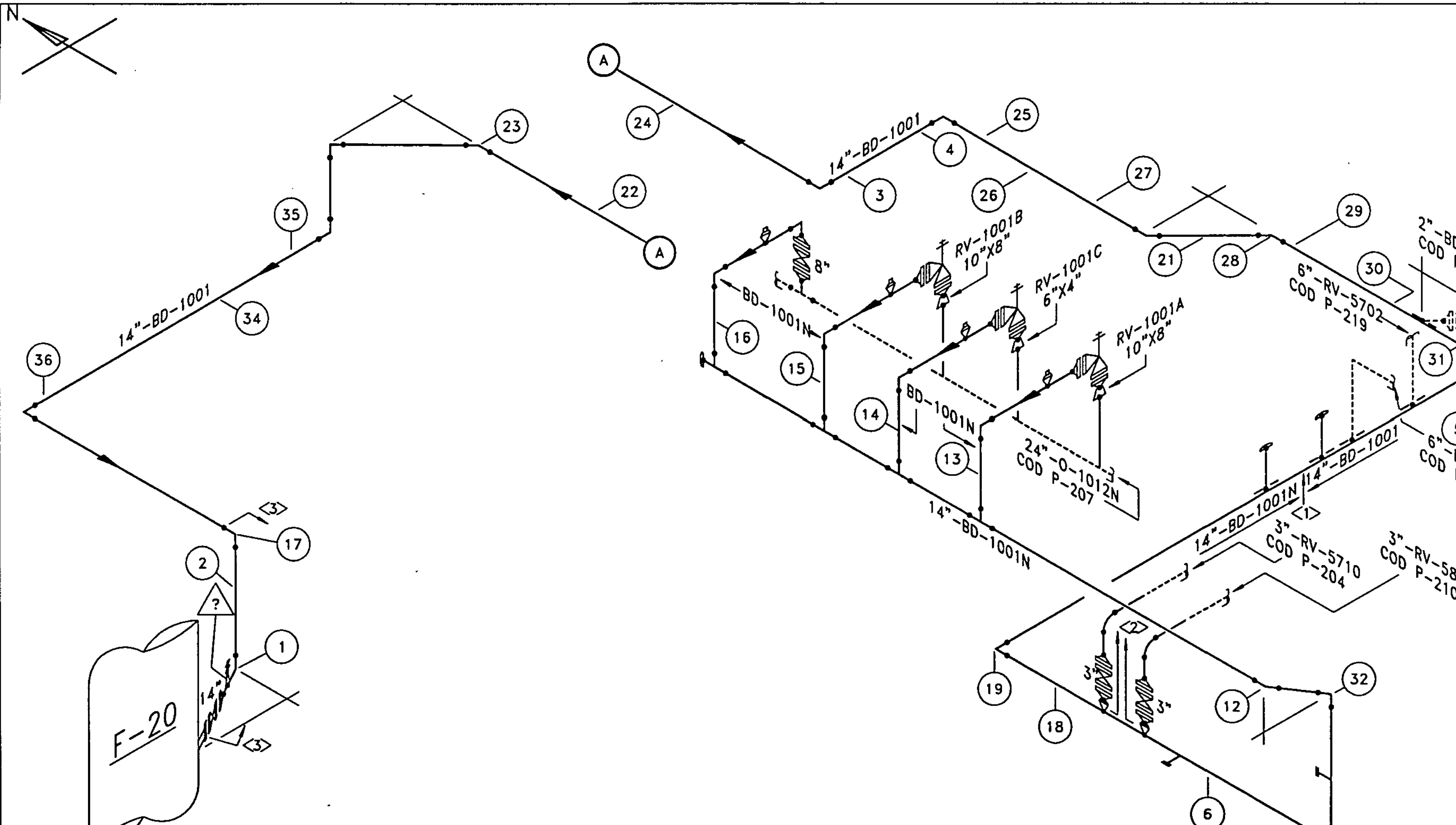
Raffinate Splitter Emergency Relief System Detail



PACKER ENGINEERING, INC 1950 North Wahington Naperville, IL 60566	FILE NUMBER	FILE NAME Raffinate ISO	DRW NO 2D
	DRAWING TITLE	Raffinate Tower Dimensions	BY tel/jem

RAFFINATE SPLITTER OVERHEAD VAPOR LINE AND PIPING

THE OVERHEAD VAPOR LINE AND PROCESS PIPING FROM THE RAFFINATE SPLITTER TOWER TO THE OVERHEAD CONDENSER CONSISTS OF 24-, 18- AND 10-INCH DIAMETER PIPES WITH AN APPROXIMATE LIQUID-FULL VOLUME OF 5,460 GALLONS. THE 24-INCH DIAMETER OVERHEAD VAPOR LINE HAS AN APPROXIMATE HEIGHT OF 140.5 FEET FROM THE TOP OF THE TOWER AND OVERHEAD VAPOR LINE TO THE LOCATION OF THE BOTTOM-MOUNTED SAFETY RELIEF VALVES.



SCRD PIPING SURVEY	
DATE	BY

<2>INSTALLED 97 TAR

<3>REPLACED 14" VA., PIPING, INSTALLED 3/4" VOL

±885' OF PIPE UNINSULATED

15

LINE NO.	DESCRIPTION	TEMP	PRESS	SCH	MAT'L
6,10,14"-BD-100	FROM RV-100A,B,C, TP BD-1001\	360	3/12	40	C.S.
8"-BD-1001N	FROM O-1012N TO14"-BD-1001N	360	3/12	40	C.S.
14"-BD-1001	FROM BD-1001N TO F-20	360	3/12	40	C.S.

MARK	DESCRIPTION	BY	DATE	CHKD
3	REV.	PLP	1/03	
2	REV. REDRAWN	GRA	10/27	
	UPDATED	DRS	7/90	

ALTERATIONS

AMOCO OIL COMPANY
TEXAS CITY REFINERY

ISOM #1
PIPING RECORDS
HUF SPLITTER

SCALE NONE DATE 10-21-97

COMPANY AMOCO OIL COMPANY

OFFICE TC	DEPT INSPEC	REVISION DATE: 02-07-03
DRAWN GRA	ENGR	
APPROVED 19		

DWG. D- ISOM - 1 - P - 222

AutoCAD DWG. 11P222

BLOWDOWN DRUM AND STACK

THE BLOWDOWN DRUM IS A VERTICAL ASME PROCESS VESSEL WITH A MAWP OF 30 PSIG, AN INSIDE DIAMETER OF 10 FEET, AN APPROXIMATE HEIGHT OF 27.0 FEET, A BOTTOM ASME 2:1 ELLIPTICAL HEAD, AND A TOP CONICAL TRANSITION SECTION (FRUSTRUM OF A RIGHT CIRCULAR CONE) WITH AN APPROXIMATE HEIGHT OF 10.0 FEET AND INLET AND OUTLET DIAMETERS OF 10 AND 2.71 FEET (32.5 INCHES), RESPECTIVELY. THE APPROXIMATE LIQUID-FULL VOLUME OF THE BLOWDOWN DRUM IS 19,475 GALLONS.

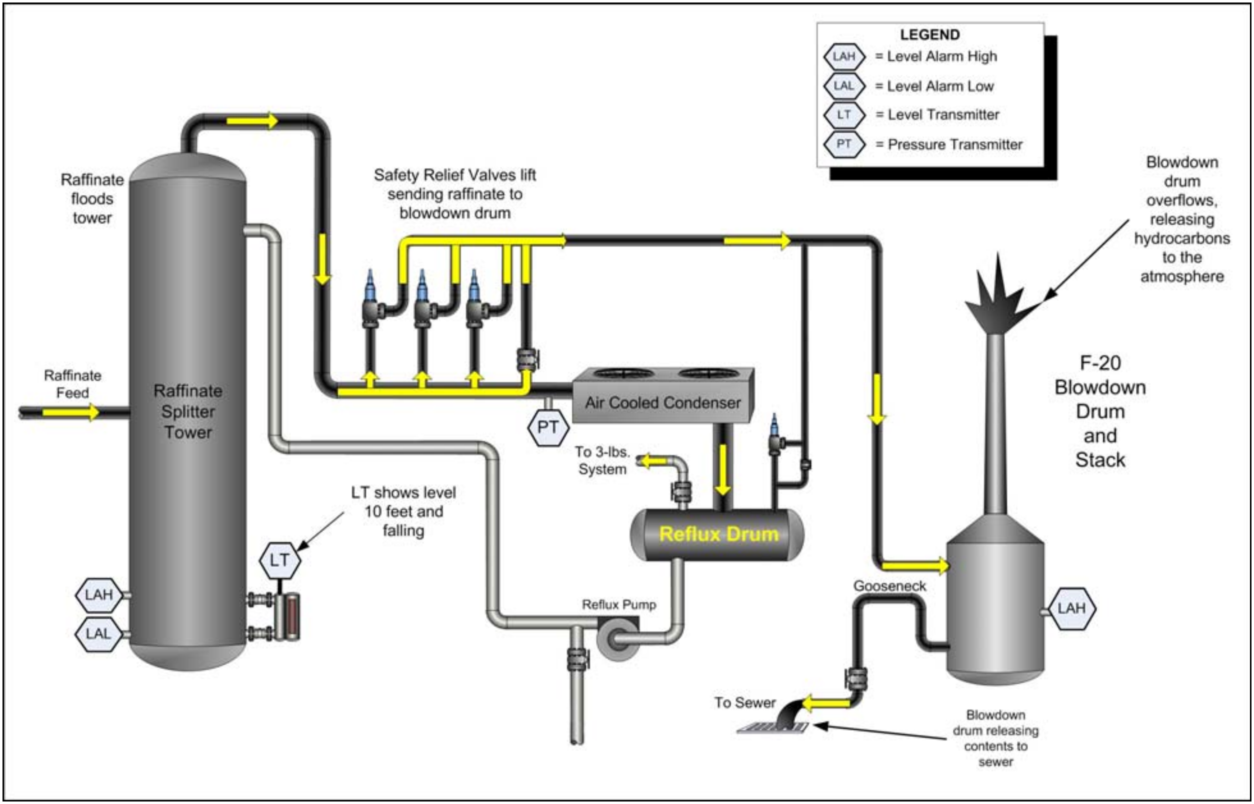
A LIQUID LEVEL, NORMALLY WATER, OF APPROXIMATELY 3,915 GALLONS IS MAINTAINED IN THE BOTTOM OF THE BLOWDOWN DRUM BY A GOOSENECK SEAL. THE 6-INCH DIAMETER GOOSENECK PIPE IS OPEN AND DRAINS TO THE PROCESS SEWER.

THE BLOWDOWN DRUM STACK HAS AN INSIDE DIAMETER OF 2.71 FEET (32.5 INCHES) AND A HEIGHT OF 76.5 FEET. THE DISCHARGE TO THE ATMOSPHERE FROM THE TOP OF THE STACK OCCURS AT AN APPROXIMATE HEIGHT OF 119.3 FEET. THE APPROXIMATE LIQUID-FULL VOLUME OF THE BLOWDOWN DRUM STACK IS 3,295 GALLONS. THE APPROXIMATE LIQUID-FULL VOLUME OF THE BLOWDOWN DRUM AND STACK IS 22,770 GALLONS.

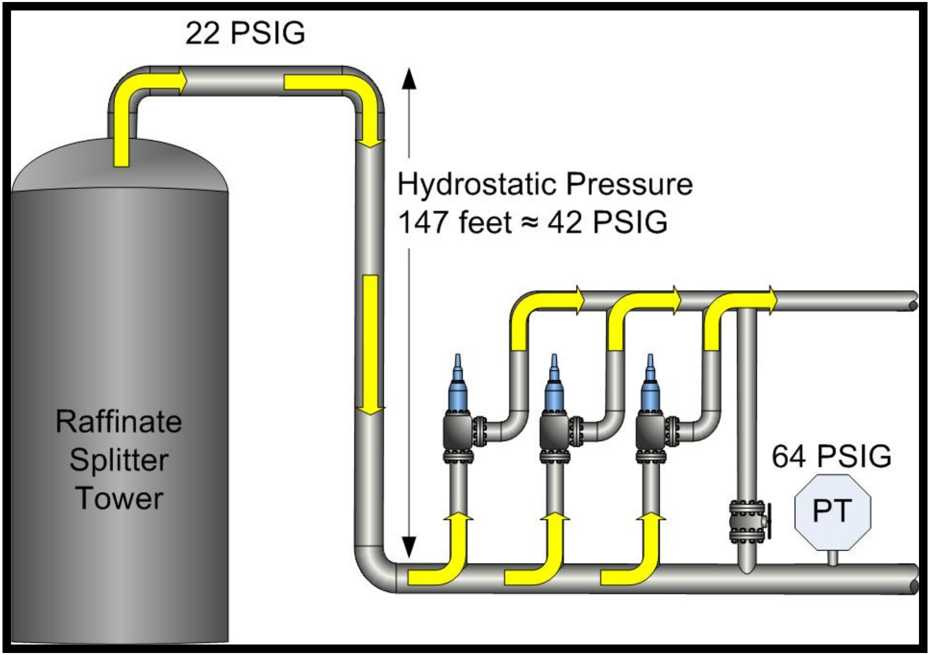
BLOWDOWN DRUM TRANSITION

RAFFINATE SPLITTER SRV HEADER ENTERING BLOWDOWN DRUM

BLOWDOWN DRUM SEAL LEG TO SEWER



TOWER OVERFILLS AND BLOWDOWN DRUM RELEASES HYDROCARBONS



WEIGHT OF LIQUID IN OVERHEAD PIPING LEADS TO THE THREE SAFETY RELIEF VALVES OPENING

PROCESS EQUIPMENT SUMMARY

A SUBCOOLED LIQUID DUE TO VAPOR-LIQUID FLASHING WITHIN THE RAFFINATE SPLITTER TOWER AND THERMAL EXPANSION FLOWED FROM THE RAFFINATE SPLITTER TOWER INTO THE OVERHEAD VAPOR LINE. HYDROSTATIC HEAD DUE TO THE LIQUID LEVEL IN THE OVERHEAD VAPOR LINE PROVIDED SUFFICIENT PRESSURE TO SUCCESSIVELY OPEN THE THREE BELLOWS SAFETY RELIEF VALVES PROTECTING THE TOWER. SUBCOOLED LIQUID THEN FLOWED INTO THE RAFFINATE SPLITTER SAFETY RELIEF VALVE DISCHARGE PIPE HEADER, THE BLOWDOWN DRUM, TWO DISCHARGE PIPE HEADERS OF OTHER SAFETY RELIEF VALVES, AND THE BLOWDOWN DRUM STACK. LIQUID HYDROCARBONS SIMULTANEOUSLY FLOWED TO THE SEWER WHILE FILLING THE BLOWDOWN DRUM AND STACK AND THE TWO OTHER SAFETY RELIEF VALVE DISCHARGE PIPE HEADERS. THE HYDROCARBONS THAT FLOWED TO THE ATMOSPHERE FROM THE BLOWDOWN DRUM STACK FORMED THE VAPOR CLOUD THAT EXPLODED AND THE LIQUID POOLS THAT BURNED. THE VOLUMES OF THE RAFFINATE SPLITTER TOWER, EACH PIECE OF ANCILLARY EQUIPMENT, THE BLOWDOWN DRUM AND STACK, AND THE THREE SAFETY RELIEF VALVE DISCHARGE PIPE HEADERS ARE SHOWN IN TABLE 1.

Table 1

VOLUMES OF THE RAFFINATE SPLITTER TOWER AND ANCILLARY EQUIPMENT

Equipment		Volume (gal)
Raffinate Splitter Tower		154,845
Overhead Vapor Line (Total)		5,460
Overhead Condenser		2,270
Reflux Drum		6,430
	Subtotal	14,160
RS SRV Discharge Pipe Header		6,975
Blowdown Drum and Stack		22,770
Other SRV Discharge Pipe Headers*		6,300
	Subtotal:	36,045
Initial Volume at Gooseneck Pipe		(3,915)
	Subtotal	32,130

* @ 0.675 of actual volume due to inert gas compression

EMERGENCY RELIEF SYSTEM EVALUATION
(INCIDENT ANALYSIS)

SUBCOOLED LIQUID FLOW FROM THE RAFFINATE SPLITTER TO THE BLOWDOWN DRUM

THE SAFETY RELIEF VALVES INSTALLED TO PROTECT THE RAFFINATE SPLITTER WERE MOUNTED ON THE OVERHEAD VAPOR LINE APPROXIMATELY 140.5 FEET BELOW THE TOP OF THE TOWER. THE HEAD TEMPERATURE OF THE TOWER WAS LOW ENOUGH AND THE HYDROSTATIC HEAD OF THE LIQUID HIGH ENOUGH TO ENSURE THAT A SUBCOOLED LIQUID FLOWED THROUGH THE SAFETY RELIEF VALVES AND INTO THE BLOWDOWN DRUM. THESE SAFETY RELIEF VALVE FLOWS ARE REDUCED BY THE BACKPRESSURE DUE TO THE DISCHARGE PIPE HEADER AND THE HYDROSTATIC HEAD OF THE BLOWDOWN DRUM STACK.

SAFETY RELIEF VALVES

THE THREE CONSOLIDATED BELLOWS SAFETY RELIEF VALVES PROTECTING THE RAFFINATE SPLITTER HAD COEFFICIENTS OF DISCHARGE AND ACHIEVE THEIR RATED CAPACITY FLOWS AS FOLLOWS:

	VAPOR FLOW	LIQUID FLOW
Kd	0.95	0.74
RATED CAPACITY	1.1 PSET	1.25 PSET

A KP FACTOR IS APPLIED TO A SAFETY RELIEF VALVE WITH VAPOR TRIM WHEN FLOWING A LIQUID. THE FLOW FROM THESE VALVES THEREFORE INCREASES FROM 0.6 TO 1.0 OF THE RATED CAPACITY AS THE FLOWING PRESSURE INCREASES FROM 1.1 TO 1.25 PSET.

SPECIFICATIONS OF SAFETY RELIEF VALVES INSTALLED ON THE RAFFINATE SPLITTER TOWER

Safety Relief Valve	Pressure Setting (psig)	Inlet Pipe Diameter (inches)	Discharge Pipe Diameter (inches)
4P6	40	6	6
8T10	41	10	10
8T10	42	10	10

FLOW FROM THE SAFETY RELIEF VALVES

ON THE RAFFINATE SPLITTER

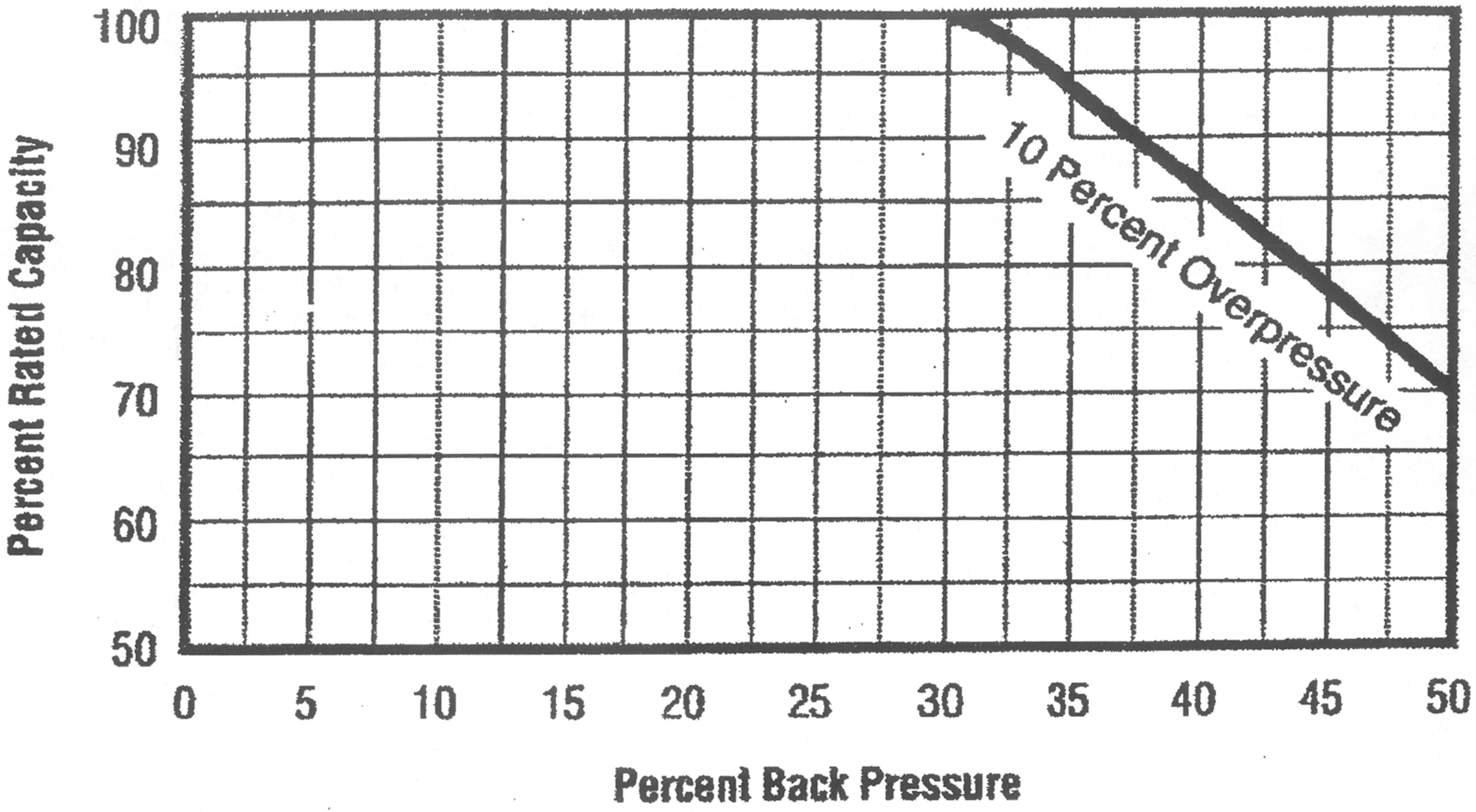
NO DISCHARGE PIPE HEADER		14-INCH DISCHARGE PIPE HEADER		20-INCH DISCHARGE PIPE HEADER	
Tower	Flow Rate (pph)	Flow Rate (pph)	% Back Pressure	Flow Rate (pph)	% Back Pressure
Vapor Flow	416,465	213,340	60.8	378,860	37.0
Liquid Flow	5,216,830	3,358,340	90.0	-	-

SAFETY RELIEF VALVE Kb AND Kw

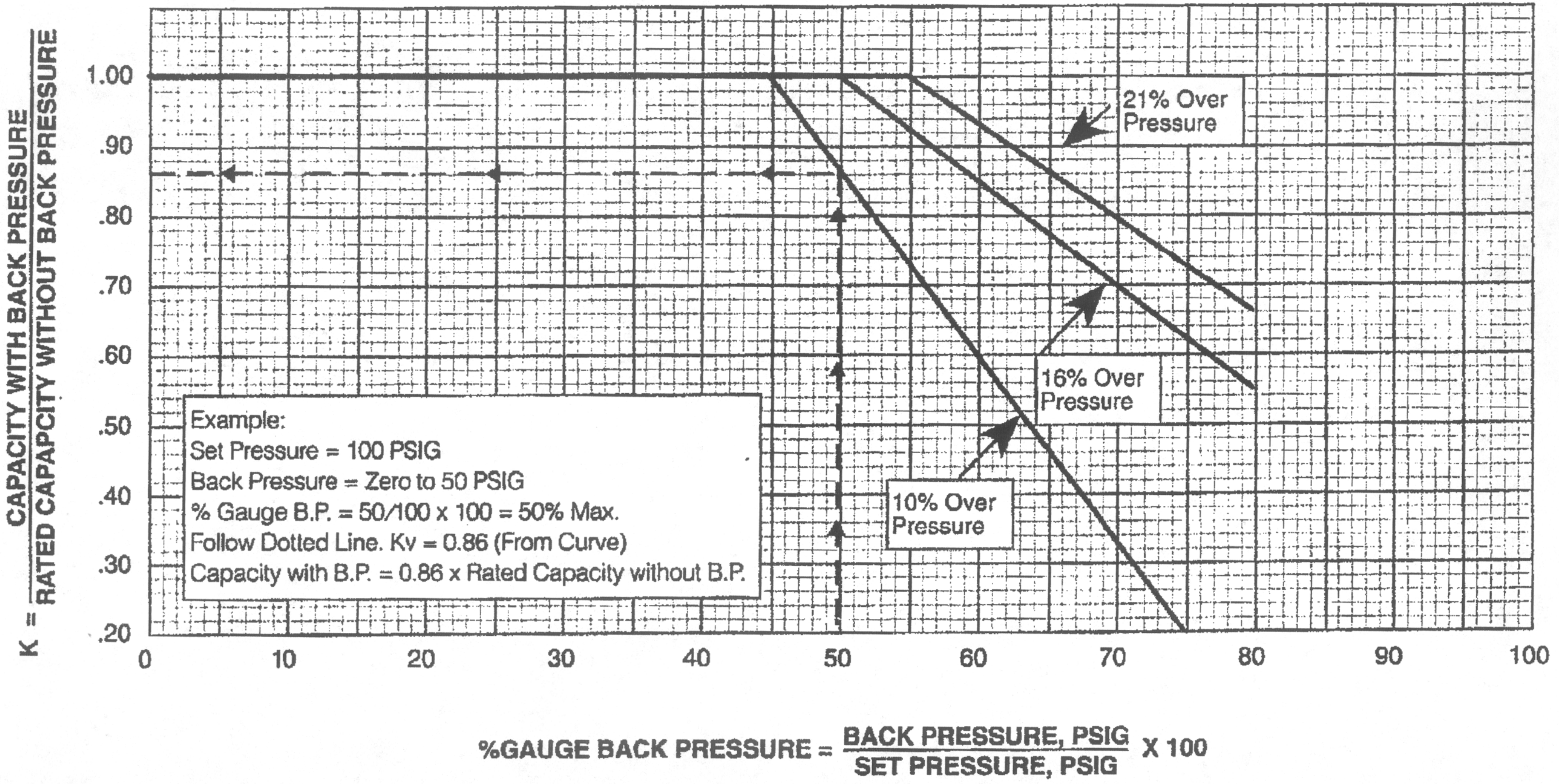
WHEN THE FLOWING AND CONSTANT SUPERIMPOSED BACKPRESSURES ON THE SAFETY RELIEF VALVES EXCEED CERTAIN VALUES, KB AND KW FACTORS ARE ALSO APPLIED TO REDUCE THE VAPOR OR LIQUID FLOWS, RESPECTIVELY, TO ACCOUNT FOR THE VALVES GOING OUT OF FULL LIFT. THE KB AND KW VALUES IN API RP 520 ARE CONSENSUS VALUES OF VARIOUS MANUFACTURERS. EACH MANUFACTURER ALSO PUBLISHES KB AND KW CURVES FOR THEIR OWN VALVES. A SINGLE CURVE IS USED TO REPRESENT THE PERFORMANCE OF ALL MODELS IN A MANUFACTURER'S LINE OF VALVES. THESE VALUES WERE SUPPOSEDLY MEASURED, BUT MOST OF THE DATA ARE NOT NOW AVAILABLE TO JUSTIFY THE PUBLISHED CURVES. THE PUBLISHED VALUES OF THE VARIOUS MANUFACTURES DO NOT AGREE.

SAFETY RELIEF VALVE Kb AND Kw

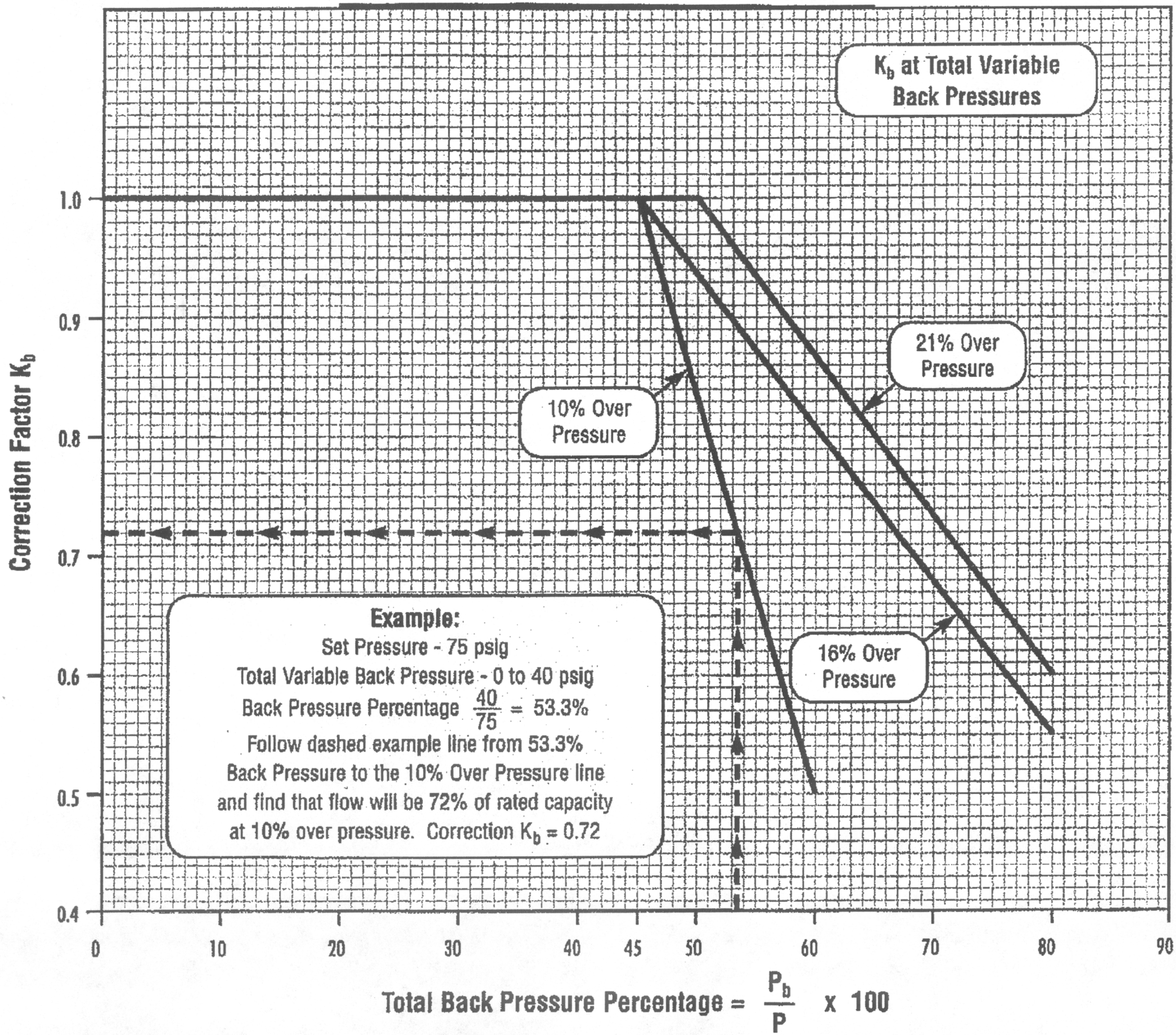
SAFETY RELIEF VALUES WITH MODIFIED LIQUID TRIM ARE NOW REQUIRED BY THE ASME CODE TO HAVE THEIR RATED CAPACITY FLOW CERTIFIED AT 1.1 PSET. THE MANUFACTURERS, HOWEVER, HAVE NEVER CHANGED THE KW CURVES PUBLISHED FOR THEIR OLDER MODEL LIQUID VALVES WITH THE RATED CAPACITY CERTIFIED AT 1.25 PSET. DUE TO THE HIGH OVERPRESSURE OF THE LIQUID FLOW FROM THE SAFETY RELIEF VALVES, A KW VALUE OF 1.0 WAS FOUND TO BEST REPRESENT THE INCIDENT INFORMATION.



Effects of Back Pressure on Relief Capacity of Typical Balanced Direct Spring PRV on Gas

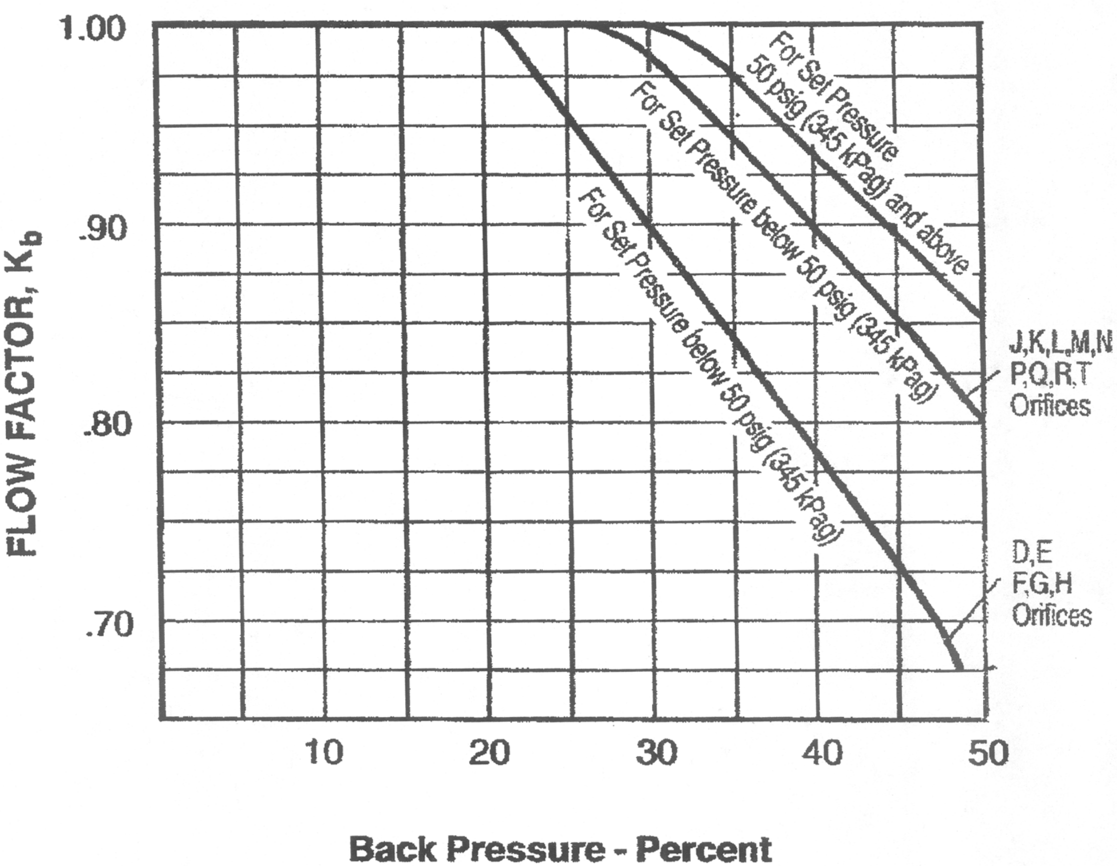


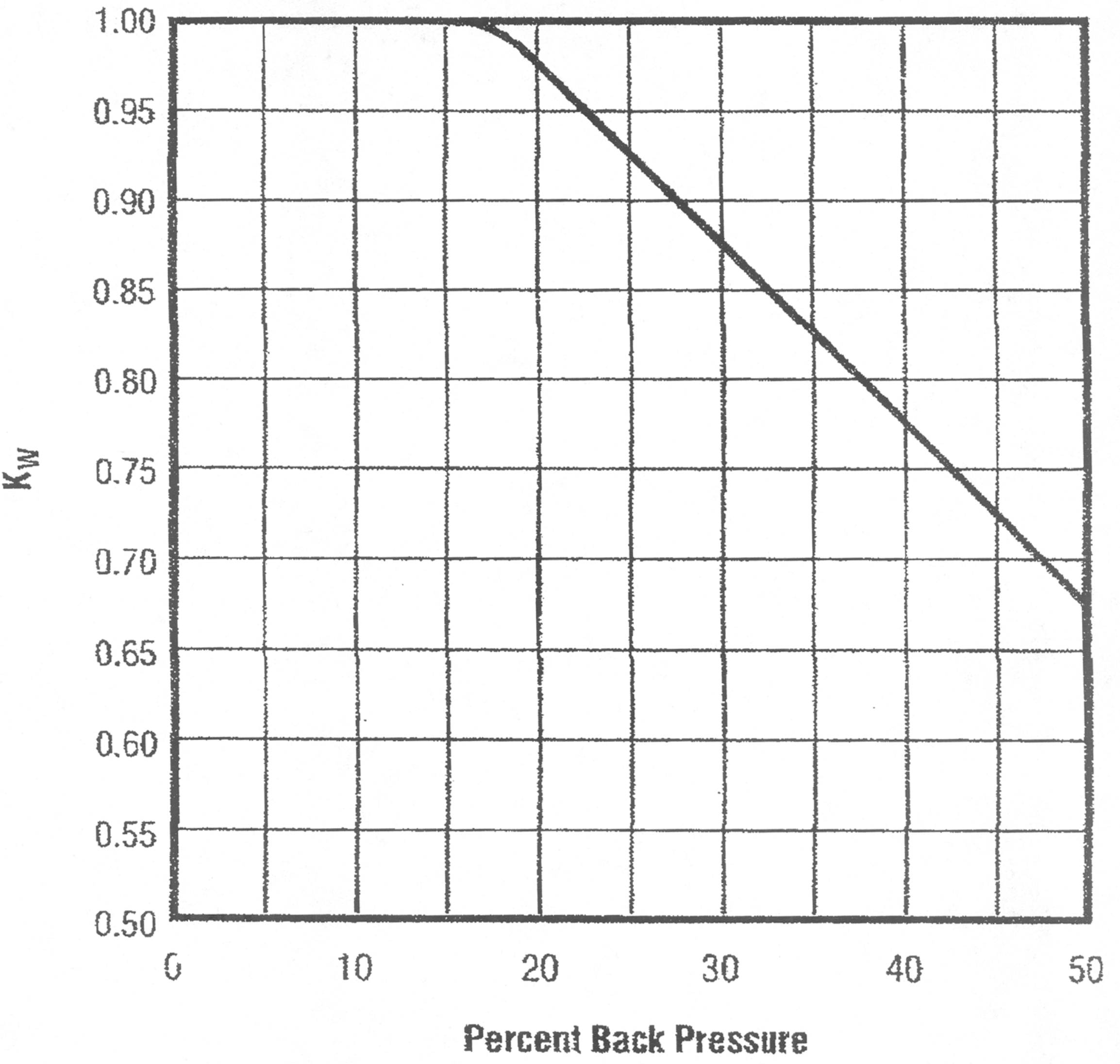
K = CAPACITY WITH BACK PRESSURE / RATED CAPAPCITY WITHOUT BACK PRESSURE
1.00
.90
.80
.70
.60
.50
.40
.30
.20
0
10
20
30
40
50
60
70
80
90
100
21% Over Pressure
16% Over Pressure
10% Over Pressure
Example:
Set Pressure = 100 PSIG
Back Pressure = Zero to 50 PSIG
% Gauge B.P. = 50/100 x 100 = 50% Max.
Follow Dotted Line. Kv = 0.86 (From Curve)
Capacity with B.P. = 0.86 x Rated Capacity without B.P.
%GAUGE BACK PRESSURE = BACK PRESSURE, PSIG / SET PRESSURE, PSIG X 100



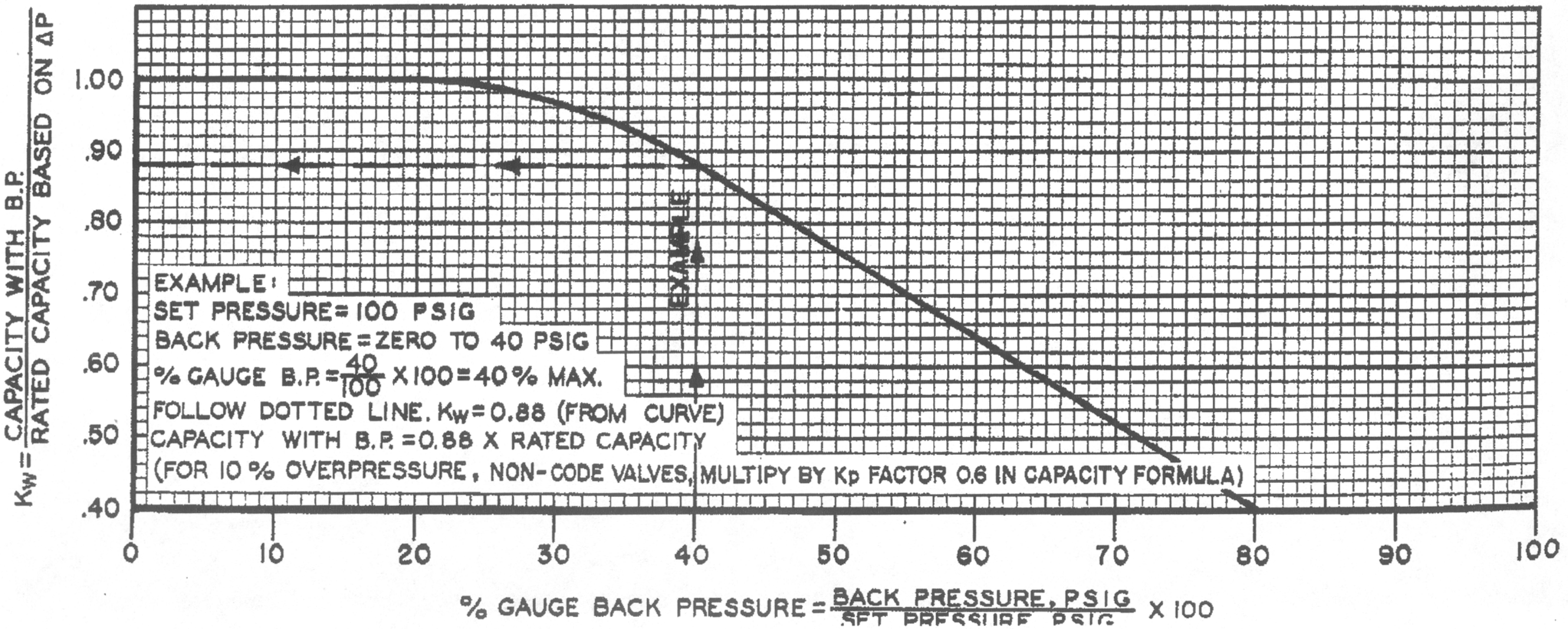
K_b at Total Variable Back Pressures
1.0
0.9
0.8
0.7
0.6
0.5
0.4
Correction Factor K_b
21% Over Pressure
10% Over Pressure
16% Over Pressure
Example:
Set Pressure - 75 psig
Total Variable Back Pressure - 0 to 40 psig
Back Pressure Percentage $\frac{40}{75} = 53.3\%$
Follow dashed example line from 53.3% Back Pressure to the 10% Over Pressure line and find that flow will be 72% of rated capacity at 10% over pressure. Correction $K_b = 0.72$
0
10
20
30
40
45
50
60
70
80
90
Total Back Pressure Percentage = $\frac{P_b}{P} \times 100$

Correction Factor for Vapors and Gases, K_b
for Balanced Bellows Valves - 10% Overpressure

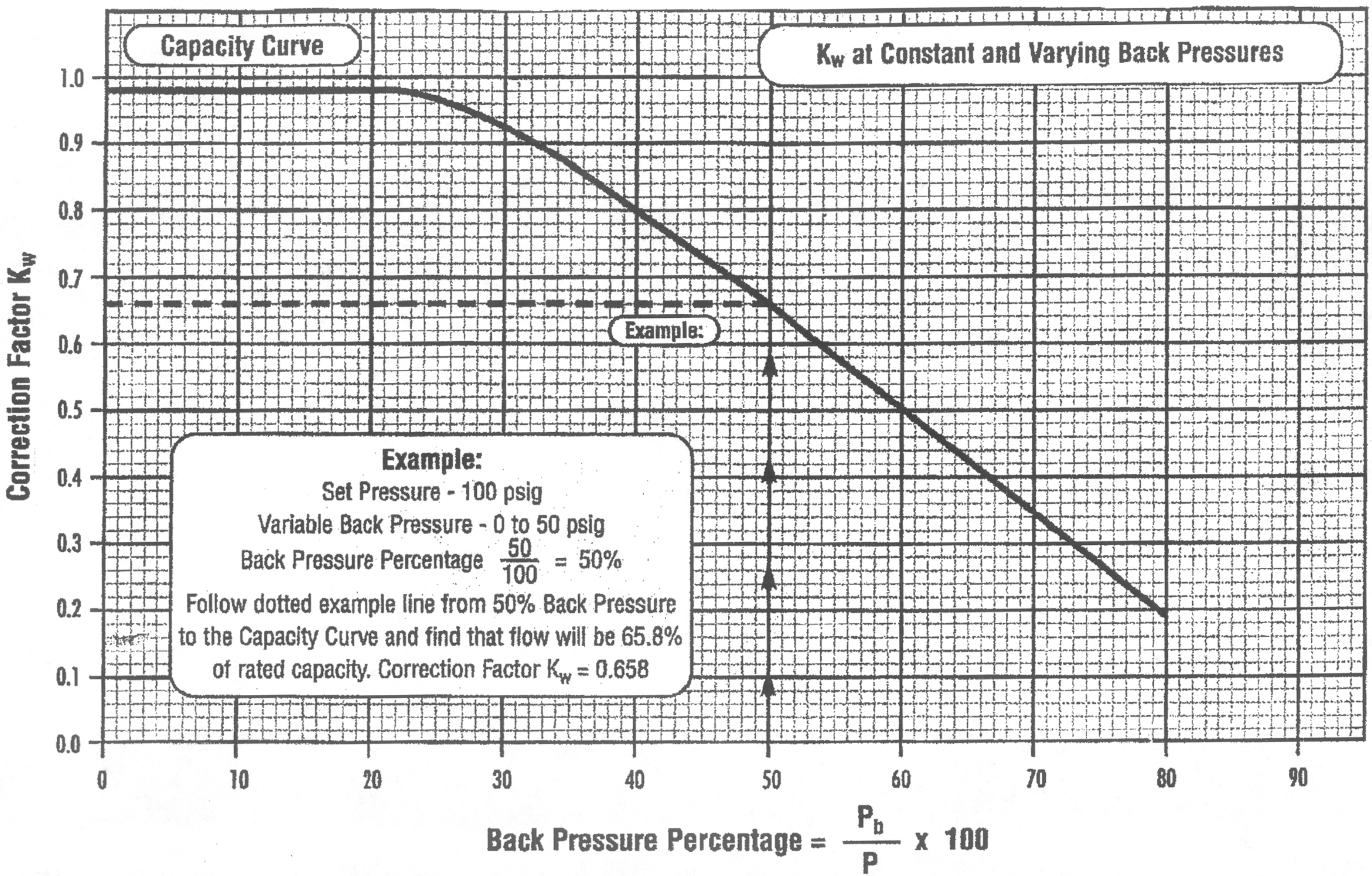


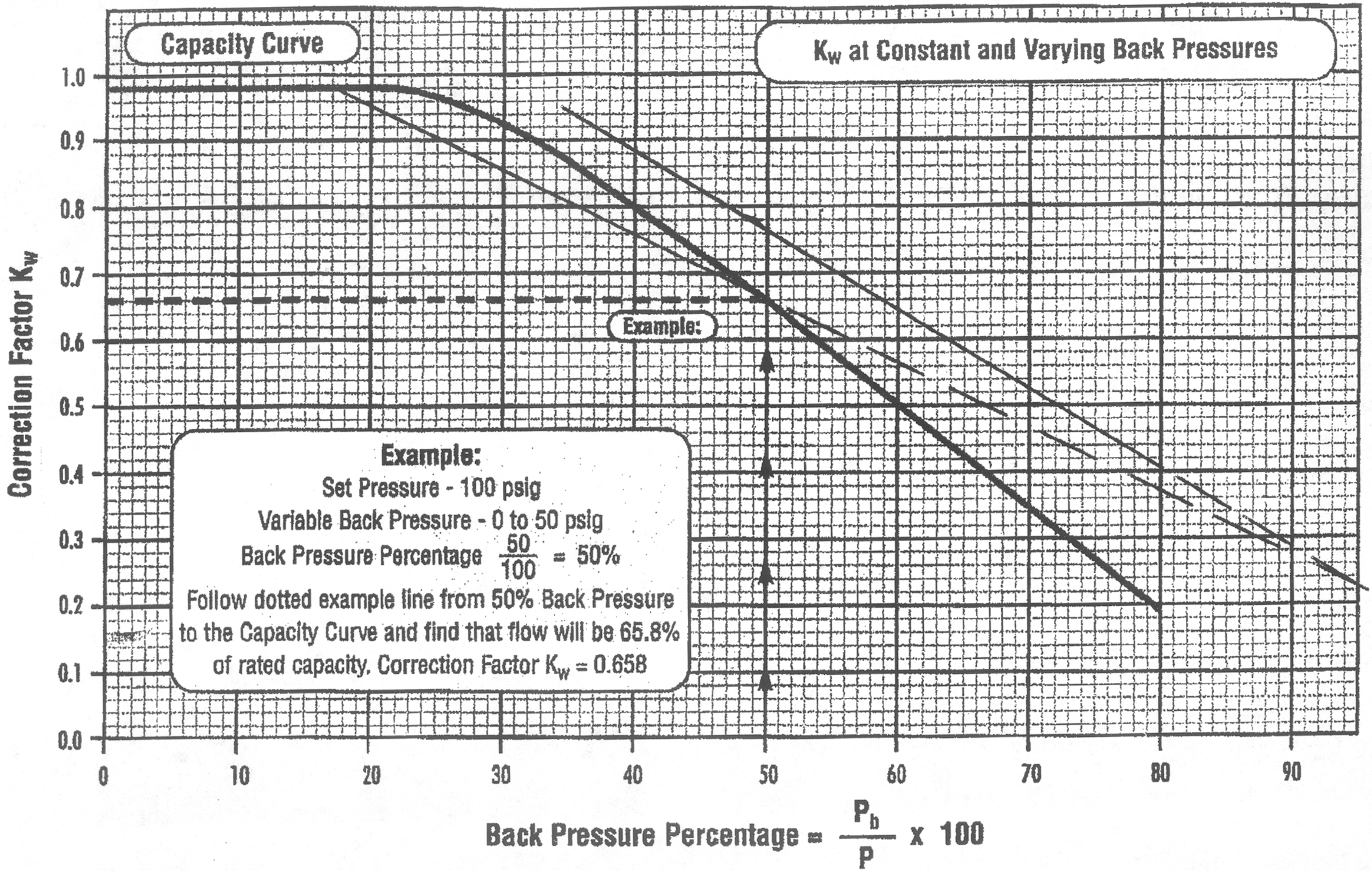


K_W for Balanced Bellows Spring Valves on Liquid



$K_W = \frac{\text{CAPACITY WITH B.P.}}{\text{RATED CAPACITY BASED ON } \Delta P}$
1.00
.90
.80
.70
.60
.50
.40
EXAMPLE
EXAMPLE:
SET PRESSURE = 100 PSIG
BACK PRESSURE = ZERO TO 40 PSIG
% GAUGE B.P. = $\frac{40}{100}$ X 100 = 40 % MAX.
FOLLOW DOTTED LINE. K_W = 0.88 (FROM CURVE)
CAPACITY WITH B.P. = 0.88 X RATED CAPACITY
(FOR 10 % OVERPRESSURE, NON-CODE VALVES, MULTIPY BY Kp FACTOR 0.6 IN CAPACITY FORMULA)
0
10
20
30
40
50
60
70
80
90
100
% GAUGE BACK PRESSURE = $\frac{\text{BACK PRESSURE, PSIG}}{\text{SET PRESSURE, PSIG}}$ X 100


Capacity Curve
K_W at Constant and Varying Back Pressures
Correction Factor K_W
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Example:
Example:
Set Pressure - 100 psig
Variable Back Pressure - 0 to 50 psig
Back Pressure Percentage $\frac{50}{100}$ = 50%
Follow dotted example line from 50% Back Pressure
to the Capacity Curve and find that flow will be 65.8%
of rated capacity. Correction Factor K_w = 0.658
0
10
20
30
40
50
60
70
80
90
Back Pressure Percentage = $\frac{P_b}{P}$ x 100



Capacity Curve
K_W at Constant and Varying Back Pressures
Correction Factor K_W
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Example:
Example:
Set Pressure - 100 psig
Variable Back Pressure - 0 to 50 psig
Back Pressure Percentage $\frac{50}{100} = 50\%$
Follow dotted example line from 50% Back Pressure
to the Capacity Curve and find that flow will be 65.8%
of rated capacity. Correction Factor $K_W = 0.658$
0
10
20
30
40
50
60
70
80
90
Back Pressure Percentage $= \frac{P_b}{P} \times 100$

EMERGENCY RELIEF SYSTEM EVALUATION
(DESIGN ANALYSIS)

THE THREE BELLOWS SAFETY RELIEF VALVES ON THE TOWER REMAINED OPEN FOR APPROXIMATELY SIX MINUTES AND FLOWED APPROXIMATELY 32,120 GALLONS OF A SUBCOOLED LIQUID THAT FILLED THE RAFFINATE SPLITTER BLOWDOWN HEADER, BLOWDOWN DRUM AND BLOWDOWN DRUM STACK ASSEMBLY AND TWO ADDITIONAL SAFETY RELIEF VALVE HEADERS. ADDITIONAL SUBCOOLED LIQUID SIMULTANEOUSLY FLOWED FROM THE BOTTOM OF THE BLOWDOWN DRUM THROUGH A 6-INCH DIAMETER GOOSENECK PIPE TO THE CLOSED SEWER WHILE THE VESSEL WAS FILLING.

A “GEYSER”, REPORTED BY EYEWITNESSES TO BE APPROXIMATELY 20-FEET IN HEIGHT AND HAVING THE 32.5-INCH DIAMETER OF THE BLOWDOWN DRUM STACK, FLOWED TO THE ATMOSPHERE AT AN ELEVATION OF APPROXIMATELY 119.3 FEET. EYEWITNESSES REPORTED A TIME LAG FROM THE INITIAL VENTING TO THE IGNITION OF THE VAPOR CLOUD OF 15 SECONDS TO TWO MINUTES.

SUBCOOLED LIQUID FLOW

THE SUBCOOLED LIQUID FLOW THROUGH THE SAFETY RELIEF VALVES DISCHARGE PIPE HEADER IS BERNOULLI FLOW. THE AVAILABLE PRESSURE DIFFERENTIAL FOR FLOW DECREASES AS THE BACKPRESSURE IN THE PIPE INCREASES. THE SAFETY RELIEF VALVES ON THE RAFFINATE SPLITTER ARE MOUNTED APPROXIMATELY 50 FEET ABOVE THE GROUND LEVEL. ONCE THE BLOWDOWN DRUM FILLS AND THE LIQUID LEVEL RISES INTO THE STACK, THE HYDROSTATIC HEAD INCREASES DUE TO THE INCREASE OF THE LIQUID LEVEL FURTHER DECREASING THE AVAILABLE PRESSURE DIFFERENTIAL. THE FLOW FROM THE SAFETY RELIEF VALVES THEREFORE ALSO DECREASES.

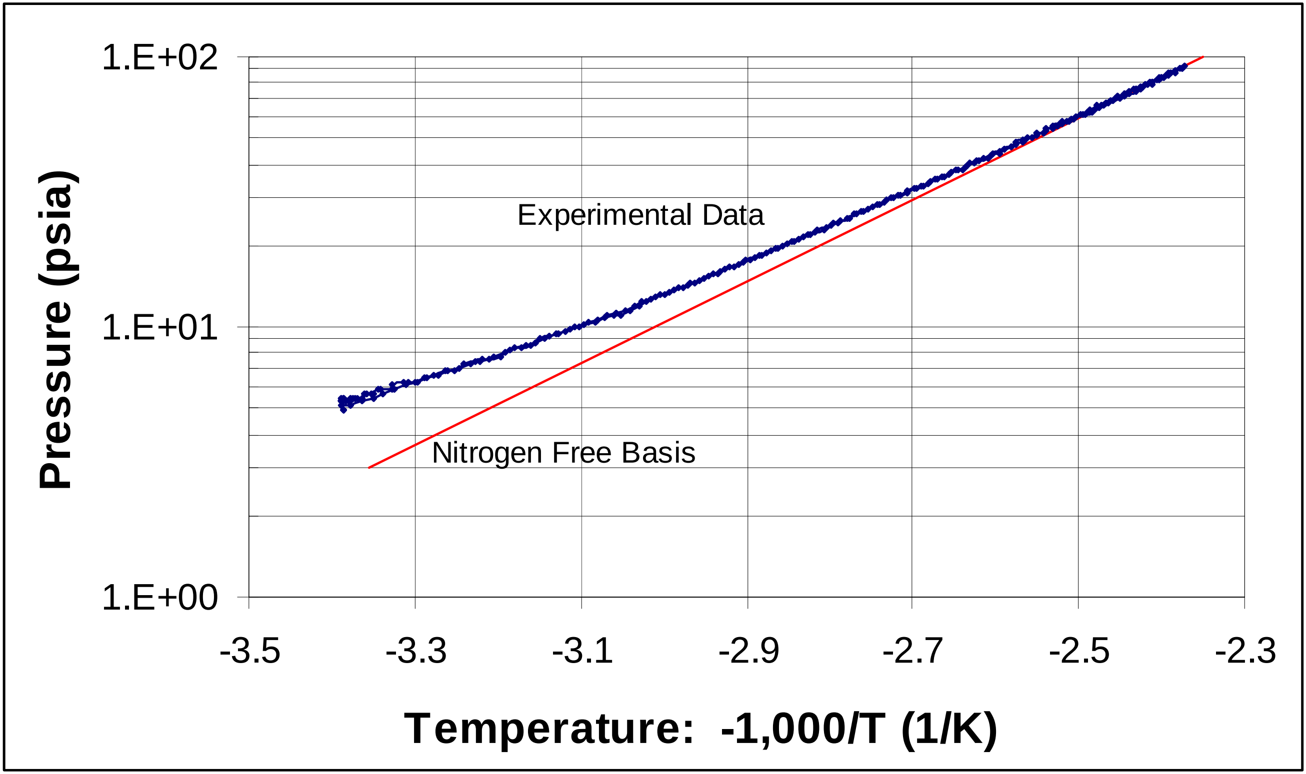
RAFFINATE SPLITTER –

COMPOSITION USED TO MODEL THE INCIDENT

Compound	Weight Fraction
n-pentane	0.0383
i-pentane (2-methyl butane)	0.0263
n-hexane	0.1519
i-hexane (2-methyl pentane)	0.2950
n-heptane	0.3072
n-octane	0.1300
n-nonane	0.0409
heavy (n-decane)	0.0104
Total	1.0000

Distillation Analysis

Sample	IBP		5 % Recovery	
	°F	°C	°F	°C
Raffinate Overhead Product	149	65.0	158	70.0
Raffinate Feed	150	65.5	160	71.1
Raffinate Bottoms	155	68.3	164	73.3

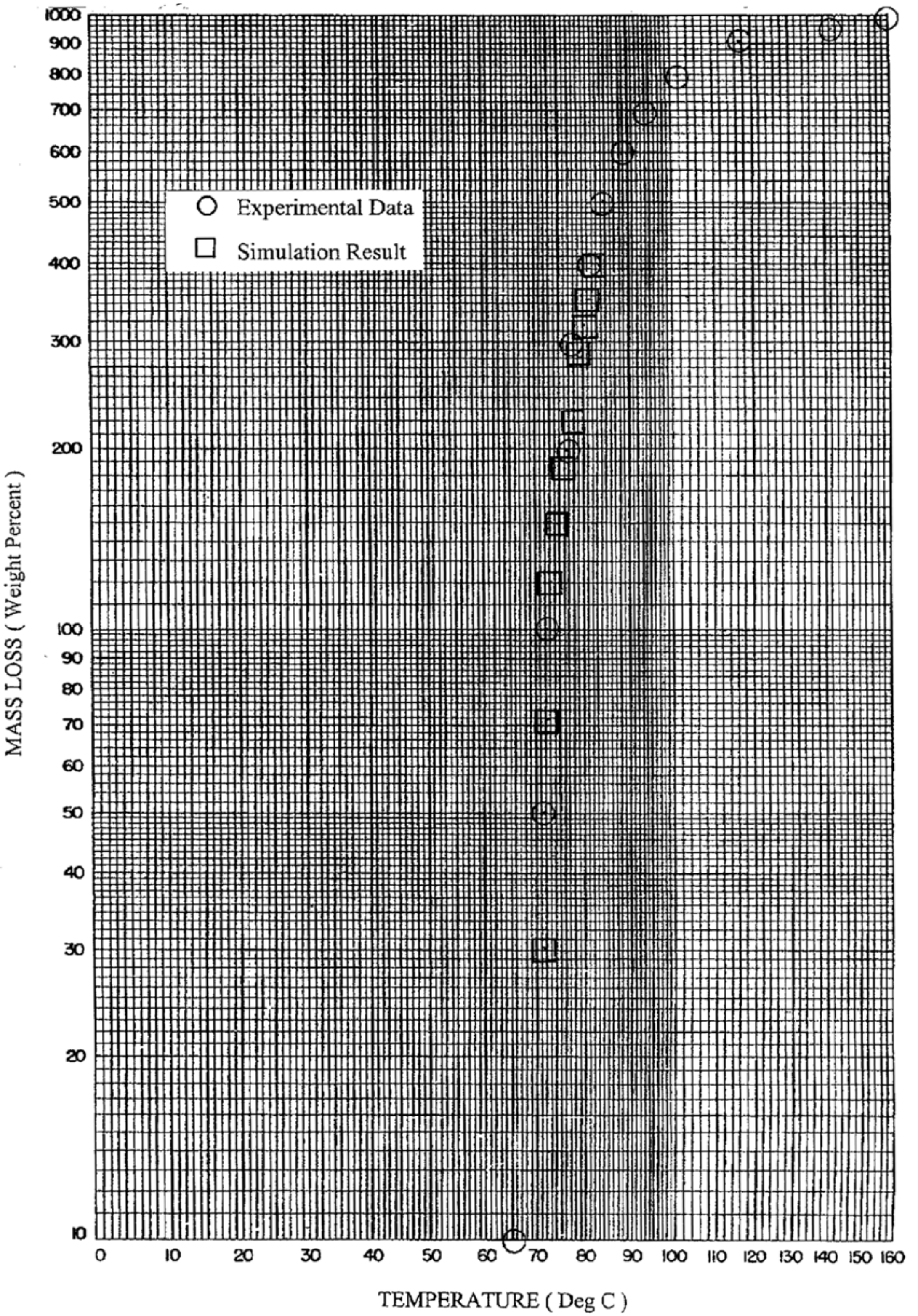


EXPERIMENTAL BOILING (BUBBLE) POINT DATA

(FAUSKE & ASSOCIATES, 2005)

Figure C-2

Incident Distillation Curve - Mass Loss versus Temperature - Comparison to the Simulation Prediction



BUNCEFIELD MAJOR INCIDENT INVESTIGATION

Initial Report to the Health and Safety Commission and the Environment Agency of the investigation into the explosions and fires at the Buncefield oil storage and transfer depot, Hemel Hempstead, on 11 December 2005

Buncefield Major Incident Investigation Board

Formation of the vapour cloud

25 The Third Progress Report described extensive tests undertaken to model the behaviour of fuel escaping from Tank 912 during overfilling. Tank 912 was fitted with a deflector plate, installed to direct water from sprinklers on the tank's top to its sides to provide cooling in the event of fire. The tests demonstrated that the deflector plate channelled some of the escaped fuel onto the tank wall, but the rest ran over the top of the plate, fragmenting into droplets that cascaded through the air. Most of the fuel running down the wall hit a wind girder (a structural stiffening ring) and detached from the tank wall, creating a second cascade of droplets.

26 These conditions would promote the evaporation of the lighter components of petrol, eg butanes, pentanes and hexanes. The free-fall of droplets leads to entrainment of air and mixing between the air and fuel vapour, and the formation of a rich fuel/air mixture. Cooling of the surrounding air, already saturated with water vapour by the evaporation, would cause some of the water content to precipitate as an ice mist, which is consistent with the cloud of mist visible on

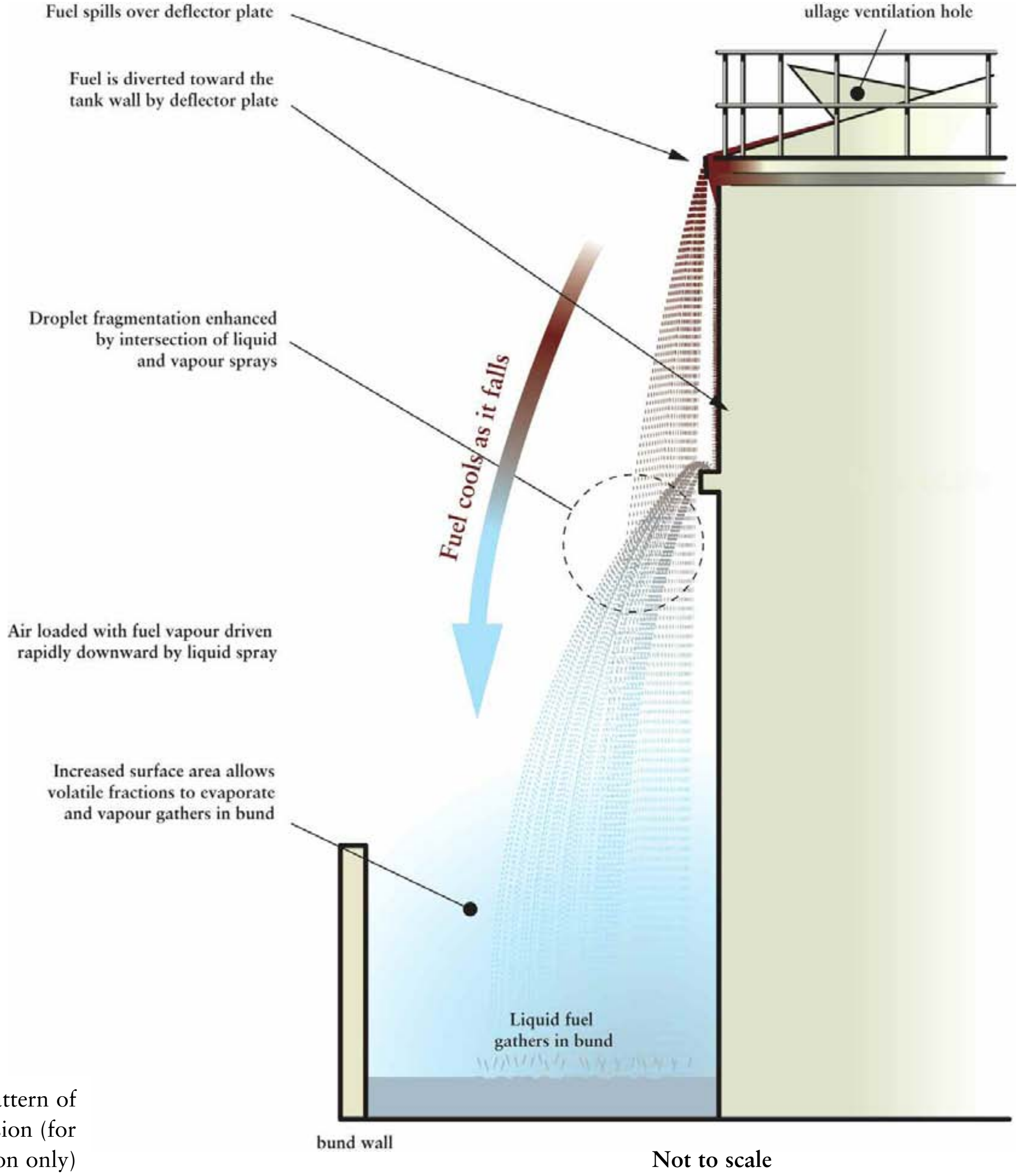


Figure 2 The pattern of fuel dispersion (for illustration only)

VAPOR CLOUD FORMATION

MOST OF THE GROUND AREA NEAR THE BLOWDOWN DRUM IS COVERED BY VERTICAL OR HORIZONTAL EQUIPMENT, PIPE RACKS WITH MULTIPLE PIPES, CONCRETE PADS WITH CURBS, ETC. THERE IS LIMITED FLAT, HORIZONTAL AREA IN WHICH A POOL OF LIQUID COULD FORM AND SPREAD IN A CONCENTRIC CIRCLE. CORRELATIONS, TYPICALLY USED BY OTHERS, TO DETERMINE THE MASS OF THE FLAMMABLE VAPOR CLOUD WITHIN A FIXED TIME PERIOD WILL NOT NECESSARILY APPLY. ANOTHER MECHANISM, OTHER THAN A LARGE DIAMETER POOL, APPEARS TO BE REQUIRED TO ACCOUNT FOR THE AMOUNT OF LIQUID REQUIRED TO HAVE VAPORIZED BY EVAPORATION WITHIN A LIMITED TIME PERIOD TO FORM THE FLAMMABLE VAPOR CLOUD OF THE SIZE REQUIRED TO EXPLAIN THE PLANT DAMAGE.

VAPOR CLOUD FORMATION

A SUBSTANTIAL PORTION OF THE SUBCOOLED LIQUID THAT FLOWED FROM THE STACK ON THE BLOWDOWN DRUM FELL ONTO NEARBY PROCESS EQUIPMENT AND PIPES IN PIPE RACKS BASED ON THE WIND DIRECTION AT THE TIME OF THE INCIDENT. SOME OF THE LIQUID WOULD ALSO HAVE FALLEN TO THE GROUND. THE ONGOING INCIDENT INVESTIGATION ANALYSIS OF A RECENT GASOLINE SPILL [BUNCEFIELD] SUGGESTS THAT LIQUID CASCADING DOWN THE SIDE OF EQUIPMENT DIVIDES INTO SMALL DROPLETS. SPLASHING UPON IMPACT OF FALLING LIQUID WITH EQUIPMENT, PIPES AND THE GROUND ALSO PROMOTES FRAGMENTATION INTO RELATIVELY SMALL DROPLETS AND EVAPORATION. THE FREE FALL OF DROPLETS THROUGH AIR LEADS TO ENTRAINMENT OF AIR AND WATER VAPOR. THESE CONDITIONS PROMOTE EVAPORATION AND FORMATION OF A VAPOR CLOUD. THE VAPORIZATION (EVAPORATION) AND VAPOR CLOUD DISPERSION COULD APPROACH THAT RESULTING FROM THE MECHANICAL BREAKUP OF A HIGH MOMENTUM SUBCOOLED LIQUID RELEASE, WHICH IS TREATED BY EXISTING VAPOR CLOUD DISPERSION MODELS. THERE WOULD ALSO BE SOME CONTRIBUTION TO THE VAPOR

CLOUD DUE TO FORCED CONVECTION FROM THE GROUND LEVEL POOL. VAPORIZATION OF THE LIQUID ON THE WETTED EQUIPMENT AND PIPES SURFACES AND THE AND GROUND DUE TO THE INITIAL FIREBALL AND EXPLOSION STAGES WOULD ALSO HAVE CONTRIBUTED TO THE FLAMMABLE VAPOR CLOUD THAT WAS INITIALLY PARTIALLY DISPERSED AND WAS THEN FURTHER MOVED AND ACCELERATED BY THE EXPLOSION AND PARTIAL CONFINEMENT AND CONGESTION OF THE UNIT.

THE PORTION OF A FLAMMABLE VAPOR CLOUD BOUNDED BY THE LOWER AND UPPER FLAMMABLE LIMITS IS CALCULATED BY AN APPROPRIATE VAPOR DISPERSION MODEL. SEVERAL OF THESE MODELS ARE BASED ON THE PREMISE THAT A VAPOR CLOUD EXPLOSION CAN OCCUR ONLY WITHIN THAT PORTION OF A FLAMMABLE CLOUD THAT IS CONGESTED OR PARTIALLY CONFINED.

A FLAMMABLE LIQUID WAS ON THE GROUND IN THE VICINITY OF THE BASE OF THE RAFFINATE SPLITTER BLOWDOWN DRUM. THE FLAME FROM THE VAPOR CLOUD IGNITED THE LIQUID. A POOL FIRE RESULTED. THE FIRE WAS OF SUFFICIENT DURATION AND PRODUCED SUFFICIENTLY HIGH TEMPERATURES IN COMBINATION WITH THE FIRE FIGHTING WATER SPRAY TO HAVE CAUSED THE CONCRETE OF THE PIPE RACK COLUMNS TO SPALL.

SUBJECT: “GENERAL OBSERVATIONS”

AS THE BP INVESTIGATION CONCLUDES AND REPORTS ARE BEING WRITTEN, THE FOLLOWING OBSERVATIONS ARE OFFERED FOR CONSIDERATION BY THE CHEMICAL SAFETY BOARD:

BUILDINGS (IN ADDITION TO CONTROL ROOMS AND TRAILERS)

1. SECURITY VULNERABILITY ANALYSIS SUGGESTS INSTALLATION OF PLASTIC SHEETING ON WINDOWS TO PREVENT SHATTERING OF THE GLASS DUE TO A BLAST.

2. REINFORCEMENT OF WINDOW FRAMES TO PROVIDE ADDITIONAL BLAST PROTECTION IS OFTEN DESIRABLE.

3. THE APPLICATION OF CALIFORNIA EARTHQUAKE STANDARDS SHOULD BE CONSIDERED FOR STABILIZATION OF OVERHEAD FIXTURES (LIGHTING, PROCESS MONITORS, VENTILATION DUCT WORK, ETC) WITHIN BUILDINGS DURING A BLAST.

4. THE WALLS OF METAL-SIDED BUILDINGS AND THEIR TYPICAL INTERNAL PARTITIONS WALLS PROVIDE VERY LITTLE PROTECTION AGAINST A BLAST AT VERY SIGNIFICANT DISTANCES.

5. BARRIERS ARE TYPICALLY SUGGESTED TO PROVIDE OFFSET SEPARATION DISTANCES FOR BUILDINGS FROM MOBILE BLAST SOURCES. THE MESSAGE MAY BE THAT SOME BUILDINGS CONTAINING NON-ESSENTIAL PERSONNEL SHOULD NOT BE LOCATED IN A PLANT WITHIN THE SIGNIFICANT DAMAGE DISTANCE OF A POTENTIAL BLAST.

   NOTE: VAPOR CLOUDS MOVE AND CAN RESULT IN SIGNIFICANT DAMAGE AT LOCATIONS REMOTE FROM THE ORIGINAL SOURCE OF THE FLAMMABLE MATERIAL RELEASE.

DISPERSION MODELS

1. THE SIZES OF POOLS USED TO MODEL / EXPLAIN THE FORMATION OF FLAMMABLE VAPOR CLOUDS BY EVAPORATIVE VAPORIZATION MUST BE REALISTIC. IF YOU DO NOT HAVE A LARGE OPEN AREA, YOU SHOULD NOT USE A LARGE DIAMETER POOL WHEN MODELING A FLAMMABLE MATERIAL RELEASE INCIDENT WITHIN A CONGESTED AREA.

2. MODELS FOR SPLASHING AND EVAPORATION OF LOW MOMENTUM, SUBCOOLED LIQUID RELEASES DO NOT APPEAR TO BE ADEQUATELY TREATED IN THE LITERATURE OR INCLUDED IN THE DISPERSION MODELS TYPICALLY USED TO PREDICT VAPOR CLOUD DEFLAGRATIONS AND BLAST DAMAGE. MORE RESEARCH IS NEEDED.

PHRA REVALIDATIONS

1. ORIGINAL PHRA PREPARATION AND REVALIDATION SHOULD BE THE WORK OF EXPERTS WITH EXPERTISE IN THE CRITICAL AREAS BEING EXAMINED. CALCULATIONS, RATHER THAN ENGINEERING JUDGMENTS, SHOULD BE THE BASIS FOR FINDINGS AND CONCLUSIONS.

2. A PHRA REVALIDATION SHOULD NOT BE CONSIDERED TO BE COMPLETE UNLESS THE CALCULATIONS THAT SUPPORT THE ORIGINAL FINDINGS AND CONCLUSIONS ARE EXAMINED BY THE PRESENT TEAM.

3. THE OCCURRENCE OF A MAJOR PROCESS HAZARD IS, HOPEFULLY, A RARE EVENT ON THE ORDER OF ONE IN THOUSANDS OF YEARS. EXPERIENCE IS THEREFORE NOT A QUALIFIER OF EXPERTISE. A RIGOROUS, SCIENTIFIC METHODOLOGY IS USUALLY REQUIRED TO QUANTIFY THE RISK (BOTH FREQUENCY AND CONSEQUENCE) OF RARE EVENTS.

4. THE GREATER THE PERCEIVED CONSEQUENCE OF A POTENTIAL EVENT, THE MORE EFFORT THAT IS USUALLY REQUIRED TO ADEQUATELY EXAMINE THAT EVENT.

5. PHRA REVALIDATIONS MUST NOT ONLY REVIEW THE FINDINGS OF THE ORIGINAL TEAM, BUT ALSO WHAT THE FIRST TEAM MISSED AND THE CHANGED CIRCUMSTANCES, TECHNOLOGY IMPROVEMENTS, AND PERCEPTION OF ACCEPTABLE RISK THAT OCCUR OVER TIME.

SAFETY RELIEF VALVES

1. SAFETY RELIEF VALVES ARE NOT PRESENTLY REQUIRED TO BE CERTIFIED FOR TWO-PHASE, VAPOR-LIQUID FLOW BY THE NATIONAL BOARD OF BOILER AND PRESSURE INSPECTORS. THE INTERNAL CONSTRUCTION DIFFERENCES BETWEEN THE VALVE MODELS OF VARIOUS MANUFACTURERS CAN RESULT IN A FLOW VARIATION OF A FACTOR OF TWO TO THREE. THE CALCULATION OF PIPE PRESSURE DROPS AND EFFLUENT FLOWS IS HIGHLY UNCERTAIN.

2. THE KB AND KW BACKPRESSURE FLOW REDUCTION CURVES PUBLISHED FOR USE WITH BELLOWS SAFETY RELIEF VALVES BY THE MANUFACTURERS AND THE AMERICAN PETROLEUM INSTITUTE (API) ARE NOT BACKED BY PUBLISHED DATA, ARE TECHNICALLY INCONSISTENT, APPLY TO VALVES NOT LONGER PERMITTED FOR USE BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS (ASME) CODE, AND DO NOT APPLY TO TWO-PHASE, VAPOR-LIQUID FLOW.

DIERS Users Group Mtg.
April, 2008   Las Vegas Nevada

“Odds and Ends”
Relief Valve Stability
And
Inlet Pressure Loss—
“Searching for Understanding”

Michael A. Grolmes

Centaurus Technology, Inc.

4590 Webb Road

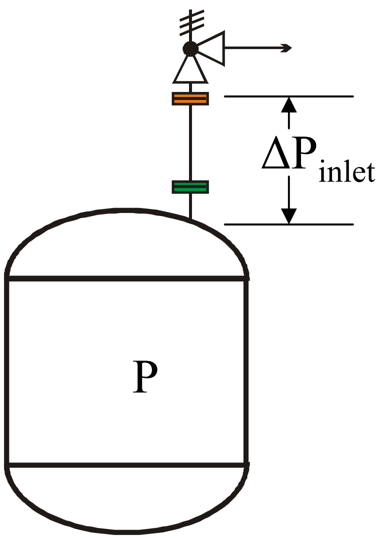
Simpsonville, KY 40067

502-243-9678

ctimag@att.net

Centaurus Technology, Inc.

Relief Valve Inlet Pressure Loss



$LET:$

$$\Delta P_{inlet} = x\, P_{set\_g} \quad where \quad 0 \le x \le 1.0$$

and

$$P = 1.1\, P_{set\_g} + P_{atm}$$

$Then$

$$\Delta P_{inlet} = \frac{1}{2}\rho\, U^2\, K \quad where \quad K = 4\, f\, (L/D)_{eq}$$

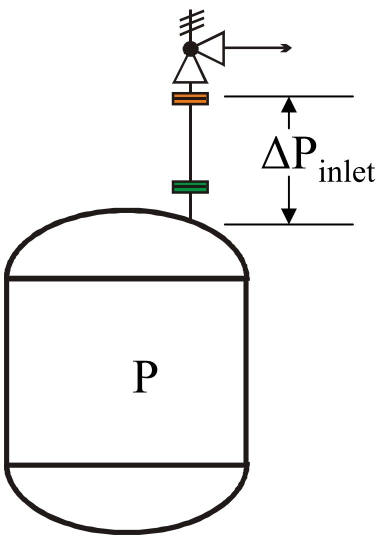
but

$$\rho\, U^2 = \frac{1}{\rho}\left(\frac{w}{A}\right)^2$$

and

$$w = C_d A_o G_n$$

Relief Valve Inlet Pressure Loss *(continued)*



$$Now:$$

$$\Delta P_{inlet} = \frac{1}{2}\left[\frac{C_d\ A_o}{A_1}\right]^2 \frac{G_n^2}{\rho} K = x\ P_{set\_g}$$

also define

$$A^* = \frac{A_1}{C_d\ A_o}$$

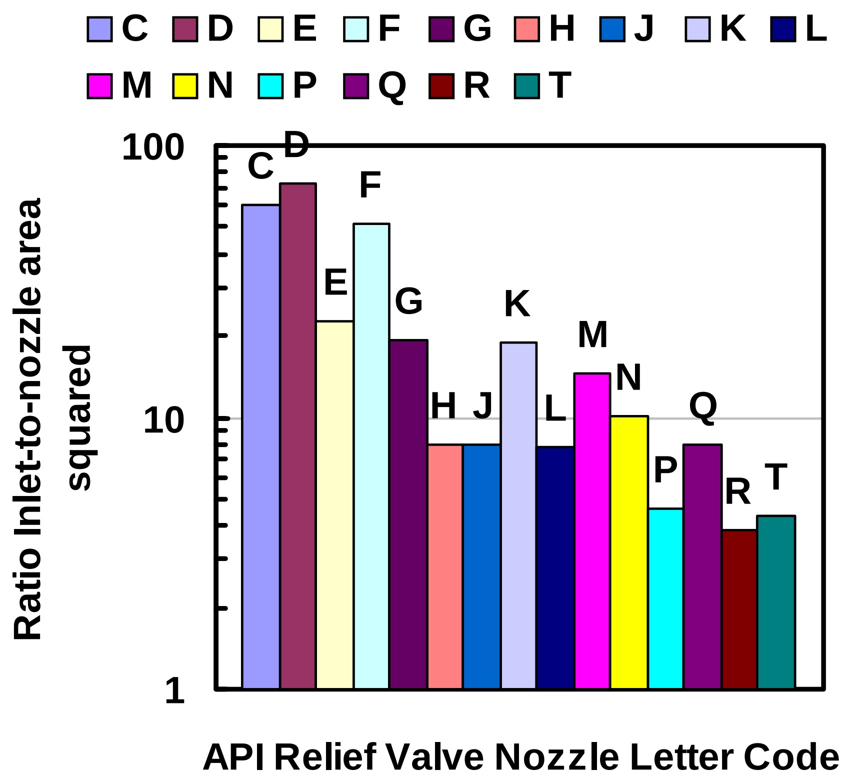
So that

$$\frac{K}{(A^*)^2} \leq \frac{2\ x\ P_{set\_g}}{\frac{G_n^2}{\rho}}$$

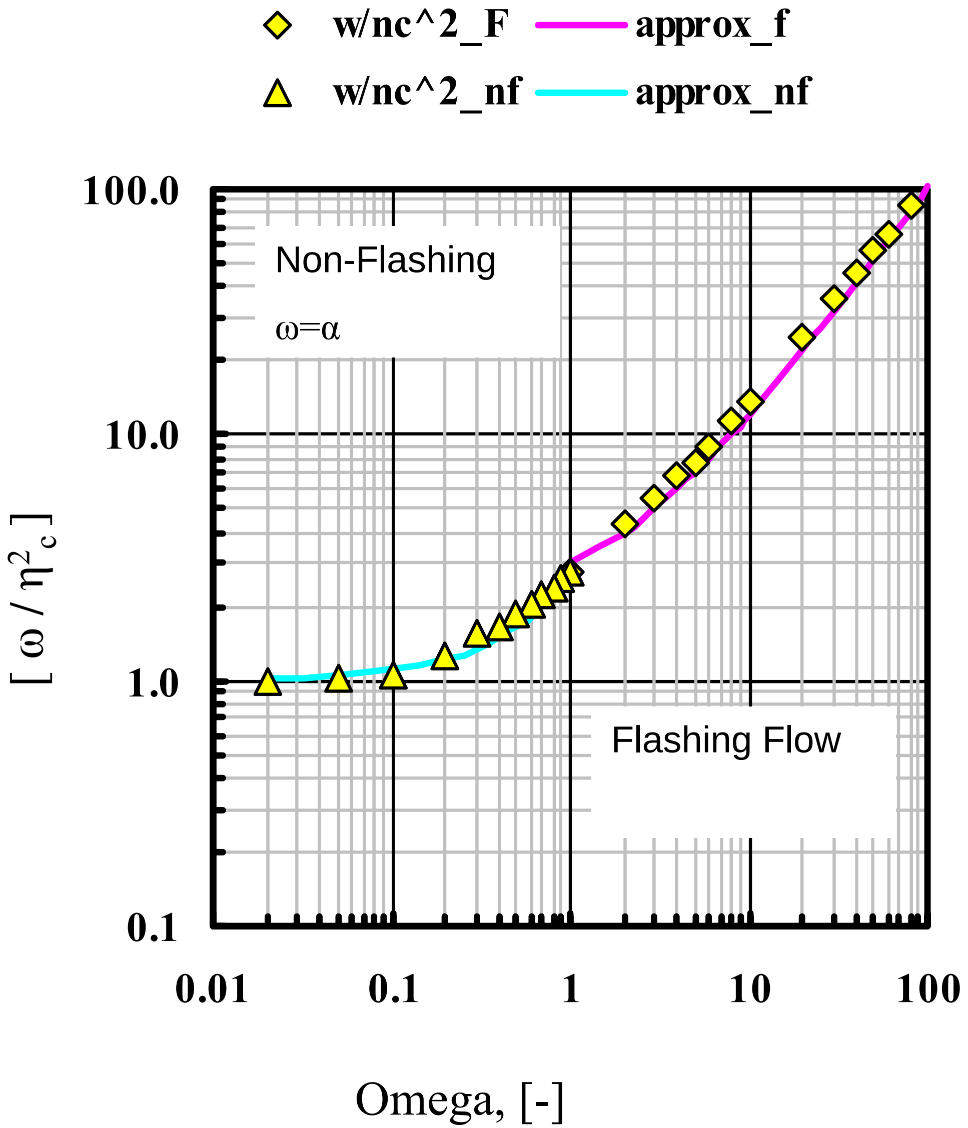
Relief Valve Inlet Pressure Loss
(summary table)

CASE	GAS	LIQUID	Two-Phase
G_n	$\dfrac{\varepsilon\, P}{\sqrt{z\, R\, T\, /\, Mw}}$	$\sqrt{2\, \rho\, 1.1\, P_{set\_g}}$	$\dfrac{\eta_c}{\sqrt{\omega}}\, \sqrt{\rho_{2\varphi}\, P}$
$\dfrac{G_n^2}{\rho}$	$\varepsilon^2\, P$	$2\ \ 1.1\ P_{set\_g}$	$\dfrac{\eta_c^2}{\omega}\, P$
$\dfrac{K}{(A^*)^2}$	$\dfrac{2\, x\, /\, \varepsilon^2}{\left[1.1 + \dfrac{P_{atm}}{P_{set\_g}}\right]}$	$\dfrac{x}{1.1}$	$\dfrac{[2\, x\, \omega\, /\, \eta_c^2]}{\left[1.1 + \dfrac{P_{atm}}{P_{set\_g}}\right]}$

Relief Valve Nozzle Letter Size Code	API Nozzle Area A_o	Inlet Pipe Area A_1	$\left[\frac{A_1}{C_d\ A_o}\right]$	$\left[\frac{A_1}{C_d\ A_o}\right]^2$
	[sq. inch]	[sq in]	[-]	[-]
C	0.074	0.533	7.791	60.69
D	0.110	0.864	8.494	72.15
E	0.196	0.864	4.767	22.72
F	0.307	2.036	7.169	51.40
G	0.503	2.036	4.376	19.15
H	0.785	2.036	2.804	7.86
J	1.287	3.356	2.819	7.95
K	1.838	7.393	4.348	18.91
L	2.853	7.393	2.801	7.85
M	3.600	12.730	3.823	14.61
N	4.340	12.730	3.171	10.06
P	6.380	12.730	2.157	4.65
Q	11.050	28.890	2.826	7.99
R	16.000	28.890	1.952	3.81
T	26.000	50.027	2.080	4.33



These are A* values for conventional safety relief valves.



Approximation for:

Non-Flashing Flow

$$\frac{\omega}{\eta_c^2} = \exp(\omega)$$

Approximation for:

Flashing Flow

$$\frac{\omega}{\eta_c^2} = \omega + 2$$

Example Application

Set-up Parameters

$x = 0.03$ (3% rule)

$\varepsilon = 0.65$ (useful in most cases)

$P_{atm} = 14.7$ psia

$P_{set\_g} = 50$ psig

For Flashing Two-Phase Flow:

$\omega = 20$ and $\eta_c = 0.895$

$(\omega / \eta_c^2) = 25$

For Non-Flashing Two-Phase Flow:

$\omega = 0.4$ and $\eta_c = 0.4$

$(\omega / \eta_c^2) = 1.25$

Case	$K/(A^*)^2$ Must be < =
Gas	0.102
Liquid	0.0273
2-P Flashing	1.076
2-P Non-Flashing	0.054

Example Application (continued)

Now, Extend Example to 4P6

Relief valve with

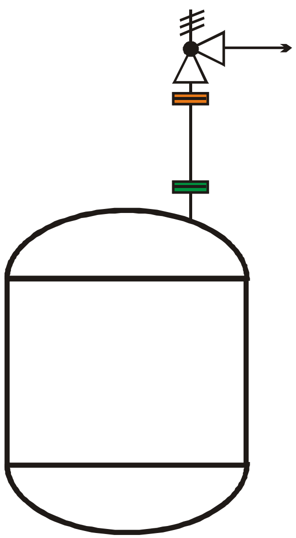
$(A^*)^2 = 4.65$

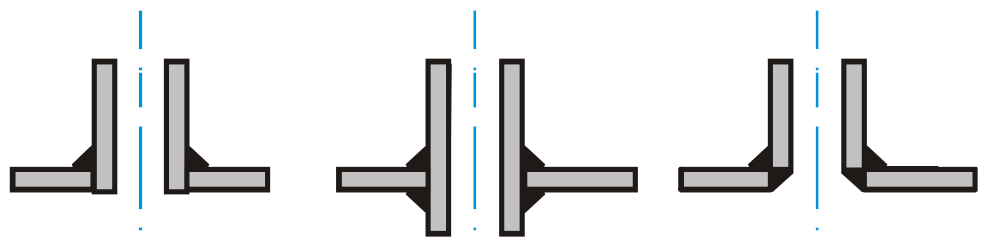
These are 3% rule

Limits for 4P6 at 50 psig set pressure.

Case	K Must be < =	$(L/D)_{eq}$ Must be < =
Gas	0.474	30
Liquid	.127	8
2-P Flashing	5.0	312
2-P Non-Flashing	.251	16

Entrance Configurations and Entrance Loss Coefficients





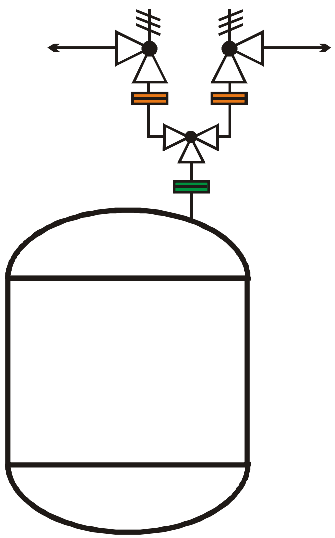
Square Cut Entrance $K_e = 0.5$

Re-entrant Entrance $K_e = 0.8$

Rounded Entrance $K_e = 0.04$

One could be dead in the water with either square-cut or re-entrant configuration

Entrance Loss due to 3-Way Valve



Relief Valve Nozzle Letter Size Code	API Nozzle Area A_o	inlet x outlet flange pipe size	Brand#1 3-way valve	Inlet Del P P	Brand#2 3-way valve	Inlet Del P P
	[sq. inch]	150#	Cv	[%]	Cv	[%]
C	0.074	3/4 x 1	7	3.49	14	0.87
D	0.110	1 x 2	20	0.94	22	0.78
E	0.196	1 x 2	20	3.00	22	2.48
F	0.307	1 1/2 x 2	40	1.84	57	0.91
G	0.503	1 1/2 x 2 1/2	40	4.94	57	2.43
H	0.785	1 1/2 x 3	40	12.02	57	5.92
J	1.287	2 x 3	70	10.55	110	4.27
K	1.838	3 x 4	100	10.54	260	1.56
L	2.853	3 x 4	100	25.41	260	3.76
M	3.600	4 x 6	175	13.21	446	2.03
N	4.340	4 x 6	175	19.20	446	2.96
P	6.380	4 x 6	175	41.49	446	6.39
Q	11.050	6 x 8	350	31.11	-	-
R	16.000	6 x 8	350	65.23	-	-
T	26.000	8 x 10	475	93.52	-	-

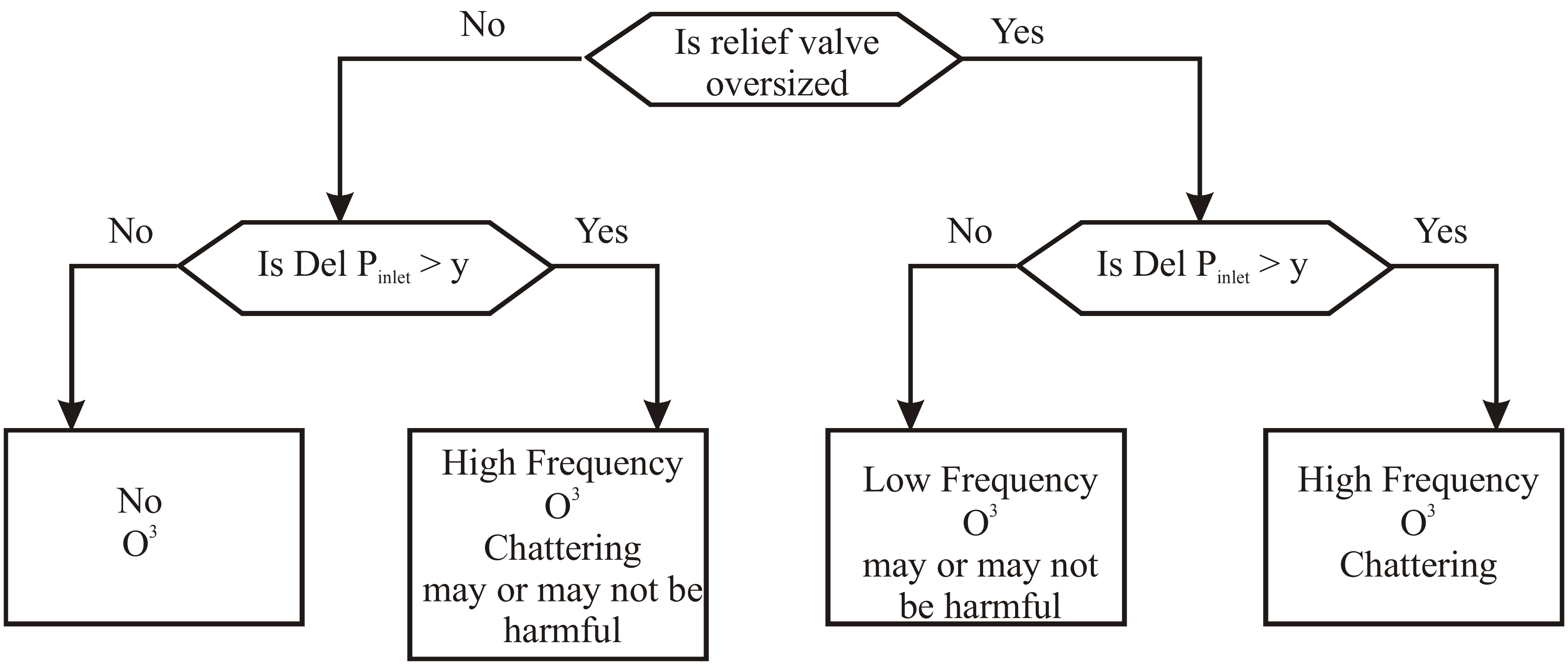
3% Rule Summary:

- Many entrance configurations won't work with the vessel nozzle size the same as the relief valve inlet size.
- Many 3-way valves won't work with the 3-way valve size the same as the relief valve inlet size.
- So how important is the 3% rule?
- What has this to do with relief valve on-off instability?

Relief Valve Instability
or
ON-OFF OPERATION (O^3)

- O^3 can occur if valve is oversized for the demand
- O^3 can occur with excess inlet pressure loss
- It is generally believed that O^3 can be harmful to the relief valve if such leads to cyclic on-off operation at the relief valve natural frequency

Possible Valve Sizing Outcomes



Note that only one out of four possible outcomes avoids O^3

And one does not really know what the value of "y" is.

A Conjecture

So let us define "truth" as being a correct statement for most circumstances.[1] Then we conjecture that the following statements are true:

(a) Del P_{inlet} < 0.03 Pset is a sufficient condition for avoiding high frequency O^3 - <u>but not a necessary condition</u>

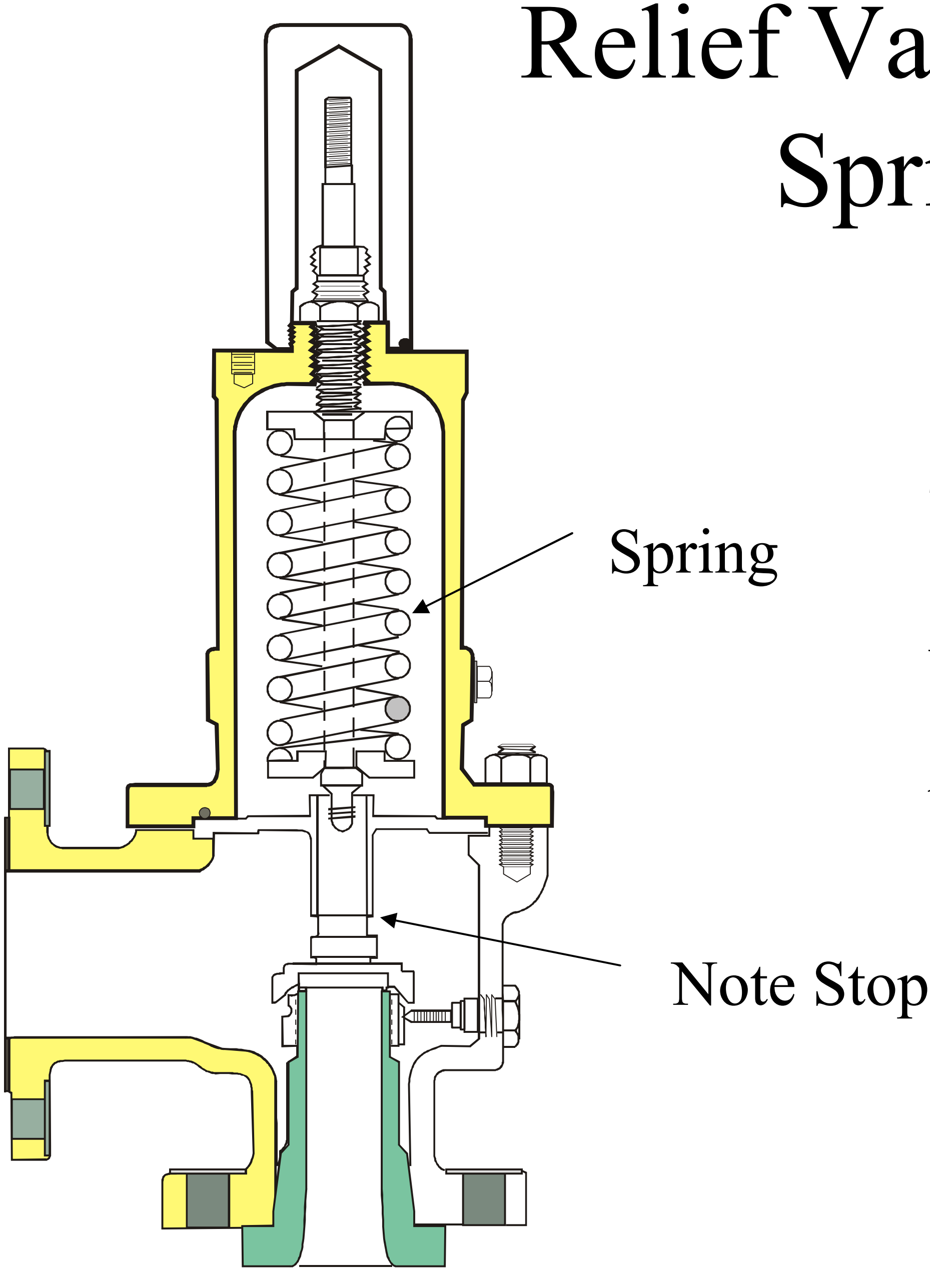
(b) Del P_{inlet} < 0.03 Pset is neither sufficient or necessary for avoiding O^3 for an oversized relief valve.

(1) Note: this being an election year, this criterion for "truth" exceeds all threshold levels for truth in current political discourse.

Relief Valve Operation as a Spring –Mass System

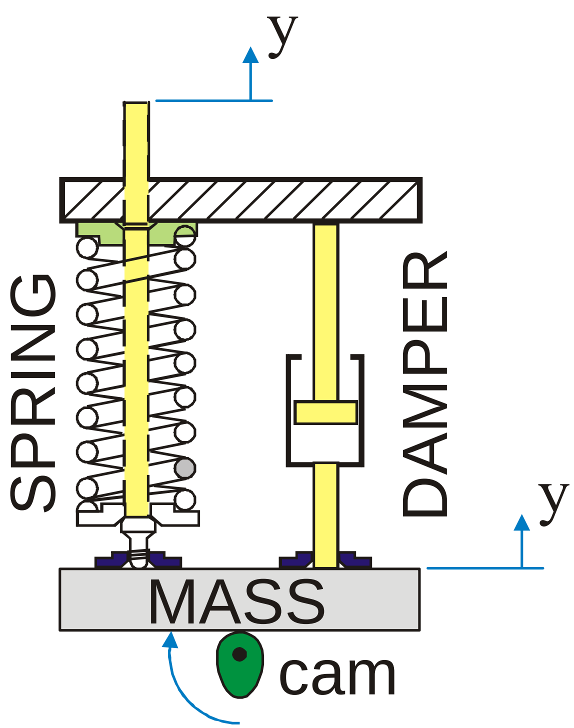
Implicit in the preceding comments is the notion that high frequency O^3 is related to resonance with the relief valve natural frequency. <u>And we turn to this issue next.</u>

Relief Valve Operation as a Spring –Mass System



The mass is there, but not in one lump. There is the valve cap, a slider piece, the spring rod, a restraining button and some part of the spring itself.

Textbook Spring – Mass – Damper System



$$m\frac{d^2y}{dt^2} + c\frac{dy}{dt} + k\,y \;=\; F_o \sin(\omega t)$$

has solution

$$y \;=\; Y\sin(\omega t - \varphi)$$

where the amplitude factor Y

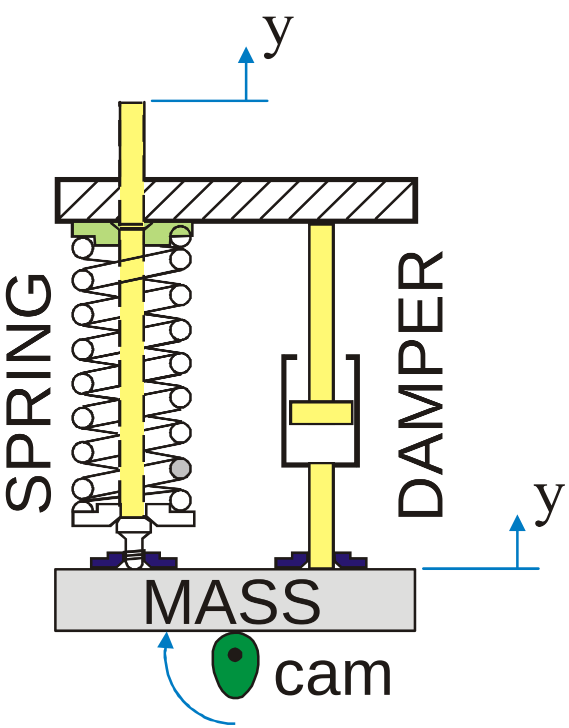
$$Y = \frac{F_o / k}{\left\{\left(1 - \dfrac{m\,\omega^2}{k}\right)^2 + \left(\dfrac{c\,\omega}{k}\right)^2\right\}^{1/2}}$$

m – mass

c – damping coefficient

k – spring constant

F_o - force



m – mass

c – damping coefficient

k – spring constant

F_o - force

The phase shift angle

$$\varphi = \tan^{-1}\left(\frac{c\,\omega}{k - m\,\omega^2}\right)$$

and the natural frequency is

$$\omega_n^2 = \frac{k}{m}$$

The dimensionless *damping factor is*

$$DF = \frac{c}{m\,\omega_n} = \frac{c}{\sqrt{k\,m}}$$

Now let

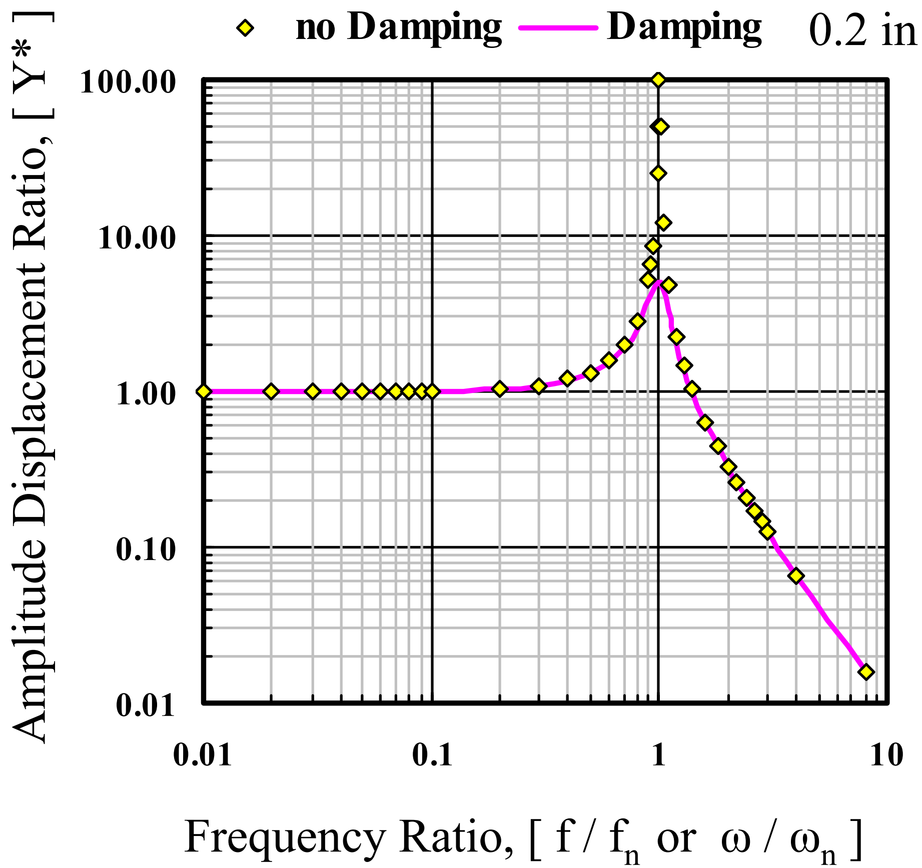
$$Y^* = \frac{Y}{F_o / k}$$

So that

$$Y^* = \frac{1}{\left\{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(DF\,\frac{\omega}{\omega_n}\right)^2\right\}^{1/2}}$$

Amplitude Response for Spring – Mass System with and without damping

Damping Factor = 0.2 in illustration



If the damping factor is 0.2, the phase shift angle is less than 4°

Frequency of the forcing function to the system natural frequency

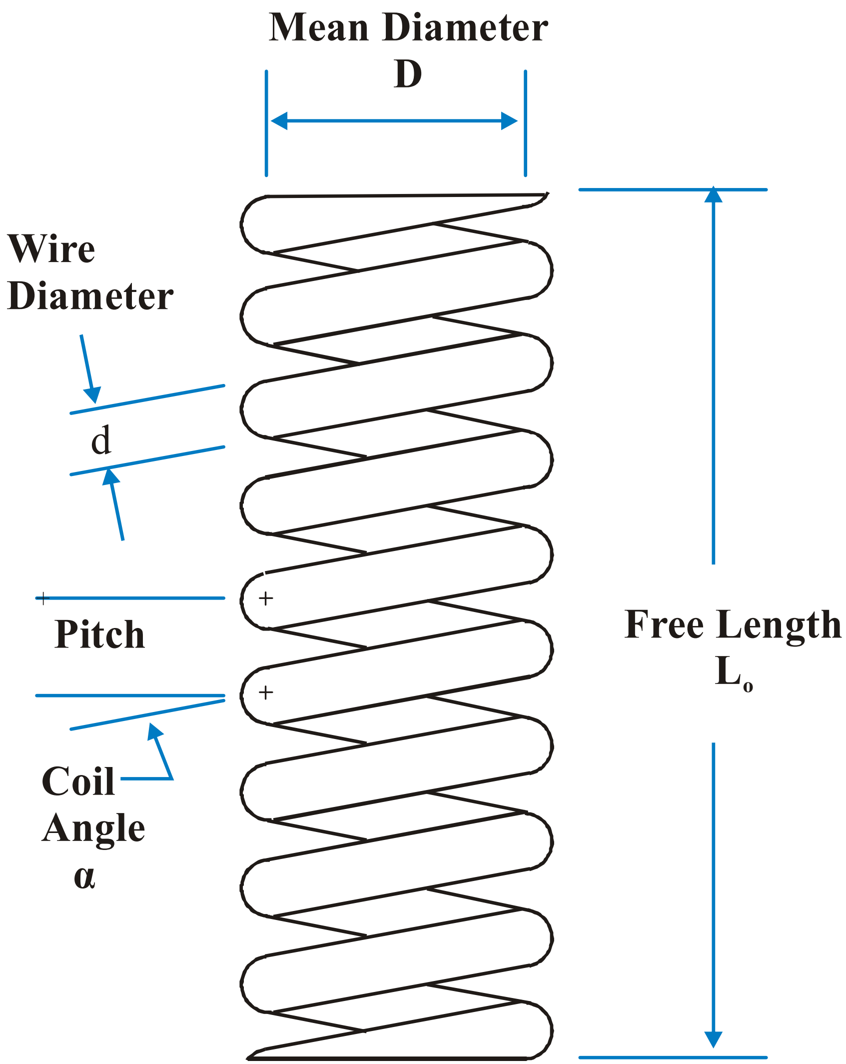
Summary of this Part:

If one wishes to avoid the resonant frequency, one can do so without any further coupling of the spring mass system to the rest of the problem.

However, to avoid resonance, one needs to know the <u>spring constant</u> (k), and the <u>mass</u> (m), in order to identify the <u>natural frequency</u> (ω_n)

This we take up next.

Helical Springs

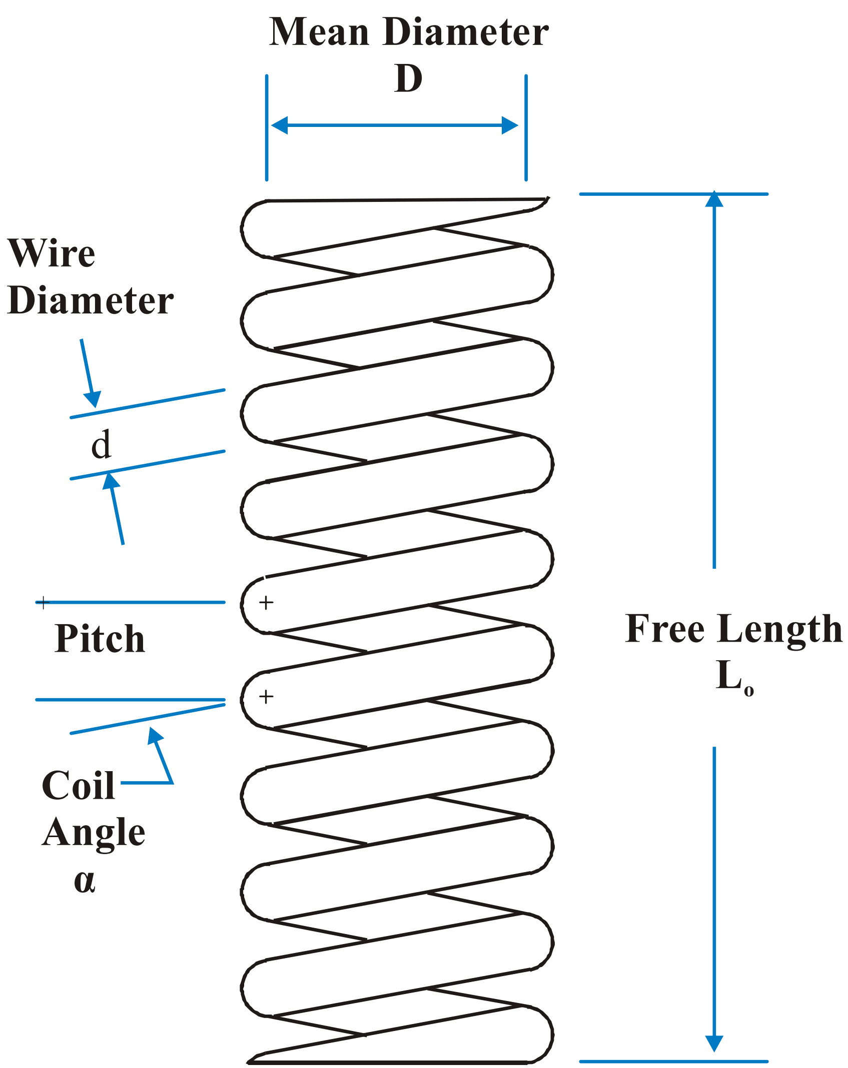


Relief Valves typically

have springs with closed ends ground such that the number of active coils is $Na = Nt - 2$

Where Nt is the total number of coils

Helical Springs



Nomenclature:

N_a is the number of active coils

G is the modulus of torsion or rigidity

C is the diameter modulus; C= D/d

D is the mean diameter

d is the coil diameter

Spring Constant k

$$k = \frac{G\,d}{8\,C^3\,N_a}$$

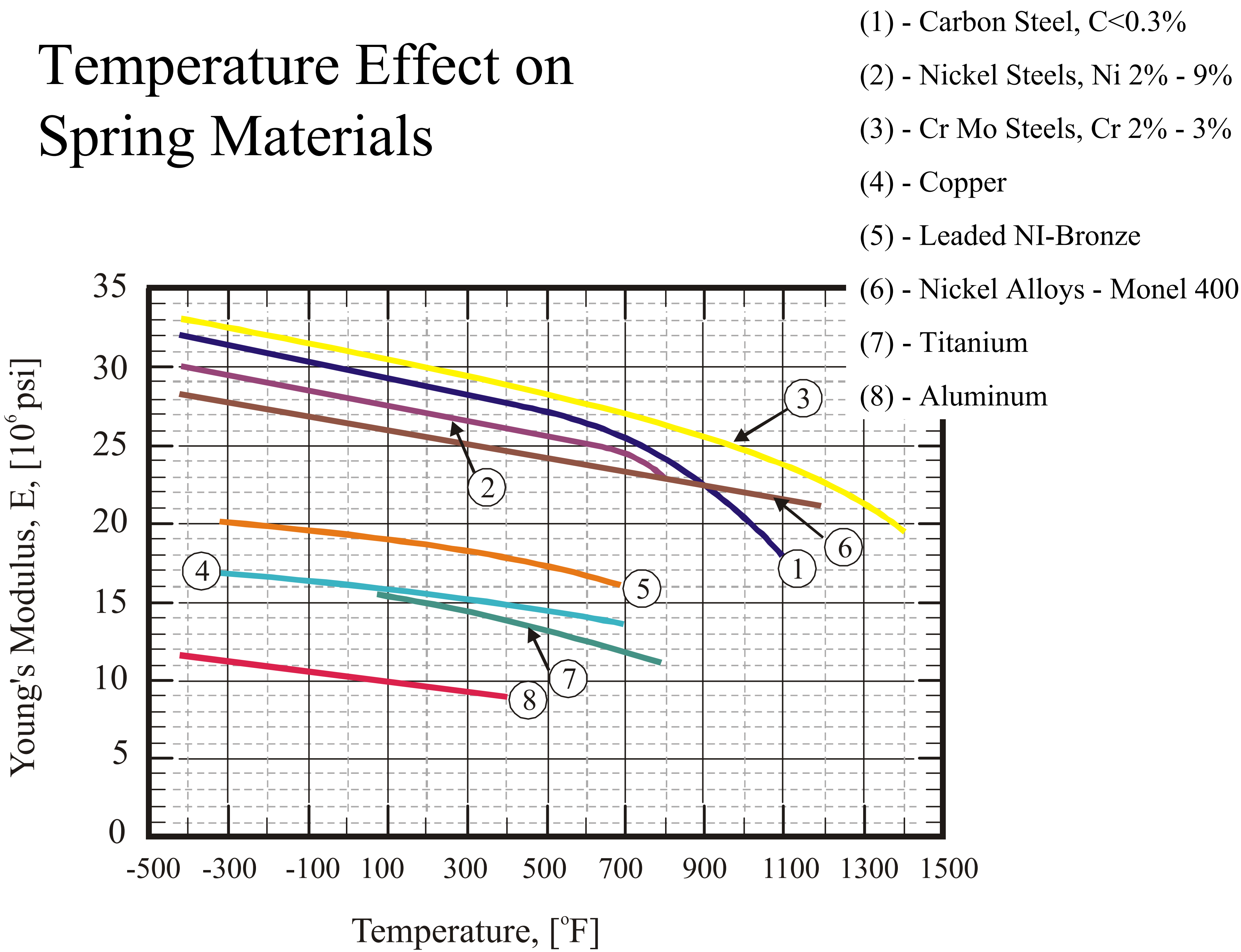
Young's Modulus (E) and Modulus of Torsion (G)

Metal	Poisson's Ratio	E [10^9 Pa]	G [10^9 Pa]
Aluminum	0.330	69	27
Copper	0.360	117	43
Ni-Steel	0.310	213	76
Stainless Steel 18-8	0.300	201	73
Carbon Steel	0.303	202	79
High Carbon Steel	0.295	210	81
Inconel	0.290	214	79

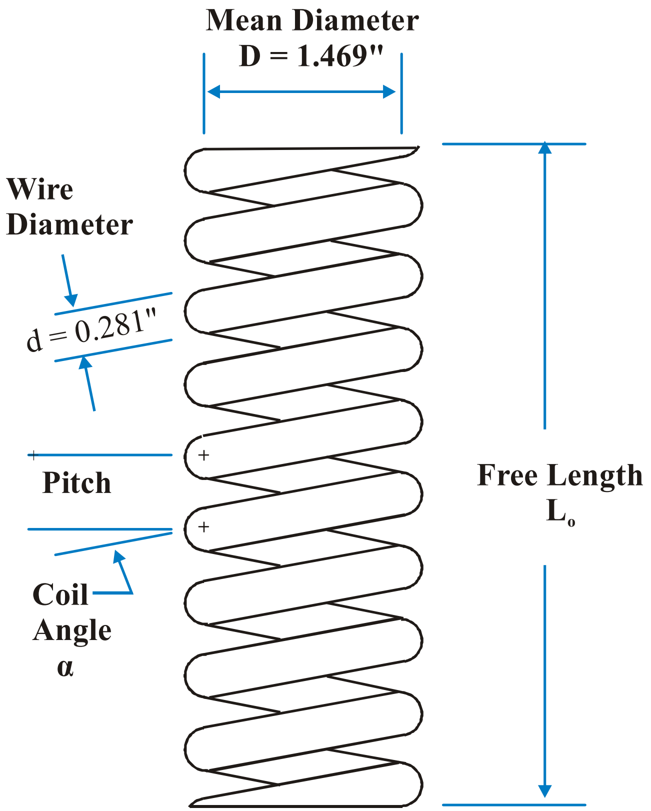
$$G = \frac{E}{2(1+v)}$$

Poisson's Ratio

Temperature Effect on Spring Materials



Real Example:

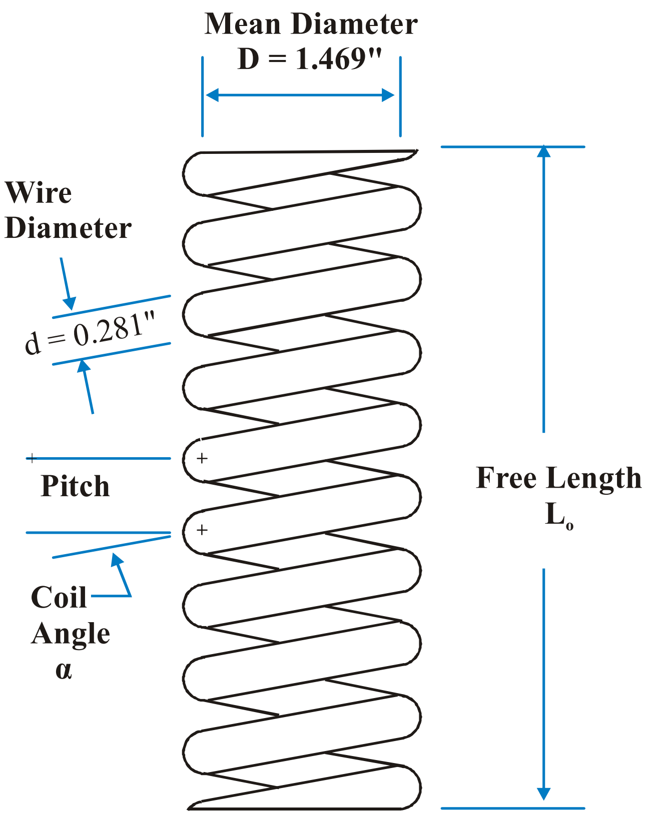


Spring from
Farris 26 FA 10 - 120
180 psig set

$N_a = 8$
$C = D/d = 5.288$
Assume:
$E = 205$ GPa or 29.7×10^6 psi
$\nu = 0.31$
$G = 78.2$ GPa or 11.35×10^6 psi

$$k = \frac{11.35E6\, x\, 0.281}{8\, x\, 5.288^3\, x\, 8} = 349\, \frac{lbf}{in}$$

Real Example: (continued)



Spring from
Farris 26 FA 10 - 120
180 psig set

Wgt. of Spring = 360 gm
Wgt. of other
parts in motion = 435 gm
Assume mass in motion is
435gm + 360 / 3 = 555 gm
or approximately 1.22 lb

Real Example: (continued)

So now we have

k = 350 lb_f / inch or 61294 N / meter

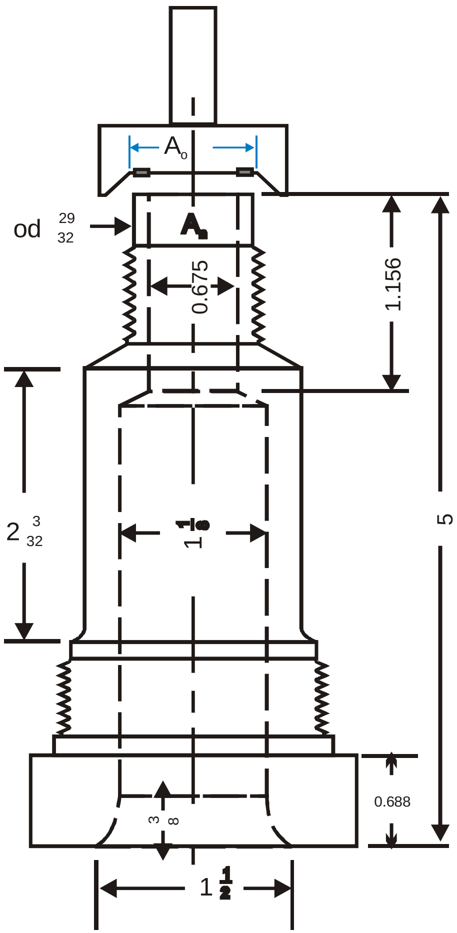
And

m = 0.555 kg

Therefore the natural frequency is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{61294}{0.555}} = 53\,Hz$$

Can we do better ??



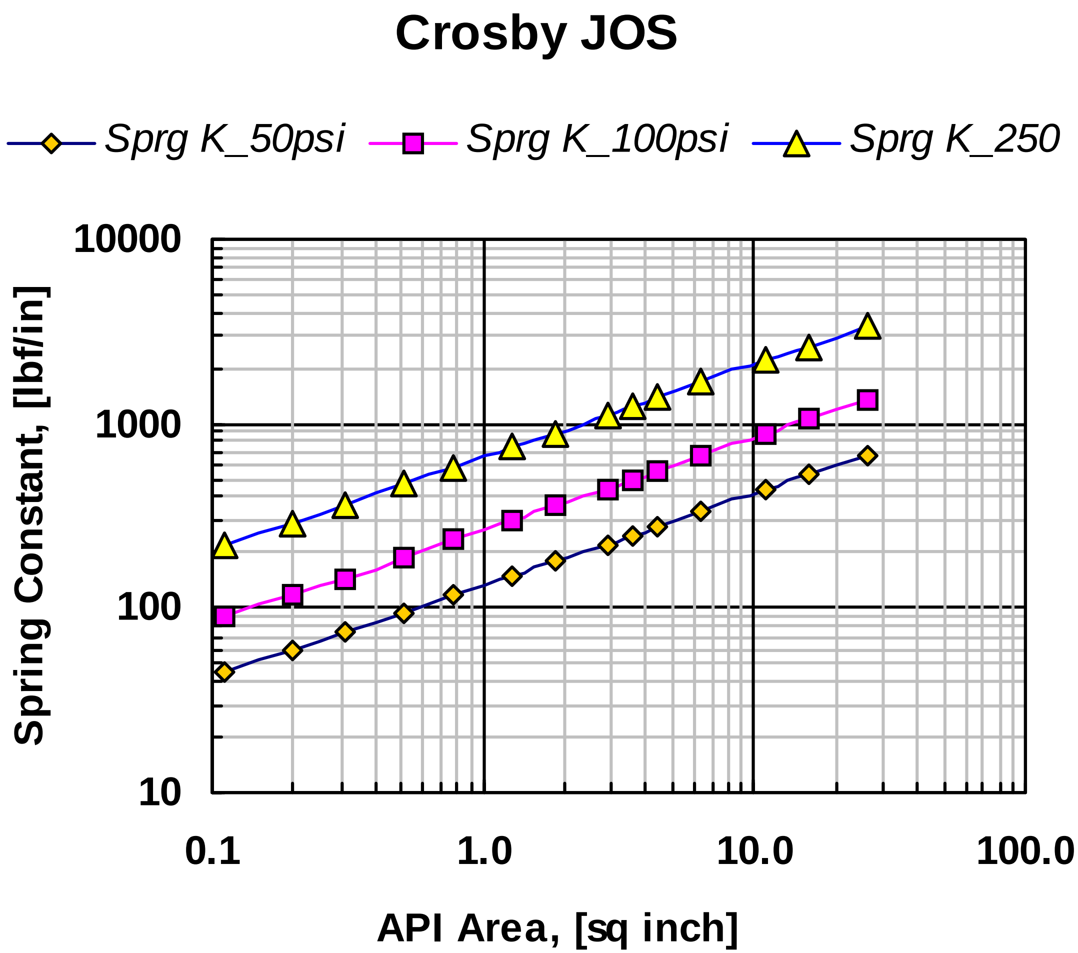
By examining the geometry of the nozzle and cap and doing a little calibration with the known example one can arrive at the following relations:

$$k = 1.08 \frac{P_{set} \; A_{redbook}}{y_{lift}}$$

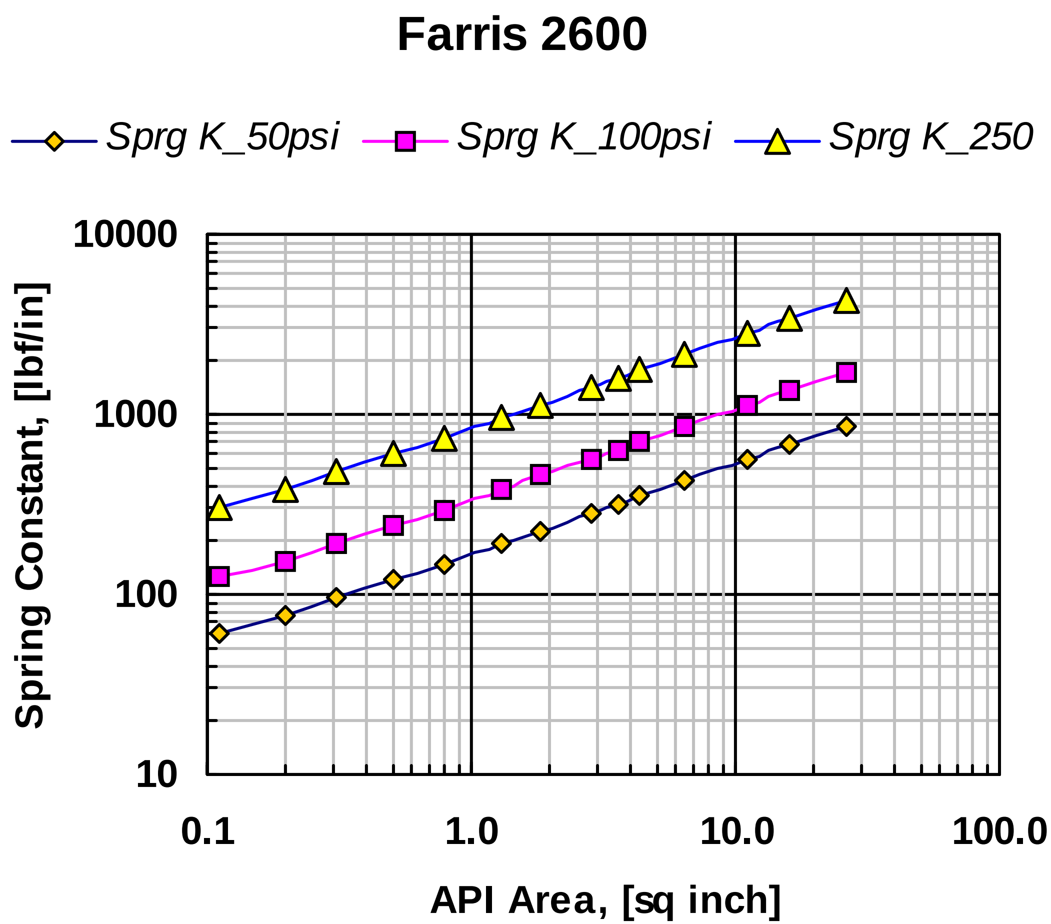
and

$$m = 0.025 \, M_{valve}$$

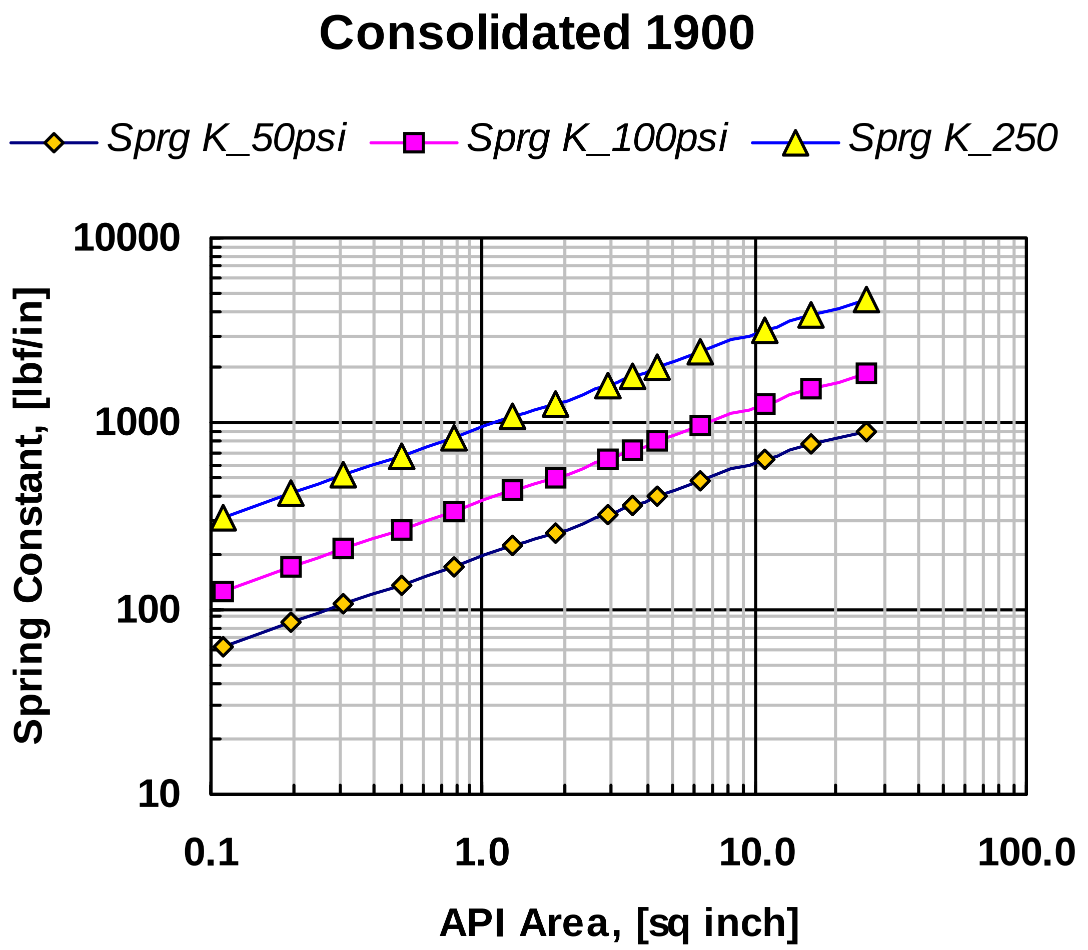
Relief Valve Spring Constant Estimate



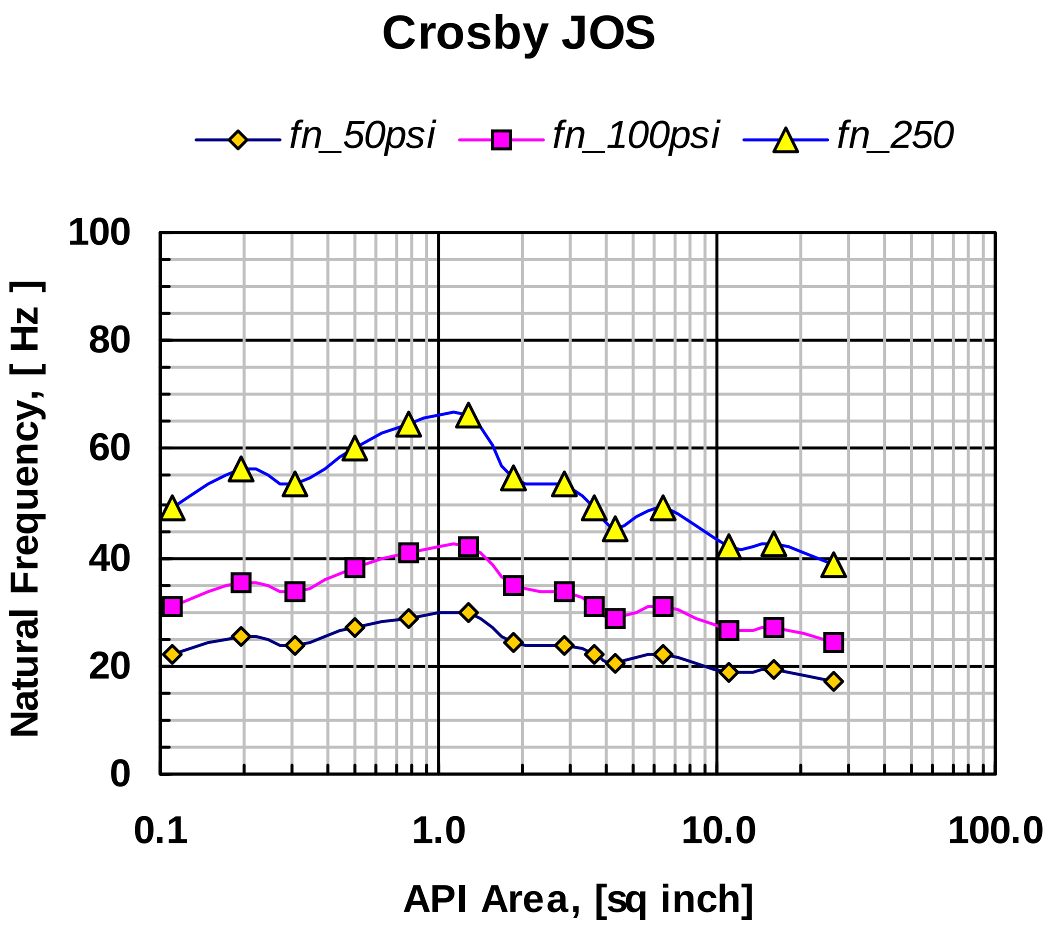
Relief Valve Spring Constant Estimate



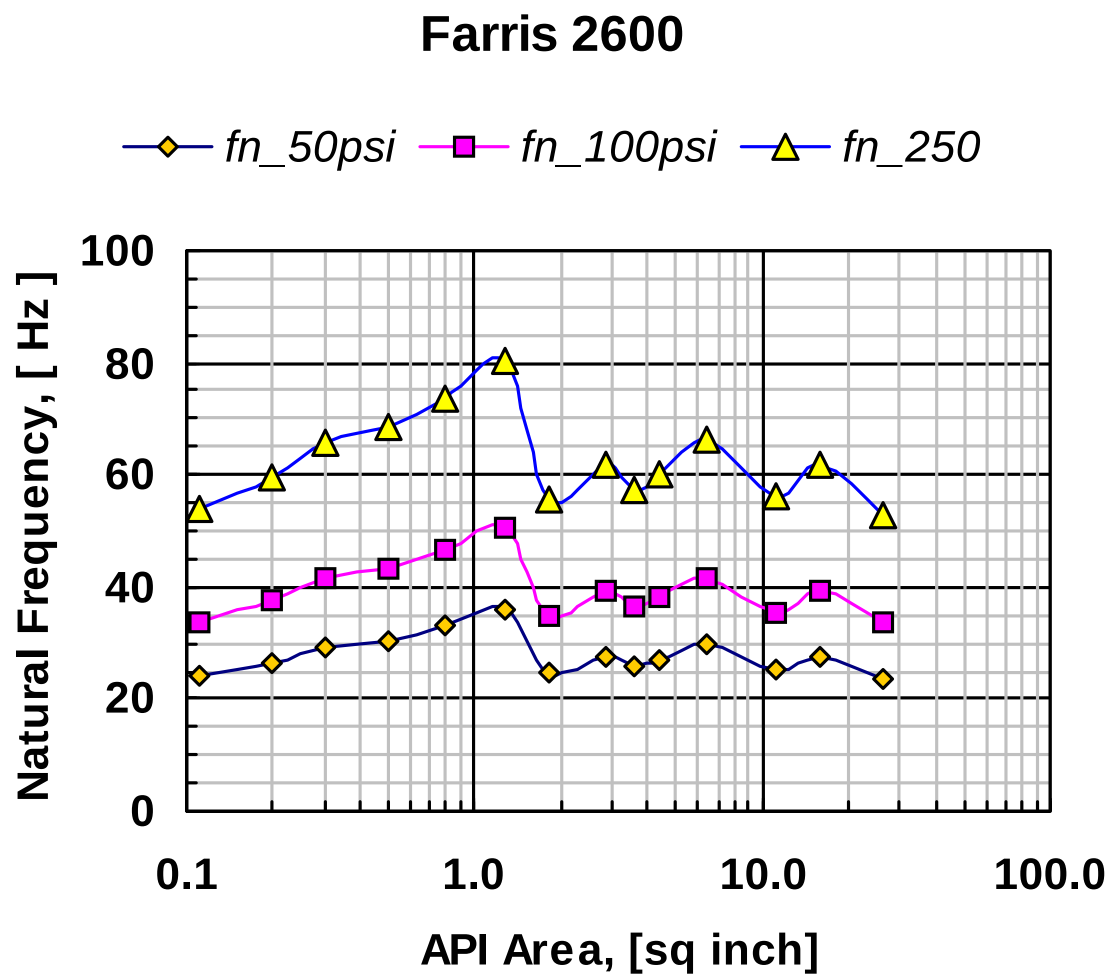
Relief Valve Spring Constant Estimate



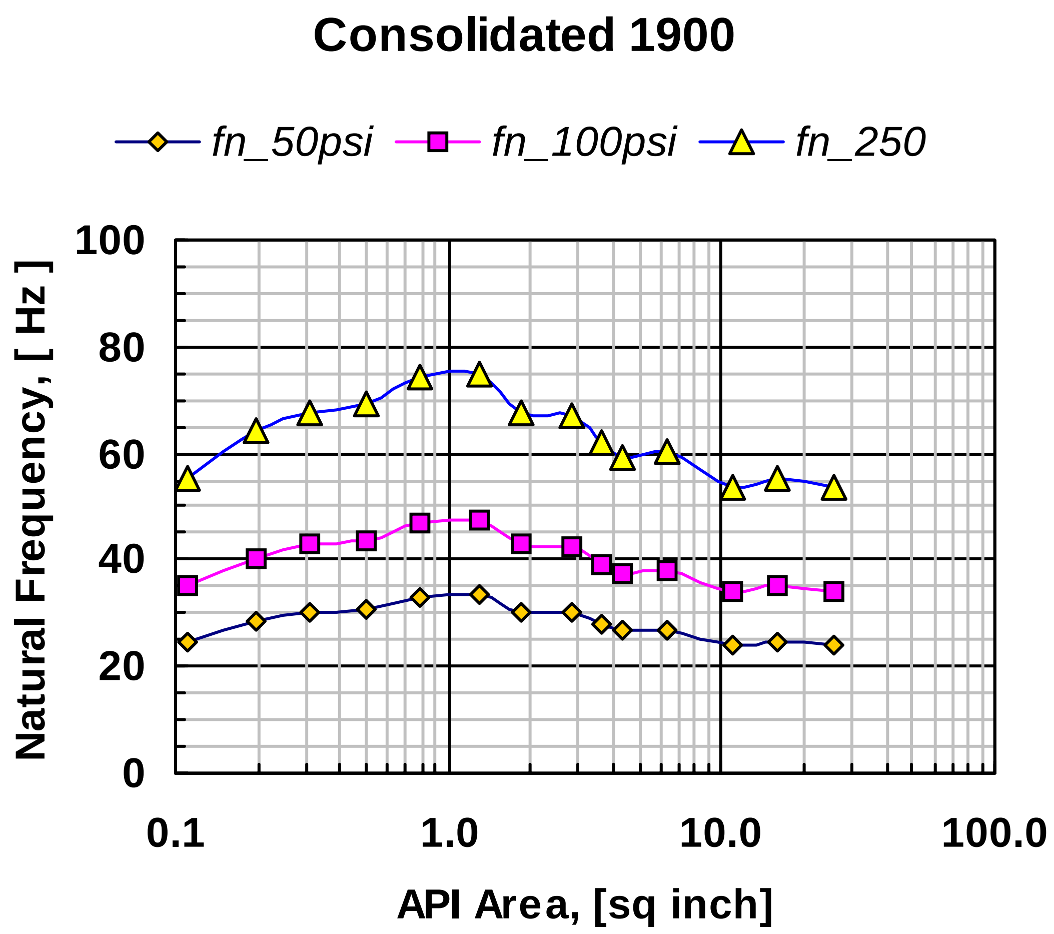
Relief Valve Natural Frequency Estimate



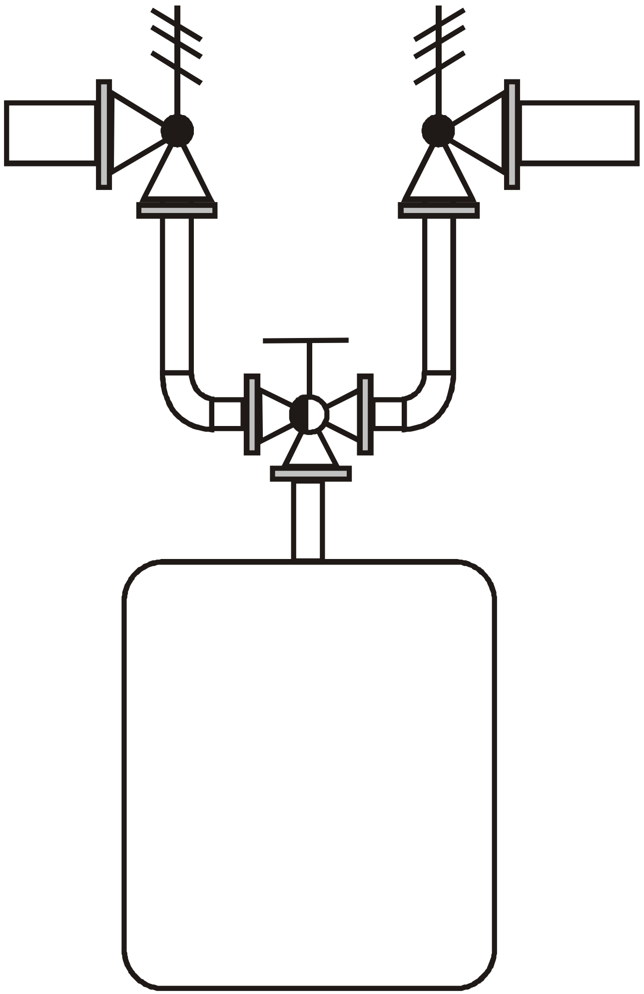
Relief Valve Natural Frequency Estimate



Relief Valve Natural Frequency Estimate



Back to Inlet Pressure Drop and Relief Valve O^3

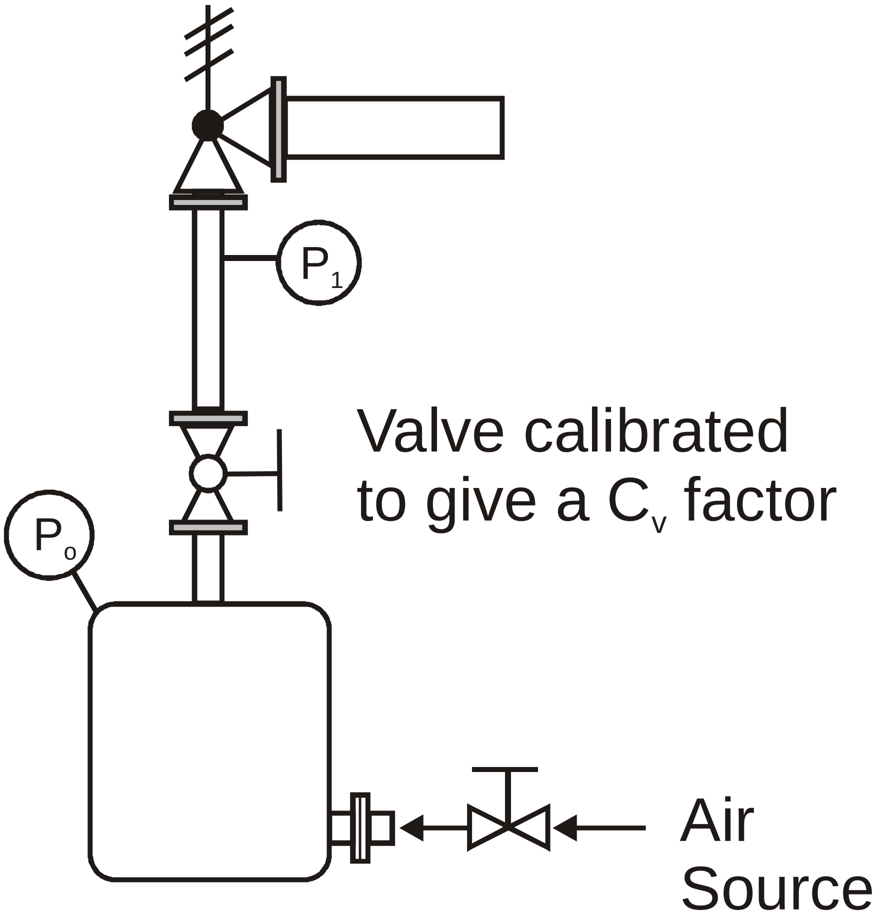


Kieth A. Kastor, "A Dynamic Stability Model for Predicting Chatter in Safety Relief Valve Installations, Part I – Model Development, Verification and Applications" circa 1986

Story Line:

DuPont had a large number of relief valves mounted on 3-way valves that did not satisfy the 3% inlet preeeure loss rule as previously illustrated.

Kastor's Tests



<u>Relief Valve:</u>

Consolidated 1905 E (1 E 2)

P_{set} = 50 psig

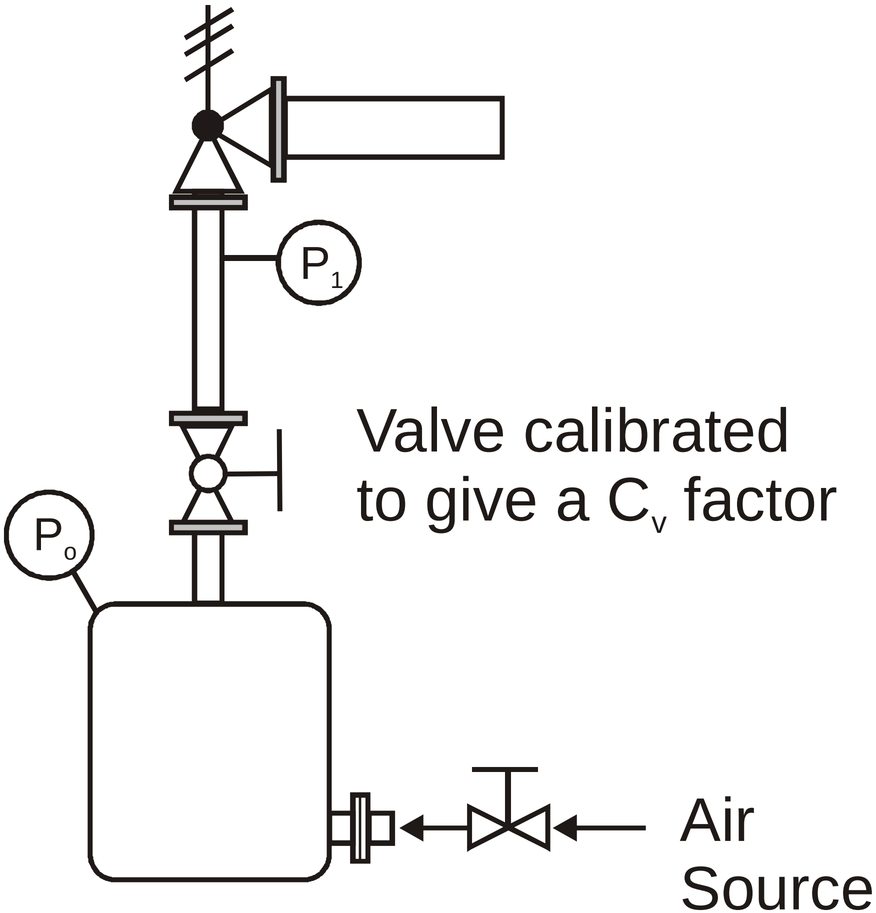
E nozzle: 0.196 sq in API; 0.2279 sq in ASME

Rated Flow at 50 psig (1.1op); 1115 lb/hr air

Reference Test Air Flow:

1200 lb/hr air

Initial Observations :



<u>Relief Valve:</u>

Redbook Valve Lift: 0.147 inch

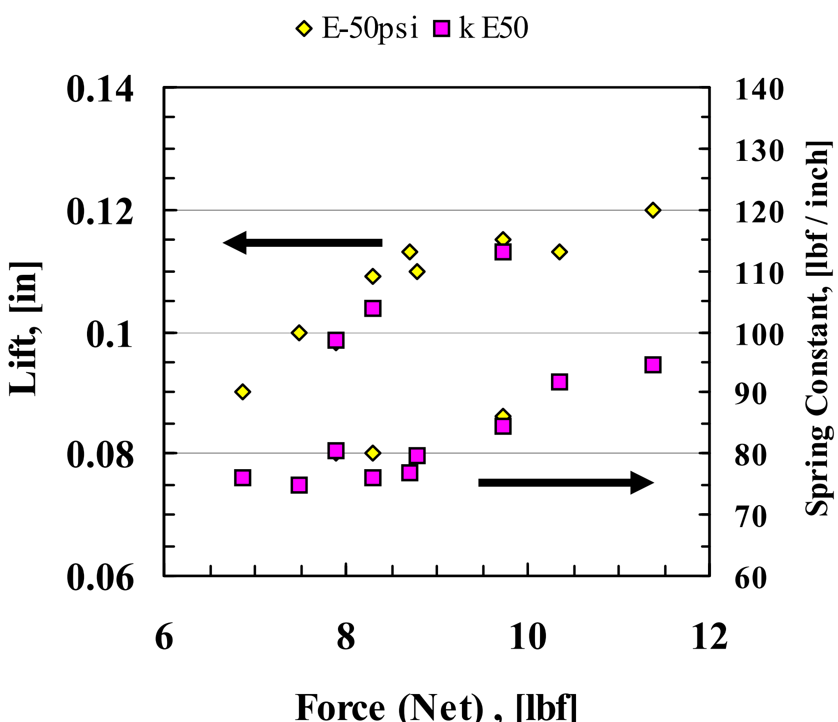
From previous material, we estimate the spring constant and natural frequency as

$k = 84$ lbf / in

$f_n = 29$ Hz

Kastor's data show chattering at 30 Hz and his lift vs pressure data show $k_{avg} = 87$ lbf / in

Initial Observations :



Relief Valve:

Redbook Valve Lift: 0.147 inch

From previous material, we estimate the spring constant and natural frequency as

k = 84 lbf / in

f_n = 29 Hz

Kastor's data show chattering at 30 Hz and his lift vs pressure data show k_{avg} =87 lbf / in

Also Stated:

Avg Opening Pressure 50.6 psig

Avg Reclosing Pressure 44 psig

Damping coefficient 0.2

Note: Blowdown, P_{bd} is over 10% relative to P_{set}

Short Summary of Test Results :

Test	1	**field**	2	3	4
Air Flow [lb/hr]	1200	**1200**	1200	1200	1200
Cv	25	**18**	10	5	5
Del Pi [psi]	1.88	**3.62**	11.7	47	47
Del Pi [%]	3.8	**7.24**	23.4	choked	choked
Results	Normal	**(1)**	(2)	(3)	(4)

Short Summary of Results : Notes

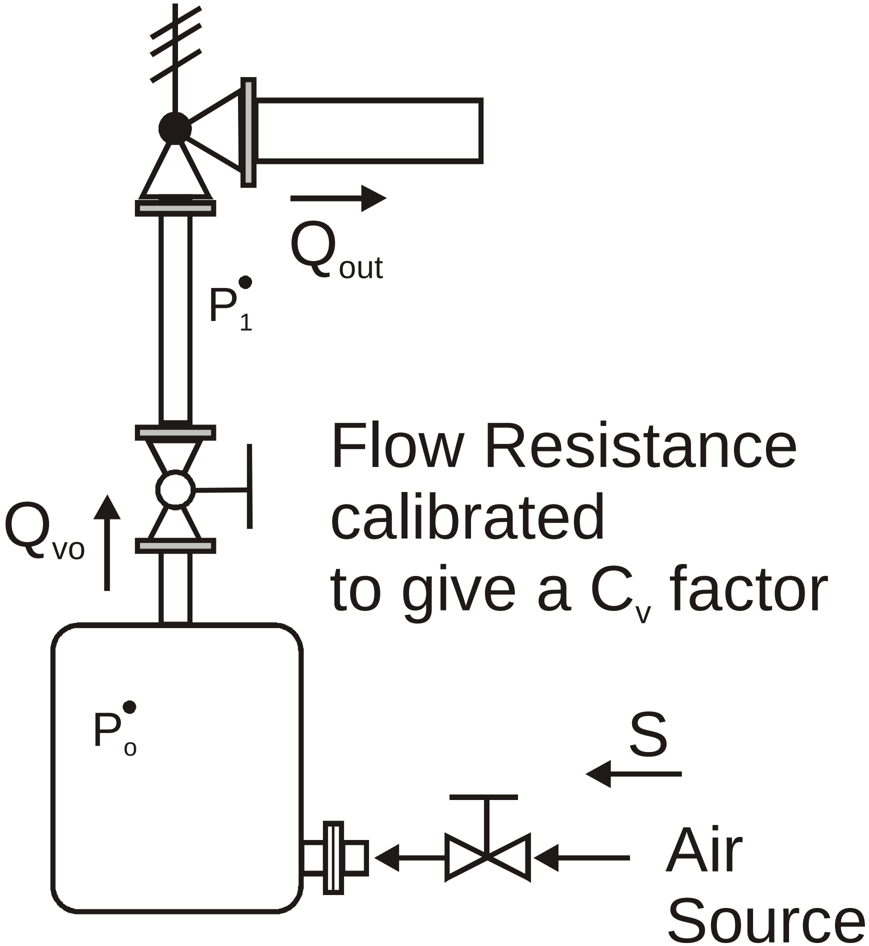
(1) "Field" column represents a typical field condition with greater than 3% inlet pressure loss

(2) $C_v = 10$: Test results show 3 to 4 O^3 cycles then stable operation, but P_o increases just enough to keep $P_1 > P_{bd}$

(3) $C_v = 5$: Test results show sustained O^3 (chatter) at 30 Hz (natural frequency of spring – mass system)

(4) $C_v = 5$: This test had a 10 ft inlet pipe before relief valve. Results showed limited initial chatter, but vessel pressure increased and chatter ceased. Tank pressure reached 80 psig where $C_v = 5$ passed the air source flow

Brief Model Outline



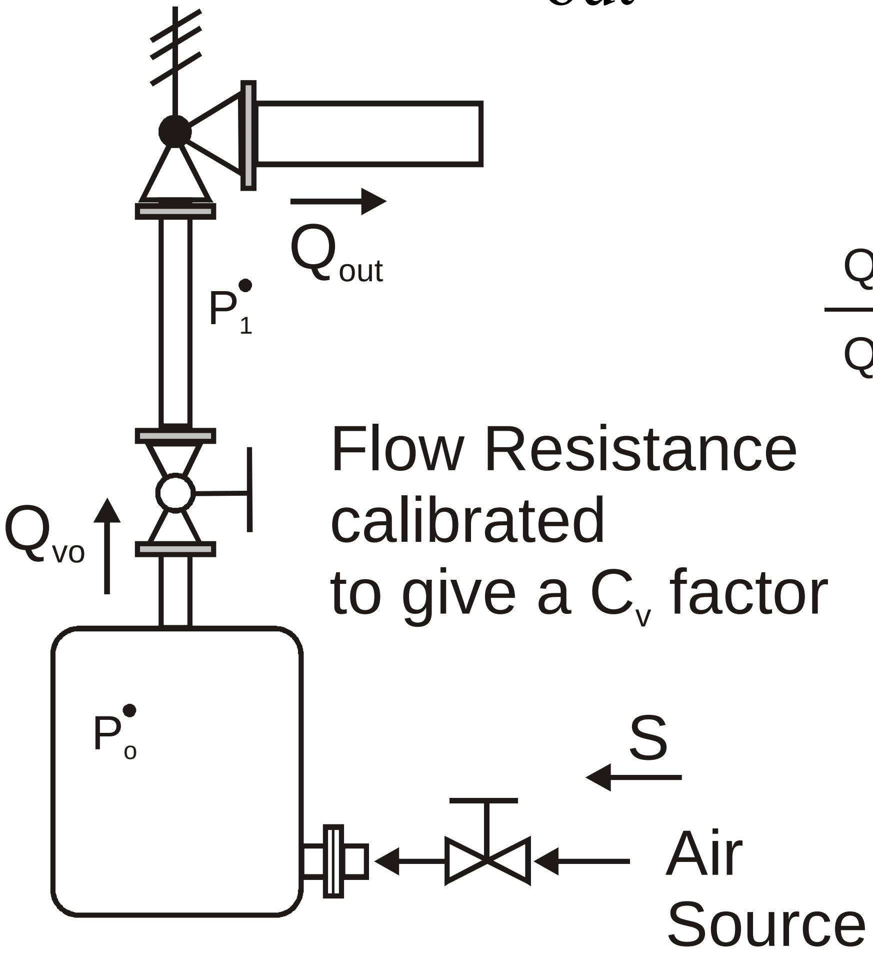
$$\dot{P}_1 = Q_{vo} - Q_{out}$$

and

$$\dot{P}_o = S - Q_{vo}$$

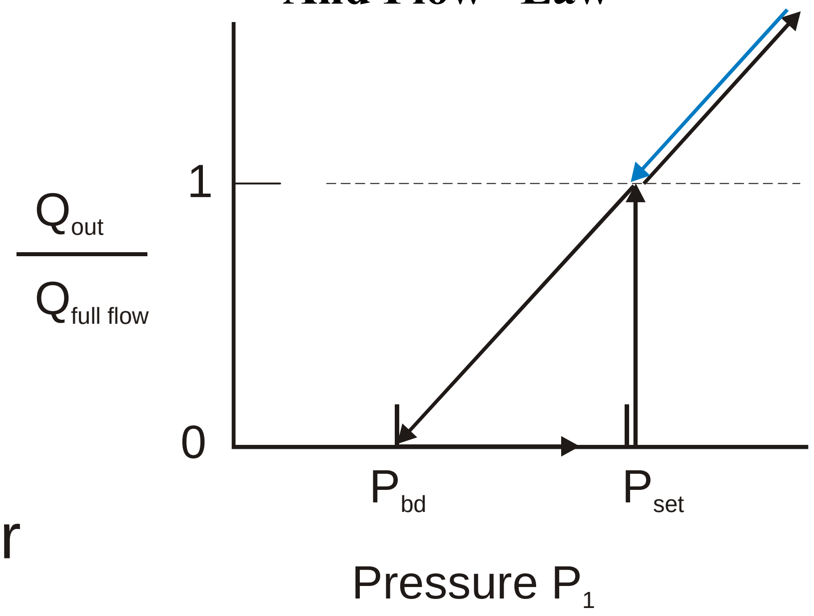
Note: This is Kastor's model without the relief valve spring mass system

A note on Q_{out}



Relief Valve Opening

And Flow "Law"



Short Summary of Model Results

(1) Equation system is very stiff

(2) Model generally reproduces test data results without relief valve spring-mass equation added. O^3 is judged simply by Q_{out}

(3) Results are not overly sensitive to precise shape of relief valve "Law", but the "pop" action is all-important

(4) It becomes immediately apparent that flow does not go to zero unless P_1 falls below P_{bd} – this is consistent with Kastor data

Short Summary of Model Results (continued)

(5) In many cases moderate excess inlet pressure loss will be overcome by increase in P_o so as to establish required relief flow without O3.

Implications for 3% rule

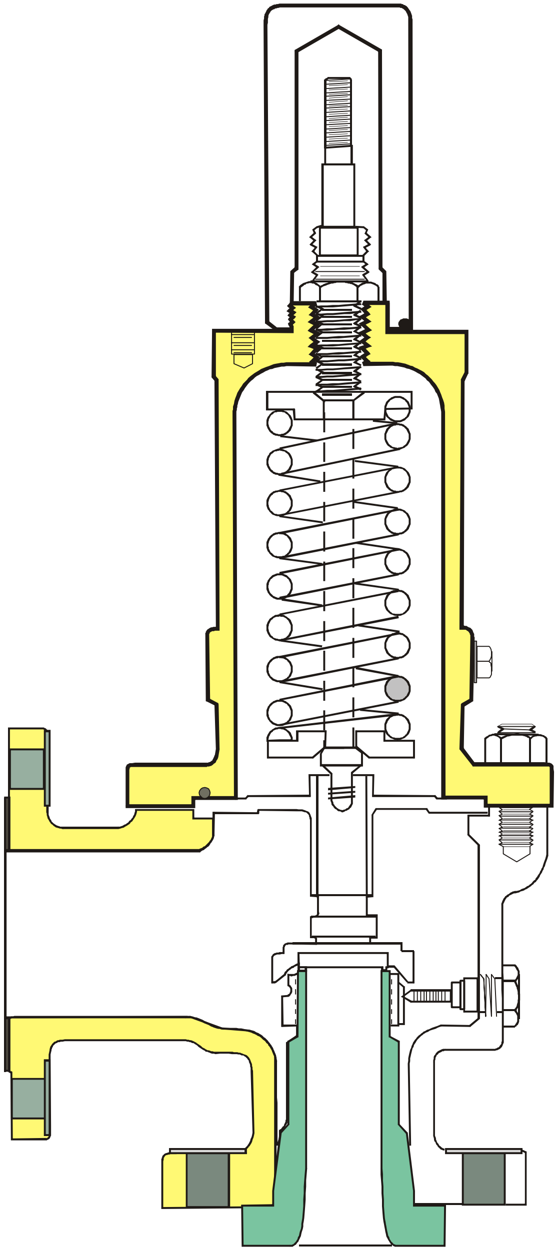
(1) An alternative and technically superior way of avoiding O^3 and chatter is to require inlet pressure loss to be related to P_{bd}. That is, if P_{bd} is 5% of P_{set}, then inlet pressure loss should not get one any closer than 2 percent points to shutoff. ***We need to have better and reliable information for the blowdown or reclosing pressure.***

(2) The present 3% rule does not consider the positive effects of allowance for overpressure. For example the fire case allowance for 21% overpressure should allow for at least 10% to 15% inlet pressure loss as long as this effect is properly included in the sizing calculations

Implications for 3% rule

(3) Reducing the relief valve set pressure below the vessel design pressure can also have the same positive effect on allowance for larger inlet pressure loss

(4) A very good situation is to have the relief valve "undersized" at the relief set pressure and properly sized for some defined overpressure. In this case 3% inlet pressure loss is irrelevant if the relief valve design properly considers the actual inlet pressure loss in the sizing calculation. Undersized at the relief set pressure is common practice for runaway tempered reactions as a class.



There is much more to this story, but for now

That's All Folks!!

Centaurus Technology, Inc.

Comments from ISO 4126-10 Committee Meeting
New Orleans, April 16, 2008

Chair: Alan West

Secretary: Joe Ball

ISO web page has all relevant info.

WG-1 Two Phase Flow – Friedel not present (has retired). Schmidt not present (German group led by Hans-Dieter Perko of Sempell). Alan West summarized results.

Previous vote was 11 – 2 in favor (US and Italy opposed), with serious reservations by some others.

John Harrer (UK), Tom Bevilaqua and Ron Darby spoke to the applicability of the Omega method. The appropriate method to use for 2-phase flow depends entirely upon the quality and amount of data available for determining the properties of the fluids Hans-Dieter:

- If complete thermo properties are available, a rigorous method (such as HDI) is the most general, precise and simple method with the least restrictions.

- If limited data are available, other more approximate methods are available (e.g. API, TPHEM, etc.) which can be used. These methods are less rigorous and less general, with more restrictions and limitations.

- If very little data are available, the Omega (single point) method can be used. This requires the least amount of data but the greatest uncertainty with regard to linear extrapolation of properties from one point, and hence offers the least accuracy, least generality and most restrictions of the available methods.

It was pointed out that the document doesn't really address piping analysis for two-phase flow.
ISO 23251 (old API 521) – equation for 2-phase flow in piping (Zamjec).
Use "Informative Annex" to describe two-phase flow in piping?

"Standards" vs "Guidelines" or "Recommended Practice"???

Options:
- Issue as a Technical Specification instead of a "Standard"?
- Joint standard or joint WG with CEN ISO TC/67/SC6 WG-12 (petroleum, petrochemical and natural gas) or API 520?
- Continue the work of WG 1 on the DIS to work out the differences (??)
- Revise the scope of the DIS, and continue to work out differences (?)

April 17

Japan recommended continuing the Working Group to revise the standard.

US recommended the first option (Technical Specification), possibly with joint input from other TC or WG committees. This was not agreed to by the committee, which voted to continue the work toward an ISO standard.

US and France will draft a resolution to continue the efforts of WG-1 on the standard.

Prof Friedel has retired, and leadership of WG-1 will be turned over to Juergen Schmidt (BASF).

Resolutions were submitted to continue the WG-1, to delete any reference having negative implications or inferences on using alternative sizing methods and to recognize that such alternate methods may be used, and to establish liaisons with several other working groups concerned with relief systems design in various systems to coordinate any common material and make the documents compatible.

New experts added to the WG-1 working group from each country. (Marc Levin and Georges Melhem from US).

Development of Spring Loaded Safety Valves

Fall 2007 DIERS

Roger D. Danzy

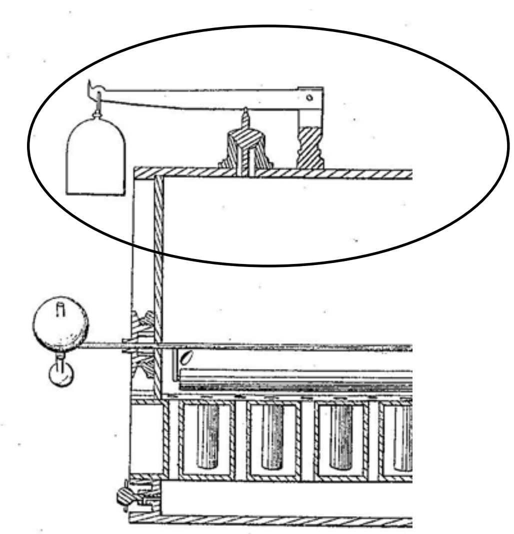
Dresser Inc.

Development

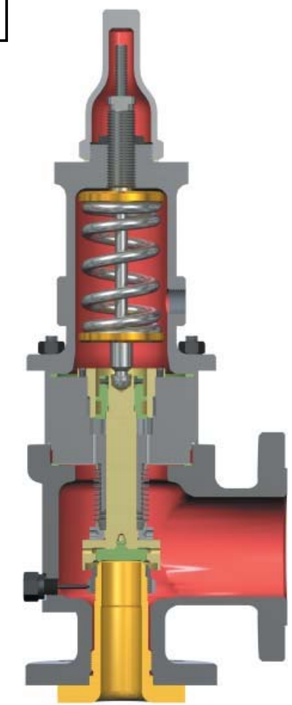
- US Patent and Trademark Office
- On line Searching
- Patents 1790 – 1975
  - Issue date and classification
  - Must view for content
- Patents 1976 forward – text search

From There to Here - 170 Years in 40 Minutes

1836



2006

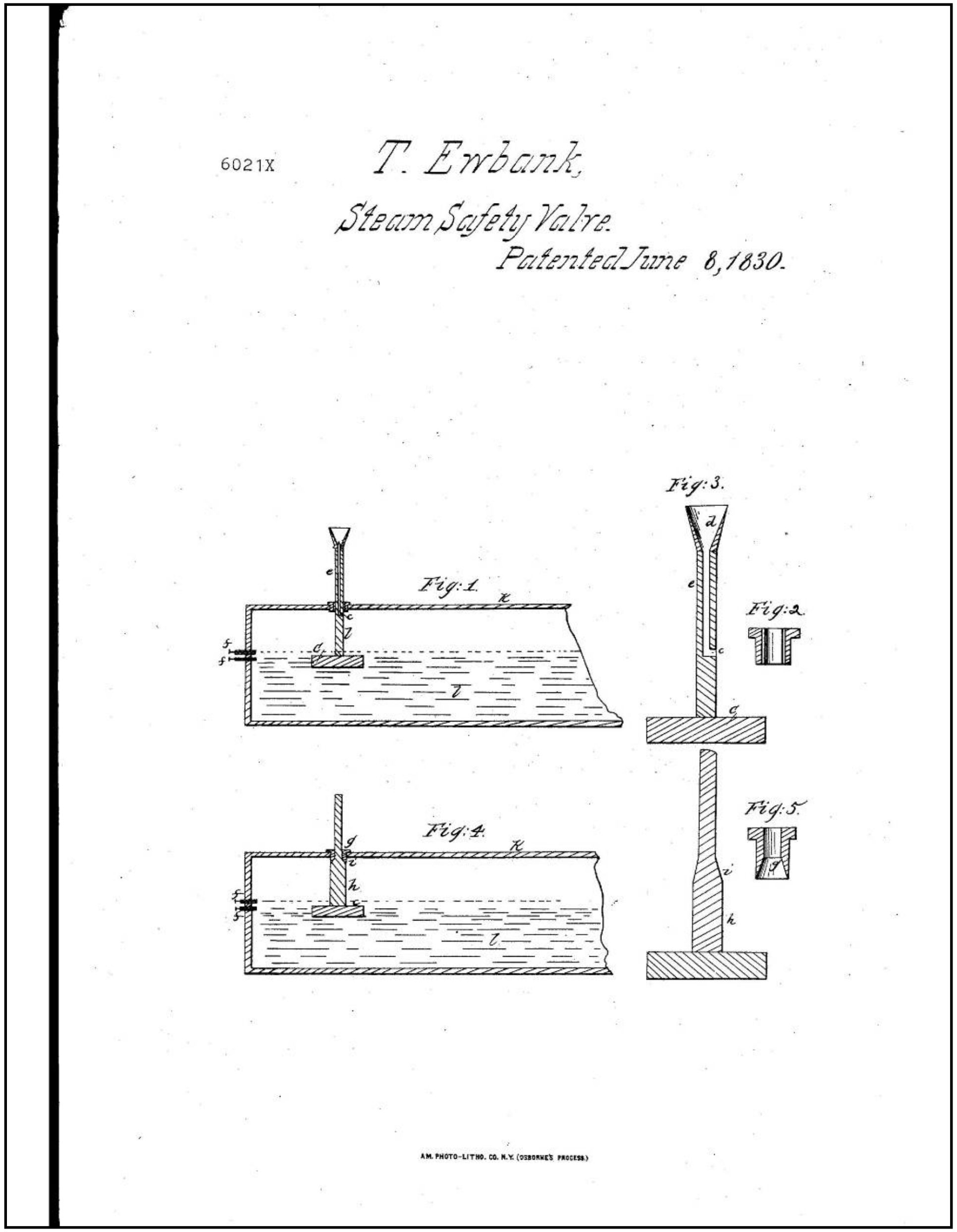


The Non-Spring Years

- First Safety Valve Patent – 1830
- First Safety Valve Patent with Spring - 1860

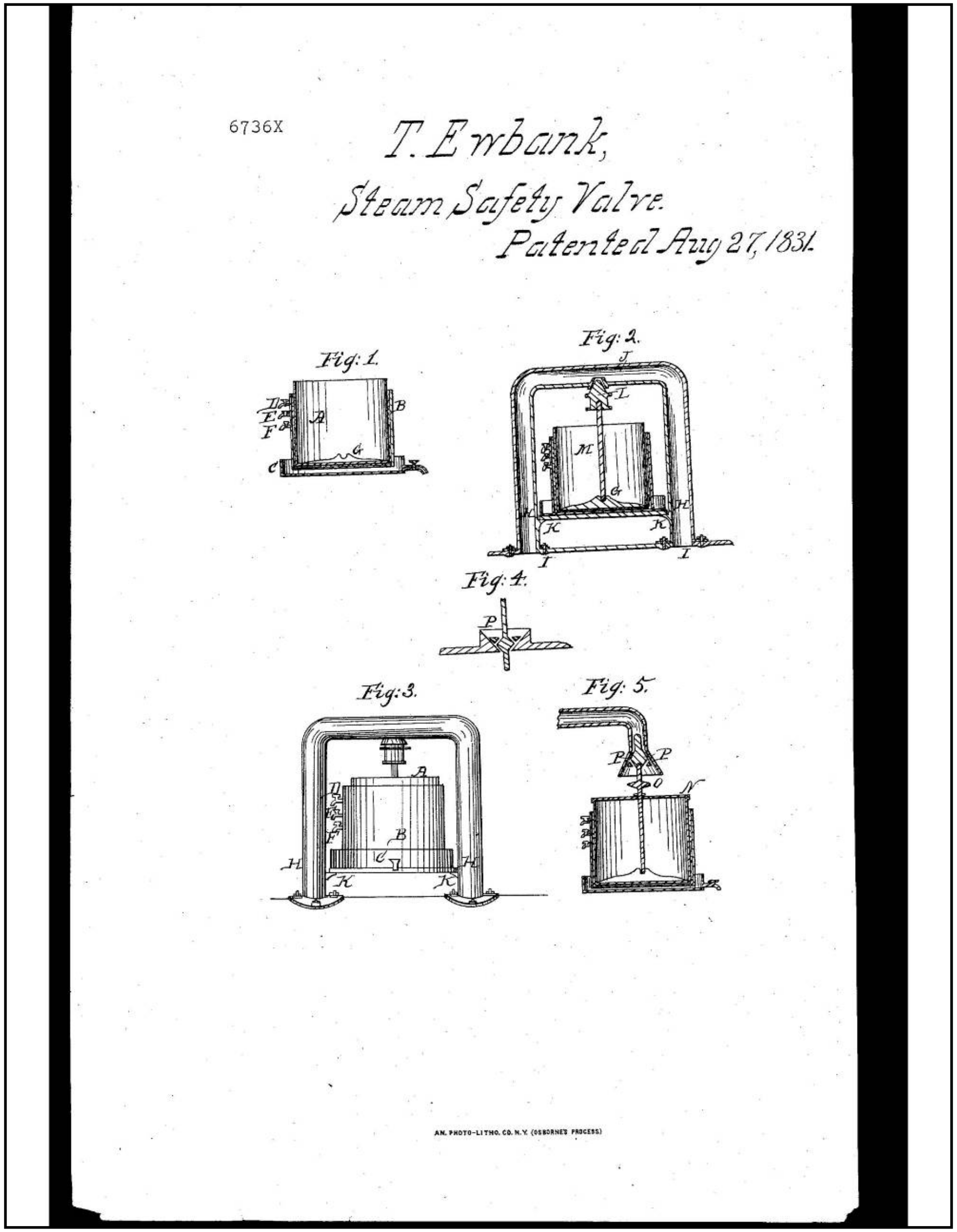
1830

- First Patent for Safety Valve
- Low Water Level



1831

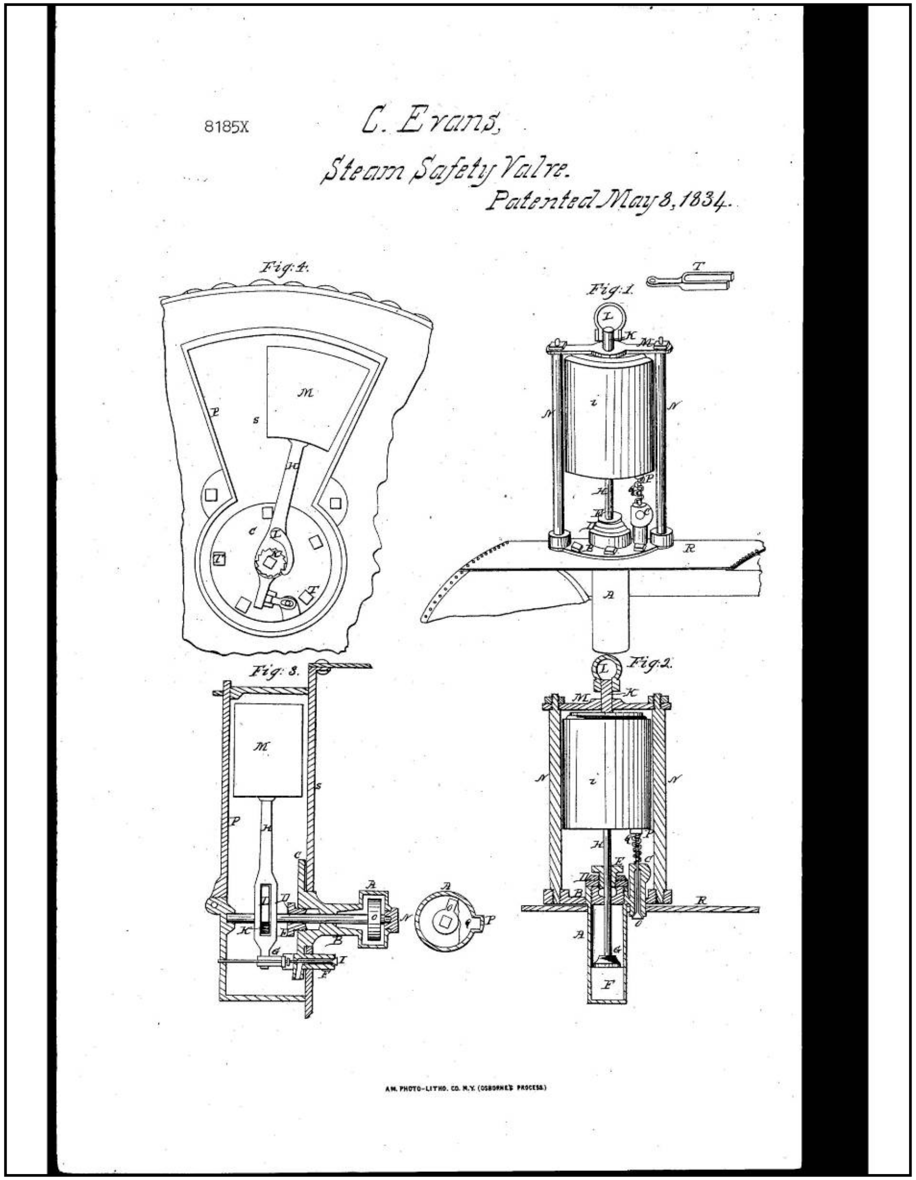
- Water Column
- Differential Area
- Biasing Force



1834

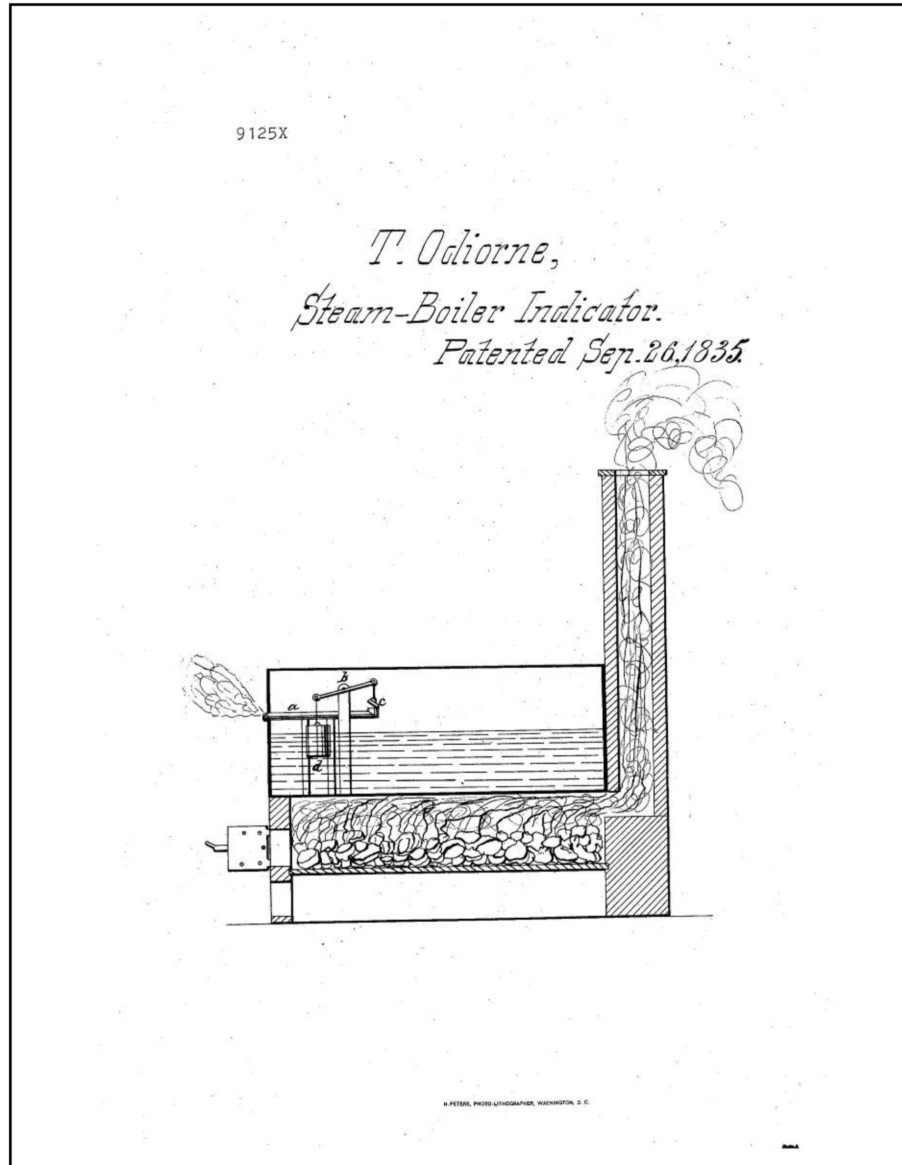
- •Fusible Plug
- •Pressure to Internal Piston

- •Pendulum - Marine Applications
- •Opened Valve – Boat Rocking



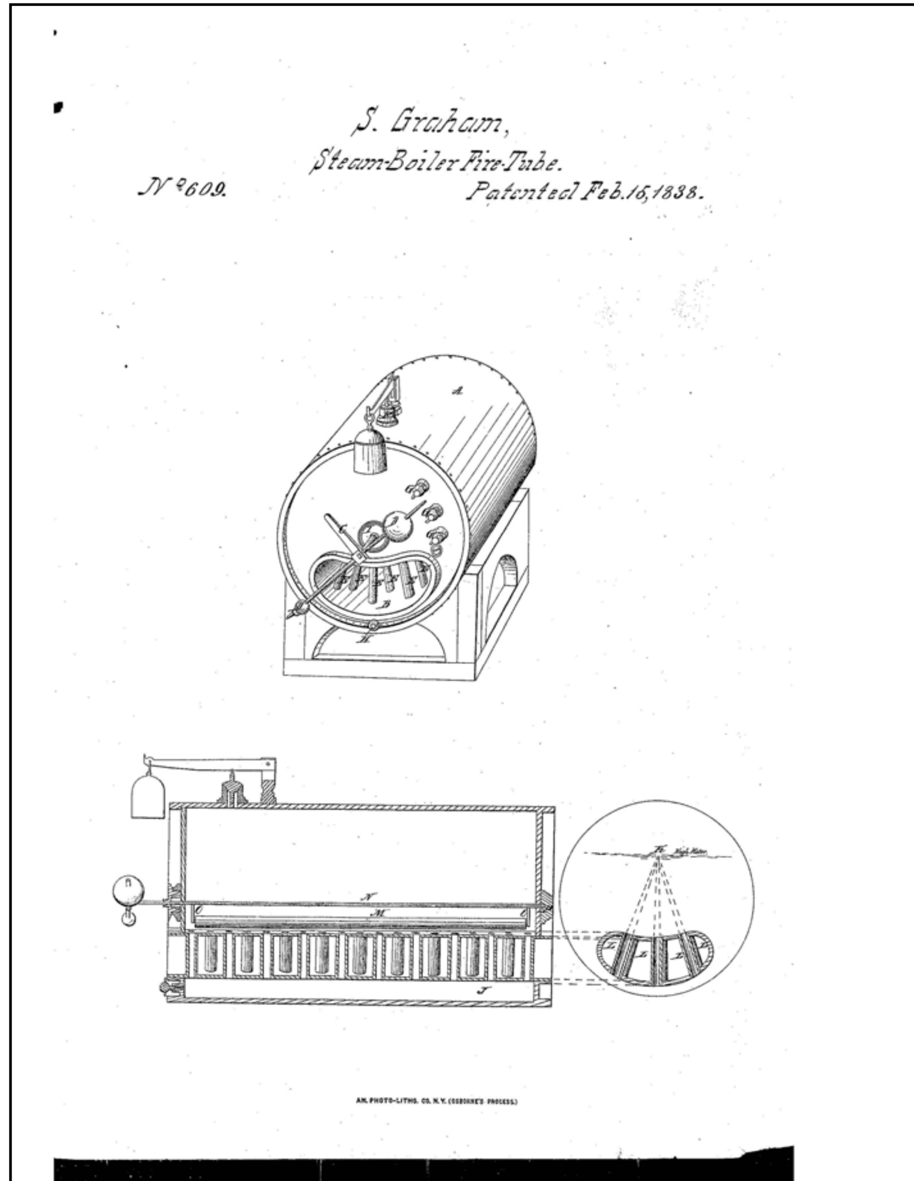
1835

•Water Level Safety



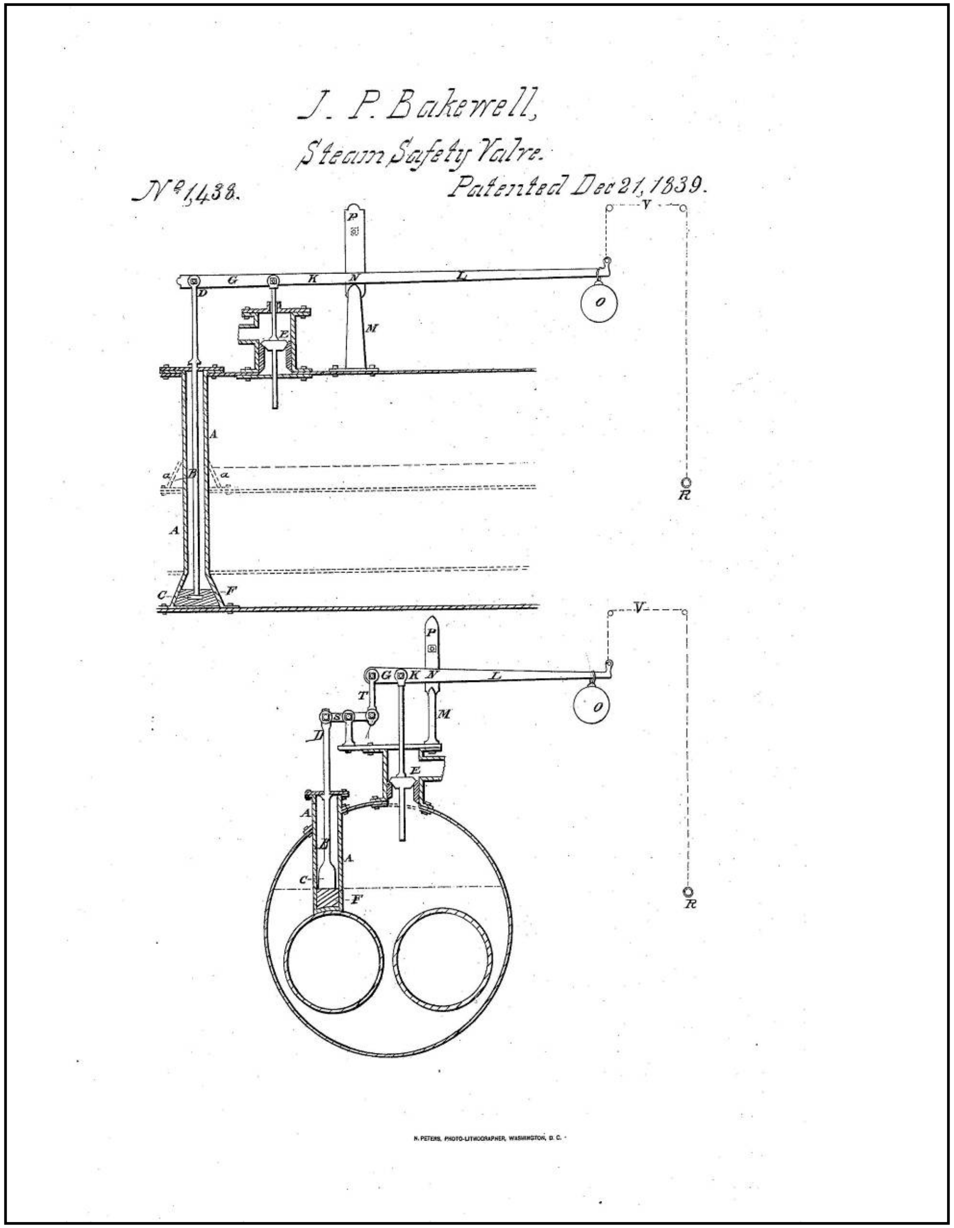
1836

- Weighted Lever Safety Valve
- No Patents – This type device



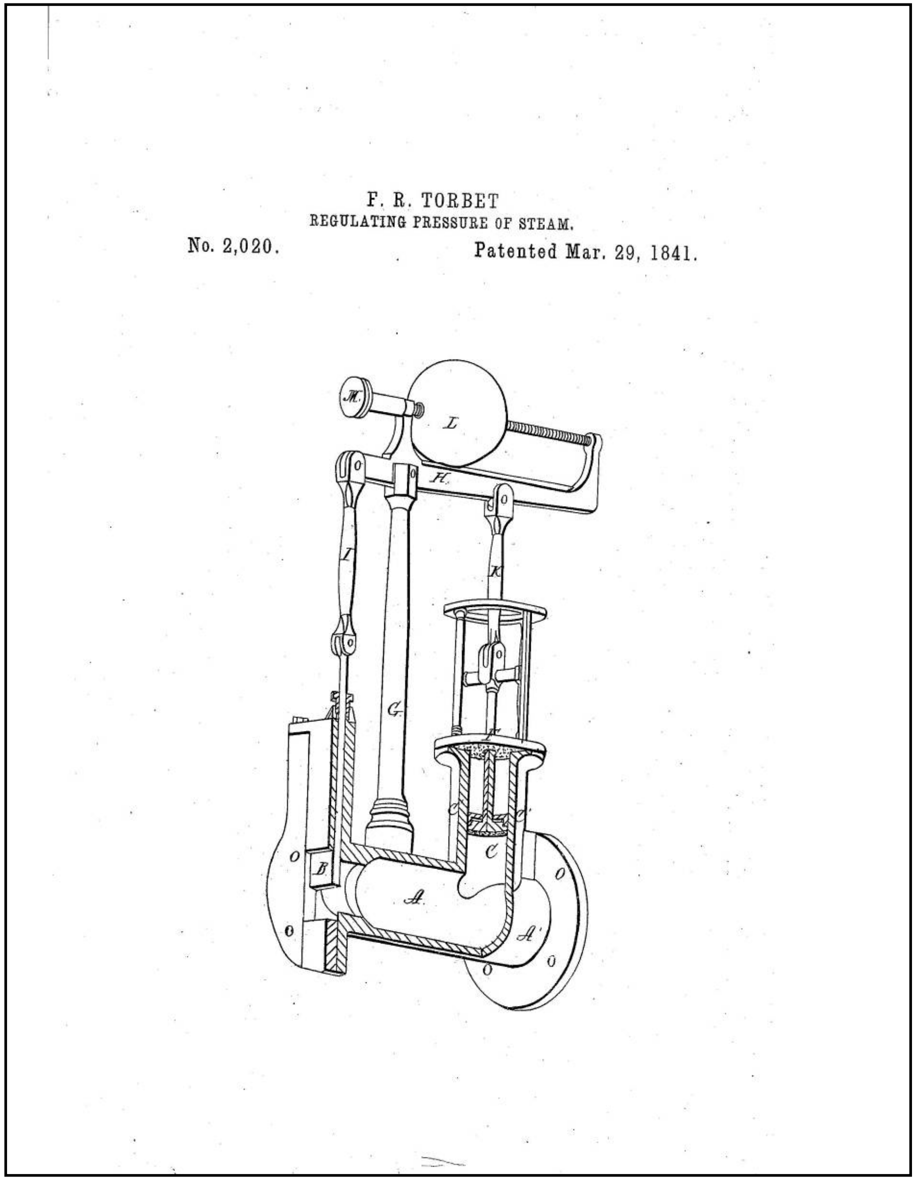
1839

- Temperature
- Pressure
- Safety Valve



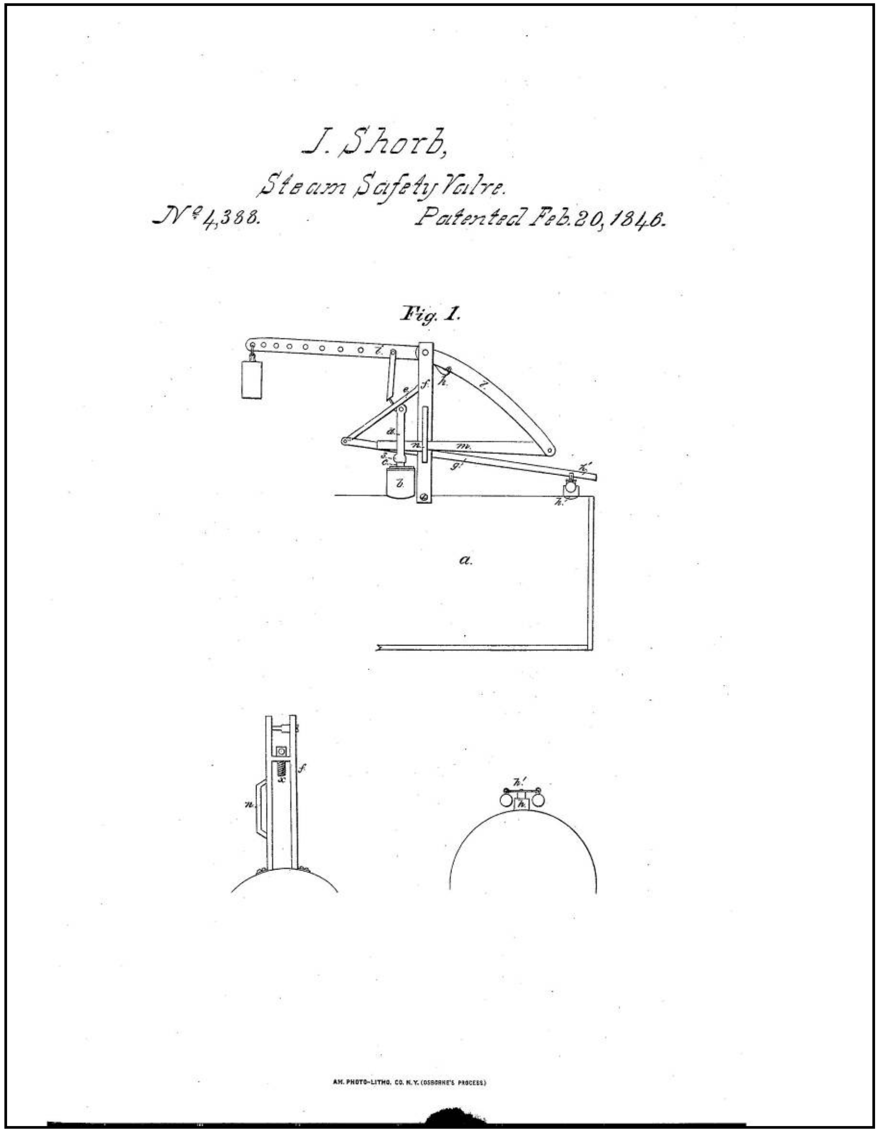
1841

First Control Valve Patent



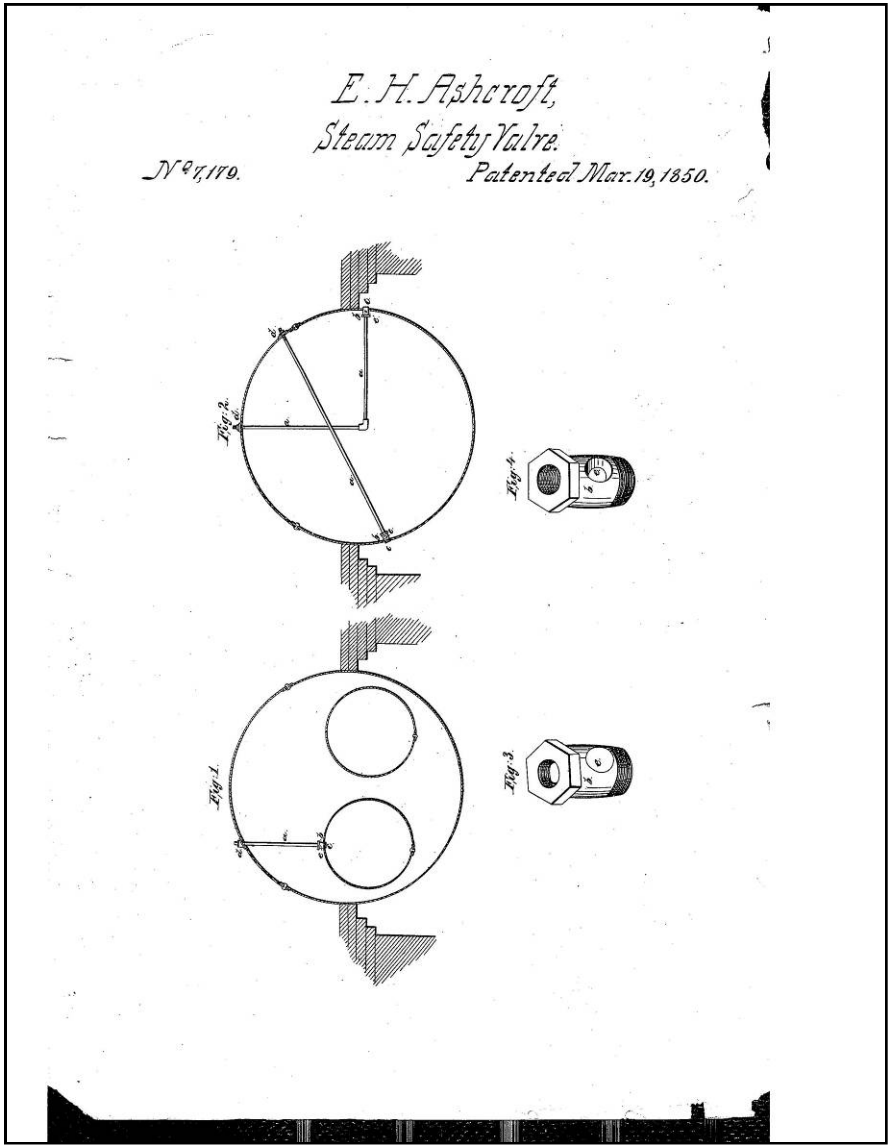
1846

- Compound Lever
- Weighted Safety Valve
- Improve Opening Action



1850

•Fusible Plug
•Ashcroft – Consolidated® Co-Founder

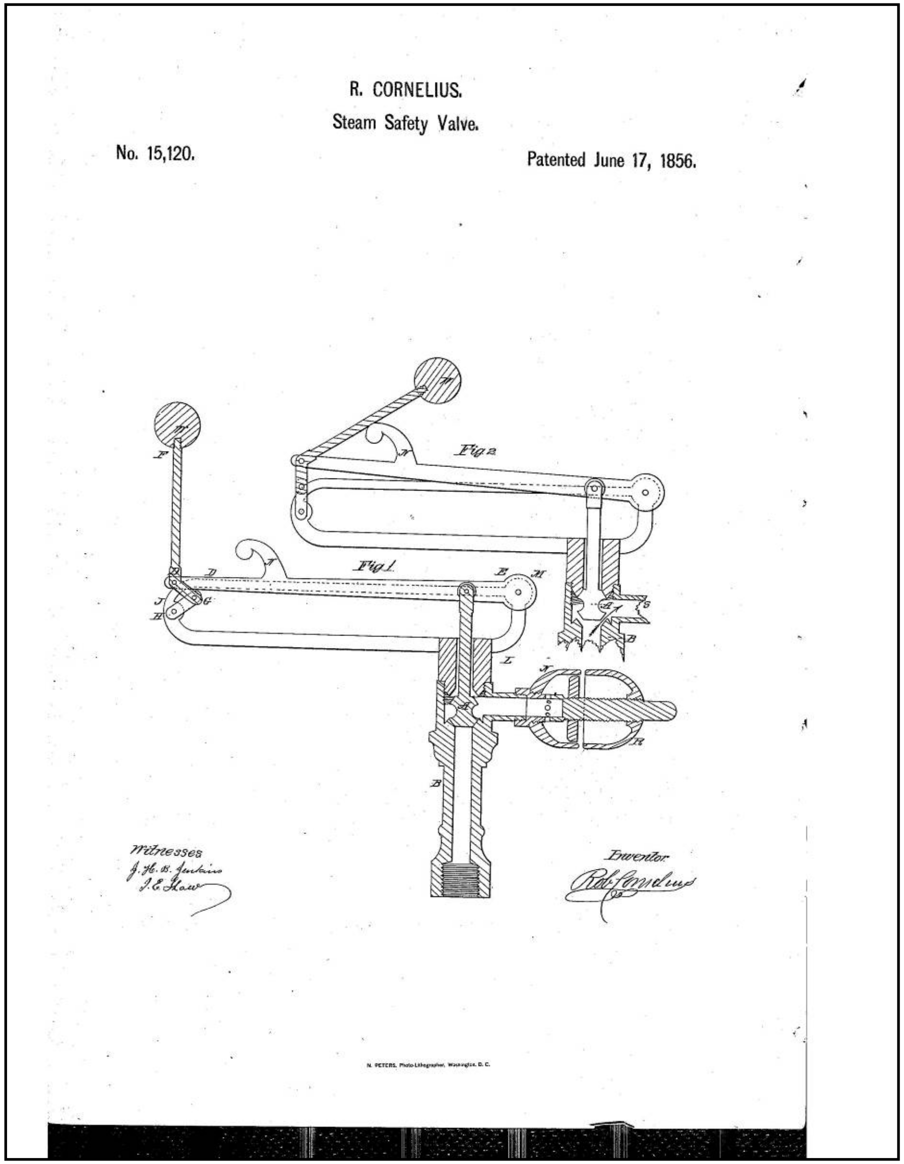


1856

•Compound Lever

•Weighted Safety Valve

•Improve Opening Action

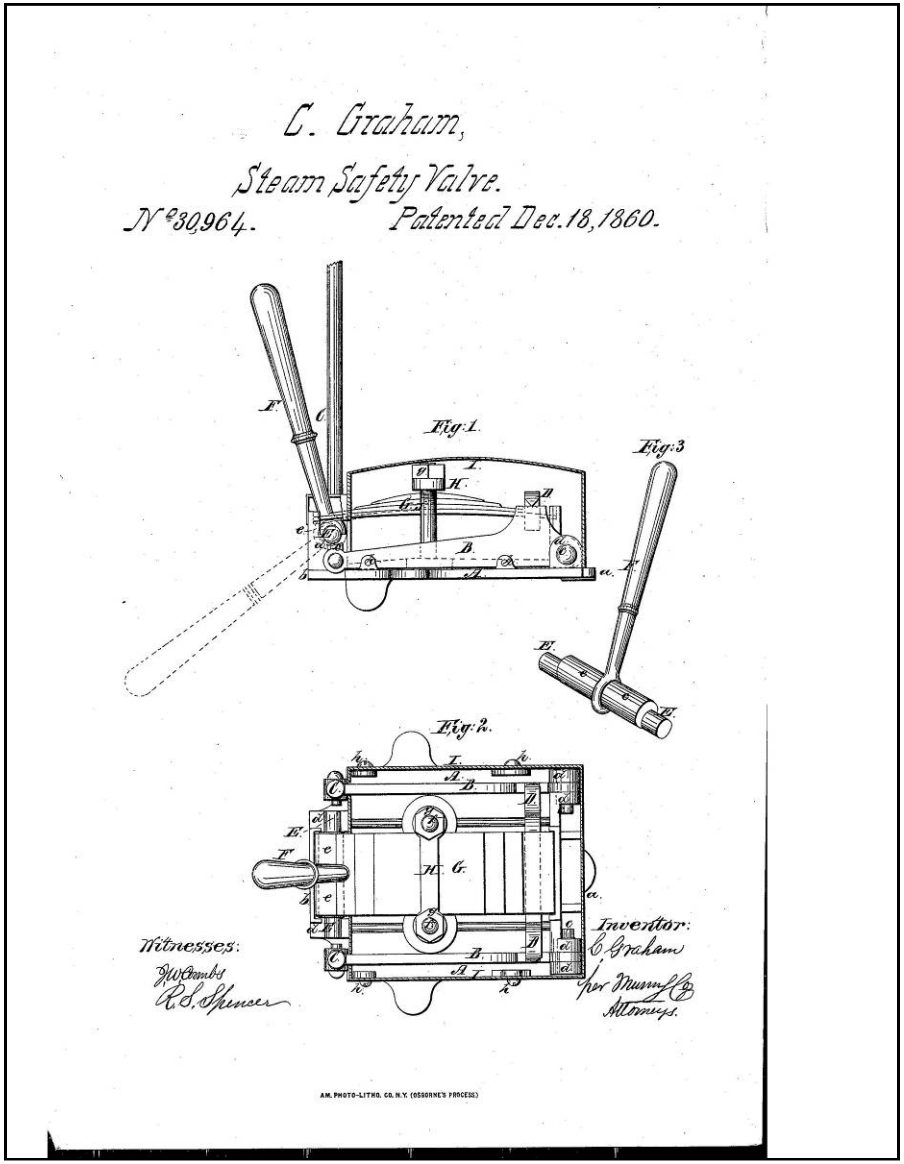


Spring Loaded

- Spring Loaded Valves
- More Compact than Dead Weight
- More Reliable
  - Locomotives
  - Marine Applications

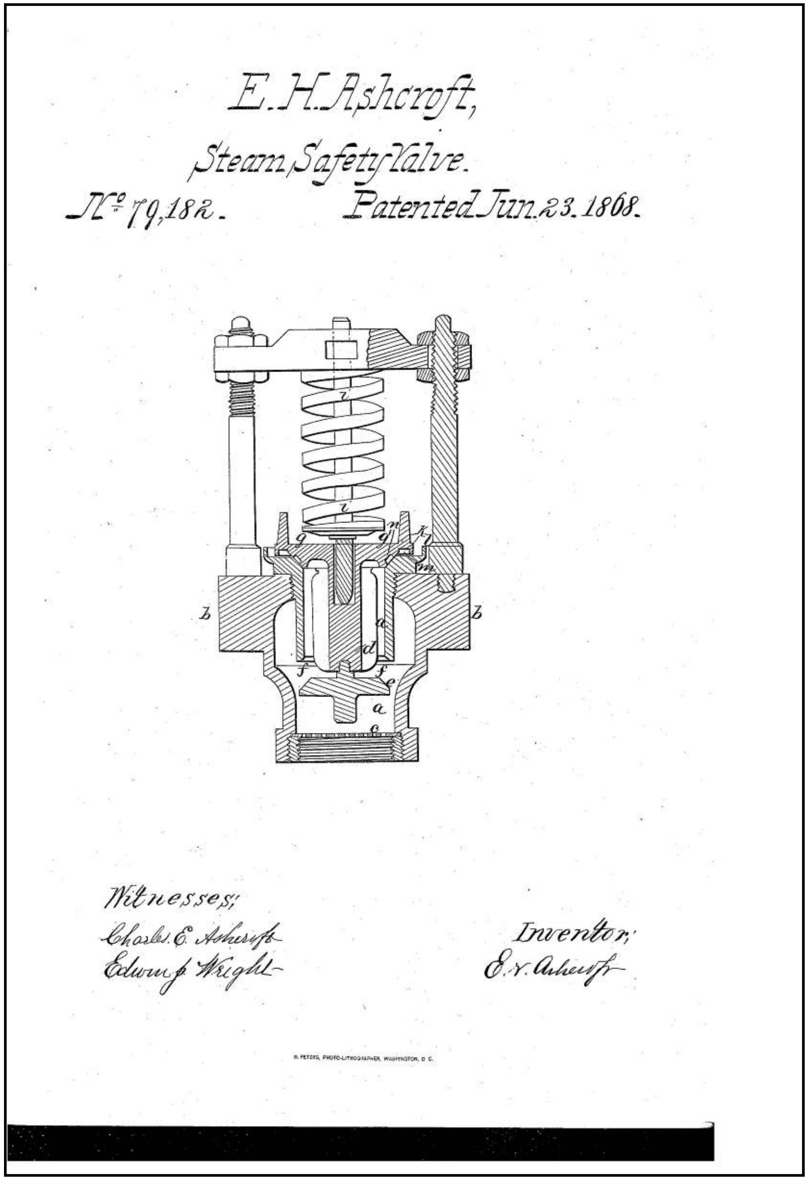
1860

- •First Spring Loaded Noted
- •Spring Loaded Never Patented
- •Elliptical Spring
- •Patent – Adjusting Device



1868

- First Helical Spring Noted
- No Helical Spring Patents
- Reduced Water Draw
- Fail Safe Close



High Capacity Pop Action

- Expansion of Steam Utilized for Pop Action
- High Capacity
- Quick Action

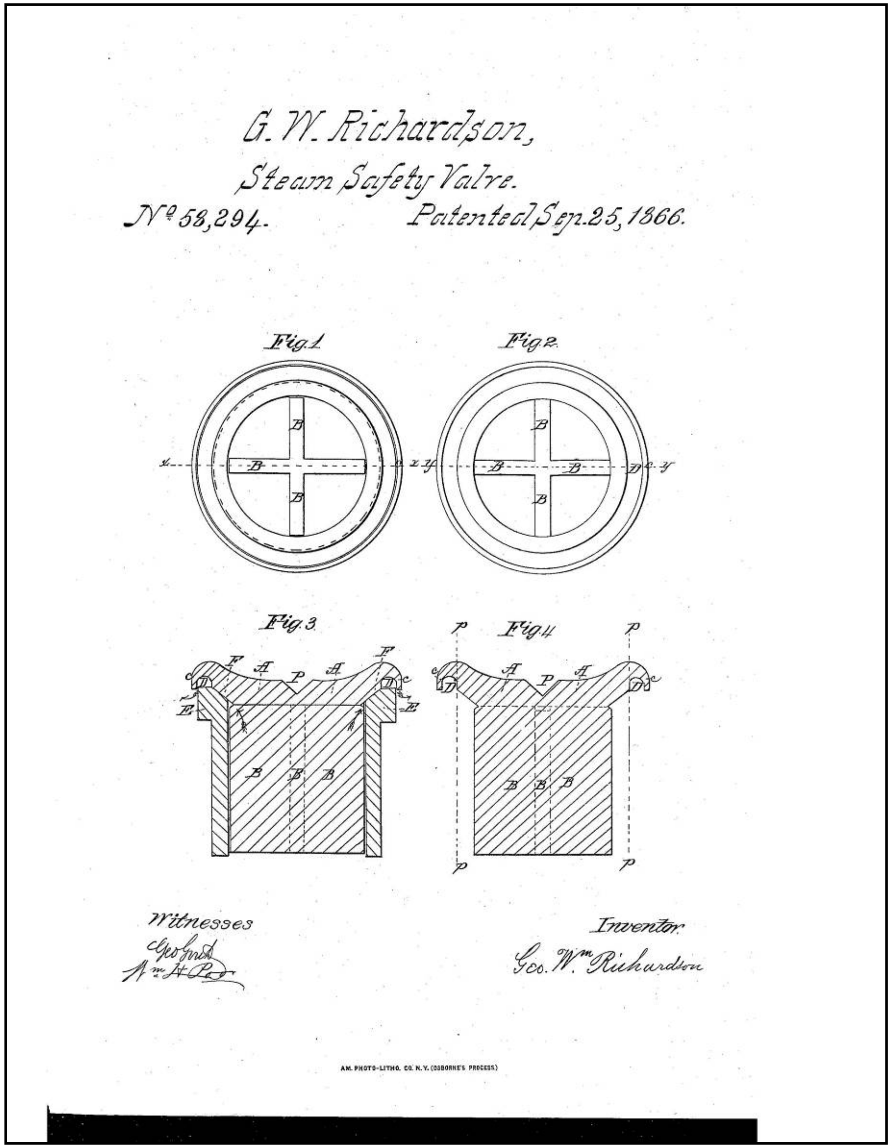
1866

•First Design – Huddling Chamber

•Additional Reaction Force

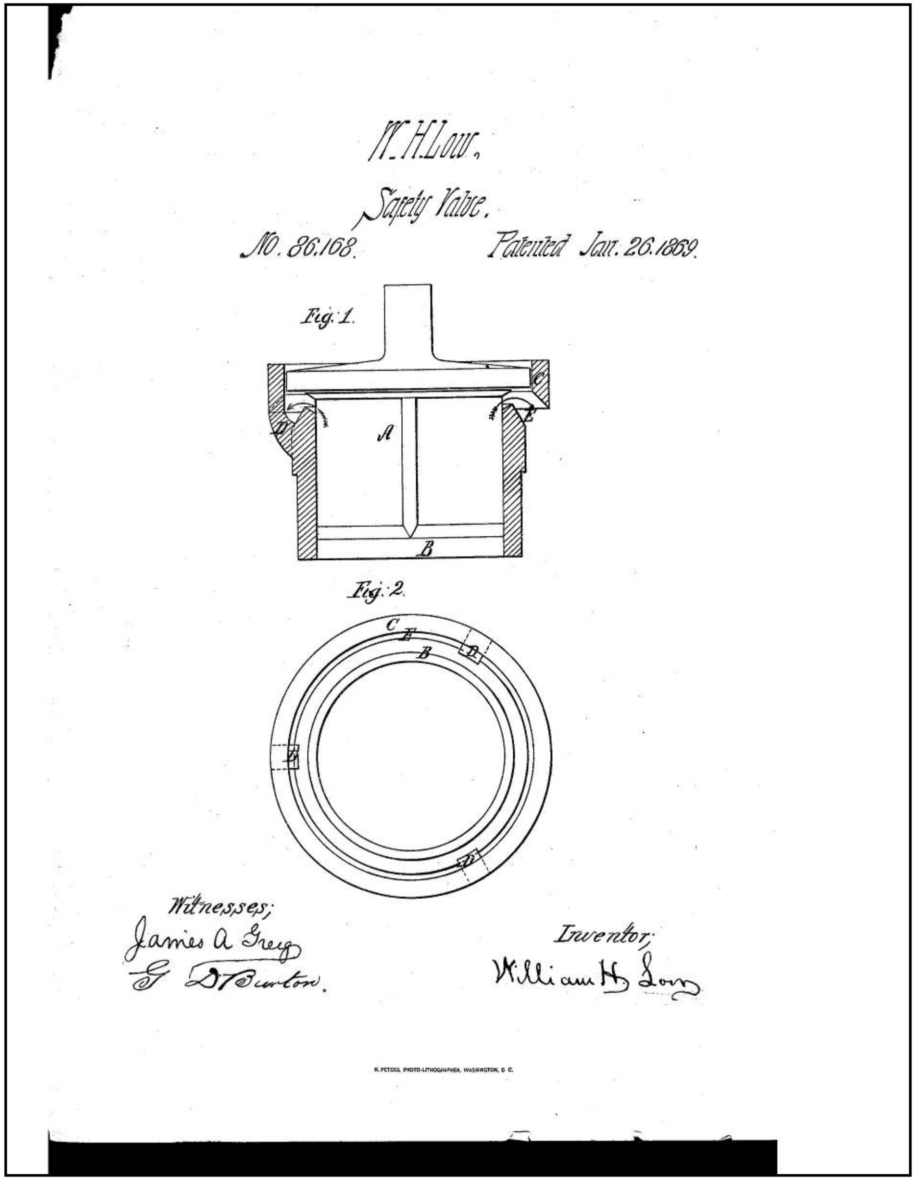
•Improved Opening Action

•Richardson – Consolidated® Co-Founder



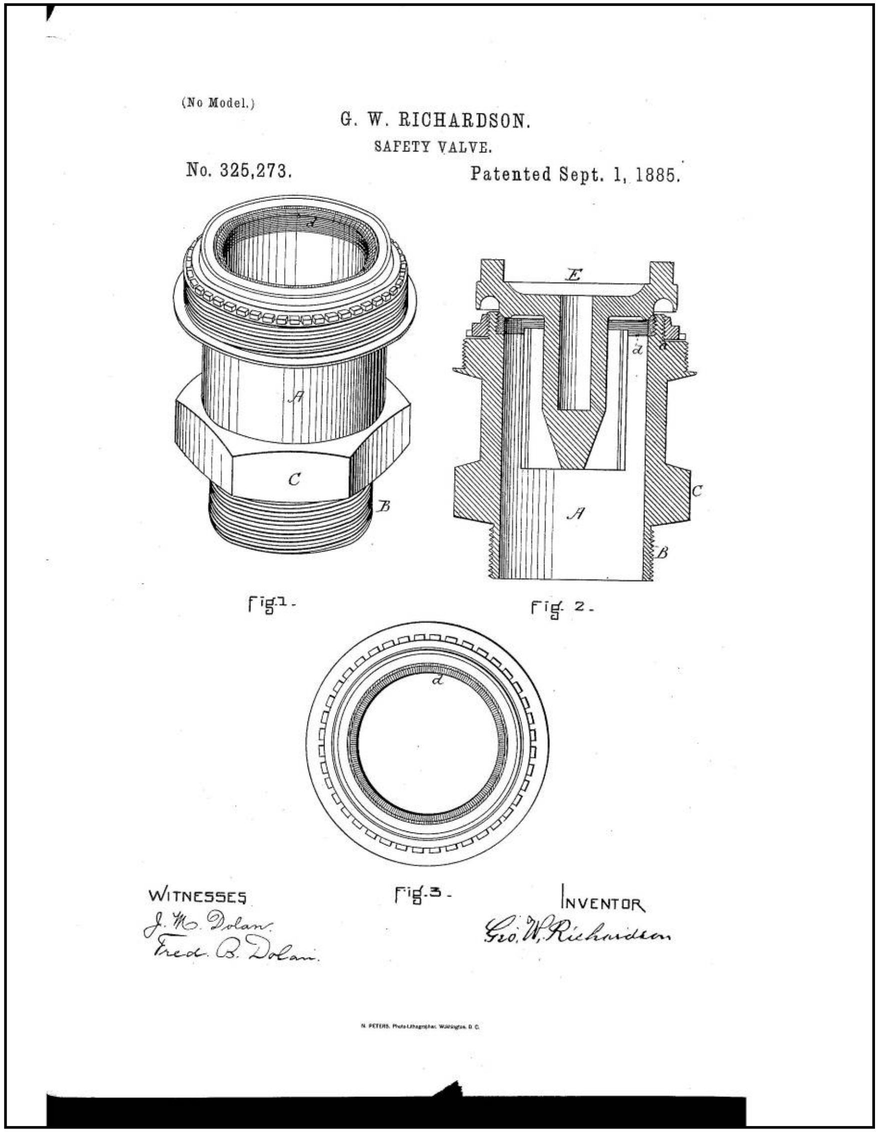
1869

- Stationary Ring
- Improved Opening Action



1886

- First Adjustable Lower Ring
- Opening Action Adjustment
- Blowdown Adjustment

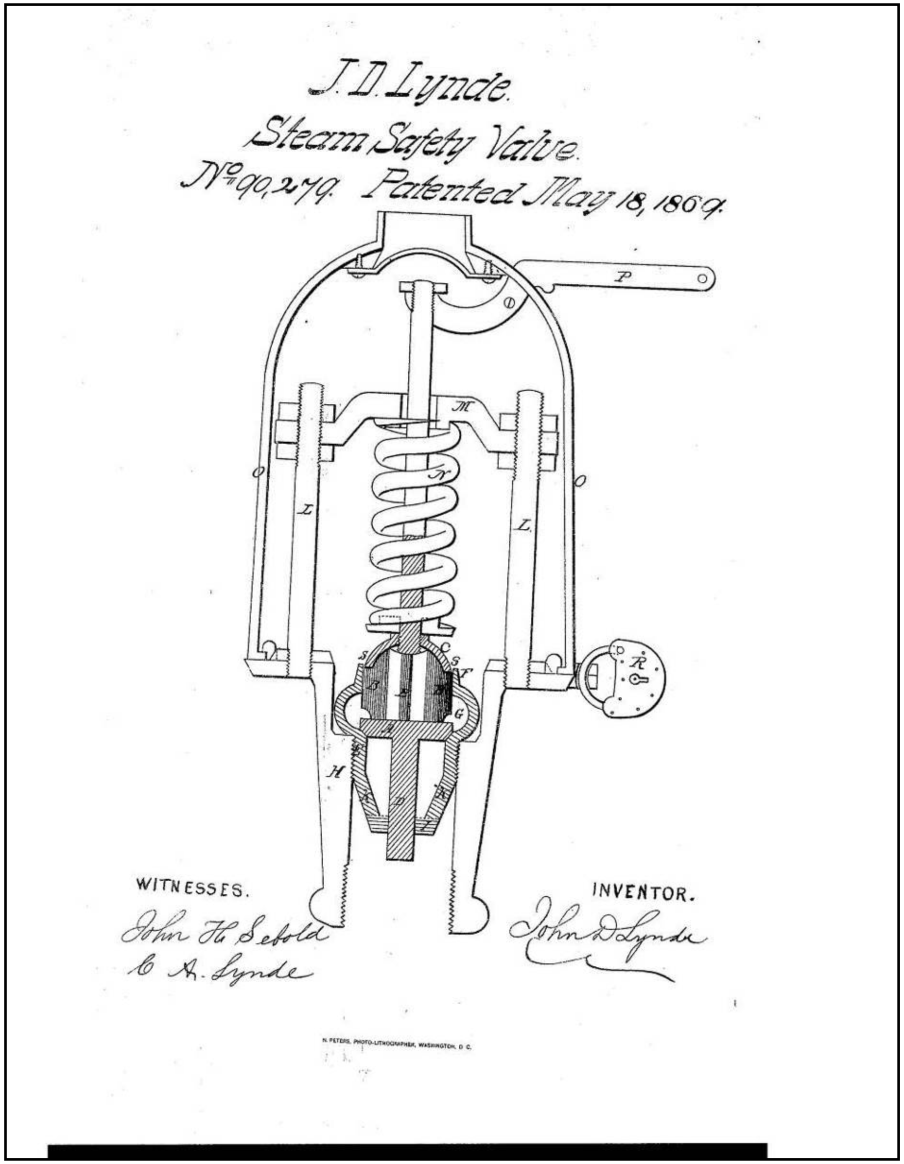


Other 1800 Developments

- Tamper Proofing
- Process Valves
- Soft Seat
- Deadweight Conclusion

1869

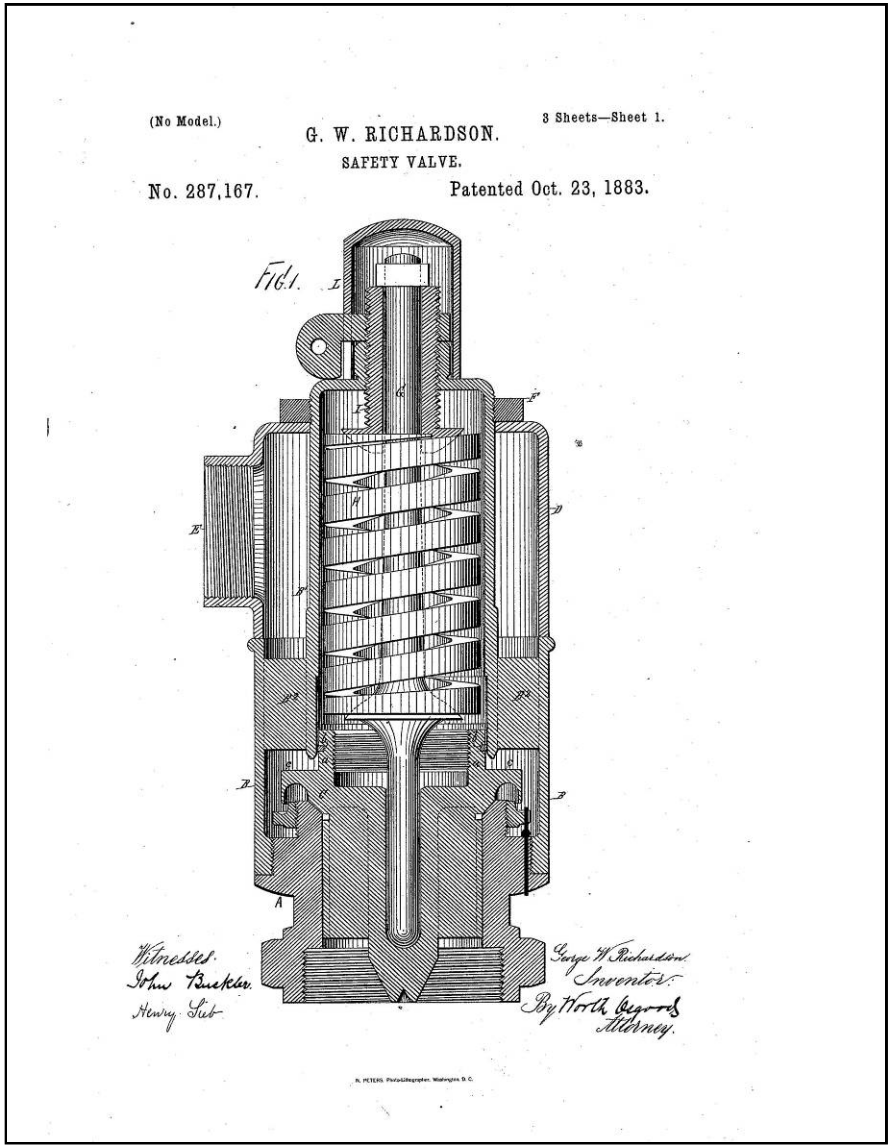
- Tamper Proofing
- Lifting Lever



1883

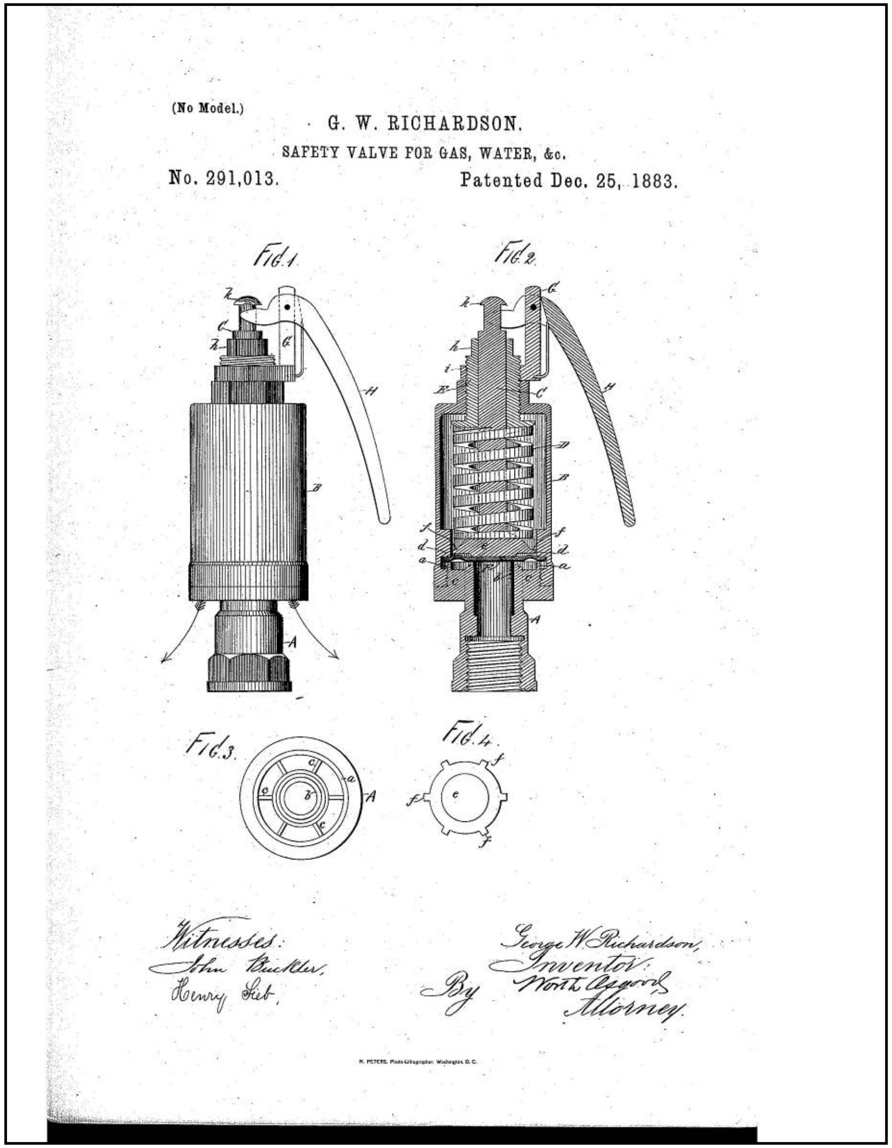
•First Closed Bonnet

•First Right Angle Outlet



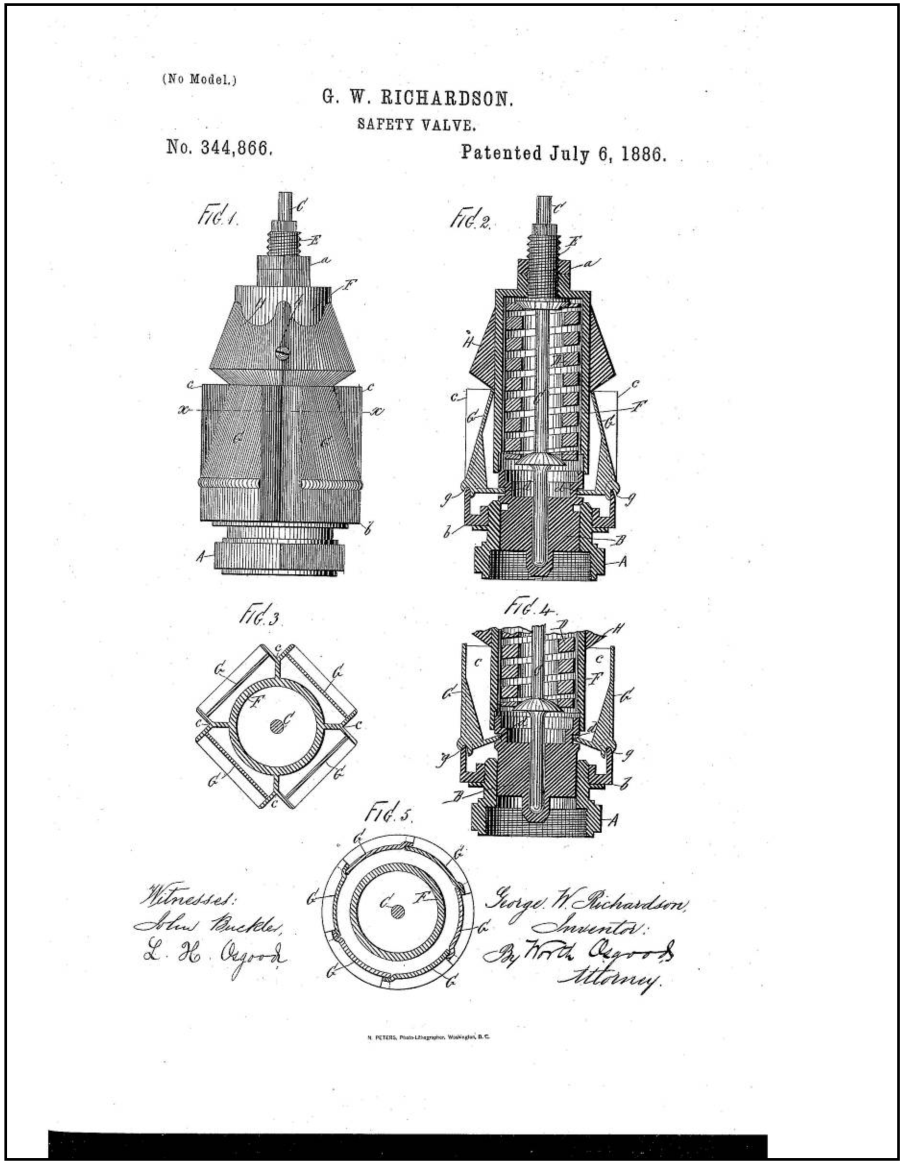
1883

- Process Fluids
- First Soft Seat



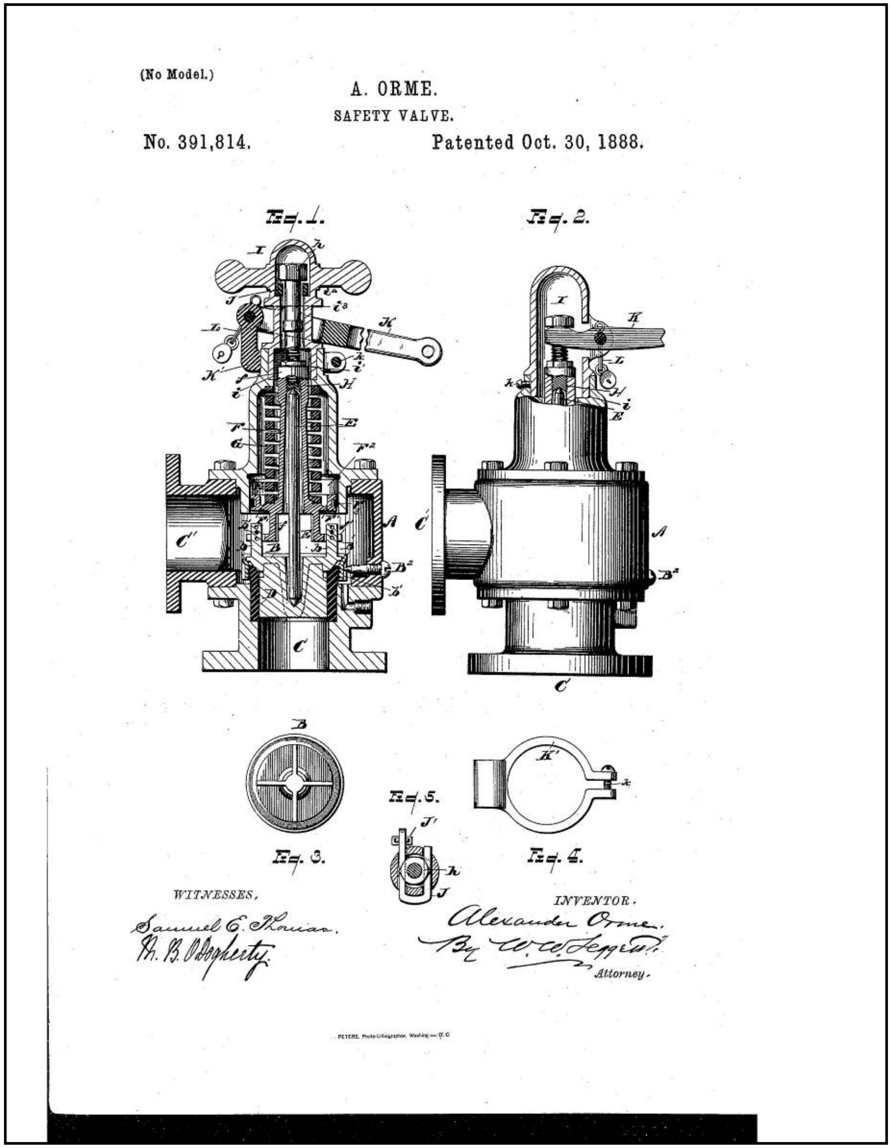
1886

•Not All Ideas are Good



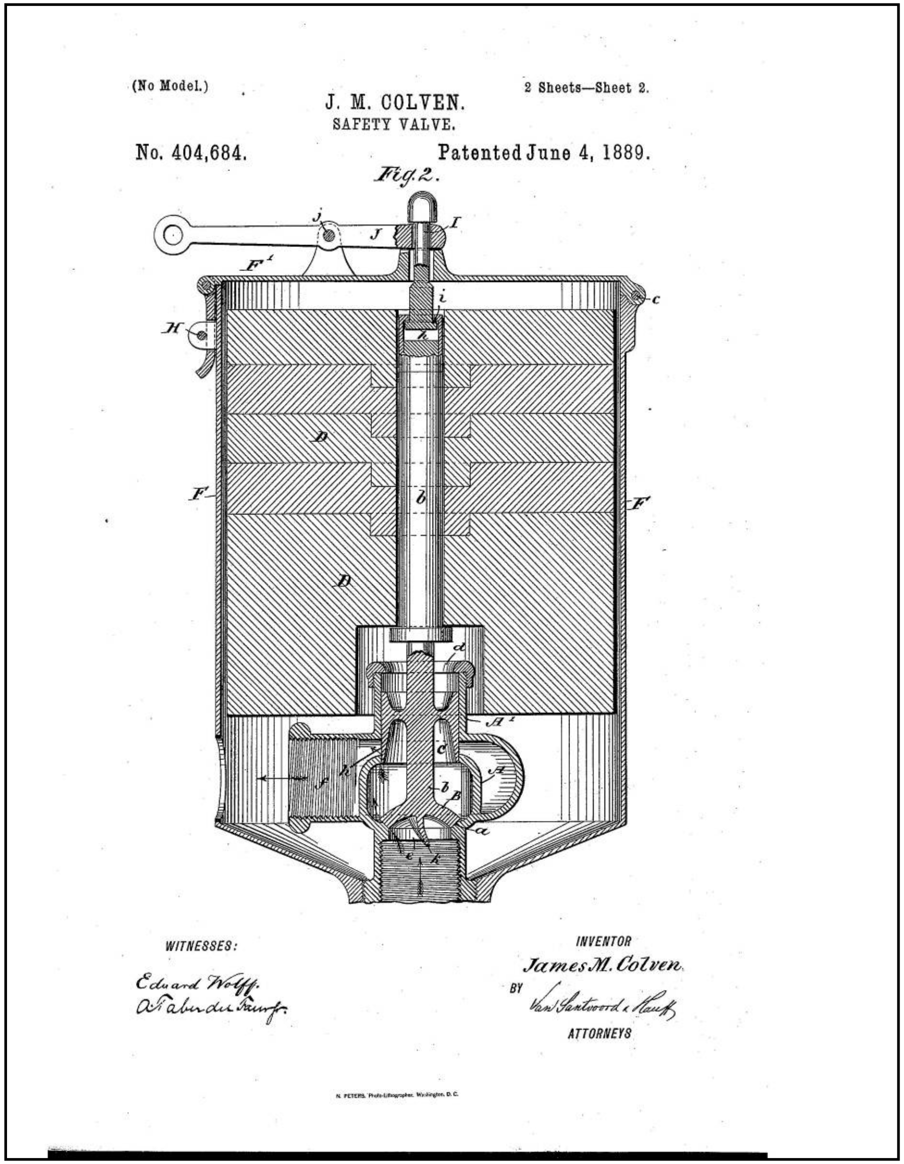
1888

- First Example
- Process Safety Valve
- Tamperproof Patent



1889

- Last Patent Noted
- Deadweight Safety Valve

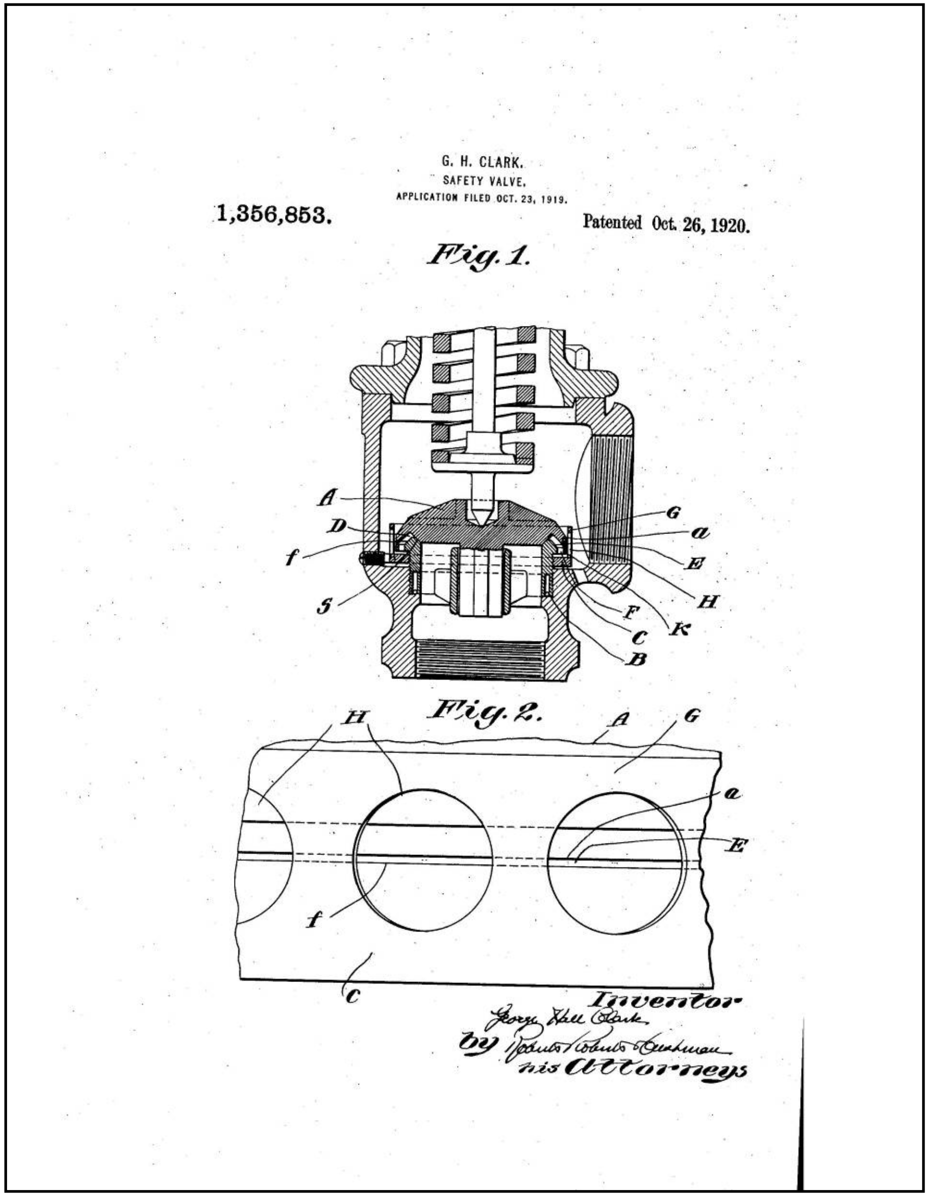


Blowdown Control

- With High Lift and Capacity Attain
- Attention Focused on Blowdown Control
- Minimize Steam Waste

1920

- Lower Ring with Orifices
- Crosby Valve Patent

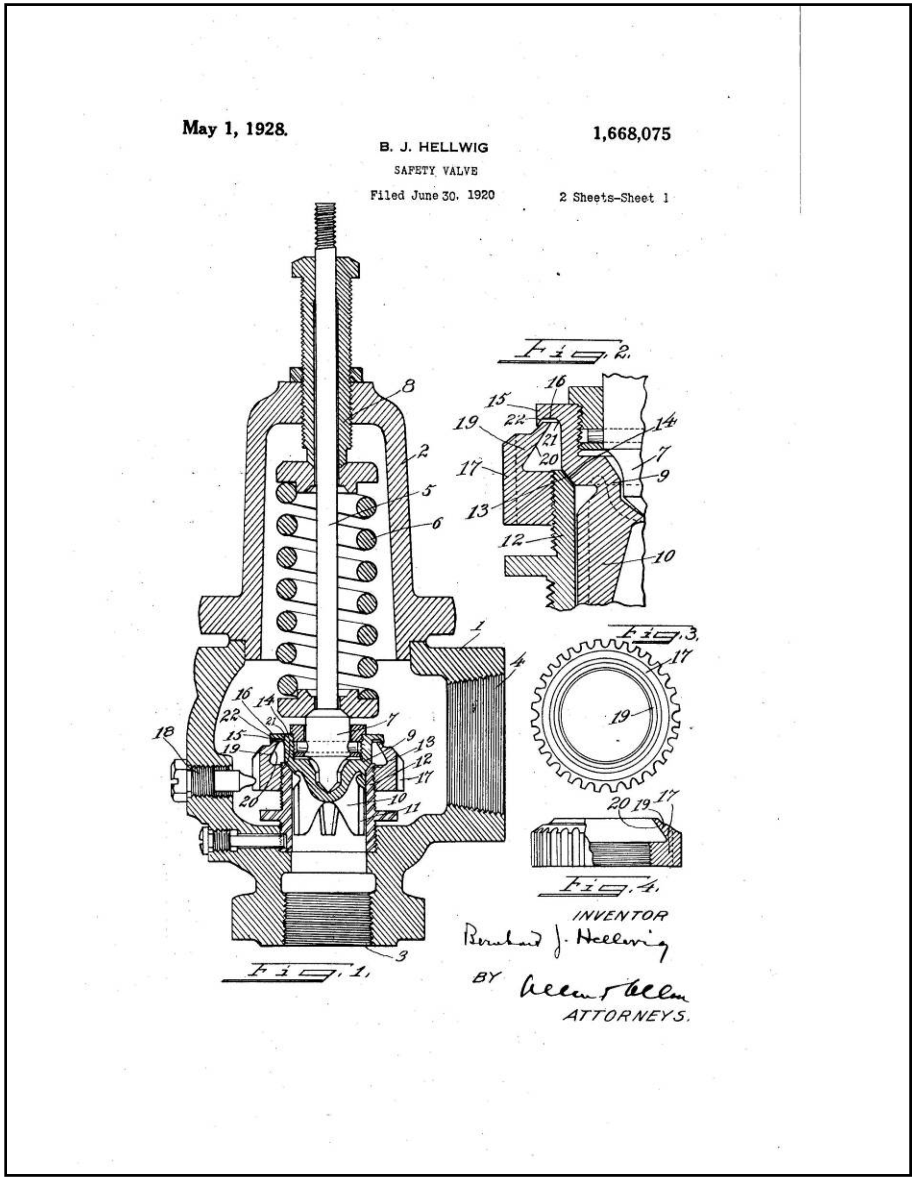


1928

•Lower Ring Geometry

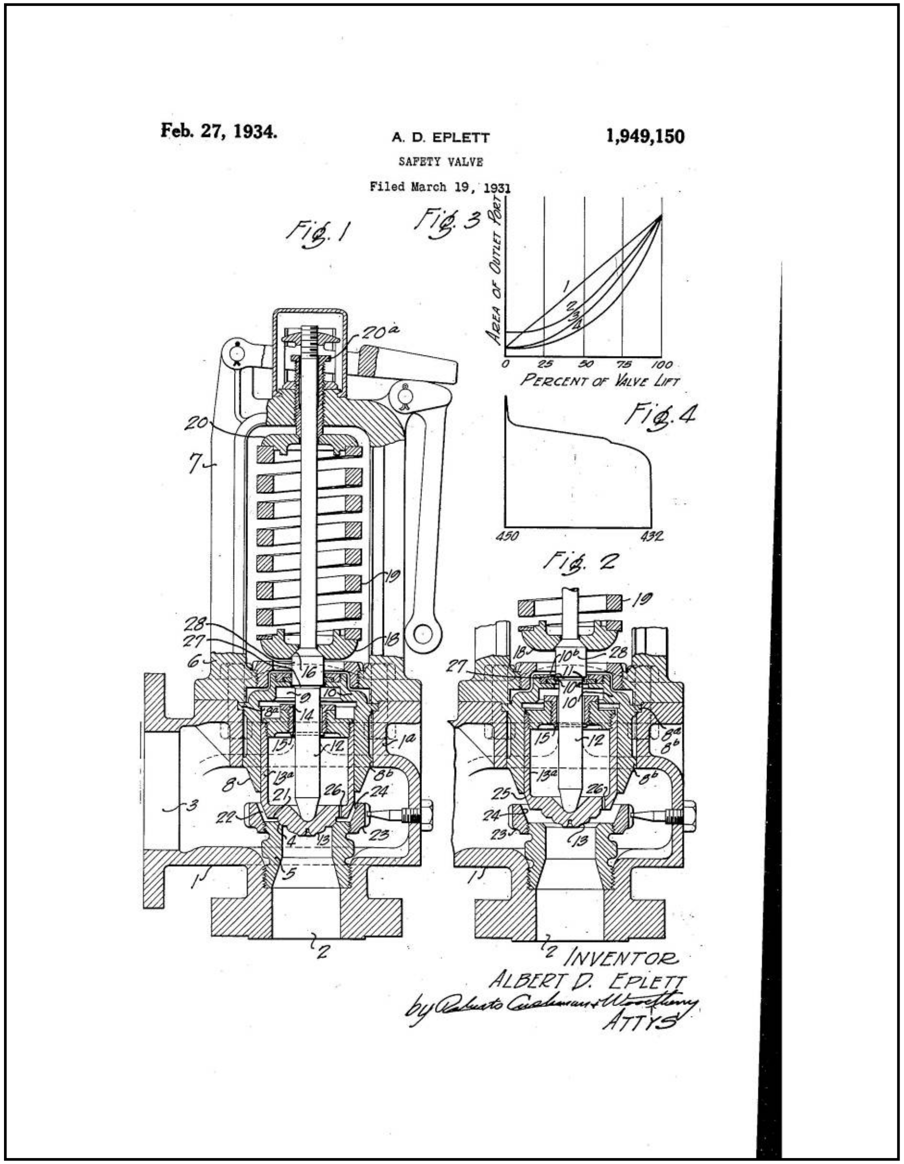
•Improved Opening Action

•Blowdown Action



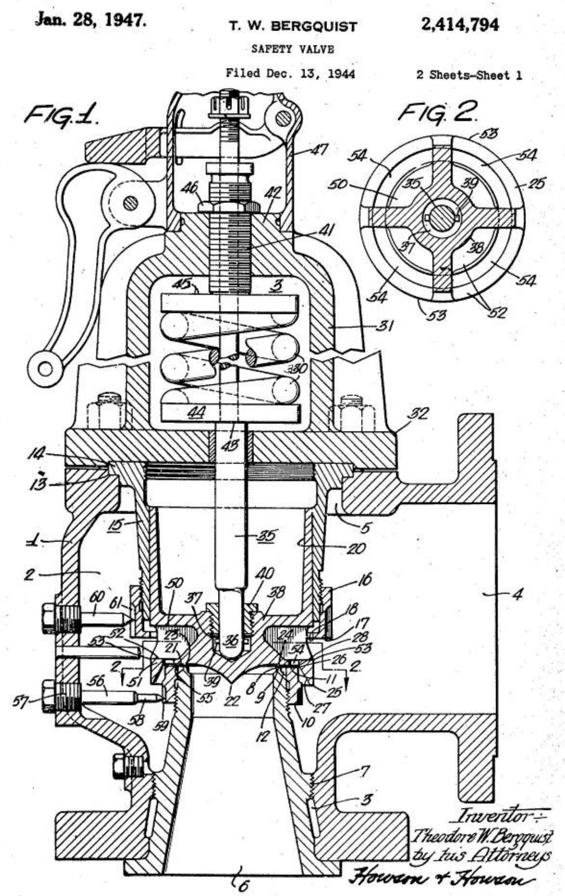
1934

•Pressure Assisted Closing



1947

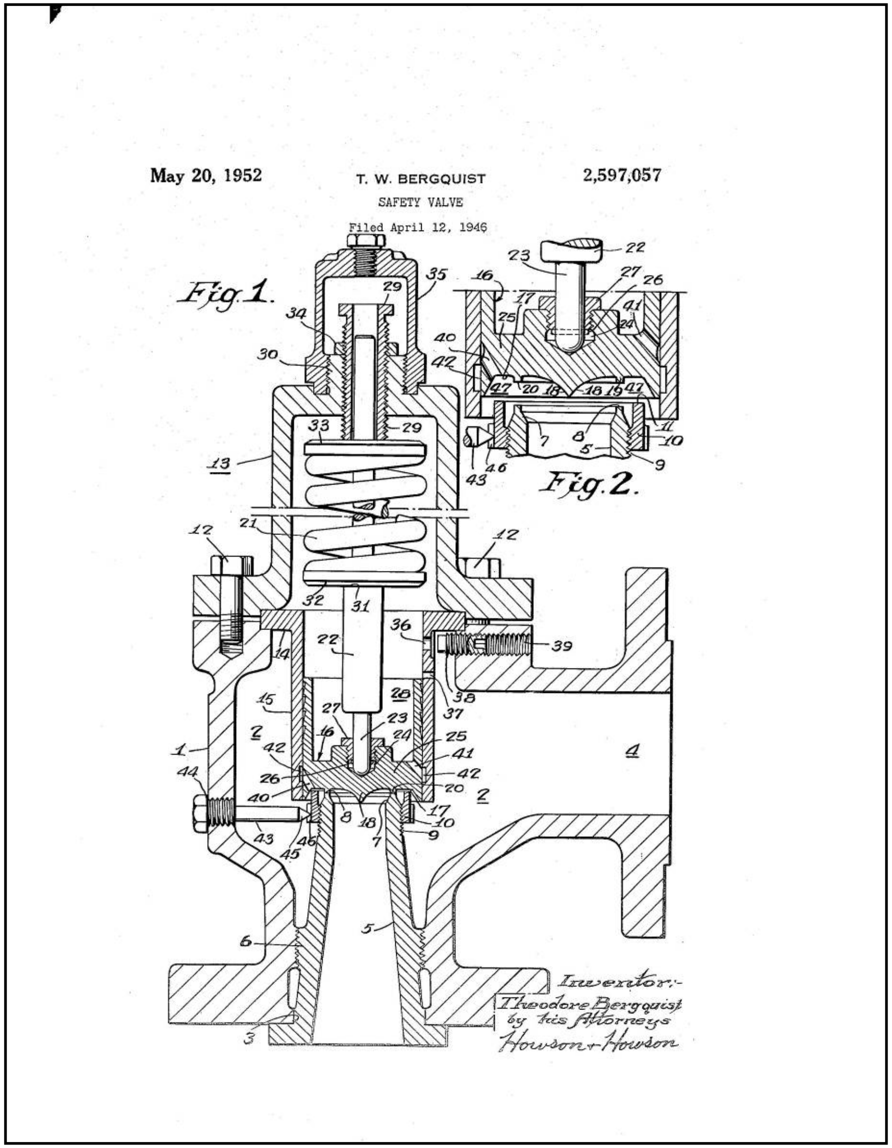
•First Two Ring Safety Valve



1952

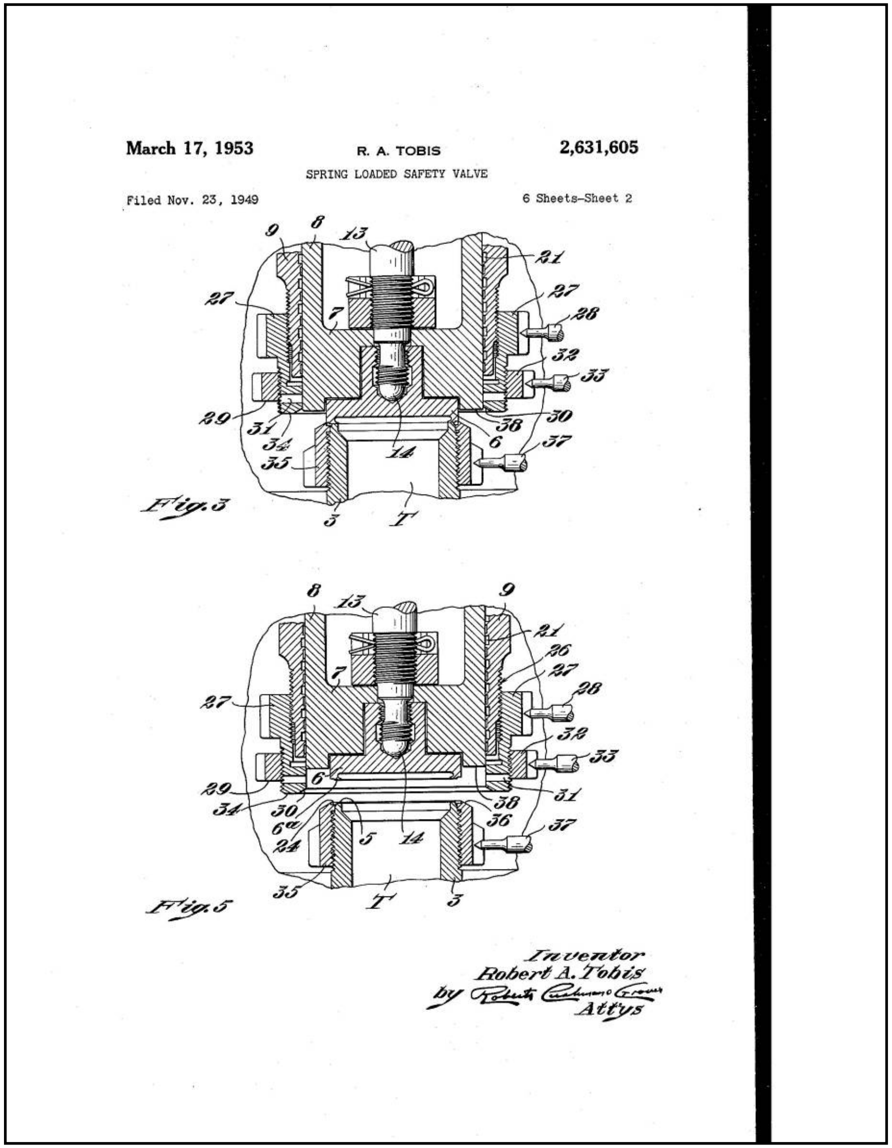
•Pressure Closing Assist

•External Adjustment



1953

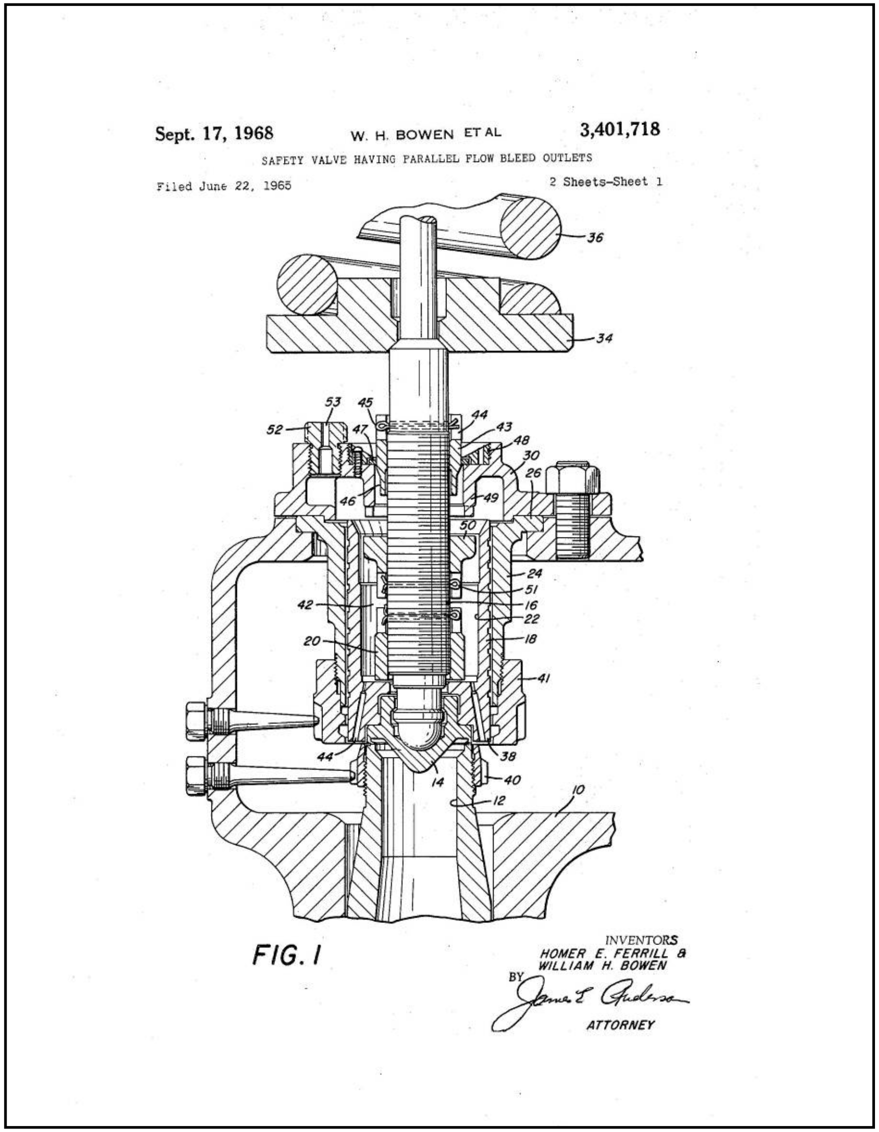
- Three Ring Design
- Improved Blowdown Control



1968

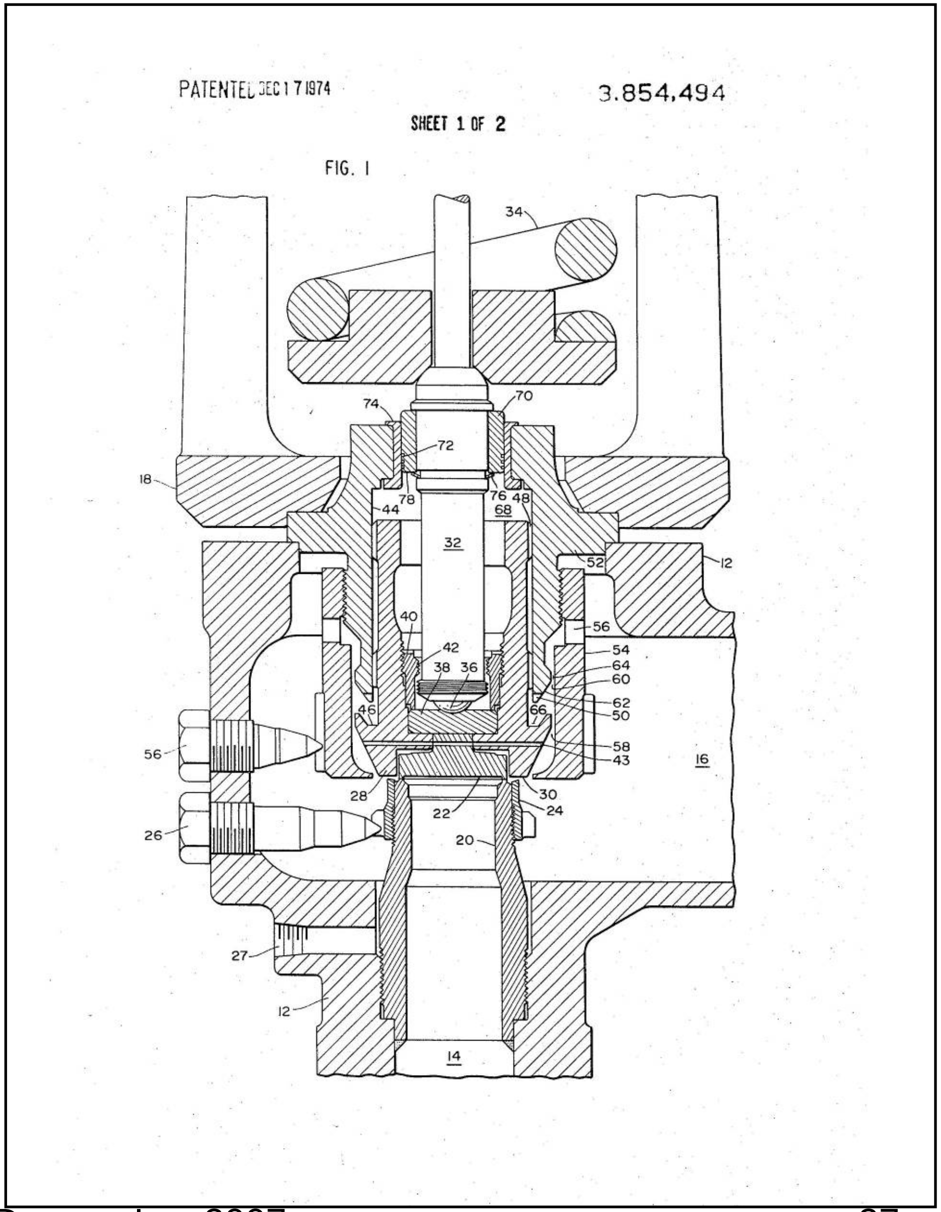
•Controlled Venting

•Pressure Assisted Closing



1974

•Pressure Assisted Closing

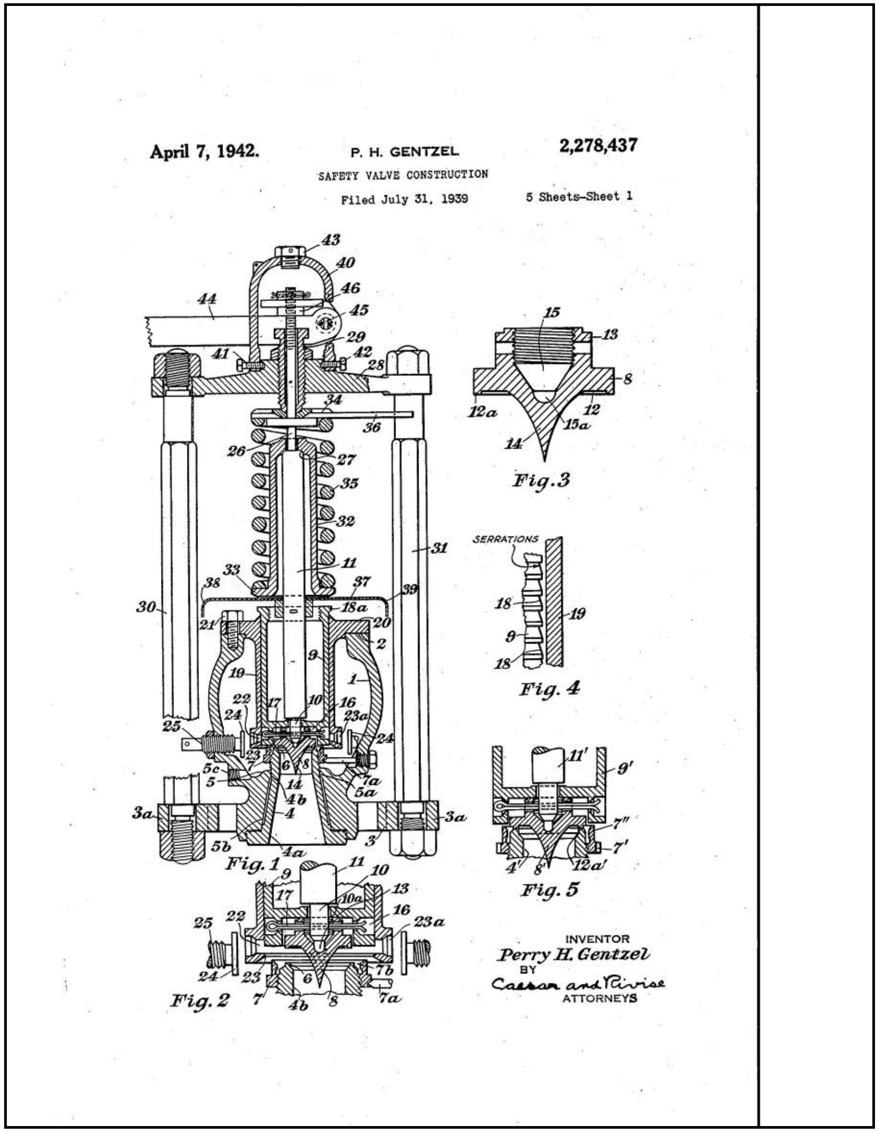


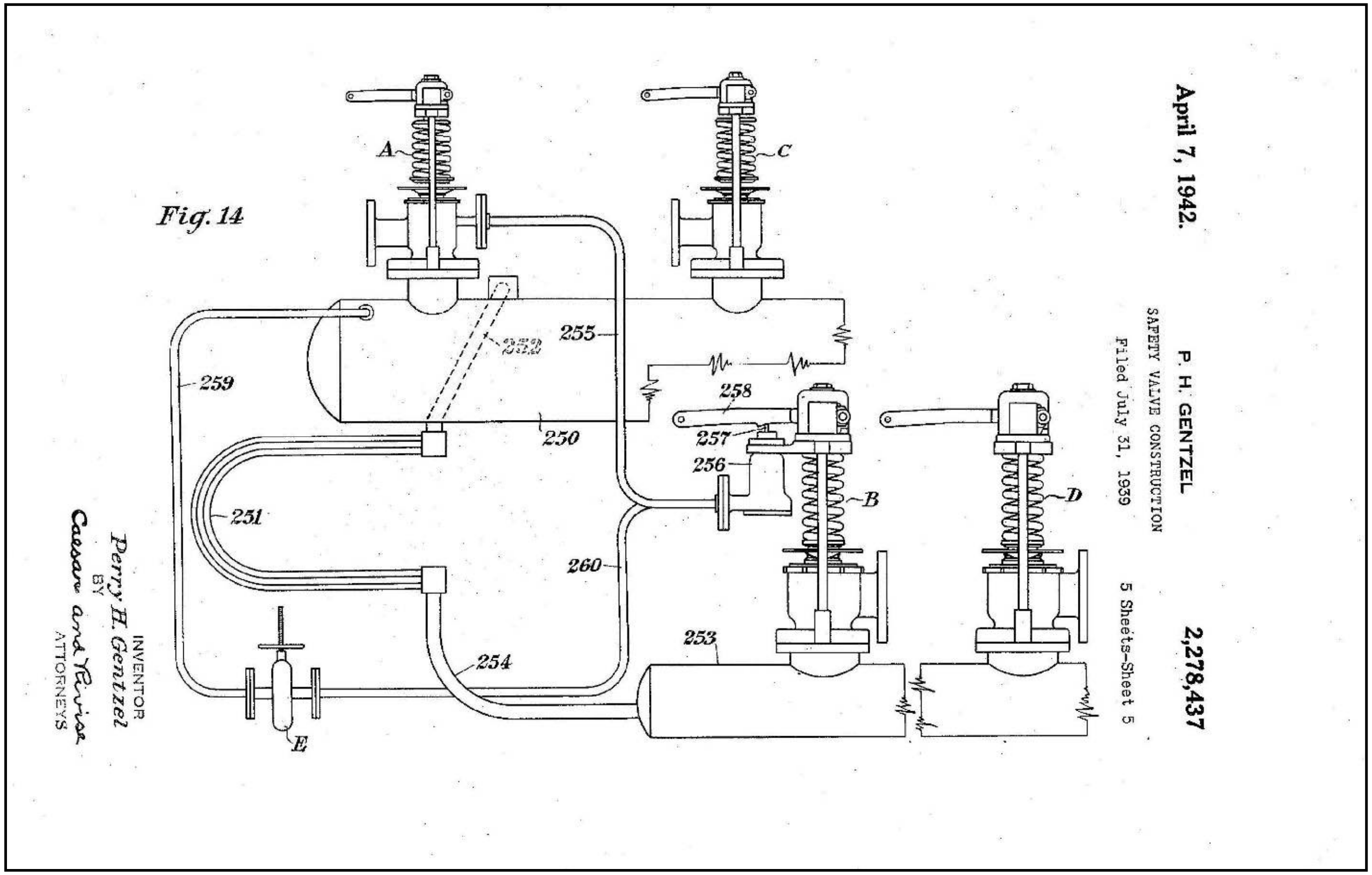
Other Developments

- Increased Flow
- Set Point Stability
- Pilot Operation

1942

- High Flow Disc Geometry
- Serrations on Discholder





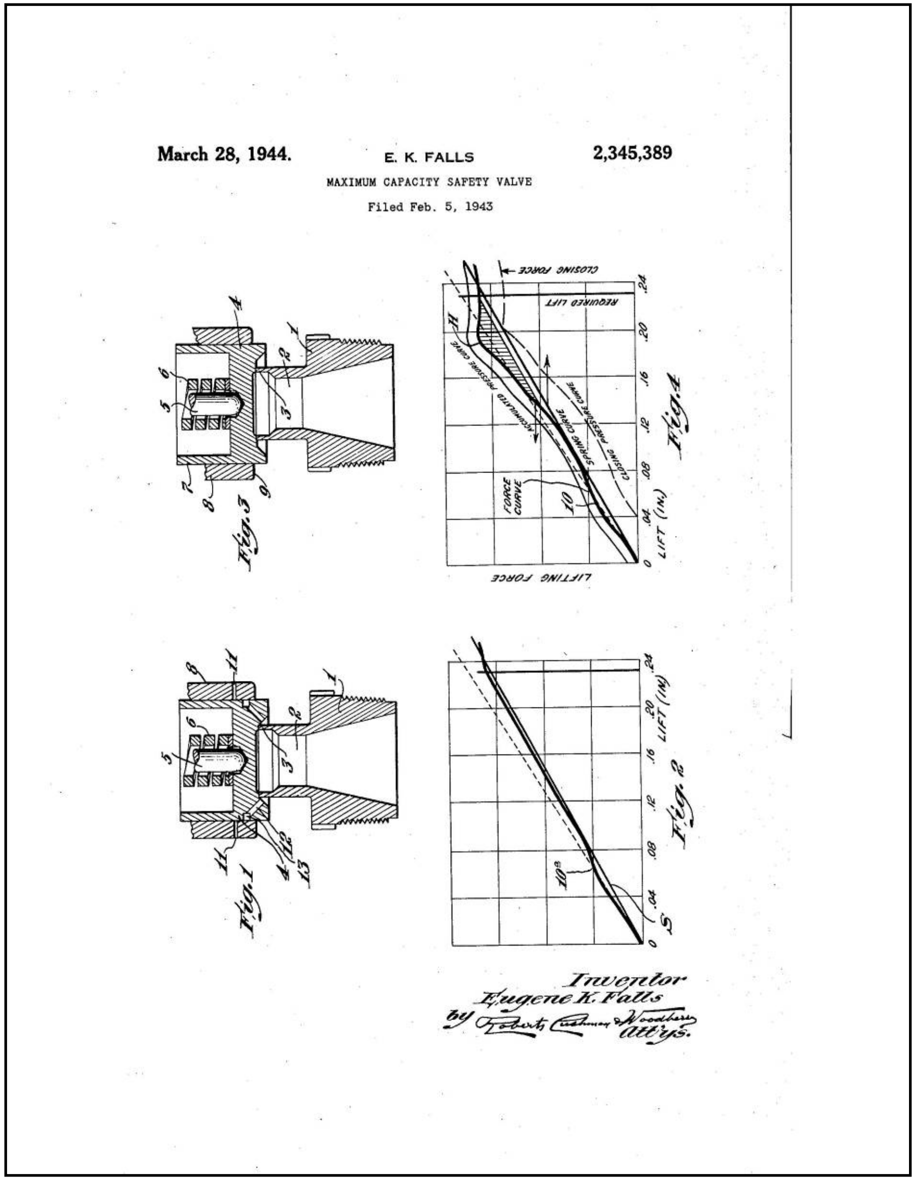
1942

•Pilot Operated

•Superheater Safety

1944

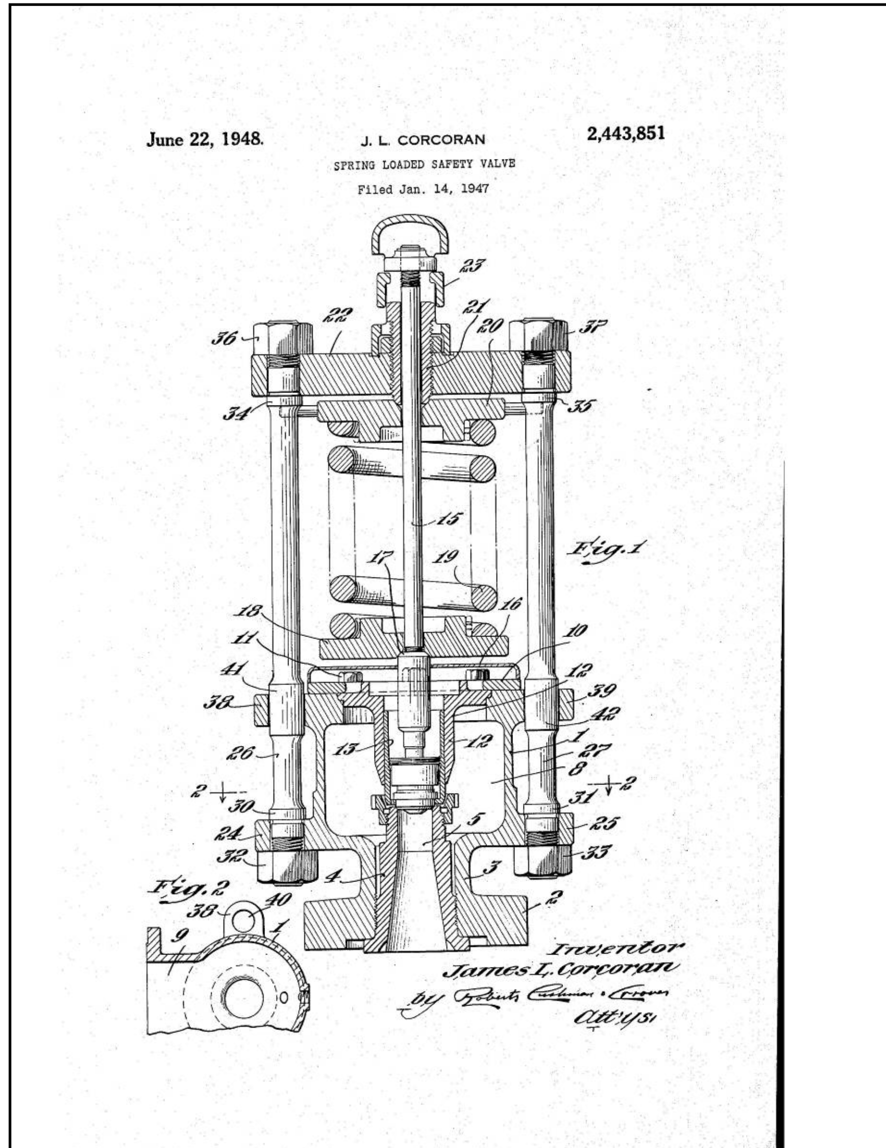
•High Flow Nozzle Design



1948

•Sturdy Side Rod Construction

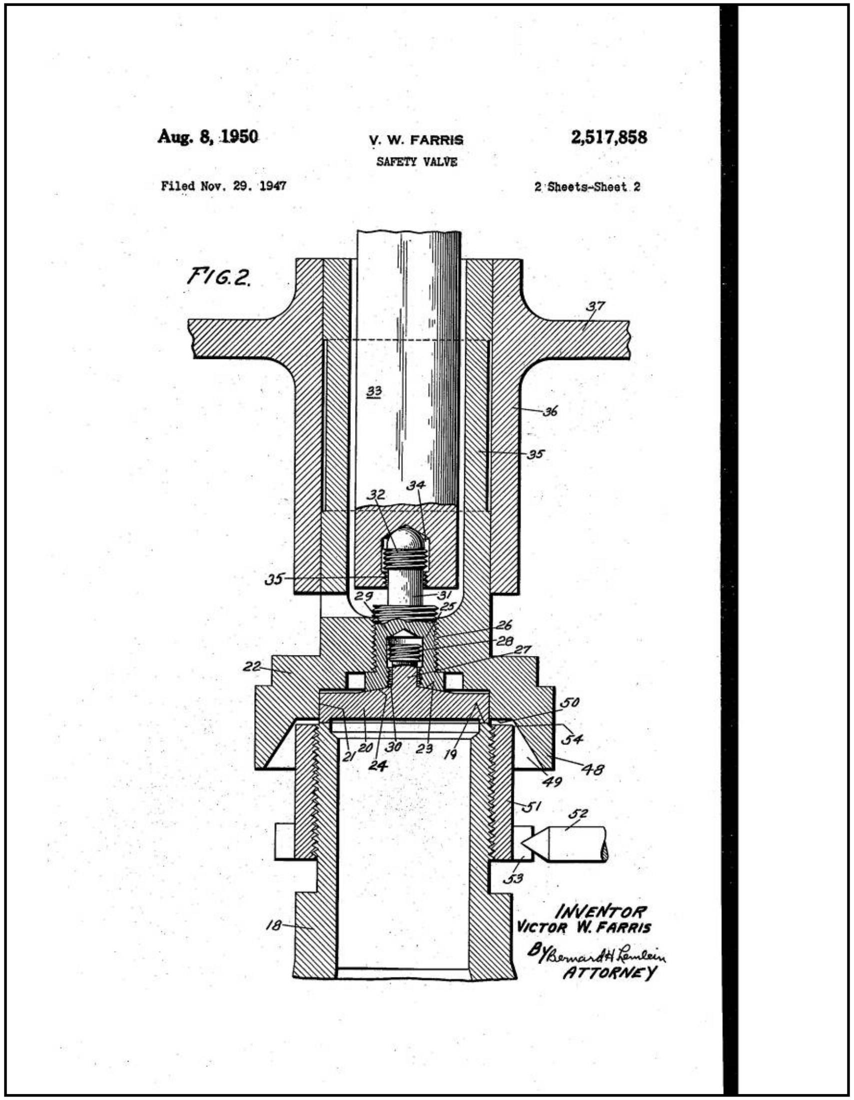
•Set Pressure Variation



1950

•2:1 Guiding

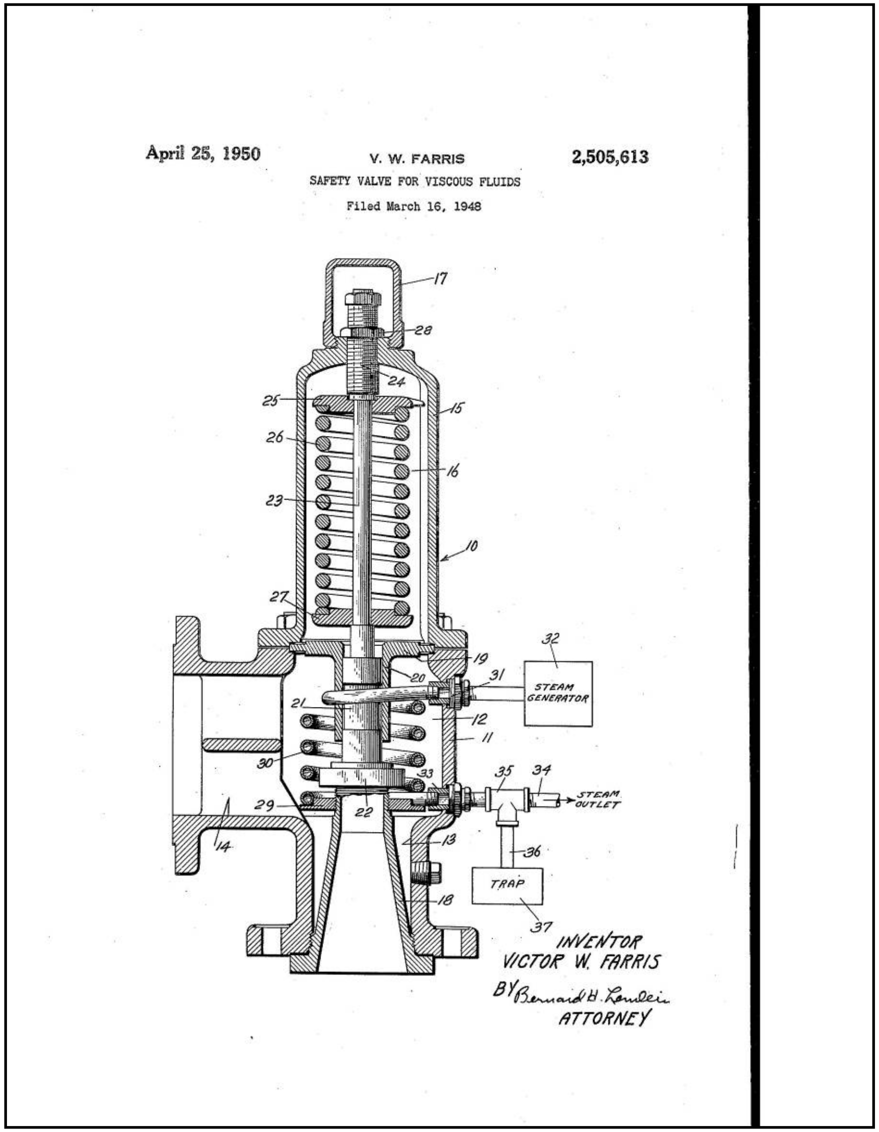
•Disc/Discholder



1950

•Heating

•Highly Viscous Liquids

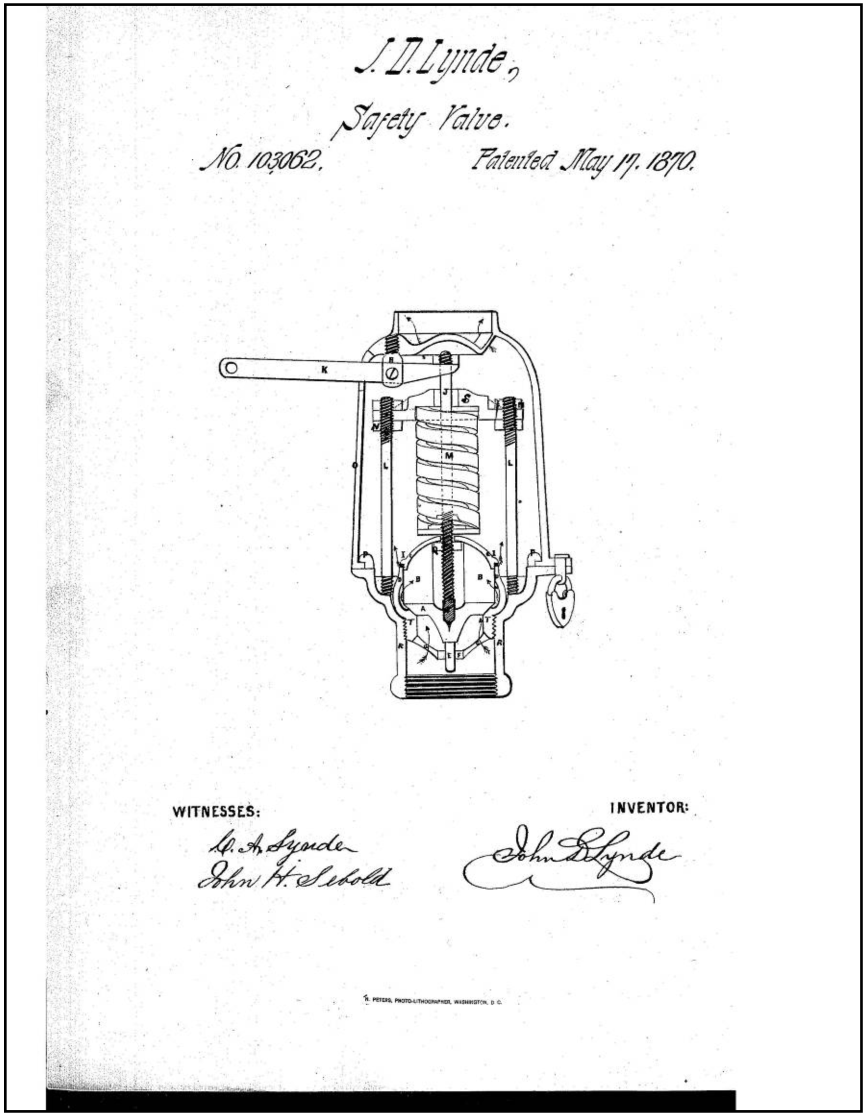


Seat Tightness

- Seat Tightness – Operating Pressure
- Flexible Disc
  - Increased Unit Loading
  - Negated Temperature Effects of Throttling

1870

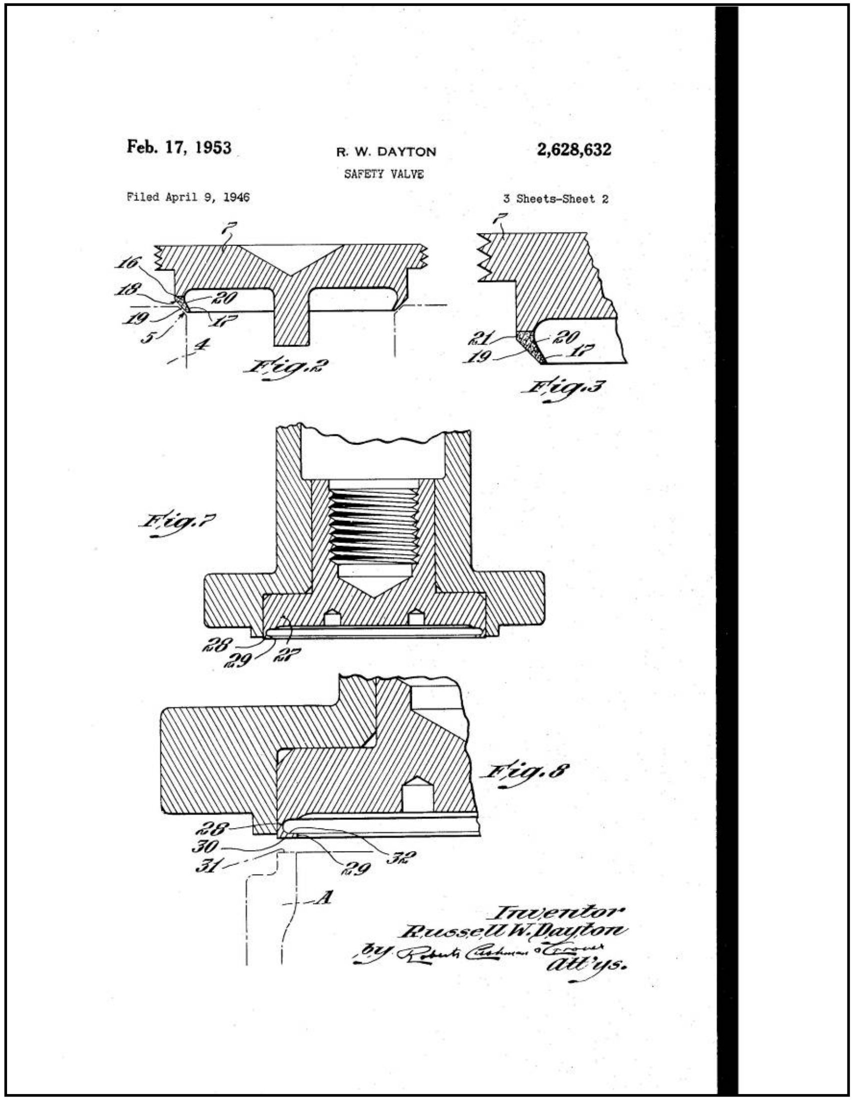
•Bearing Point Below Valve Seat



1953

•Flexible Disc

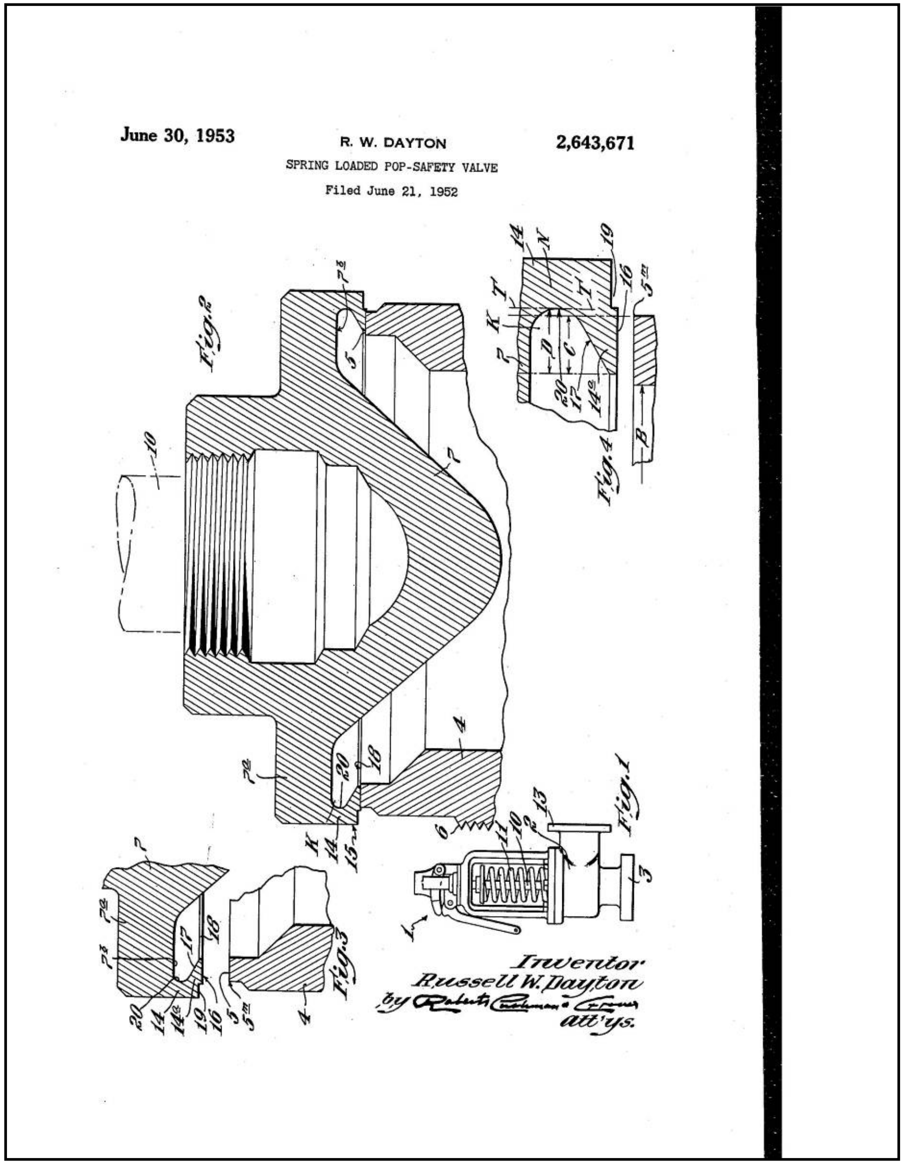
•Improved Seat Tightness



1953

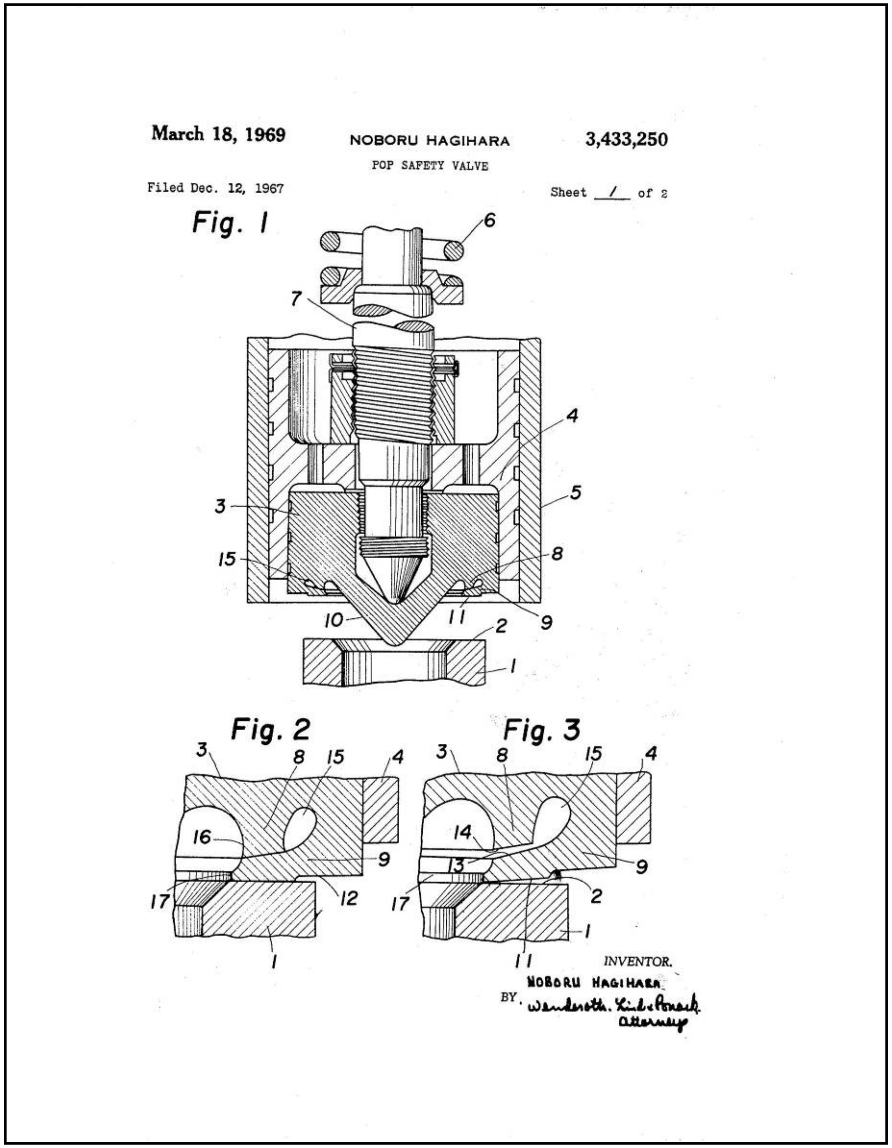
•Flexible Disc

•Most Common Usage



1969

- Supported Flexible Disc
- Questionable Manufacturing



1982

- Supplementary Loading
- Improved Seat Tightness

United States Patent [19]
Richter et al.

[11] **4,362,183**
[45] **Dec. 7, 1982**

[54] **SPRING LOADED SAFETY VALVE**

[75] Inventors: **Herbert Richter,** Weinheim; **Karl Schaaf,** Mannheim, both of Fed. Rep. of Germany

[73] Assignee: **Bopp & Reuther GmbH,** Mannheim, Fed. Rep. of Germany

[21] Appl. No.: **162,397**

[22] Filed: **Jun. 23, 1980**

[30] **Foreign Application Priority Data**

Jun. 30, 1979 [DE] Fed. Rep. of Germany 2926522

[51] **Int. Cl.³** **F16K 17/06**
[52] **U.S. Cl.** **137/489;** 92/13.5; 137/492.5; 251/63.4
[58] **Field of Search** 137/485, 489, 492.5; 251/63.4; 92/13.5

[56] **References Cited**

U.S. PATENT DOCUMENTS

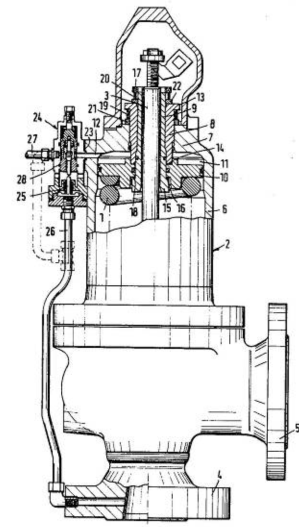
3,512,549 5/1970 Wiegand 137/489
3,865,132 2/1975 Wiegand .... 137/489 X

Primary Examiner—Robert G. Nilson
Attorney, Agent, or Firm—Michael J. Striker

[57] **ABSTRACT**

A spring loaded safety valve includes a control device which admits the pressurized medium to a supplemental loading system which consists of a spring housing which serves as a cylinder for a spring retainer which serves as a supplemental loading piston that is arranged about two adjustable sleeves which are arranged around the valve stem and which control the response pressure and the supplemental loading of the valve spring.

9 Claims, 1 Drawing Figure



1994

•Last Flexible Disc Patent

US005370151A

United States Patent [19]

Smart

[11] **Patent Number: 5,370,151**

[45] **Date of Patent: Dec. 6, 1994**

[54] **SAFETY VALVE**

[75] Inventor: **Larry W. Smart,** Winnfield, La.

[73] Assignee: **Dresser Industries Inc.,** Dallas, Tex.

[21] Appl. No.: **216,927**

[22] Filed: **Mar. 23, 1994**

[51] **Int. Cl.**^{5} **F16K 17/08**

[52] **U.S. Cl.** **137/468**; 137/469

[58] **Field of Search** 137/468, 469, 478, 536, 137/542

[56] **References Cited**

U.S. PATENT DOCUMENTS

733,641	11/1904	Hayden	137/477 X
2,264,656	12/1941	Briscoe et al.	137/536
3,433,250	3/1969	Hagihara	137/469
4,708,164	11/1987	Scallan	137/476
4,858,642	8/1989	Fain, Jr.	137/474
5,011,116	4/1991	Alberts et al.	137/469 X
5,234,023	8/1993	Lai	137/489

OTHER PUBLICATIONS

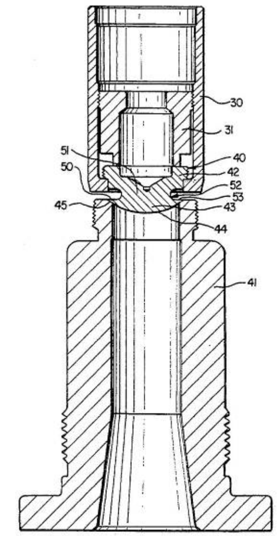
Bulletin SV-7 1555, 1556 and 1557 Cast Steel Safety Valves published Jul. 1986 by Industrial Valve Operation, Dresser Valve and Controls Division.

Primary Examiner—Stephen M. Hepperle
Attorney, Agent, or Firm—Johnson & Wortley

[57] **ABSTRACT**

A safety valve for releasing excessive fluid pressure from a pressure vessel including a valve body having an inlet and an outlet, a seat bushing connected into the inlet defining an inlet flow passage, a valve seat on the inward end of the seat bushing, a valve disc having a seat thereon movably mounted in the valve body for opening and closing the valve, a spring coupled with the valve disc for biasing the valve disc toward a closed position. The valve disc includes a convex end portion to reduce turbulence during flow into the valve to permit improved seating of the valve disc and an external annular recess defining an annular lip on the valve disc including a seat having a frustoconical seat surface, the lip being flexible to improve the disc seat contact with the seat around the valve inlet to increase the seat tightness of the valve.

5 Claims, 4 Drawing Sheets



Back Pressure

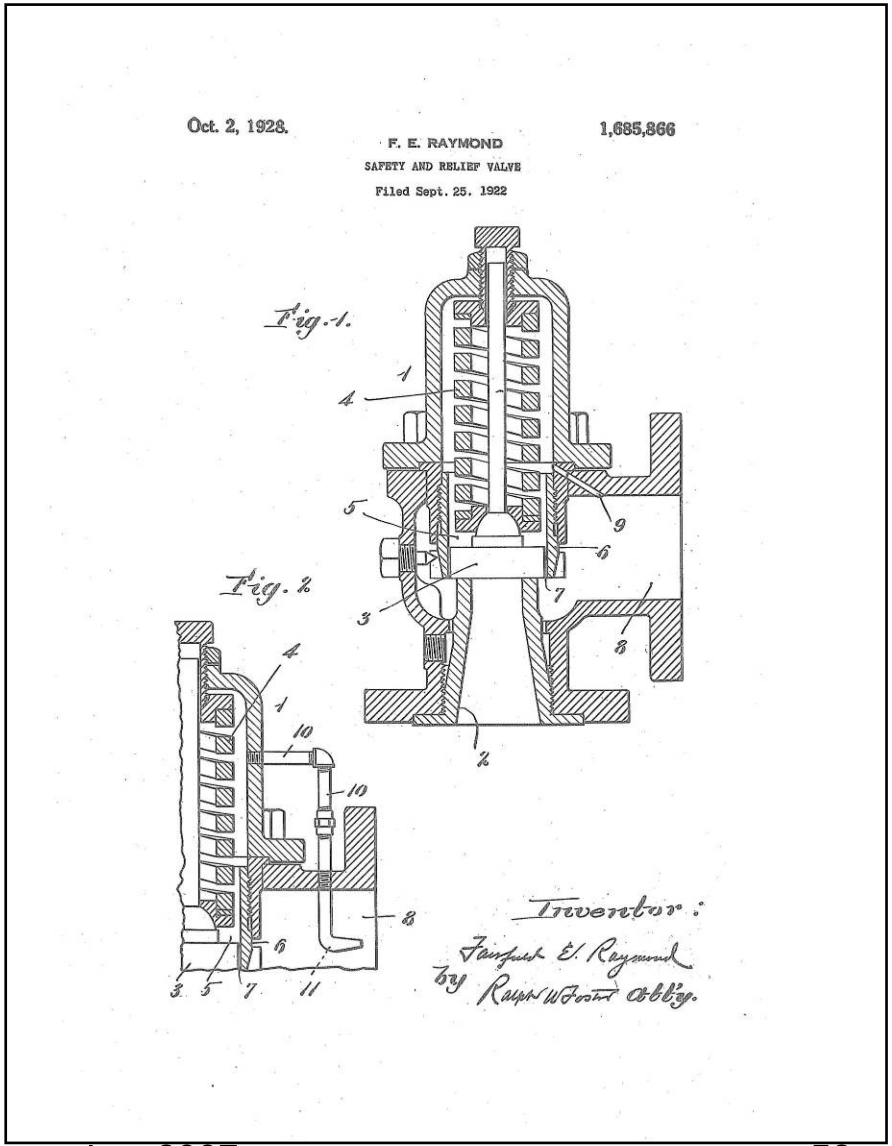
- Bellows
- Bonnet Venting
- Pressure Assist
- Combat Actions of Back Pressure

1928

•Venting

•Bonnet Pressure

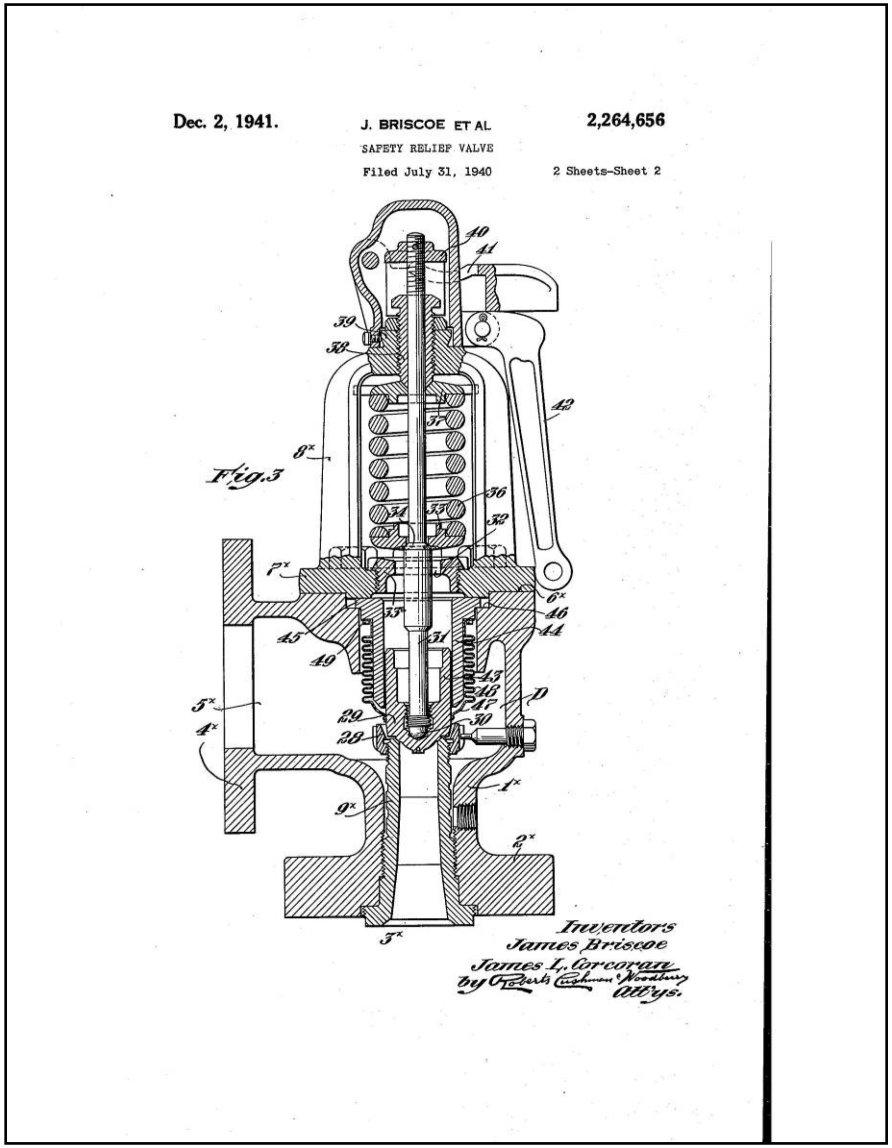
•Non-Wing Guided



1941

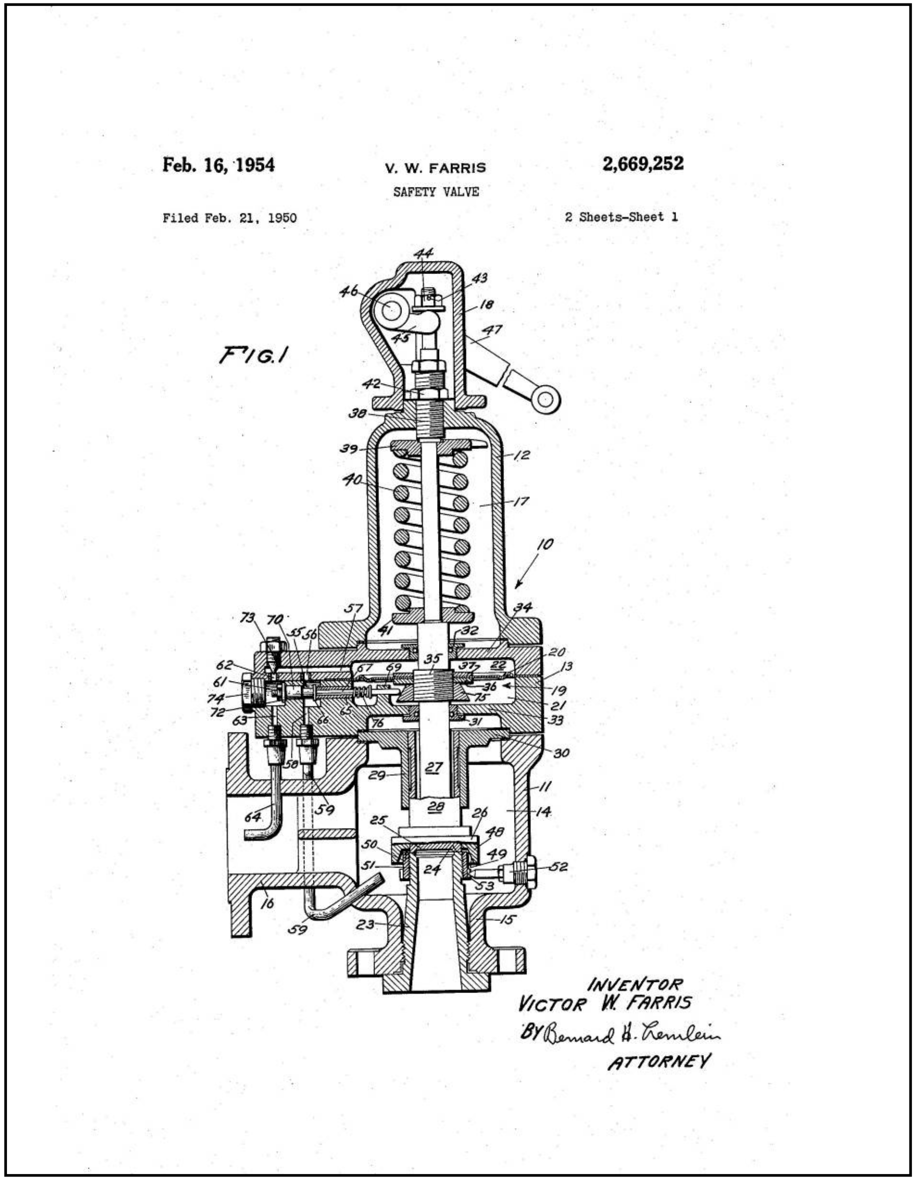
•Bellows

•Backpressure Effects



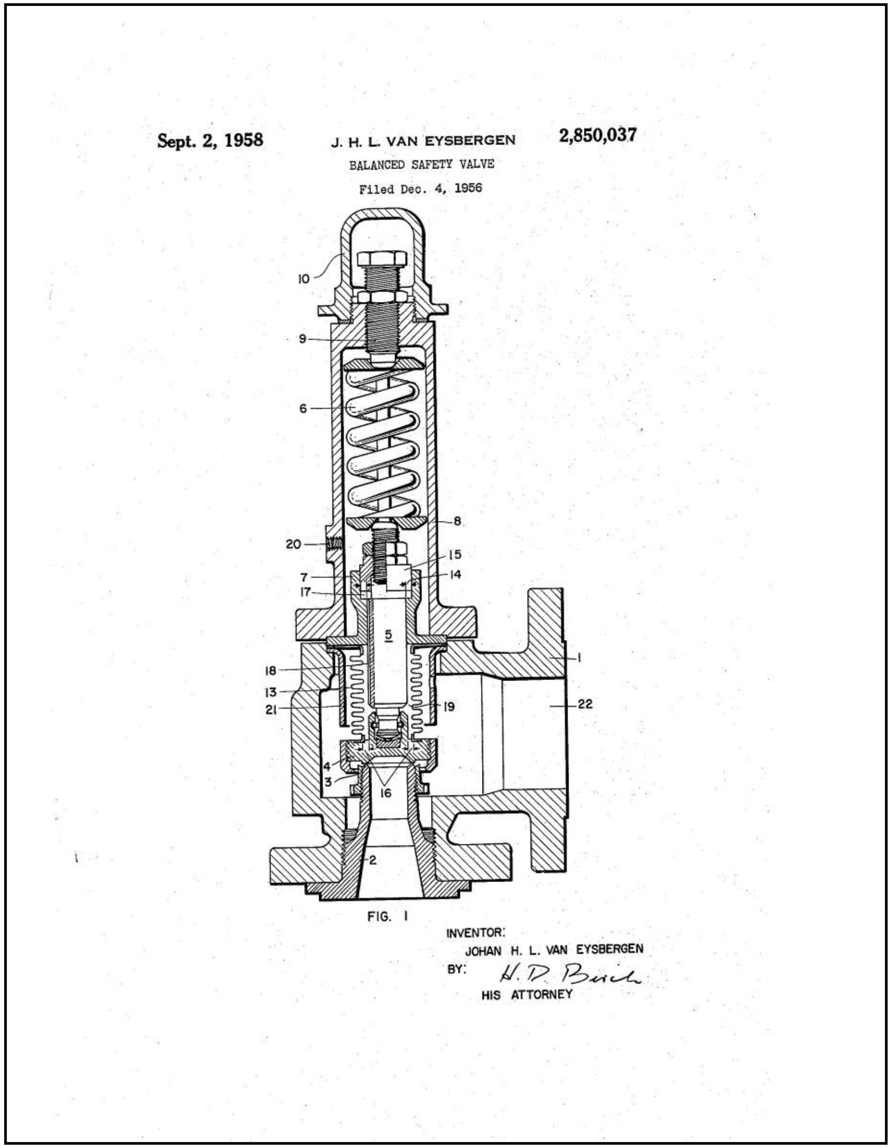
1954

•Pressure Assisted Opening



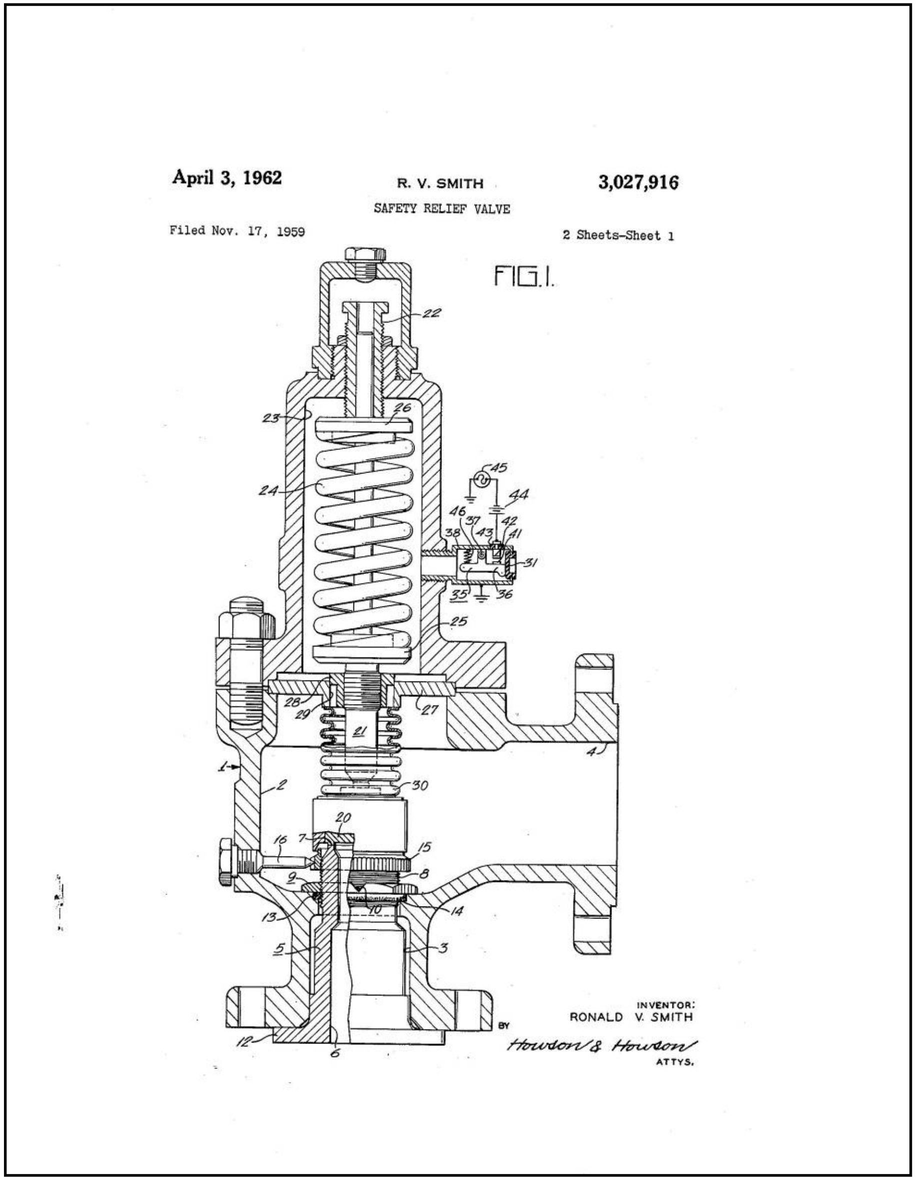
1958

- •Balanced Bellows
- •Piston Backup
- •Shell® Patent
- •ISO/CEN 4126-1 Requirement



1962

•Bellows Leakage Detector



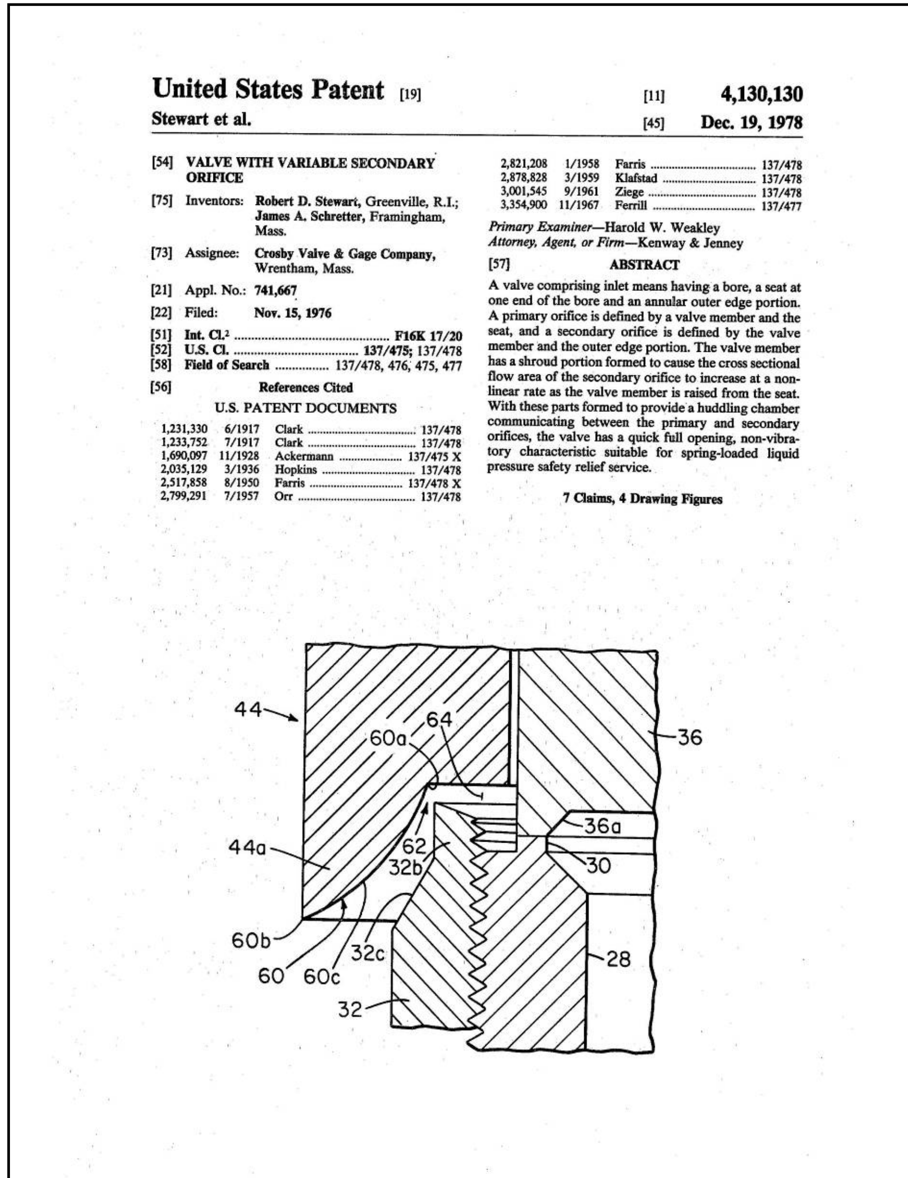
Liquid Applications

- Liquid Applications Prone to Chatter
- External and Fluid Force Designs

1978

•First Liquid Specific

•Chatter Elimination



United States Patent [19] [11] **4,130,130**

Stewart et al. [45] **Dec. 19, 1978**

[54] **VALVE WITH VARIABLE SECONDARY ORIFICE**

[75] Inventors: **Robert D. Stewart**, Greenville, R.I.; **James A. Schretter**, Framingham, Mass.

[73] Assignee: **Crosby Valve & Gage Company**, Wrentham, Mass.

[21] Appl. No.: **741,667**

[22] Filed: **Nov. 15, 1976**

[51] **Int. Cl.**[2] **F16K 17/20**

[52] **U.S. Cl.** **137/475**; 137/478

[58] **Field of Search** 137/478, 476, 475, 477

[56] **References Cited**

U.S. PATENT DOCUMENTS

1,231,330	6/1917	Clark	137/478
1,233,752	7/1917	Clark	137/478
1,690,097	11/1928	Ackermann	137/475 X
2,035,129	3/1936	Hopkins	137/478
2,517,858	8/1950	Farris	137/478 X
2,799,291	7/1957	Orr	137/478
2,821,208	1/1958	Farris	137/478
2,878,828	3/1959	Klafstad	137/478
3,001,545	9/1961	Ziege	137/478
3,354,900	11/1967	Ferrill	137/477

Primary Examiner—Harold W. Weakley
Attorney, Agent, or Firm—Kenway & Jenney

[57] **ABSTRACT**

A valve comprising inlet means having a bore, a seat at one end of the bore and an annular outer edge portion. A primary orifice is defined by a valve member and the seat, and a secondary orifice is defined by the valve member and the outer edge portion. The valve member has a shroud portion formed to cause the cross sectional flow area of the secondary orifice to increase at a non-linear rate as the valve member is raised from the seat. With these parts formed to provide a huddling chamber communicating between the primary and secondary orifices, the valve has a quick full opening, non-vibratory characteristic suitable for spring-loaded liquid pressure safety relief service.

7 Claims, 4 Drawing Figures

1984

•Chatter Elimination

•Friction Elements

United States Patent [19] — **Schmitt et al.**

[11] **Patent Number: 4,481,974**

[45] **Date of Patent: Nov. 13, 1984**

[54] **SAFETY VALVE WITH FRICTION MEANS FOR DAMPING VALVE VIBRATION**

[75] Inventors: **Manfred Schmitt,** Friedelsheim; **Emil Zitzelsberger,** Heppenheim/Kirschhausen, both of Fed. Rep. of Germany

[73] Assignee: **Bopp & Reuther GmbH,** Mannheim, Fed. Rep. of Germany

[21] Appl. No.: **424,977**

[22] Filed: **Sep. 27, 1982**

[30] **Foreign Application Priority Data**

Mar. 24, 1982 [DE] Fed. Rep. of Germany 3210768

[51] **Int. Cl.**[3] **F16K 17/04**

[52] **U.S. Cl.** **137/514;** 137/542; 188/67; 188/381; 251/64

[58] **Field of Search** .... 137/514, 542, DIG. 5; 188/67, 381; 251/64

[56] **References Cited**

U.S. PATENT DOCUMENTS

1,608,392 11/1926 Hannah .... 188/67 X
2,017,297 10/1935 Sorensen 137/DIG. 5
3,848,632 11/1974 Powell 137/514
4,194,527 3/1980 Schonwald 137/514
4,357,956 11/1982 Anselmann 137/514

FOREIGN PATENT DOCUMENTS

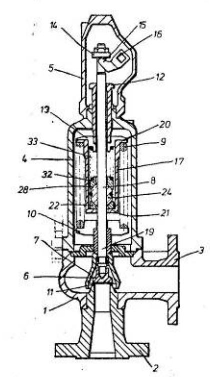
2654077 11/1976 Fed. Rep. of Germany .
1159648 2/1958 France 188/67
477276 7/1975 U.S.S.R. 137/514

Primary Examiner—Robert G. Nilson
Attorney, Agent, or Firm—Michael J. Striker

[57] **ABSTRACT**

A safety valve has movable and immovable valve elements, and a friction unit arranged therebetween for damping vibrations, wherein the friction unit has a plurality of friction members movable radially relative to a valve axis, a pressing member arranged to move the friction members so that their friction surfaces are pressed against a friction countersurface of a movable valve element of the movable valve means and for abutment against a stationary abutment member at their side facing toward a valve closing member, and a pretensioned auxiliary spring acting upon the friction members at their side facing away from the valve closing member and in a valve closing direction, wherein the auxiliary spring has such force and such a smaller resistance that in the event of seizing of the friction members with the movable valve element, the auxiliary spring is compressed within a predetermined opening pressure which does not considerably exceed a response pressure over approximately a valve stroke.

11 Claims, 5 Drawing Figures



1987

- Chatter Elimination
- Overlapping
    - Ring
    - Discholder

United States Patent [19]
Scallan

[11] **Patent Number:** **4,708,164**
[45] **Date of Patent:** **Nov. 24, 1987**

[54] **SAFETY RELIEF VALVE**

[75] Inventor: **David J. Scallan,** Pineville, La.

[73] Assignee: **Dresser Industries, Inc.,** Dallas, Tex.

[21] Appl. No.: **874,921**

[22] Filed: **Jun. 16, 1986**

[51] **Int. Cl.**^{4} **F16K 17/20**
[52] **U.S. Cl.** **137/476;** 137/477; 137/478
[58] **Field of Search** 137/476, 477, 478

[56] **References Cited**

U.S. PATENT DOCUMENTS

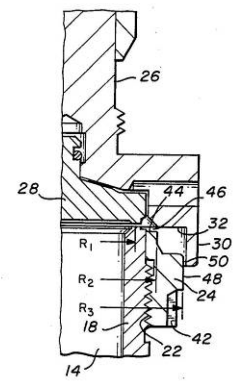
630,826	8/1899	Downs	137/478 O
1,231,330	6/1917	Clark	137/478
1,233,752	7/1917	Clark	137/478
1,696,452	12/1928	Raymond	137/477
2,261,461	11/1941	Falls	137/478
2,821,208	6/1958	Farris	137/428
3,354,900	11/1967	Ferrill	137/478 X
3,406,712	10/1968	Weise	137/477 X
3,572,372	3/1971	Moore	137/478 X
3,897,802	8/1979	Bass	137/428

Primary Examiner—Harold W. Weakley

[57] **ABSTRACT**

An improved safety relief valve that includes an adjusting ring located on the valve seat member. The adjusting ring has one end located to provide an orifice which will control the area governing the rate of closure or "blow down" of the valve and a much larger peripheral surface which cooperates with a valve member carrier to provide an orifice that governs the force applied to the valve carrier and valve member to lift the valve which has been previously "cracked."

1 Claim, 4 Drawing Figures



1993

•Chatter Elimination

•Shearing Current Dampening

US005224511A

United States Patent [19]

Schnettler

[11] **Patent Number:** **5,224,511**

[45] **Date of Patent:** **Jul. 6, 1993**

[54] **SPRING-LOADED SAFETY VALVE**

[75] Inventor: **Armin Schnettler,** Grevenbroich, Fed. Rep. of Germany

[73] Assignee: **Babcock Sempell AG,** Korschenbroich, Fed. Rep. of Germany

[21] Appl. No.: **427,120**

[22] PCT Filed: **Apr. 22, 1988**

[86] PCT No.: **PCT/EP88/00342**

§ 371 Date: **Oct. 23, 1989**

§ 102(e) Date: **Oct. 23, 1989**

[87] PCT Pub. No.: **WO88/08496**

PCT Pub. Date: **Nov. 3, 1988**

[30] **Foreign Application Priority Data**

Apr. 25, 1987 [DE] Fed. Rep. of Germany 3713974

[51] **Int. Cl.**[5] **F16K 21/10**
[52] **U.S. Cl.** **137/514.3;** 137/514
[58] **Field of Search** 137/514, 514.3, 514.5

[56] **References Cited**

U.S. PATENT DOCUMENTS

957,311	5/1910	Davis et al.	137/514.3 X
2,392,009	1/1946	Stern	137/514.3
3,048,188	8/1962	Hunter	137/514.3 X
3,789,873	2/1974	Westwood	137/514.3
3,850,405	11/1974	White	137/514.3 X
4,018,248	4/1977	Carr	137/514.3
4,756,334	7/1988	Panet et al.	137/514.3

FOREIGN PATENT DOCUMENTS

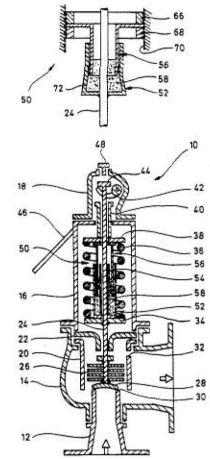
0032346	7/1981	European Pat. Off. .	
3010520	9/1981	Fed. Rep. of Germany .	
244923	12/1925	United Kingdom	137/514.3
258163	9/1926	United Kingdom	137/514.3

Primary Examiner—John Rivell
Attorney, Agent, or Firm—Lowe, Price, LeBlanc & Becker

[57] **ABSTRACT**

A spring-loaded safety valve provided with an integrated damping element that achieves a damping effect by the dissipative action of a shearing current and which involves a single-layer or multiple-layer virtually coaxial arrangement of tubes or active surfaces that can be moved telescopically with respect to one another.

10 Claims, 5 Drawing Sheets



1996

•Multi Fluid

US005515884A

United States Patent [19]

Danzy et al.

[11] **Patent Number:** **5,515,884**

[45] **Date of Patent:** **May 14, 1996**

[54] **MULTI-MEDIA SAFETY RELIEF VALVE**

[75] Inventors: **Roger D. Danzy; John E. Fain, Jr.,** both of Pineville, La.

[73] Assignee: **Dresser Industries Inc.,** Dallas, Tex.

[21] Appl. No.: **245,475**

[22] Filed: **May 18, 1994**

[51] **Int. Cl.**^{6} **F16K 17/08**

[52] **U.S. Cl.** **137/478**; 137/476

[58] **Field of Search** 137/469, 476, 137/478

[56] **References Cited**

U.S. PATENT DOCUMENTS

488,020	12/1892	Lohbiller .	
498,803	6/1893	Lunken .	
849,532	4/1907	Coutant .	
1,356,853	10/1920	Clark	137/478
1,690,097	11/1928	Ackermann .	
2,787,127	4/1957	Benz	62/1
2,875,978	3/1959	Kmiecik	251/333
2,917,072	12/1959	Saville	137/469
3,131,720	6/1964	Horvath	137/543
3,149,643	9/1964	Breitsprecher	137/469
3,520,326	7/1970	Bowen	137/478 X
3,572,372	3/1971	Moore	137/478 X
3,605,793	9/1971	Kinsel	137/469 X
4,036,250	7/1977	Dashner	137/476
4,130,130	12/1978	Stewart et al.	137/478 X
5,094,266	3/1992	Ledbetter	137/469

OTHER PUBLICATIONS

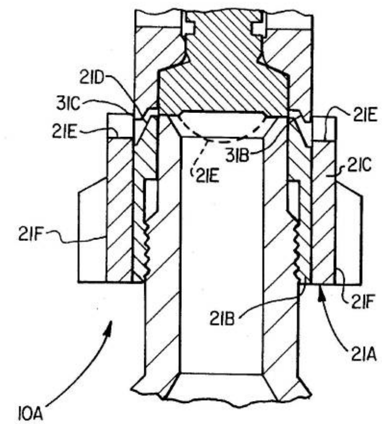
Dresser Industries Catalog published in 1991, entitled Safety Relief Valves, Sizing and Selection, Fall, 1991, SRV–1.

Primary Examiner—Stephen M. Hepperle
Attorney, Agent, or Firm—Jenkens & Gilchrist

[57] **ABSTRACT**

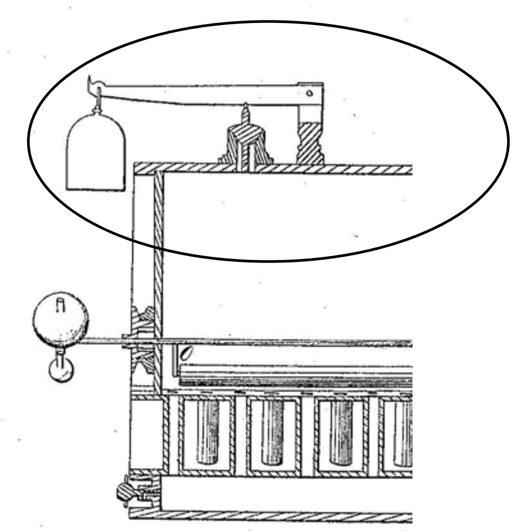
A safety valve for relieving excessive pressure in a fluid system including a valve body having a central chamber, a discharge flow nozzle connected into the chamber having a valve seat on the end of the flow nozzle in the chamber, a valve disc movably mounted in the chamber for movement between a closed position on the nozzle seat and an open position spaced from the nozzle seat, a valve disc holder connected with the valve disc having a deflecting rim circumscribing the valve disc seat provided with an internal outwardly divergent annular deflecting surface, a blowdown ring mounted on the nozzle having an inner ring portion provided with an outer frustoconical deflecting surface substantially parallel with and spaced from the valve disc holder deflecting surface, the blowdown ring including an outer cylindrical rim provided with circumferentially spaced semi-elliptical discharge ports and an internal cylindrical surface substantially parallel with and spaced from the outer cylindrical surface of the valve disc holder, and a spring connected with the valve disc holder for biasing the valve disc holder and valve disc closed. The blowdown ring is adjustable for changing the blowdown of the valve. The configuration of the valve disc holder and the blowdown ring deflecting surfaces provides chatter free operation.

4 Claims, 3 Drawing Sheets

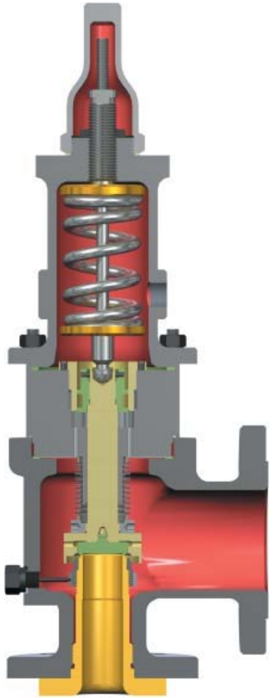


From There to Here - 170 Years in 40 Minutes

1836



2006



Back Pressure Effects on Spring Loaded PRV's

DIERS Users Group Meeting

Fall 2007 – Manchester, New Hampshire

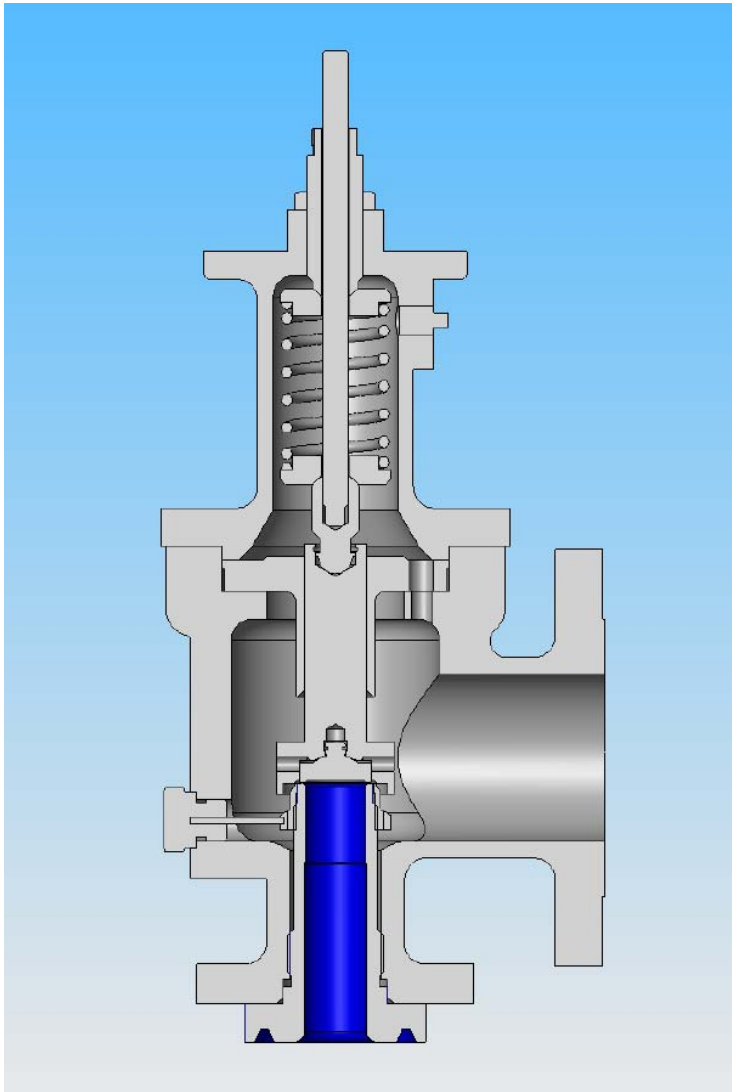
Outline

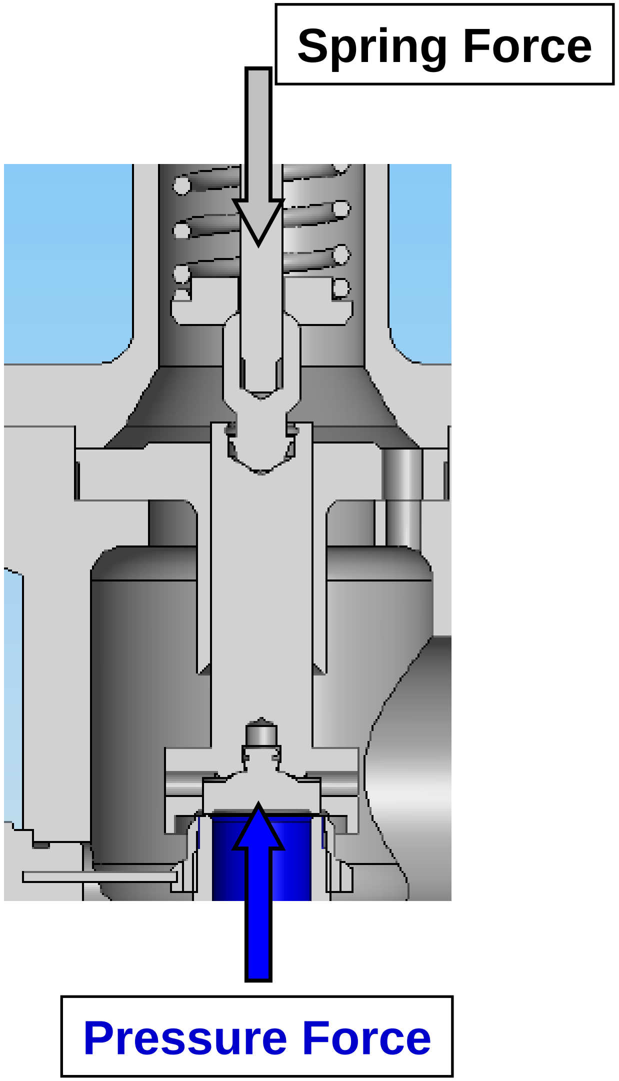
- Conventional
  - Set pressure change
  - Accumulation increase with backpressure
- Bellows
  - Bellows area tolerance
  - set pressure balance
  - flowing backpressure
  - back pressure > set pressure

Information

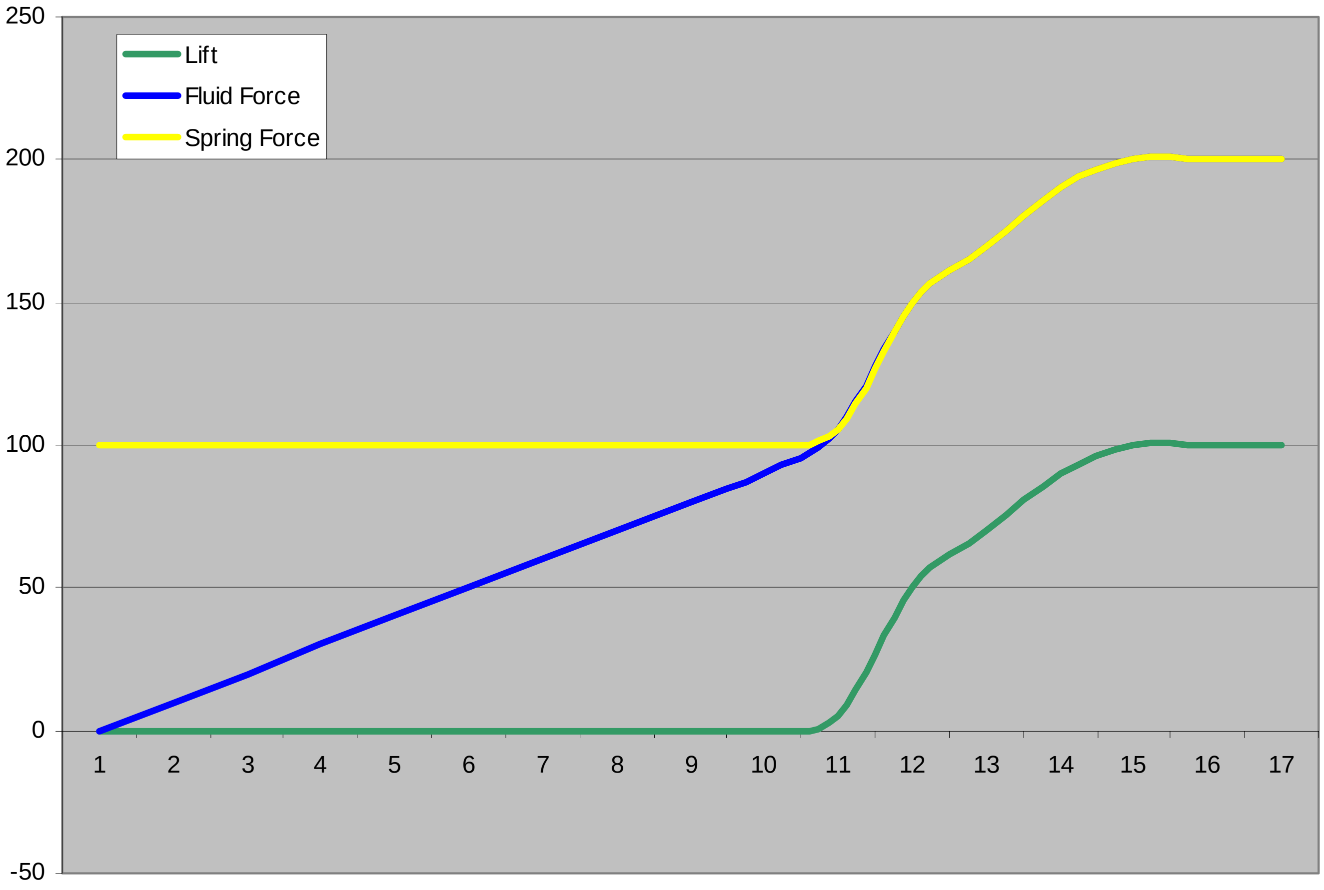
- Performance Graphs – Gas
- Graphs are representative of performance of Consolidated® PRV's
- Images are Consolidated® 1900 Series
- Color Coding
  - Blue – Inlet Pressure/Fluid Force
  - Red – Back Pressure
  - Green – PRV Lift
  - Yellow – Huddle Chamber

Conventional Spring Loaded PRV





Spring Loaded Valve Forces - Lift



Fluid Force, FF;

PRV Disc Lift, L;

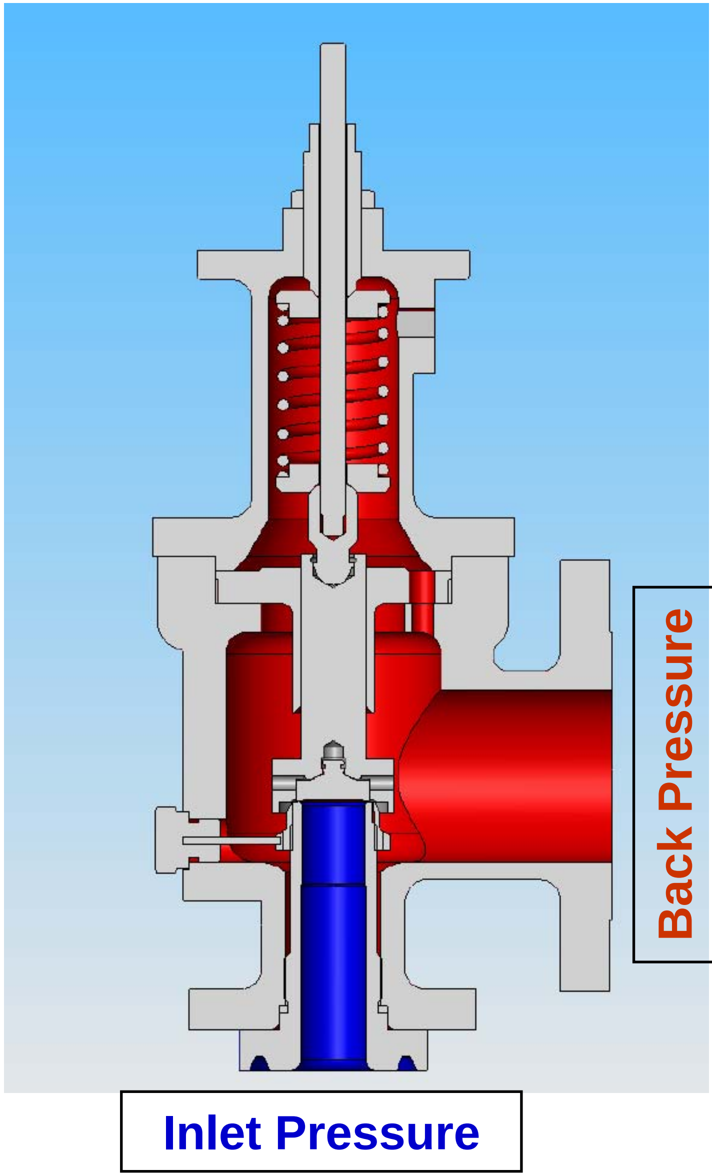
Spring Force, SF;

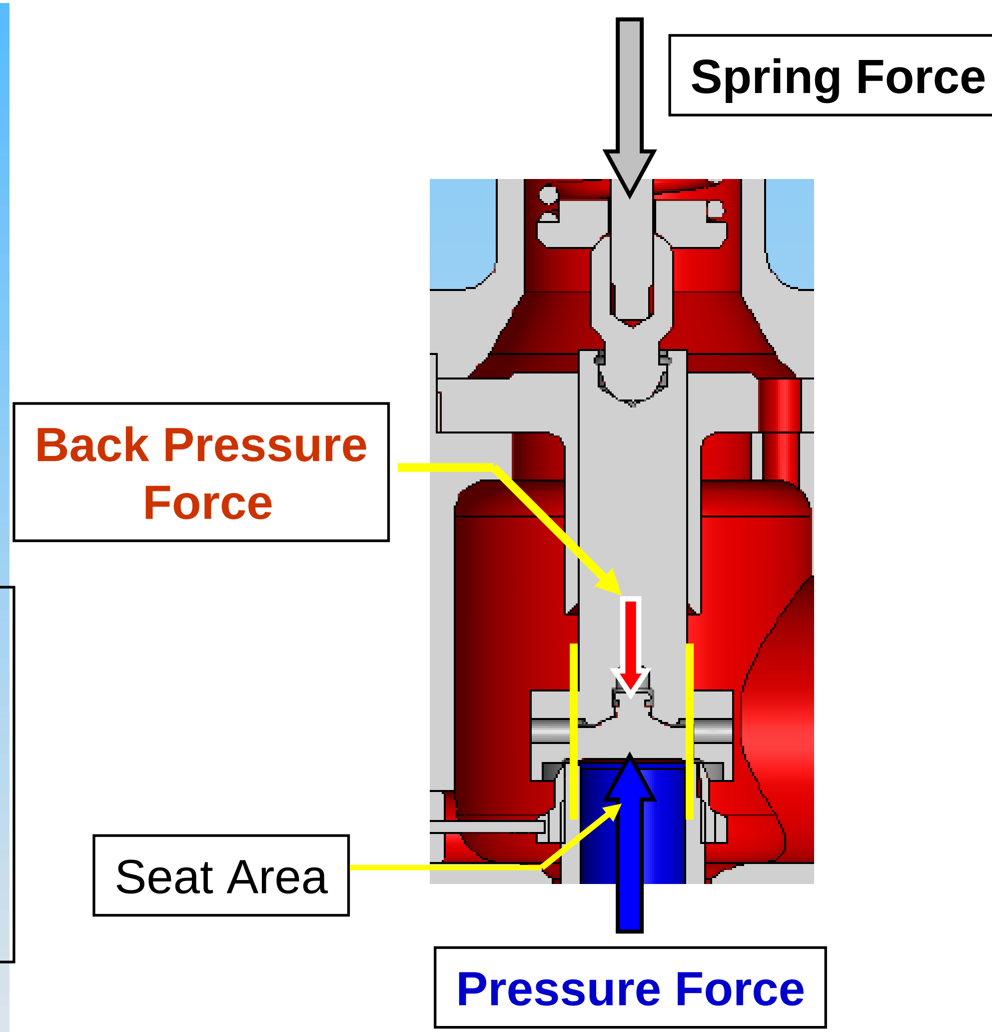
Spring Constant, k

Spring Preload, SF_i

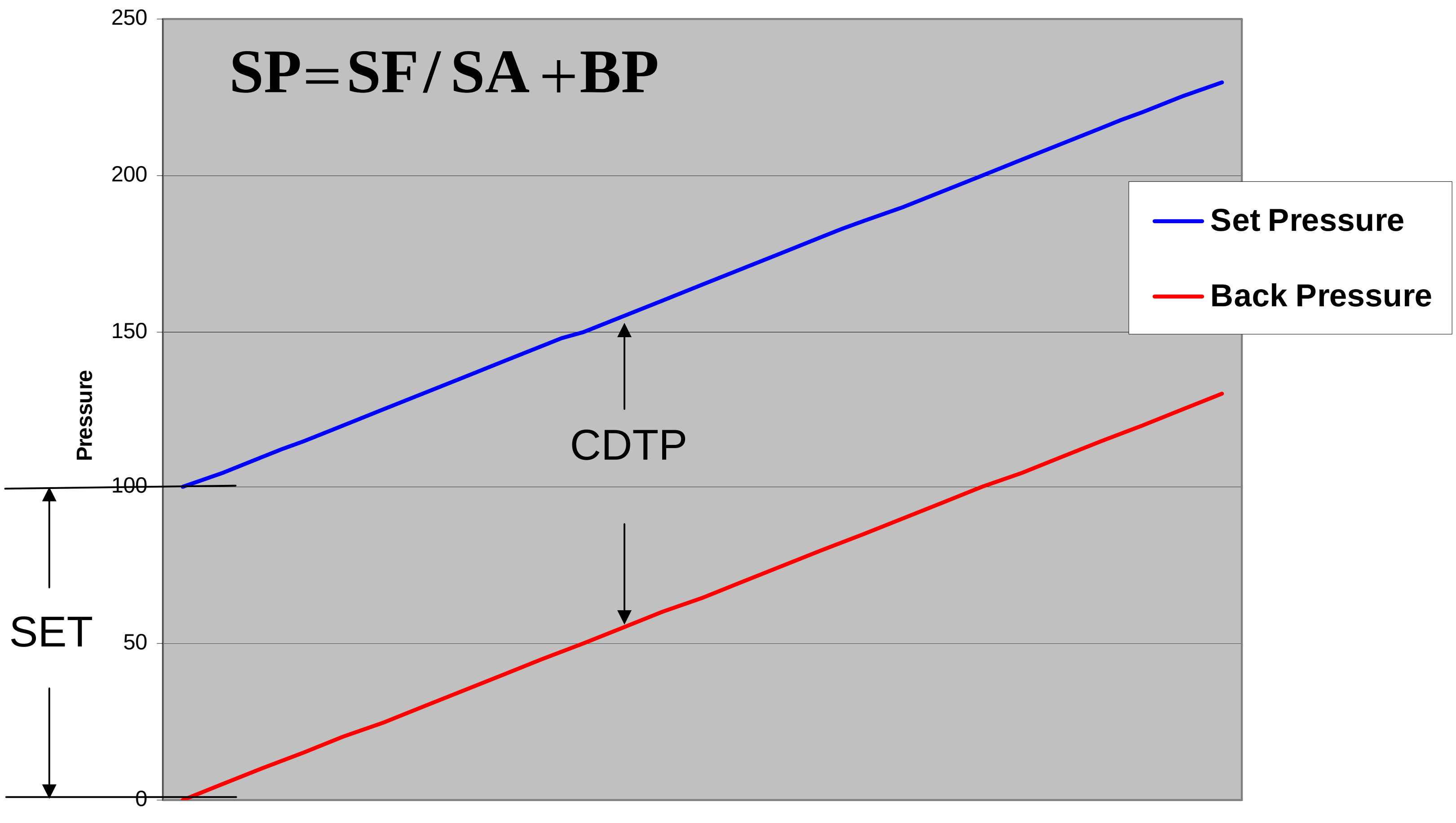
$$L = \frac{(FF - SF_i)}{k}$$

Set Pressure Effect

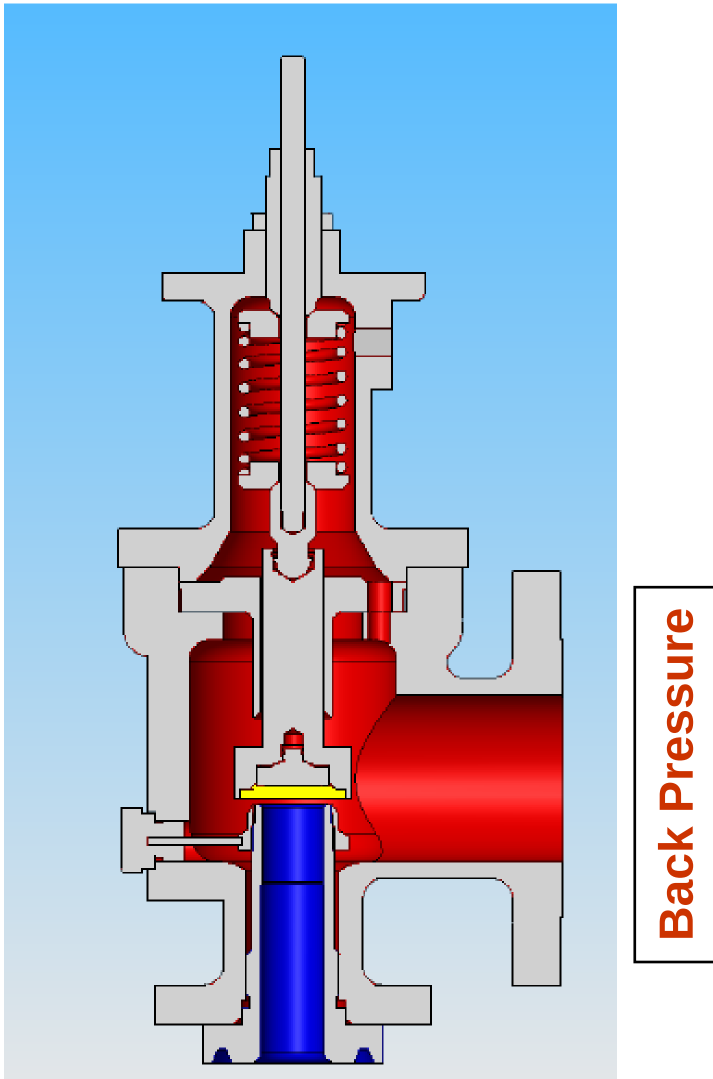




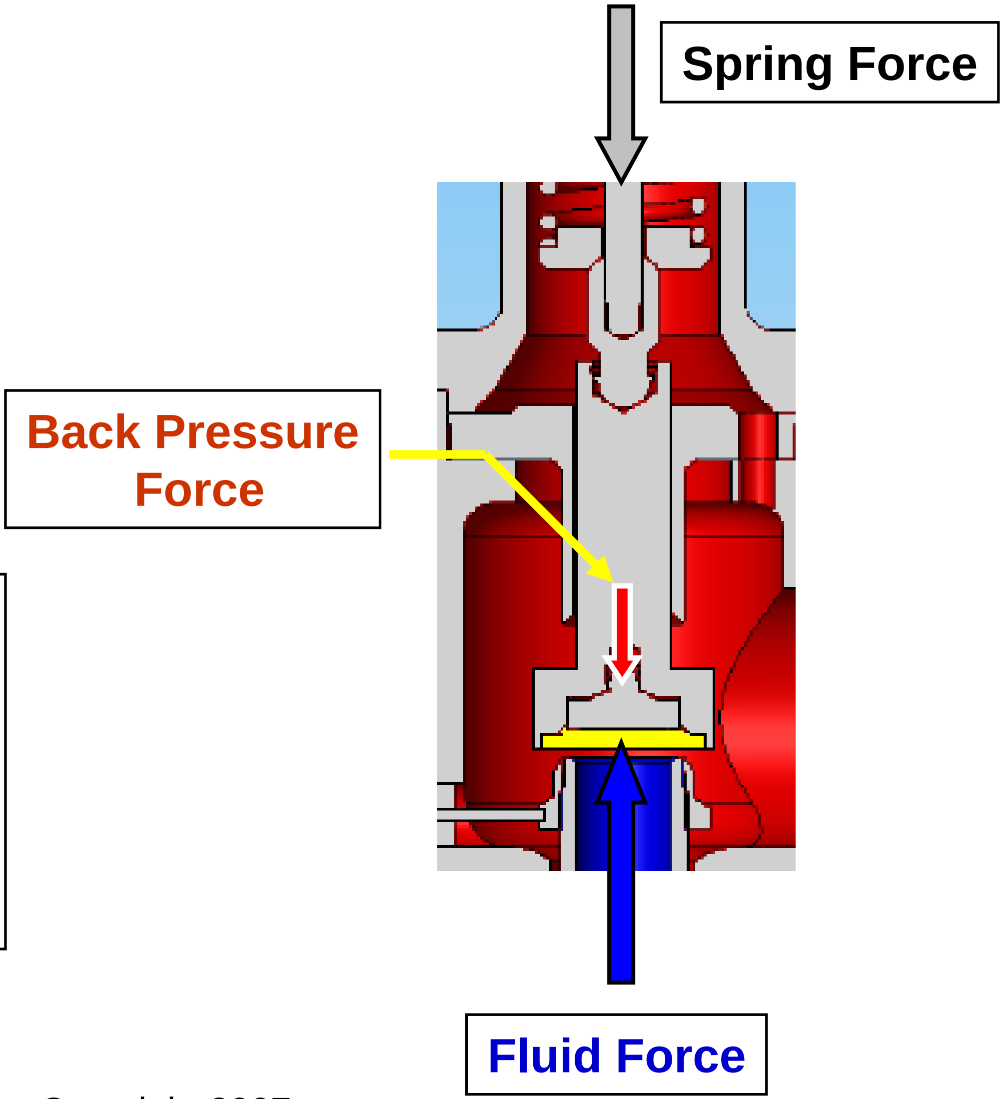
Back Pressure Effect on Set Pressure



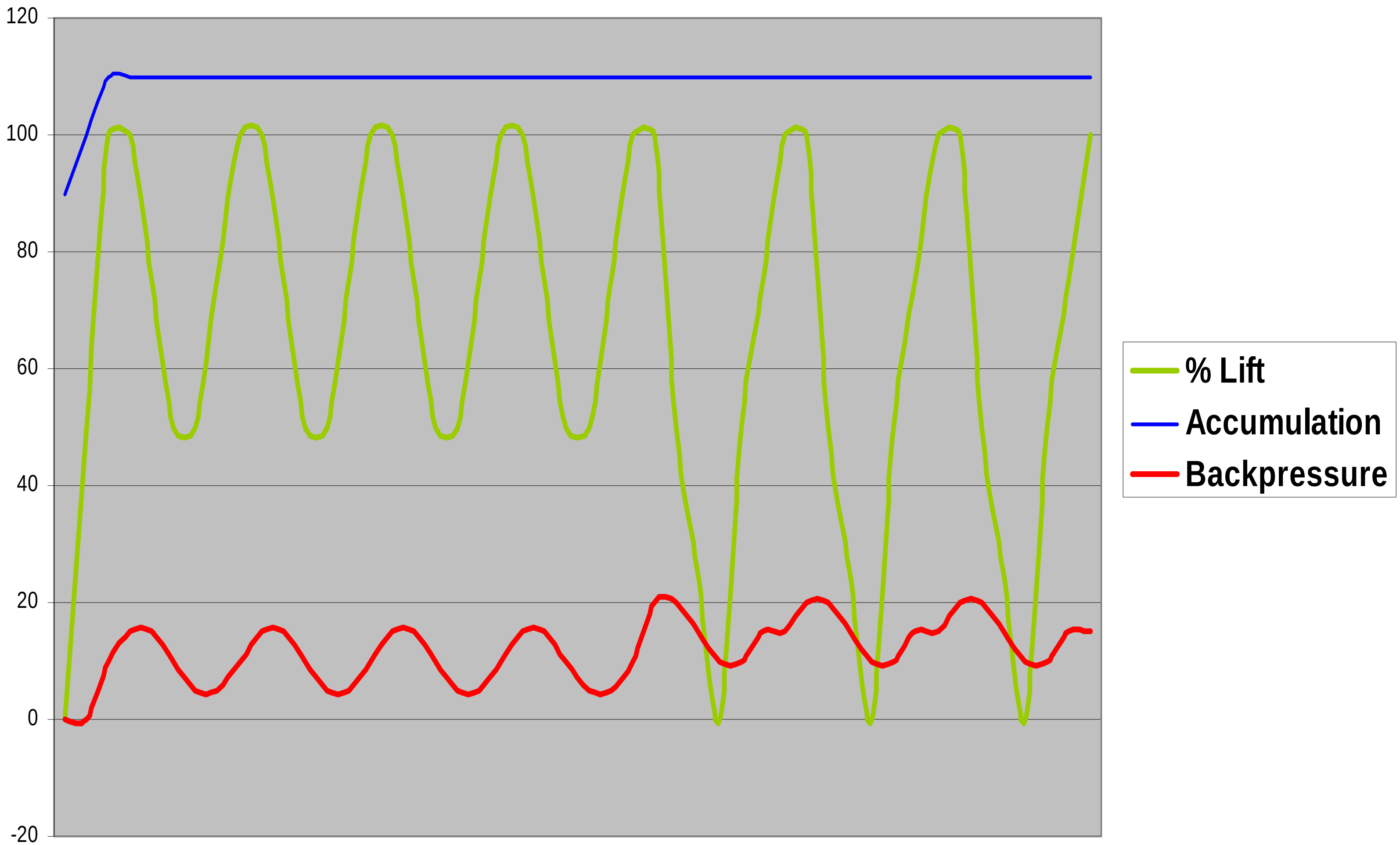
Flowing Condition Effects



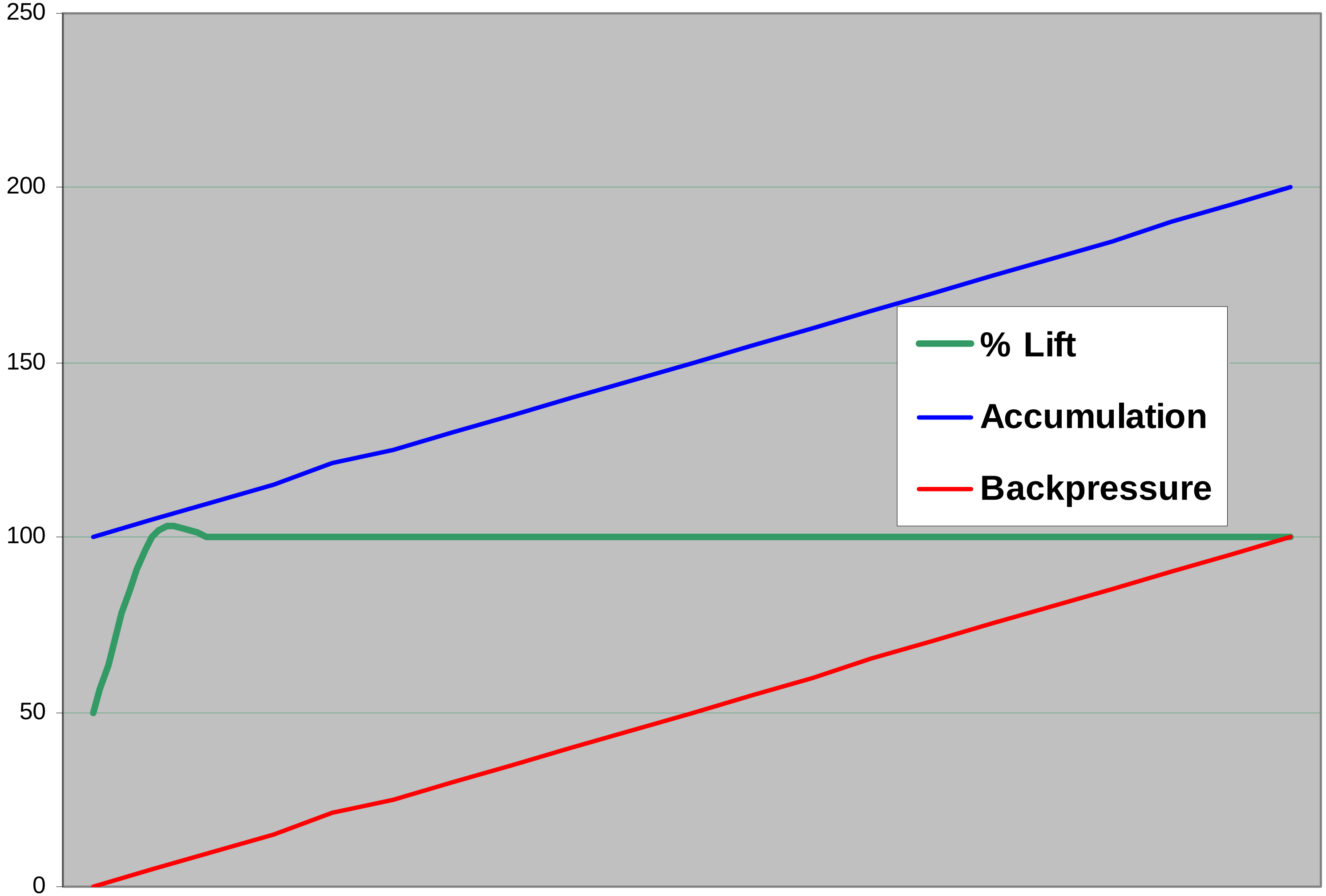
Inlet Pressure



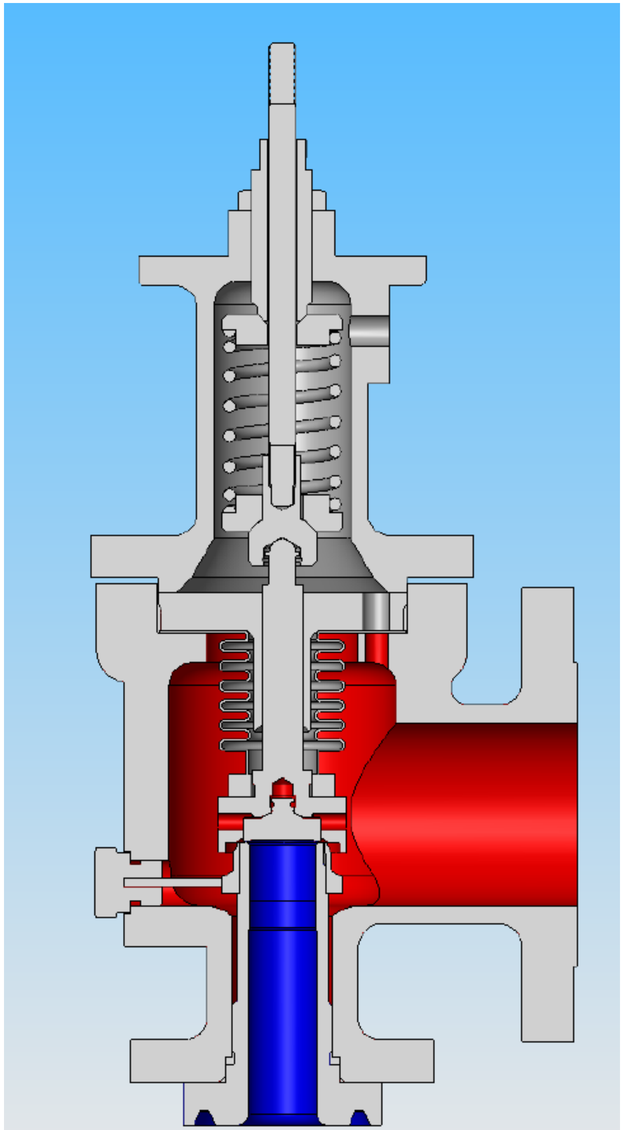
Backpressure Effect at Constant Accumulation

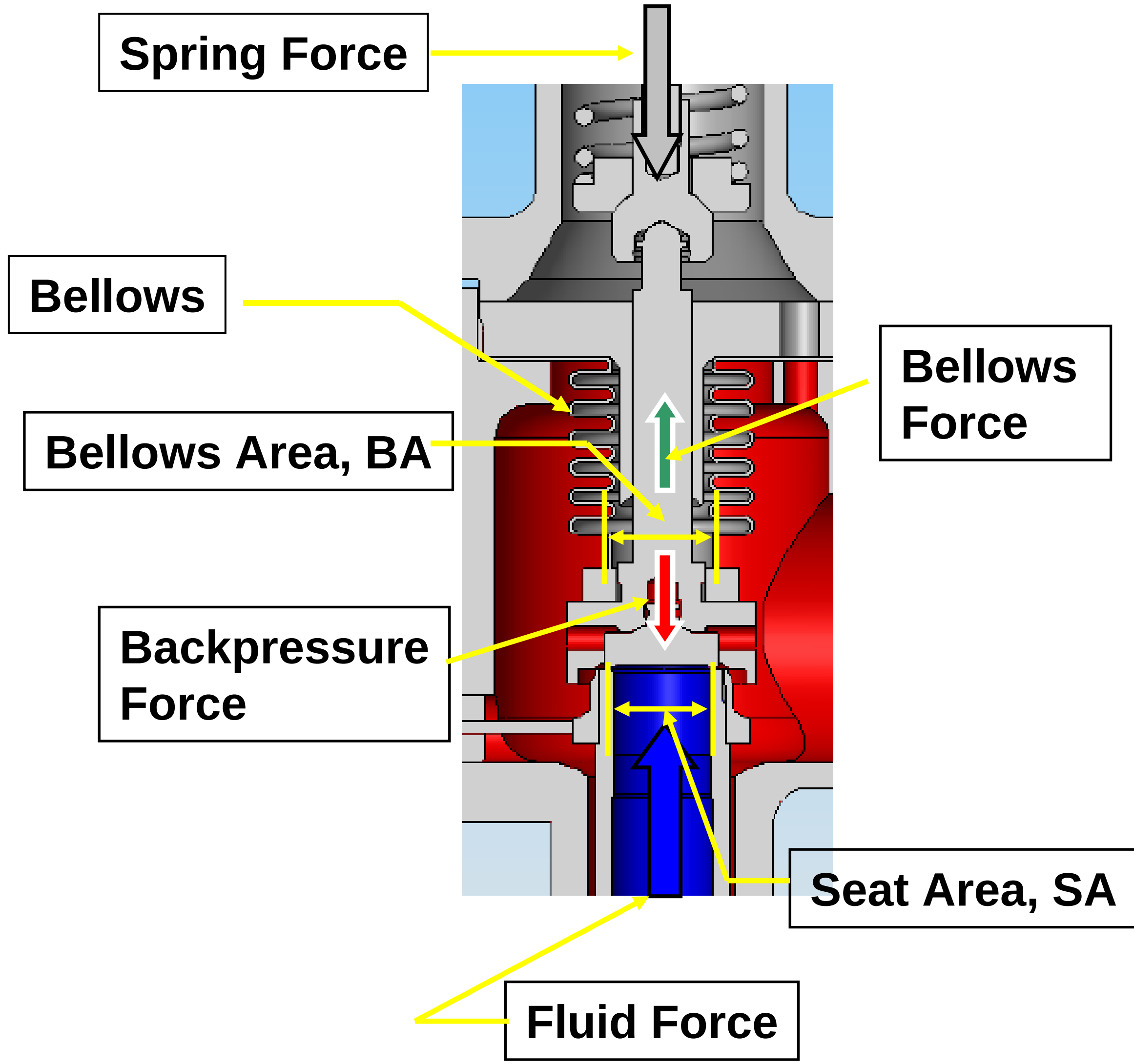


Backpressure Effect at Constant Flow Rate



Bellows Spring Loaded PRV

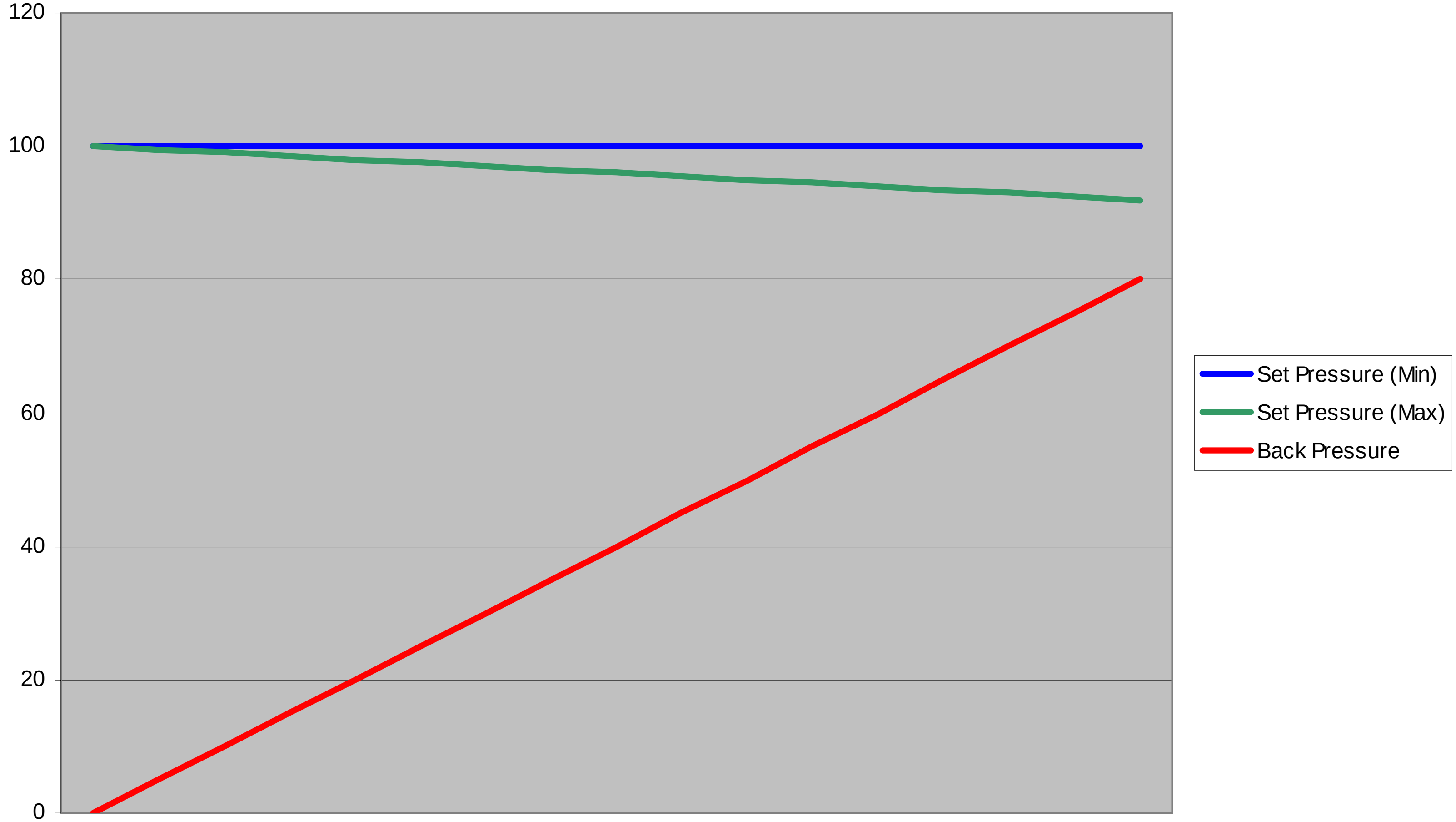




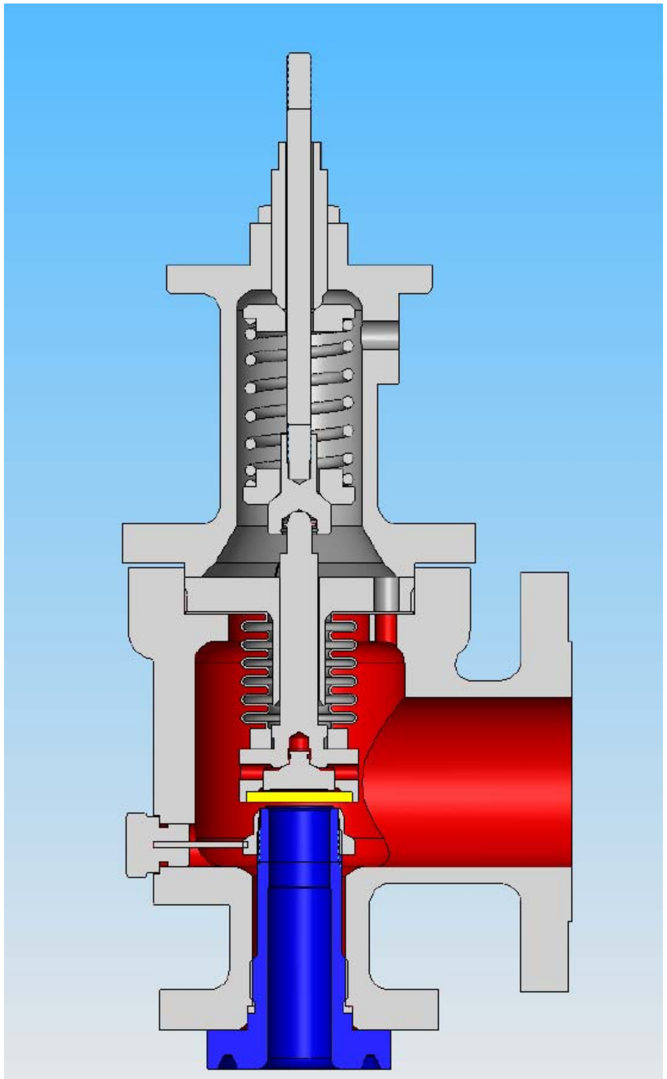
“Balanced Bellows”

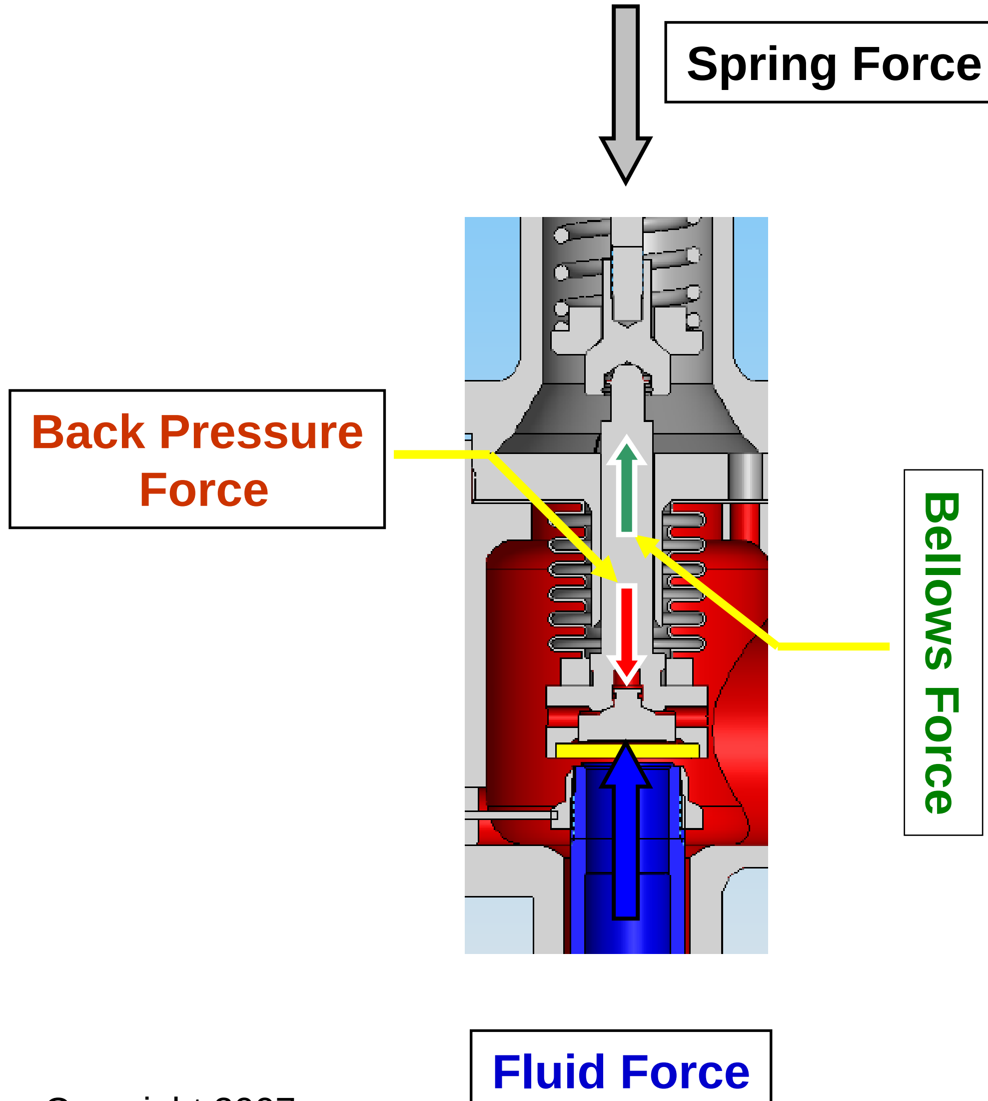
- Bellows are formed by rolling
- Bellows Effective Area +- 5% or greater
- Ensure Set Pressure <= Stamped Value
- Bellows Effective Area > Seat Area

Backpressure Effect on Set Pressure of Balanced Bellows PRV

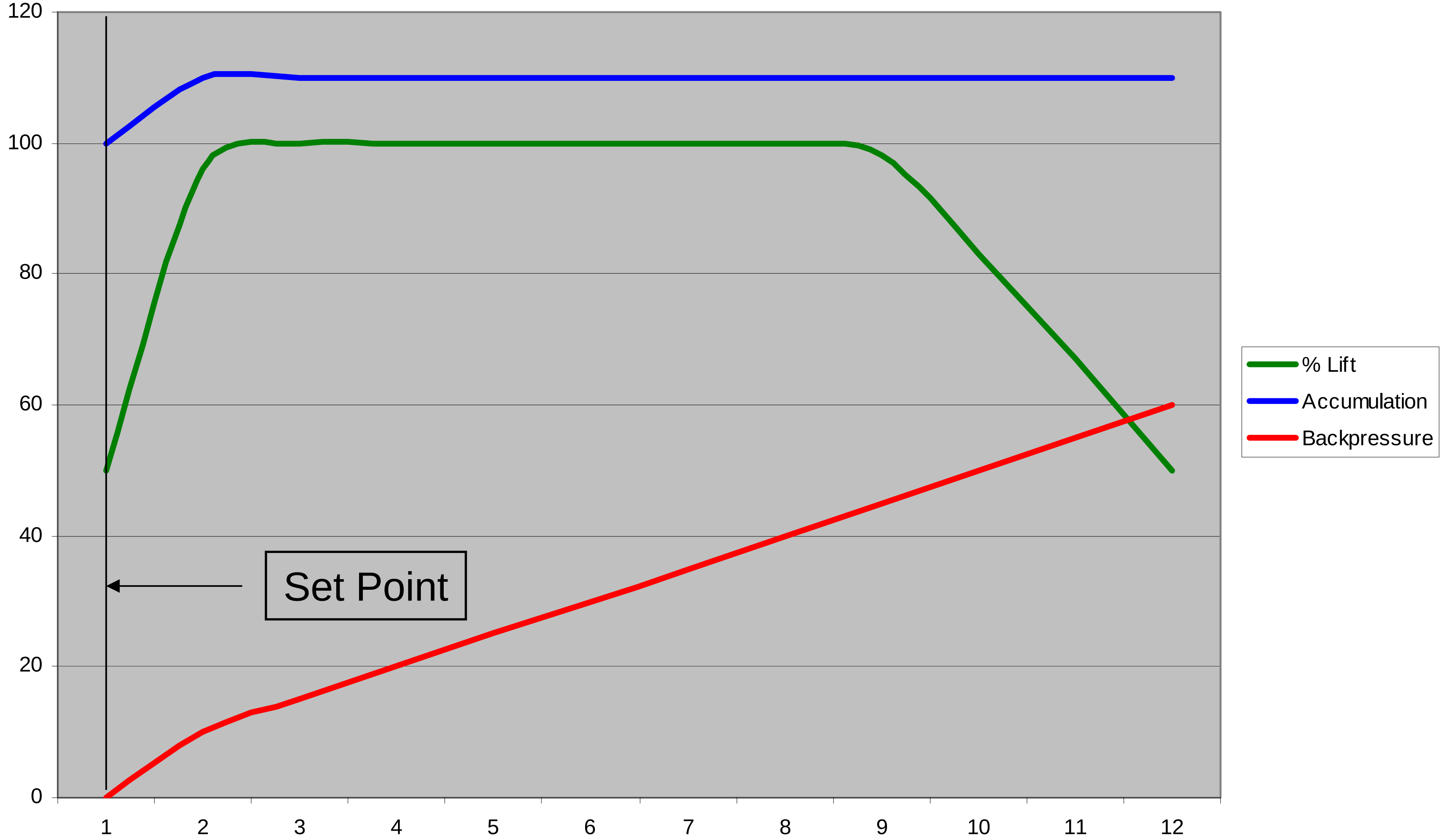


Balanced Bellows, Flowing,Backpressure

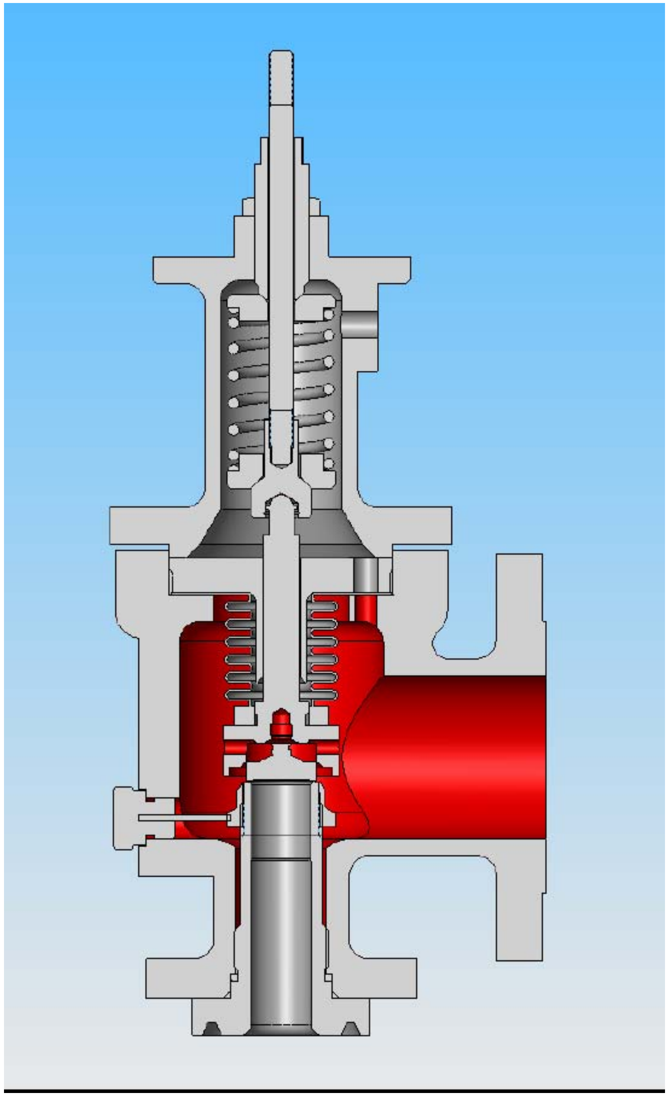


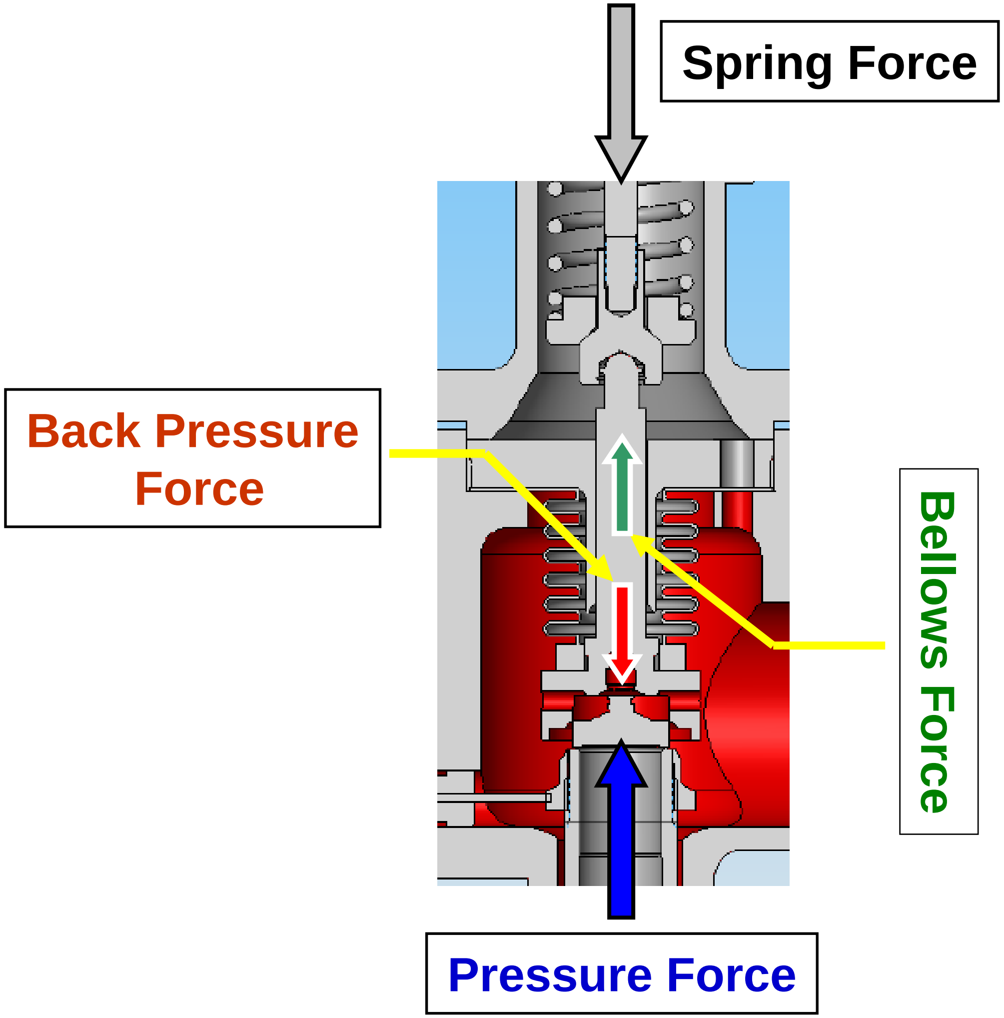


Backpressure Effect on Lift

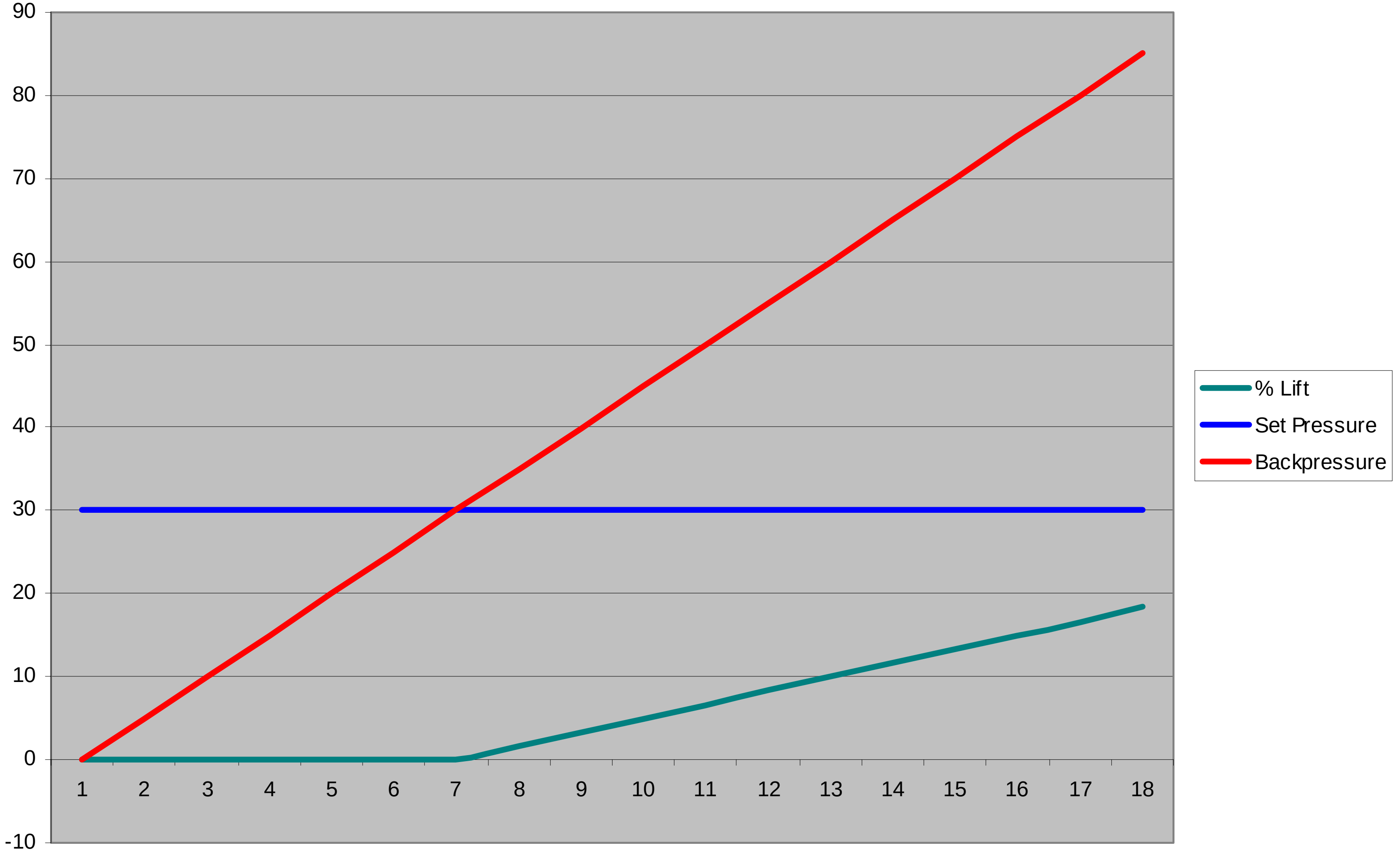


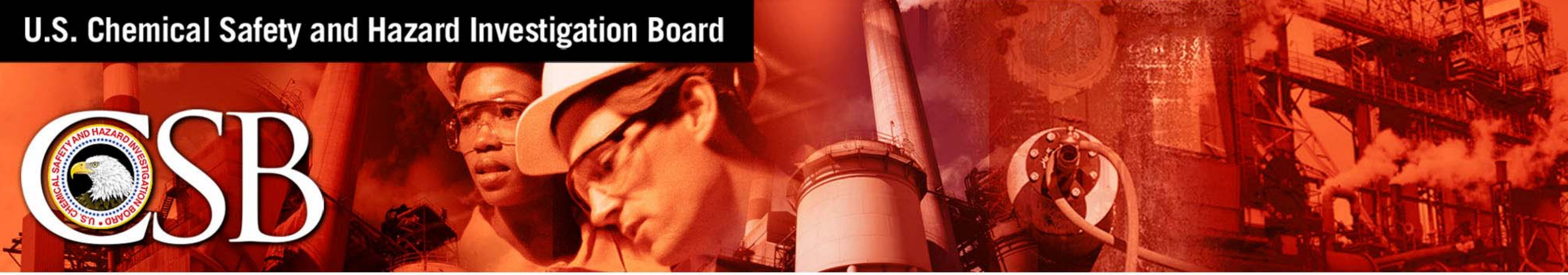
Disc Separation - Backpressure





Backpressure - Disengagement





BP-Texas City Relief System Modeling

DIERS Users Group - Philadelphia

Jim Lay – CSB

Harold Fisher – Fauske

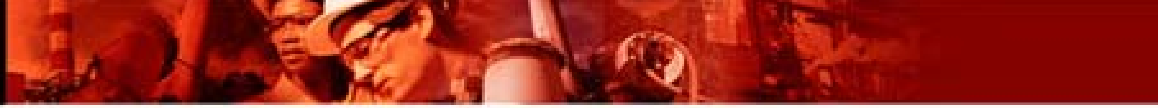
May 7, 2007

Presentation Contents

1. Introductions
2. Animation
3. Incident background & consequences
4. Key system components
5. Modeling approach
6. Findings
7. Q&A

What is the CSB?

- The CSB is an independent U.S. federal agency charged with investigating chemical accidents
- Authorized by United States Congress in 1990
- Modeled after the National Transportation Safety Board
- Professional investigation staff
- Located in Washington, DC



What Does the CSB Do?

- We investigate chemical process incidents
- We determine root causes
- We make recommendations

Our goal is prevention.

Animation

Incident Background

- BP – Texas City refinery Isom unit
- March 23, 2005
- Raffinate splitter column overfilled on initial liquid fill (packing)
  - Mass balance (stuff went in, little came out, the difference accumulated)
  - Liquid expansion due to heating
  - Vapor volume due to heating
  - Multiple instrument failures, many other factors
- Sub-cooled “cap” of liquid overflowed into the vapor product line

Incident Background

- Column pressure + hydrostatic head opened relief valves
  - 63 psig versus SP of 40 psig
- Blowdown drum overfilled
- “Geyser-like” discharge from stack
- Subsequent vapor cloud formation / ignition / VCE

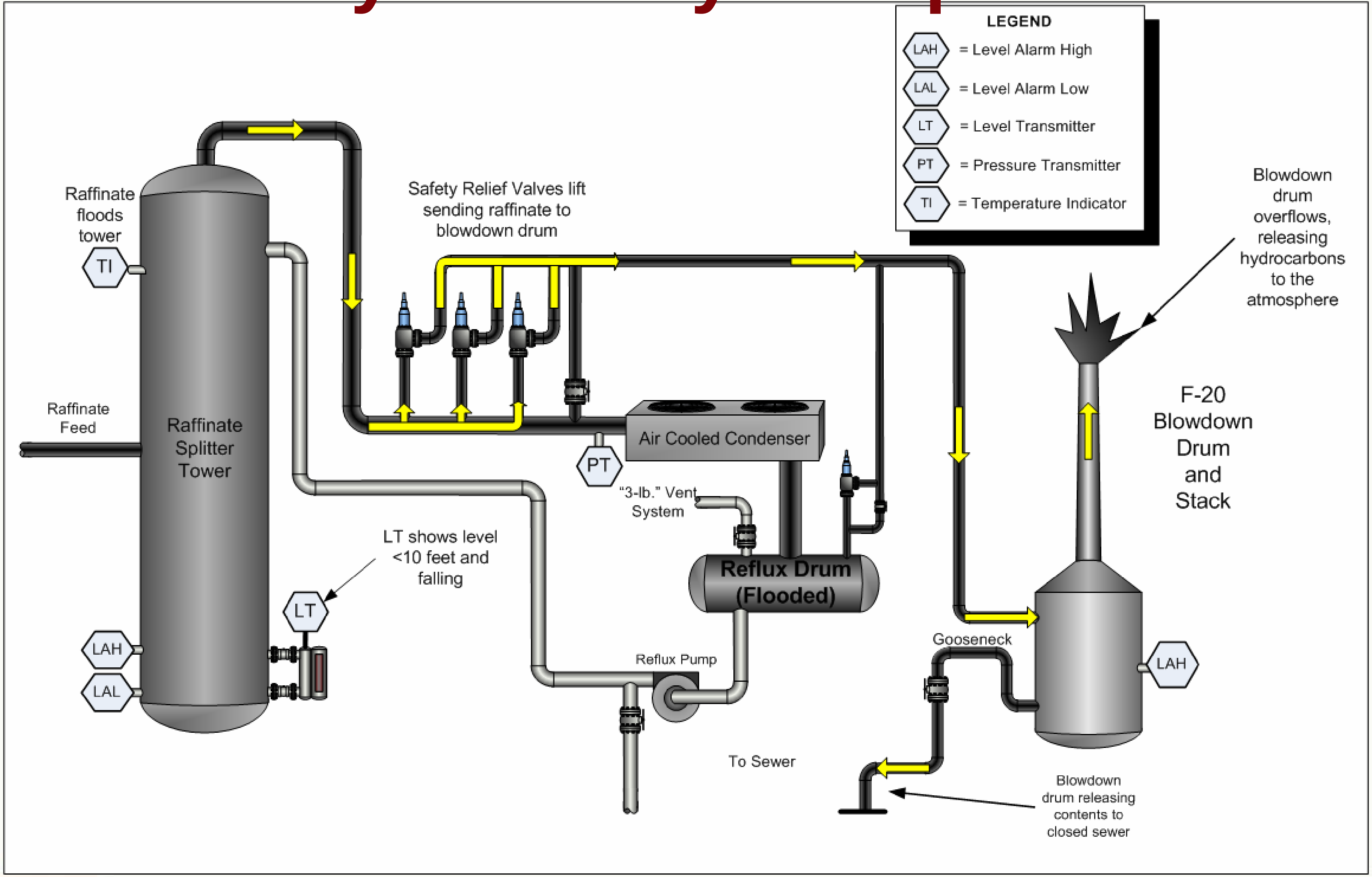
Incident Consequences

- 15 fatalities
- ~180 injured
- Serious consequences for BP
  - Record OSHA fine
  - Extensive litigation
  - Damage to corporate image
  - Assorted other impacts

Incident Consequences

- Major CSB investigation to determine root and contributing causes
  - Trailer urgent recommendation
  - Baker Commission – BP North American refinery safety culture
  - CSB recommendations to BP, API, OSHA, others

Relief System Key Components



Inlet Header

- 16” line descending 142’ from top of column to relief valve inlets
- Temperature at top of column
- Column pressure at relief valves
- P&T consistent with column pressure + hydrostatic head of sub-cooled liquid
- Pressure sustained above relief set-points for 6 minutes

Relief Valves

- Three Consolidated 1910-30 valves
  - One 4P6
  - Two 8T10’s
  - Set at 40, 41, & 42 psig
  - Bellows equipped
- Sized for vapor only
- Valves not changed when column derated from 70 to 40 psig
- Credit taken for partial duty on fin-fans and reflux following derate

Blowdown Header

- 14” diameter
- 885 feet long
- 12 - 90º, 4 - 45º elbows
- 1 full ported gate valve
- Flow area of 137 in^2 is less than combined discharge area of relief valves (188 in^2)
- Major backpressure issues expected (by inspection)!

Blowdown Drum & Stack

- 10 ft. diameter x 27 ft. tall drum
- 10 ft. high conical transition
- 76.5 ft high x 32.5 in. diameter vapor discharge stack
- 6 in. “gooseneck” liquid drain to process sewer
- Equipped with sprays (OOS) to condense heavy hydrocarbons

Blowdown Drum & Stack

- Old design, replaced nearly-in-kind in 1997
- ~50 other reliefs tied in
- Stack velocity / back pressure issues
- History of HC vapors reaching ground level

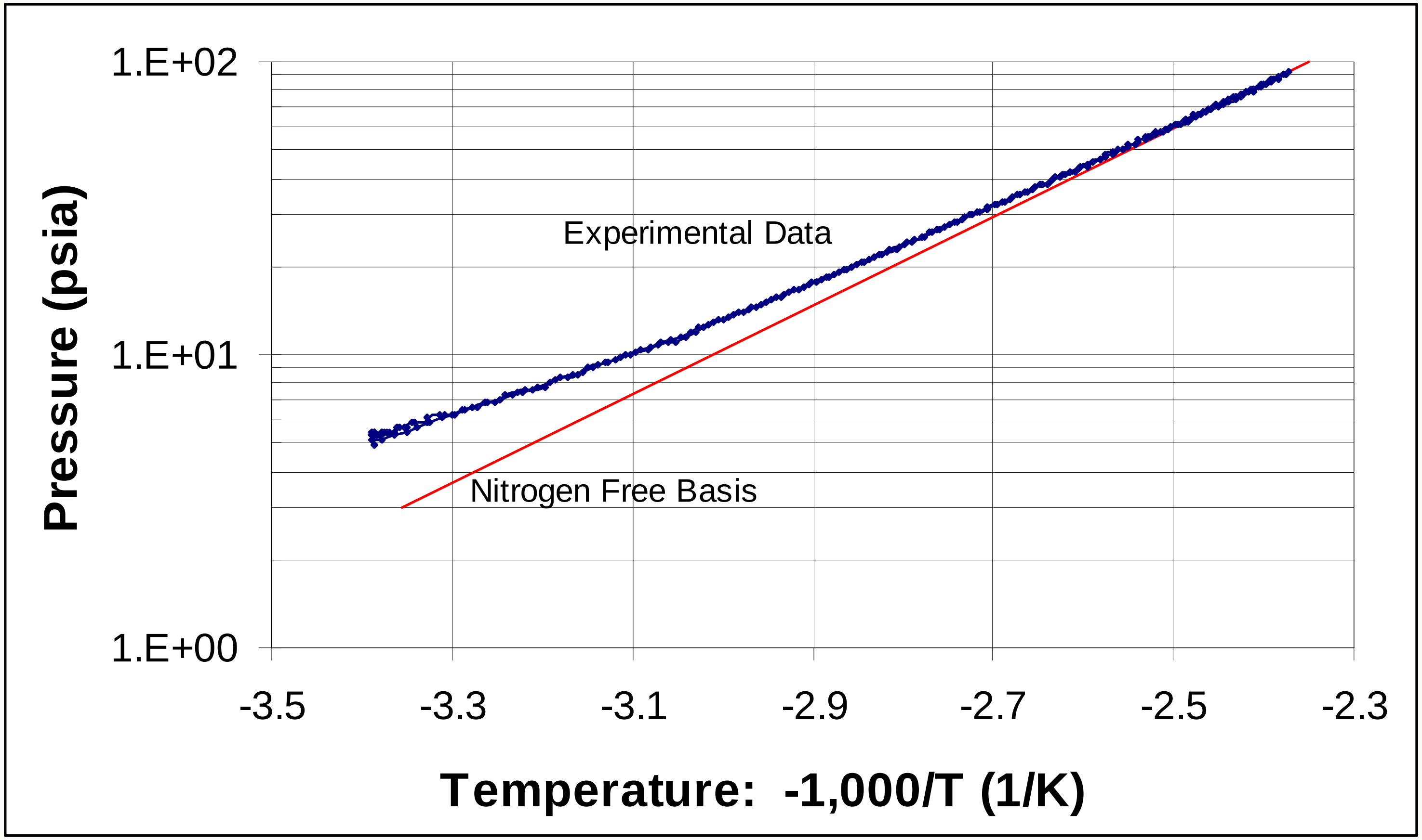
Modeling Approach

- Numerical integration of equations
  - Variable time-step
- Simplified mixture composition with 8 components
  - Good fit to Fauske experimental VP data
- R-K EOS for vapor phase enthalpies
- DIPPR liquid properties
- Raoult's law (activity coefficients = 1) gave excellent fit

Modeling Approach - Composition

Compound	Weight Fraction
n-pentane	0.0383
2-methyl butane	0.0263
n-hexane	0.1519
2-methyl pentane	0.2950
n-heptane	0.3072
n-octane	0.1300
n-nonane	0.0409
Heavies as n-decane	0.0104
Total	1.0000

Modeling Approach – VP Data



Fauske Experimental Vapor Pressure Data

Modeling Approach - Parameters

- H. Fisher Relief Valve Model (with DIERS™ !)
- Older style vapor trim valves with liquid capacity rated at 1.25 SP
- For vapor:
  - K_d = 0.95 (Consolidated)
  - K_b = API RP 520 consensus values
  - ASME Code reduction factor = 1.0
  - Capacities evaluated at 1.16 MAWP

Modeling Approach - Parameters

- **For liquid:**
  - **K_d = 0.74 (Consolidated w/o ASME 0.9 derate)**
  - **K_p from 0.6 → 1.0 as pressure increases from 1.1 → 1.25 SP**
  - **K_w taken as 1.0 at incident conditions**
    - manufacturer's curves for liquid trim rated at 1.25 SP
    - experimental data not available to support manufacturer's curves
    - curves for larger valves, higher BPs extrapolated
    - For our study, K_w set to 1.0 given available 58% OP

Findings – Vapor Design Case

- Valve SP's too close
- Credit for fin-fan cooling not appropriate
- No liquid relief evaluated, even for flooded reflux drum relief valve
- Valves too small to protect column @ 40 psig SP
  - 416,500 lb/hr required vapor flow
  - 213,300 lb/hr available flow capacity
- Sizing adequate at original 70 psig SP

Findings – Relief Vapor Design Case

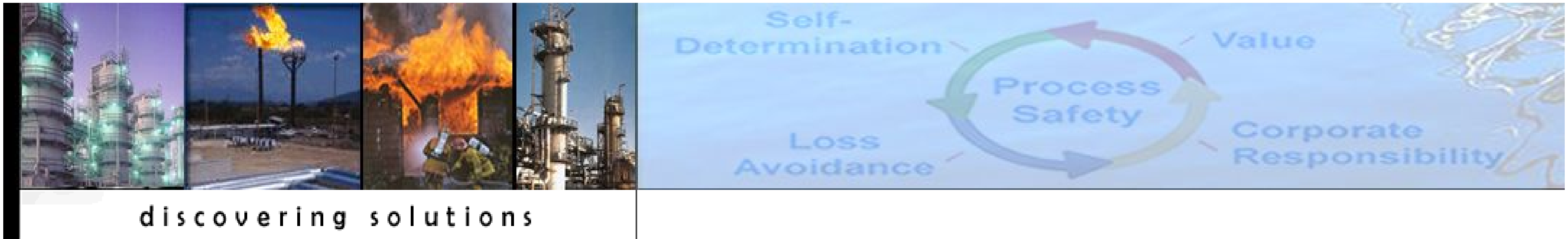
- At 40 psig SP (with inadequate flow!) blowdown header is undersized, with vapor case BP of 60.8%
  - Backpressure at required flow prohibitive
- At original 70 psig SP the blowdown header is undersized with 61.8% BP

Findings – March 23 Incident:

- Sub-cooled liquid relief until final seconds
- Relief valves open for 6 minutes
- Before Explosion:
  - Equipment Fill of 32,130 gallons (4.2 min.)
  - Liquid to sewer of 11,470 gallons
  - Discharge out stack of 6,735 gallons (1.6 min.)
- Flow out stack continued for ~0.2 min. after initial blast
- Maximum BP on valves 90% of SP

Thank You!

Q&A



Understand Flare Noise

By
G. A. Melhem, Ph.D.

Overview

- Establish how API noise estimation method was derived
- Provide a simple help tutorial about noise in SuperChems

ioMosaic

What is noise?

- Sound is the result of a source setting a medium into vibration, usually air
- The vibration produces alternating compression and rarefaction waves
- The resulting variation in normal ambient pressure is translated by the ear and perceived as sound
- Sound waves can be reflected, scattered, and/or refracted
- Sound waves travel at the speed of sound in the vibrating medium
  - 344 m/s in air
  - 1433 m/s in water
  - 3962 m/s in wood
  - 5029 m/s in steel
- Noise is unwanted sound

All units are in SI unless otherwise specified

ioMosaic

What is sound?

- Sound can be described in terms of amplitude (loudness), frequency (pitch), and duration
- Amplitude is a measure of the difference between atmospheric pressure (with no sound present) and the total pressure (with sound present)
- The amplitude of a sound wave equates to the sound pressure
- Sound pressure is used as a fundamental measure of sound amplitude
- Sound pressure is measured using root mean square (rms) sound pressure level, L_p, in decibels (dB); absolute pressure is not measured

$$L_p = 10\log\left(\frac{P_1}{P_r}\right)^2 = 20\log\left(\frac{P_1}{P_r}\right) \text{ in dB}$$

1 Pa = 1 N/m^2 = 1 J/m^3 = 1 $kg/m/s^2$

$$P_r = 0.00002 \ \text{Pascal}$$

ioMosaic

What is sound?

- For each increase of 20 dB, there is a tenfold increase in sound pressure

- Sound pressure expressed in decibels (dB) is not additive linearly

$$L_{p,total} = 10\log\sum_{i=1}^{n} 10^{\left[\frac{L_{p,i}}{10}\right]}$$

- Adding two values from different sources producing the same sound pressure level (L_p) results in a total sound pressure level of L_p+3 dB. If L_p equals 80 dB, than the total will be 83 dB.

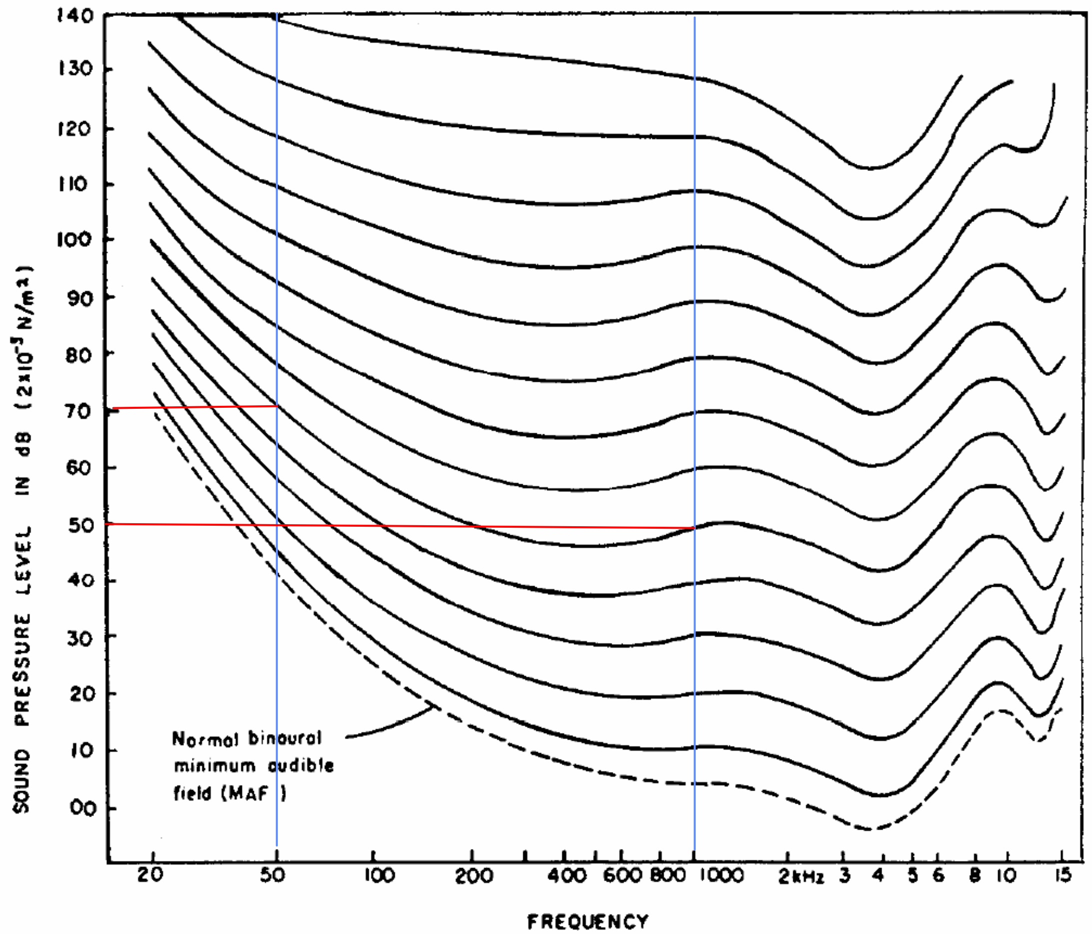
$$L_{p,total} = 10\log\left[10^{\left[\frac{80}{10}\right]} + 10^{\left[\frac{80}{10}\right]}\right] = 10\times 8.30103 = 83.01 \text{ dB}$$

- If there is a 10 dB difference in the sound pressure level of two sources, the cumulative sound pressure level will be approximately equal to the higher value of the two

$$L_{p,total} = 10\log\left[10^{\left[\frac{90}{10}\right]} + 10^{\left[\frac{80}{10}\right]}\right] = 10\times 9.0413 = 90.41 \text{ dB}$$

What is sound?

- The ear is less sensitive to low frequencies than high frequencies
- A 50 Hz tone at 70 dB sounds as loud as 1000 Hz tone at 50 dB



ioMosaic

What is sound?

- Another relationship used to represent sound or noise is <u>sound power level</u>, L_w

$$L_w = 10\log\left(\frac{W_1}{W_r}\right) \text{ in dB}$$

$$W_r = 10^{-12} \quad \text{Watts, W}$$

- A relationship between sound power (W) and sound intensity (I)can also be established

$$W = I\,A \quad \text{and} \quad L_I = 10\log\left(\frac{I}{I_r}\right) \text{ and } I = \frac{P^2}{\rho c}$$

$$I_r = 10^{-12}\,\text{W/m}^2,\ c = 344\,\text{m/s, and } \rho = 1.2\,\text{kg/m}^3$$

$$A = 4\pi r^2 (\text{for spherical radiation}) \ \ \textit{or}$$

$$A = 2\pi r^2 (\text{for hemispherical radiation})$$

ioMosaic

What is sound?

- Sound pressure level L_p can be related to sound power level L_w

$$W = I\,A \;\; = \frac{P^2}{\rho c} A$$

$$L_w = 10\log\left(\frac{W_1}{W_r}\right) = 10\log\left(\frac{P_1^2}{\rho c 10^{-12}} A\right) = 10\log\left(\frac{P_1^2}{1.2\times 344\times 10^{-12}}\right) + 10\log A$$

$$L_w = 10\log\left(\frac{P_1^2}{0.00002^2}\right) + 10\log A$$

$$L_w = L_p + 10\log A$$

- For spherical radiation of sound and at r=30 meters

$$L_w = L_{p,30} + 10\log\left(4\pi r^2\right) = L_{p,30} + 10\times 4.053 \approx L_{p,30} + 41$$

$$L_{p,30} = L_w - 41$$

What is sound?

- Sound power level can be estimated for flow from a flare tip using the following relation:

$$L_w = 10\log\left[\eta\frac{\frac{1}{2}mc^2}{10^{-12}}\right] = 10\log\left[\frac{\eta mc^2}{2}\right] + 120$$

½ mc^2 is the kinetic power of the flow through the flare tip in Watts
m is the flow rate in kg/s
c is the speed of sound in the gas medium in m/s
η is the acoustic efficiency associated with transforming a portion of the kinetic power to sound power with a value ranging from 10^{-5} to 10^{-2}

$$L_p = L_w - 10\log A = 10\log\left[\frac{\eta mc^2}{2}\right] + 120 - 10\log\left(4\pi r^2\right)$$

ioMosaic

What is sound?

- The sound pressure level derived earlier can be evaluated at 30 meters to yield an expression very similar to the API-521 noise equation for choked flow

$$L_{p,30} = L_w - 41 = 10\log\left[\frac{\eta m c^2}{2}\right] + 120 - 41 = 10\log\left[\frac{\eta m c^2}{2}\right] + 79$$

$$L_{p,30} = 10\log\left[\frac{m c^2}{2}\right] + \left\{10\log(\eta) + 79\right\}$$

$$L_{p,30} - 10\log\left[\frac{m c^2}{2}\right] = \left\{10\log(\eta) + 79\right\}$$

- L_p at any distance is then evaluated

$$L_p = L_{p,30} - 20\log\left(\frac{r}{30}\right)$$

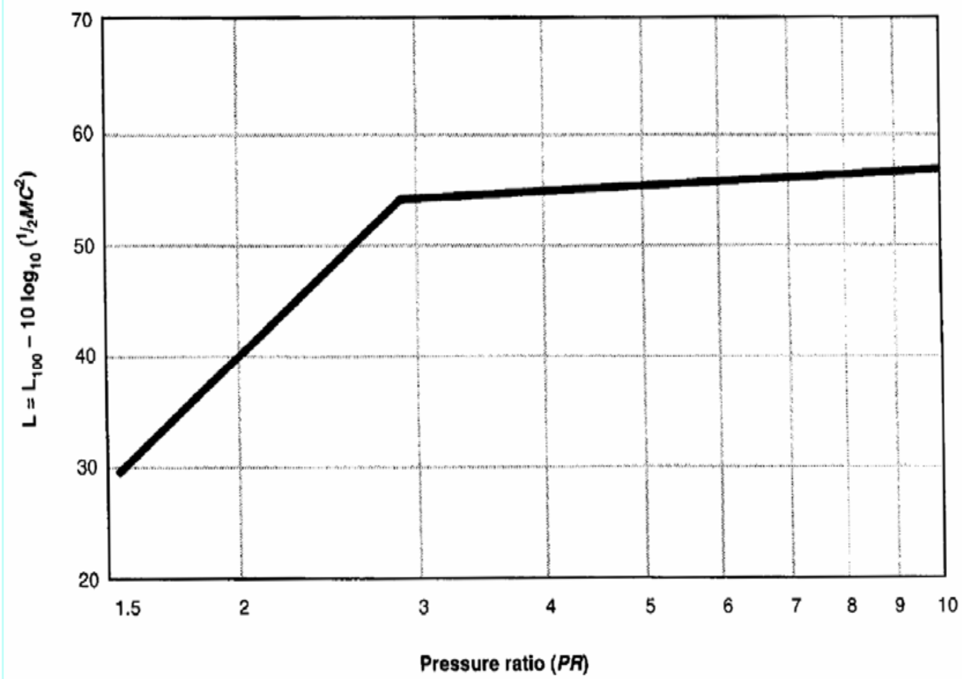
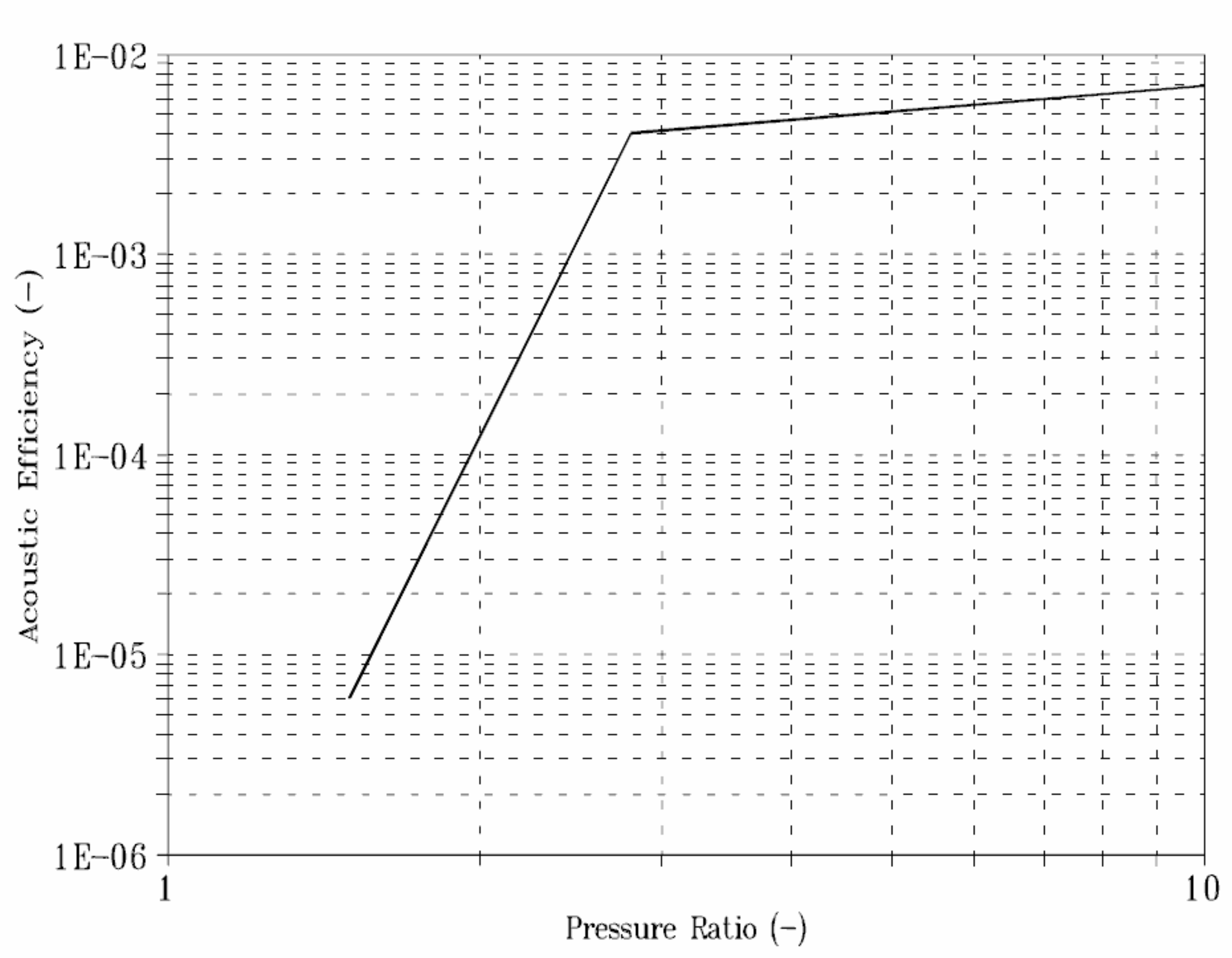


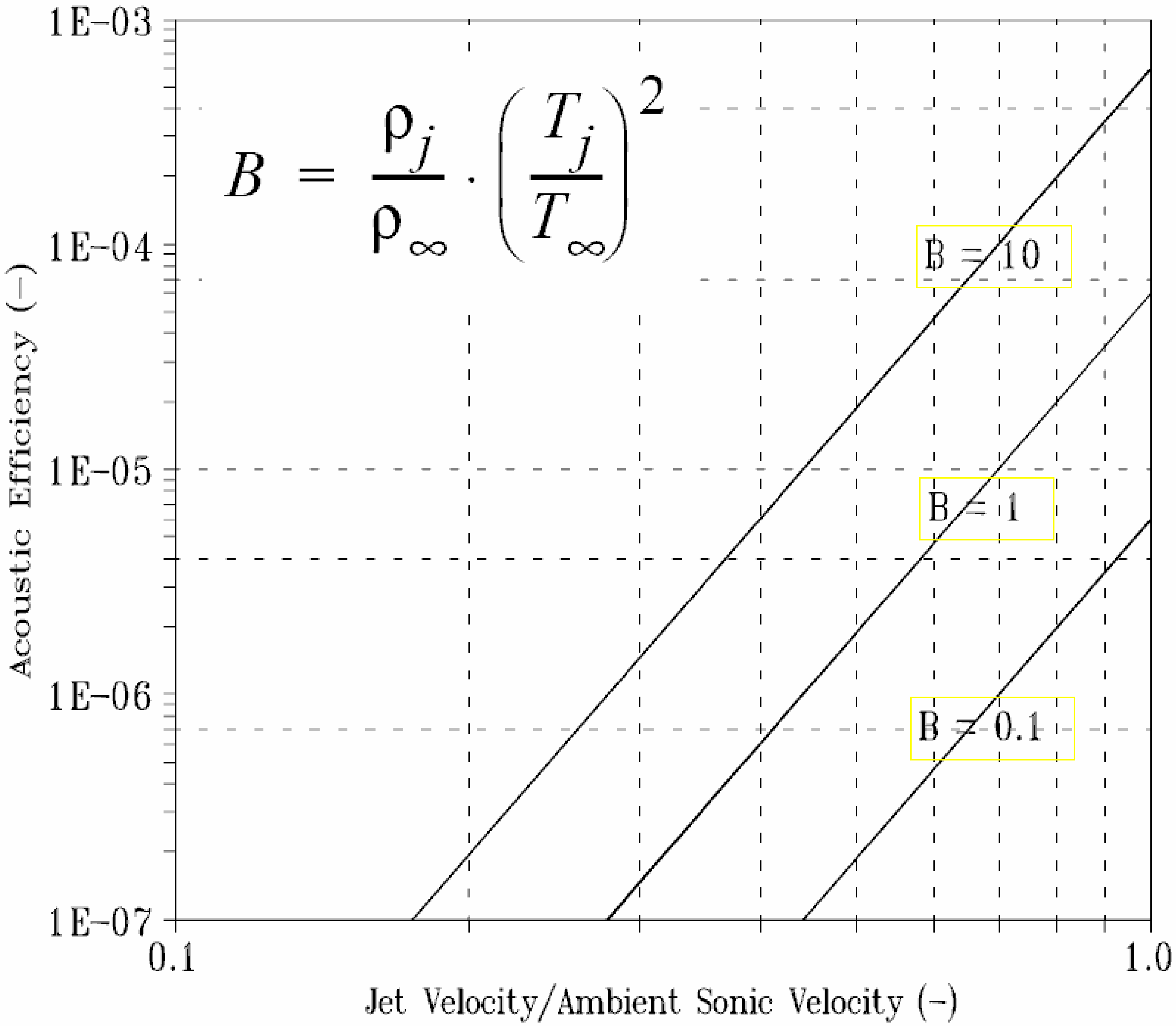
Figure 23—Noise Intensity at 100 Feet (30 Meters) From the Stack Tip

ioMosaic

Sonic Flow Acoustic Efficiency



Subsonic Flow Acoustic Efficiency



Process Piping Sound Power Level Equations

- The sound power level (L_w) in piping systems can be calculated for gas flow using the following equations:

$$L_{w,source} = 10\log\left[\left(\frac{P_1 - P_2}{P_1}\right)^{3.6} m^2 \left(\frac{T}{M_w}\right)^{1.2}\right] + 126.1$$

$$L_{w,branch} = L_{w,source} - 60\frac{L}{D_i}$$

$$L_{w,branch} = 10\log\sum_{i=1}^{n} 10^{\left[\frac{L_{w,branch,i}}{10}\right]}$$

T is upstream temperature in Kelvins

M_w is gas molecular weight

D_i is the main pipe internal diameter, mm

L is the distance between the source and the branch, m

Reference: Guidelines for the avoidance of vibration induced fatigue in process pipework, MTD Ltd., 1999

Combustion noise

- Combustion at the flare tip also produces noise which is approximately 16 dB higher than flow noise. However, the flow noise is predominant above 500 Hz
- For a typical hydrocarbon, the acoustical efficiency for burning on a standard flare tip is estimated at $\eta=5\times10^{-8}$

$$L_{w,combustion} = 10\log\left(\frac{5\times10^{-8} m\Delta H}{10^{-12}}\right) = 10\log\left(m\Delta H\right) + 36.98$$

$$L_{p,combustion} = 10\log\left(m\Delta H\right) + 36.98 - 10\log\left(4\pi r^2\right) = 10\log\left(\frac{m\Delta H}{r^2}\right) + 26$$

m is mass flow rate in kg/s

ΔH is heat of combustion in J/kg

r is distance from the combustion source in meters

Steam Injection Noise

- Steam assisted flares also generate noise due to the high pressure steam jets and injectors
- Additional noise is due to steam flow and enhanced combustion efficiency
- Steam jet noise can be reduced by using multi-port steam injectors
- Steam sound power level can be calculated using the equations specified previously
- Enhanced combustion sound power level can be estimated using the following expression:

$$L_{w,combustion} = 10\log\left(m_{gas}\Delta H\right) + 46.98 + 15\log\left(\frac{m_{steam}}{m_{gas}}\right)$$

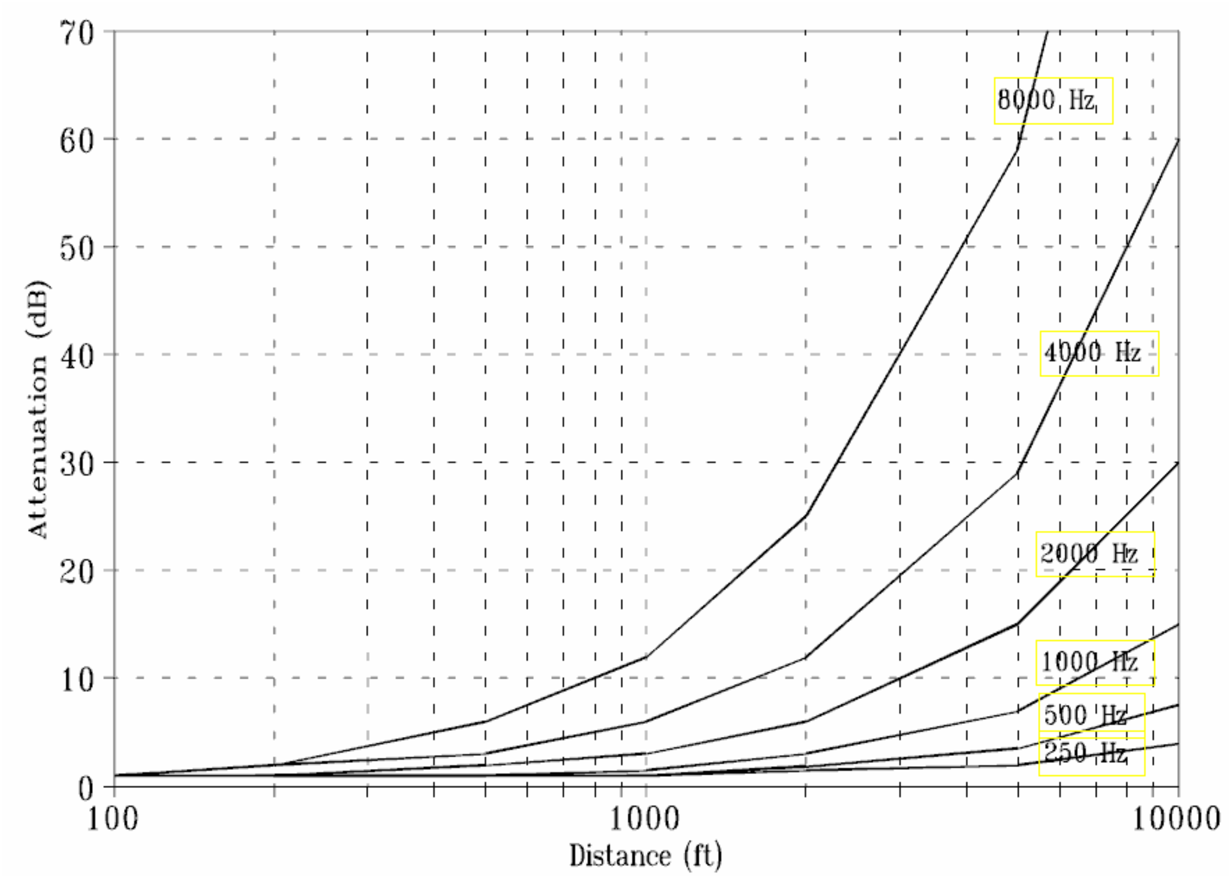
ioMosaic

Overall Flare Noise

- To find the overall flare noise add steam, combustion, and flow sound power level contributions logarithmically at the each frequency of interest

Atmospheric Attenuation

- At distances greater than 30 meters noise is attenuated due to absorption by the atmosphere. Higher frequencies of noise are more readily attenuated than lower frequencies



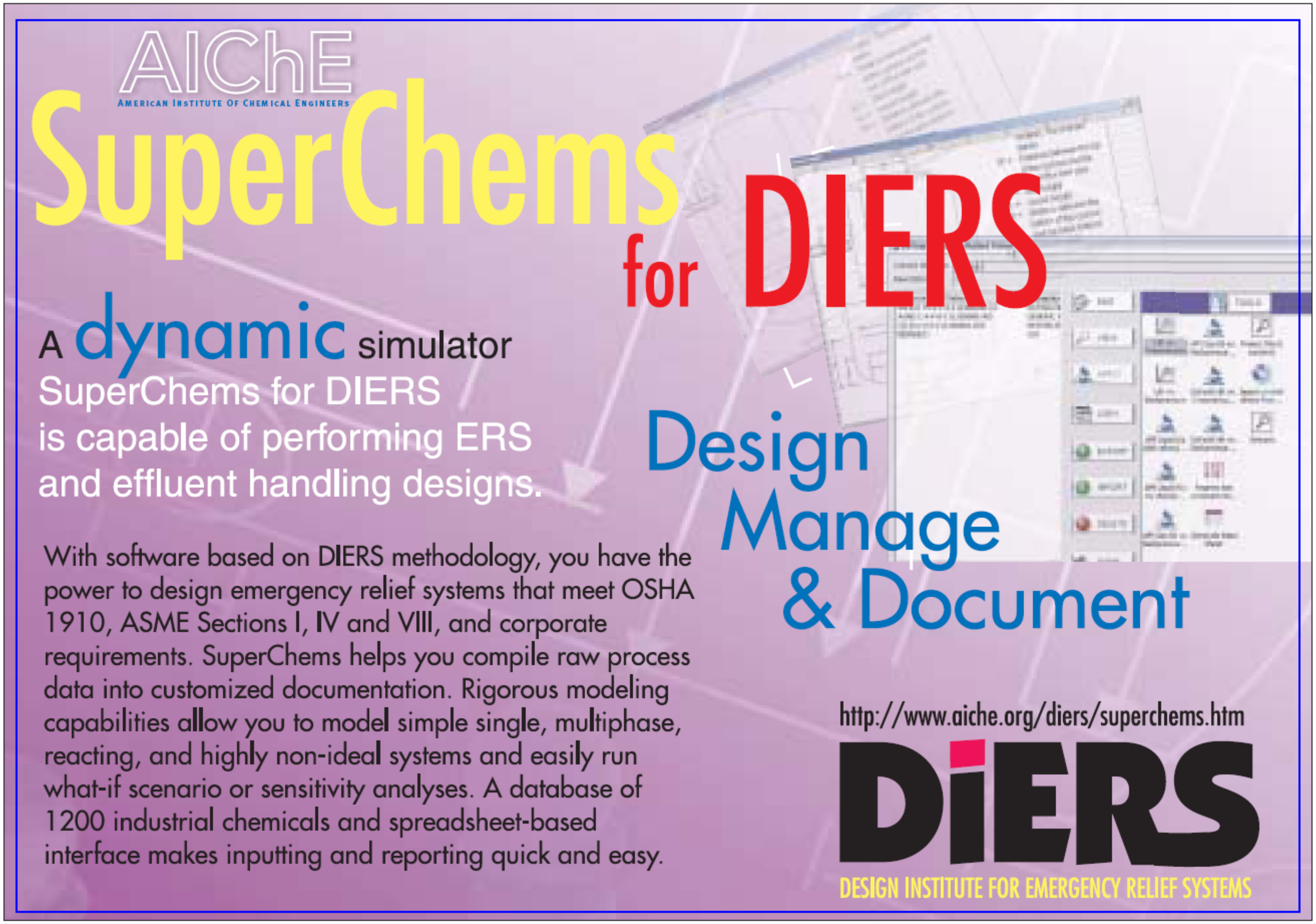
Graph is valid for zero wind speed, ambient temperature of 70 F, and a relative humidity greater than 60 %.

For temperatures below 70 F increase the attenuation by 10 % for each 10 F

Typical Noise Limits

- 0 dB → hearing threshold
- 30 dB → Whisper
- 60 dB → Talking
- 90 dB → City traffic
- 120 dB → Rock concert
- 150 dB → Jet engine at 10 meters standoff distance
- There are OSHA limits for continuous exposure to noise
- 85 dB is typically used as an exposure limit for emergency flaring

One Platform for Relief and Flare Systems



ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic Corporation is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire and Houston, Texas.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief systems design services and solutions. Its pressure relief system applications are used by over 250 users at the world's largest operating companies. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009
Fax: 603-251-8384
Email: traininingondemand@iomosaic.com
Web: www.iomosaic.com

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, TX 77057
Tel: 713-490-5220
Fax: 832-553-7283
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

Back pressure
DIERS Users Group Meeting
Charleston, March 21 - 23, 2005

LESER

Contents

- **LESER Company profile**
- **Reasons for malfunction of safety valves**
- **Back pressure**
- **LESER's back pressure test arrangements**
- **Analysis and discussion of API 520, Fig. 30**
- **Analysis and discussion of ISO 4126-1**
- **Conclusion**

Company profile – LESER

- **Product: safety valves, safety valves, safety valves**
- **Total turnover: € 33 million equals USD 43 million p.a.**
- **Yearly valve production: 62,000 pieces**
- **No. of employees: 300, thereof 25 apprentices**
- **Test lab: High capacity test labs for steam, water and gas, NB certified**

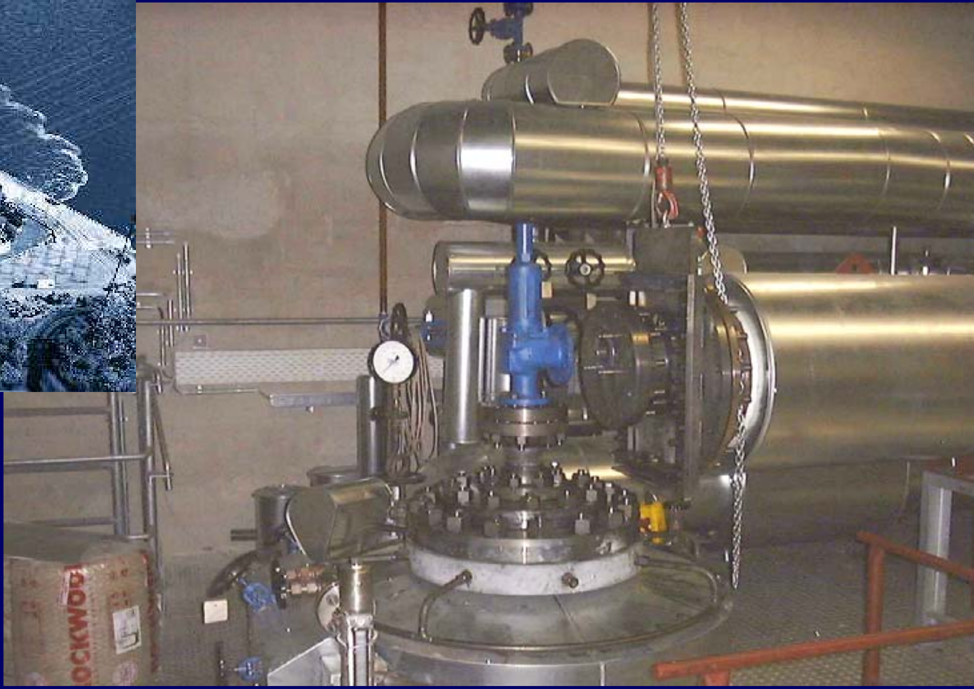
LESER

LESER Test lab
– water and air –

Total investments: € 1.5 million

LESER

LESER test lab
– steam –

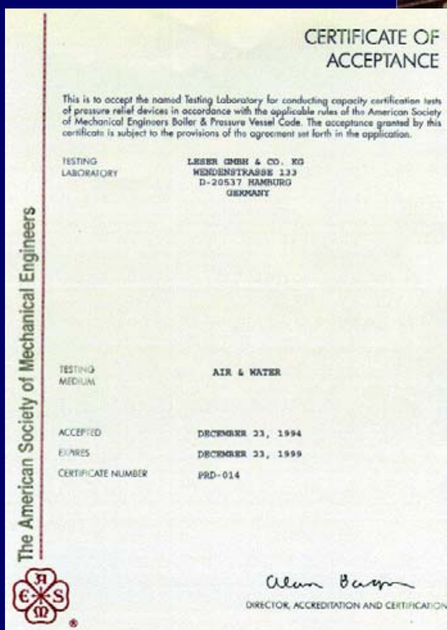


Total investments: € 1.2 million

LESER

Test lab approvals

ASME/NB



The American Society of Mechanical Engineers

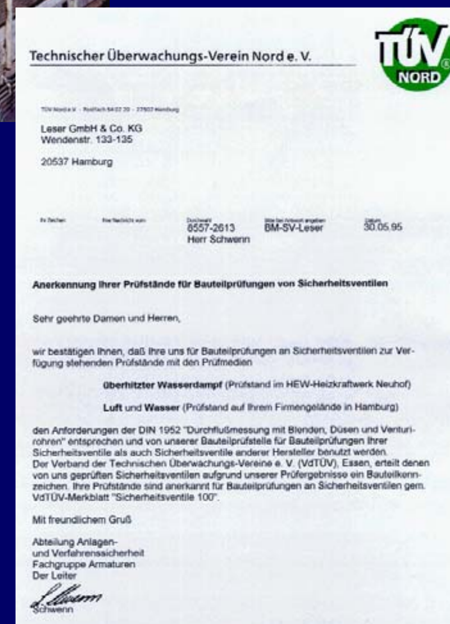
CERTIFICATE OF ACCEPTANCE

This is to accept the named Testing Laboratory for conducting capacity certification tests of pressure relief devices in accordance with the applicable rules of the American Society of Mechanical Engineers Boiler & Pressure Vessel Code. The acceptance granted by this certificate is subject to the provisions of the agreement set forth in the application.

TESTING LABORATORY	LESER GMBH & CO. KG WENDENSTRASSE 133 D-20537 HAMBURG GERMANY
TESTING MEDIUM	AIR & WATER
ACCEPTED	DECEMBER 23, 1994
EXPIRES	DECEMBER 23, 1999
CERTIFICATE NUMBER	PRD-014

DIRECTOR, ACCREDITATION AND CERTIFICATION

TÜV



Technischer Überwachungs-Verein Nord e. V.

TÜV NORD

Leser GmbH & Co. KG
Wendenstr. 133-135

20537 Hamburg

8557-2613
Herr Schwenn

BM-SV-Leser

30.05.95

Anerkennung Ihrer Prüfstände für Bauteilprüfungen von Sicherheitsventilen

Sehr geehrte Damen und Herren,

wir bestätigen Ihnen, daß Ihre uns für Bauteilprüfungen an Sicherheitsventilen zur Verfügung stehenden Prüfstände mit den Prüfmedien

überhitzter Wasserdampf (Prüfstand im HEW-Heizkraftwerk Neuhof)

Luft und **Wasser** (Prüfstand auf Ihrem Firmengelände in Hamburg)

den Anforderungen der DIN 1952 "Durchflußmessung mit Blenden, Düsen und Venturirohren" entsprechen und von unserer Bauteilprüfstelle für Bauteilprüfungen Ihrer Sicherheitsventile als auch Sicherheitsventile anderer Hersteller benutzt werden.
Der Verband der Technischen Überwachungs-Vereine e. V. (VdTÜV), Essen, erteilt denen von uns geprüften Sicherheitsventilen aufgrund unserer Prüfergebnisse ein Bauteilkennzeichen. Ihre Prüfstände sind anerkannt für Bauteilprüfungen an Sicherheitsventilen gem. VdTÜV-Merkblatt "Sicherheitsventile 100".

Mit freundlichem Gruß

Abteilung Anlagen-
und Verfahrenssicherheit
Fachgruppe Armaturen
Der Leiter

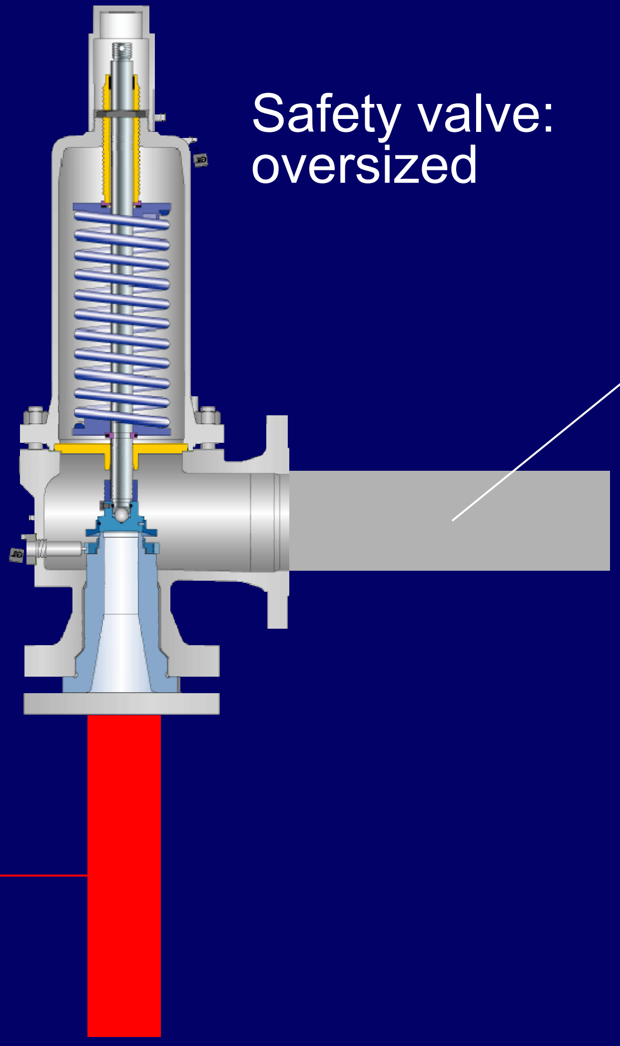
Schwenn

LESER

Contents

LESER

Reasons for malfunction of safety valves



Reasons for malfunction of safety valves

Mal-functions of safety valves

- **Flattering**
- **Chattering**
- **Untightness**

Reasons for malfunctions

- **Safety valve: oversized**
- **Inlet pressure drop: too high**
- **Back pressure: too high**

LESER

Reasons for malfunction of safety valves

Mal-functions of safety valves

Flattering

Chattering

Untightness

Reasons for malfunctions

- **Safety valve: oversized**
- **Inlet pressure drop: too high**
- **Back pressure: too high**

The main focus is on back pressure in this presentation

Contents

LESER

Target of this lecture

Verification
of analysis and comparison
of back pressure statement
in API 520 and
ISO 4126 Part 1

LESER

What is back pressure?

In API 520 item 3.3.1.1 back pressure is defined as "… pressure existing at the outlet of a pressure relief valve …"

Two kinds of back pressure

Built-up back pressure

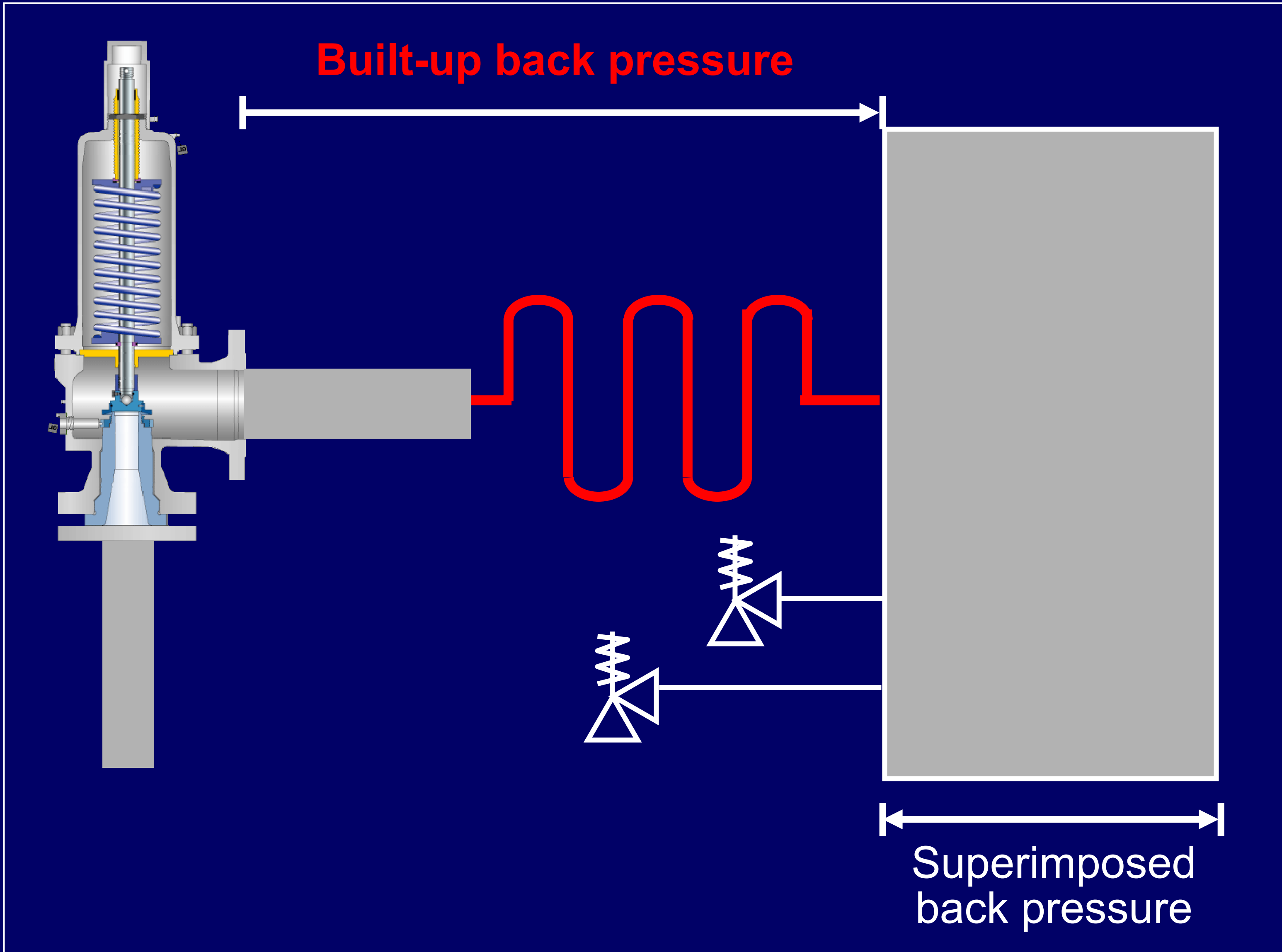
- Caused by pressure losses in the outlet piping
- Amounts to its maximum value at full discharge of the safety valve
- Built-up pressure is 0 when the valve is closed

Superimposed back pressure

- Caused by blow-down system or other valves
- Can be either constant or variable. Variable especially when other valves are discharging in the blow-down system

LESER

Back pressure



Back pressure in rules

In the different rules and regulations throughout the world roughly the following information or guidelines are given regarding back pressure:

- **Manufacturer should be consulted for correction factors for back pressure and operating limits**
- **Outlet piping should be designed in such a way that permissible back pressure specified by the manufacturer of safety valve is not exceeded**

Back pressure in API 520 Edition 2000

- **Conventional safety valve: built-up back pressure should not exceed 10% (API 520 item 3.3.3.1.3)**
- **Balanced valves can typically be applied whether total back pressure does not exceed approximately 50% of the back pressure (e.g. API 520 item 3.3.2.1)**
- **Tables for back pressure correction factors for balanced bellows pressure relief valves (e.g. at different overpressure situations such at 10%, 16% and 21% (e.g. API 520 fig. 30 and 31)**

LESER

Missing information

- **Rules and regulations do not provide formulas for the calculation of built-up back pressures in existing piping.**
  In that case the manufacturer has take it from other sources.
  Exception: The new AD-A2 Merkblatt gives some formulas but they are not mutual agreed among all committee members.
- **In ASME and API there is no information about subcritical flow available**

Aspects / raised questions in previous DIERS Meetings

Do current makers of safety valves follow the back pressure correction factor curve as shown in API 520 part 1 figure 30?

What are the operating limits (expressed in ratio back pressure / set pressure) of existing valve makes with regard to back pressure?

Can we expect other results when operating bigger valves because they are not exactly geometric comparable to a tested valve?

Set pressure changes due to variable superimposed back pressure

LESER

Why does LESER perform back pressure tests?

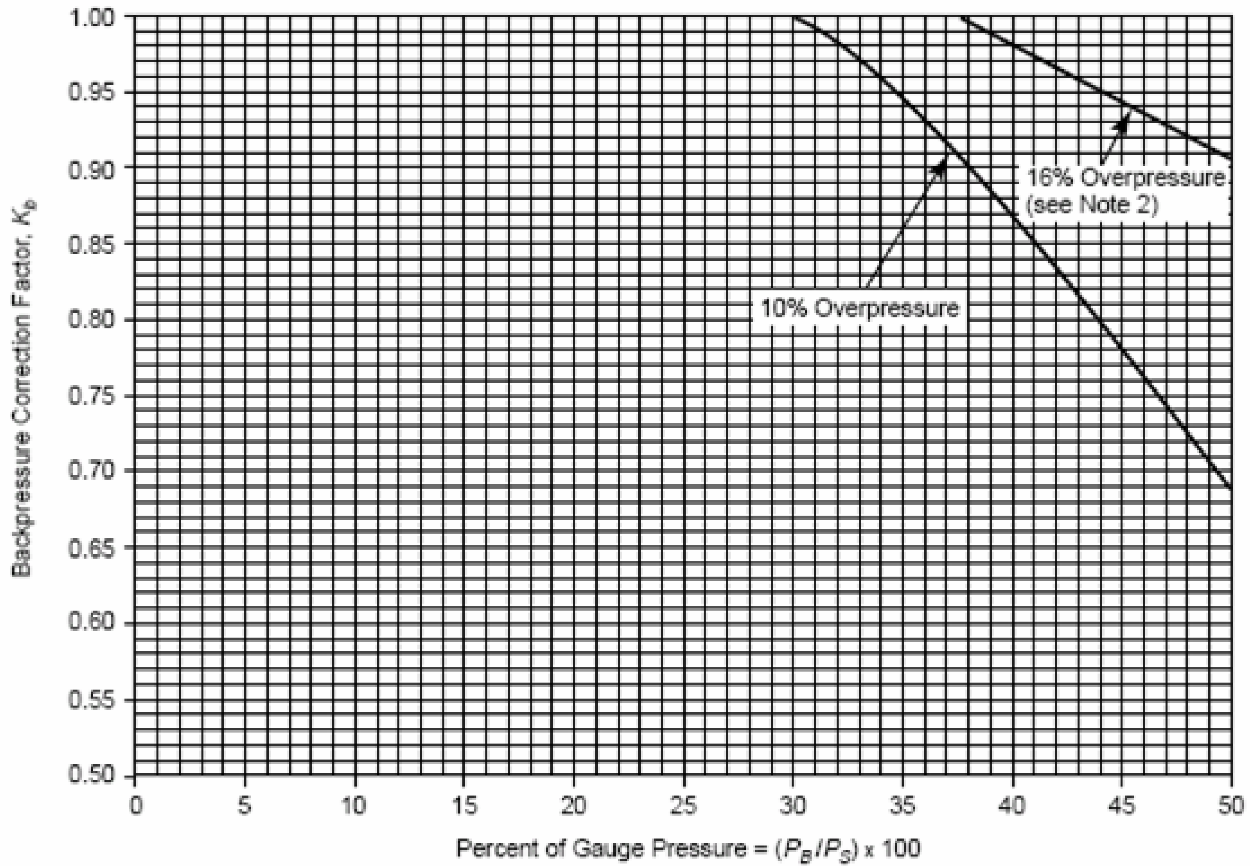
Reasons

LESER has the capability

Customers ask for review of existing plants

There are more and more subcritical flow situations in Europe

API 520, Edition 2000



P_B = back pressure, in psig.
P_S = set pressure, in psig.

Figure 30 - Back Pressure Correction Factor, K_b, for Balanced-Bellows Pressure Relief Valve (Vapors and Gases)

LESER

ISO 4126, Part 1

Chapter 8: Determination of safety valve performance

8.1 Determination of coefficient of discharge

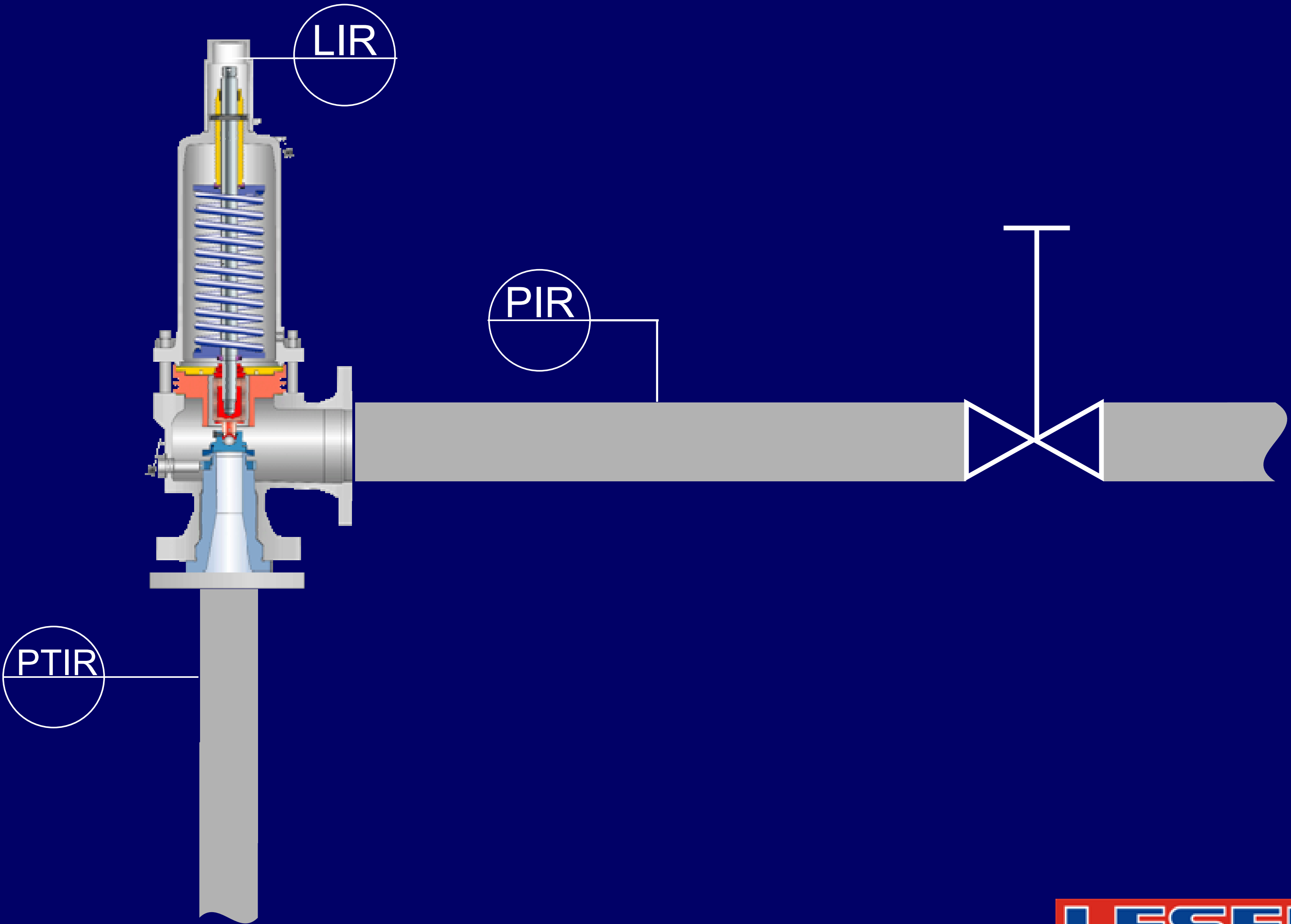
8.2 Critical and subcritical flow

8.3 Discharge capacity at critical flow

8.4 Discharge capacity for any gas at subcritical flow

Contents

Test arrangement



Test arrangement

LESER
The Safety Valve

LESER

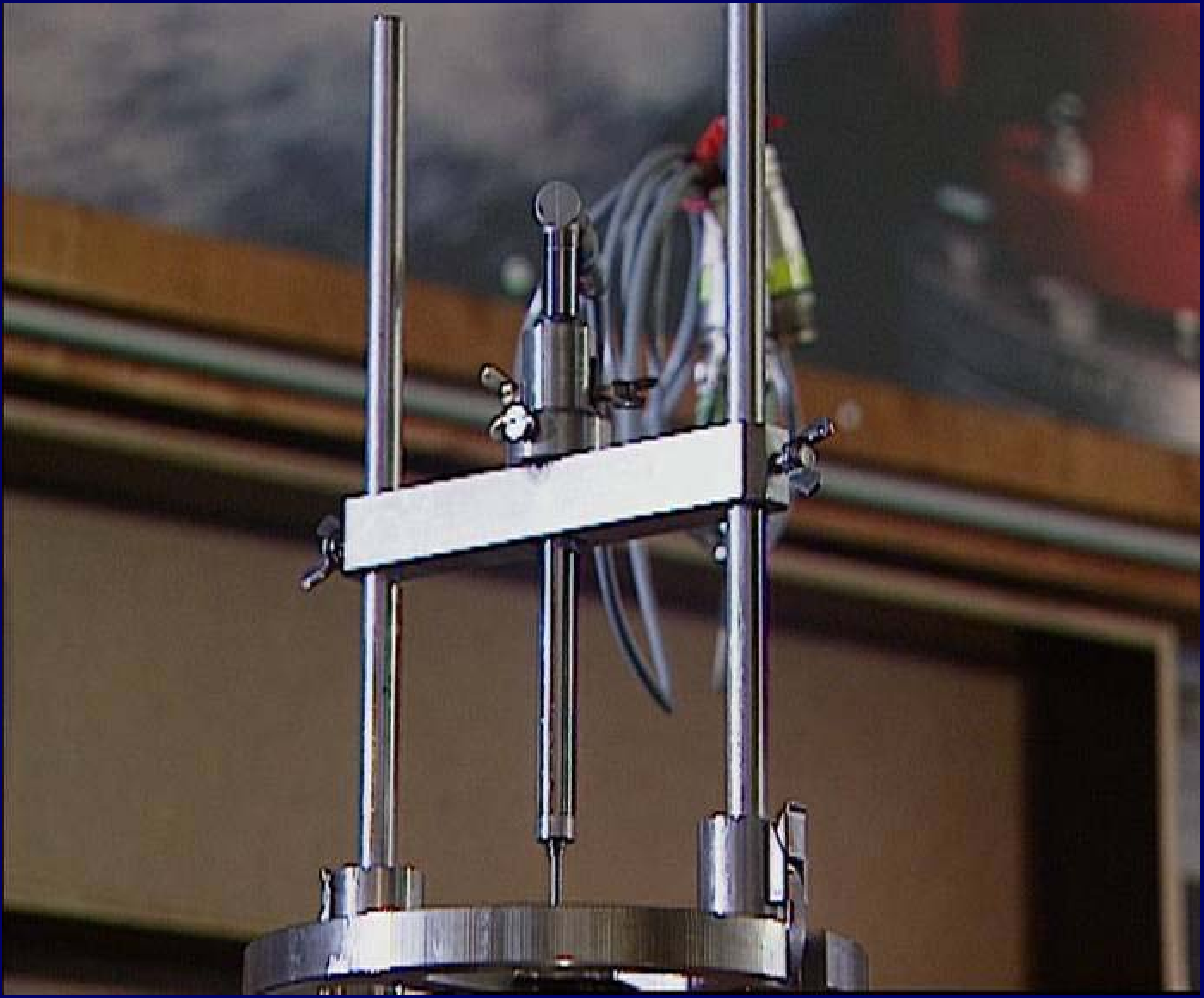
The Safety Valve

Inlet situation according to ASME PTC 25

Test equipment

LESER

Lift measurement



Back pressure gauge and pressure transducer

Test object

- LESER tested its entire product range
- The API Series was tested as follows (up to now):

Valve size	Pressure level (barg)	(psig)
1E2 2J3 3L4	4 / 6 / 10	58 / 87 / 145

Test object

- LESER tested its entire product range
- The API Series was tested as follows (up to now):

All diagramms in this lecture show test results of LESER Type 526 3L4

2J3	4 / 6 / 10	58 / 87 / 145
3L4		

Performance of 526 3L4, conventional, without bellows, without back pressure

LESER

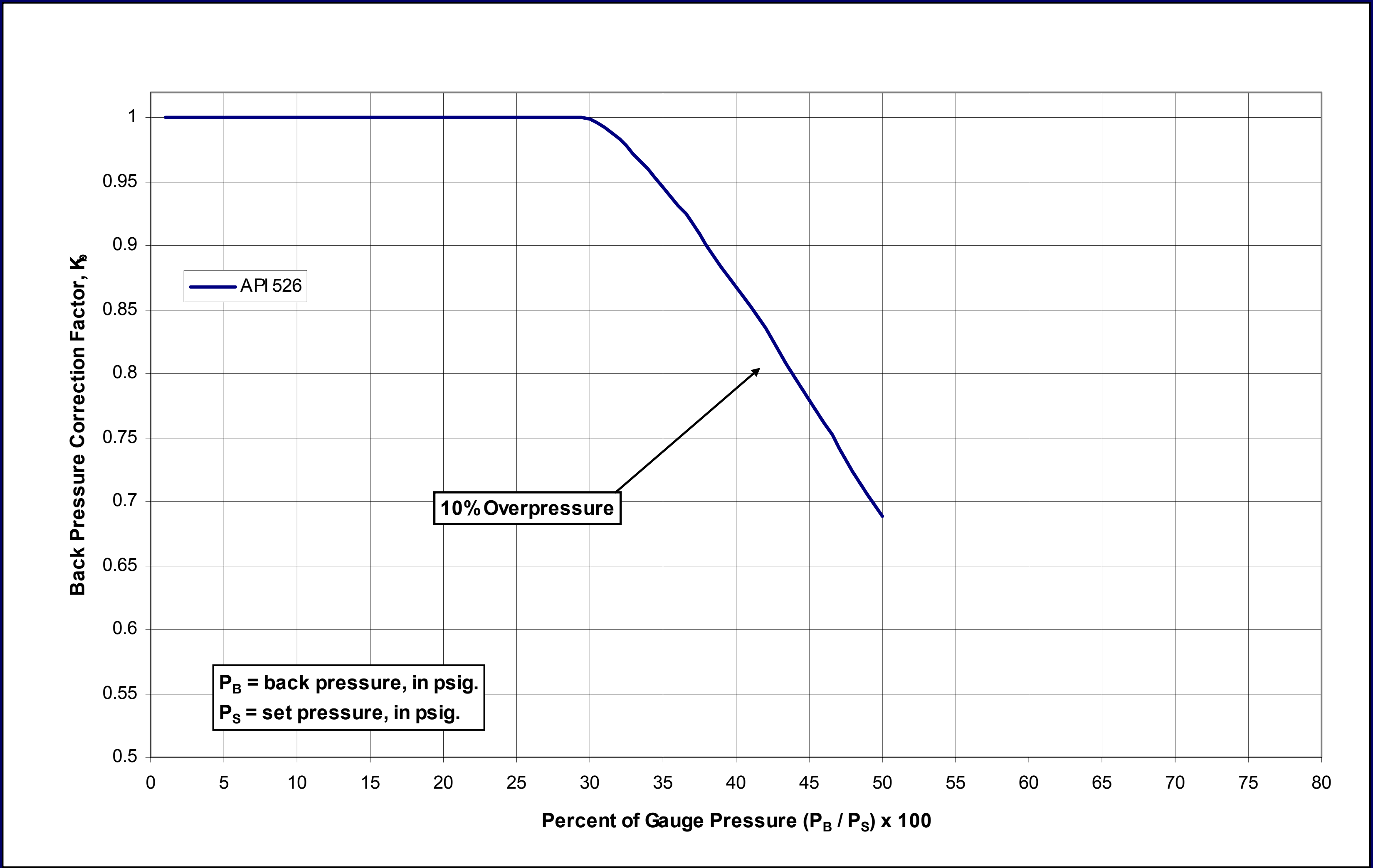
The Safety Valve

Contents

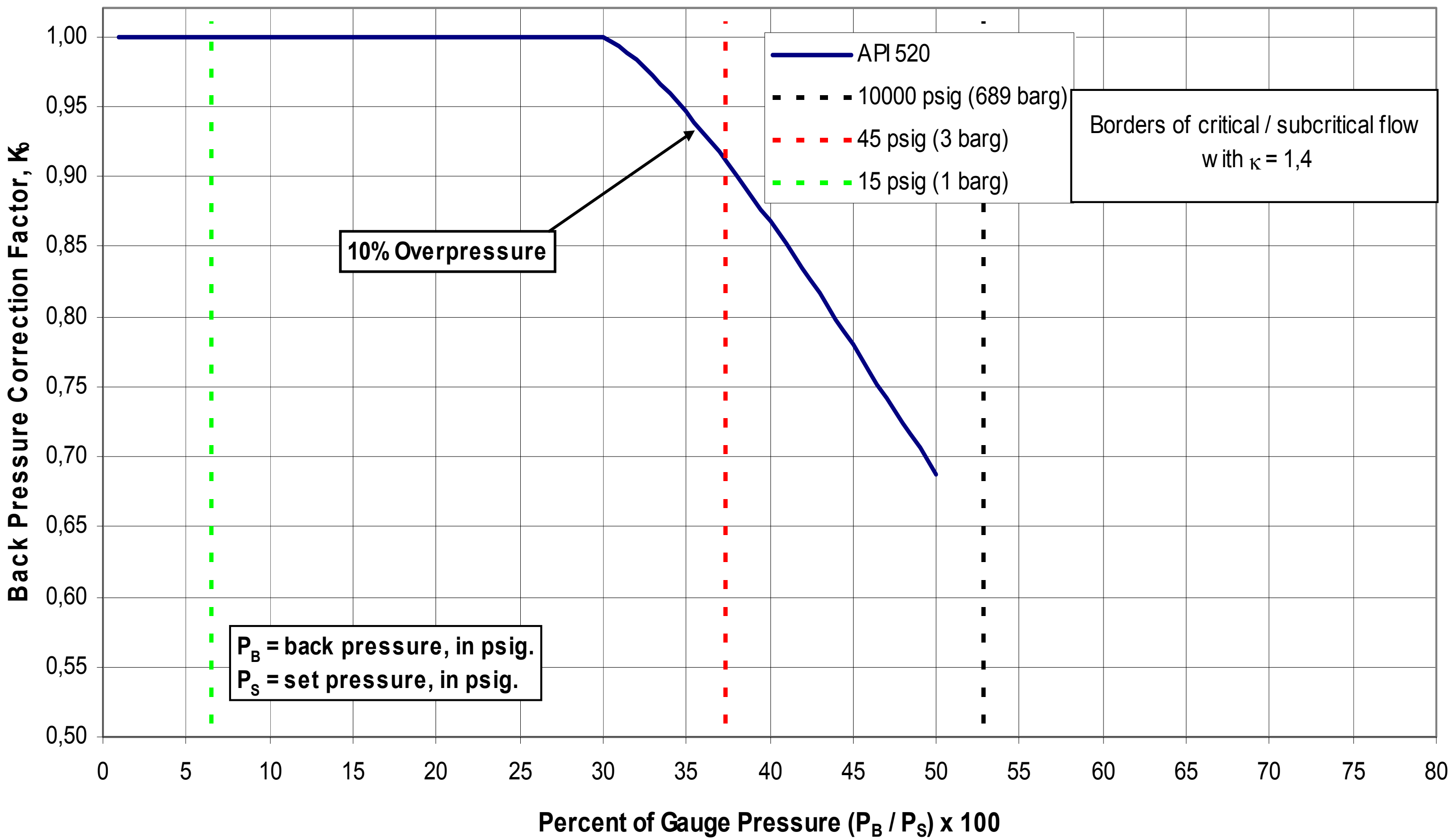
- LESER Company profile
- Reasons for malfunction of safety valves
- Back pressure
- LESER's back pressure test arrangements
- Analysis and discussion of API 520, Fig. 30
- Analysis and discussion of ISO 4126-1
- Conclusion

LESER

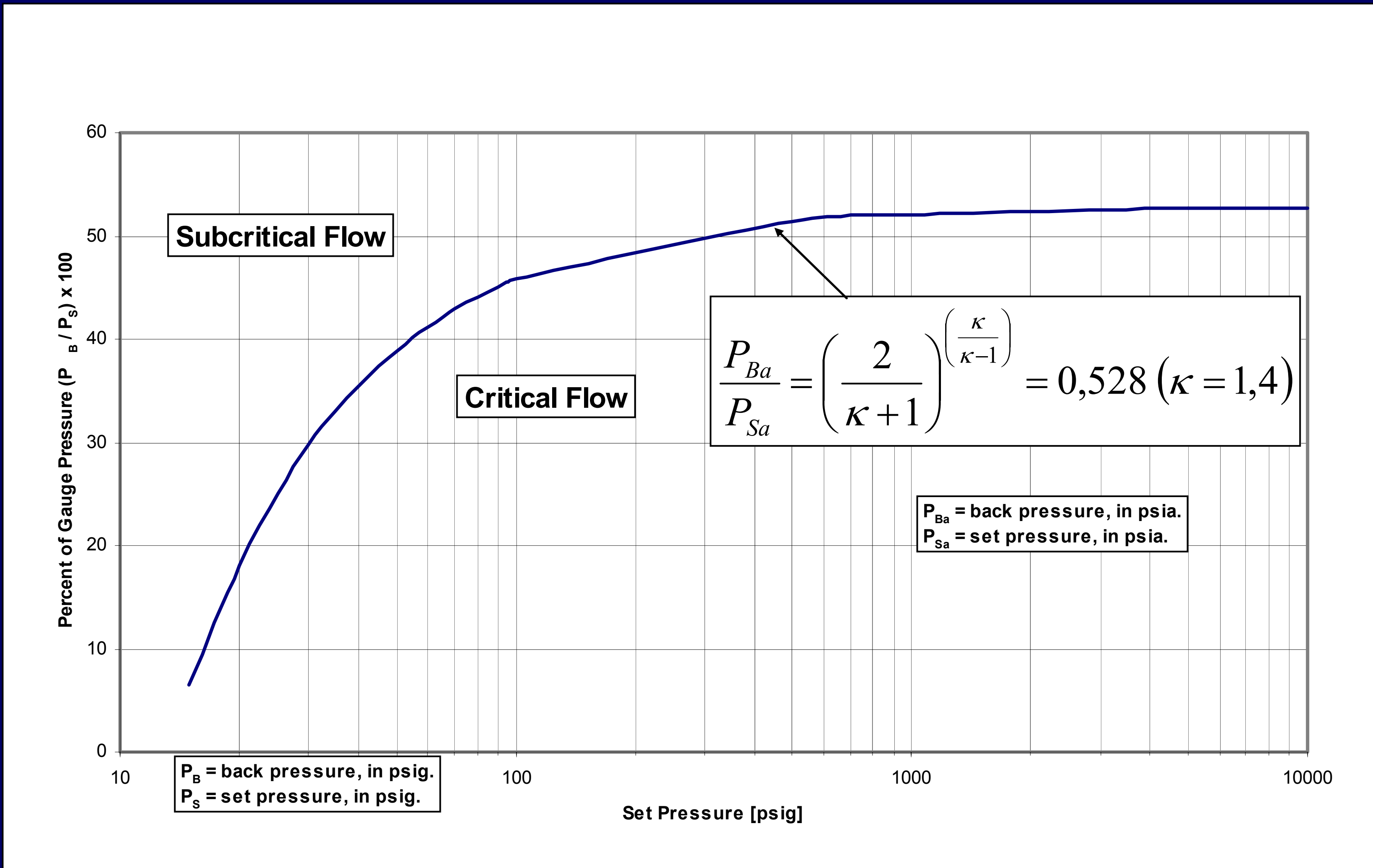
Back pressure correction factor K_b (converted to Excel)



API 520, fig. 30 and limits showing critical and subcritical flow

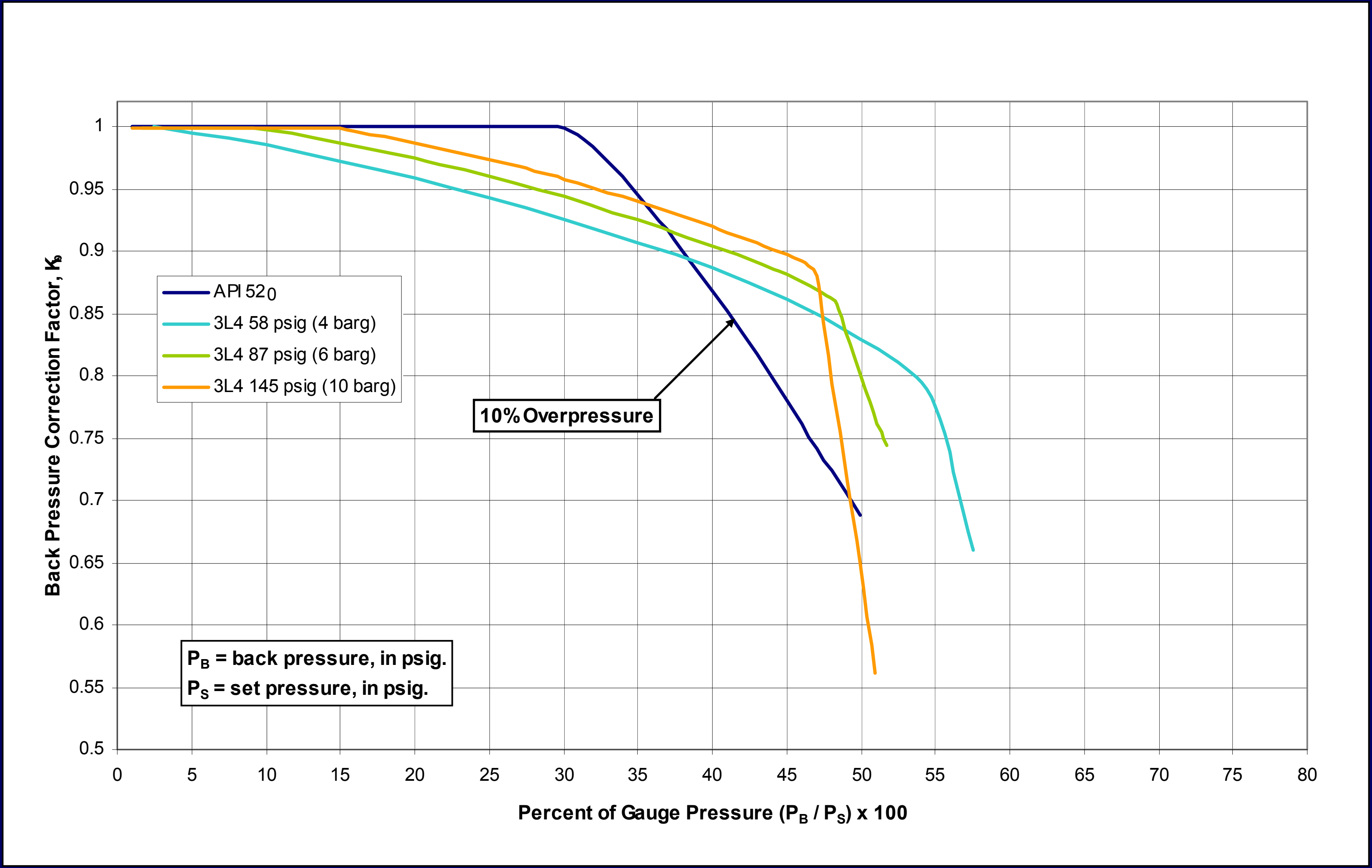


Border of critical flow



Back Pressure Correction Factor, K_b

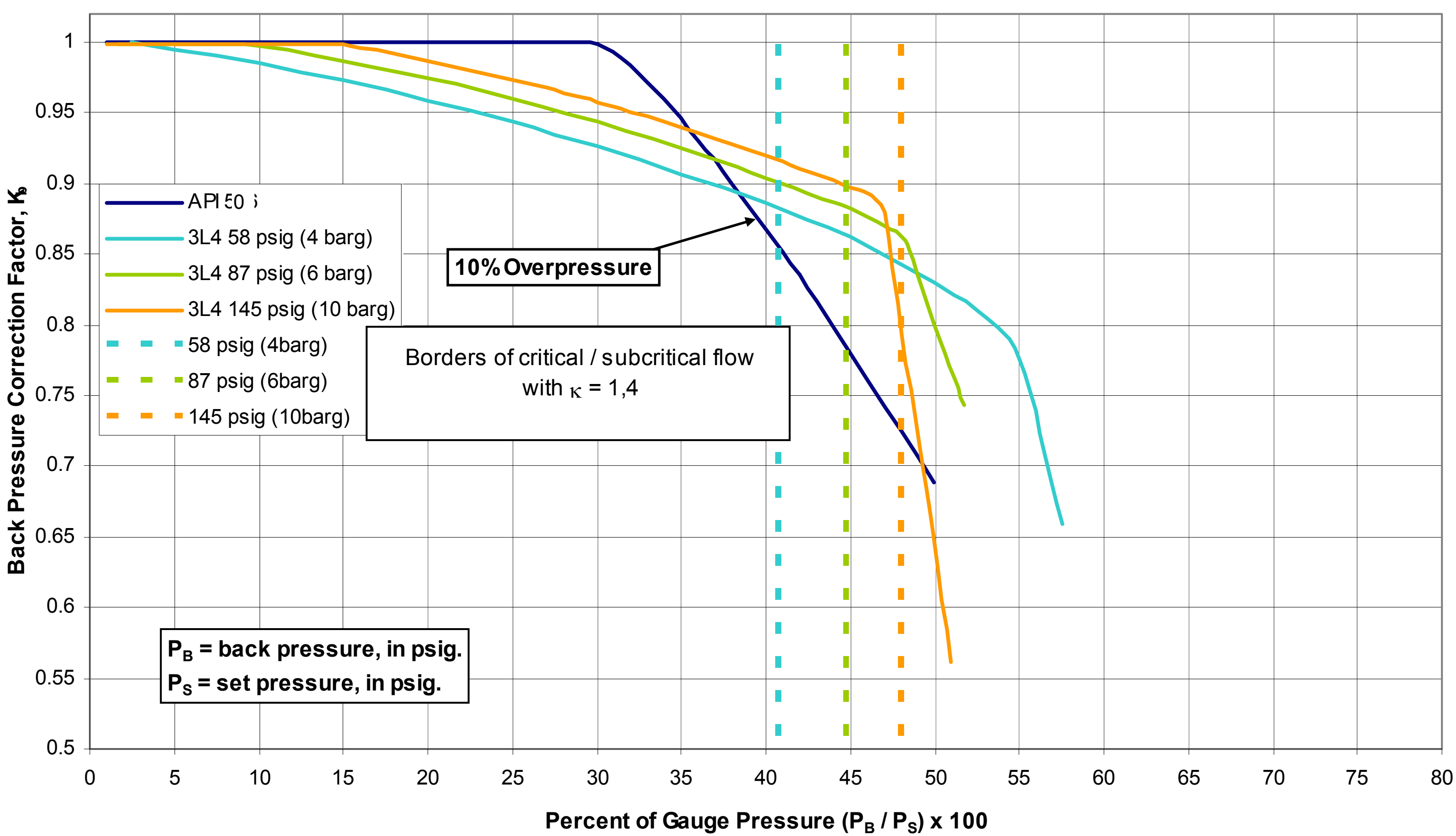
for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures, κ = 1.4, Test results LESER 526 3L4



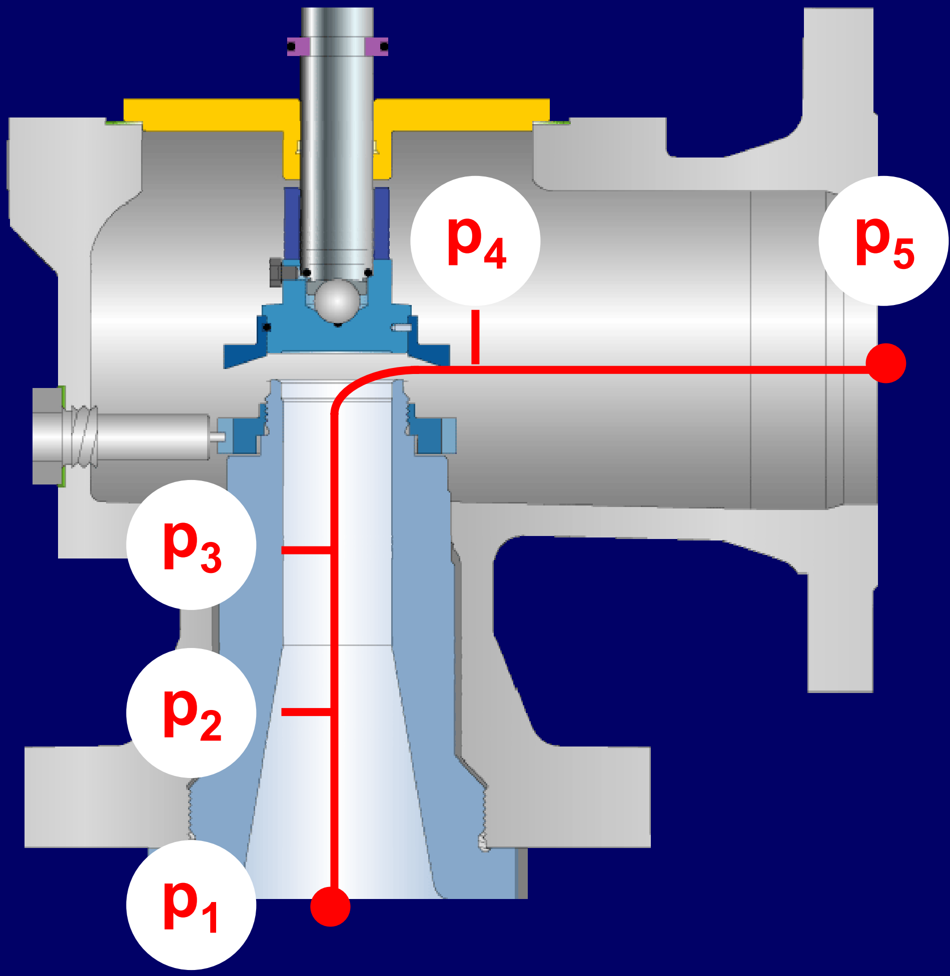
LESER

Back Pressure Correction Factor, K_b

for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures, κ = 1.4, Critical / subcritical line



Pressure cascades



Critical flow:

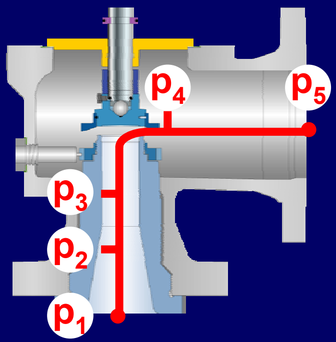
$$\frac{P_{n+1}}{P_n} \leq \left[\frac{2}{(\kappa+1)}\right]^{\left(\frac{\kappa}{(\kappa-1)}\right)}$$

Subcritical flow:

$$\frac{P_{n+1}}{P_n} > \left[\frac{2}{(\kappa+1)}\right]^{\left(\frac{\kappa}{(\kappa-1)}\right)}$$

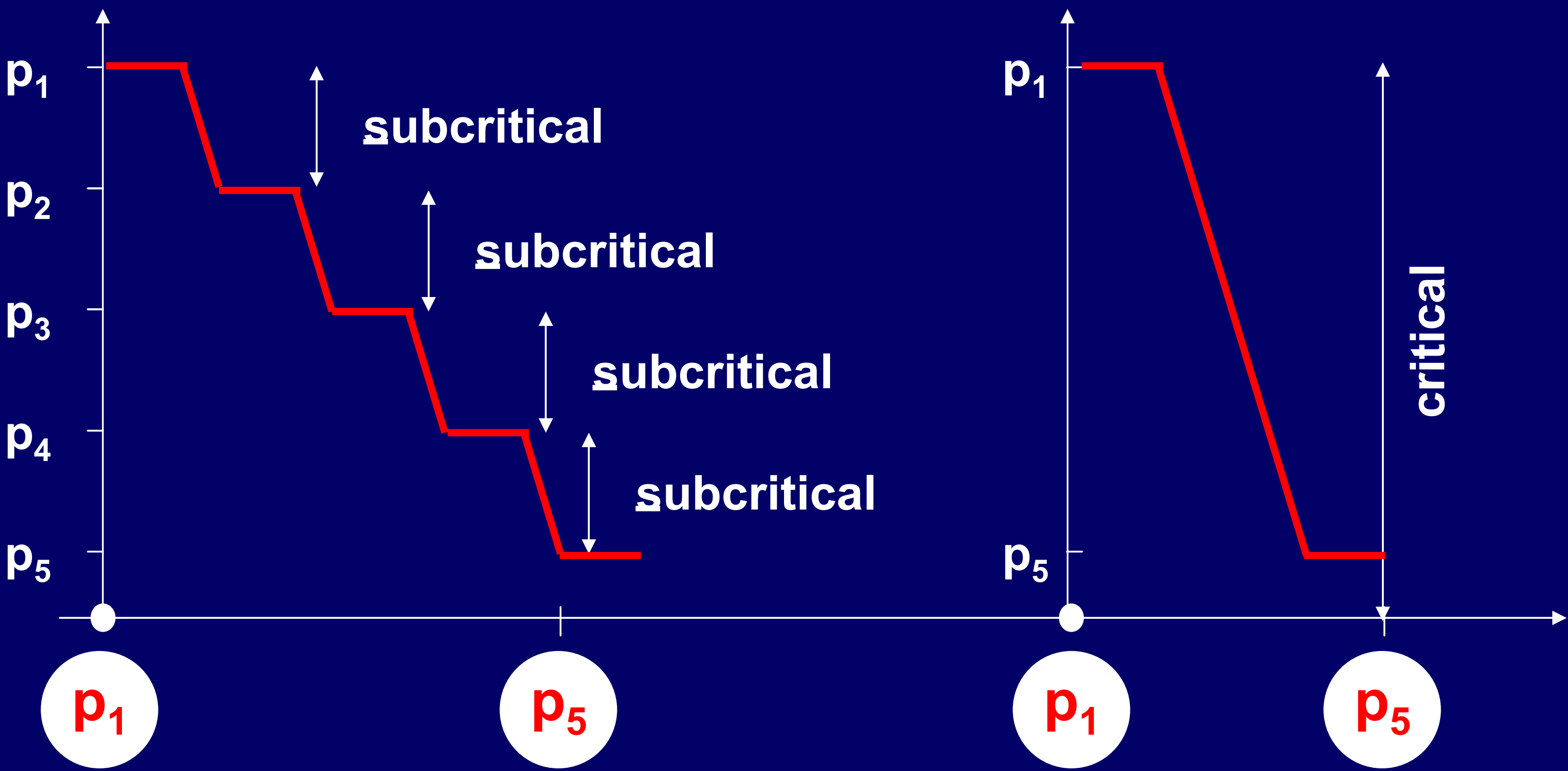
LESER

Pressure cascades



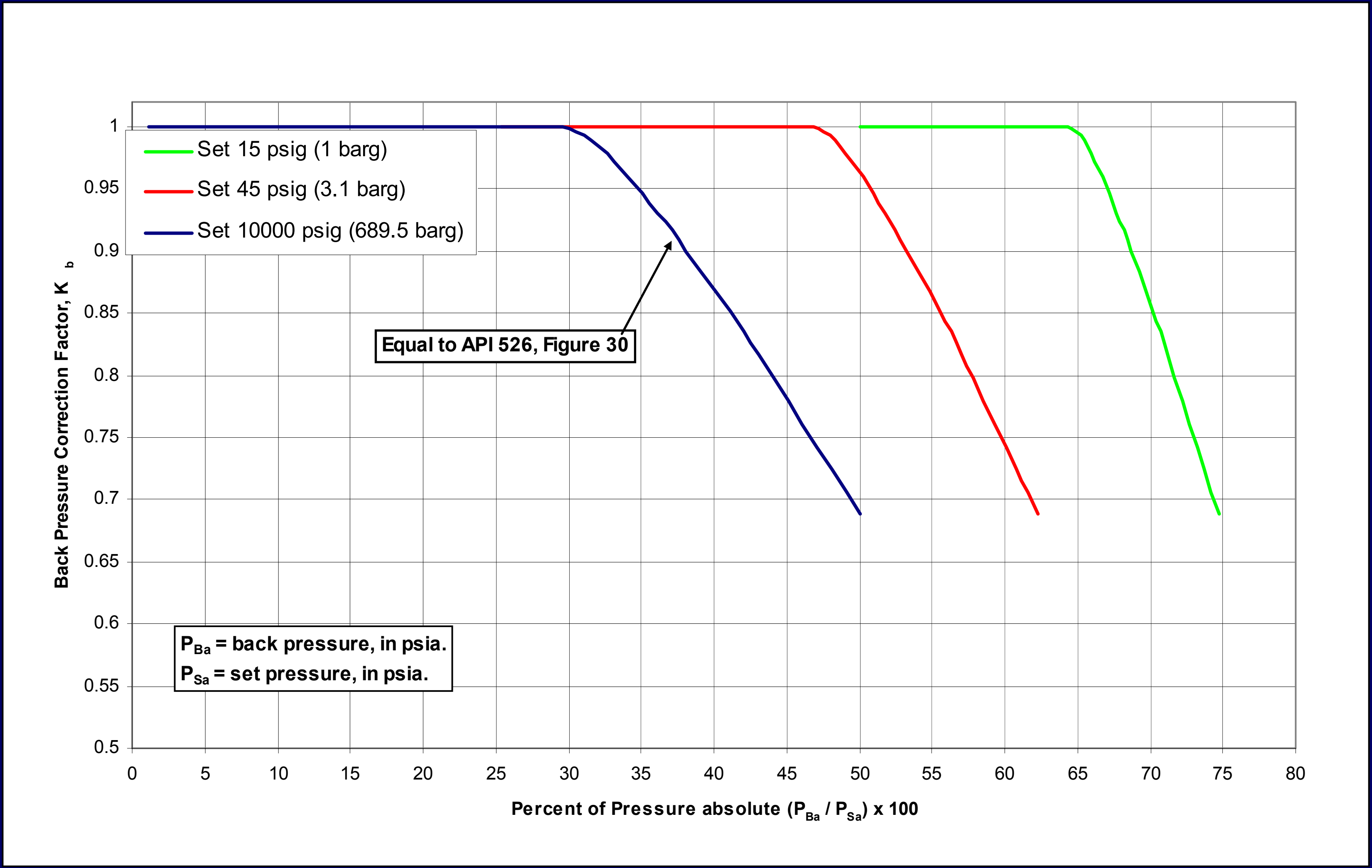
$$\frac{P_{n+1}}{P_n} > 0.528$$

$$\frac{P_5}{P_1} \leq 0.528$$



Back Pressure Correction Factor, K_b

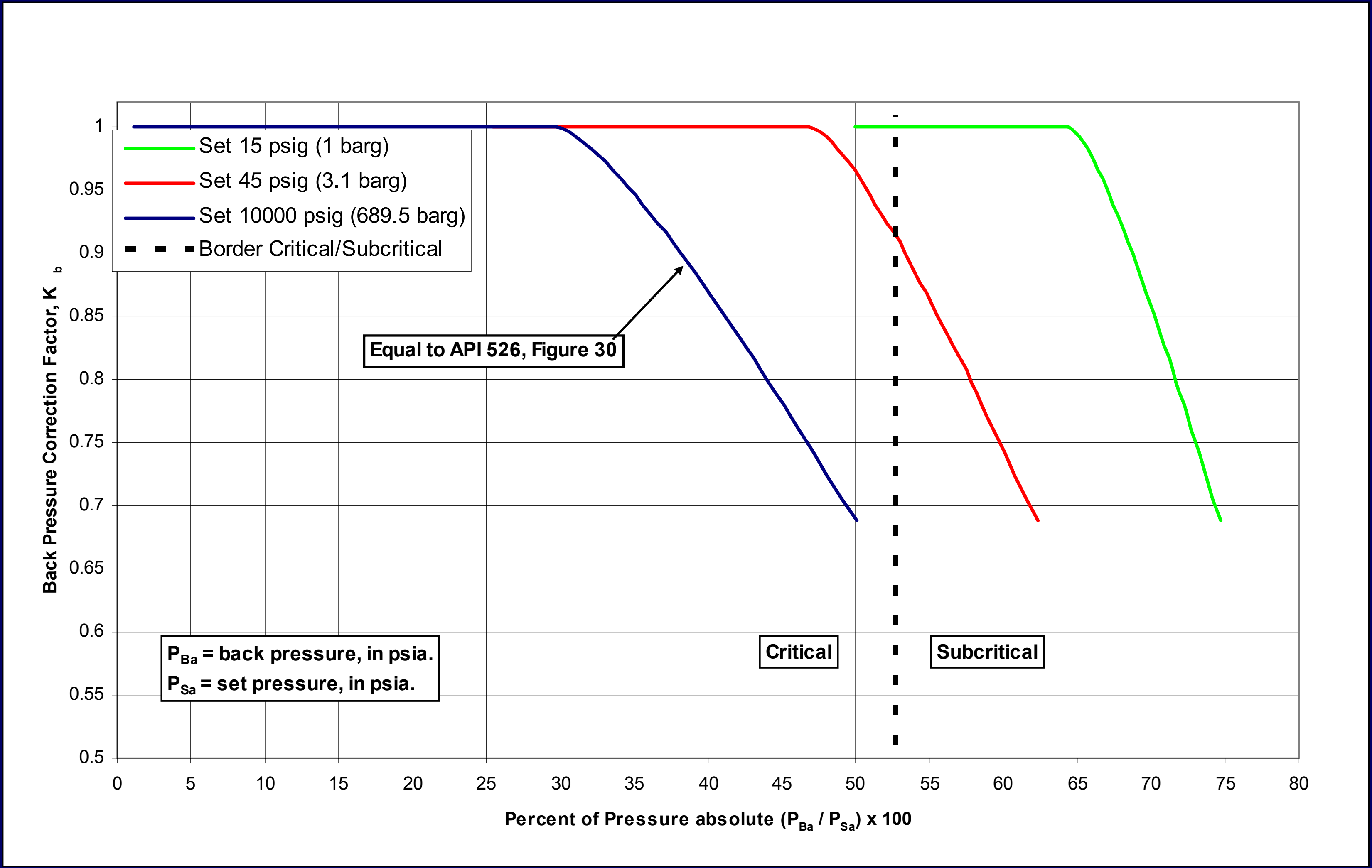
for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures at different set pressures



LESER

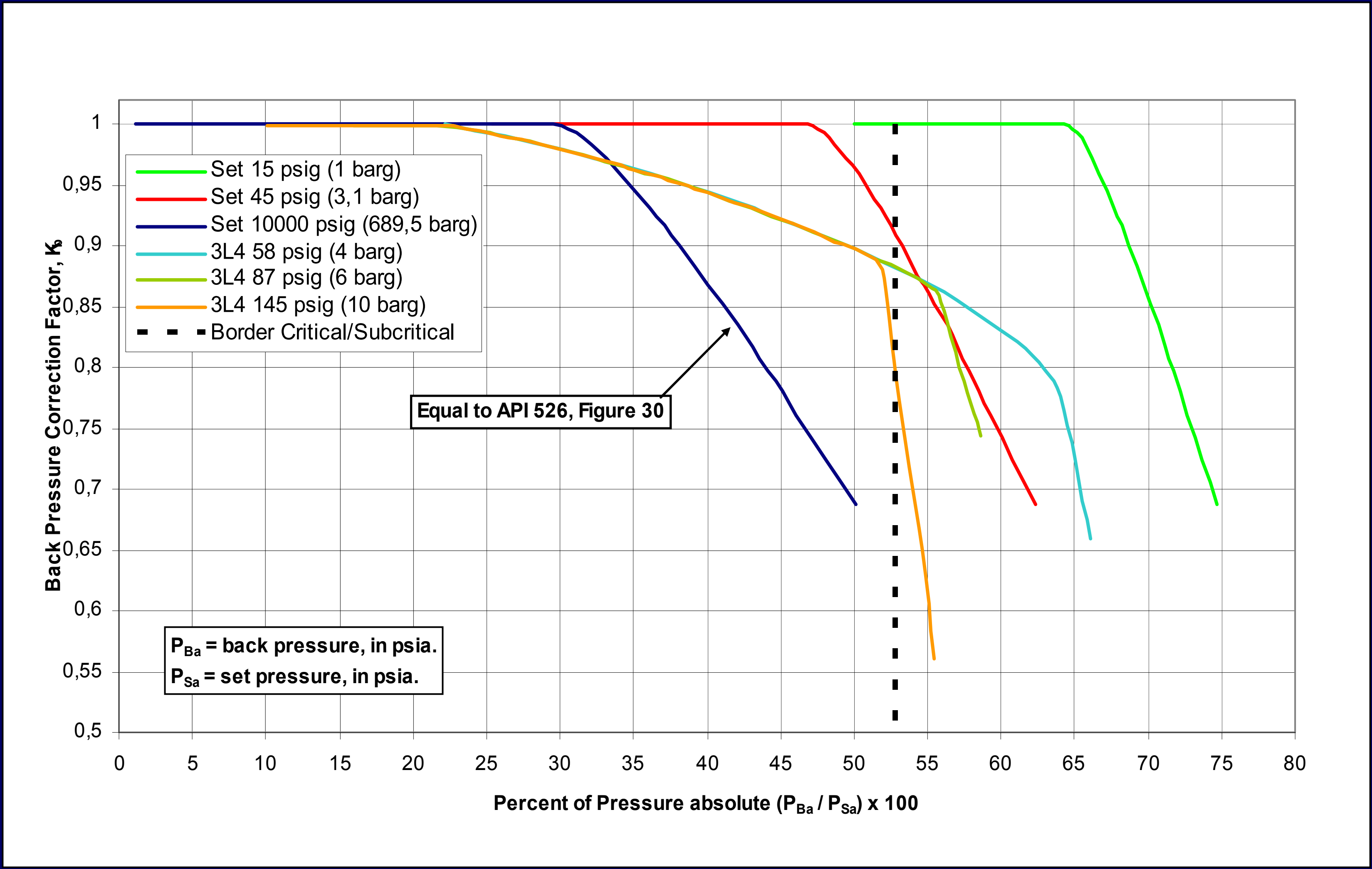
Back Pressure Correction Factor, K_b

for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures, κ = 1.4



Back Pressure Correction Factor, K_b

for balanced bellows – Pressure Relief Valve (Vapors and Gases), using absolute pressures, κ = 1.4



LESER

Performance of 526 3L4, bellows design, with back pressure

LESER

The Safety Valve

Contents

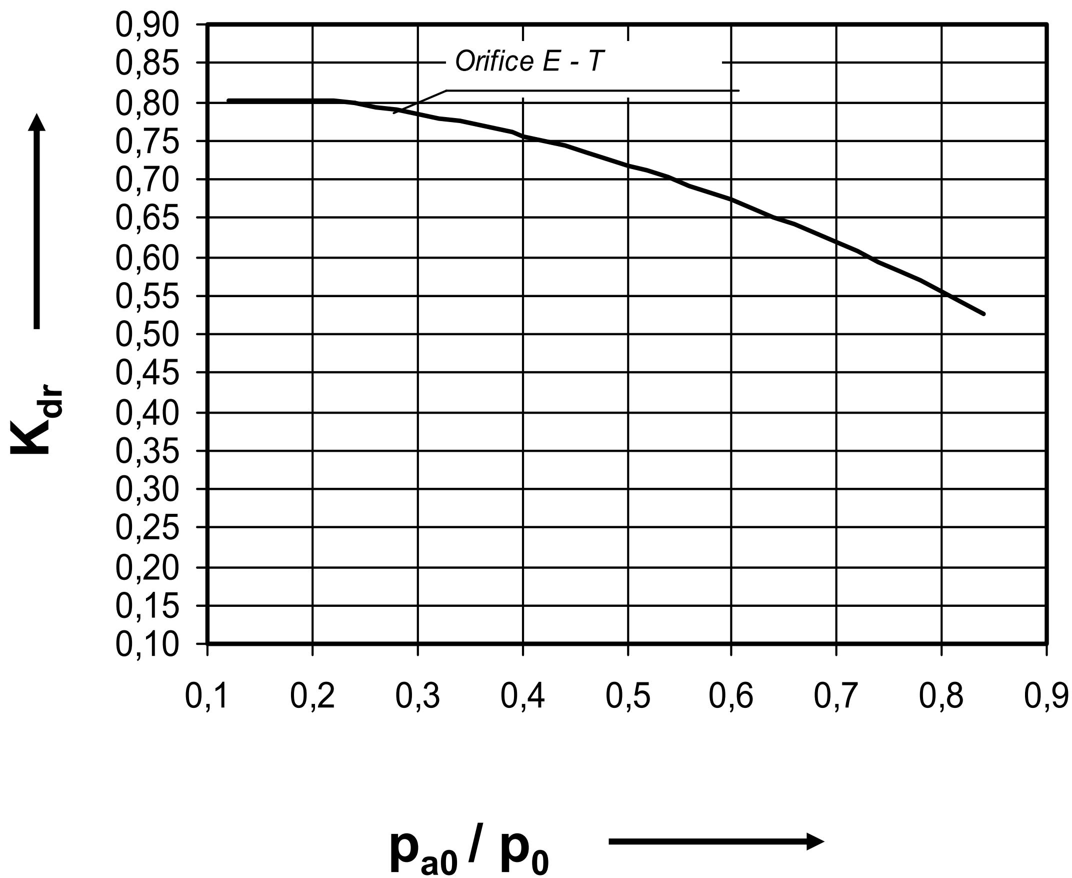
- LESER Company profile
- Reasons for malfunction of safety valves
- Back pressure
- LESER's back pressure test arrangements
- Analysis and discussion of API 520, Fig. 30
- Analysis and discussion of ISO 4126-1
- Conclusion

Backpressure from the European point of view

ISO 4126-1, Chapter 7.3.3.4

“For compressible fluids when the ratio of absolute back pressure to absolute relieving pressure exceeds the value of 0.25, the coefficient of discharge can be largely depending on this ratio.”

Backpressure acc. to ISO 4126-1

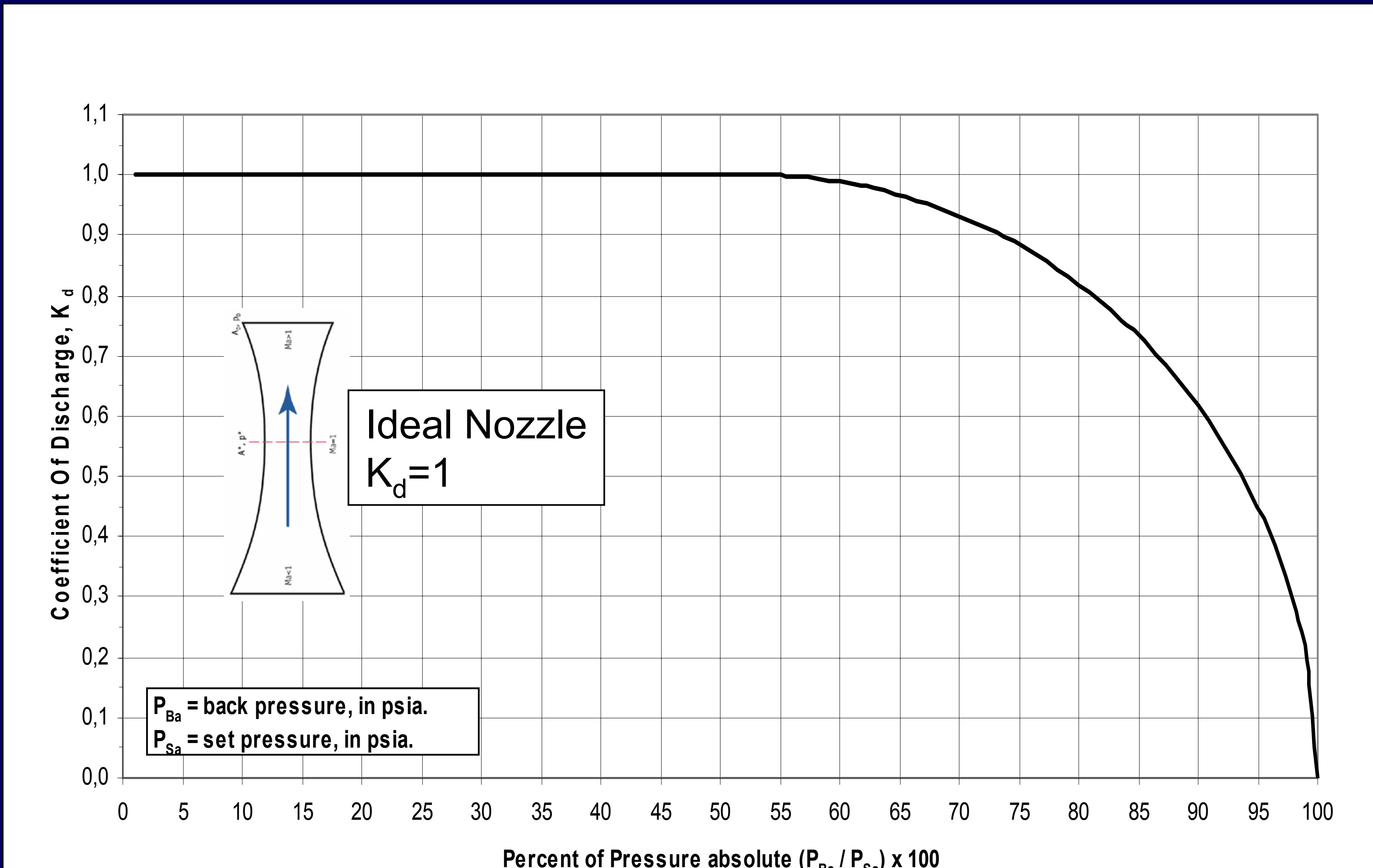


Certified coefficient of discharge K_{dr} in relation to the quotient p_{a0} / p_0 for steam and gases

Coefficient Of Discharge, K

using absolute pressures

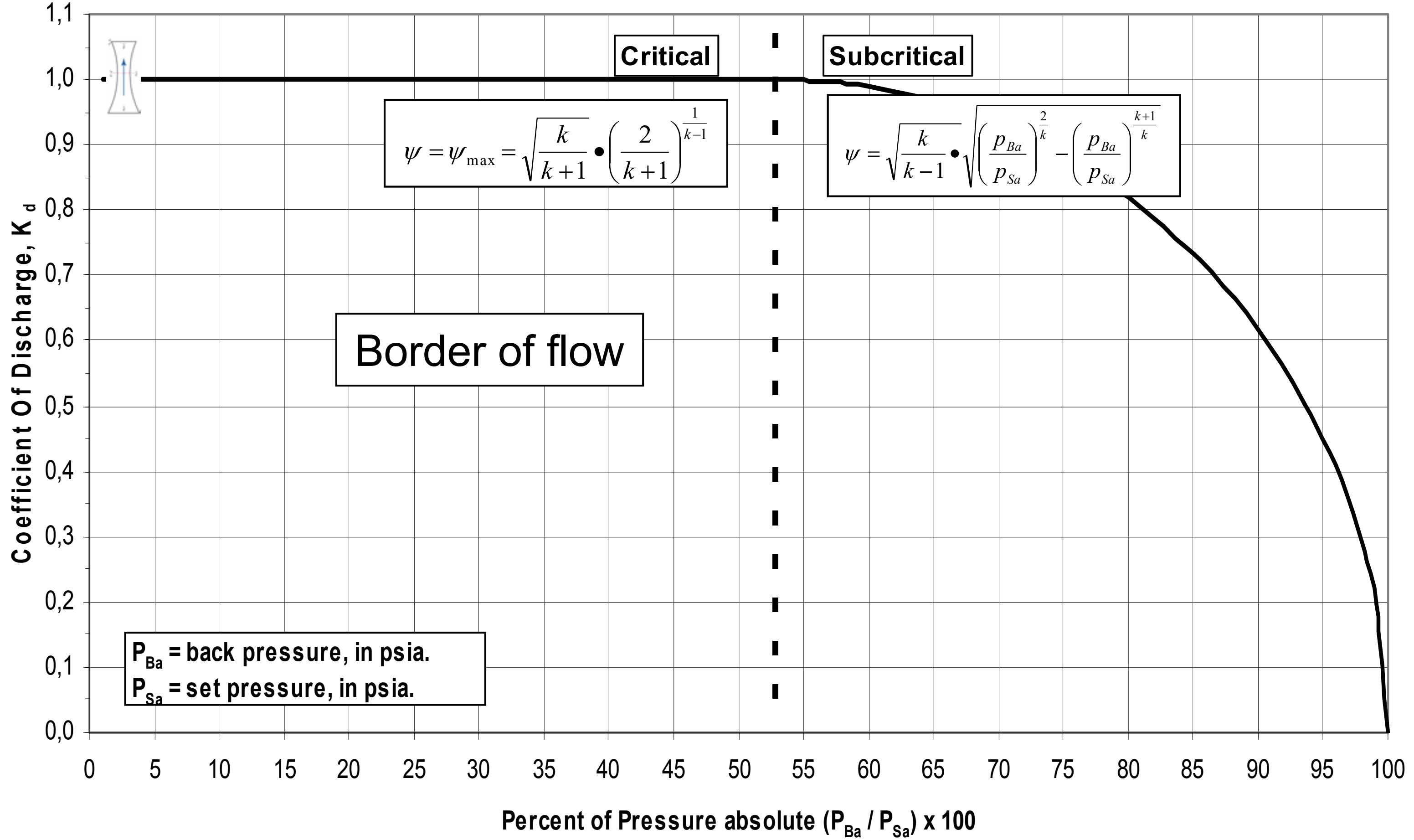
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

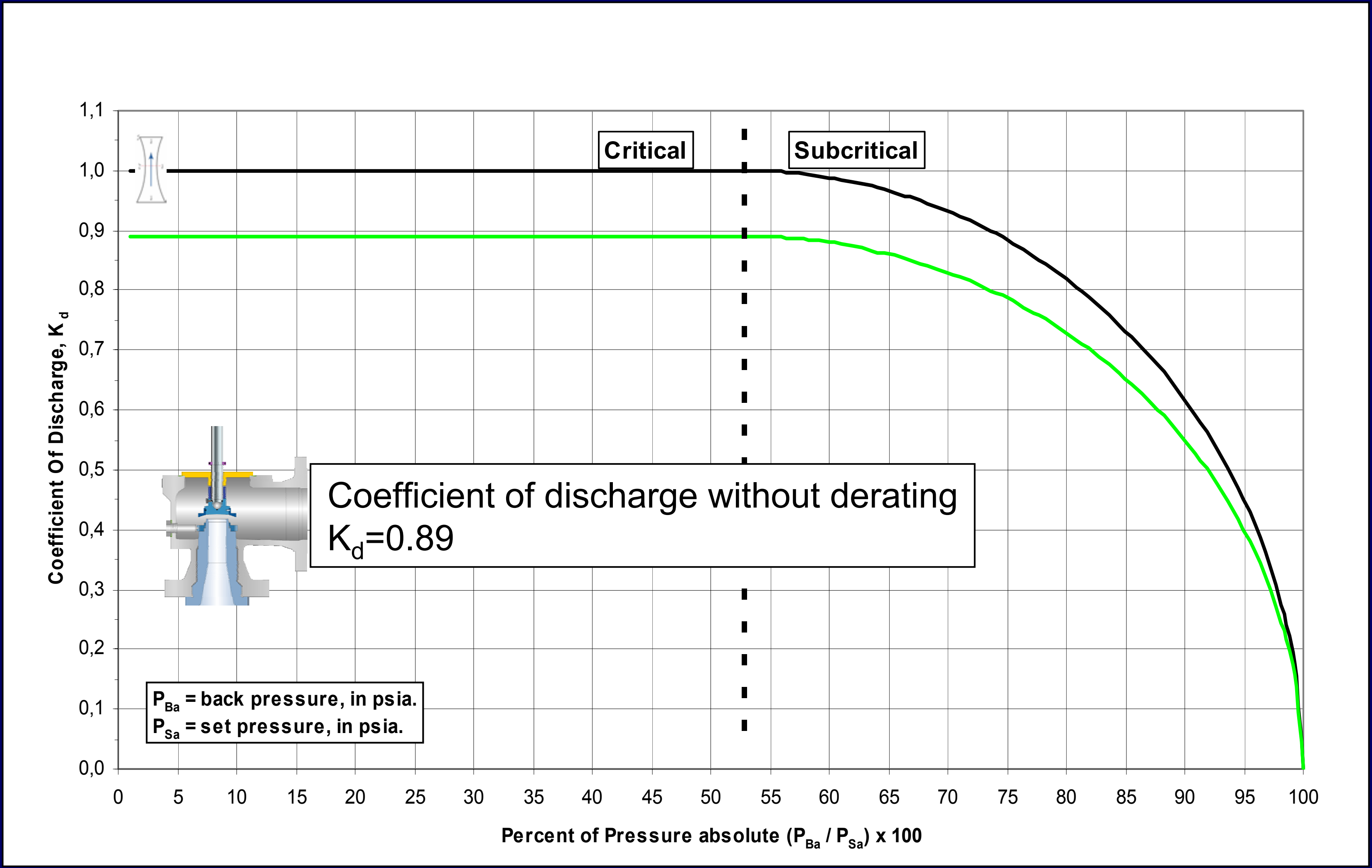
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

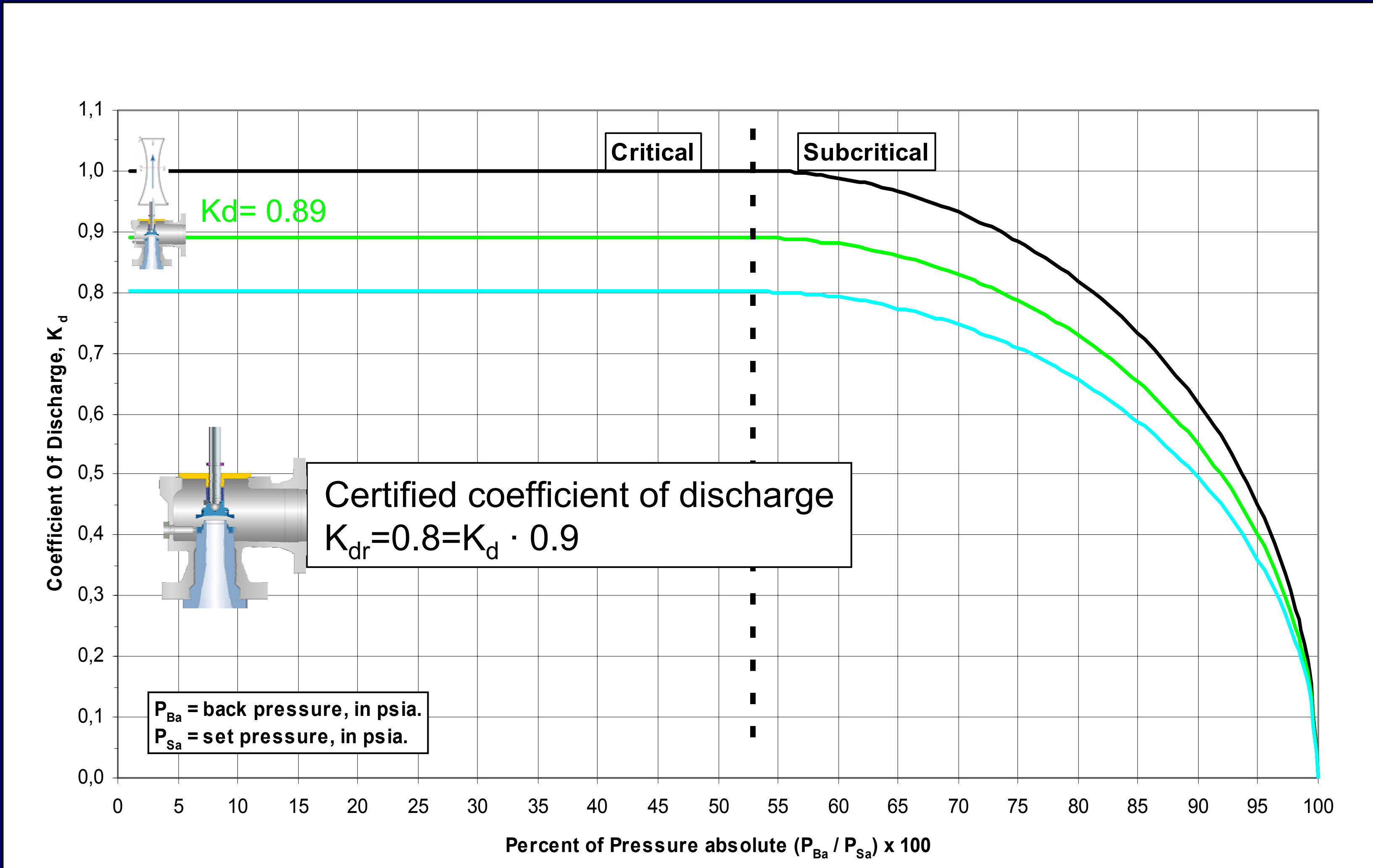
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

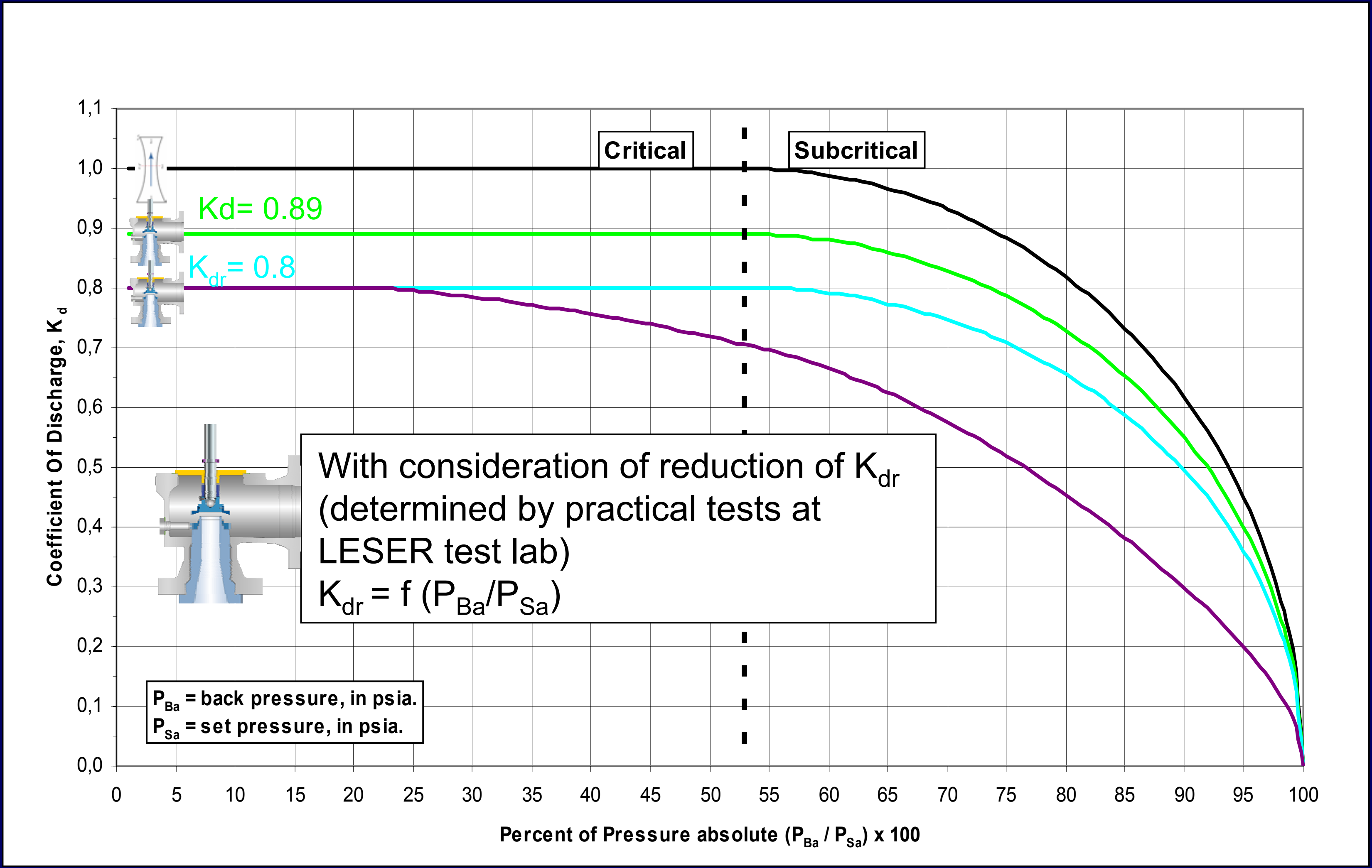
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

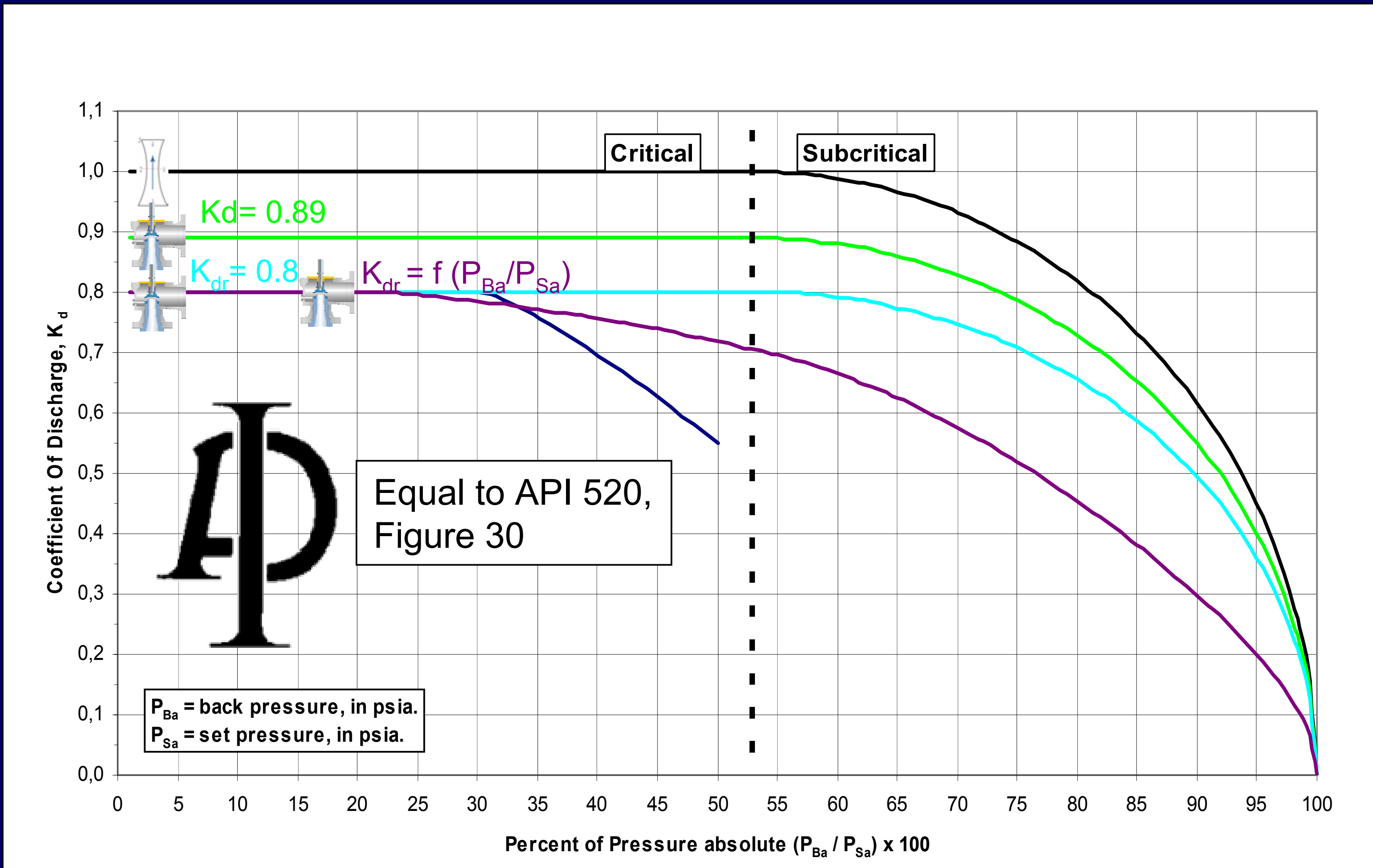
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

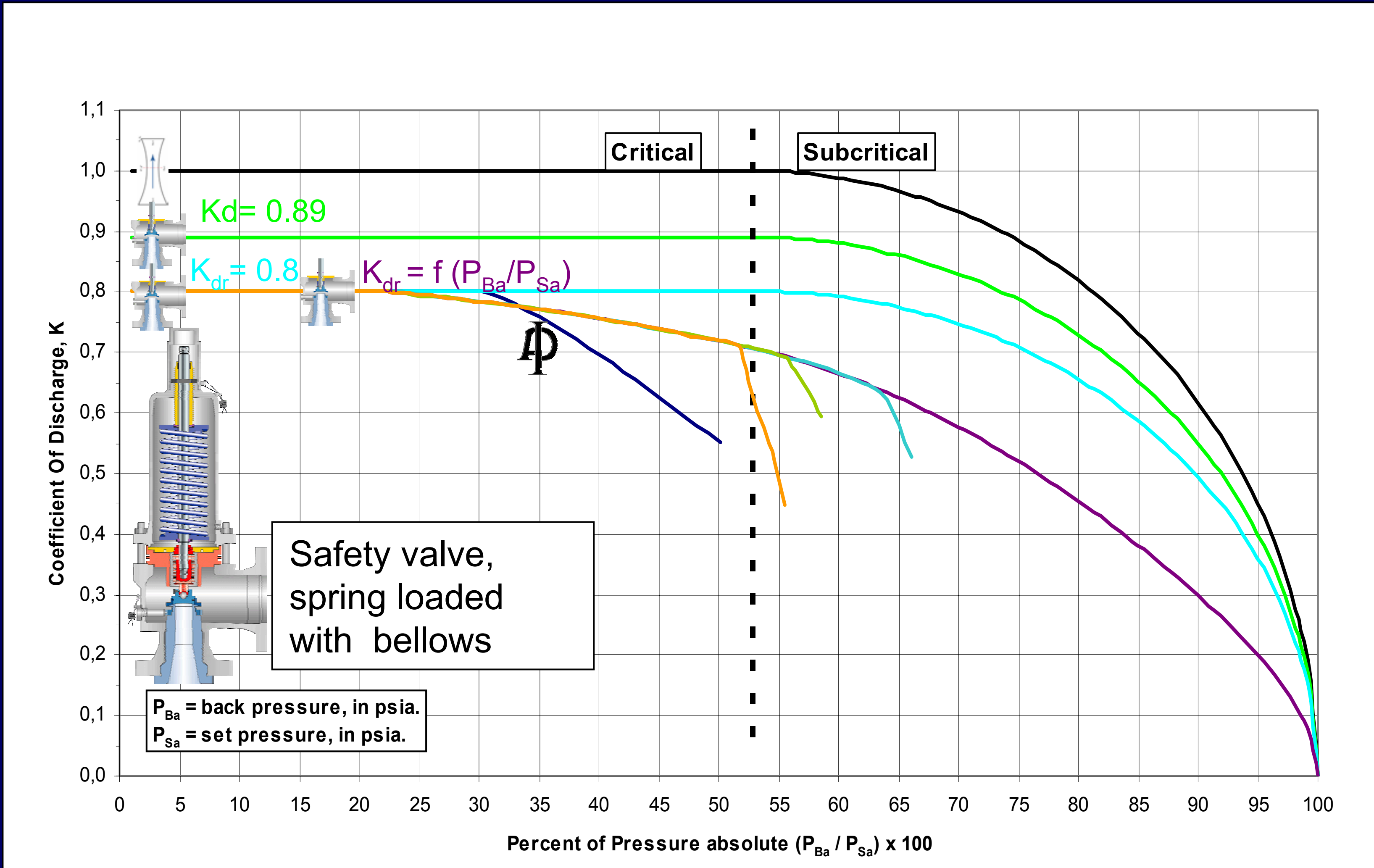
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

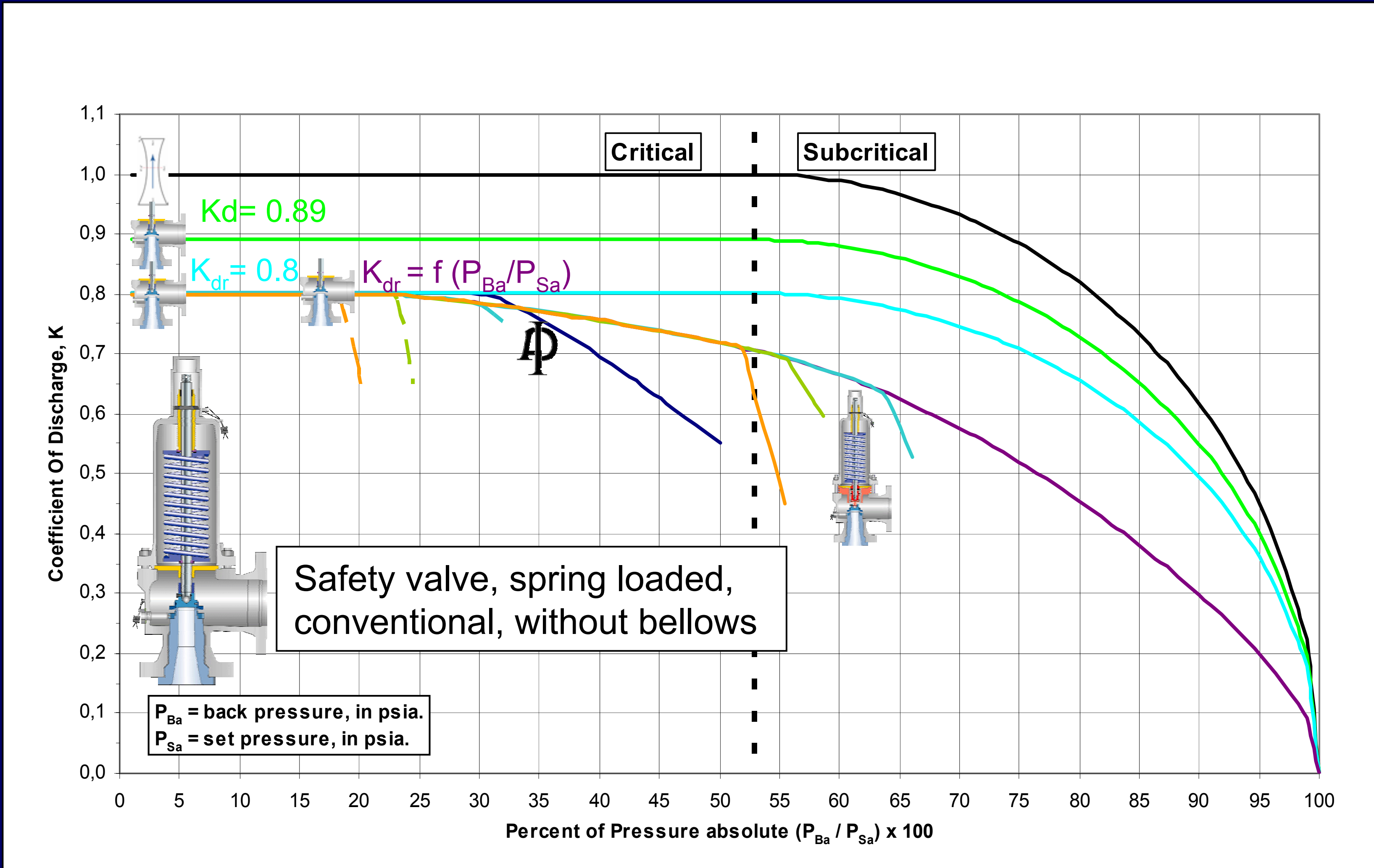
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

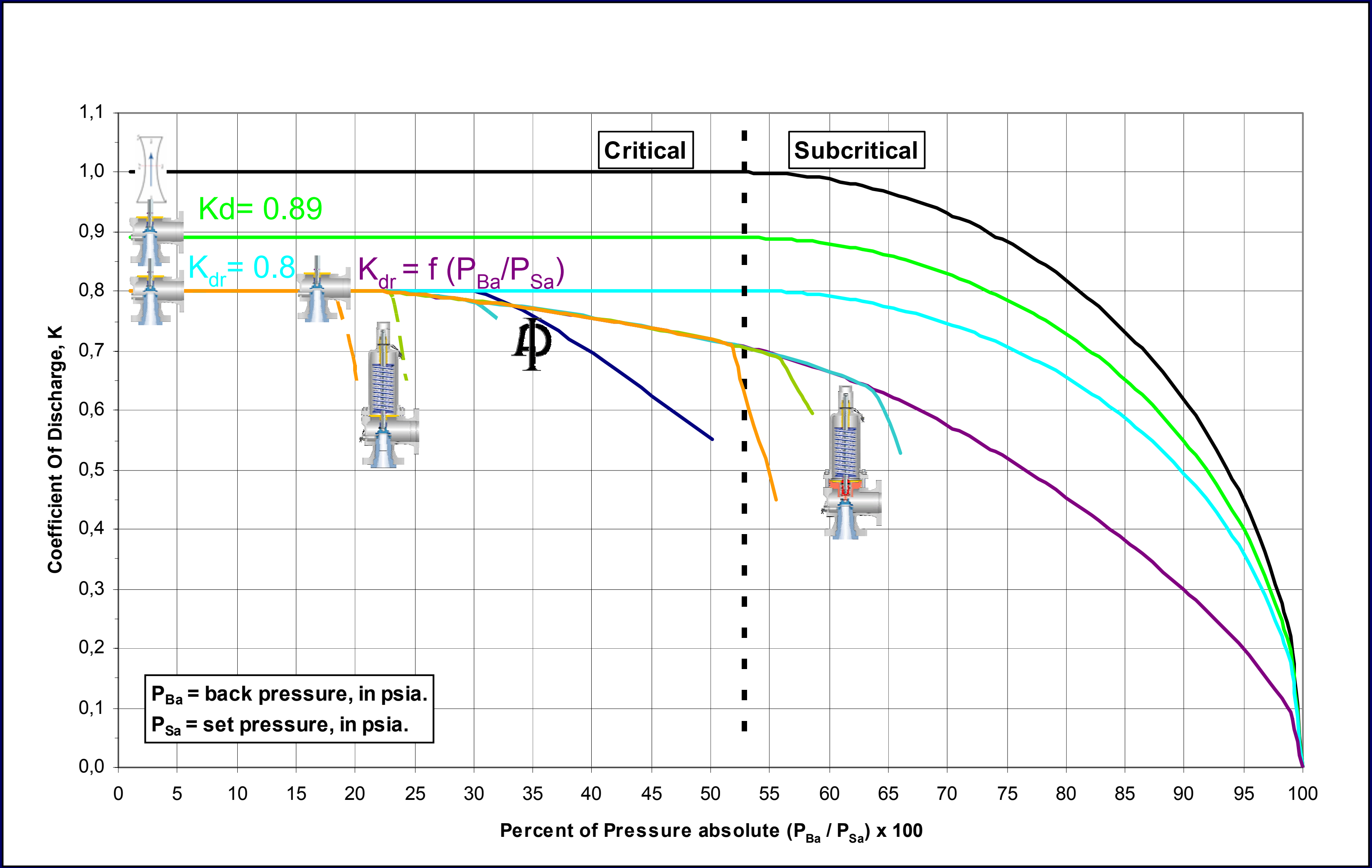
$\kappa = 1.4$



Coefficient Of Discharge, K

using absolute pressures

κ = 1.4



Performance of 526 3L4, conventional, without bellows, with back pressure

LESER

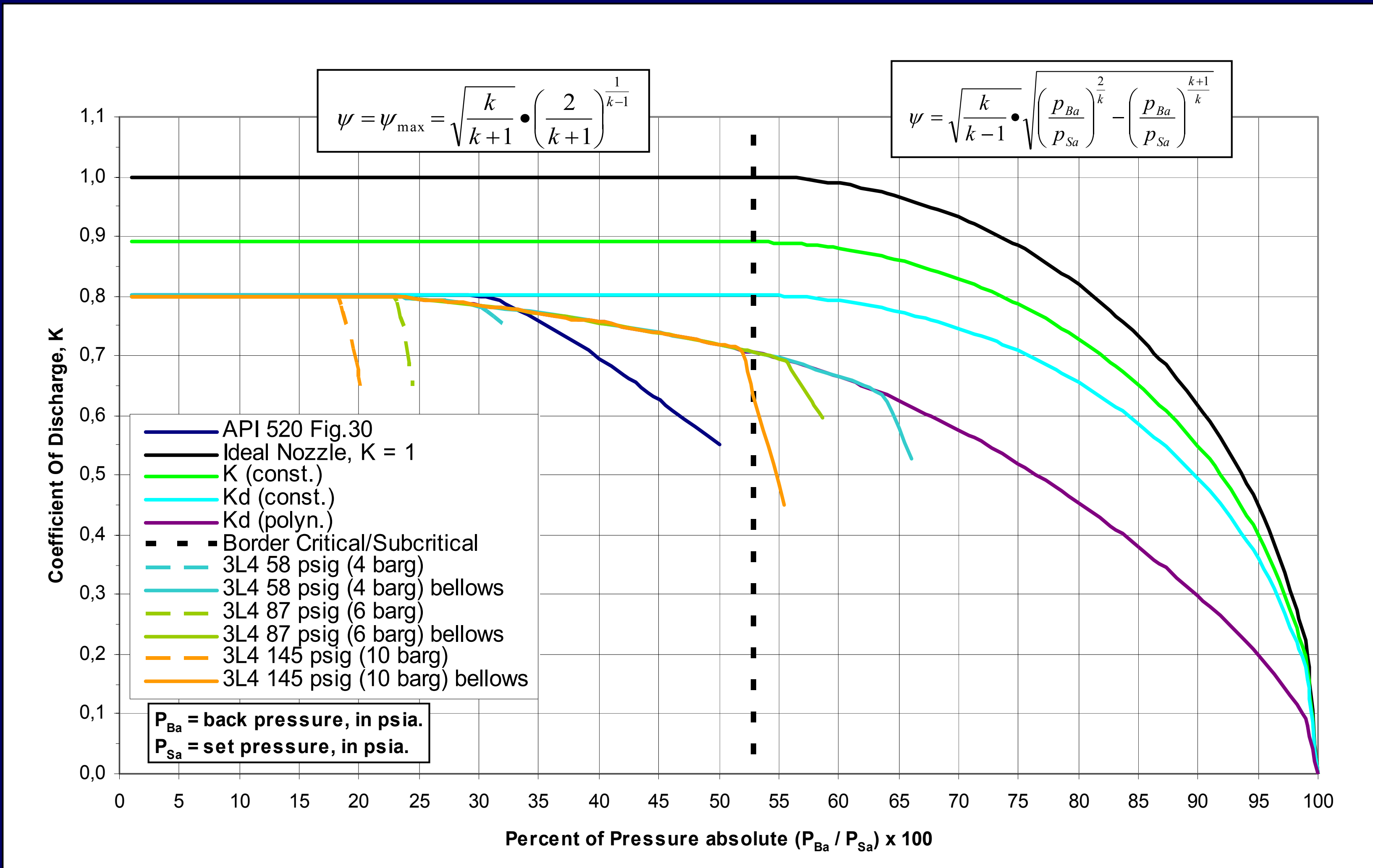
The Safety Valve

The Safety Valve

Coefficient Of Discharge, K

using absolute pressures

κ = 1.4



Contents

LESER

Conclusions

Back pressure fluids should be shown depending on the absolute pressure ratio of back pressure and set pressure

Conclusions

Consideration of reduced coefficient of discharge in the back pressure correction in API Standard

Conclusions

Introduction of manufacturer statements about operation of safety valves in case of back pressure

LESER

Conclusions

Special valve options keeping the valve stable even in situations with reduced valve lift

Performance of 526 3L4, conventional, without bellows, with O-Ring-Damper

The Safety Valve

Back pressure
DIERS Users Group Meeting
Charleston, March 21 - 23, 2005

Pressure Safety Valve Thrust Forces for Compressible Gas or Vapor Flow

Robert D'Alessandro, P.E.
Degussa Corporation
Mobile, Alabama
robert.dalessandro@degussa.com

ABSTRACT

When a pressure safety valve actuates, thrust forces are developed. These thrust forces must be considered for the proper installation of the relieving device. For gas or vapor compressible flow, Simpson developed thrust force plots that can be utilized to calculate the magnitude of the thrust force. Although these charts are available in the open literature, the assumptions behind the charts and the derivation of the underlying equations are not. This paper addresses this issue by re-deriving the equations necessary for reproducing the charts. All fundamental assumptions are stated and all flow models are explained. The limitations of the equations are clearly stated. Numerical examples are given to illustrate the use of the equations and the corresponding Simpson Charts. The examples are also used to compare calculated results from other methods. In addition, several important modifications to the original charts are noted. The thrust force equations for a relief valve operating with a subcritical nozzle and a subcritical outlet, which are absent from the original Simpson Charts, are included here. The extent of the region where the relief valve operates with a critical flow nozzle and a subcritical flow outlet is modified to account for backpressure effects. Finally, the concept of a minimum relief valve outlet to nozzle area ratio is introduced.

1. INTRODUCTION

When a pressure safety valve actuates, thrust forces are developed. If the relief valve is mounted directly to the vessel, there is no net force at the relief valve nozzle, since the relief valve disk acts as a thrust plate that balances the momentum of the flowing fluid. In other words, the vessel restraints and weight are normally sufficient to balance the forces that are developed at the relief valve nozzle. However, the balance of forces at the relief valve exit may not be zero. The thrust force at the relief valve exit acts in a direction opposite to the direction of the mass flow discharging from the relief valve. In order to properly secure relief valves and prevent damage to connected components such as pipes and equipment nozzles, it is necessary to estimate the magnitude of the thrust forces that are developed. Once known, these reaction forces can be balanced by the installation of appropriate restraints at key locations. For the typical emergency relief system, the restraints necessary to balance the relief valve thrust forces are usually moderate and the costs associated with these restraints are typically low. However, large pressure safety valves operating at high pressure can produce substantial thrust forces.

One method for calculating the thrust forces generated when a pressure safety valve actuates was described by Simpson [1]. The Simpson method is typically presented in the form of charts where the thrust force parameter is plotted against the backpressure ratio. The ratio of outlet area to relief valve nozzle area is used as a parameter. The charts can only be used for compressible gas flow in relief valves. The effect of the fluid properties is incorporated via the heat capacity ratio of the gas under consideration. Each value of the heat capacity ratio yields a new Simpson Chart. The Simpson Chart for a gas with a heat capacity ratio of 1.40 is shown in Figure 1.

For this diagram, the stagnation pressure is the vessel static relieving pressure in absolute pressure units while the backpressure is the static pressure immediately downstream of the relief valve exit plane in absolute pressure units as well. The parameter k is the heat capacity ratio for the gas under consideration and is evaluated at the vessel stagnation conditions. A_n is the cross-sectional area of the relief valve nozzle. A_e is the cross-sectional area of the relief valve outlet. The thrust force parameter is a dimensionless ratio defined by Equation (01), where T_R is the thrust force and P_0 is the vessel stagnation pressure.

$$Thrust\ Force\ Parameter = \frac{T_R}{P_0 A_n} \tag{01}$$

The Simpson Chart can be broken into three regions as shown in Figure 1. Each region represents a different flow regime within the relief valve geometry. For Region A, the flow is choked at the relief valve nozzle and at the relief valve exit plane. For Region B, the flow is choked at the relief valve nozzle, but is unchoked at the relief valve exit plane. In Region C, the flow is unchoked throughout the relief valve geometry.

Region A illustrates a very important flow regime. In this region, the flow is choked at the relief valve exit plane. This type of relief valve flow pattern is often referred to as "body-bowl" choking. It must be considered carefully when evaluating the effect of backpressure on the performance of a relief valve.

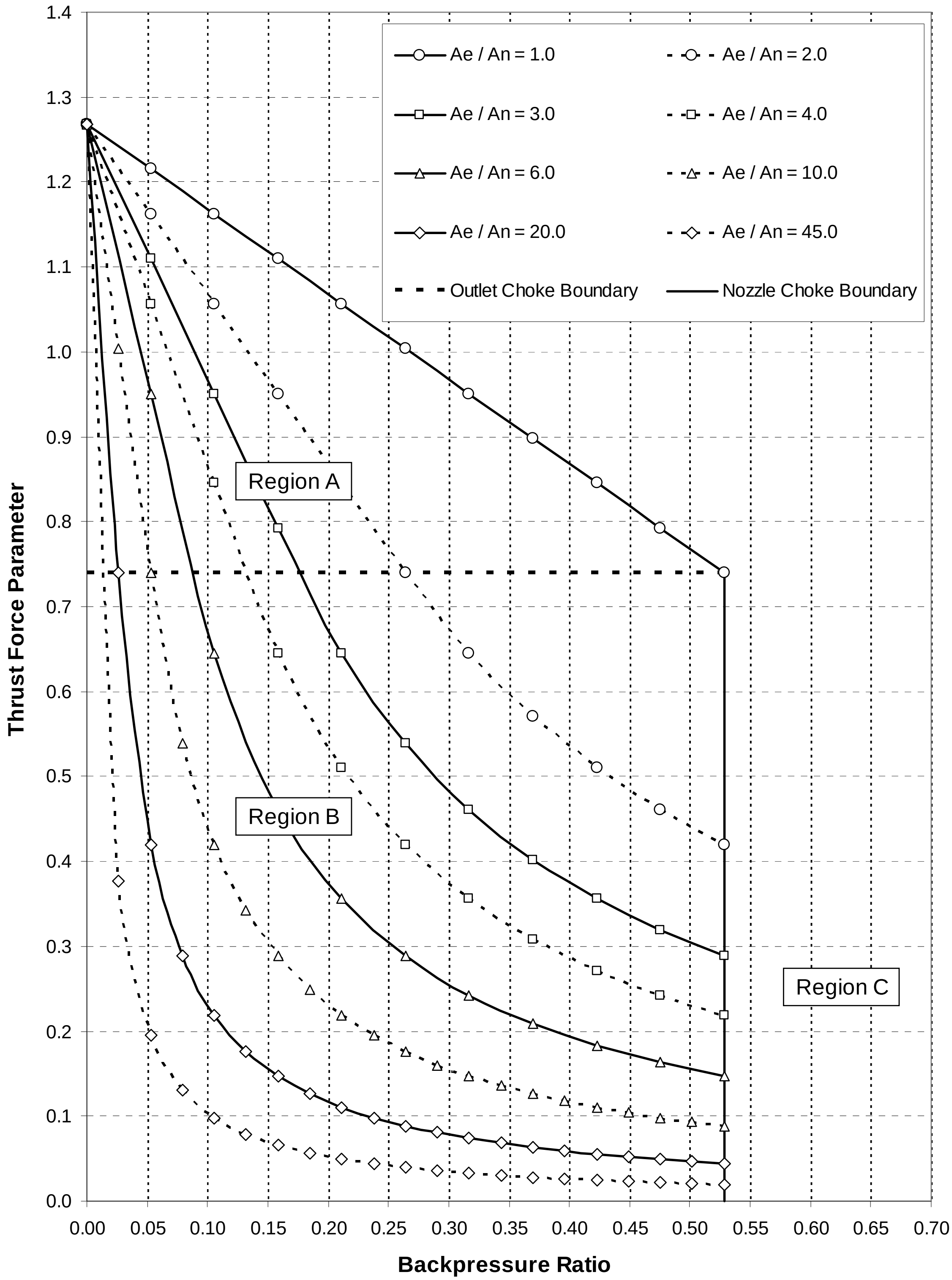


Figure 1. Simpson Chart - Thrust Force for a Compressible Gas or Vapor with a Heat Capacity Ratio of 1.40

Obviously, most relief valve applications have operating points that lie toward the left side of this diagram. For conventional spring loaded full lift pressure safety valves, proper functioning of the relief valve can only be assured if the built-up backpressure is limited to 10% of the relief valve set pressure on a gauge basis at 10% overpressure. For balanced spring loaded full lift pressure safety valves, backpressure limitations can be extended to 30% to 50% of the relief valve set pressure on a gauge basis. Therefore, on the Simpson Charts, most relief valve operating points will lie in either Region A or Region B. Few traditional relief valve applications operate at backpressures greater than 50% on a gauge pressure basis. Pilot operated relief valves can operate at extremely high backpressure ratios. At times operating points for pilot operated relief valves can lie in Region C.

It is obvious from Figure 1 that the thrust force on a relief valve decreases dramatically when the cross-sectional area of the outlet is significantly larger than the cross-sectional area of the relief valve nozzle. A summary of typical relief valve geometries (Appendix A) indicates that the ratio of the cross-sectional area of the outlet to the cross-sectional area of the nozzle is usually between three and forty. In most cases, this ratio is greater than six. Super capacity valves (orifice designations greater than T) typically have outlet to nozzle area ratios less than three. Generally speaking, smaller relief valves tend to have larger ratios and larger relief valves tend to have smaller ratios.

There are many references to the Simpson Charts in the literature. However, the actual derivations of the equations that generate the charts do not appear in any of these open literature references. The original work was documented in the form of an internal Union Carbide memorandum (October 1969) written by Simpson [1]. This reference was not published in the open literature and is therefore unavailable to most pressure relief practitioners.

Huff [2] references the Simpson Charts and actually includes one of the charts (k =1.4) in his paper. However, the underlying equations are not shown.

The DIERS Project Manual [3] provides reproductions of all three original charts (pages 338 to 340). Equations for Region A are also given, but not for Region B. The erratum to the DIERS Project Manual [4] provides a set of equations for both Region A and Region B. However, this set of equations does not yield thrust force parameters that correspond to the values obtained from the charts.

The CCPS Guidelines for Pressure Relief and Effluent Handling Systems [5] (pages 220 to 222) also includes the charts. However, no underlying equations are given. The charts in the CCPS book do appear to be original plots, and therefore, were probably generated from the basic equations.

The Workbook for Chemical Reactor Relief System Sizing [6], published by the Health & Safety Executive (HSE), gives an equation for the thrust force on a pressure safety valve in compressible gas service. This publication references the DIERS Project Manual as the source of the equation, and therefore, it is presumed that the HSE equation corresponds to the Simpson Charts. However, if the equations from these two references are compared, a discrepancy is noted. In addition, a numerical calculation using the HSE equation does not reproduce the thrust force parameter obtained from the corresponding Simpson Chart.

The general utility of the Simpson Charts is quite obvious. Thrust forces are easily calculated without the need for determining either the capacity of the relief valve or the internal pressure profile. The only requirements for the calculation are the stagnation pressure, properties of the gas at the stagnation condition, the relief valve geometry in terms of nozzle area and outlet area, and the backpressure.

It is quite easy to use the charts to calculate relief valve thrust forces. However, if computer calculations are desired, the charts become a burden. Therefore, to calculate the thrust force on a relief valve using algebraic equations, it becomes necessary to derive the underlying equations that form the basis for the thrust force charts.

This paper addresses this issue by deriving the equations that represent the thrust force charts. All fundamental assumptions are stated and all flow models are explained. The limitations of the equations are clearly stated. Numerical examples are given to illustrate the use of the equations and the corresponding thrust force charts. The examples are also used to compare calculated results from other methods. In addition, several important modifications to the original charts are noted. The thrust force equations for Region C, which are absent from the original Simpson Charts, are included here. The extent of Region B is modified to account for backpressure when the exit of the relief valve is unchoked. Finally, the concept of a minimum outlet to nozzle area ratio is introduced.

2. BASIC FORCE BALANCE FOR A PRESSURE SAFETY VALVE

The basic steady state force balance for a venting relief valve is given by Equation (02).

$$T_R = Wu_e + A_e\left(P_e - P_b\right) \qquad (02)$$

The following nomenclature and units are used.

T_R is the thrust force [N]
W is the fluid mass flow [kg/s]
u_e is the fluid velocity at the relief valve outlet [m/s]
A_e is the cross-sectional area of the relief valve outlet [m^2]
P_e is the pressure at the relief valve outlet [Pa]
P_b is the backpressure at the relief valve outlet [Pa]

The first term on the right side of this equation represents the thrust force due to fluid momentum. The second term represents the thrust force due to the potential pressure discontinuity at the relief valve outlet.

Dividing Equation (02) by the stagnation pressure, P_0, and relief valve nozzle cross-sectional area, A_n, yields the dimensionless steady state thrust force equation.

$$\frac{T_R}{P_0 A_n} = \frac{Wu_e}{P_0 A_n} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\} \qquad (03)$$

Simpson was able to evaluate the above expression for a compressible perfect gas. The results of this evaluation are the Simpson Charts. In each chart, the thrust force parameter is plotted against the backpressure ratio (P_b / P_0). The ratio of the relief valve outlet area to relief valve nozzle area (A_e / A_n) is used as a parameter.

The mass flow and exit velocity can be eliminated from Equation (03) by utilizing the mass flux at the relief valve outlet.

$$W = G_e A_e \tag{04}$$

$$u_e = \frac{G_e}{\rho_e} \tag{05}$$

Substitution of Equation (04) and Equation (05) into Equation (03) yields the working thrust force equation:

$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{G_e^2}{P_0 \rho_e} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\} \tag{06}$$

3. MODEL

In order to evaluate the thrust force equation, it is necessary to solve the flow and pressure balances from the stagnation condition through the relief valve nozzle into the relief valve body and finally through the relief valve exit plane.

The important variables and parameters necessary to solve these balances are shown in Figure 2.

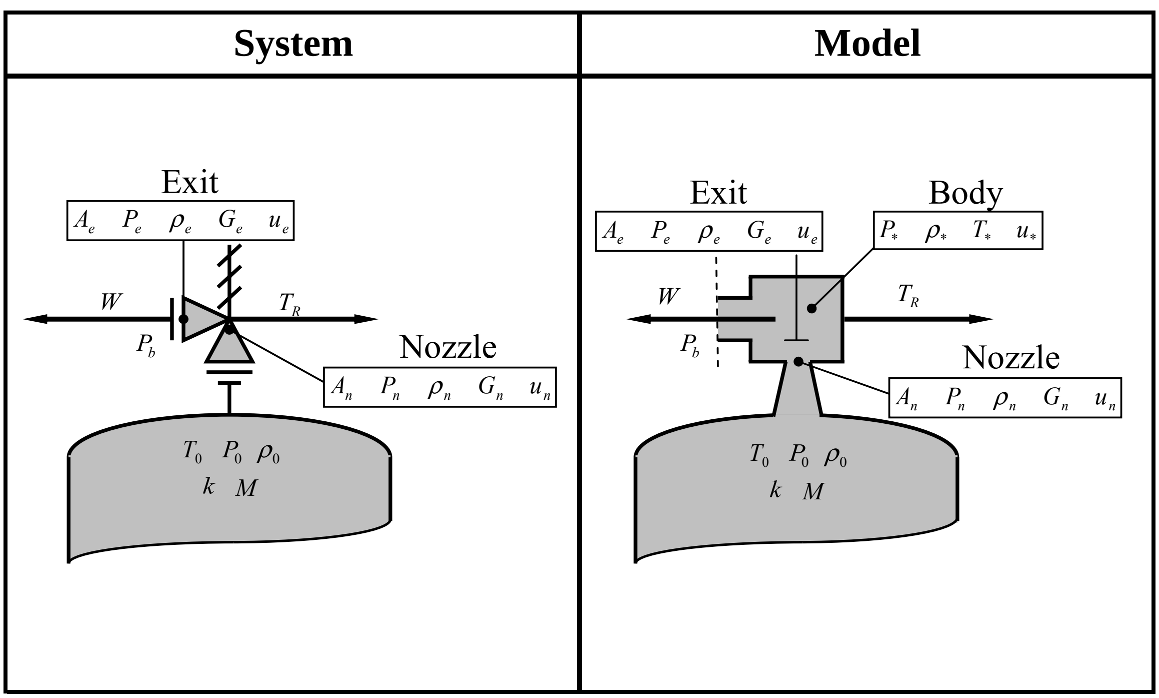


Figure 2. System Sketch and Model Sketch

The following nomenclature for the variables and parameters shown in Figure 2 is utilized.

T_R Thrust force [N]
A Cross-sectional area [m^2]
W Mass flow [kg/s]
u Velocity [m/s]
P Pressure [Pa]
T Temperature [K]
ρ Density of the gas [kg/m^3]
G Mass flux [kg/m^2-s]
M Molecular weight of the gas [kg/kmol]
k Specific heat ratio of the gas [unitless]

The subscript "e" denotes the relief valve exit plane. The subscript "n" denotes the relief valve nozzle. The subscript "$*$" denotes the relief valve body. The subscript "b" corresponds to the conditions immediately downstream of the relief valve exit plane.

Three cases are considered. The first case assumes the flow through the relief valve is choked at the relief valve nozzle and the flow at the relief valve exit plane is also choked. For the second case, the flow through the relief valve nozzle is still choked, but the flow at the relief valve exit plane remains unchoked. The third case considers unchoked flow throughout the relief valve geometry.

The flow process to bring the fluid from the stagnation condition to the nozzle throat is assumed to be isentropic, namely adiabatic and reversible. The flow from the nozzle throat to the relief valve body is assumed to be adiabatic, but irreversible. Finally, the flow through the outlet nozzle is also assumed to be isentropic. All flow patterns are assumed to be frictionless and at steady state. In essence, the relief valve is modeled as two ideal frictionless nozzles that are fed from an infinite reservoir and separated by a plenum.

The body of a relief valve has a very complicated geometry. In addition, there are few, if any, geometric similarities between the bodies of various relief valve sizes and relief valve manufacturers. As a result of this complex geometry, the flow pattern of the relieving fluid in the relief valve body is also very complicated. To assume that the body and outlet act as a frictionless ideal nozzle is, in many respects, a drastic simplification. However, in making this assumption the mathematics becomes tractable and reasonable answers are obtained.

4. RELATIONSHIP BETWEEN CRITICAL MASS FLUX AND CRITICAL PRESSURE

In evaluating the mass and pressure balances through the relief device, it is useful to have a relationship between the critical mass flux and the local flow conditions. When the pressure of the flowing gas is high enough and the flow geometry experiences a change in the cross-sectional area, critical mass flow conditions may result. The desired relationship between the critical mass flux and the local flowing conditions can be derived from basic thermodynamic principles.

Using energy conservation, the mass flux through an ideal nozzle can be related to the enthalpy change by the following equation.

$$G = \frac{1}{v}\sqrt{2(h_0 - h)} \tag{07}$$

where v is the specific volume of the flowing fluid, h_0 is the enthalpy of the fluid at the stagnation condition and h is the enthalpy of the fluid under flowing conditions.

A graph of this equation results in a maximum value for the mass flux. This maximized mass flux is usually referred to as the critical mass flux (G_c) while the corresponding pressure is typically referred to as the critical pressure (P_c). To seek this maximum, the general expression for the mass flux, Equation (07), can be differentiated with respect to pressure and the subsequent derivative can be set to zero.

From thermodynamics, a change in enthalpy can be related to a change in entropy and a change in pressure as follows.

$$dh = TdS + vdP \tag{08}$$

For an isentropic process such as flow through an ideal frictionless nozzle, $dS = 0$. Therefore,

$$\frac{dh}{dP} = v \tag{09}$$

By substituting Equation (09) into the maximizing expression obtained from Equation (07), the following relationship for the critical mass flux is obtained.

$$G_c = \left\{-\frac{dP}{dv}\right\}_{P_c}^{\frac{1}{2}} \tag{10}$$

Equation (10) is only valid if the derivative is evaluated at the critical pressure. Now, this generalized expression for the critical mass flux can be applied to the equation for a perfect gas undergoing an isentropic flow process.

$$Pv^k = P_0 v_0^k \tag{11}$$

When this is done, the desired relationship between the critical mass flux and the local pressure condition is obtained.

$$G_c^2 = \frac{kP_c}{v_c} = k\rho_c P_c \quad or \quad \frac{G_c^2}{\rho_c} = kP_c \tag{12}$$

5. ADIABATIC, IRREVERSIBLE FLOW OF A PERFECT COMPRESSIBLE GAS

To flow from the relief valve nozzle into the relief valve body, the fluid experiences a discontinuity in pressure if the flow through the relief valve nozzle is choked. Once in the body of the relief valve, the fluid velocity is assumed to be zero. The energy balance shown in Equation (13) for the relief valve nozzle can describe this flow process. Here h_* is the enthalpy of the fluid at the conditions in the body of the relief valve and h_n is the enthalpy of the fluid at the conditions of the nozzle throat.

$$h_* = h_n + \frac{u_n^2}{2} \tag{13}$$

However, since the fluid is a perfect gas, the enthalpy difference between the nozzle throat and the relief valve body is only a function of the fluid temperature.

$$h_* - h_n = \int_{T_n}^{T_*} C_p dT = \frac{R}{M}\left\{\frac{k}{k-1}\right\}\int_{T_n}^{T_*} dT = \left\{\frac{k}{k-1}\right\}\frac{R}{M}\left(T_* - T_n\right) \tag{14}$$

Utilizing the perfect gas law, this equation can be rewritten as.

$$h_* - h_n = \frac{k}{k-1}\left\{\frac{P_*}{\rho_*} - \frac{P_n}{\rho_n}\right\} \tag{15}$$

The velocity at the nozzle throat can be expressed in terms of the nozzle mass flux.

$$u_n = \frac{G_n}{\rho_n} \tag{16}$$

When Equation (13), Equation (15), and Equation (16) are combined, the following expression is obtained.

$$\frac{G_n^2}{\rho_n^2} = \frac{2k}{k-1}\left\{\frac{P_*}{\rho_*} - \frac{P_n}{\rho_n}\right\} \tag{17}$$

If the flow at the nozzle throat is choked, the local choking expression, Equation (12), can also be used to relate the nozzle mass flux to the nozzle choking pressure yielding Equation (18). Combining Equation (17) and Equation (18) yields an expression that relates the nozzle choking pressure to the pressure in the relief valve body.

$$G_n^2 = k\rho_n P_n \tag{18}$$

$$\frac{P_*}{\rho_*} = \left\{\frac{k+1}{2}\right\}\frac{P_n}{\rho_n} \tag{19}$$

6. CASE 1 – REGION A: CHOKING AT THE RELIEF VALVE NOZZLE AND RELIEF VALVE EXIT PLANE

For Case 1, the entire flow process can be described by eight independent equations, each with its own origin. These eight independent nonlinear algebraic equations are summarized in Table 1.

Table 1. Equation Set for Case 1 – Region A

$G_e A_e = G_n A_n$	(20)
$\dfrac{P_0}{\rho_0^k} = \dfrac{P_n}{\rho_n^k}$	(21)
$G_n^2 = k\rho_n P_n$	(22)
$\dfrac{P_n}{P_0} = \left\{\dfrac{2}{k+1}\right\}^{\frac{k}{k-1}}$	(23)
$\dfrac{P_*}{\rho_*} = \left\{\dfrac{k+1}{2}\right\}\dfrac{P_n}{\rho_n}$	(24)
$\dfrac{P_*}{\rho_*^k} = \dfrac{P_e}{\rho_e^k}$	(25)
$G_e^2 = k\rho_e P_e$	(26)
$\dfrac{P_e}{P_*} = \left\{\dfrac{2}{k+1}\right\}^{\frac{k}{k-1}}$	(27)

The continuity of mass flow, Equation (20), provides a relationship between the pressure and fluid density at the nozzle throat and the pressure and fluid density at the relief valve exit plane. Equation (21) represents the isentropic flow of a perfect gas from the vessel stagnation condition to the relief valve nozzle. Choking at the relief valve nozzle yields Equation (22) and Equation (23). Adiabatic flow of a perfect gas from the relief valve nozzle to the relief valve body yields Equation (24). Isentropic flow of a perfect gas from the relief valve body to the relief valve exit yields Equation (25). Finally, choking at the relief valve exit yields Equation (26) and Equation (27).

In most problems of interest, the following parameters are known.

$$P_0 \quad P_b \quad A_e \quad A_n \quad k$$

The eight independent expressions [Equations (20) through (27)] summarized in Table 1 must be solved simultaneously for the following eight unknowns in terms of the five known parameters.

$$P_n \quad P_* \quad P_e \quad \rho_n \quad \rho_* \quad \rho_e \quad G_n \quad G_e$$

In doing so, the following expression can be derived.

$$\frac{A_e}{A_n}\frac{P_e}{P_0}=\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \tag{28}$$

Now we return to the thrust force expression, Equation (06), and note that the exit pressure can replace the exit mass flux and exit density. When this is done, and the resulting expression is simplified, the following relationship is obtained.

$$\frac{T_R}{P_0 A_n}=\frac{A_e}{A_n}\frac{P_e}{P_0}(k+1)-\frac{A_e}{A_n}\frac{P_b}{P_0} \tag{29}$$

Combining Equation (28) and Equation (29) and performing some algebra leads to the desired expression for the thrust force parameter.

$$\frac{T_R}{P_0 A_n}=(k+1)\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}-\frac{A_e}{A_n}\frac{P_b}{P_0}=2\left\{\frac{2}{k+1}\right\}^{\frac{1}{k-1}}-\frac{A_e}{A_n}\frac{P_b}{P_0} \tag{30}$$

Equation (30) is the Simpson equation for the thrust force on a relief valve when the flow through the relief valve is choked at both the relief valve nozzle and the relief valve exit. This equation yields the lines shown in Region A of Figure 1.

If the Thrust Force Parameter is identified as the dependent variable, and the backpressure ratio is identified as the independent variable, this expression is the equation of a straight line with the following characteristics.

$$Slope=-\frac{A_e}{A_n} \qquad Intercept=2\left\{\frac{2}{k+1}\right\}^{\frac{1}{k-1}} \tag{31}$$

From this equation one easily determines that the maximum thrust occurs when the backpressure is zero. Under these circumstances, the Thrust Force Parameter is given by:

$$\left\{\frac{T_R}{P_0 A_n}\right\}_{\max}=2\left\{\frac{2}{k+1}\right\}^{\frac{1}{k-1}} \quad when\ P_b=0 \tag{32}$$

For k =1.01, the maximum Thrust Force Parameter is 1.215. For k = 1.40, the maximum Thrust Force Parameter is 1.268. Finally, for k = 1.80, the maximum Thrust Force Parameter is 1.313. Generally speaking, larger heat capacity ratios result in larger relief valve thrust forces.

7. CASE 2 – REGION B: CHOKING AT THE RELIEF VALVE NOZZLE ONLY

Again, eight independent expressions are developed. As before, the eight expressions are based on the various flow processes that take the fluid from the vessel stagnation conditions to the conditions at the relief valve exit. These eight independent nonlinear algebraic equations are summarized in Table 2.

Table 2. Equation Set for Case 2 – Region B

$G_e A_e = G_n A_n$	(33)
$\dfrac{P_0}{\rho_0^k} = \dfrac{P_n}{\rho_n^k}$	(34)
$G_n^2 = k\rho_n P_n$	(35)
$\dfrac{P_n}{P_0} = \left\{\dfrac{2}{k+1}\right\}^{\frac{k}{k-1}}$	(36)
$\dfrac{P_n}{\rho_n} = \left\{\dfrac{2}{k+1}\right\}\dfrac{P_*}{\rho_*}$	(37)
$\dfrac{P_*}{\rho_*^k} = \dfrac{P_e}{\rho_e^k}$	(38)
$G_e^2 = P_* \rho_* \left\{\dfrac{2k}{k-1}\right\}\left\{\left(\dfrac{P_e}{P_*}\right)^{\frac{2}{k}} - \left(\dfrac{P_e}{P_*}\right)^{\frac{k+1}{k}}\right\}$	(39)
$P_e = P_b$	(40)

The continuity of mass flow, Equation (33), provides a relationship between the mass flux at the relief valve nozzle and the mass flux at the relief valve exit. Equation (34) represents the isentropic flow of a perfect gas from the vessel stagnation condition to the relief valve nozzle. The local choking condition at the relief valve nozzle provides Equation (35). Choking at the relief valve nozzle yields Equation (36). Adiabatic flow of a perfect gas from the relief valve nozzle to the relief valve body yields Equation (37). Isentropic flow of a perfect gas from the relief valve body to the relief valve exit yields Equation (38). Next, Equation (39) relates the mass flux at the relief valve exit to the fluid conditions in the relief valve body. This is the traditional mass flux equation for the subcritical flow of a perfect gas through a nozzle.

Since the flow at the relief valve exit is unchoked, Equation (40) sets the exit pressure equal to the backpressure. Equation (40) can be used to simplify the basic dimensionless thrust force relationship, Equation (06). When the flow at the relief valve exit is unchoked, the force contribution due to a pressure discontinuity becomes zero. The second term on the right side of Equation (06) is eliminated. The entire thrust force is caused by flow alone.

As before, in most problems of interest, the following parameters are known.

$P_0 \quad P_b \quad A_e \quad A_n \quad k$

The eight independent expressions [Equations (33) through (40)] shown in Table 2 must be solved simultaneously for the following eight unknowns.

$P_n \quad P_* \quad P_e \quad \rho_n \quad \rho_* \quad \rho_e \quad G_n \quad G_e$

In doing so, the following two expressions are derived.

$$\frac{G_e^2}{P_0 \rho_e} = k \frac{A_n^2}{A_e^2} \frac{P_0}{P_e} \left\{ \frac{P_e}{P_*} \right\}^{\frac{k-1}{k}} \left\{ \frac{2}{k+1} \right\}^{\frac{k+1}{k-1}} \tag{41}$$

$$\left\{ \frac{P_e}{P_*} \right\}^{\frac{k-1}{k}} = \left[\frac{1}{2} + \frac{1}{2} \frac{P_0}{P_e} \left\{ \frac{P_e^2}{P_0^2} + 2(k-1) \frac{A_n^2}{A_e^2} \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}} \right\}^{\frac{1}{2}} \right]^{-1} \tag{42}$$

Equations (40), (41) and (42) can now be substituted into the thrust force expression, Equation (06), yielding the Simpson expression, Equation (43), for the thrust force on a relief valve when the flow is choked at the relief valve nozzle and unchoked at the relief valve exit. This equation yields the curves shown in Region B of Figure 1.

$$\frac{T_R}{P_0 A_n} = \frac{2k \dfrac{A_n}{A_e} \left\{ \dfrac{2}{k+1} \right\}^{\frac{k+1}{k-1}}}{\dfrac{P_b}{P_0} + \left[\dfrac{P_b^2}{P_0^2} + 2(k-1) \dfrac{A_n^2}{A_e^2} \left\{ \dfrac{2}{k+1} \right\}^{\frac{k+1}{k-1}} \right]^{\frac{1}{2}}} \tag{43}$$

8. CASE 3 – REGION C: SUBCRITICAL FLOW THROUGH THE ENTIRE RELIEF VALVE

Generally speaking, thrust forces are small for this case. The pressure discontinuity term in the thrust force equation is again zero, since the flow is unchoked at the relief valve exit plane. As for Case A and Case B, eight independent expressions are developed. These eight independent nonlinear algebraic equations are summarized in Table 3.

Table 3. Equation Set for Case 3 – Region C

$$G_e A_e = G_n A_n \tag{44}$$

$$\frac{P_0}{\rho_0^k} = \frac{P_n}{\rho_n^k} \tag{45}$$

$$G_n^2 = P_0 \rho_0 \left\{ \frac{2k}{k-1} \right\} \left\{ \left(\frac{P_n}{P_0} \right)^{\frac{2}{k}} - \left(\frac{P_n}{P_0} \right)^{\frac{k+1}{k}} \right\} \tag{46}$$

$$P_n = P_* \tag{47}$$

$$G_n^2 = \frac{2kP_n \rho_n}{k-1} \left\{ \frac{\rho_n}{\rho_*} - 1 \right\} \tag{48}$$

$$\frac{P_*}{\rho_*^k} = \frac{P_e}{\rho_e^k} \tag{49}$$

$$G_e^2 = P_* \rho_* \left\{ \frac{2k}{k-1} \right\} \left\{ \left(\frac{P_e}{P_*} \right)^{\frac{2}{k}} - \left(\frac{P_e}{P_*} \right)^{\frac{k+1}{k}} \right\} \tag{50}$$

$$P_e = P_b \tag{51}$$

The set of independent equations for this case is developed in a procedure similar to that used for Region A and Region B. The continuity of mass flow, Equation (44), provides a relationship between the mass flux at the relief valve nozzle and the mass flux at the relief valve exit. Isentropic flow of a perfect gas from the vessel stagnation condition to the relief valve nozzle yields Equation (45). The mass flux at the relief valve nozzle is expressed in terms of the vessel stagnation conditions. Equation (46) is the traditional mass flux equation for the subcritical flow of a perfect gas through a nozzle. Since the flow at the relief valve nozzle is unchoked, the nozzle pressure must be equal to the pressure in the relief valve body. This criterion is expressed as Equation (47). Adiabatic flow of a perfect gas from the relief valve nozzle to the relief valve body yields Equation (48), which is simply a different algebraic version of Equation (17). This relationship reflects the deceleration of the gas into the relief valve body. Isentropic flow of a perfect gas from the relief valve body to the relief valve exit yields Equation (49). The next to last expression, Equation (50), relates the mass flux at the relief valve exit to the fluid conditions in the relief valve body. This is the traditional mass flux equation for the subcritical flow of a perfect gas through a nozzle. Finally, since the flow at the relief valve exit is unchoked, the exit pressure must be equal to the backpressure. This criterion is expressed as Equation (51).

Again, in most problems of interest, the following parameters are known.

$P_0 \quad P_b \quad A_e \quad A_n \quad k$

As in Region A and Region B, the eight independent expressions [Equations (44) through (51)] shown in Table 3 must be solved simultaneously for the following eight unknowns.

$P_n \quad P_* \quad P_e \quad \rho_n \quad \rho_* \quad \rho_e \quad G_n \quad G_e$

At the October 2004 meeting of the DIERS Users Group, D'Alessandro [7] presented approximate equations for Region C, since analytical expressions could not be obtained. Since then, a complete analytical description of Region C was found.

When the eight equations in Table 3 are solved simultaneously, Equation (52) is obtained. For a given heat capacity ratio, outlet to nozzle area ratio, and backpressure ratio, this expression yields a specific body pressure ratio. Trial and error calculations are necessary. Since this equation has multiple roots, care is required to obtain the physically real solution for the body pressure ratio. Starting the trial and error calculation with a body pressure ratio of 0.999 assures that the correct solution is obtained.

$$\left\{\frac{P_*}{P_0}\right\}^{\frac{3-k}{k}} - \left\{\frac{P_*}{P_0}\right\}^{\frac{2}{k}} - \left\{\frac{A_e}{A_n}\right\}^2 \left\{\frac{P_b}{P_0}\right\}^{\frac{2}{k}} \left\{\frac{P_*}{P_0}\right\}^{\frac{k-1}{k}} + \left\{\frac{A_e}{A_n}\right\}^2 \left\{\frac{P_b}{P_0}\right\}^{\frac{k+1}{k}} = 0 \tag{52}$$

When the working equation for the thrust force, Equation (17), is combined with Equations (49), (50), and (51), the following expression is obtained.

$$\frac{T_R}{P_0 A_n} = \left\{\frac{A_e}{A_n}\right\}\left\{\frac{2k}{k-1}\right\}\left[\left\{\frac{P_*}{P_0}\right\}^{\frac{k-1}{k}} \left\{\frac{P_b}{P_0}\right\}^{\frac{1}{k}} - \left\{\frac{P_b}{P_0}\right\}\right] \tag{53}$$

Once a body pressure ratio is calculated via Equation (52), the corresponding thrust force parameter is calculated using Equation (53). The two expressions shown above, Equations (52) and (53), represent Region C on the Thrust Force Chart. The graphical representation and underlying equations for Region C were not included in the original Simpson work. As should be expected, Equation (52) and Equation (53) yield the correct limits when the backpressure equals the stagnation pressure. Namely, at this limit, the body pressure equals the stagnation pressure and the thrust force parameter equals zero.

9. BOUNDARY CONDITIONS

As mentioned previously and illustrated in Figure 1, the Simpson Charts are segregated into several distinct regions. In Region A, the flow through the relief valve is choked at both the relief valve nozzle and the relief valve exit. For Region B, the flow is only choked at the relief valve nozzle, but not at the relief valve exit where the flow is subcritical. In Region C, the flow is neither choked at the relief valve nozzle nor the relief valve exit. The equations derived thus far can now be utilized to develop expressions for the boundaries between these regions.

9.1 Boundary between Region A and Region B

For the boundary between Region A and Region B, the relief valve exit pressure is compared to the backpressure to determine if the flow is choked or unchoked. First, Equation (28) is utilized to calculate the relief valve exit pressure. Then the following two criteria are used to determine the flow condition at the relief valve exit plane.

$$If \quad \frac{P_b}{P_0} < \frac{P_e}{P_0}, \quad then\ the\ flow\ is\ choked. \tag{54}$$

$$If \quad \frac{P_b}{P_0} \geq \frac{P_e}{P_0}, \quad then\ the\ flow\ is\ unchoked\ and\ P_e = P_b. \tag{55}$$

To determine the thrust force parameter at the boundary between Region A and Region B, P_e is set equal to P_b in Equation (28). The resulting expression is combined with Equation (30) to yield the desired expression, Equation (56).

$$\left\{\frac{T_R}{P_0 A_n}\right\}_{AB} = k\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \quad when\ P_e = P_b \tag{56}$$

This equation describes the horizontal line between Region A and Region B on the Simpson Charts shown in Figure 1. This boundary is independent of the backpressure until the backpressure becomes high enough to unchoke the relief valve exit.

9.2 Boundary between Region B and Region C

For the boundary between Region B and Region C, the Simpson Chart compares the nozzle choking pressure, Equation (36), directly to the backpressure. The result is the vertical line between Region B and Region C shown in Figure 1. This assumes that there is no pressure drop between the relief valve body and the relief valve exit plane when the relief valve outlet is unchoked. A different approach is utilized here. The body pressure is compared against the nozzle choking pressure instead.

$$If \quad \frac{P_*}{P_0} < \frac{P_n}{P_0}, \quad then\ the\ flow\ is\ choked. \tag{57}$$

$$If \quad \frac{P_*}{P_0} \geq \frac{P_n}{P_0}, \quad then\ the\ flow\ is\ unchoked\ and\ P_n = P_*. \tag{58}$$

Equation (36) is still utilized to calculate the nozzle choking pressure. The body pressure is calculated by utilizing Equation (59), which is an alternate form of Equation (42) with the exit pressure set equal to the backpressure.

$$\frac{P_*}{P_0}=\left\{\frac{P_b}{P_0}\right\}\left\{\frac{1}{2}+\frac{1}{2}\left[1+2(k-1)\left\{\frac{A_eP_b}{A_nP_0}\right\}^{-2}\left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}\right\}^{\frac{k}{k-1}} \tag{59}$$

An expression for the boundary between Region B and Region C is obtained by setting Equation (36) equal to Equation (59). Algebraic manipulation yields the desired result.

$$\left\{\frac{P_b}{P_0}\right\}_{BC}^{\frac{k+1}{k}}-\left\{\frac{2}{k+1}\right\}\left\{\frac{P_b}{P_0}\right\}_{BC}^{\frac{2}{k}}+\left\{\frac{k-1}{2}\right\}\left\{\frac{A_n}{A_e}\right\}^2\left\{\frac{2}{k+1}\right\}^{\frac{2}{k-1}}=0 \tag{60}$$

For a given heat capacity ratio and nozzle to outlet area ratio, Equation (60) yields the backpressure coordinate of the boundary between Region B and Region C. Backpressure ratios equal to or greater than the value obtained by this equation will result in an unchoked relief valve nozzle. For the relief valve nozzle to remain choked, the actual backpressure must be less then backpressure calculated via Equation (60).

To determine the thrust force parameter at the boundary between Region B and Region C, Equation (53) is combined with Equation (59). This yields the following expression.

$$\left\{\frac{T_R}{P_0A_n}\right\}_{BC}=\left\{\frac{k}{k-1}\right\}\left\{\left[\left\{\frac{A_e}{A_n}\right\}^2\left\{\frac{P_b}{P_0}\right\}_{BC}^2+2(k-1)\left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}-\left\{\frac{A_e}{A_n}\right\}\left\{\frac{P_b}{P_0}\right\}_{BC}\right\} \tag{61}$$

This equation along with Equation (60) is used to map the new boundary between Region B and Region C.

It is interesting to note what happens to Equation (60) as the ratio of the outlet area to the nozzle area becomes very large. Under these circumstances, the third term in Equation (60) approaches zero and the expression simplifies to the following equation.

$$\left\{\frac{P_b}{P_0}\right\}_{BC\left(\frac{A_e}{A_n}\to\infty\right)}=\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \tag{62}$$

This is the equation that yields the vertical boundary line between Region B and Region C in the original Simpson Charts. Here, however, it only represents the boundary point for very large outlet to nozzle area ratios.

10. MINIMUM OUTLET TO NOZZLE AREA RATIO CONCEPT

Before describing the details of a minimum area ratio, a simple "thought" experiment is appropriate. Let us assume there are two equal diameter ideal frictionless nozzles in series, separated by a plenum and operating under steady state conditions. The flowing fluid is assumed to be a perfect gas. In addition, let us assume that the backpressure is low enough to assure choking at the downstream nozzle. Based on mass flow continuity at steady state conditions, the mass flux times the cross-sectional area for both nozzles must be equal. Since the diameters of both nozzles are equal, the mass flux through the upstream nozzle must be equal to the mass flux through the downstream nozzle. However, the pressure in the plenum between the two nozzles must also be lower than the stagnation pressure feeding the upstream nozzle. If the upstream nozzle is choked, the mass flux through this nozzle must be greater than the mass flux through the downstream nozzle. This is a logical inconsistency. Therefore, if the cross-sectional areas of both nozzles are equal and the downstream nozzle is choked, the upstream nozzle must be unchoked.

Obviously, the same conclusion is reached if the downstream nozzle has a cross-sectional area smaller than the upstream nozzle. Based on this "thought" experiment, there must be an outlet to nozzle area ratio greater than 1.0 where both the upstream and downstream nozzles cannot be choked. Assuming that such an area ratio exists, the flow and pressure equations for Region A (Table 1) can be used to determine this minimum area ratio.

The minimum outlet to nozzle area ratio is defined as follows.

$$\alpha_m = \left\{ \frac{A_e}{A_n} \right\}_{\min} \tag{63}$$

Combining Equation (27) and Equation (28) leads to the following relationship between the body pressure and the stagnation pressure.

$$\frac{P_*}{P_0} = \frac{A_n}{A_e} \tag{64}$$

For the relief valve nozzle to be unchoked, the nozzle pressure must be equal to the body pressure, namely, $P_n = P_*$. By using this equality along with Equation (23) and Equation (60), the relationship for the minimum outlet to nozzle area ratio is obtained.

$$\alpha_m = \left\{ \frac{A_e}{A_n} \right\}_{\min} = \left\{ \frac{2}{k+1} \right\}^{\frac{-k}{k-1}} \tag{65}$$

Pressure safety valves with outlet to nozzle area ratios less than those calculated via Equation (65) cannot choke at the main nozzle. For heat capacity ratios of 1.001, 1.40, and 1.80, the minimum outlet to nozzle area ratios are 1.655, 1.892, and 2.132, respectively.

If the equations for Region B are analyzed, the exact same result is obtained for the minimum outlet to nozzle area ratio. The expression for the boundary between Region B and Region C, Equation (60), is examined. For heat capacity ratios between 1.001 and 2.00, this equation normally has two roots between a backpressure ratio of zero and a backpressure ratio of one. If the left side of Equation (60) is set equal to a fictitious function, instead of zero, it can be shown that an extreme point exists between the two roots. This is accomplished by setting the first derivative of the function equal to zero. This procedure results in the following expression for the backpressure ratio where the boundary function between Region B and Region C achieves an extreme point. By examining the second derivative of boundary function, it is also determined that the extreme point is a minimum.

$$\left\{\frac{P_b}{P_0}\right\}_{BC\min} = \left\{\frac{2}{k+1}\right\}^{\frac{2k}{k-1}} \tag{66}$$

Therefore, it is possible for the boundary function to have only one root and possibly no roots. Based on this, it is postulated that the minimum outlet to nozzle area ratio occurs when the boundary function is set equal to zero and evaluated at a backpressure ratio equal to the minimum point. When this is done, Equation (65) is obtained. Outlet to nozzle area ratios less than the value given by Equation (65) result in a boundary function between Region B and Region C that has no solutions. The boundary function has only one root when the outlet to nozzle area ratio equals the value given by Equation (65). Finally, the boundary function has two roots when the outlet to nozzle area ratio is greater than the value given by Equation (65).

11. REVISED THRUST FORCE CHARTS

Based on the analysis shown in the previous sections, a revised thrust force chart is constructed. For each heat capacity ratio, a different chart can be generated. The thrust force chart for a gas or vapor with a heat capacity ratio of 1.4 is shown in Figure 3. The identical thrust force chart, noting all pertinent equations, is shown in Figure 4.

Figure 3 and Figure 4 can be easily compared with the original Simpson Chart, Figure 1. Differences include the following aspects:

(1) The extent of Region A is limited by the existence of a minimum outlet to nozzle area ratio and this minimum value is greater than 1.0.
(2) Region C has a complete set of thrust force curves that smoothly converge with the thrust force curves in Region B.
(3) The boundary line between Region B and Region C is curved rather than a straight vertical line and this curve depends on the backpressure and the heat capacity ratio.

Holding the relief valve outlet to nozzle area ratio constant and using the heat capacity ratio as a parameter can generate an alternate form of the thrust force chart. This type of thrust force chart is shown in Figure 5 for a 4M6 pressure safety valve with an outlet to nozzle area ratio of 7.12. The heat capacity ratio is varied from 1.001 to 1.80.

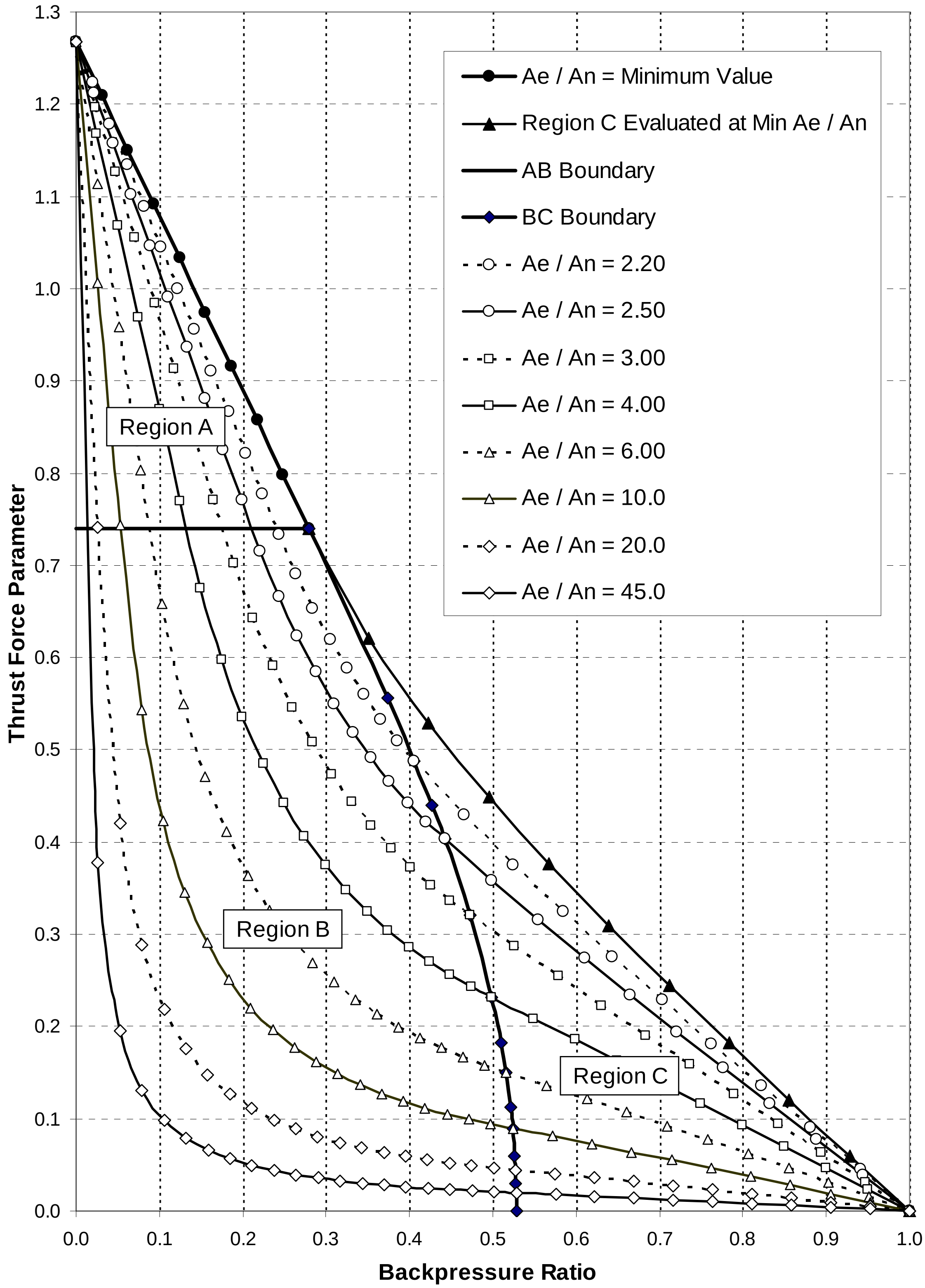


Figure 3. Thrust Force Chart for a Compressible Gas or Vapor
(k = 1.40 and $A_e / A_n \geq \alpha_m$)

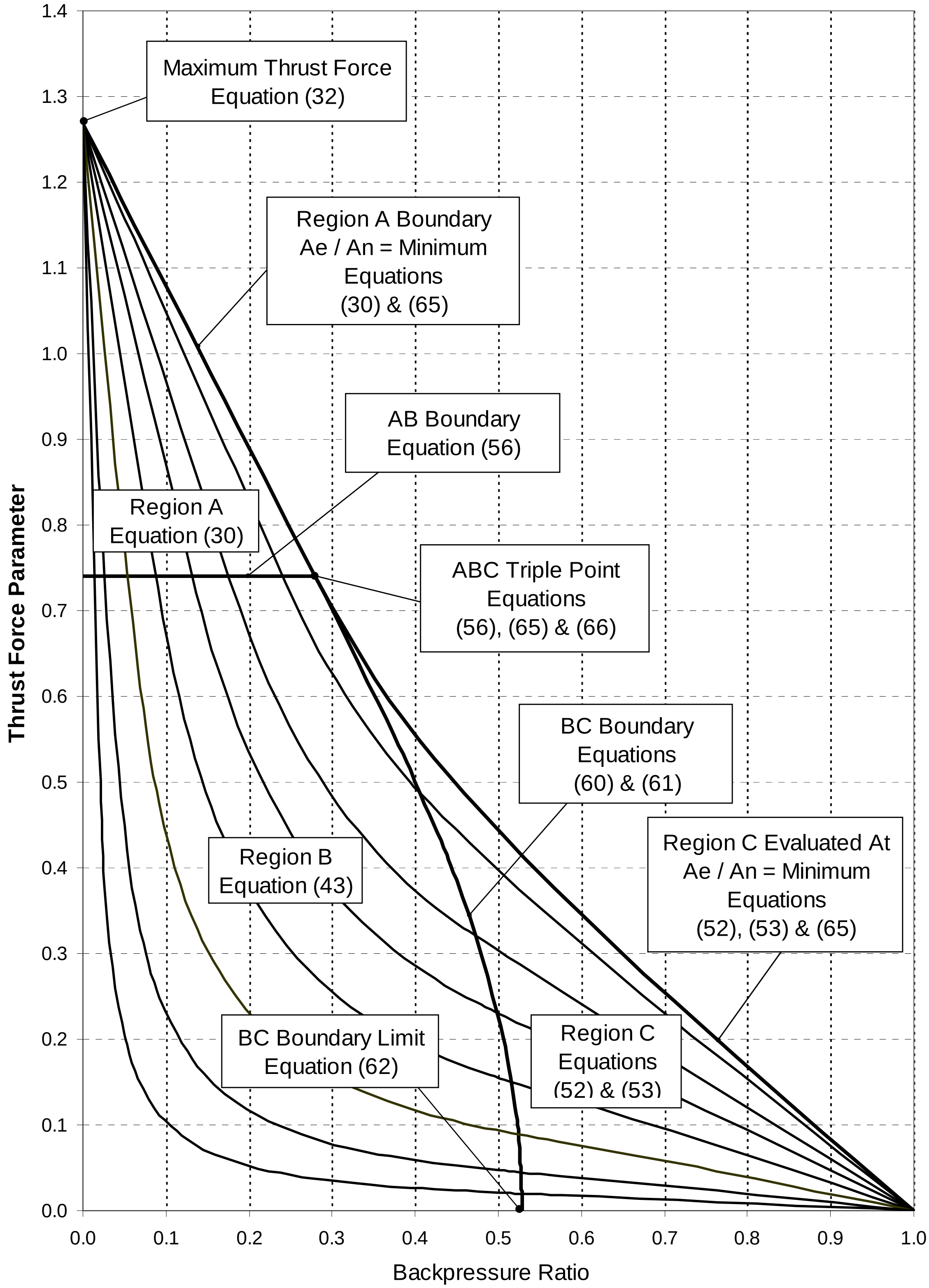


Figure 4. Thrust Force Chart for a Compressible Gas or Vapor
($k = 1.40$ and $A_e / A_n \geq \alpha_m$)

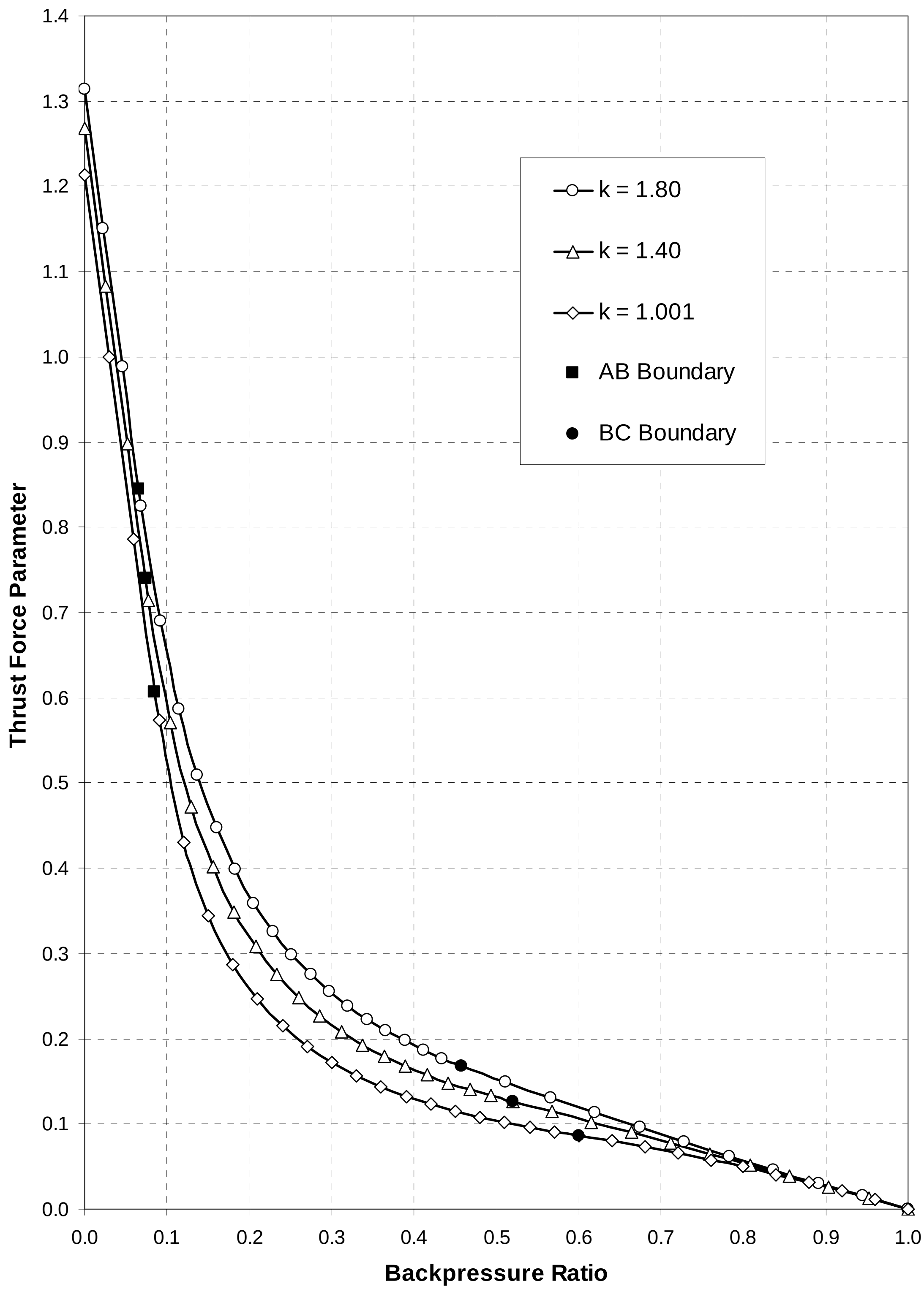


Figure 5. Thrust Force Chart - 4 M 6 PSV with Outlet to Nozzle Area Ratio of 7.12

12. OTHER RELIEF VALVE THRUST FORCE EQUATIONS

Other methods are available for calculating the thrust force when a relief valve actuates in compressible gas or vapor service. Usually, these equations use nomenclature that is different than the nomenclature utilized in this paper. Where possible, these alternate thrust force equations have been converted to the same nomenclature used here.

12.1 API RP 520 Method for Pressure Safety Valve Thrust Force

API RP 520 Part 2 [8] includes an expression for the reaction force on a pressure safety valve system in gas, vapor, or steam service. The equation applies to those relief valves connected to an outlet pipe discharging to atmosphere, which contains one 90-degree long radius elbow and a vertical pipe oriented in the upward direction. The thrust force in this case is established at the end of the discharge pipe in a direction opposite to the discharging fluid. The equation includes the effects of both momentum and static pressure. The English unit version of the API RP 520 formula is shown as Equation (67).

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + A_e P_e \qquad (67)$$

The following definitions and dimensions apply to this equation.

T_R is the thrust force [lb_f]
W is the mass flow [lb/h]
T_0 is the vessel stagnation temperature [°R]
M is the molecular weight of the fluid [lb/lbmol]
k is the heat capacity ratio [unitless]
A_e is the cross-sectional area of the pipe outlet at the point of discharge [in^2]
P_e is the static pressure within the pipe outlet at the point of discharge [psig]

API RP 520 Part 2 also shows the equivalent expression in metric units, which is given here as Equation (68).

$$T_R = 129\,W\sqrt{\frac{kT_0}{(k+1)M}} + 0.1\,A_e P_e \qquad (68)$$

The following definitions and dimensions apply to this equation.

T_R is the thrust force [N]
W is the mass flow [kg/s]
T_0 is the vessel stagnation temperature [K]
M is the molecular weight of the fluid [kg/kmol]
k is the heat capacity ratio [unitless]
A_e is the cross-sectional area of the pipe outlet at the point of discharge [mm^2]
P_e is the static pressure within the pipe outlet at the point of discharge [barg]

If the numerical coefficient in the first term on the right side of Equation (68) is brought under the square root, it becomes apparent that this term is actually two times the universal gas constant. If the cross-sectional area of the pipe outlet is taken in square meters and the static pressure within the pipe outlet is taken in absolute pressure units of Pascals, the numerical coefficient in the second term on the right side of Equation (68) disappears and the backpressure term can be added to the expression. With these modifications, Equation (68) can be rewritten as Equation (69), where *R* is the universal gas constant. In consistent units the value of *R* is 8314 Pa m^3 / kmol-K.

$$T_R = W\sqrt{\frac{2kRT_0}{(k+1)M}} + A_e(P_e - P_b) \tag{69}$$

Comparing Equation (69) to the basic thrust force expression, Equation (02), it becomes apparent that the square root term should be equivalent to the exit velocity at the pipe outlet. This relationship is expressed as Equation (70).

$$u_e = \sqrt{\frac{2kRT_0}{(k+1)M}} \tag{70}$$

An energy balance based on the adiabatic flow of a perfect gas from the vessel stagnation condition to the outlet of the discharge pipe yields the following expression. In this expression, T_e is the temperature of the gas at the exit of the discharge pipe.

$$\frac{T_0}{T_e} = \frac{k+1}{2} \tag{71}$$

Substitution of Equation (71) into Equation (70) yields the following alternate expression for the exit velocity at the pipe outlet. Equation (72) is the expression for the local speed of sound based on the flow of a perfect gas under adiabatic conditions.

$$u_e = \sqrt{\frac{kRT_e}{M}} = u_s \tag{72}$$

From this analysis, it is apparent that the API RP 520 formula for thrust force assumes that the flow is choked at the exit plane of the safety relief valve discharge pipe. This is equivalent to Region A of the Simpson Charts. The API RP 520 formula for thrust force does not apply when the flow is not choked at the exit plane, namely, conditions in Region B or Region C.

API RP 520 does not give a relationship for the pressure at the exit of the discharge pipe. However, since the flow in this model is assumed to be choked and is based on adiabatic conditions, the following expression is utilized for the exit pressure.

$$P_e = \frac{W}{A_e}\sqrt{\frac{2RT_0}{k(k+1)M}} \tag{73}$$

In addition to the above analysis, it can be shown that Equation (69) is identical to the thrust force expression derived in this paper for Region A, namely, Equation (30). To obtain this result, convert Equation (69) to dimensionless form by dividing both sides of the expression by the vessel stagnation pressure and the pressure safety valve nozzle cross-sectional area. Then eliminate the exit pressure by utilizing Equation (73). Finally, use the expression for the critical mass flux, Equation (12), along with the perfect gas law and the expression for the critical pressure ratio for the isentropic flow of a perfect gas, to eliminate the mass flow.

12.2 Crosby Pressure Relief Valve Engineering Handbook – Gas or Vapor Flow

The Crosby Pressure Relief Valve Engineering Handbook [9] gives the following equations for the thrust force on a pressure safety valve that apply to gas or vapor flow.

$$T_R = \frac{C K_d A_n P_0}{332.7}\sqrt{\frac{k}{k+1}} + F_g \tag{74}$$

$$F_g = \left\{\frac{K_d A_n P_0 K_r}{1.383 A_e} - P_b\right\} A_e \tag{75}$$

English units are used in these equations and the variables are defined as follows.

T_R	is the thrust force [lb_f]
F_g	is the thrust force due to static pressure at the valve outlet [lb_f]
K_d	is the relief valve nozzle discharge coefficient [unitless]
A_n	is the relief valve nozzle cross-sectional area [in^2]
P_0	is the relieving or stagnation pressure [psia]
k	is the heat capacity ratio for the gas [unitless]
A_n	is the relief valve outlet cross-sectional area [in^2]
K_r	is a correction factor for heat capacity ratios other than 1.4 [unitless]
C	is a coefficient given by Equation (76)

$$C = 520\sqrt{k\left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}} \tag{76}$$

The Crosby Relief Valve Handbook indicates that if the value of F_g is less than zero, then F_g should be set to zero in Equation (74). In addition, the Crosby Relief Valve Handbook indicates that the API effective orifice area should be used for A_n.

The value for K_r is a function of the heat capacity ratio only and is given in the Crosby Relief Valve Handbook in the form of a table. The value of K_r varies from a maximum of 1.15 when the heat capacity ratio is 1.01 to a minimum value of 0.84 when the heat capacity ratio is 2.0. For a heat capacity ratio of 1.40, the value of K_r is exactly 1.00.

With a discharge coefficient of 1.00, the combination of Equation (74) and Equation (75) yields the following expression for the thrust force.

$$T_R = \frac{CA_n P_0}{332.7}\sqrt{\frac{k}{k+1}} + \frac{K_r A_n P_0}{1.383} - A_e P_b \tag{77}$$

If the actual nozzle area is used instead of the API effective orifice area, the numerical coefficients in the first two terms on the right side of Equation (77) are increased by a factor of 1.10. In addition, the mass flow can be introduced into Equation (77) by utilizing the standard expression for critical gas flow through a pressure safety valve with a discharge coefficient of 1.0.

$$W = CA_n P_0\sqrt{\frac{M}{T_0}} \quad \Rightarrow \quad CA_n P_0 = W\sqrt{\frac{T_0}{M}} \tag{78}$$

With these two modifications, Equation (77) becomes Equation (79).

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + \frac{K_r A_n P_0}{1.521} - A_e P_b \tag{79}$$

Comparing this equation with the API RP 520 formula for the pressure safety valve thrust force, Equation (67), it is apparent that the first term on the right side of both equations is equivalent. If the effect of backpressure is included in the API RP 520 formula, it is apparent that the second and third terms of Equation (79) are equivalent to the second term of Equation (67). This relationship is illustrated in Equation (80).

$$\frac{K_r A_n P_0}{1.521} - A_e P_b = A_e(P_e - P_b) \quad \Rightarrow \quad K_r = 1.521\frac{A_e}{A_n}\frac{P_e}{P_0} \tag{80}$$

If Equation (28) is substituted into Equation (80), the resulting relationship is only a function of the heat capacity ratio. Comparing the calculated values of K_r based on this equation with the tabular values given in the Crosby Relief Valve Handbook, it becomes apparent that a constant multiplier of 1.25 is missing from Equation (80). Using this constant multiplier and Equation (28), the following equation reproduces the K_r values given by Crosby.

$$K_r = (1.25)(1.521)\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} = 1.90\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \tag{81}$$

By combining Equation (79) and Equation (81), we obtain the following expression.

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + 1.25 P_0 A_n\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - A_e P_b \tag{82}$$

Equation (82) can be converted to dimensionless form by dividing both sides of the equation by the vessel stagnation pressure and the relief valve nozzle cross-sectional area. In addition, Equation (76) and Equation (78) can be used to eliminate the mass flow term in Equation (82). The result is an alternate form of the formula for thrust force that can be directly compared with the thrust force formulation developed in this paper.

$$\frac{T_R}{A_n P_0} = \left\{ \frac{520k}{366\sqrt{2}} + 1.25 \right\} \left\{ \frac{2}{k+1} \right\}^{\frac{k}{k-1}} - \frac{A_e P_b}{A_n P_0} = \left(k + 1.25 \right) \left\{ \frac{2}{k+1} \right\}^{\frac{k}{k-1}} - \frac{A_e P_b}{A_n P_0} \qquad (83)$$

Comparing Equation (83) with the formula for the thrust force parameter for Region A, Equation (30), it is apparent that the formula given in the Crosby Relief Valve Handbook includes a 25% safety factor on that portion of the thrust force resulting from the discontinuity in static pressure at the outlet of the pressure safety valve. In addition, the Crosby formula only applies to pressure safety valves that are choked at both the nozzle and the exit plane, namely, Region A. The effect of high backpressure is not accounted for in the Crosby equations. Clearly the thrust force developed when the relief valve is not choked at the relief valve exit plane cannot be predicted by the Crosby equations. A plot of Equation (83) is shown in Figure 6. A heat capacity ratio of 1.40 is used to generate the plot.

To emphasize the differences, the Crosby expression, Equation (83), can be directly compared with the results for the thrust force equations developed in this paper. For this comparison, a heat capacity ratio of 1.40 and a relief valve outlet to nozzle area ratio of 6.0 is used. The results for both methods are plotted in Figure 7.

13. EXAMPLE PROBLEM 1

To illustrate the use of the revised thrust force charts and the associated equations developed in this paper, the following example, extracted from the CCPS Guidelines for Pressure Relief and Handling Systems [5], is utilized. To complete the comparison, the alternate methods of computing the thrust force that are given in this paper are also used.

Consider acetone flowing through a typical 2J3 pressure safety valve. Namely, the valve has a 2” nominal inlet, a J orifice designation, and a 3” nominal outlet. The ASME actual nozzle area is 1.452 in^2 and the ASME discharge coefficient is 0.864. The valve outlet has an inside diameter of 3.068 inches (7.393 in^2 or 0.004770 m^2). The pressure safety valve has a set pressure of 50 psig and operates with a 5 psi or 10% overpressure. Based on this, the relieving pressure is 55 psig or 69.7 psia. For a relieving condition of 69.7 psia, the CCPS example indicates a relieving temperature of 109.7 °C (382.85 K) with a corresponding heat capacity ratio of 1.102. Acetone has a molecular weight of 58.08 lb/lbmole. The backpressure immediately downstream of the pressure safety valve exit plane is 5 psig or 19.7 psia.

The CCPS Book claims that the relieving rate for this relief valve under the operating conditions stated above is 8,429 lb/h (1.0619 kg/s). The relieving rate is not

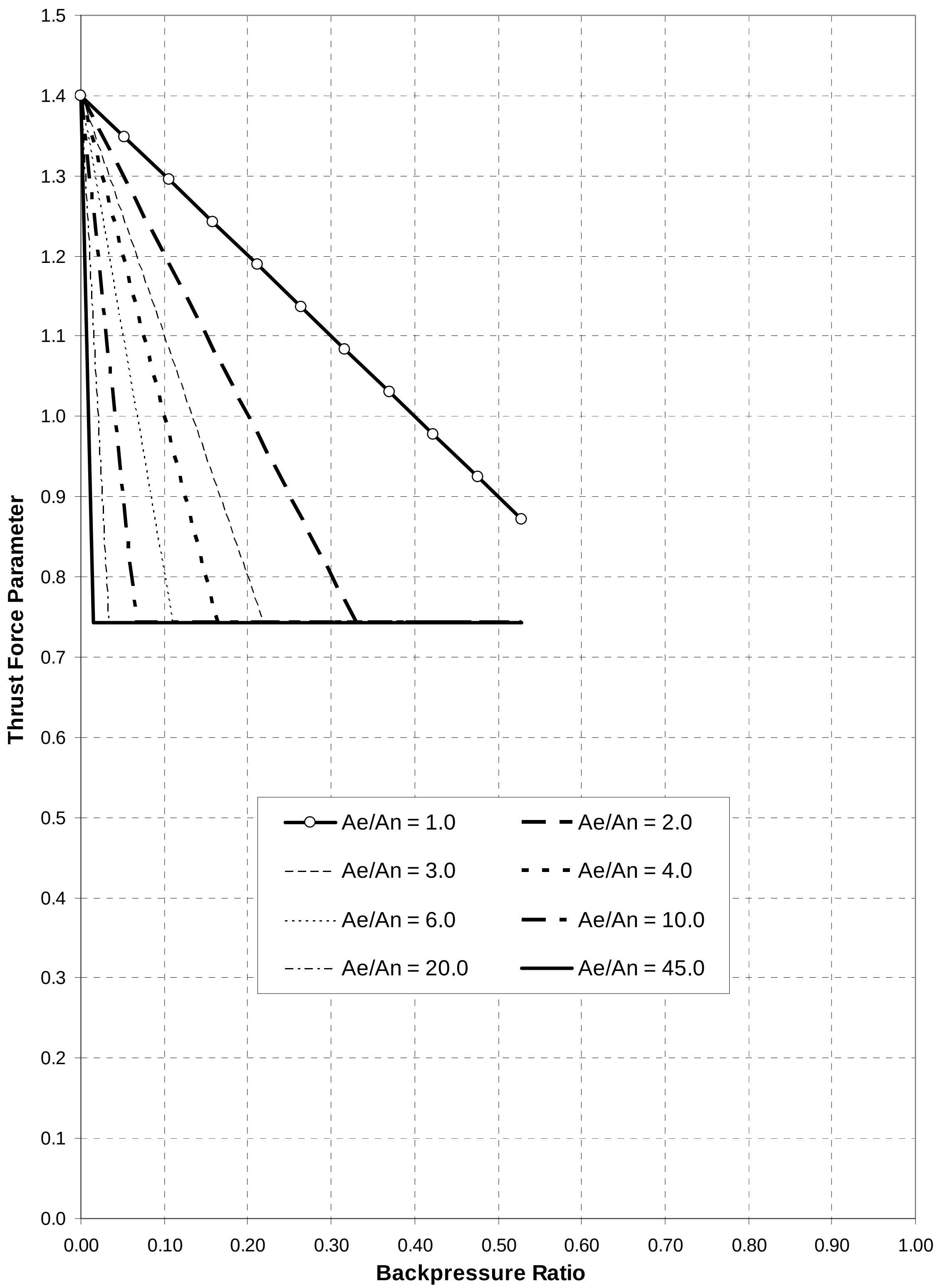


Figure 6. Thrust Force Parameter Utilizing the Crosby Method (k = 1.40, K_d = 1.0)

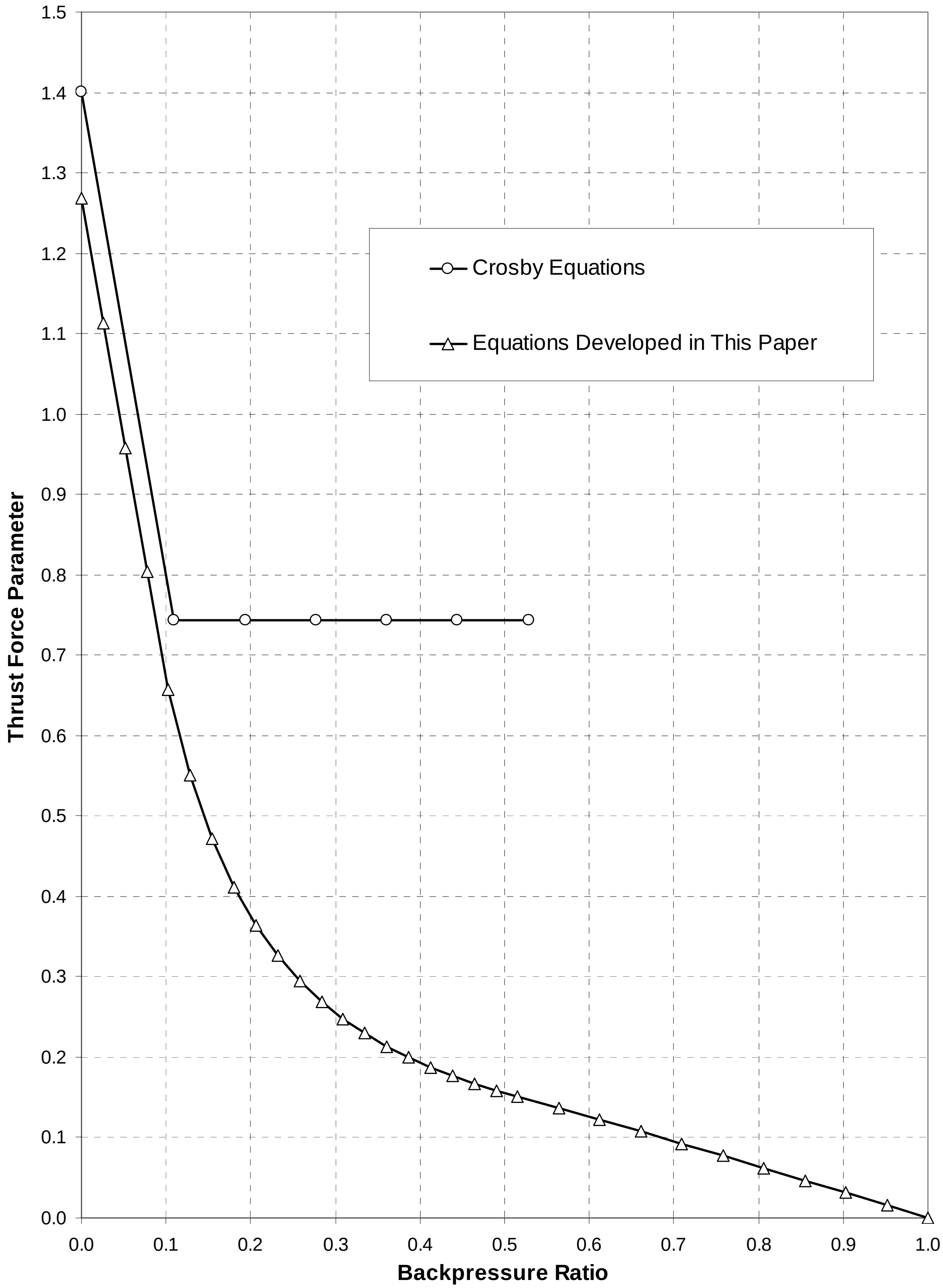


Figure 7. Comparison - Crosby Method versus Methods Developed in this Paper
(k = 1.40, K_d = 1.0, A_e / A_n = 6.0)

needed if either the thrust force charts or their underlying equations are used to evaluate the thrust force. However, several of the alternate methods given in this paper do require the relieving rate for the calculation of the thrust force.

13.1 Methods Developed in This Paper

Although the recommendation in this paper is to utilize the ASME actual nozzle area and a discharge coefficient of one for calculating the thrust force, the CCPS example reduces the nozzle area by multiplying by the ASME discharge coefficient. To allow an equivalent comparison, the same approach is taken here.

The outlet to nozzle area ratio and the backpressure ratio are calculated first. The outlet to nozzle area ratio is above the minimum value for a gas with a heat capacity ratio 1.102.

$$\frac{A_e}{A_n} = \frac{7.393}{(1.452)(0.864)} = 5.893$$

$$\frac{P_b}{P_0} = \frac{19.7}{69.7} = 0.283$$

Determine the choking pressure at the relief valve exit by utilizing Equation (28).

$$\frac{P_e}{P_0} = \frac{(1.452)(0.864)}{7.393}\left\{\frac{2}{1.102+1}\right\}^{\frac{1.102}{1.102-1}} = 0.0991 \quad \Rightarrow \quad P_e = (0.0991)(69.7) = 6.91\,psia$$

Since the choking pressure at the relief valve exit plane is less than the given backpressure, the flow is unchoked at the relief valve outlet. Therefore, the relief valve exit pressure is set equal to the backpressure. In addition, the operating condition of this relief valve lies in either Region B or Region C on the thrust force chart.

Now, determine the choking pressure at the relief valve nozzle by utilizing Equations (23).

$$\frac{P_n}{P_0} = \left\{\frac{2}{1.102+1}\right\}^{\frac{1.102}{1.102-1}} = 0.5843 \quad \Rightarrow \quad P_n = (0.5843)(69.7) = 40.7\,psia$$

The body pressure is determined by using Equation (59).

$$\frac{P_*}{P_0} = \left\{\frac{19.7}{69.7}\right\}\left\{\frac{1}{2} + \frac{1}{2}\left[1 + 2(1.102-1)\left\{\frac{7.393 \bullet 19.7}{1.452 \bullet 0.864 \bullet 69.7}\right\}^{-2}\left\{\frac{2}{1.102+1}\right\}^{\frac{1.102+1}{1.102-1}}\right]^{\frac{1}{2}}\right\}^{\frac{1.102}{1.102-1}}$$

$$= 0.3033 \quad \Rightarrow \quad P_* = (0.3033)(69.7) = 21.1\,psia$$

Since the calculated relief valve body pressure is less than the relief valve nozzle choking pressure (40.7 psia), the flow is choked at the relief valve nozzle. Therefore, the operating point for this relief valve lies in Region B on the thrust force chart.

Since the flow is choked at the relief valve nozzle and unchoked at the relief valve exit plane, Equation (43) is the correct expression for evaluating the thrust force parameter.

$$\frac{T_R}{P_0 A_n} = \frac{2(1.102)\frac{1}{5.893}\left\{\frac{2}{1.102+1}\right\}^{\frac{1.102+1}{1.102-1}}}{\frac{19.7}{69.7}+\left\{\left[\frac{19.7}{69.7}\right]^2+2(1.102-1)\left[\frac{1}{5.893}\right]^2\left[\frac{2}{1.102+1}\right]^{\frac{1.102+1}{1.102-1}}\right\}^{\frac{1}{2}}} = 0.2358$$

Now the thrust force can be calculated using the definition of the thrust force parameter, Equation (01).

$$T_R = (0.2358)(69.7)(1.452)(0.864) = 20.62\, lb_f = 91.7\, N$$

This value can be compared with the Simpson Chart value of 20 lb_f given in the CCPS book. For an additional comparison, the alternate COMFLOW value of 21.2 lb_f is also obtained from the CCPS Book.

13.2 API RP 520 Method

This method does not apply for this example, since the flow is subcritical at the pressure safety valve outlet. This can be demonstrated by utilizing Equation (73).

$$P_e = \frac{1.0619}{0.00477}\left\{\frac{(2)(8314)(382.85)}{(1.102)(1.102+1)(58.08)}\right\}^{\frac{1}{2}} = 48{,}426\, Pa = 7.02\, psia$$

This is very close to the value obtained by utilizing Equation (28). As before, this calculation shows that the pressure at the exit plane of the pressure safety valve must fall to below the given backpressure for critical conditions to exist at the outlet.

13.3 Crosby Method

Again, this method does not apply for this example. However, the thrust force is calculated anyway. First, the pertinent parameters must be determined using Equation (76) and Equation (81). If the Crosby Table for K_r were used instead, the same value would be obtained.

$$C = 520\sqrt{1.102\left\{\frac{2}{1.102+1}\right\}^{\frac{1.102+1}{1.102-1}}} = 327.0\frac{lb_m^{\frac{1}{2}}\, lbmol^{\frac{1}{2}}\, R}{lb_f\, h} \Rightarrow K_r = 1.90\left\{\frac{2}{1.102+1}\right\}^{\frac{1.102}{1.102-1}} = 1.11$$

Now Equation (75) is used to calculate the thrust force due to static pressure at the valve outlet. The result is a value less than zero, and therefore, the Crosby method sets F_g equal to zero. Then Equation (74) is used to calculate the total thrust force.

$$F_g = 7.393\left\{\frac{(0.864)(1.452)(69.7)(1.11)}{(1.383)(7.393)} - 19.7\right\} = -75.5\,lb_f < 0 \quad \Rightarrow \quad F_g = 0$$

$$T_R = \frac{(327.0)(0.864)(1.452)(69.7)}{332.7}\sqrt{\frac{1.102}{1.102+1}} = 62.2\,lb_f$$

The Crosby equation leads to a thrust force that is considerably larger than the thrust force calculated by using the methods developed in this paper.

14. EXAMPLE PROBLEM 2

The same basic example is used here. Except, a significantly higher stagnation pressure is utilized. This higher stagnation pressure causes critical flow at the pressure safety valve exit plane, and therefore, the operating point lies in Region A of the thrust force chart.

Again acetone is flowing through a typical 2J3 pressure safety valve. The ASME actual nozzle area is 1.452 in^2 and the ASME discharge coefficient is 0.864. The valve outlet has an inside diameter of 3.068 inches (7.393 in^2 or 0.004770 m^2). The pressure safety valve has a set pressure of 250 psig and operates with a 25.0 psi or 10% overpressure. Based on this, the relieving pressure is 275 psig or 289.7 psia. The saturation temperature of acetone at this pressure is 179.8 °C (452.95 K) and the heat capacity ratio is 1.333. The molecular weight is again 58.08 lb/lbmole. The backpressure immediately downstream of the pressure safety valve exit plane is still 5 psig or 19.7 psia.

14.1 Methods Developed in This Paper

As in Example 1, the ASME discharge coefficient is applied. The outlet to nozzle area ratio and the backpressure ratio are calculated first. The outlet to nozzle area ratio is still above the minimum valve for a gas with a heat capacity ratio 1.333.

$$\frac{A_e}{A_n} = \frac{7.393}{(1.452)(0.864)} = 5.893$$

$$\frac{P_b}{P_0} = \frac{19.7}{289.7} = 0.068$$

Determine the choking pressure at the relief valve exit by utilizing Equation (28).

$$\frac{P_e}{P_0} = \frac{(1.452)(0.864)}{7.393}\left\{\frac{2}{1.333+1}\right\}^{\frac{1.333}{1.333-1}} = 0.0916 \quad \Rightarrow \quad P_e = (0.0916)(289.7) = 26.54\, psia$$

Since the choking pressure at the relief valve exit plane is greater than the given backpressure, the flow is choked at the relief valve outlet. Therefore, the relief valve exit pressure is set equal to the exit choking pressure. In addition, the operating condition of this relief valve lies in Region A on the thrust force chart.

Check the choking pressure at the relief valve nozzle by utilizing Equations (23).

$$\frac{P_n}{P_0} = \left\{\frac{2}{1.333+1}\right\}^{\frac{1.333}{1.333-1}} = 0.5398 \quad \Rightarrow \quad P_n = (0.5398)(289.7) = 156.4\, psia$$

The body pressure is determined by using Equation (59).

$$\frac{P_*}{P_0} = \left\{\frac{19.7}{289.7}\right\}\left\{\frac{1}{2}+\frac{1}{2}\left[1+2(1.333-1)\left\{\frac{7.393 \bullet 19.7}{1.452 \bullet 0.864 \bullet 289.7}\right\}^{-2}\left\{\frac{2}{1.333+1}\right\}^{\frac{1.333+1}{1.333-1}}\right]^{\frac{1}{2}}\right\}^{\frac{1.333}{1.333-1}}$$

$$= 0.1805 \quad \Rightarrow \quad P_* = (0.1805)(289.7) = 52.3\ psia$$

Since the calculated relief valve body pressure is less than the relief valve nozzle choking pressure (156.4 psia), the flow is choked at the relief valve nozzle. Therefore, it is confirmed that the operating point for this relief valve lies in Region A on the thrust force chart.

Since the flow is choked at the relief valve nozzle and choked at the relief valve exit plane, Equation (30) is the correct expression for evaluating the thrust force parameter.

$$\frac{T_R}{P_0 A_n} = 2\left\{\frac{2}{1.333+1}\right\}^{\frac{1}{1.333-1}} - (5.893)(0.068) = 0.8587$$

Now the thrust force can be calculated using the definition of the thrust force parameter, Equation (01).

$$T_R = (0.8587)(289.7)(1.452)(0.864) = 312.1\, lb_f = 1{,}388.3\, N$$

14.2 API RP 520 Method

For this example, the API RP 520 method does apply, since the flow is choked at the pressure safety valve outlet. Since the stagnation condition is different, a new relieving rate must be determined. This is accomplished by utilizing Equation (76) and Equation (78). However, as in Example 1, the ASME discharge coefficient is utilized.

$$C = 520\left[1.333\left\{\frac{2}{1.333+1}\right\}^{\frac{1.333+1}{1.333-1}}\right]^{\frac{1}{2}} = 350.0\frac{lb_m^{\frac{1}{2}}\ lbmol^{\frac{1}{2}}\ R}{lb_f\ h}$$

$$W = (350)(1.452)(0.864)(289.7)\sqrt{\frac{58.08}{355.64+459.69}} = 33{,}950\,lb/h = 4.277\,kg/s$$

Now use Equation (73) to determine the exit pressure.

$$P_e = \frac{4.277}{0.00477}\left\{\frac{(2)(8314)(452.95)}{(1.333)(1.333+1)(58.08)}\right\}^{\frac{1}{2}} = 183{,}097\,Pa = 26.56\,psia$$

Again, this is very close to the value obtained by utilizing Equation (28) from the methods described in this paper. This calculation shows that the pressure at the exit plane of the pressure safety valve is higher than the backpressure of 19.7 psia (135,827 Pa). Therefore, the flow is choked at the outlet.

Now Equation (68) can be utilized for calculating the thrust force.

$$T_R = (129)(4.277)\sqrt{\frac{(1.333)(452.95)}{(1.333+1)(58.08)}} + (0.00477)(183{,}097 - 135{,}827) = 1{,}390.2N$$

This is essentially the same result obtained by utilizing the methods described in this paper.

14.3 Crosby Method

The Crosby method applies for this example since the flow is choked at both the pressure safety valve nozzle and exit plane. First, the K_r must be determined using Equation (81).

$$K_r = 1.90\left\{\frac{2}{1.333+1}\right\}^{\frac{1.333}{1.333-1}} = 1.026$$

Now Equation (75) is used to calculate the thrust force due to static pressure at the valve outlet. In this case the value obtained for F_g is greater than zero. Then Equation (74) is used to calculate the total thrust force.

$$F_g = 7.393\left\{\frac{(0.864)(1.452)(289.7)(1.026)}{(1.383)(7.393)} - 19.7\right\} = +124.0\,lb_f > 0$$

$$T_R = \frac{(350.0)(0.864)(1.452)(289.7)}{332.7}\sqrt{\frac{1.333}{1.333+1}} + 124.0 = 413.0\,lb_f = 1837\,N$$

The Crosby equation leads to a thrust force that is somewhat larger than the thrust force calculated by using the methods described in this paper. This is primarily due to the 25% safety factor described in the main discussion on the Crosby method.

15. ASSUMPTIONS AND LIMITATIONS

In order to derive the equations shown in this paper, several assumptions were necessary. These assumptions limit the applicability of these equations. Therefore, apply these equations with care when conditions fall outside of the stated limitations.

In developing the flow models, the following primary assumptions concerning the fluid and its flow characteristics were made.

(1) The fluid obeys the perfect gas law.

(2) The flow through the entire relief valve is adiabatic, namely, heat is neither lost nor gained by the fluid as it flows from the vessel stagnation condition to the relief valve exit.

(3) The flow through the entire relief valve is frictionless.

(4) The flow from the vessel stagnation condition to the relief valve nozzle is isentropic.

(5) The fluid stagnates in the relief valve body.

(6) The flow from the relief valve body to the relief valve exit plane is isentropic.

(7) The heat capacity ratio for the gas is held constant throughout the expansion process, namely, is independent of decreasing pressure and decreasing temperature.

For the most part, these seven assumptions do not severely limit the applicability of the charts or their underlying equations.

Deviations from the perfect gas assumption in terms of compressibility factors less than one would tend to increase the capacity of the relief valve, and therefore, increase the thrust force. However, at low to moderate pressures, the increase is relatively small. The assumption of perfect adiabaticity is a standard technique in the study of compressible gas flow and deviations are usually minor. Based on measured values of flow through standard relief valves under choking conditions in the relief valve nozzle, it is quite apparent that deviations from isentropic flow are minor and these deviations tend to decrease the capacity of the relief valve, and therefore, the thrust force. Generally speaking, the flow path through the relief valve nozzle is short, and therefore, frictional effects are small. Based on Figure 5, it is apparent that the thrust force parameter is relatively insensitive to the gas heat capacity ratio.

However, the assumptions that the gas stagnates after entering the relief valve body and that isentropic flow occurs from the relief valve body to the relief valve outlet are tentative. As mentioned before, the relief valve body has a very complex geometry. This complex geometry causes the relieving fluid to have a complex flow pattern. The fluid must change direction several times and there are certainly some frictional effects within the relief valve body. However, by making these two assumptions, analytical expressions can be developed and these expressions yield interesting information and valuable insight about a rather complex problem. More complicated methods can certainly be utilized to analyze the body of the relief valve. However, these added complexities will certainly require a more detailed description of the geometry within the relief valve body and will almost certainly require numerical methods for computations. Therefore, when using the models presented in this paper, due consideration must be given to how these two assumptions may alter the calculated results.

It is assumed that the net force at the relief valve nozzle is zero since the relief valve disk acts as a thrust plate that transfers the momentum of the flowing fluid to the relief valve mounting nozzle. The pressure discontinuity that may exist at the relief valve nozzle is balanced by the relief valve mounting nozzle as well.

The thrust force equations in the Crosby Relief Valve Handbook incorporate a relief valve discharge coefficient. This discharge coefficient accounts for the difference in mass flow between a theoretical nozzle and real nozzle. For critical flow of a compressible gas through a relief valve, the measured discharge coefficient has a typical value between 0.94 and 0.98. The discharge coefficient reduces the predicted flow through the relief valve, and therefore, when used in a thrust force formulation would tend to reduce the calculated thrust force. The charts and the underlying equations given in this paper assume a discharge coefficient of 1.0. Therefore the thrust forces calculated using the charts are slightly conservative in this respect.

The thrust force parameters obtained from the equations in this paper yield ideal theoretical thrust forces. The real thrust force will probably be larger. The derivations assume steady state flow through the relief valve. This is almost never true. Flow through a relief valve is almost always dynamic, with the relief valve opening and closing, sometimes over very short periods of time. This dynamic behavior tends to increase the reaction forces through harmonic effects. It is common practice to multiply the thrust force values obtained from the equations presented in this paper by a Dynamic Load Factor (DLF). The DLF has a maximum theoretical value of 2.0 and, in most installations, this maximum value can be applied without severe cost implication [10].

16. OPEN LITERATURE METHODS FOR PSV THRUST FORCES IN GAS/VAPOR SERVICE

Table 4 below summarizes the status of the thrust force methods for gas or vapor pressure safety valves in several of the major open literature sources. Most of these do not include a method when the pressure safety valve exit is unchoked and none include a method when both the nozzle and outlet are unchoked. Some are even in error.

Table 4. Summary of Open Literature Methods for Pressure Safety Valve Thrust Forces in Gas/Vapor Service

Literature Method	Region A	Region B	Region C	Remarks
DIERS Project Manual [3]	Correct Equation	Graph Only	Not Included	
DIERS Project Manual Errata [4]	Incorrect Equation	Incorrect Equation	Not Included	Probably a Typographical Error
CCPS Guidelines for ERS [5]	Graph Only	Graph Only	Not Included	
HSE Workbook [6]	Incorrect Equation	Not Included	Not Included	Probably a Typographical Error
API RP 520 [8]	Correct Equation	Not Included	Not Included	No Equation for Exit Pressure
Crosby Relief Valve Handbook [9]	Correct Equation	Not Included	Not Included	Includes a 25% Safety Factor

NOTE TO READERS

The author has also extended the methods shown in this paper to outlet to nozzle area ratios less than the minimum value given by Equation (65). The results are consistent with the results shown in the preceding discussions. This information was not included here since relief valves rarely, if ever, have nozzle to outlet area ratios less than the minimum. Interested readers are welcome to contact the author for additional information.

17. ACKNOWLEDGMENTS

The author gratefully acknowledges the valuable input obtained from Mr. Harold Fisher, Chairman of the DIERS Users Group, during the preparation of this manuscript. In addition, thanks are extended to Mr. Andrew Jones, Mr. Jeff Seay, P.E. and Mr. Frank Fearn, P.E. of Degussa Corporation for reviewing the document several times prior to publication.

A very special acknowledgement must also be extended to Mr. Jim Huff who reviewed the present paper and provided extremely beneficial feedback on some of the detailed technical aspects of the proposed methods [11]. Huff maintains that the trust force at the relief valve outlet can be determined without specifying the flow path within the relief valve body. In other words, the assumption of isentropic flow from the relief valve body to the relief valve exit is unnecessary. Instead, simply considering adiabatic flow from the relief valve nozzle to the relief exit in conjunction with isentropic nozzle flow from the stagnation condition to the relief valve nozzle and a continuity equation should be sufficient for calculating the thrust force.

Based on this insight, the author of the present paper developed the set of corresponding equations shown in Table 5.

Table 5. Isentropic Relief Valve Nozzle and Adiabatic Relief Valve Body

Common Equations for Regions A and B

$$G_e A_e = G_n A_n \tag{84}$$

$$G_n^2 = k\rho_n P_n \tag{85}$$

$$\frac{P_n}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \tag{86}$$

$$\frac{P_n}{\rho_n} = \frac{RT_n}{M} \quad and \quad \frac{P_e}{\rho_e} = \frac{RT_e}{M} \tag{87}$$

$$T_n\left\{1+\frac{k-1}{2}M_n^2\right\} = T_e\left\{1+\frac{k-1}{2}M_e^2\right\} \quad with \quad M_n = 1.0 \tag{88}$$

$$M_e = \frac{u_e}{u_{se}} = \frac{G_e}{\rho_e u_{se}} \tag{89}$$

$$u_{se} = \sqrt{\frac{kRT_e}{M}} \tag{90}$$

Equations for Region A Only

$$G_e^2 = k\rho_e P_e \tag{91}$$

$$M_e = 1.0 \tag{92}$$

Equations for Region B Only

$$P_e = P_b \tag{93}$$

In Table 5, the following nomenclature is used for any newly introduced variables and parameters. All previously introduced nomenclature still applies.

T_n Temperature at the throat of the relief valve nozzle [K]
T_e Temperature at the relief valve exit plane [K]
M_n Mach number at the throat of the relief valve nozzle [unitless]
M_e Mach number at the relief valve exit plane [unitless]
u_{se} Local speed of sound at the relief valve exit plane [m/s]

All of the equations in Table 5 are independent of the flow path through the relief valve body. In other words, neither the relief valve body pressure (P_*) nor the fluid density in the relief valve body (ρ_*) appears in these expressions. Yet this set of equations, when combined properly yield the exact thrust force equations derived in the present paper for both Region A, Equation (30) and Region B, Equation (43). This should be expected since the assumption of isentropic flow through the body is by definition also adiabatic. However, this approach has disadvantages.

Since the equations in Table 5 do not include the relief valve body pressure, the method offers limited insight into the effects of backpressure on the relief valve nozzle, namely, the BC boundary. In addition, the potential problems associated with a minimum outlet to nozzle area ratio cannot be ascertained. Finally, the equations for Region C cannot be derived.

The Simpson chart shows the BC boundary as a vertical line. This implies that there is no pressure drop between the relief valve body and the relief valve exit when the relief valve exit is unchoked. This cannot be the case. The assumption of isentropic flow from the relief valve body to the relief valve exit may be simplistic, but it definitely yields interesting information and a starting point for further analysis. In considering the location of the BC boundary and its effect on the operation of a relief valve, the assumption of isentropic flow from the relief valve body to the relief valve exit is conservative. The real BC boundary probably lies somewhere between the one obtained in the present paper versus the one shown on the original Simpson Charts.

In Region C, the relief valve nozzle is unchoked. There is no pressure discontinuity between the relief valve nozzle and the relief valve body. The mass flow through the entire relief valve is now determined by the pressure loss aspects of both the relief valve nozzle, the relief valve body, and the relief valve exit. Therefore, to evaluate the thrust force in Region C, a pressure loss model for the relief valve body is necessary.

18. REFERENCES

1. Simpson, Larry L., "Reaction Forces from Process Venting", unpublished Union Carbide Corporation memorandum (October 1969).
2. Huff, James E., "Relief System Reaction Forces in Gas and Two-Phase Flow", Paper Presented at the *25th Annual AIChE Loss Prevention Symposium*, August 18-22, 1991.
3. Fisher, Harold G., *et al.*, *Emergency Relief System Design Using DIERS Technology – The Design Institute for Emergency Relief System DIERS Project Manual*, American Institute of Chemical Engineers / Design Institute for Emergency Relief Systems, New York, 1992.
4. Errata to the DIERS Project Manual, published January 5, 2005.
5. *Guidelines for Pressure Relief and Effluent Handling Systems*, American Institute of Chemical Engineers/ Center for Chemical Process Safety, New York, 1998.

6. Etchells, Janet and Wilday, Jill., *Workbook for Chemical Reactor Relief System Sizing*, Health and Safety Executive, Contract Research Report 136/1998, ISBN 0 7176 1389 5, first published in 1998.
7. D'Alessandro, Robert N., "Simpson Charts for PSV Thrust Force – Revisited", Paper Presented at the *34th Meeting of the DIERS Users Group*, Denver, Colorado, October 4-6, 2004.
8. API Recommended Practice 520, 4th Edition, "Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries, Part II-Installation", American Petroleum Institute, December 1994.
9. Crosby Valve & Gage Company, "Crosby Pressure Relief Valve Engineering Handbook", Technical Document No. TP-V300, May 1997.
10. Huff, James E., "Flow through Emergency Relief Devices and Reaction Forces", *Journal of Loss Prevention in the Process Industries*, Volume 3, January 1990.
11. Huff, James E., Personal communication to R. D'Alessandro, dated February 8, 2005.

Appendix A – Typical Relief Valve Geometries

Nozzle Designation	Typical Actual Nozzle Area in^2	Nominal Outlet Size inches	Outlet Inside Diameter (Sch. 40) inches	Outlet to Nozzle Area Ratio unitless
D	0.131	2.5 2.0	2.469 2.067	36.5 25.5
E	0.223	2.5 2.0	2.469 2.067	21.5 15.0
F	0.354	2.5 2.0	2.469 2.067	13.5 9.5
G	0.567	3.0 2.5	3.068 2.469	13.0 8.4
H	0.886	3.0	3.068	8.3
J	1.452	4.0 3.0	4.026 3.068	8.8 5.1
K	2.073	6.0 4.0	6.065 4.026	13.9 6.1
L	3.217	6.0 4.0	6.065 4.026	9.0 4.0
M	4.060	6.0	6.065	7.1
N	4.894	6.0	6.065	5.9
P	7.195	6.0	6.065	4.0
Q	12.46	8.0	7.981	4.0
R	18.04	10.0 8.0	10.02 7.981	4.4 2.8
T	28.93	10.0	10.02	2.7
T2	31.0	10.0	10.02	2.5
U	31.5	10.0	10.02	2.5
W	63.6	16.0	15.00	2.8
W2	104.0	18.0	16.876	2.2
X	113.1	20.0	18.812	2.5
Y	143.1	24.0	22.624	2.8
Z	176.7	24.0	22.624	2.3

degussa.

Technology & Process

Simpson Charts
for
PSV Thrust Force Revisited

DIERS Users Group Meeting
Denver, Colorado
October 2004

Robert D'Alessandro, P.E.
Degussa Corporation

Overview

degussa.

Technology & Process

- **Motivations**
- **Thrust Force Fundamentals**
- **Simpson Charts**
  - **Model for Region A**
  - **Model for Region B**
  - **Model for Region C**
  - **Boundary Conditions**
- **Comparisons**
  - **API RP 520 Method**
  - **Crosby Method**
- **Summary & Concluding Remarks**

Motivations

Large Glass-Lined Storage Vessel

Discrepancies Noted in Several Literature Sources

Simpson Reference not in my possession at the time (*)

Lack of Functional Relationships for Computer Based Calculations (Note)

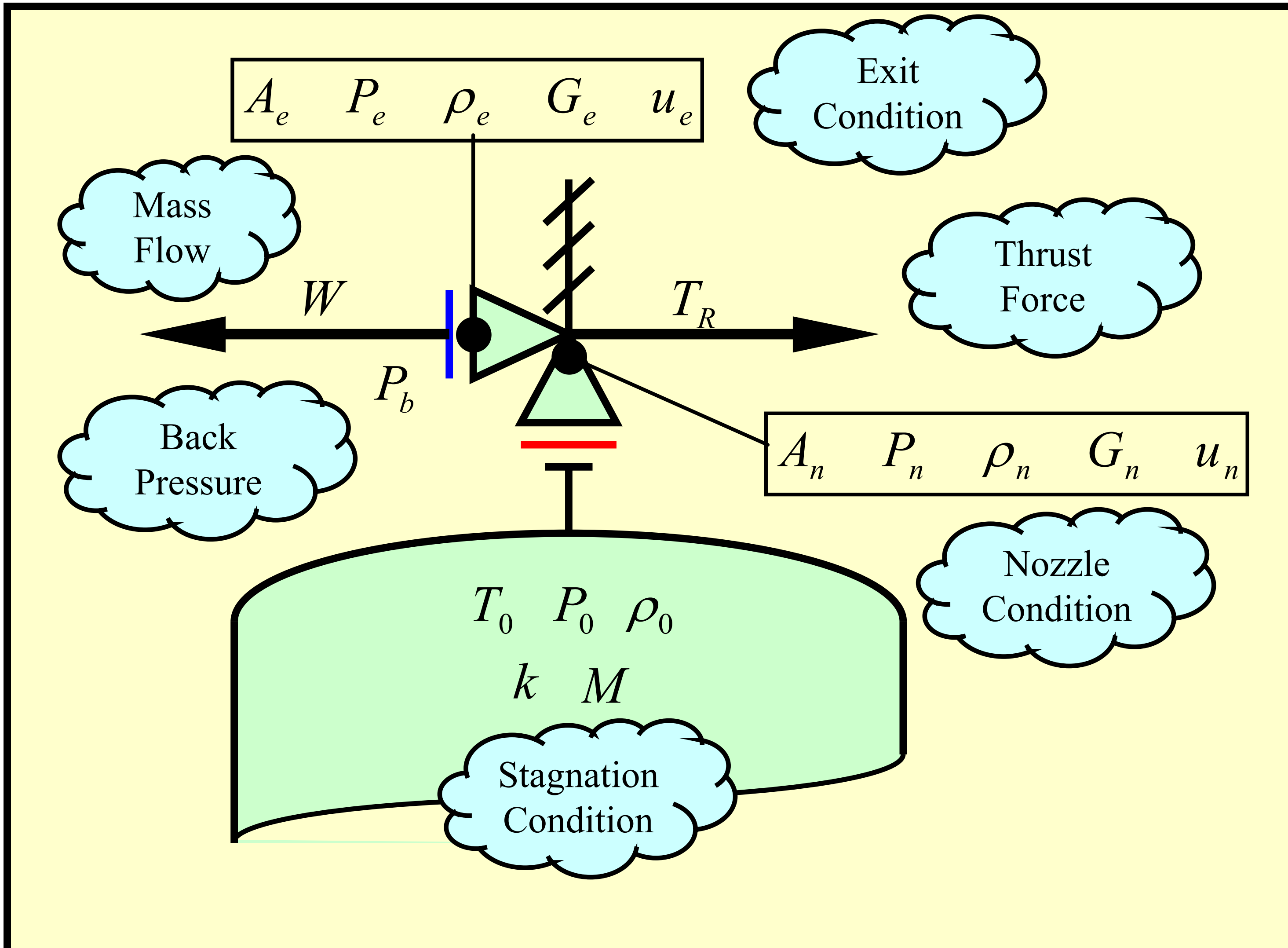
Lack of Underlying Assumptions

* Simpson 1969 – Internal Union Carbide Memoradum

System Sketch

degussa.

Technology & Process



PSV Thrust Force Fundimentals

$$T_R = Wu_e + A_e\left(P_e - P_b\right)$$

Basic Force Balance

Momentum

Pressure Discontinuity

$$\frac{T_R}{P_0 A_n} = \frac{Wu_e}{P_0 A_n} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\}$$

Dimensionless Form

$$W = G_e A_e$$

$$u_e = \frac{G_e}{\rho_e}$$

Eliminate Mass Flow & Exit Velocity

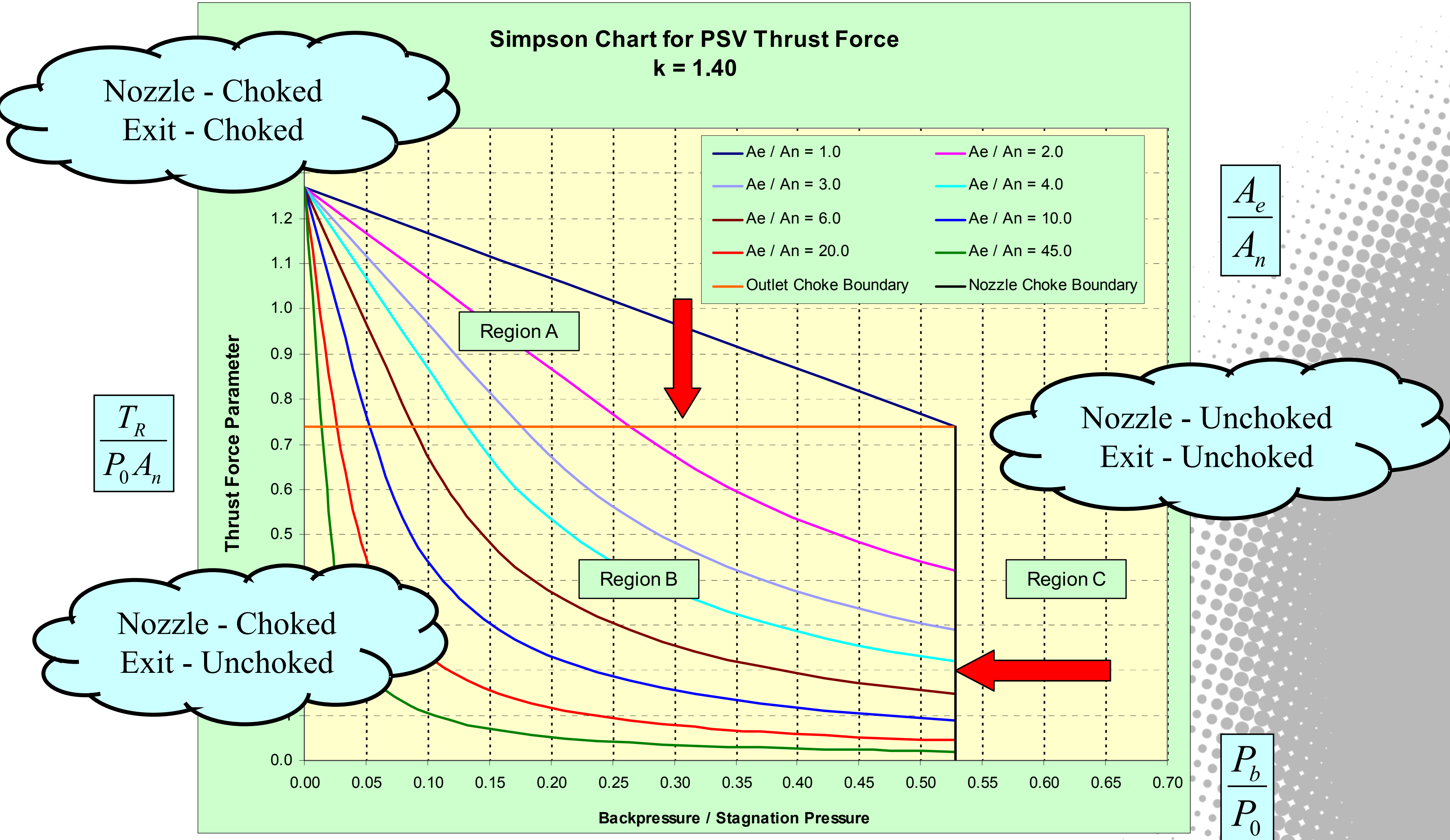
$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{G_e^2}{P_0 \rho_e} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\}$$

Working Equation

Simpson Chart

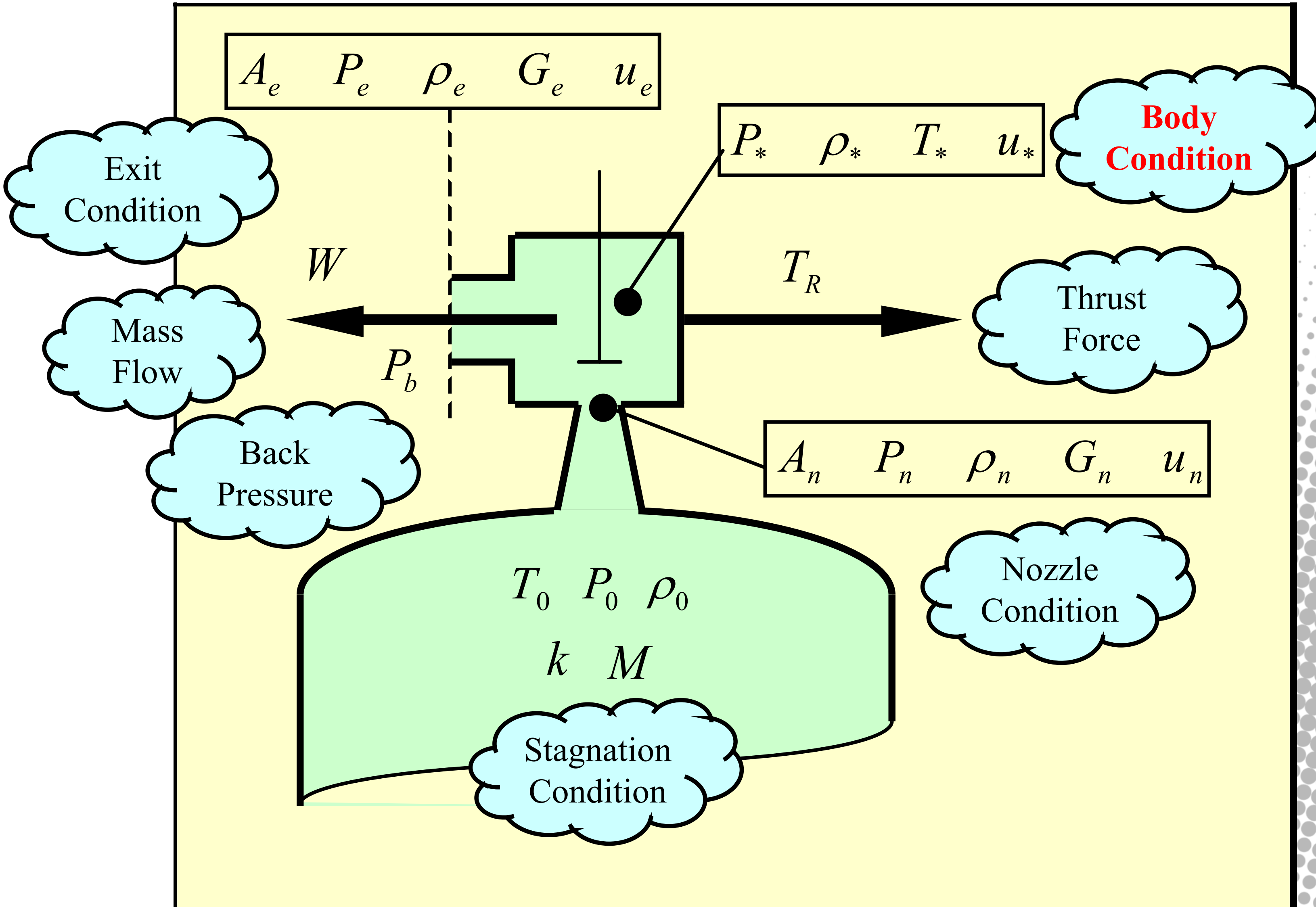
degussa.

Technology & Process



Robert D'Alessandro, P.E.

System Model



Important Flow Expressions

degussa.

Technology & Process

$$G = \frac{1}{v}\sqrt{2\left(h_0 - h\right)}$$

Isentropic Ideal Nozzle Mass Flux

Thermodynamics

$$G_c = \left\{-\frac{dP}{dv}\right\}_{P_c}^{\frac{1}{2}}$$

Isentropic Ideal Nozzle Critical Mass Flux

$$Pv^k = P_0 v_0^k$$

For Perfect Gases

$$G_c^2 = \frac{kP_c}{v_c} = k\rho_c P_c \quad or \quad \frac{G_c^2}{\rho_c} = kP_c$$

Local Choking Condition for a Perfect Gas

Important Flow Expressions

degussa.

Technology & Process

$$h_* = h_n + \frac{u_n^2}{2}$$

Adiabatic Energy Balance to a Stagnant State

$$u_n = \frac{G_n}{\rho_n}$$

$$h_* - h_n = \int_{T_n}^{T_*} C_p\,dT = \frac{R}{M}\left\{\frac{k}{k-1}\right\}\int_{T_n}^{T_*} dT = \left\{\frac{k}{k-1}\right\}\frac{R}{M}\left(T_* - T_n\right)$$

For Perfect Gases

$$\frac{P_*}{\rho_*} = \left\{\frac{k+1}{2}\right\}\frac{P_n}{\rho_n}$$

For Nozzle Choking Only

degussa.

Technology & Process

Region A

Choked Nozzle / Choked Exit

$$G_e A_e = G_n A_n \quad \Rightarrow \quad \frac{A_e^2}{A_n^2}\frac{\rho_e}{\rho_n} = \frac{P_n}{P_e}$$

Continuity

$$\frac{P_0}{\rho_0^k} = \frac{P_n}{\rho_n^k}$$

Isentropic Flow

$$\frac{P_n}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Nozzle Choking

$$\frac{P_*}{\rho_*} = \left\{\frac{k+1}{2}\right\}\frac{P_n}{\rho_n}$$

Adiabatic Flow

$$\frac{P_*}{\rho_*^k} = \frac{P_e}{\rho_e^k}$$

Isentropic Flow

$$\frac{P_e}{P_*} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Exit Choking

6 Equations
6 Unknowns

$P_n \quad P_* \quad P_e \quad \rho_n \quad \rho_* \quad \rho_e$

Solve in Terms of Known Parameters

$P_0 \quad P_b \quad A_e \quad A_n \quad k$

Region A

Choked Nozzle / Choked Exit

Analytical Solution

$$\frac{A_e}{A_n}\frac{P_e}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Exit Choking

$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{G_e^2}{P_0 \rho_e} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\}$$

$$G_e^2 = k\rho_e P_e$$

$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{P_e}{P_0}(k+1) - \frac{A_e}{A_n}\frac{P_b}{P_0}$$

Intercept

Independent Variable

Simpson Equation for Region A

$$\frac{T_R}{P_0 A_n} = (k+1)\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - \frac{A_e}{A_n}\frac{P_b}{P_0} = 2\left\{\frac{2}{k+1}\right\}^{\frac{1}{k-1}} - \frac{A_e}{A_n}\frac{P_b}{P_0}$$

Straight Line

Slope

Region B

Choked Nozzle / Unchoked Exit

degussa.

Technology & Process

$$G_e A_e = G_n A_n$$

Continuity

$$\frac{P_0}{\rho_0^k} = \frac{P_n}{\rho_n^k}$$

Isentropic Flow

$$G_n^2 = kP_0\rho_0 \left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}} \qquad \frac{P_n}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Nozzle Choking

$$\frac{P_*}{\rho_*} = \left\{\frac{k+1}{2}\right\}\frac{P_n}{\rho_n}$$

Adiabatic Flow

$$\frac{P_*}{\rho_*^k} = \frac{P_e}{\rho_e^k}$$

Isentropic Flow

$$G_e^2 = P_*\rho_* \left\{\frac{2k}{k-1}\right\}\left\{\left(\frac{P_e}{P_*}\right)^{\frac{2}{k}} - \left(\frac{P_e}{P_*}\right)^{\frac{k+1}{k}}\right\}$$

Exit Flow

$$P_e = P_b$$

Region B Condition

8 Equations
8 Unknowns

$$\begin{array}{ccc} P_n & P_* & P_e \\ \rho_n & \rho_* & \rho_e \\ G_n & G_e & \end{array}$$

Solve in Terms of Known Parameters

$$P_0 \quad P_b \quad A_e \quad A_n \quad k$$

Robert D'Alessandro, P.E.

Region B

Choked Nozzle / Unchoked Exit

degussa.

Technology & Process

Analytical Solution

$$\frac{G_e^2}{P_0 \rho_e} = k \frac{A_n^2}{A_e^2} \frac{P_0}{P_e} \left\{\frac{P_e}{P_*}\right\}^{\frac{k-1}{k}} \left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}$$

$$\left\{\frac{P_e}{P_*}\right\}^{\frac{k-1}{k}} = \left[\frac{1}{2} + \frac{1}{2}\frac{P_0}{P_e}\left\{\frac{P_e^2}{P_0^2} + 2(k-1)\frac{A_n^2}{A_e^2}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}\right\}^{\frac{1}{2}}\right]^{-1}$$

$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{G_e^2}{P_0 \rho_e} + \frac{A_e}{A_n}\left\{\frac{P_e - P_b}{P_0}\right\}$$

$$\frac{T_R}{P_0 A_n} = \frac{A_e}{A_n}\frac{G_e^2}{P_0 \rho_e}$$

$$P_e = P_b$$

Simpson Equation for Region B

$$\frac{T_R}{P_0 A_n} = \frac{2k \dfrac{A_n}{A_e}\left\{\dfrac{2}{k+1}\right\}^{\frac{k+1}{k-1}}}{\dfrac{P_b}{P_0} + \left[\dfrac{P_b^2}{P_0^2} + 2(k-1)\dfrac{A_n^2}{A_e^2}\left\{\dfrac{2}{k+1}\right\}^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}}$$

Region C

Unchoked Nozzle / Unchoked Exit

degussa.

Technology & Process

$$G_e A_e = G_n A_n$$

Continuity

$$\frac{P_0}{\rho_0^k} = \frac{P_n}{\rho_n^k}$$

Isentropic Flow

$$G_n^2 = P_0 \rho_0 \left\{\frac{2k}{k-1}\right\} \left\{ \left(\frac{P_n}{P_0}\right)^{\frac{2}{k}} - \left(\frac{P_n}{P_0}\right)^{\frac{k+1}{k}} \right\}$$

Nozzle Flow

$$G_n^2 = \frac{2kP_n\rho_n}{k-1} \left\{\frac{\rho_n}{\rho_*} - 1\right\}$$

Adiabatic Flow

$$\frac{P_*}{\rho_*^k} = \frac{P_e}{\rho_e^k}$$

Isentropic Flow

$$G_e^2 = P_* \rho_* \left\{\frac{2k}{k-1}\right\} \left\{ \left(\frac{P_e}{P_*}\right)^{\frac{2}{k}} - \left(\frac{P_e}{P_*}\right)^{\frac{k+1}{k}} \right\}$$

Exit Flow

$$P_n = P_* \qquad P_e = P_b$$

Region C Conditions

8 Equations
8 Unknowns

P_n P_* P_e
ρ_n ρ_* ρ_e
G_n G_e

Solve in Terms of Known Parameters

P_0 P_b A_e A_n k

Region C

Unchoked Nozzle / Unchoked Exit

degussa.

Technology & Process

Analytical Solution Not Found

Assume Linear Relationship

$$\frac{\dfrac{T_R}{P_0 A_n} - y_1}{\dfrac{P_b}{P_0} - x_1} = \frac{y_2 - y_1}{x_2 - x_1}$$

$$\left.\begin{array}{r} T_R = 0 \quad when \quad P_b = P_0 \\ \\ T_R\,@\,Boundary\ BC = T_R\,@\,Boundary\ CB \end{array}\right\}$$

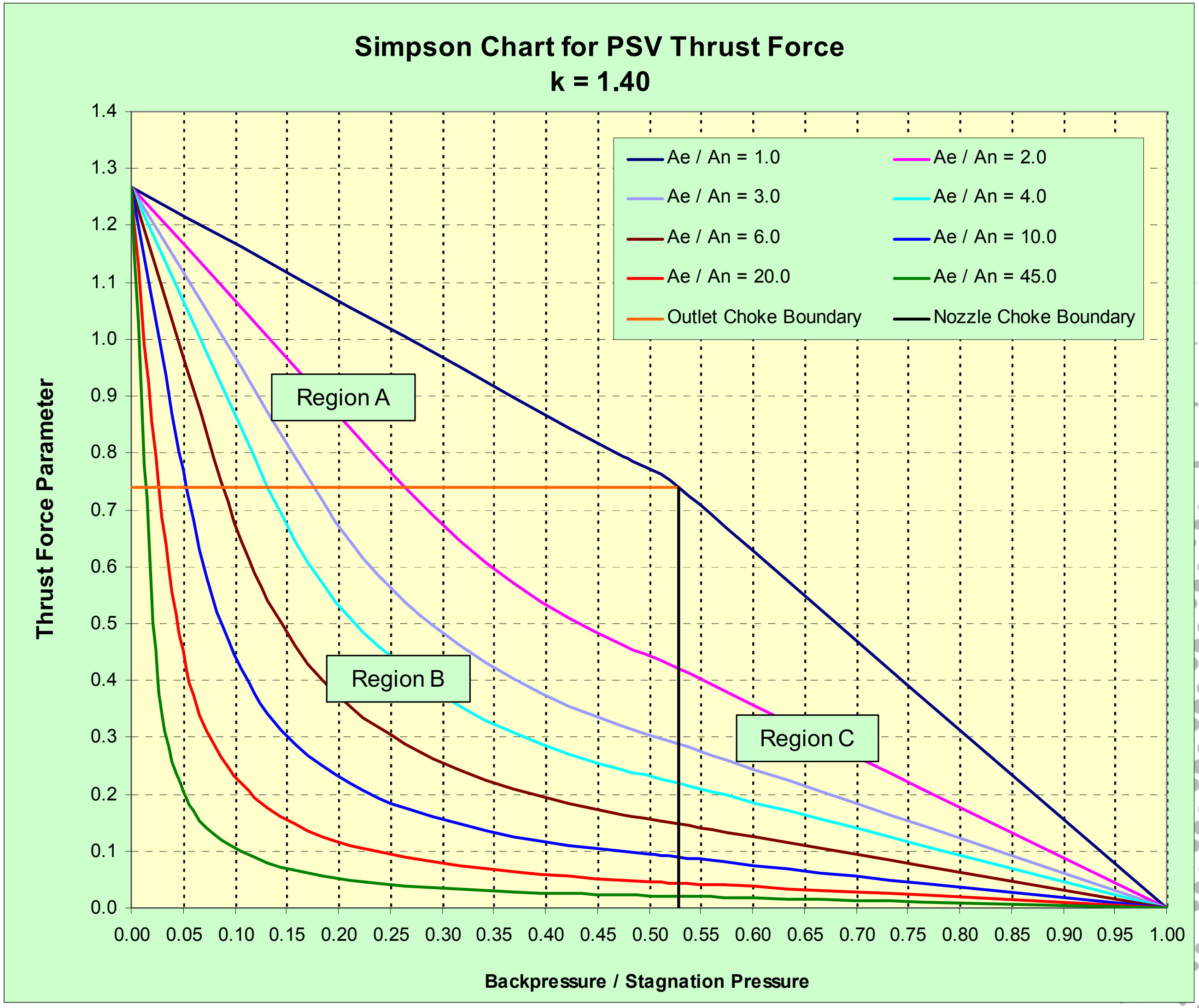
Equation for Region C

$$\frac{T_R}{P_0 A_n} = \frac{2k\dfrac{A_n}{A_e}\left\{\dfrac{2}{k+1}\right\}^{\frac{1}{k-1}}\left\{\dfrac{P_b}{P_0} - 1\right\}}{\left\{1 + \left[1 + 2(k-1)\dfrac{A_n^2}{A_e^2}\left\{\dfrac{2}{k+1}\right\}^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}\right\}\left[\left\{\dfrac{2}{k+1}\right\}^{\frac{k}{k-1}} - 1\right]}$$

Simpson Chart - Revisited

degussa.

Technology & Process

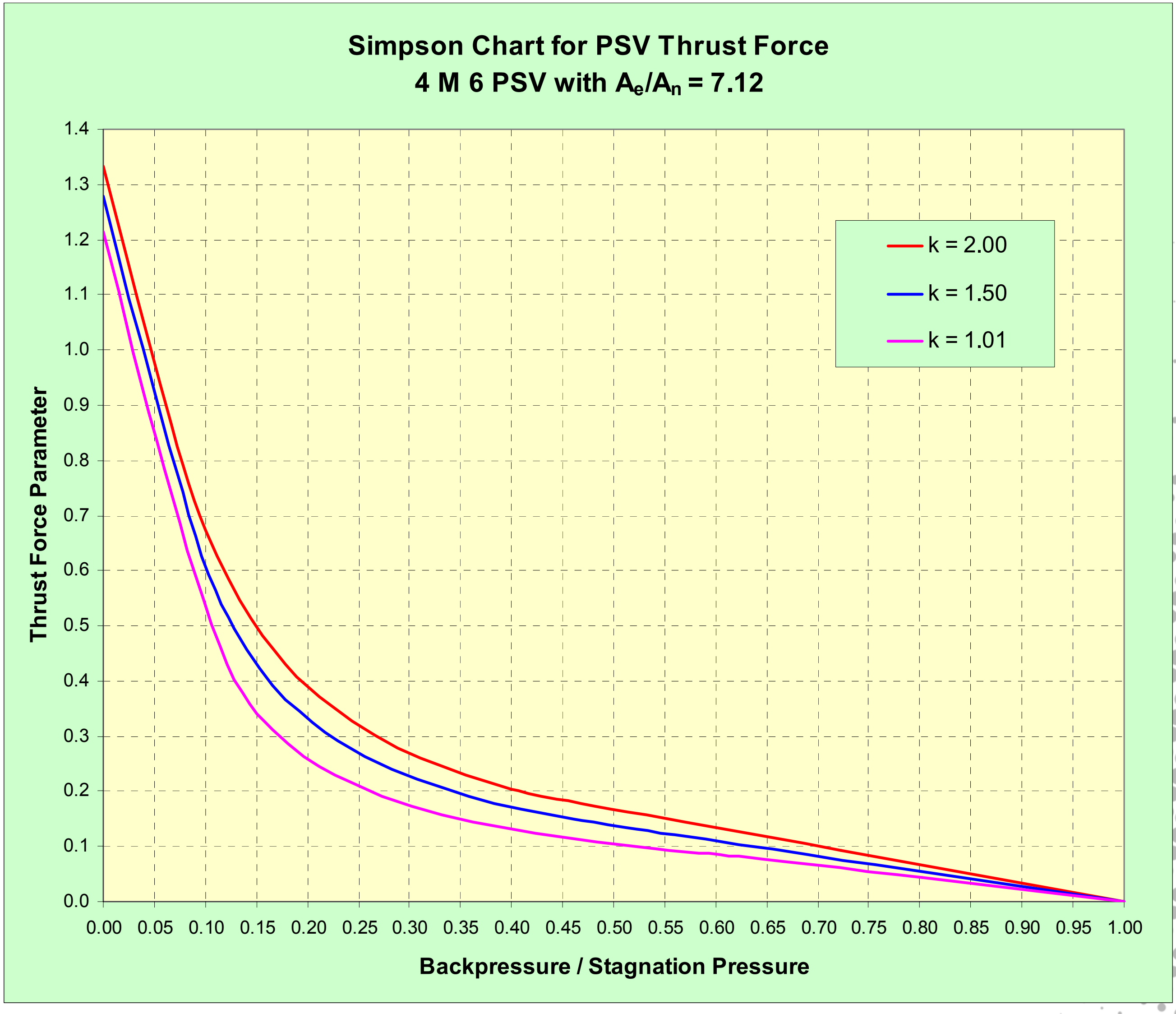


Robert D'Alessandro, P.E.

Simpson Chart - Revisited

degusso.

Technology & Process



Robert D'Alessandro, P.E.

Boundary AB Conditions

degussa.

Technology & Process

Choking versus No Choking at Exit

$$\frac{P_e}{P_0} = \frac{A_n}{A_e}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

$$\frac{P_b}{P_0} < \frac{P_e}{P_0}$$

Choked at Exit – Operation in Region A

$$\frac{P_b}{P_0} \geq \frac{P_e}{P_0}$$

Unchoked at Exit – Operation in Regions B or C

$$\frac{T_R}{P_0 A_n} = k\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} \quad when\ P_e = P_b$$

Horizontal Line Between Regions A & B

Boundary BC Conditions

degussa.

Technology & Process

Choking versus No Choking at Nozzle

Deviation from Simpson Analysis

Simpson Uses P_b to Determine Choking Condition

$$\frac{P_b}{P_0} < or > \frac{P_n}{P_0}$$

$$\frac{P_n}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

$$\frac{P_*}{P_0} < \frac{P_n}{P_0}$$

Choked at Nozzle – Operation in Region B

$$\frac{P_*}{P_0} \geq \frac{P_n}{P_0}$$

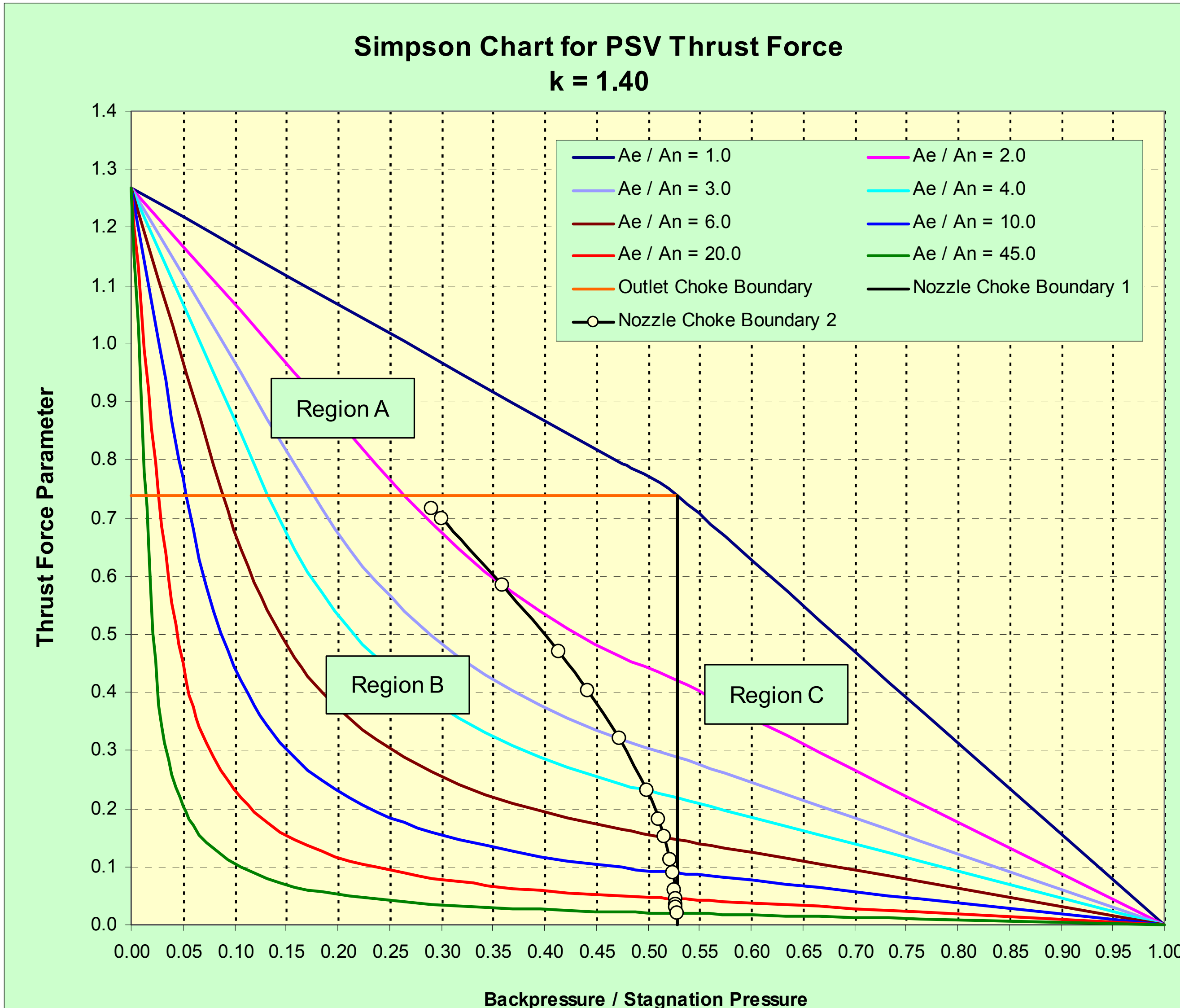
Unchoked at Nozzle – Operation in Regions C

$$\left\{\frac{P_*}{P_0}\right\}^{\frac{k-1}{k}} = \frac{1}{2}\left\{\frac{P_b}{P_0}\right\}^{\frac{k-1}{k}} + \frac{1}{2}\left\{\frac{P_b}{P_0}\right\}^{-\frac{1}{k}}\left[\frac{P_b^2}{P_0^2} + 2(k-1)\frac{A_n^2}{A_e^2}\left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}\right]^{\frac{1}{2}}$$

Simpson Chart - Revisited

degussa.

Technology & Process



Typical PSV Characteristics

degussa.
Technology & Process

Orifice Designation	Typical Actual Orifice Area in^2	API Standard Orifice Area in^2	Ratio Standard to Actual unitless	Nominal Inlet Size inches	Nominal Outlet Size inches	Inlet Inside Diameter (Sch 40) inches	Orifice / Inlet Area Ratio unitless	Outlet Inside Diameter (Sch 40) inches	Orifice / Outlet Area Ratio unitless	Outlet / Orifice Area Ratio unitless
D	0.131	0.110	0.840	1.0	2.0	1.049	0.152	2.067	0.039	25.6
				1.5	2.0	1.610	0.064	2.067	0.039	25.6
				1.5	2.5	1.610	0.064	2.469	0.027	36.5
E	0.223	0.196	0.879	1.0	2.0	1.049	0.258	2.067	0.066	15.0
				1.5	2.0	1.610	0.110	2.067	0.066	15.0
				1.5	2.5	1.610	0.110	2.469	0.047	21.5
F	0.354	0.307	0.867	1.5	2.0	1.610	0.174	2.067	0.105	9.5
				1.5	2.5	1.610	0.174	2.469	0.074	13.5
G	0.567	0.503	0.887	1.5	2.5	1.610	0.279	2.469	0.118	8.4
				2.0	3.0	2.067	0.169	3.068	0.077	13.0
H	0.886	0.785	0.886	1.5	3.0	1.610	0.435	3.068	0.120	8.3
				2.0	3.0	2.067	0.264	3.068	0.120	8.3
J	1.452	1.287	0.886	2.0	3.0	2.067	0.433	3.068	0.196	5.1
				2.5	4.0	2.469	0.303	4.026	0.114	8.8
				3.0	4.0	3.068	0.196	4.026	0.114	8.8
K	2.073	1.838	0.887	3.0	4.0	3.068	0.280	4.026	0.163	6.1
				3.0	6.0	3.068	0.280	6.065	0.072	13.9
L	3.217	2.853	0.887	3.0	4.0	3.068	0.435	4.026	0.253	4.0
				4.0	6.0	4.026	0.253	6.065	0.111	9.0
M	4.060	3.600	0.887	4.0	6.0	4.026	0.319	6.065	0.141	7.1
N	4.894	4.340	0.887	4.0	6.0	4.026	0.384	6.065	0.169	5.9
P	7.195	6.380	0.887	4.0	6.0	4.026	0.565	6.065	0.249	4.0
Q	12.46	11.05	0.887	6.0	8.0	6.065	0.431	7.981	0.249	4.0
R	18.04	16.00	0.887	6.0	8.0	6.065	0.624	7.981	0.361	2.8
				6.0	10.0	6.065	0.624	10.020	0.229	4.4
T	28.93	26.00	0.899	8.0	10.0	7.981	0.578	10.020	0.367	2.7

Robert D'Alessandro, P.E.

API 520 Method

degussa.

Technology & Process

English Version

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + A_e P_e$$

T_R in lb_f
W in lb/h
T_0 in °R
A_e in in^2
P_e in psig

Relief Valve Vents to Atmosphere

Metric Version

$$T_R = 129\,W\sqrt{\frac{kT_0}{(k+1)M}} + 0.1\,A_e P_e$$

T_R in N
W in kg/s
T_0 in K
A_e in mm^2
P_e in barg

$$T_R = W\sqrt{\frac{2kRT_0}{(k+1)M}} + A_e(P_e - P_b)$$

T_R in N
W in kg/s
T_0 in K
A_e in m^2
P_e in Pa

R = Gas Constant (8314 Pa m^3 / K kmol)

API 520 Method

Comparing to
Basic Thrust Force Equation

$$u_e = \sqrt{\frac{2kRT_0}{(k+1)M}}$$

Adiabatic Flow from
Stagnation to Relief Valve Exit

$$\frac{T_0}{T_e} = \frac{k+1}{2}$$

Therefore

$$u_e = \sqrt{\frac{kRT_e}{M}} = u_s$$

Formula for Speed of
Sound under
Adiabatic Conditions

API 520 Formula Only Applies to Region A

Algebraic Manipulation Converts the API 520 Formula to the
Simpson Formula for Region A

API 520 Does Not Give a Formula for Exit Pressure

$$P_e = \frac{W}{A_e}\sqrt{\frac{2RT_0}{k(k+1)M}}$$

Crosby Method

degussa.
Technology & Process

$$T_R = \frac{CK_d A_n P_0}{332.7}\sqrt{\frac{k}{k+1}} + F_g$$

$$F_g = \left\{\frac{K_d A_n P_0 K_r}{1.383 A_e} - P_b\right\} A_e$$

$$C = 520\sqrt{k\left\{\frac{2}{k+1}\right\}^{\frac{k+1}{k-1}}}$$

T_R in lb_f
P_0 in psia
F_g in lb_f
W in lb/h
A_n in in^2
A_e in in^2
P_b in psia

A_n = API Effective Nozzle Area

If $F_g < 0$, then $F_g = 0$

K_r –Function of k only
Tabulated Values

Using ASME Nozzle Area rather than API Effective Nozzle Area

Assume $K_d = 1.00$

$$W = CA_n P_0\sqrt{\frac{M}{T_0}} \quad \Rightarrow \quad CA_n P_0 = W\sqrt{\frac{T_0}{M}}$$

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + \frac{K_r A_n P_0}{1.521} - A_e P_b$$

Crosby Method

degussa.
Technology & Process

Comparison with API 520 formula implies

$$\frac{K_r A_n P_0}{1.521} - A_e P_b = A_e\left(P_e - P_b\right) \quad \Rightarrow \quad K_r = 1.521\frac{A_e}{A_n}\frac{P_e}{P_0}$$

From Simpson derivation for Region A

$$\frac{A_e}{A_n}\frac{P_e}{P_0} = \left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

To match tabulated values, a 1.25 factor is needed

$$K_r = (1.25)(1.521)\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} = 1.90\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}}$$

Reformulated Crosby Equation

$$T_R = \frac{W}{366}\sqrt{\frac{kT_0}{(k+1)M}} + 1.25 P_0 A_n\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - A_e P_b$$

Replace *W* and convert to dimensionless form

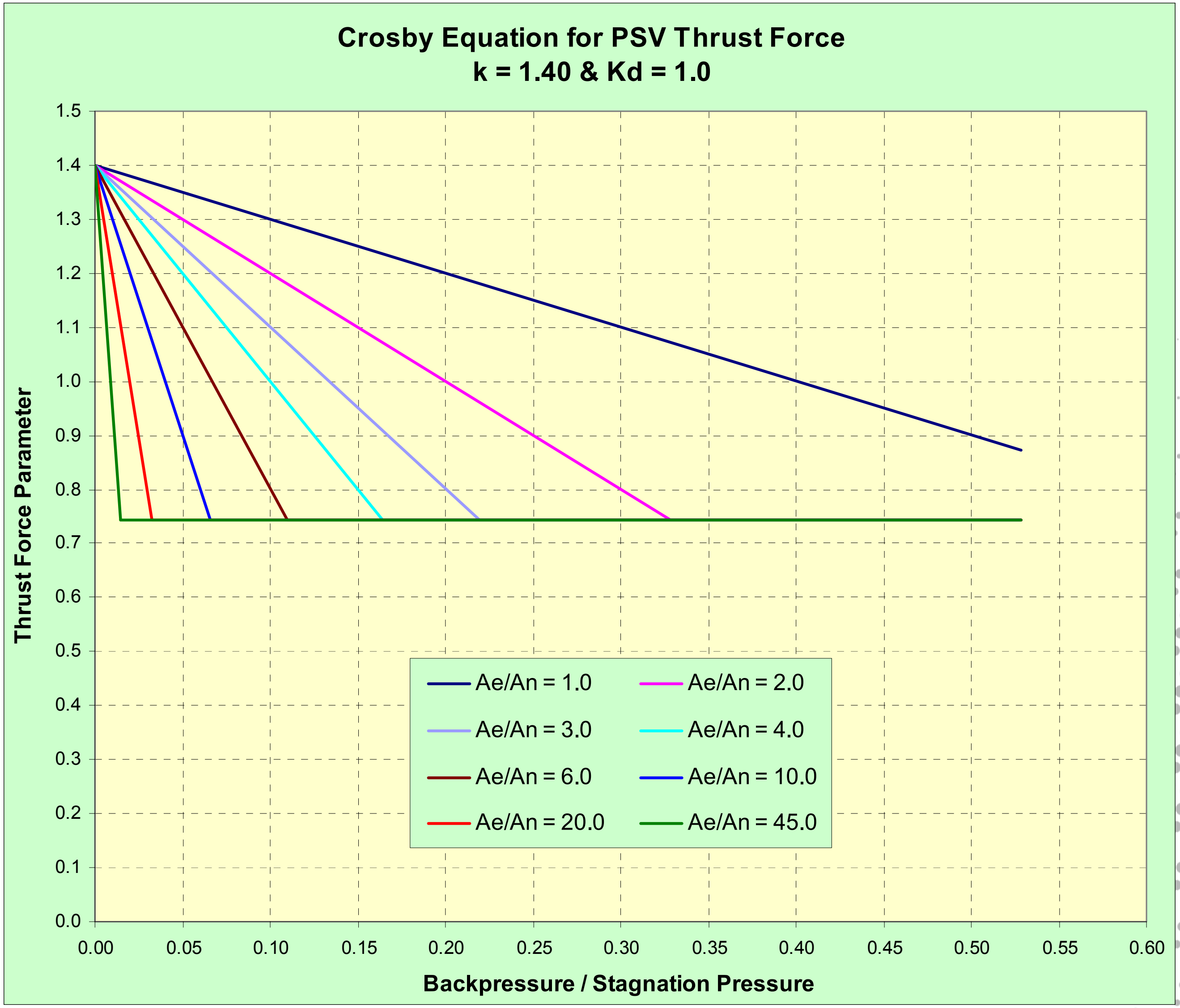
$$\frac{T_R}{A_n P_0} = \left\{\frac{520k}{366\sqrt{2}} + 1.25\right\}\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - \frac{A_e P_b}{A_n P_0} = (k + 1.25)\left\{\frac{2}{k+1}\right\}^{\frac{k}{k-1}} - \frac{A_e P_b}{A_n P_0}$$

Region A Only

Crosby Method

degussa.

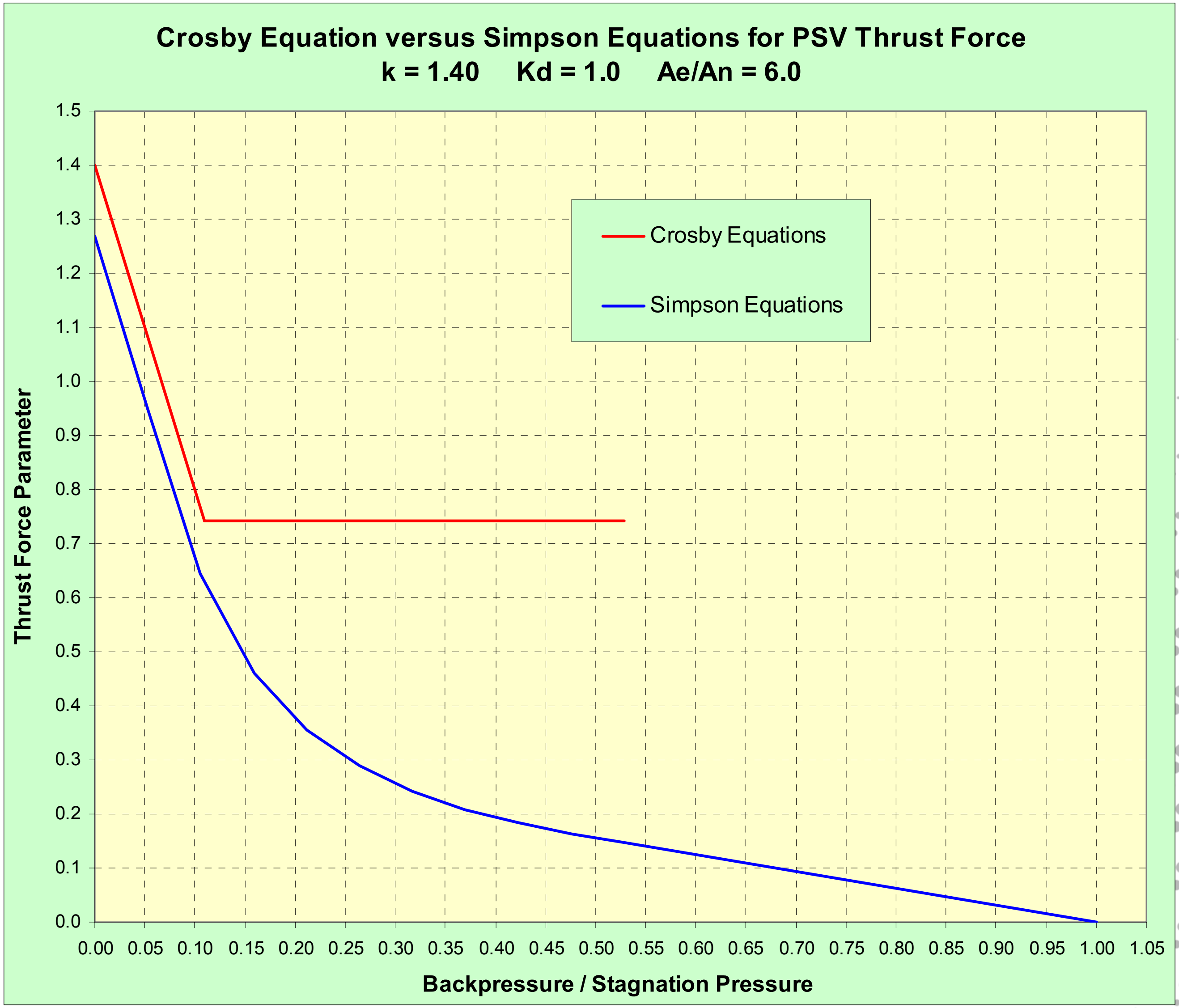
Technology & Process



Crosby Method

degussa.

Technology & Process



Summary of What is Out There

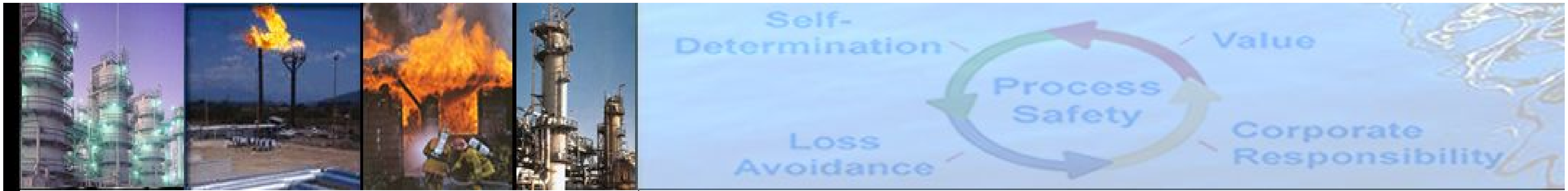
degussa.

Technology & Process

Reference	Region A	Region B	Region C	Remarks
DIERS Project Manual	Correct Equation	Graph Only	Not Included	
DIERS Project Manual Errata	Incorrect Equation	Incorrect Equation	Not Included	Probably a Typos
CCPS Guidelines for ERS	Graph Only	Graph Only	Not Included	
HSE Workbook	Incorrect Equation	Not Included	Not Included	Probably a Typo
API 520	Correct Equation	Not Included	Not Included	No Equation for Exit Pressure
Crosby Handbook	Correct Equation	Not Included	Not Included	Includes 25% Safety Factor

Concluding Remarks

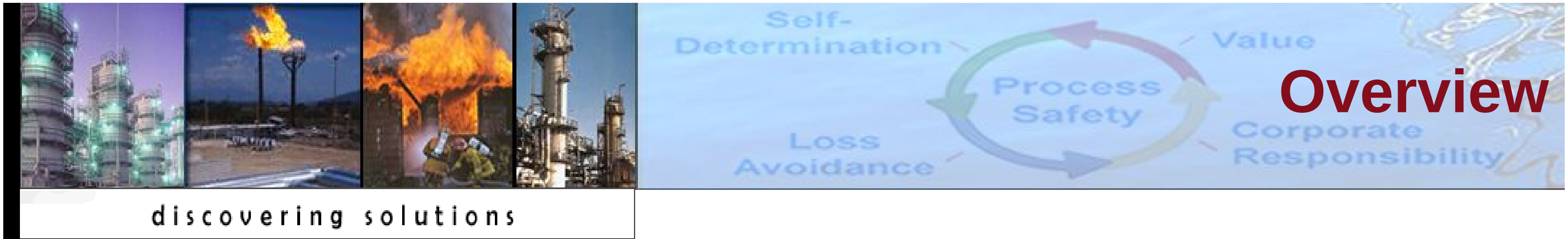
- Dynamic Load Factor (DLF)
- Better Model for Region C
- Extensions to Two-Phase Flow
- Applications to "Body-Bowl" Choking



discovering solutions

Practical Guidelines for Dealing with Excessive Pressure Drop in Relief Systems

Georges A. Melhem, ioMosaic Corporation
and
Harold G. Fisher, Fisher Inc.



Overview

- A large fraction of existing relief device installations suffer from excessive pressure loss (inlet and/or outlet)
- Excessive pressure loss can lead valve instability and possibly valve failure
- Operating companies are faced with significant upgrade/mitigation costs
- We examine key factors leading to valve instability and provide some practical guidance on mitigation options

ioMosaic

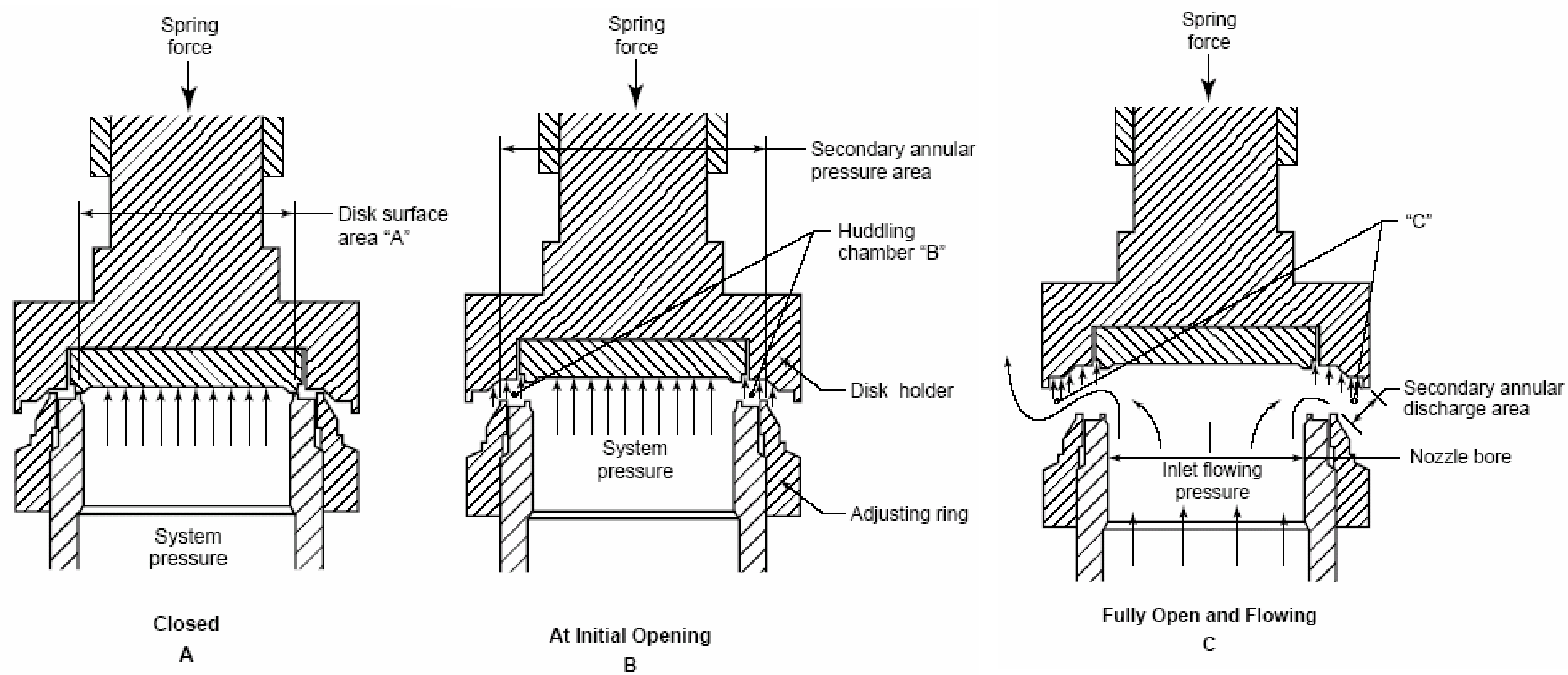
discovering solutions

Chatter, Flutter, Simmering, and Blowdown

- **Chatter is an abnormal rapid reciprocating motion of the movable parts of a pressure relief valve in which the disc contacts the seat**
- **The impact on the seat is usually very strong, and therefore can damage the valve. Chatter can occur in either vapor or liquid service**
- **Flutter is the same reciprocating motion, but the disc does not contact the seat**
- **Simmering is the audible or visible escape of fluid between the seat and disc at an inlet static pressure below the popping pressure and at no measurable capacity**
- **Blowdown is the difference between the popping pressure and reseating pressure expressed either as a percentage of the popping pressure or in pressure units**

Relief Valve Model: Force Dynamics During Flow

discovering solutions



Pressure relief valve Operation: Vapor/Gas Service. Source API RP 520

ioMosaic

Relief Valve Model: Force Dynamics During Relief

discovering solutions

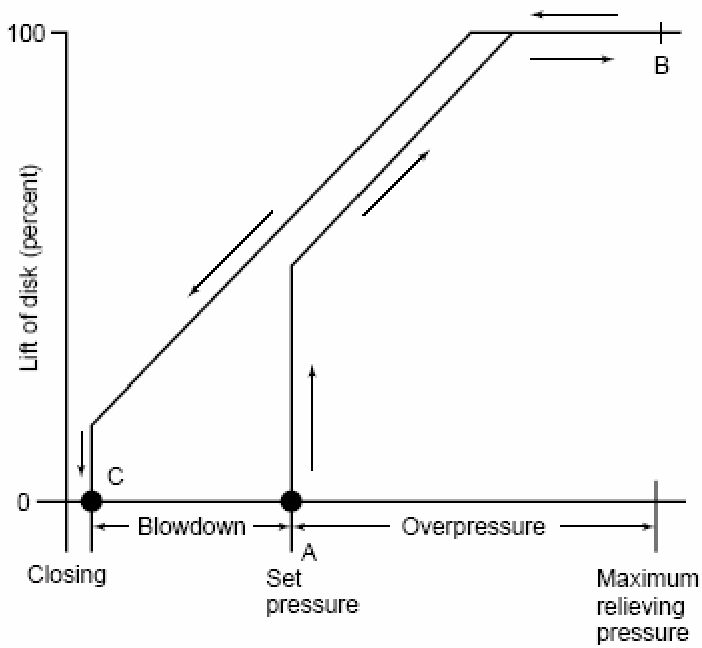
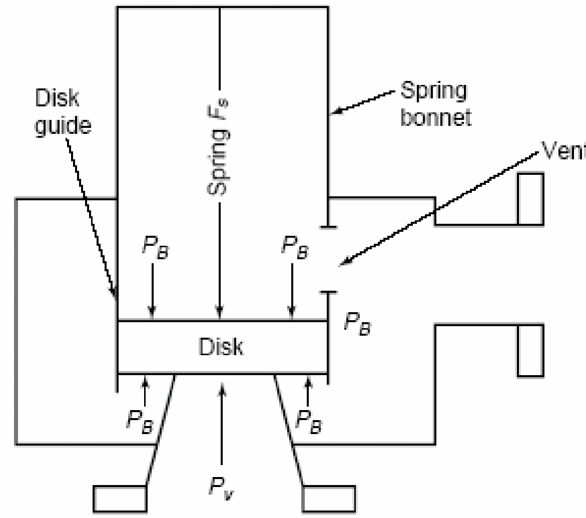


Figure 20—Typical Relationship Between Lift of Disk in a Pressure Relief Valve and Vessel Pressure

Spring Bonnet
Vented to Valve Discharge



Back pressure increases set pressure

$$P_V A_N = F_S + (P_B A_N)$$

$A_D > A_N$

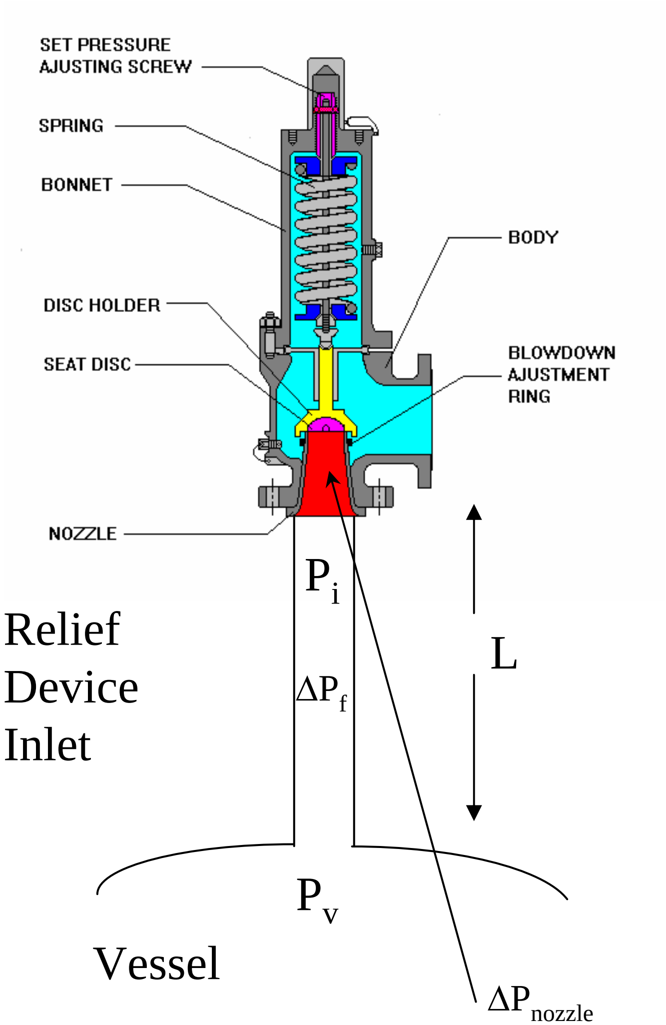
A_D = disk area,
A_N = nozzle seat area,
F_S = spring force,
P_V = vessel pressure in pounds per square inch gauge,
P_B = superimposed back pressure, in pounds per square inch gauge.

Figure 22—Typical Effects of Superimposed Back Pressure on the Opening Pressure of Conventional Pressure Relief Valves

Source API RP 520

discovering solutions

It is Not a Perfect Valve!



$$t_{Chatter} \approx 2\frac{L}{c_o}$$

$$\text{Blowdown} \approx \Delta P_f + (\Delta P_{nozzle} + \Delta P_{safety\,margin})$$

$$7\% \qquad \approx 3\% + 4\%$$

For short pipe lengths, $t_{chatter}$ may be smaller than the actual valve opening or closing time

The valve can close due to excessive inlet pressure loss or excessive backpressure development

Co = speed of sound in fluid

ioMosaic

It is Not a Perfect Valve

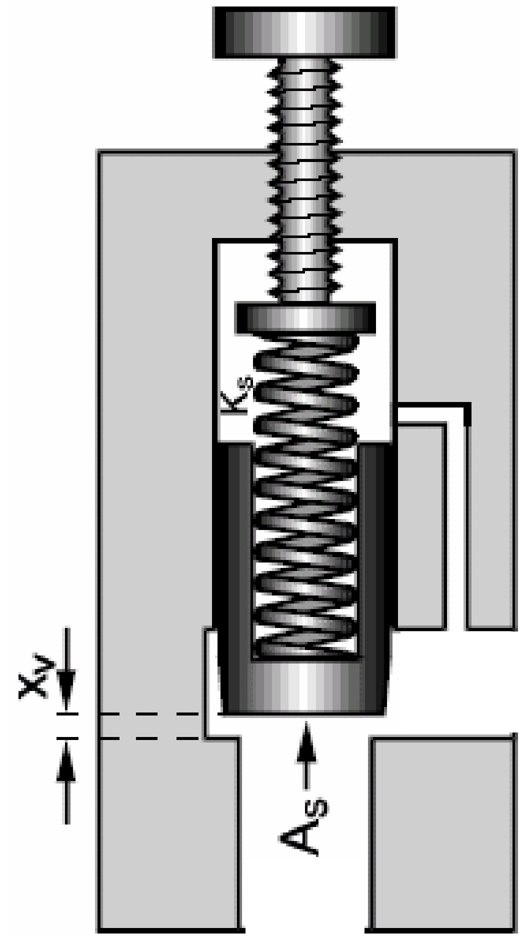
- **The momentum exchange between the fluid impinging on disk surface and the disk is an important factor**
- **If the fluid characteristic chatter time is less than the valve closure time, the valve will flutter and not chatter**
- **If the fluid characteristic chatter time more than the valve closure time, the valve will chatter**
- **If the fluid characteristic chatter time is equal to the valve closure time, the valve will chatter with increased severity due to acoustic coupling**

Relief Valve Model: Set Pressure

discovering solutions

$$P_s = \frac{F_{s_0} \pm F_r}{A_s} = \frac{k_s x_0 \pm F_r}{A_s}$$

- P_s = Set pressure
- F_{s0} = Spring force preload
- k_s = Spring rate
- x_0 = Pre-compressed spring length
- F_r = Coulomb friction force. F_r acts in the direction opposite to that of spool motion. F_r contributes to the hysteresis exhibited by the valve during opening and reseating.
- A_s = Spool area normal to flow



ioMosaic

Relief Valve Model: Force Balance During Flow

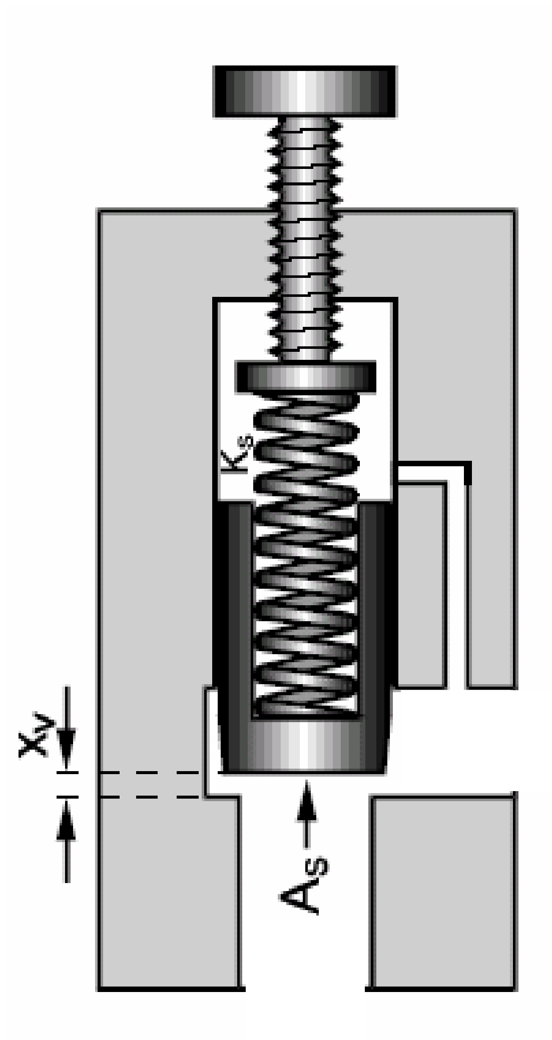
$$P_V = \frac{k_s(x_0 + x_V) + F_{sf} \pm F_r}{A_s}$$

$$F_{sf} = C_f P_V x_V = 2\pi D C_d C_V \cos\phi$$

$$P_V = \left(P_s + \frac{k_s x_V}{A_s}\right)\left(\frac{A_s}{A_s - C_f x_V}\right)$$

$$P_V \approx P_s \text{ if } \frac{k_s x_V}{A_s} \approx 0 \text{ and if } C_f x_V \text{ is negligible}$$

- **F_{sf} is the steady state flow force, D is the spool diameter, C_v is the flow velocity coefficient, C_f is the flow force area gradient, Φ is the flow jet angle, C_d is the flow discharge coefficient, and P_v is the vessel pressure**



ioMosaic

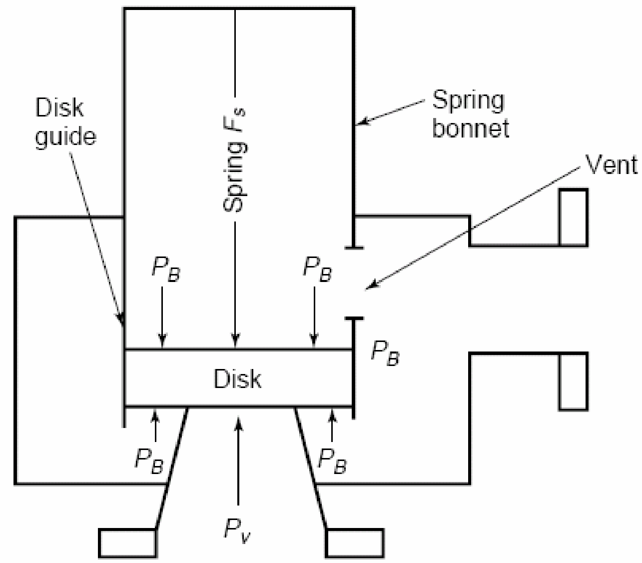
Relief Valve Model: Force Balance During Flow

discovering solutions

$$P_V A = k_s(x_0 + x_V) + P_B A_D - P_B(A_D - A) \pm F_r + m\frac{d}{dt}\left(\frac{dx_V}{dt}\right) + b\frac{dx_V}{dt}$$

$$\frac{dA}{dt} = f(x_V, P_V, P_B, \ldots) \text{ at t} = 0, A = A_N, \text{at t} = \text{t}_{\text{fully open}}, A = A_D$$

- **m is the spool mass plus 1/3 mass of spring**
- **b is the viscous damping coefficient**
- **The above equation can be solved numerically if a suitable value for dA_N/dt is provided for valve opening and closing**
- **This is difficult because the pressure profile on the disk surface is non-uniform**
- **These equations can be used to estimate the opening/closing time of the relief device and can be integrated with vessel balances used to establish Pv as a function of time, etc.**



ioMosaic

Relief Valve Model: Force Balance During Flow

discovering solutions

$$\frac{dx_v}{dt} = y$$

$$\frac{dy}{dt} = \left(\frac{1}{m}\right)\left(P_V A - k_s(x_0 + x_V) - P_B A_D + P_B(A_D - A) \pm F_r - b\frac{dx_V}{dt}\right)$$

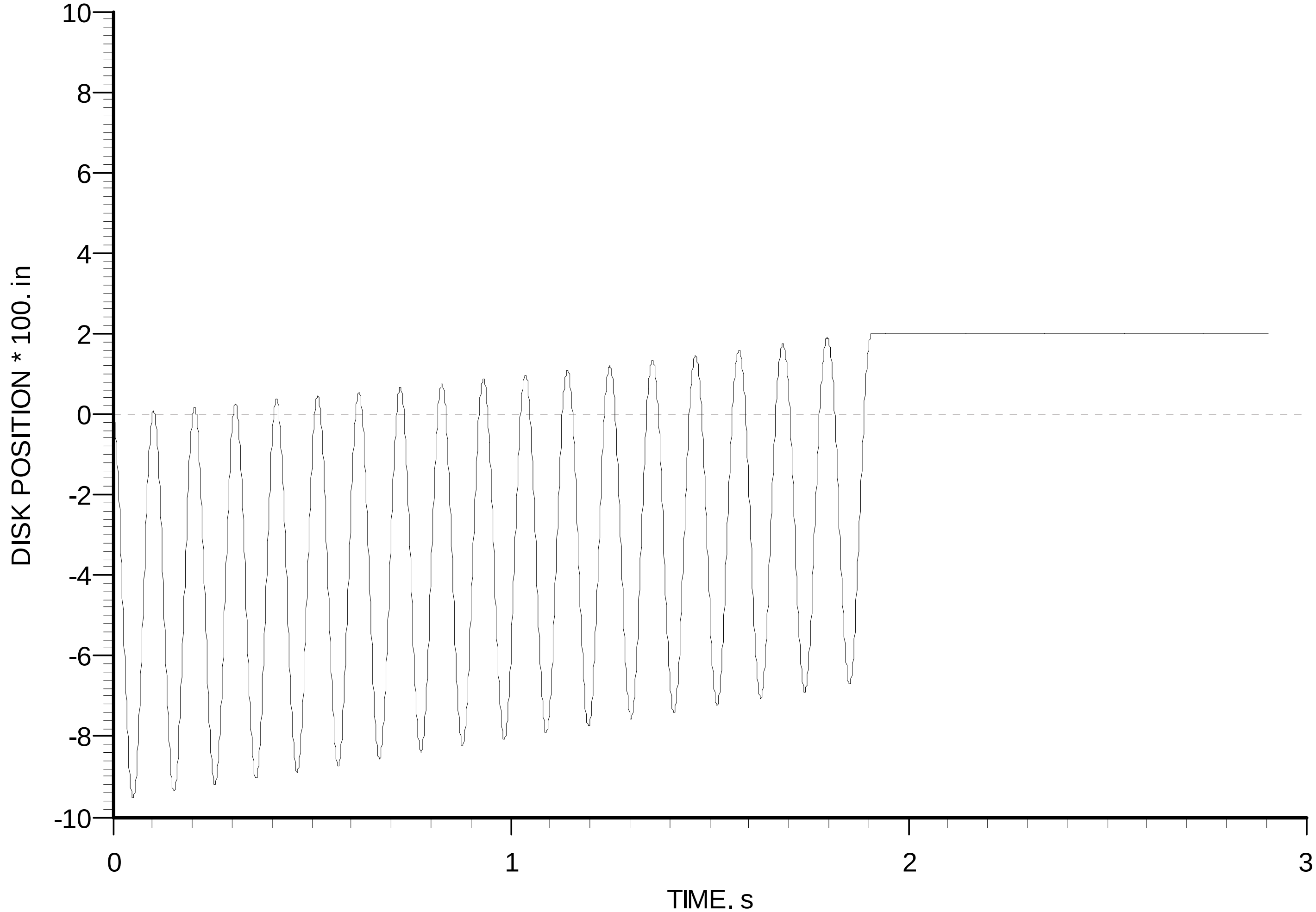
$$\frac{dP_V}{dt} = \text{function of venting dynamics}$$

$$A = A_s + \left|\frac{x_v}{x_{v,\max}}\right|(A_D - A_s)$$

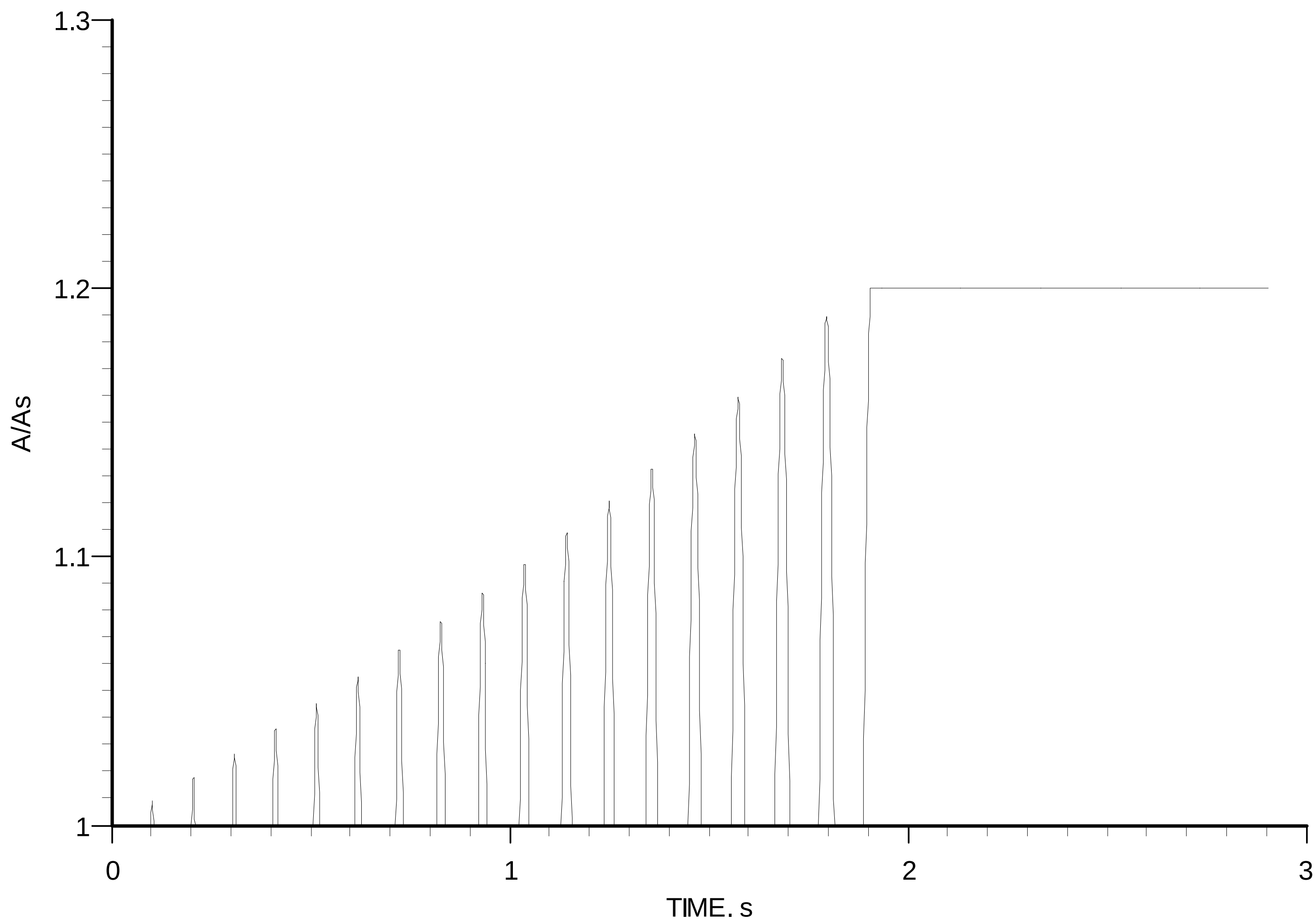
ioMosaic

Sample Calc

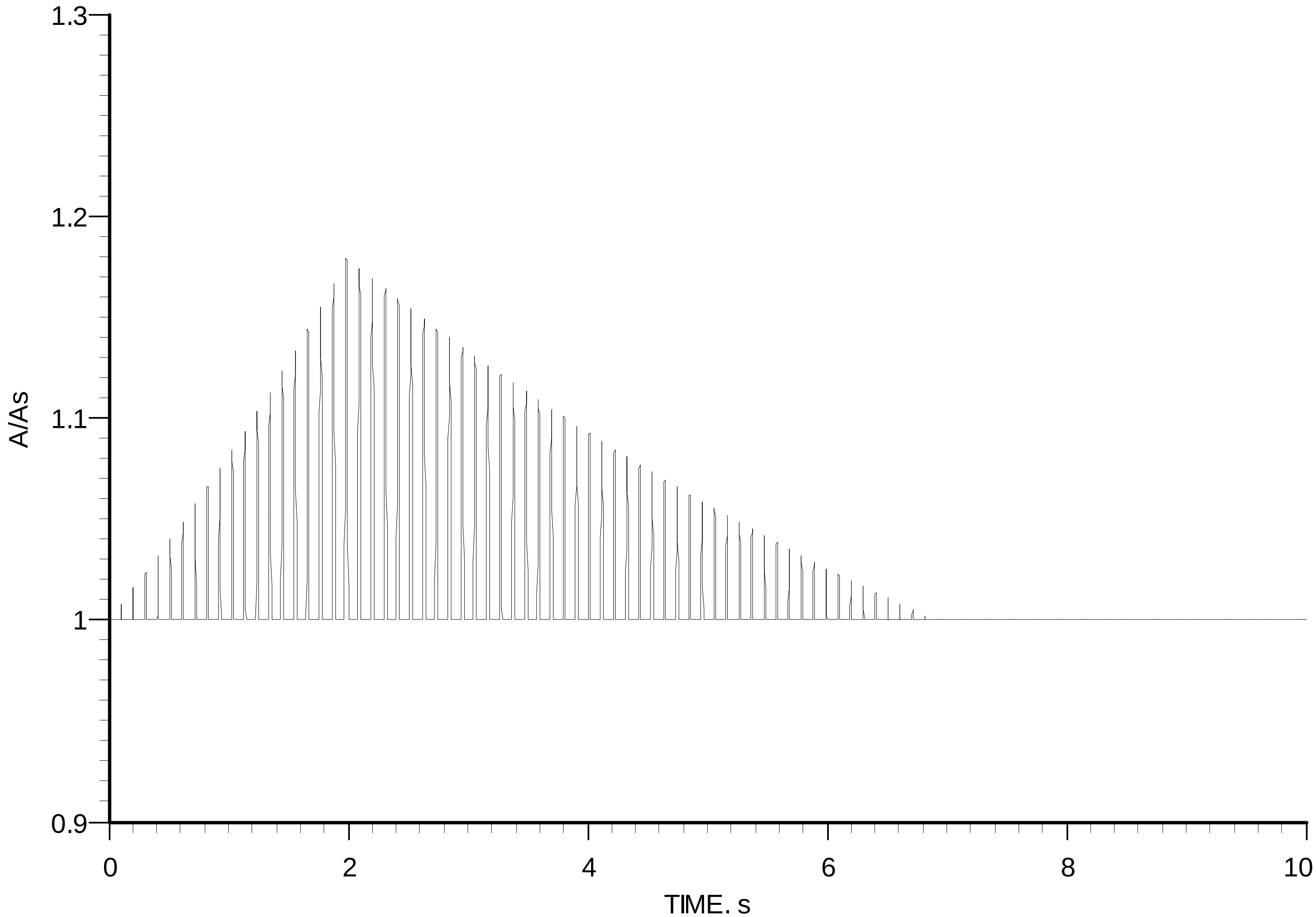
discovering solutions



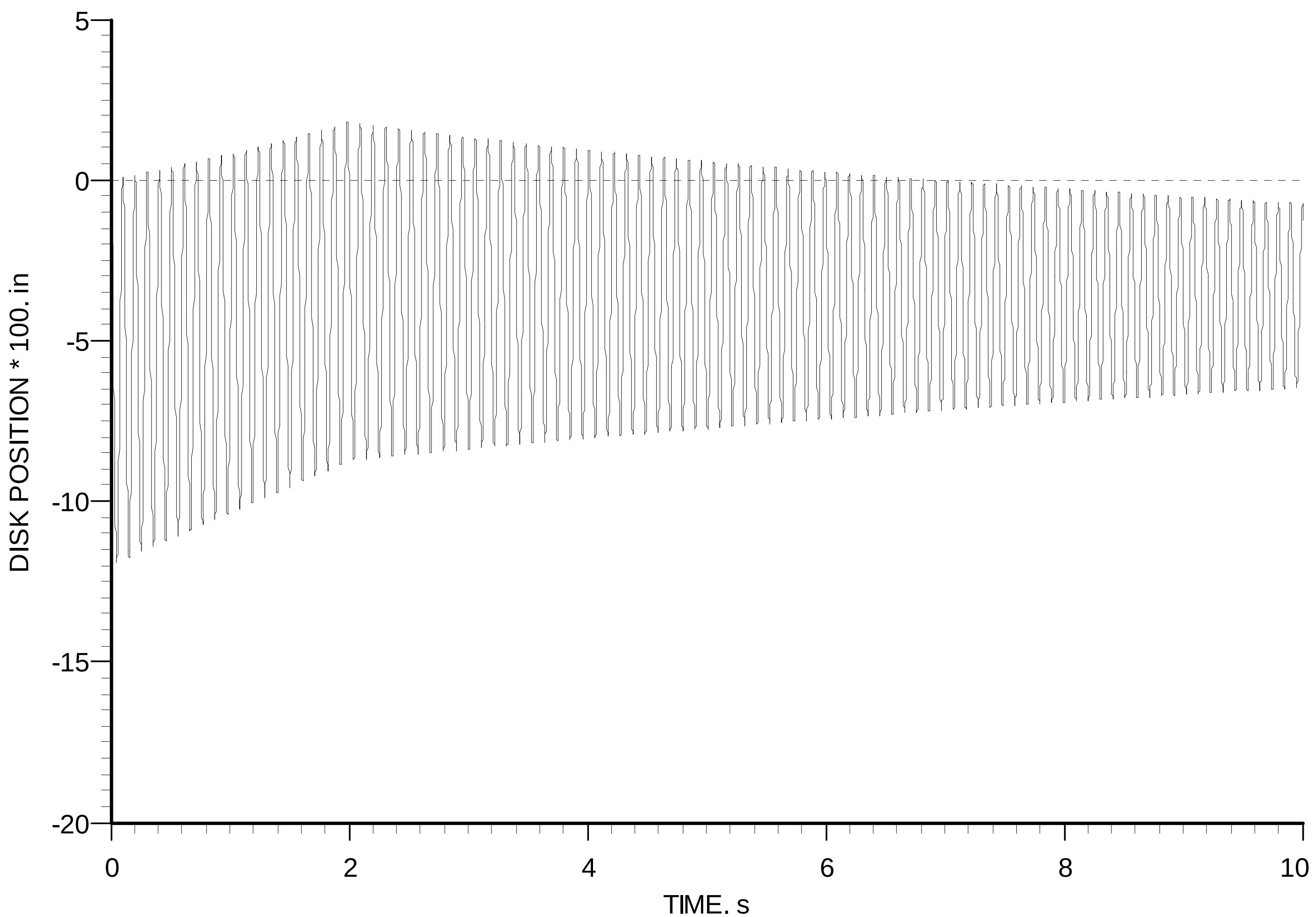
Sample Calc



Sample Calc



Sample Calc



discovering solutions

Typical Causes of Chatter

- **PRV oversized for installation**
  - » Flow is < 25 % of rated capacity
  - » Valve is handling widely different rates
- **Inlet piping has excessive length.** $P_{inlet} = P_{vessel} - \Delta P_{loss} <= P_{blowdown}$
- **Inlet piping is undersized for PRV (Starving PRV)**
- **Outlet piping has excessive length**
- **Outlet piping is undersized for PRV**
- **Upper adjusting ring too high**

ioMosaic

Chatter Solutions

- **Avoid turns, elbows and sharp area reductions in inlet and outlet lines**
- **Use long radius elbows**
- **Use larger piping**
- **If you cannot change the piping**
  - » Increase blowdown (for example, 5 % inlet loss can be tolerated if blowdown is set at 7 %)
  - » Install a smaller PRV or use a restricted lift valve
  - » Install a different type of PRV (for example, a pilot valve)
- **Use multiple valves with staggered set pressures when the lowest required contingency rate is less than 25% of highest rate**

discovering solutions

Inlet Line Considerations

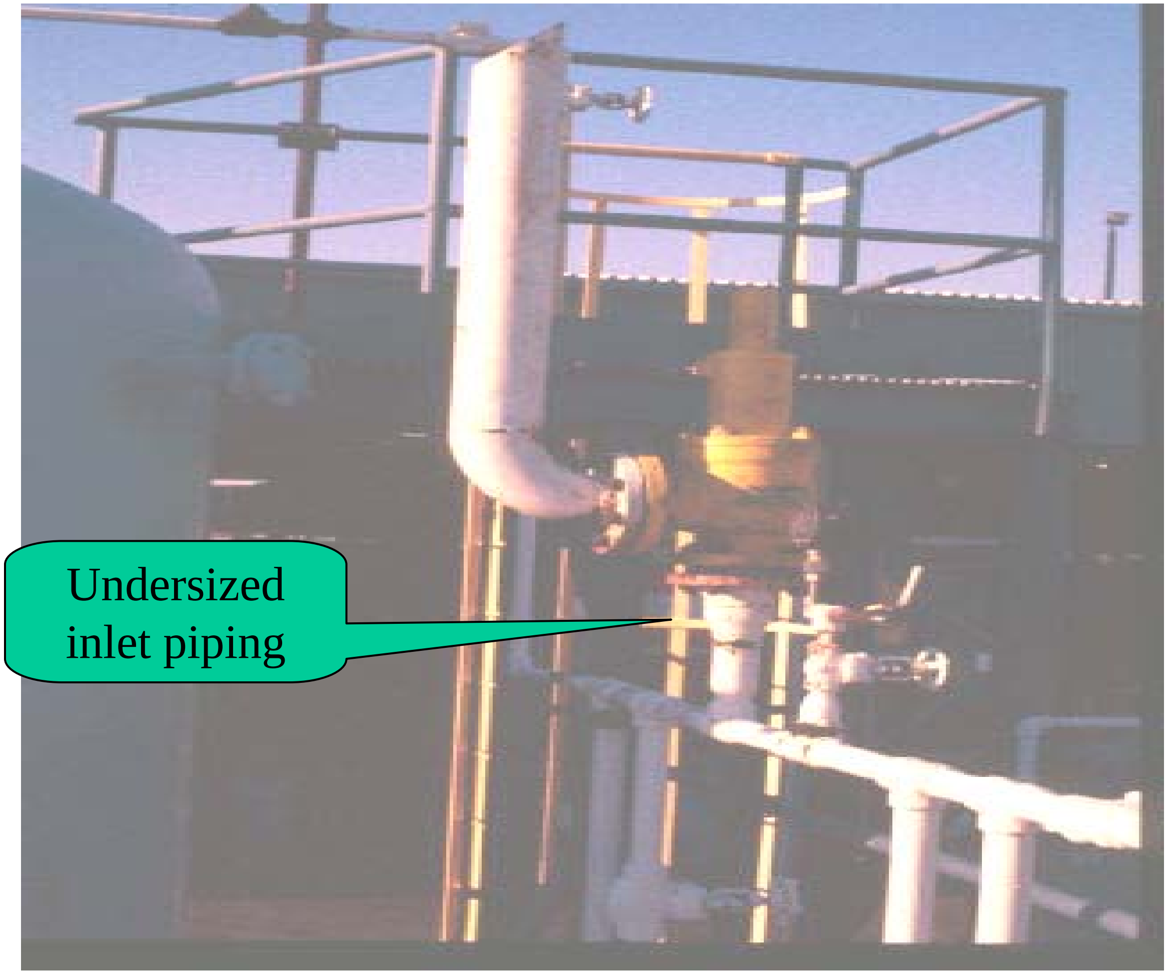
- Inlet line size must be at least equal to PRV inlet flange size
- Inlet piping should slope continuously upward from vessel to avoid traps
- Inlet piping should be heat traced if freezing or congealing of viscous liquids could occur
- A continual clean purge should be provided if coke/polymer formation or solids deposition could occur
- Etc.

ioMosaic

Causes of Chatter

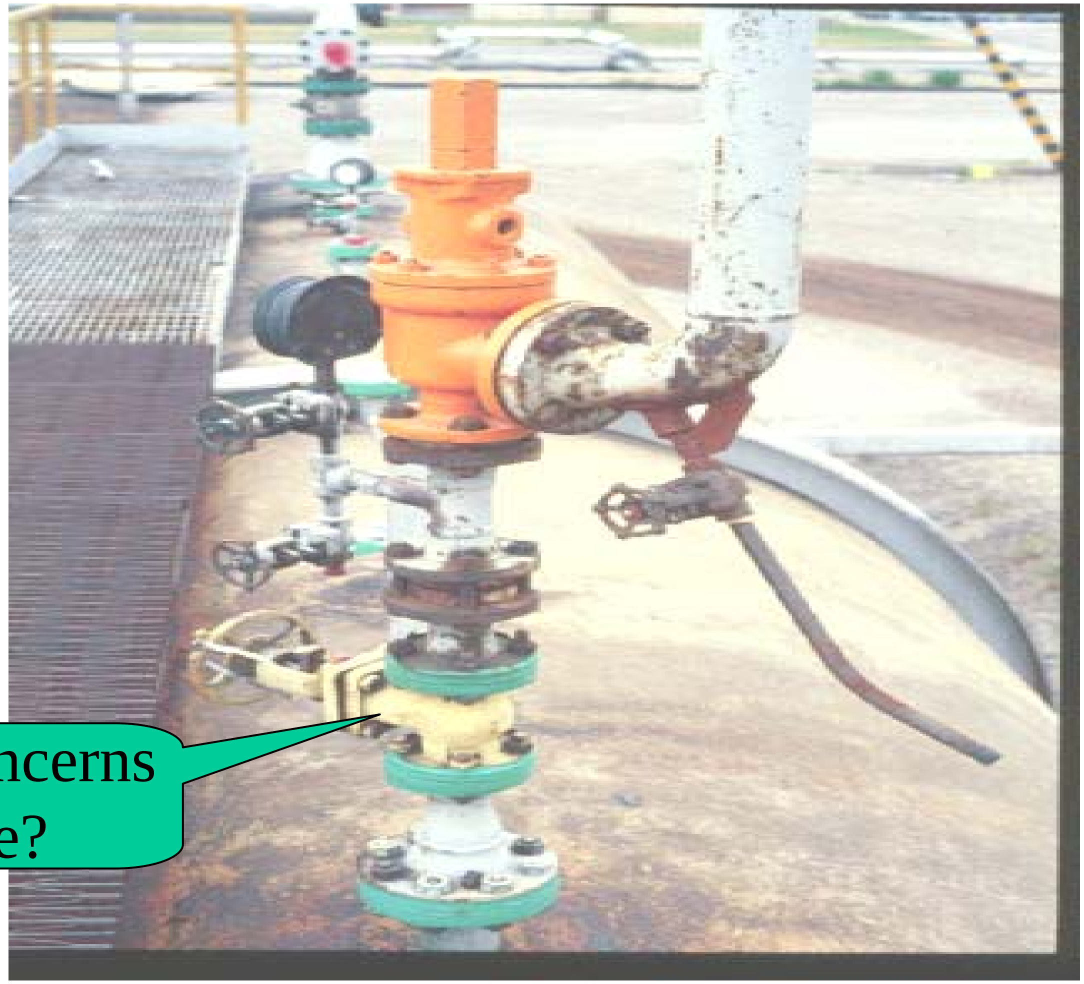
Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Causes of Chatter



Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

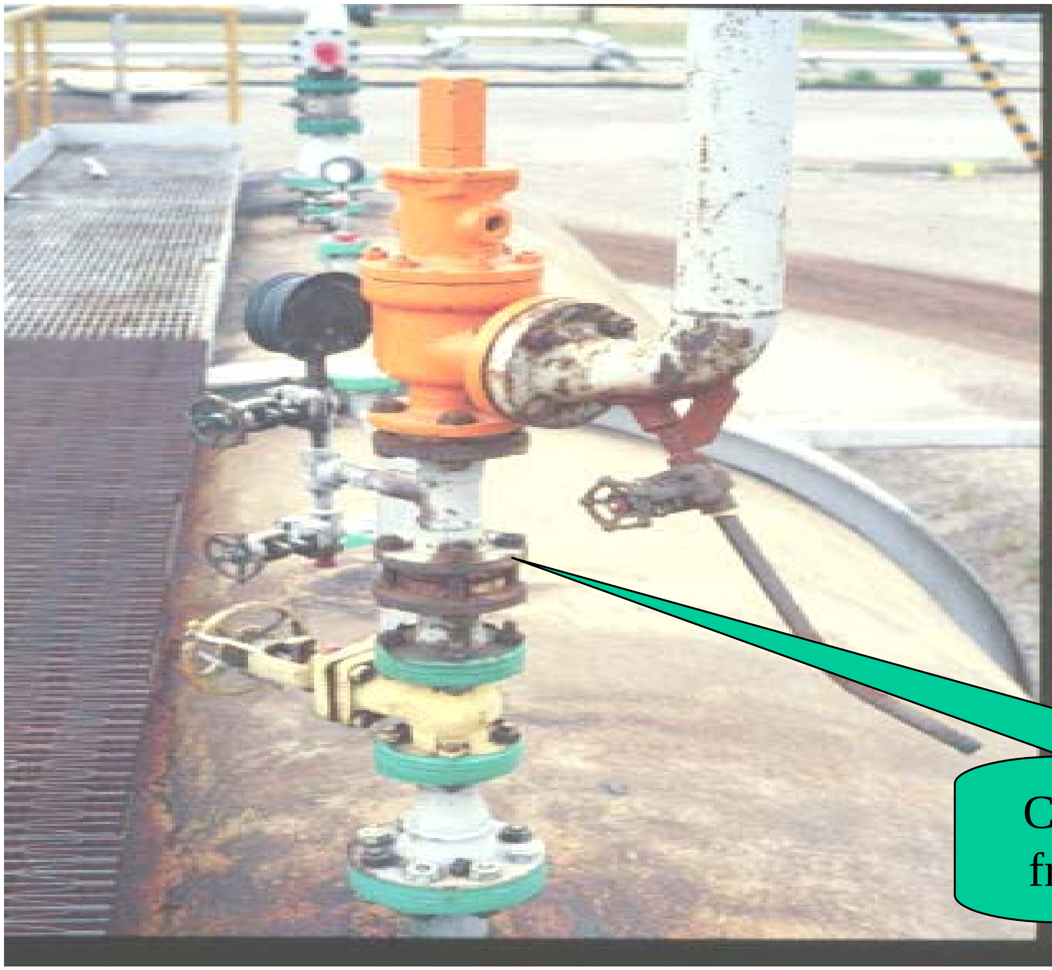
Causes of Chatter



Any concerns here?

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Causes of Chatter



Consider the pressure drop from all these connections

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Outlet Line Considerations

- **Discharge line diameter must be at least equal to PRV outlet flange size**
- **Maximum discharge velocity should not exceed 75% of sonic velocity**
- **For flammable releases to atmosphere, minimum velocity should be no less than 100 ft/sec**
- **Atmospheric risers should discharge at least 10 ft above platforms within 50 ft horizontally**
- **Radiant heat due to ignition of release should be considered**

Outlet Line Considerations

- No check valves, orifice plates or other restrictions permitted
- Atmospheric discharge risers should have drain hole
- Piping design must consider thermal expansion due to hot/cold release
- Auto-refrigeration and need for brittle fracture resistant materials
- Closed discharge piping should slope continuously downward to header to avoid liquid traps
- Etc.

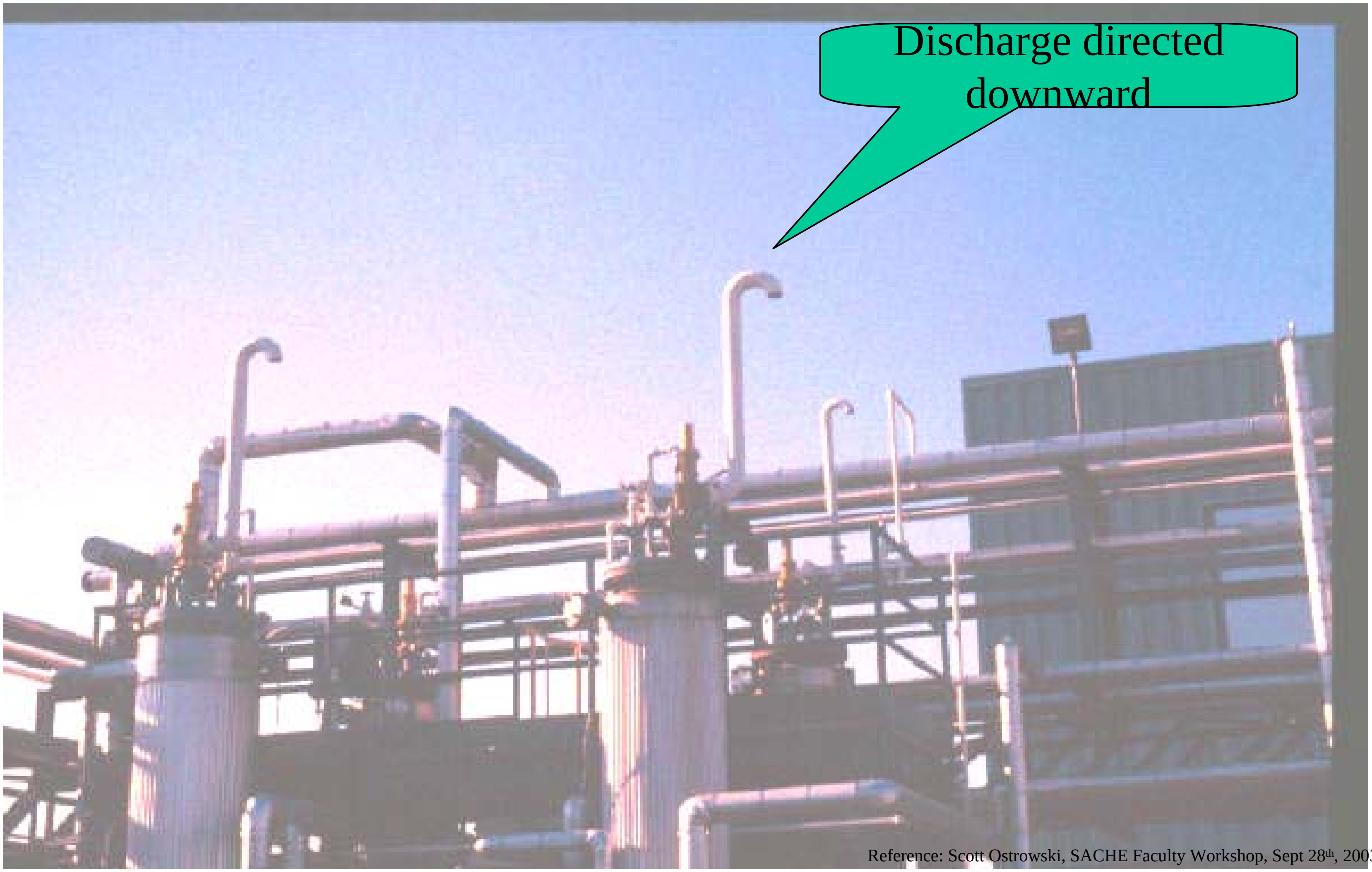
Quick Rules

- **Enlarged inlet pipe diameter is almost always required for:**
  - » 4P6, 6R8, 6R10, and 8T10
  - » All safety valves used in series with rupture disks
  - » 1.5H3, 2J3, 3L4, and 6Q8 with shutoff valve and L/D=5
- **Enlarged outlet piping is almost always required for:**
  - » 6R8 safety valves
  - » Conventional safety valves 3L4, 4P6, and 8T10 with a set pressure > 100 psig and discharge pipe length more than 10 ft

Outlet Line Considerations

discovering solutions

Outlet Line Considerations



Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

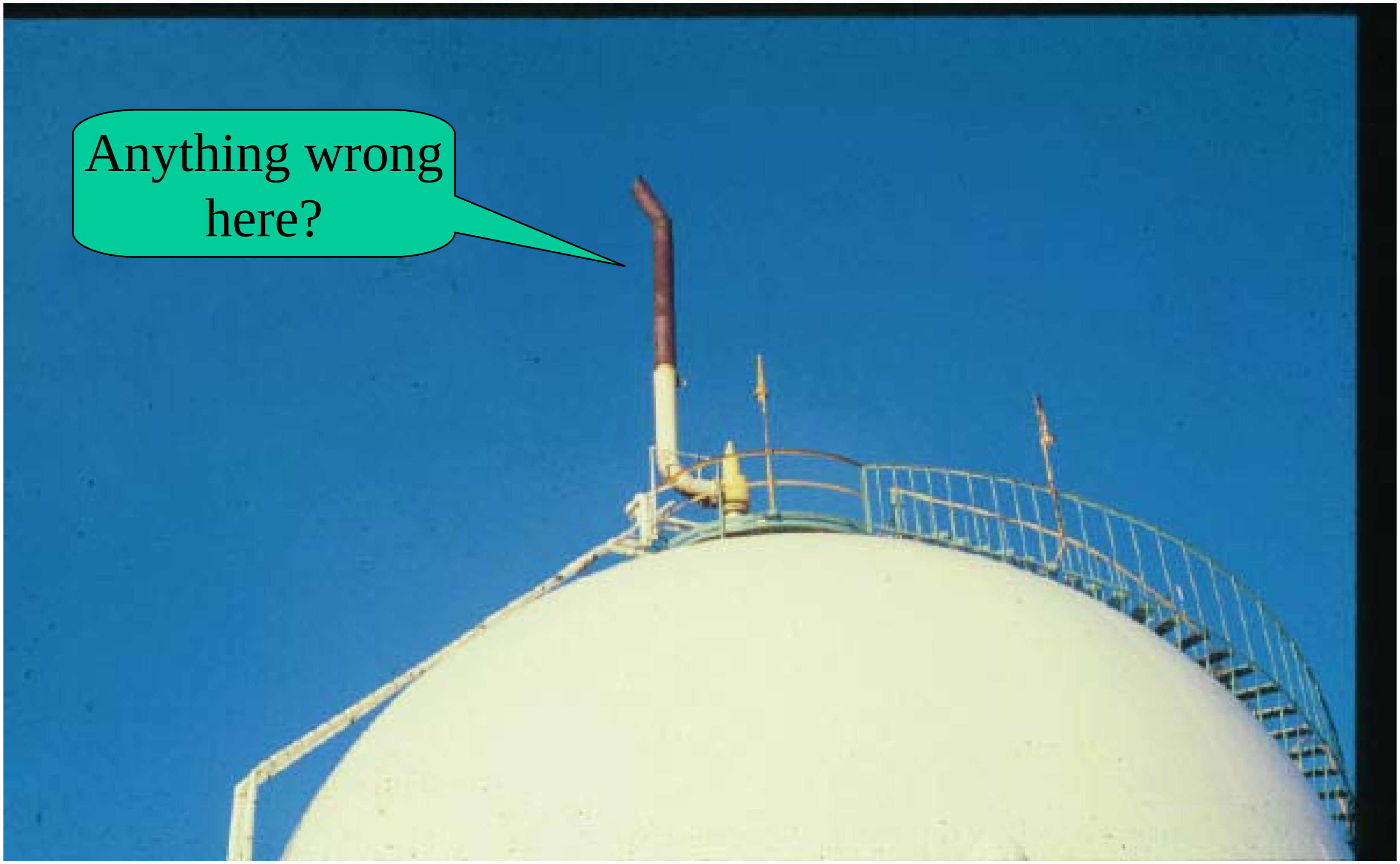
Outlet Line Considerations

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Outlet Line Considerations

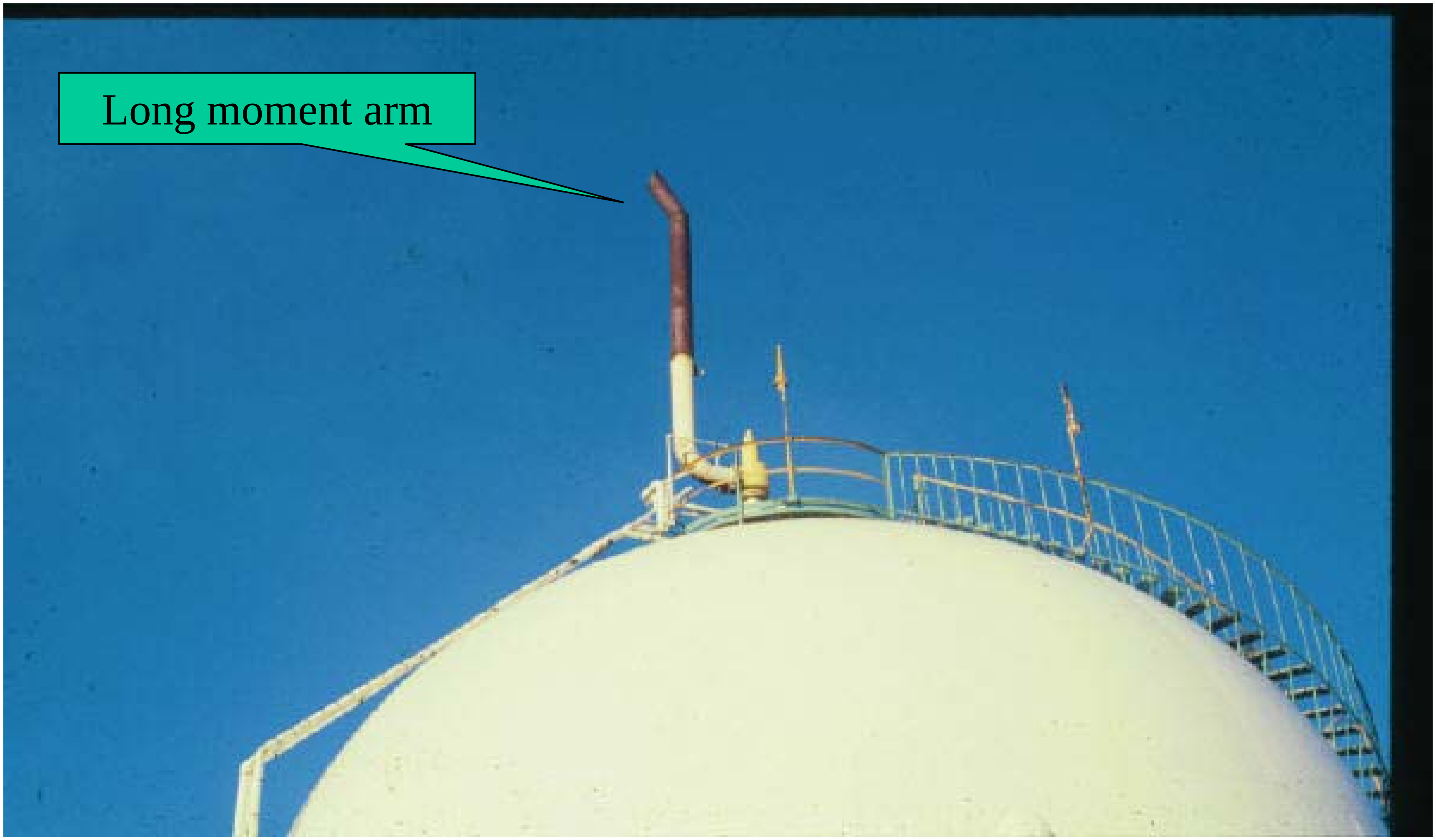
Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Outlet Line Considerations

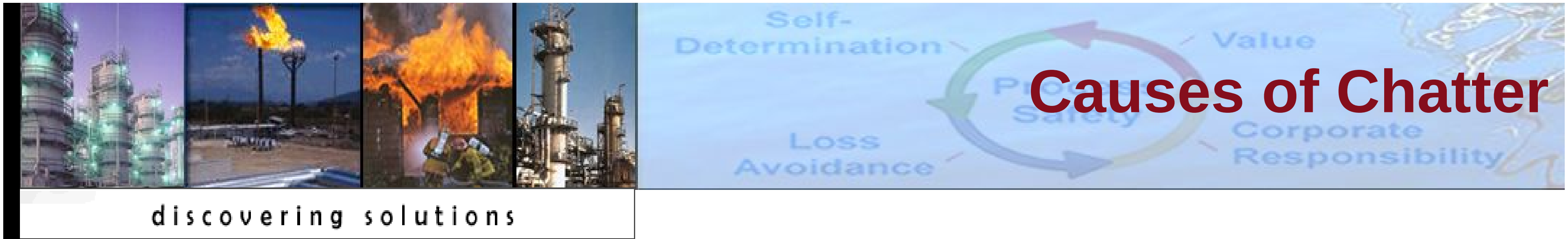


Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

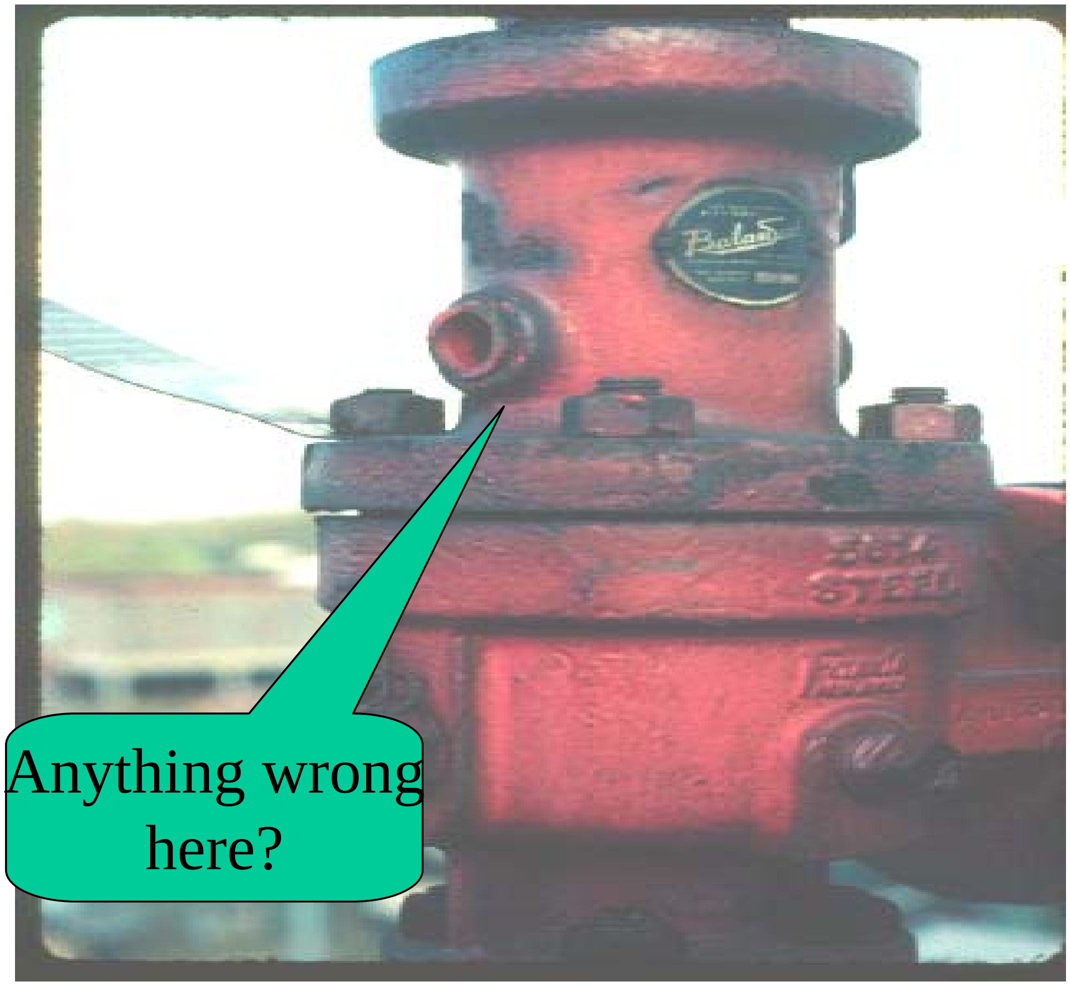
Outlet Line Considerations



Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.



Causes of Chatter

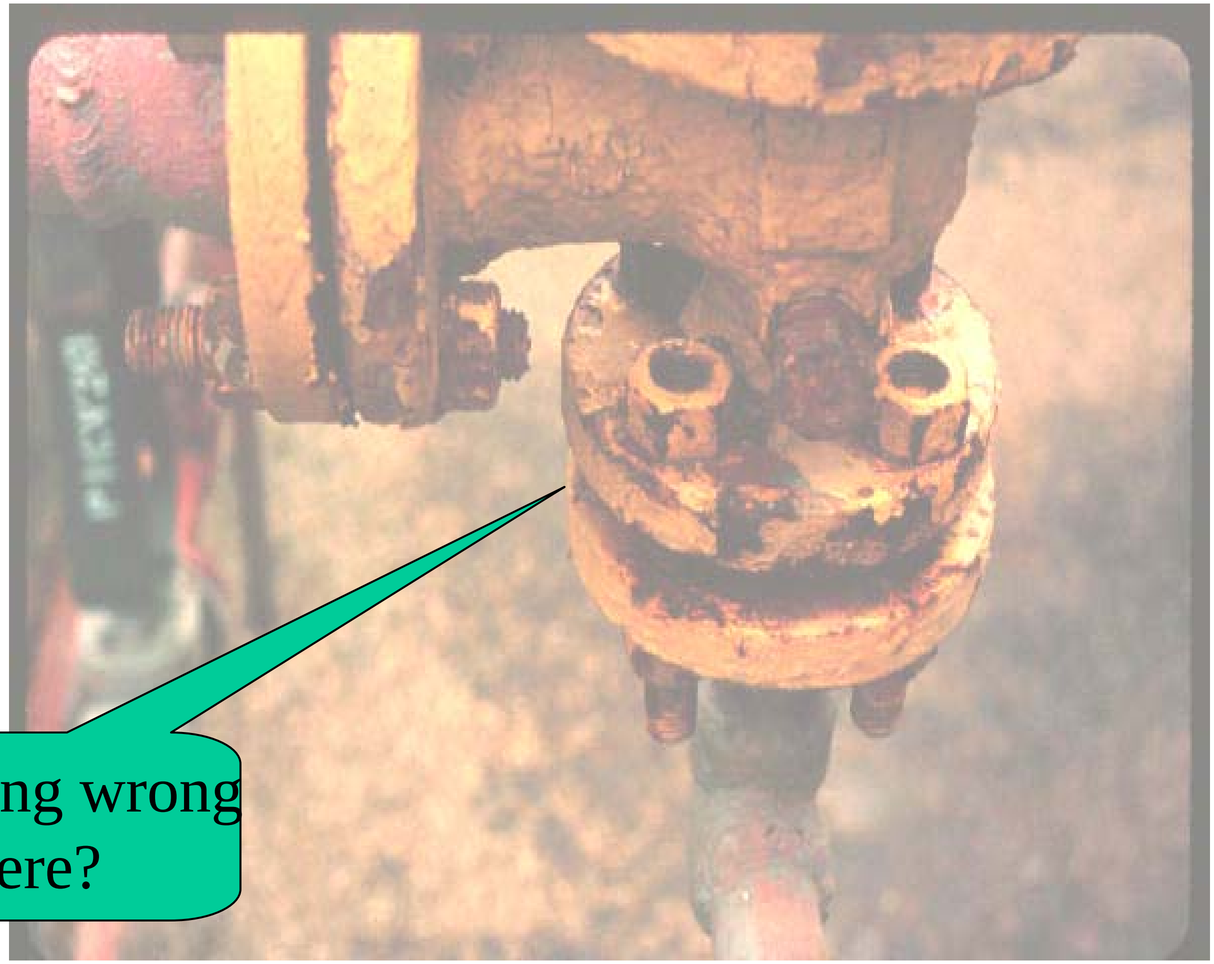


Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Causes of Chatter

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

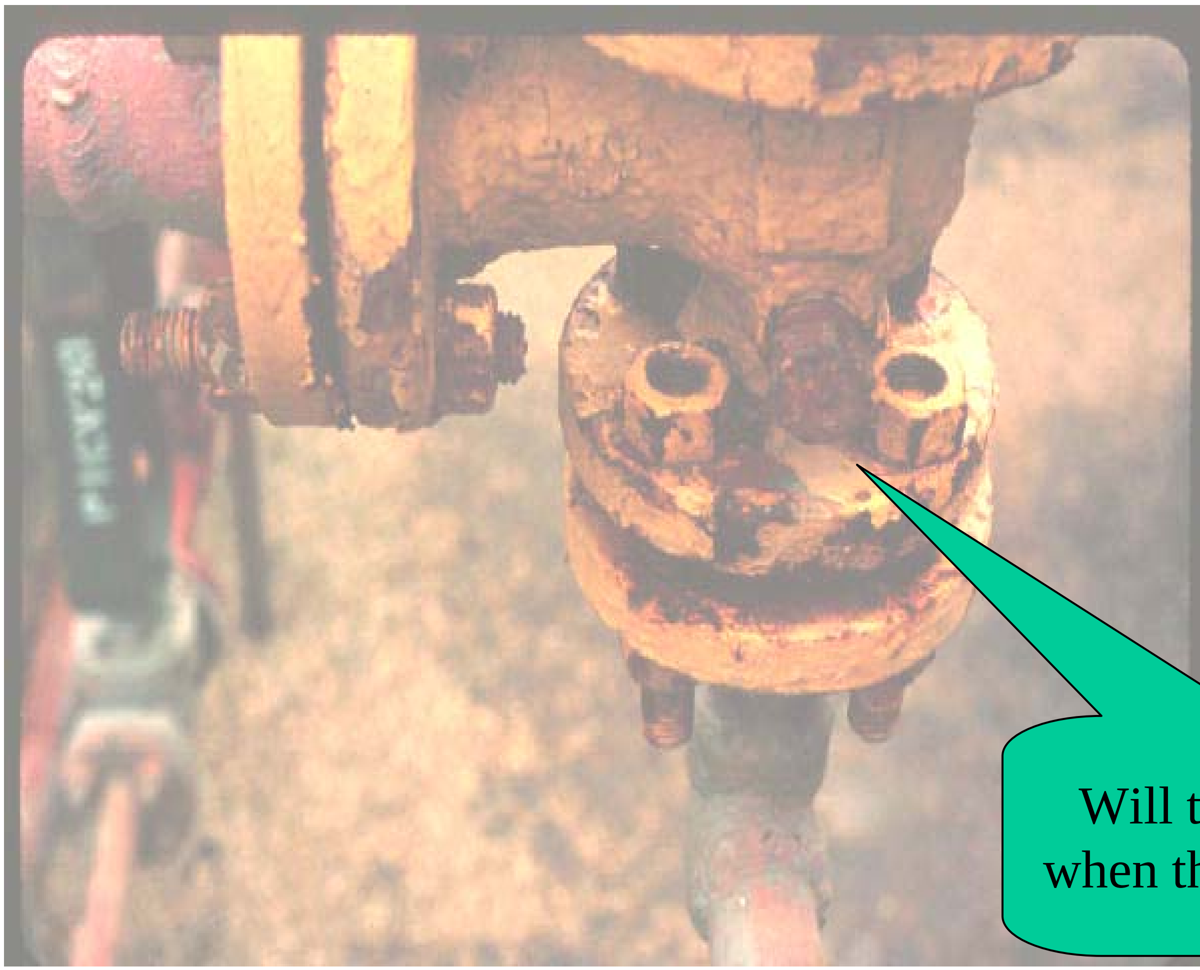
Anything Wrong Here ?



Anything wrong here?

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Anything Wrong Here?



Will these bolts hold when the PRV relieves?

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Causes of Chatter

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Causes of Chatter

Reference: Scott Ostrowski, SACHE Faculty Workshop, Sept 28th, 2003.

Conclusions & Recommendations

- **Restrict inlet pressure loss that can be tolerated to blowdown minus 2 %***
  - » If blowdown is 10 %, inlet pressure loss that can be tolerated is 8 %
  - » If blowdown is 5 %, inlet pressure loss that can be tolerated is 3 %
- **Adjust relief device blowdown and set points with care using a certified shop**
- **Increasing blowdown allows more backpressure tolerance but also reduces the flow capacity of the valve**

*Unpublished work by Prof. Ron Darby suggests that at 7 % blowdown can tolerate an inlet loss of 5 %. Also see published data of K. A. Kastor, 1992.

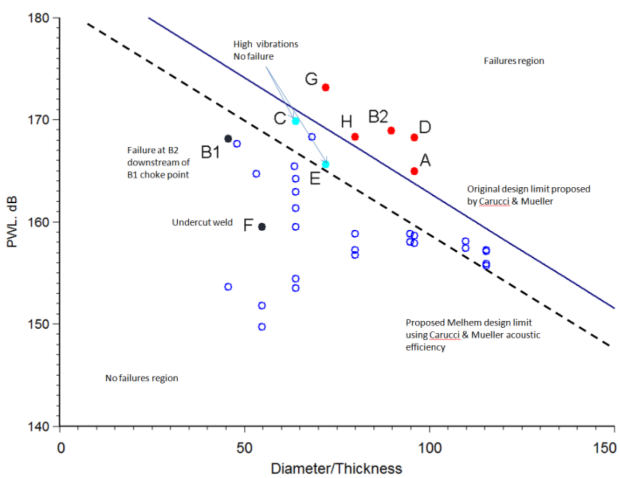
discovering solutions

Chatter Program

- **A chatter algorithm will be added to SuperChems**
- **Will Allow the user to test if chatter is possible with a specified piping layout**

Estimate Vibration Risk for Relief and Process Piping

July 17, 2012



Notice:

This report was prepared by ioMosaic for public disclosure. This report represents ioMosaic's best judgment in light of information made available to us. Opinions in this report are based in part upon data and information obtained from the open literature. The reader is advised that ioMosaic has not independently verified the data or the information contained therein. This report must be read in its entirety. The reader understands that no assurances can be made that all liabilities have been identified. This report does not constitute a legal opinion.

No person has been authorized by ioMosaic to provide any information or make any representations not contained in this report. Any use the reader makes of this report, or any reliance upon or decisions to be made based upon this report are the responsibility of the reader. ioMosaic does not accept any responsibility for damages, if any, suffered by the reader based upon this report.

ioMosaic Corporation
93 Stiles Road
Salem, New Hampshire
03079 U.S.A.

Contents

List of Tables

List of Figures

Estimate Vibration Risk for Relief and Process Piping

G. A. MELHEM
ioMosaic Corporation
93 Stiles Road
Salem, New Hampshire 03079

1 Introduction

Fatigue failure of relief and/or process piping caused by vibration can develop due to the conversion of flow mechanical energy to noise. Factors that have led to an increasing incidence of noise vibration related fatigue failures in piping systems include but are not limited to (a) increasing flow rates as a result of debottlenecking which contributes to higher flow velocities with a correspondingly greater level of turbulent energy, (b) frequent use of thin-walled piping which results in higher stress concentrations, particularly at small bore and branch connections, (c) design of process piping systems on the basis of a static analysis with little attention paid to vibration induced fatigue, (e) and lack of emphasis of the issue of vibration in piping design codes. Piping vibration is often considered on an ad-hoc or reactive basis. According to the UK Health and Safety Executive (HSE), 21 % of all piping failures offshore are caused by fatigue/vibration. Typical systems at risk include large compressor recycle systems and high capacity pressure relief depressuring systems. For relief and flare piping, flow induced turbulence and high frequency acoustic excitations are key concerns.

2 Flow Induced Turbulence

Fluid flow in pipes generates turbulent energy (pressure fluctuations). Dominant sources of turbulence are associated with flow discontinuities in the piping systems (e.g., partially closed valves, short radius, mitered bends, tees or expanders). The level of turbulence intensity is a function of pipe size, fluid density, viscosity, velocity, and structural support. High noise levels are generated by high velocity fluid impingement on the pipe wall, turbulent mixing, and if the flow is choked, shock waves downstream of flow restriction, which leads to high frequency excitation/vibration.

3 High Frequency Excitation

High frequency acoustic energy is often generated by a pressure reducing device such as a relief valve, control valve, or orifice plate. Acoustic induced piping failure is of a particular concern for safety related systems (e.g. relief and blowdown/depressuring). The severity of high frequency acoustic excitation is primarily a function of the pressure upstream and downstream of the pressure reducing device, pipe diameter, and the fluid volumetric flow. Acoustically induced piping failures are known to occur at non-asymmetric discontinuities in the downstream piping such as small bore and branch connections and welded supports.

4 Additional Causes of Vibration

Additional causes of piping vibration include mechanical excitation, pulsation, vortex shedding, surge or momentum changes associated with valves, cavitation, and flashing. Mechanical excitations are often associated with pipes connected with reciprocating compressors, pumps, or rotating machinery. Such connection machines cause vibration of the pipe and its support structure. Thermowells are intrusive fittings and are subject to static and dynamic fluid forces. Vortex shedding is the dominant concern as it is capable of forcing the thermowell into flow-induced resonance and consequent fatigue failure. The latter is particularly significant at high fluid velocities.

5 Relief and Depressuring Systems

Depressuring systems are often subjected to acoustic energy (rapidly fluctuating pressure forces) generated by flow turbulence which is accentuated by flow restricting devices within the flow path. The magnitude of pressure fluctuations depends on the mass flow rate, speed of sound, and density. For choked flow, intense noise due to large shock discontinuities and pressure fluctuations is generated. The generated noise is non-periodic due to the randomness of the pressure fluctuations. Choked flow typically leads to a wide frequency spectrum with peak values than can exceed 1000 Hz. Vibrations associated with small fittings and branch connections are of special concern because they introduce discontinuities and stress concentration points.

In many situations resonance can onset which can lead to magnification of static piping loads up to a factor of 50 times. The presence of discontinuities such as tees and welded pipe supports can further increase these loads.

6 Noise Generation

A pressure reducing device or relief device controls flow by converting internal energy into kinetic energy. Some energy is converted to heat through friction (viscous forces) by intense turbulence and shock formation. Some of the energy is also transferred to the pipe wall as vibration, and a portion of this is radiated as noise. The primary noise generating mechanism is the confined jet of fluid formed between the upstream and downstream locations. Flow noise can be modeled as noise of a confined jet. As a result, the noise-generation mechanisms are turbulent mixing, turbulence boundary interaction, shock, shock/turbulence interaction and flow separation.

Since the noise is generated downstream of a flow restriction, most of the acoustic energy is radiated to the downstream piping, which becomes the transmitting medium. As the noise travels downstream along the inside of the pipe in the fluid it radiates through the pipe wall along its entire length.

7 How to Calculate Sound Power Level

We can calculate the sound power level in a fundamental way by assigning an efficiency factor to provide the fraction of flow mechanical energy that is converted to noise:

$$W = \eta \frac{1}{2} \dot{m} u^2 \tag{1}$$

$$L_W = 10 \log_{10} \left[\frac{W}{10^{-12}} \right] = 10 \log_{10} W + 120 = 10 \log_{10} \left[\eta \frac{1}{2} \dot{m} u^2 \right] + 120 \tag{2}$$

where W is the flow mechanical power or energy, L_W is the sound power level in dB, and η is defined as the acoustical efficiency factor. If the flow is choked, then W becomes:

$$W = \eta \frac{1}{2} \dot{m} u_{sonic}^2 \tag{3}$$

If a value of η can be estimated for single and multi-phase flow, then the sound power level can be easily calculated not only for pressure reducing devices but also for pipe flow. Attenuation due to friction and temperature changes can be calculated from pipe flow equations in a more detailed manner. Computer codes such as SuperChems can then calculate the sound power level at every axial location for flow piping for single and multi-phase flow.

For incompressible flow, the value of flow velocity for an ideal nozzle can be calculated from the mechanical energy balance:

$$u = \sqrt{\frac{2}{\rho_l} \Delta P} \tag{4}$$

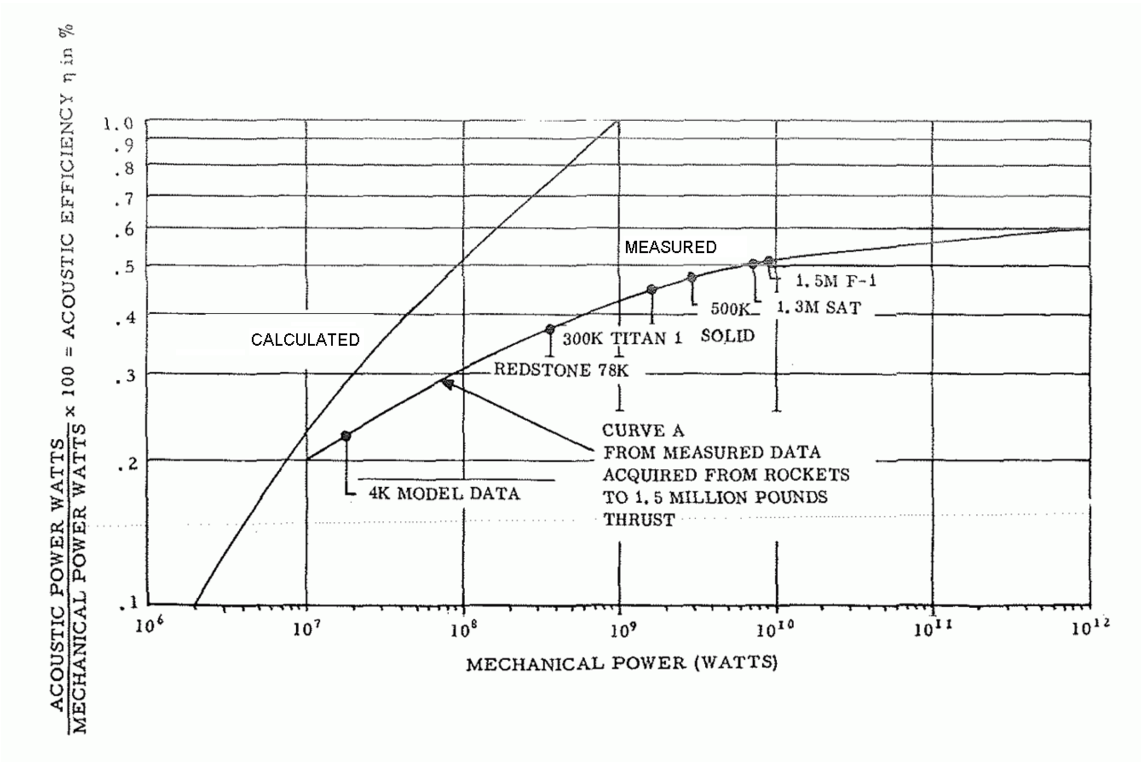
Substituting the above equation for u in Equation 1 yields the following Equation for W for liquid flow through an ideal nozzle:

$$W = \eta \dot{m} \frac{\Delta P}{\rho_l} \tag{5}$$

For liquid flow, a typical value of η is approximately 10^{-8}.

All gas flow acoustic efficiencies have been reported to approach 1 % of the total flow mechanical energy for rockets [4]. This is illustrated in Figure 1. When the acoustic efficiency is plotted against the flow mechanical power Figure 1 is obtained. The measured curve indicates that the acoustic efficiency falls off as the flow mechanical energy gets larger while the calculated curve [5] indicates increasing acoustic efficiencies.

Figure 1: Acoustic Efficiency Trends



Two-phase flow sound power level can also be calculated using Equation 1. Recent work by Singh et al. ([6], [7]) demonstrated that more attenuation of noise is exhibited by two-phase flow. Therefore, using the gas acoustic efficiency values will overpredict sound power and noise levels for two-phase flow.

8 Existing Methods and Guidance

Current API, AIChE/CCPS, and AIChE/DIERS pressure relief and flare systems guidelines and standards do not formally address vibration risk. They do not offer specific guidance on velocity limitations other than backpressure calculations and they do not offer guidance on acoustic induced or turbulence induced piping vibration fatigue failure. The Marine Technology Directorate Limited (MTD) has published in 1999 "Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework" [8]. A second edition of these guidelines were published in 2008 by the Energy Institute [9]. The methods outlined in these guides have been incorporated in the SuperChems Expert ioVIPER modules. The MTD/Energy Institute guidelines provide qualitative and quantitative methods for the assessment of piping vibration failure risk and depending on the calculated risk level they provide generic guidance for the mitigation of vibration risk.

Many operating companies have established their own internal guidance for evaluating and minimizing piping vibration risk. Although these criteria vary from company to company, they all in general include a limit of flow velocity in some form.

A common criteria used in relief systems is to limit the value of flow Mach number to a limiting value ranging from 0.3 to 0.9:

$$0.3 \leq M = \frac{u}{u_{sonic}} \leq 0.9 \tag{6}$$

Another common criteria often encountered is to limit the dynamic pressure component or kinetic energy of a flow stream to a value of 100,000 Pascals for gas flow:

$$\rho u^2 \leq 1 \times 10^5 \ \ \mathrm{kg/m/s^2} \tag{7}$$

and 50,000 for two-phase flow:

$$\rho u^2 \leq \frac{1}{2} \times 10^5 \ \ \mathrm{kg/m/s^2} \tag{8}$$

The 2006 fifth edition of the NORSOK Process Design Standard P-001 limits the flow velocities for all flare lines to $\rho u^2 \leq 200000 \ \mathrm{kg/m/s^2}$ for single and multiphase flow. The piping for flare headers and sub-headers are designed for a maximum Mach number of 0.6 and lines downstream of pressure relief valves to the first sub-header are designed for a maximum Mach number of 0.7. For process lines the maximum design velocity for gas pipes is limited to 60 $\mathrm{m/s}$ or $u \leq 175\frac{1}{\rho^{0.43}}$ whichever is less. The maximum velocity for two-phase lines is limited to $u \leq 183\frac{1}{\rho^{0.5}}$ or:

$$\rho u^2 \leq 183^2 \leq 33489 \ \ \mathrm{kg/m/s^2} \tag{9}$$

9 The Method of Carucci and Mueller

Carucci and Mueller have published guidance for the estimation of sound power levels for control valves and pressure reducing devices (see [10] and [11]).

$$\begin{aligned}
L_W &= 10 \times \log_{10}\left[\left(\frac{P_1 - P_2}{P_1}\right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w}\right)^{1.2}\right] + 126.1 && (10)\\
&= 10 \times \log_{10}\left[4 \times \left(\frac{P_1 - P_2}{P_1}\right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w}\right)^{1.2}\right] + 120 && (11)\\
&= 36\log_{10}\left(\frac{P_1 - P_2}{P_1}\right) + 20\log_{10}(\dot{m}) + 12\log_{10}\left(\frac{T_1}{M_w}\right) + 126.1 && (12)
\end{aligned}$$

where P_1 is the upstream pressure or source pressure, P_2 is the downstream pressure, T_1 is the source temperature, $\dot{m}$ is the mass flow rate, and M_w is the molecular weight. Attenuation of noise due to friction and heat conduction losses is estimated from the following equation:

$$L_{W,At} = 0.06\frac{L}{D_i} \tag{13}$$

where L is the pipe length and D_i is the internal pipe diameter. For an L/D of 50 the attenuation loss is 3 dB. Abrupt changes in flow area (expansion) in the piping can also lead to attenuation. The decrease in sound power level can be estimated from the following equation:

$$L_{W,Ex} = 2\left(\frac{D_2}{D_1} - 1\right) \tag{14}$$

where $D_2 > D_1$. A 3 dB reduction is typically applied to the flow leaving a tee in each direction or entering into header or a large drum or vessel. The Carucci and Mueller Equation 12 is used by the Energy Institute Guidelines for the assessment of failure likelihood for high frequency acoustic excitation (see Pages 59-62 in [9]). Note that the Carucci and Mueller equation cannot be easily applied to complex piping systems, multi-phase systems, and relief piping with multiple chokes. Some companies also add 6 dB to the sound power level estimate when sonic flow exists at a branch connection to account for amplified dynamic strain response.

9.1 Analysis of the Carucci and Mueller Data Set

Careful analysis of the Carucci and Mueller data set (see Table 1) indicates that the source pressures were high enough to produce choked (sonic) flow through a flow limiting orifice or valve upstream of the failure point further downstream, often at a branch or line connection. This important point was missed by Eisinger who used the downstream (non-choked) flow velocity to establish his Mach number based failure criterion. We were also able to reproduce Eisinger's estimates of the original Caruci and Mueller data set based on his published paper.

Carucci and Mueller provided enough data in their original paper to allow the calculation of the upstream flow limiting flow area and choke point conditions (this is possible because they reported the actual flow rates) as well as the conditions downstream of the flow limiting device in the discharge piping. Note that the sound power level estimate upstream of the choke points should be based on the pressure difference (or pressure ratio) of the source pressure and the choke point. The sound pressure level downstream of the choke point (the primary cause of acoustic induced fatigue failure in downstream piping of the choked point) should be based on the difference (or pressure ratio) of the choke point and the shock discontinuity pressure downstream of the choke point. The original equation proposed by Carucci and Mueller used the pressure difference (or ratio) between the source pressure and the exit pressure downstream of the choke point. This yields an overestimate of the sound pressure. This overestimate is somewhat tempered by the fact that the pressure difference ratio to the source pressure is raised to the 3.6 power.

The following information was provided by Carucci and Mueller regarding the five failures and two high vibrations cases observed in Table 1:

A Failure occurred during startup. Sonic velocity is achieved at the 6 inch branch connection of a 24 inch pipe run downstream of the recycle valve. The sound power level estimate considers the combined acoustic energy generated by the letdown valve and the sonic condition at the branch connection. Upstream pressure at branch connection estimated to be 98.8 psia.

Table 1: Vibration Risk Data Set used by Carucci and Mueller and reproduced by Melhem

CASE	SERVICE	T_1, K	P_1, bara	T_2, K	P_2, bara	D_2, m	u2, m/s	Exit Mach Number, M_2	Mass Flow Rate, kg/s	Heat Capcity Ratio, Cp/Cv	Restriction Orifice / Valve Flow Speed, m/s	Carucci and Mueller Acoustic Efficiency, %	IEC Acoustic Efficiency, %	PWL_cm, dB	PWL_mel, dB	PWL_cm - PWL_mel, dB	PWL_mel (IEC), dB
A	Propylene Compressor Recycle	319.26	20.55	238.91	1.59	0.610	32.50	0.137	29.74	1.13	224.30	0.65	0.85	168.5	165.0	3.5	161.5
B1	Natural Gas Pressure	318.71	42.75	226.00	6.27	0.254	125.60	0.399	48.26	1.22	322.99	2.54	0.91	170.9	168.1	2.8	163.6
B2	Natural Gas Letdown to Flare	318.71	5.17	270.38	2.07	0.711	58.20	0.169	48.26	1.22	357.00	0.41	0.69	171.4	168.9	2.5	163.3
C	Natural Gas Letdown to Flare	308.15	47.57	195.68	4.14	0.406	64.40	0.218	47.63	1.23	311.29	4.17	0.95	171.8	169.8	2.0	163.4
D	Nitrogen Compressor Recycle	310.93	6.96	280.31	3.72	0.610	93.80	0.297	119.70	1.20	324.76	1.05	0.79	168.3	168.2	0.1	167.0
E	Nitrogen Compressor Recycle	310.93	16.82	267.59	7.03	0.457	60.10	0.194	84.80	1.21	328.35	0.79	0.66	168.8	165.6	3.3	164.8
F	Natural Gas Compressor Recycle	330.37	51.09	220.53	13.86	0.305	28.80	0.093	35.28	1.45	272.52	0.69	0.82	164.9	159.5	5.4	160.3
G	Desuperheater Steam	802.59	99.97	480.32	10.34	0.457	49.80	0.093	36.89	1.29	619.65	2.91	0.94	175.5	173.1	2.4	168.2
H	Desuperheater Steam	658.15	43.09	476.11	10.34	0.508	49.60	0.093	45.99	1.29	586.00	0.86	0.83	163.9	168.3	-4.4	168.9
1	Process Gas Compressor Recycle	308.15	13.17	204.98	1.03	0.406	122.80	0.461	27.09	1.19	296.81	2.22	0.95	165.9	164.2	1.7	160.6
2	Process Gas Compressor Recycle	311.48	38.75	265.88	10.00	0.305	14.20	0.048	13.86	1.13	258.54	0.32	0.83	156.6	151.8	4.8	155.9
3	Propylene Compressor Recycle	316.48	20.55	262.14	3.24	0.406	15.70	0.065	12.60	1.11	209.52	0.81	0.91	156.0	153.5	2.5	154.0
4	Process Gas Compressor Recycle	300.37	2.48	257.91	1.03	0.508	145.60	0.450	34.78	1.21	325.00	0.26	0.77	161.3	156.7	4.6	161.5
5	Process Gas Compressor Recycle	310.93	15.86	215.12	2.21	0.406	68.70	0.226	25.33	1.23	357.01	1.21	0.91	165.2	162.9	2.3	161.7
6	Process Gas Compressor Recycle	300.93	39.44	250.67	15.51	0.254	26.10	0.075	20.66	1.24	330.10	0.20	0.78	158.3	153.6	4.7	159.4
7	Propylene Compressor Recycle	310.93	16.69	233.88	1.38	0.356	131.30	0.575	38.05	1.13	216.35	3.92	0.95	166.6	165.4	1.2	159.3
8	Propylene Compressor Recycle	310.93	16.69	234.72	1.38	0.508	59.20	0.259	35.03	1.13	216.35	3.61	0.95	165.9	164.7	1.2	158.9
9	Waste Steam Dump	403.71	2.76	370.82	1.03	0.406	245.10	0.566	18.90	1.10	436.60	0.15	0.77	160.5	154.4	6.1	161.4
10	Process Gas Compressor Recycle	302.59	10.69	207.90	1.03	0.406	95.80	0.357	20.92	1.19	295.15	1.49	0.94	163.2	161.3	1.9	159.3
11	Process Gas Compressor Recycle	308.15	54.81	263.43	27.51	0.305	7.20	0.016	9.15	1.29	441.77	0.10	0.79	150.5	149.7	0.8	158.5
12	Natural Gas Compressor Recycle	308.15	3.79	282.54	2.48	0.914	28.40	0.075	39.69	1.26	315.99	0.19	0.73	155.4	155.7	-0.3	161.6
13	Natural Gas Compressor Recycle	308.15	6.48	279.42	4.00	0.914	17.30	0.046	39.44	1.25	334.77	0.24	0.75	157.2	157.2	0.0	162.2
14	Natural Gas Compressor Recycle	308.15	11.45	282.82	6.76	0.762	14.40	0.039	38.93	1.19	346.05	0.28	0.76	158.1	158.1	0.0	162.5
15	Natural Gas Compressor Recycle	308.15	21.79	266.72	11.65	0.660	10.10	0.027	36.04	1.30	359.11	0.33	0.78	159.4	158.8	0.6	162.6
16	Natural Gas Compressor Recycle	308.15	40.75	272.56	21.99	0.610	6.30	0.017	35.91	1.25	344.77	0.34	0.78	159.2	158.6	0.6	162.2
17	Natural Gas Compressor Recycle	308.15	80.12	271.44	40.89	0.508	4.80	0.013	35.41	1.23	324.89	0.40	0.78	160.1	158.8	1.3	161.6
18	Natural Gas Compressor Recycle	308.15	172.99	240.53	79.98	0.406	3.80	0.010	35.41	1.47	326.41	0.71	0.82	161.5	161.3	0.2	161.9
19	Natural Gas Compressor Recycle	308.15	3.86	280.10	2.34	0.914	27.60	0.074	37.17	1.24	340.74	0.24	0.75	157.3	157.1	0.2	162.1
20	Natural Gas Compressor Recycle	308.15	6.21	281.91	3.93	0.914	16.50	0.044	36.79	1.24	325.37	0.20	0.74	155.9	155.9	0.0	161.6
21	Natural Gas Compressor Recycle	308.15	10.62	274.09	6.27	0.762	14.30	0.038	36.04	1.29	345.37	0.26	0.76	157.4	157.4	0.0	162.1
22	Natural Gas Compressor Recycle	308.15	19.72	266.35	10.62	0.660	10.10	0.027	33.01	1.31	359.23	0.30	0.78	158.4	158.0	0.4	162.2
23	Natural Gas Compressor Recycle	308.15	36.82	263.14	19.79	0.610	6.40	0.017	32.89	1.34	346.54	0.31	0.78	158.4	157.9	0.5	161.9
24	Natural Gas Compressor Recycle	308.15	64.47	270.22	36.96	0.508	4.90	0.013	32.82	1.31	319.94	0.32	0.77	157.3	157.2	0.1	161.1
25	Natural Gas Compressor Recycle	308.15	139.48	320.30	64.54	0.406	4.20	0.012	32.76	0.95	311.67	0.56	0.80	160.7	159.5	1.2	161.1
26	Propane Compressor Recycle	308.15	12.07	234.24	0.97	0.457	143.50	0.644	50.40	1.12	217.76	4.82	0.96	168.9	167.6	1.3	160.6
27	Steam Desuperheater	658.15	43.09	522.78	10.34	0.762	22.00	0.041	45.99	1.19	564.55	0.91	0.83	161.2	168.3	-7.1	167.9

B1/B2 This system consisted of a control valve letting down pressure to a safety valve / flare header system. The failure occurred after five to ten hours of operation as a crack at a 10 inch branch connection to 28 inch header. Sonic velocity was achieved at both the control valve and the 10 inch branch connection to the 28 inch flare header. As such, two pertinent data points are established:

1. Acoustic energy within 10 inch line as generated by valve (not the failure point), and
2. Combined acoustic energy within 28 in header as generated by valve and sonic condition at the 10 in branch connection (the failure point).

C High vibrations, no failures.

D Failure during startup. Cracks were observed in a 24 inch line downstream of the compressor recycle valve after twelve hours of operation. The cracks were near the branch connections of a 6 inch line and a 3/4 inch drain valve and at an I-beam support welded to the pipe at the elbow immediately downstream of the control valve.

E High vibration, no failures.

F Failure at severely undercut weld on 300 mm (12 in.) line made to fasten conduit support Clip. Points of high stress concentration later eliminated.

G This system consisted of six, parallel, high pressure steam letdown valves, each with a downstream, three pass contra-flow attemporator and an in-line silencer. Failure of the 18 inch attemporator shells occurred after four hundred hours of operation.

H This system included four parallel desuperheaters located downstream of control valves. Several cracks developed in this system after two to three months of operation. Two to three of the four letdown valves were discharging steam through a 10 inch connection to a 20 inch header which swaged up to a 30 inch diameter. Longitudinal cracking occurred at the bottom of the 20 inch header at a transverse guide 0.5 meters downstream of the fourth 10 inch branch connection. In addition, a 1 inch bypass line for a block valve downstream of the third letdown valve cracked and the 20 inch header cracked around a pressure tap downstream of the transverse guide.

Table 1 compares the acoustic efficiencies implied in the Carucci and Mueller method (at the upstream choke point) vs. acoustic efficiencies established using IEC methods described later on in this paper. The acoustic efficiencies implied by the Carucci and Mueller method at the upstream choke point ranged from 0.1 to 5 percent while the IEC acoustic efficiency ranged from 0.67 to 1 percent. The Carucci and Mueller method is based on acoustic energy theories encompassing both jet and choked flow noise as well as test data from work performed by Exxon research and engineering. As discussed in later sections, the experience based failure criteria originally developed by Carucci and Mueller can only be used with sound power levels calculated by Equation 12.

The implied acoustic efficiency by Equation 12 is proportional to the mass flow rate and inversely proportional to u^2:

$$\eta = 8\left(1-\frac{P_2}{P_1}\right)^{3.6}\left(\frac{T_1}{M_w}\right)^{1.2}\left(\frac{\dot{m}}{u^2}\right) \tag{15}$$

If we consider the case of an ideal gas undergoing isentropic choked (sonic) flow through a restriction orifice, we can establish the following expressions for variables of interest at the choke point:

$$T_2 = 2\frac{T_1}{\gamma+1} \tag{16}$$

$$P_2 = P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \tag{17}$$

$$\rho_2 = \frac{P_2 M_w}{R_g T_2} \tag{18}$$

$$u_2 = \sqrt{\frac{\gamma R_g T_2}{M_w}} \tag{19}$$

$$\dot{m} = \rho_2 u_2 A_o \tag{20}$$

Substituting the values of mass flow rate $\dot{m}$, choked flow velocity u_2, and choke pressure P_2 in Equation 15, we obtain the following simplified expression for the Carucci and Mueller acoustic efficiency after some algebraic manipulations:

$$\eta = (4.7950\gamma - 2.5882)\left(\frac{M_w}{T_1}\right)^{0.3} P_1 A_o \tag{21}$$

Where A_o is the effective restriction orifice flow area in m^2, P_1 is the upstream pressure in bara, T_1 is the upstream temperature in Kelvin, and η is the calculated Carucci and Mueller acoustic efficiency in percent. This expression shows that the Carucci and Mueller acoustic efficiency is directly proportional to upstream pressure, and the restriction orifice flow area (flow rate). It can be shown that the Carucci and Mueller acoustic efficiency produces unrealistic values for high pressure systems, large flow area, and/or mixtures with high molecular weights.

If we consider the case of methane discharging through a restriction with an effective flow area of 0.1 m^2, upstream temperature of 373.15 K, and an upstream pressure of 100 bara, the calculated acoustic efficiency is:

$$\eta = (4.7950 \times 1.282 - 2.5882)\left(\frac{16}{373.15}\right)^{0.3} \times 100 \times 0.1 \tag{22}$$

$$= 3.559 \times 0.388 \times 100 \times 0.1 = 13.8\% \tag{23}$$

The same value can also be calculated by calculating $\dot{m}$, u_2, P_2, and η directly from the above ideal gas flow equations and Equation 15:

$$T_2 = 2\frac{T_1}{\gamma+1} = 2\frac{373.15}{1.282+1} = 327.05\ \mathrm{K} \tag{24}$$

$$P_2 = P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 100\left(\frac{2}{1.282+1}\right)^{\frac{1.282}{1.282-1}} = 54.9\ \mathrm{bara} \tag{25}$$

$$
\begin{aligned}
\rho_2 &= \frac{P_2 M_w}{R_g T_2} = \frac{54.9 \times 100000 \times 16}{8314 \times 327.03} = 32.306 \text{ kg/m}^3 && (26)\\
u_2 &= \sqrt{\frac{\gamma R_g T_2}{M_w}} = \sqrt{\frac{1.282 \times 8314 \times 327.03}{16}} = 466.76 \text{ m/s} && (27)\\
\dot{m} &= \rho_2 u_2 A_o = 32.306 \times 466.76 \times 0.1 = 1507.63 \text{ kg/s} && (28)\\
\eta &= 8\left(1 - \frac{P_2}{P_1}\right)^{3.6} \left(\frac{T_1}{M_w}\right)^{1.2} \left(\frac{\dot{m}}{u^2}\right) && (29)\\
&= 8 \times (1 - 0.549)^{3.6} \times \left(\frac{373.15}{16}\right)^{1.2} \times \left(\frac{1507.63}{466.76^2}\right) && (30)\\
&= 8 \times 0.0568 \times 43.78 \times 69.2 \times 10^{-4} && (31)\\
&= 0.1376 \text{ or } 13.76\% && (32)
\end{aligned}
$$

A similar equation to 21 can be derived for the acoustic efficiency downstream of the choke point:

$$
\eta = (68.63 - 4.836\gamma)\left[1 - (1.0397 + 0.6096\gamma)\frac{P_b}{P_1}\right]^{3.6} \left(\frac{M_w}{T_1}\right)^{0.3} P_1 A_o \tag{33}
$$

where P_b is the superimposed backpressure downstream of the choke point. Applying Equation 33 to the same example above we calculate an acoustic efficiency of 227 percent at a backpressure of 1 bara:

$$
\begin{aligned}
\eta &= (68.63 - 4.836 \times 1.282)\left[1 - (1.0397 + 0.6096 \times 1.282)\frac{1}{100}\right]^{3.6} \left(\frac{16}{375}\right)^{0.3} \times 100 \times 0.1\\
&= 62.430 \times 0.9359 \times 0.388 \times 100 \times 0.1\\
&= 226.70\% && (34)
\end{aligned}
$$

It is evident from the actual data reported by Carucci and Mueller and from the above theoretical proof that the acoustic efficiency used by Carucci and Mueller in their equation can produce unrealistic values, well in excess of 1 percent. Thus, it is recommended that the Carucci and Mueller acoustic efficiency value be limited to a maximum of 2 percent if the calculated value exceeds 2 percent.

10 Failure Criteria

Several methods are now available for screening and analyzing the potential failure risk of piping caused by vibration. These methods include:

- Experience Based - D/t method or D Method
- Experience Based - Mach number method (Not recommended)
- MTD methods

- Detailed structural dynamics methods

The experience based methods center around correlating likelihood of failure based on actual experience and/or test data. The most widely cited reference is that of Carucci and Mueller (see [10], and [11]) for steel pipe.

Figure 2 illustrates the failure criteria developed originally by Carucci and Mueller and later modified by Eisinger. This failure criteria establishes a design sound power level vs. the ratio of pipe diameter to thickness. This criteria is fundamentally better than the other criteria based on pipe diameter only since thicker wall pipes are stronger than thinner wall pipes.

The original work by Carucci and Mueller suggests a limit provided by the following equation (Figure 2):

$$L_{W,limit} = 184.6 - 0.215\frac{2D_i}{D_o - D_i} = 184.6 - 0.215\frac{D_i}{\delta} \tag{35}$$

where δ is the pipe thickness.

The lower allowable limit developed by Eisinger also shown Figure 2 is given by the following equation:

$$L_{W,limit} = 173.6 - 0.125\frac{2D_i}{D_o - D_i} = 173.6 - 0.125\frac{D_i}{\delta} \tag{36}$$

The NORSOK standard published guidance in 2006 using the same equation for $L_{W,limit}$ as proposed by Eisinger above.

Note that the above limits are based on sound power levels calculated using the source pressure and the downstream exit pressure. We have recalculated the sound power levels based on the correct pressure ratios downstream of the choke point. This data is shown in Figures 3, 4, and 5.

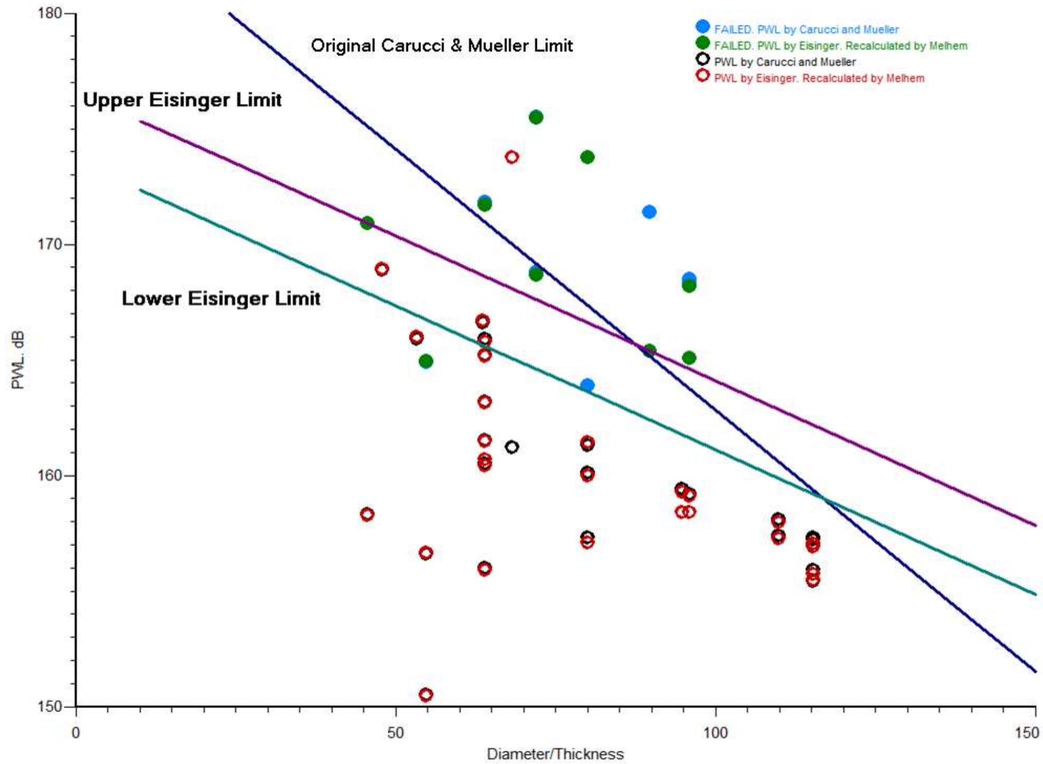
Another design limit criterion that was originally proposed by Carucci and Mueller is based on pipe diameter only for steel pipe. This criterion applies to pipe diameters ranging from 10 inch to 36 inch (approximately between 200 mm to 800 mm) and with wall thicknesses ranging from 0.219 in to 0.439 in (approximately 5.5 mm to 10 mm). The design limit is shown in Figure 3 for the corrected data and can be approximated by:

$$L_{W,limit} = 192.8 - 9.8\ln D \tag{37}$$

where D is the nominal pipe diameter in inches and L_W is the sound power limit in dB.

The design limit correlations developed in this paper can be used with computer programs such as SuperChems Expert which calculates the sound power level at every axial locations for single and multiphase flow to decide if the steel piping inside diameter to thickness ratio exceed the allowable limits as shown by Figure 6.

Figure 2: Piping failure limit as originally proposed by Carucci and Mueller and by Eisinger for steel pipe



11 Other Methods for Calculating η

The simplest method for calculating η is to assume the same efficiency as an expanded jet (see Figure 7) and apply it equally to single and multi-phase flow. The efficiency shown in Figure 7 is the same as is currently used by API for the estimation of flare noise.

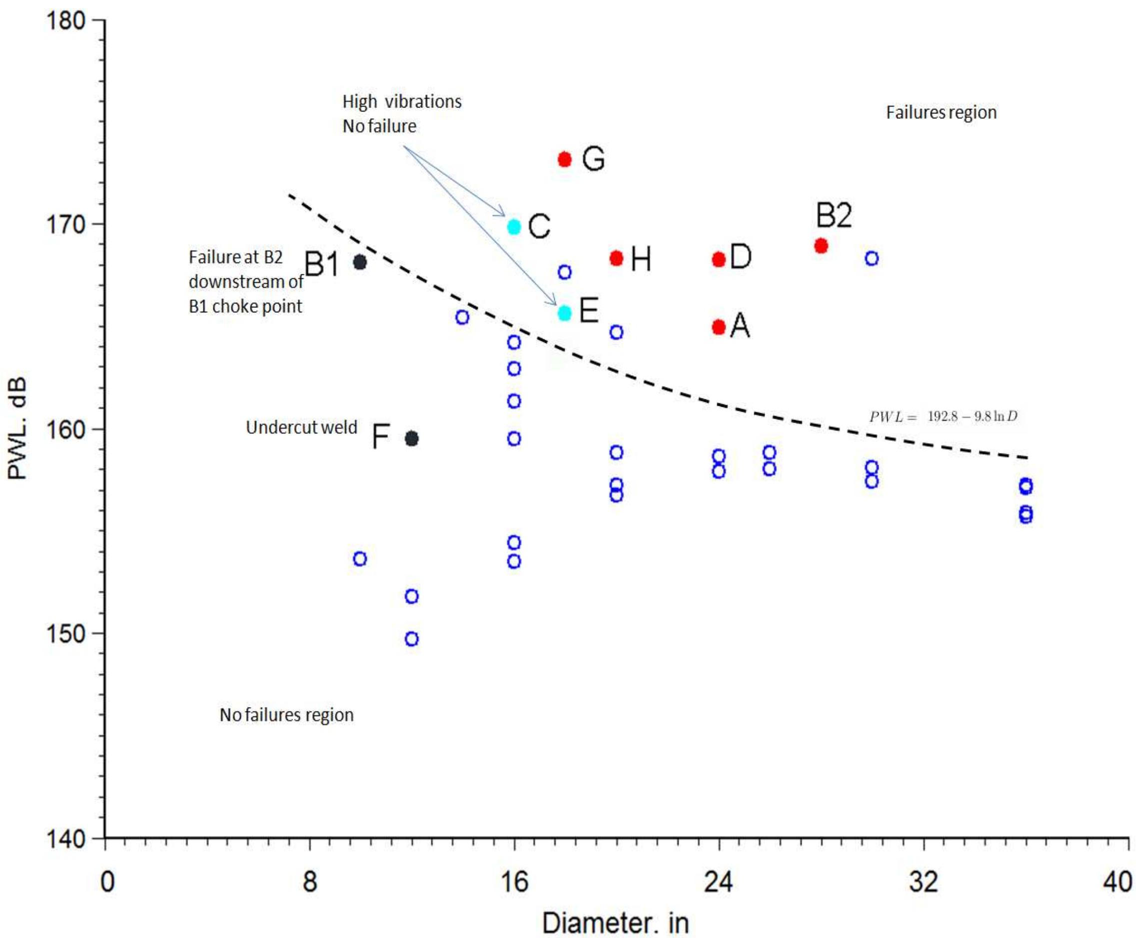
The efficiency is related to the flow pressure ratio and the flow regime as well. IEC calculates the efficiency based on five different flow regimes (see Figure 8) depending on the value of the downstream pressure, P_2:

Regime I The flow is sub-sonic. The sound generation has the character of a dipole jet. The highest Mach number is reached at the vena contracta, not exceeding Mach 1 at the maximum. Downstream of the vena contracta, the jet expands, leading to partial pressure recovery (F_L factor).

Regime II Sonic and supersonic flows exist together, which means that strongly turbulent flow and shock cell structure dominate. Pressure recovery drops until the top limit of regime II is reached.

Regime III The rise in pressure is non-isentropic. The flow is supersonic and shear turbulence predominates.

Figure 3: Piping failure correlation developed by Melhem using Carucci Mueller acoustic efficiency and corrected sound power level estimates for steel pipe. PWL vs. D

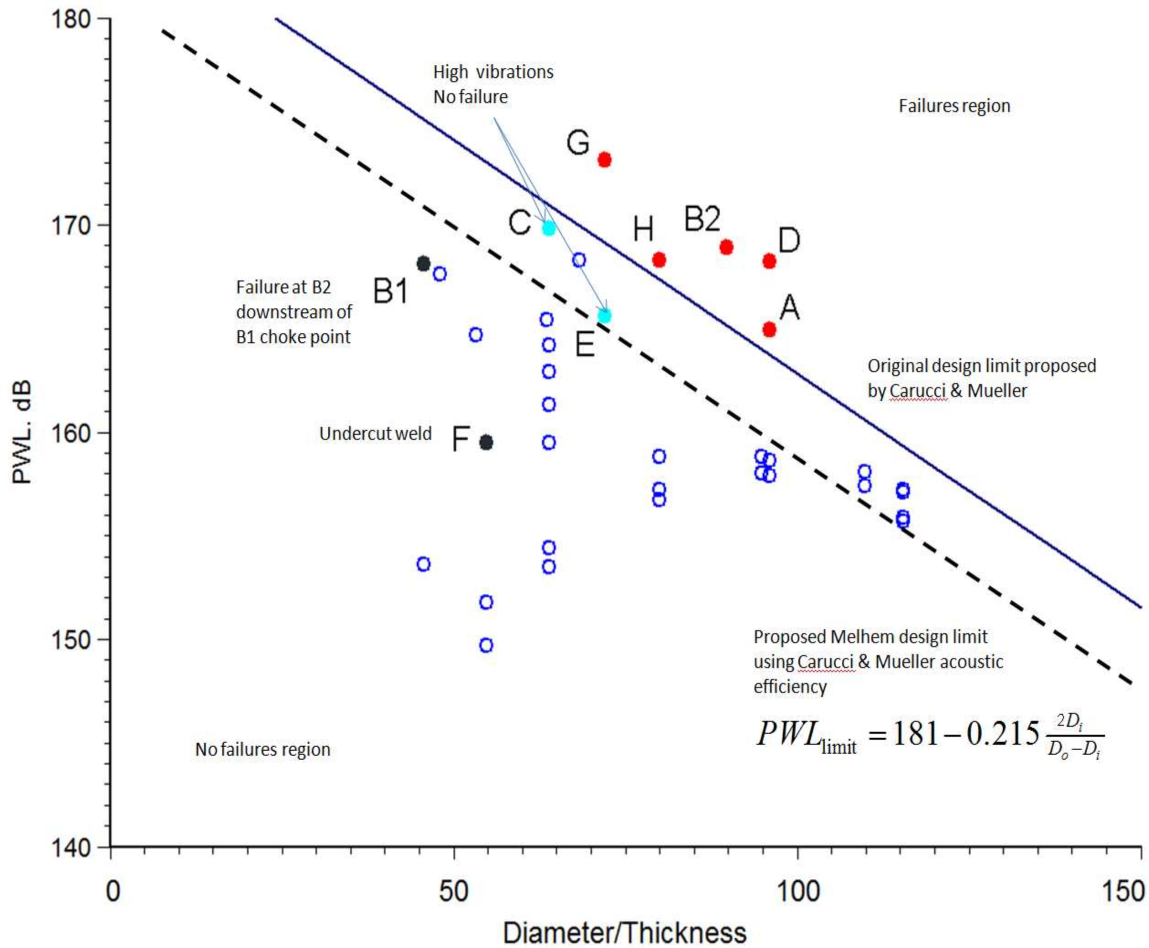


Regime IV The shock cells disappear and a Mach disk forms. The dominant mechanism is the interaction between shock cells and turbulence.

Regime V The acoustical efficiency is constant.

The flow regime types are defined by varying jet shapes in the area upstream and downstream of the vena contracta. These jets change their shape when certain differential pressure ratios are exceeded. In regimes II to IV, higher Mach numbers arise downstream of the vena contracta. Yet, M at the vena contracta itself remains unchanged at 1. The IEC method produces equivalent values of efficiency to the data shown in Figure 7 (as $F_L \rightarrow 1.0$) as shown in Figure 9. Note that Figure 9 uses $x = 1 - P_2/P_1$ as the X axis while Figure 7 uses $P_1/P_2 = \frac{1}{1-x}$.

Figure 4: Piping failure correlation developed by Melhem using Carucci Mueller acoustic efficiency and corrected sound power level estimates for steel pipe. PWL vs. D/t



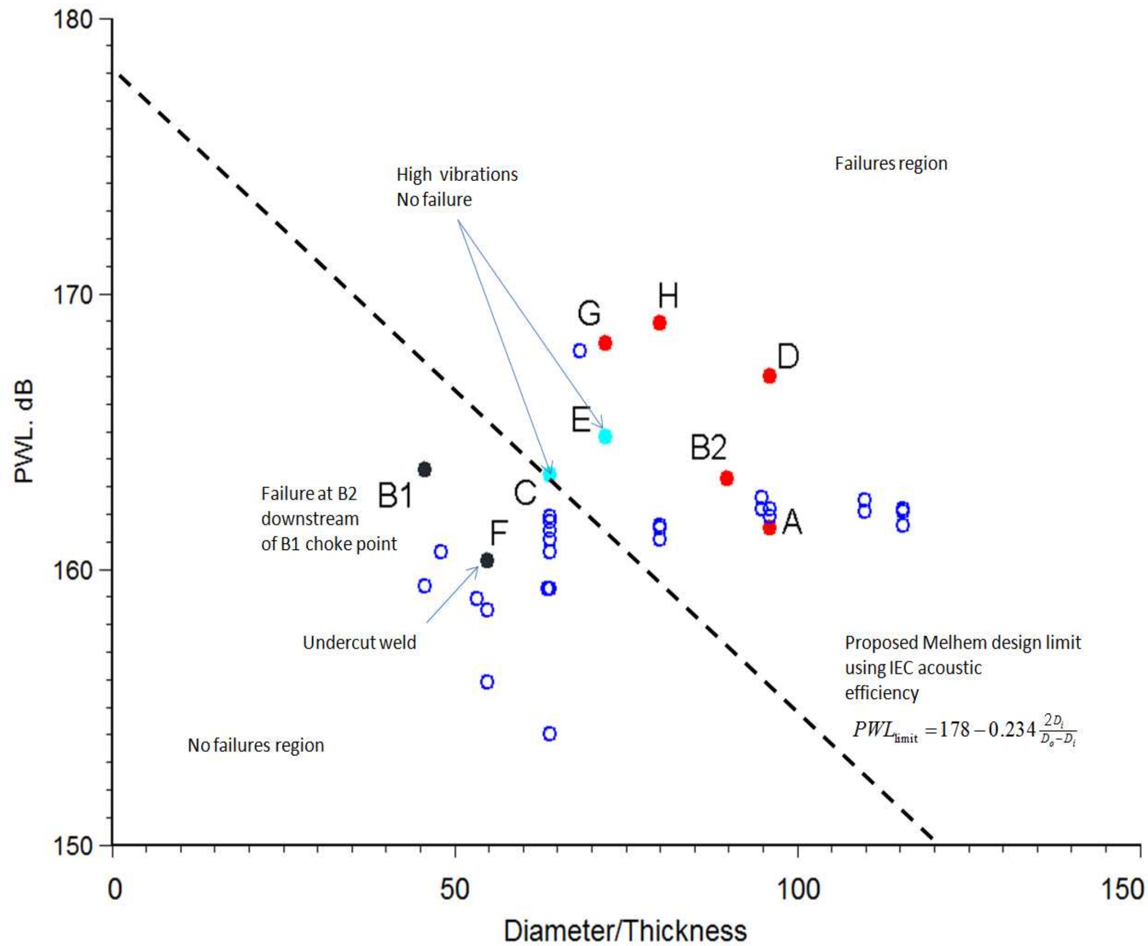
12 Vibration Frequencies

As a fluid moves through piping components there likely will be a separation of the fluid from the constraining wall as the fluid changes flow direction. As a result a vortex is formed and then swept into the main stream. This vortex shedding occurs at fairly well defined dimensionless frequencies. The strength of the vortex varies but does not need to be very strong to cause damage especially if the shedding frequency couples with the natural frequency of the piping system. The shedding peak frequency for a vortex for subsonic and sonic flows (regimes I and II up to a Mach number of 1.4) is given by:

$$f_p = \frac{N_{Str} u}{D} = 0.2 \frac{u}{D} \tag{38}$$

where u is the flow velocity in $\mathrm{m/s}$, D is a characteristic flow dimension (perpendicular to flow), f_p is the peak frequency in Hz and N_{Str} is the Strouhal Number which varies depending upon the

Figure 5: Piping failure correlation developed by Melhem using IEC acoustic efficiency and corrected sound power level estimates for steel pipe. PWL vs. D/t



geometry causing the separation of the boundary layer. For a circular cylinder its value is 0.2 over a wide range of Reynolds numbers. It is usually between 0.1 and 0.3. Vortex shedding frequencies are generally more than 30 Hz [upper limit for most piping system natural frequencies].

For flows with Mach numbers larger than 1.4 ($M > 1.4$, regimes III to V), the peak frequency $f_p$is given by:

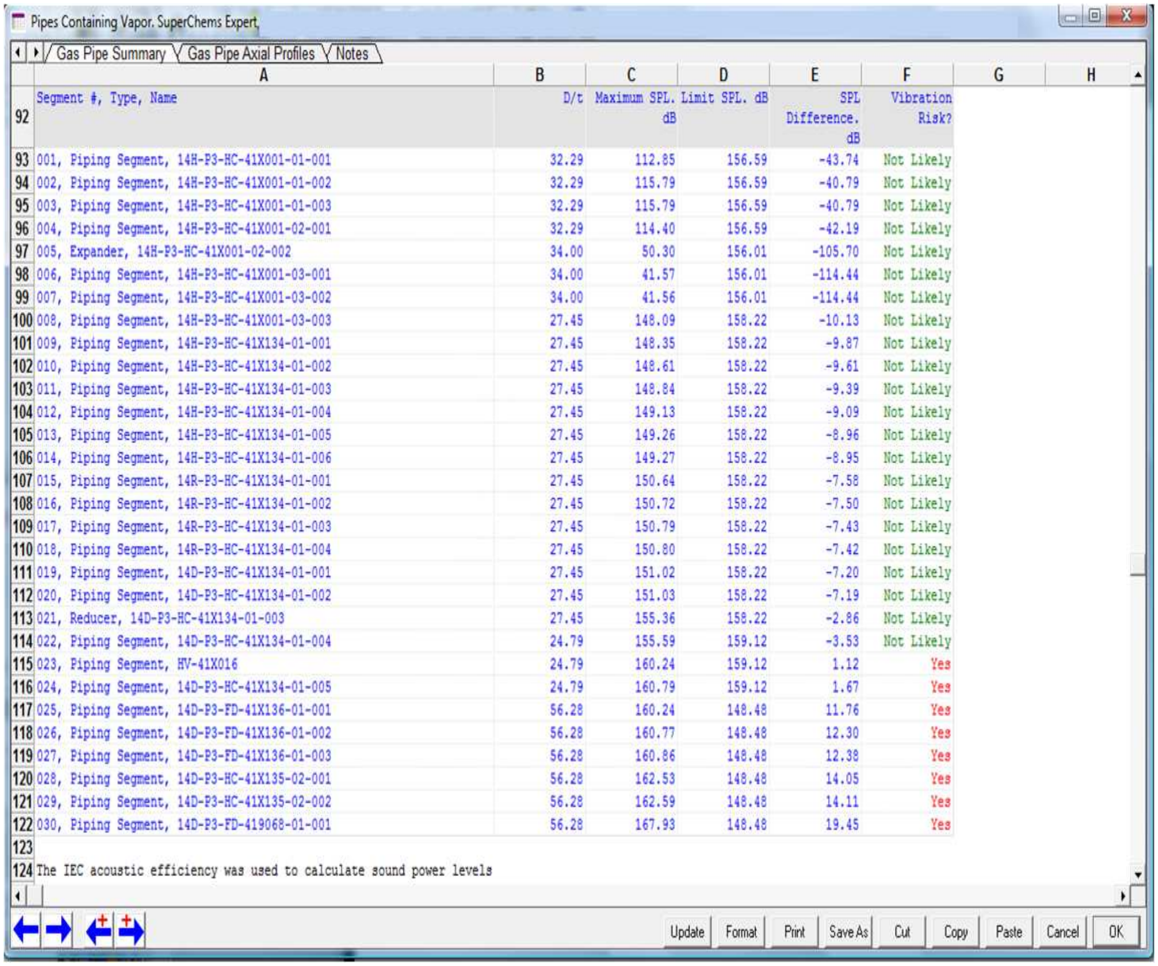
$$f_p = \frac{0.4 u_{sonic}}{1.25 D \sqrt{M^2 - 1}} \tag{39}$$

For pipe flow, D is equal to the inside flow diameter. For a control valve, D is given by:

$$D = 0.0046 \sqrt{\frac{C_v F_l}{N_o}} \tag{40}$$

where D is in meters, N_o is the number of separate flow passages, C_v is the valve flow coefficient, $\frac{C_v}{N_o}$ is the channel flow coefficient, and F_l is the valve recovery factor. The pressure recovery factor

Figure 6: Sample piping sound power level calculated by SuperChems Expert v6.4mp vs. experience based allowable limit for steel pipe



	A	B	C	D	E	F
92	Segment #, Type, Name	D/t	Maximum SPL. dB	Limit SPL. dB	SPL Difference. dB	Vibration Risk?
93	001, Piping Segment, 14H-P3-HC-41X001-01-001	32.29	112.85	156.59	-43.74	Not Likely
94	002, Piping Segment, 14H-P3-HC-41X001-01-002	32.29	115.79	156.59	-40.79	Not Likely
95	003, Piping Segment, 14H-P3-HC-41X001-01-003	32.29	115.79	156.59	-40.79	Not Likely
96	004, Piping Segment, 14H-P3-HC-41X001-02-001	32.29	114.40	156.59	-42.19	Not Likely
97	005, Expander, 14H-P3-HC-41X001-02-002	34.00	50.30	156.01	-105.70	Not Likely
98	006, Piping Segment, 14H-P3-HC-41X001-03-001	34.00	41.57	156.01	-114.44	Not Likely
99	007, Piping Segment, 14H-P3-HC-41X001-03-002	34.00	41.56	156.01	-114.44	Not Likely
100	008, Piping Segment, 14H-P3-HC-41X001-03-003	27.45	148.09	158.22	-10.13	Not Likely
101	009, Piping Segment, 14H-P3-HC-41X134-01-001	27.45	148.35	158.22	-9.87	Not Likely
102	010, Piping Segment, 14H-P3-HC-41X134-01-002	27.45	148.61	158.22	-9.61	Not Likely
103	011, Piping Segment, 14H-P3-HC-41X134-01-003	27.45	148.84	158.22	-9.39	Not Likely
104	012, Piping Segment, 14H-P3-HC-41X134-01-004	27.45	149.13	158.22	-9.09	Not Likely
105	013, Piping Segment, 14H-P3-HC-41X134-01-005	27.45	149.26	158.22	-8.96	Not Likely
106	014, Piping Segment, 14H-P3-HC-41X134-01-006	27.45	149.27	158.22	-8.95	Not Likely
107	015, Piping Segment, 14R-P3-HC-41X134-01-001	27.45	150.64	158.22	-7.58	Not Likely
108	016, Piping Segment, 14R-P3-HC-41X134-01-002	27.45	150.72	158.22	-7.50	Not Likely
109	017, Piping Segment, 14R-P3-HC-41X134-01-003	27.45	150.79	158.22	-7.43	Not Likely
110	018, Piping Segment, 14R-P3-HC-41X134-01-004	27.45	150.80	158.22	-7.42	Not Likely
111	019, Piping Segment, 14D-P3-HC-41X134-01-001	27.45	151.02	158.22	-7.20	Not Likely
112	020, Piping Segment, 14D-P3-HC-41X134-01-002	27.45	151.03	158.22	-7.19	Not Likely
113	021, Reducer, 14D-P3-HC-41X134-01-003	27.45	155.36	158.22	-2.86	Not Likely
114	022, Piping Segment, 14D-P3-HC-41X134-01-004	24.79	155.59	159.12	-3.53	Not Likely
115	023, Piping Segment, HV-41X016	24.79	160.24	159.12	1.12	Yes
116	024, Piping Segment, 14D-P3-HC-41X134-01-005	24.79	160.79	159.12	1.67	Yes
117	025, Piping Segment, 14D-P3-FD-41X136-01-001	56.28	160.24	148.48	11.76	Yes
118	026, Piping Segment, 14D-P3-FD-41X136-01-002	56.28	160.77	148.48	12.30	Yes
119	027, Piping Segment, 14D-P3-FD-41X136-01-003	56.28	160.86	148.48	12.38	Yes
120	028, Piping Segment, 14D-P3-HC-41X135-02-001	56.28	162.53	148.48	14.05	Yes
121	029, Piping Segment, 14D-P3-HC-41X135-02-002	56.28	162.59	148.48	14.11	Yes
122	030, Piping Segment, 14D-P3-FD-419068-01-001	56.28	167.93	148.48	19.45	Yes
123						
124	The IEC acoustic efficiency was used to calculate sound power levels					

is defined as:

$$F_l = \sqrt{\frac{P_1 - P_2}{P_1 - P_{vc}}} \tag{41}$$

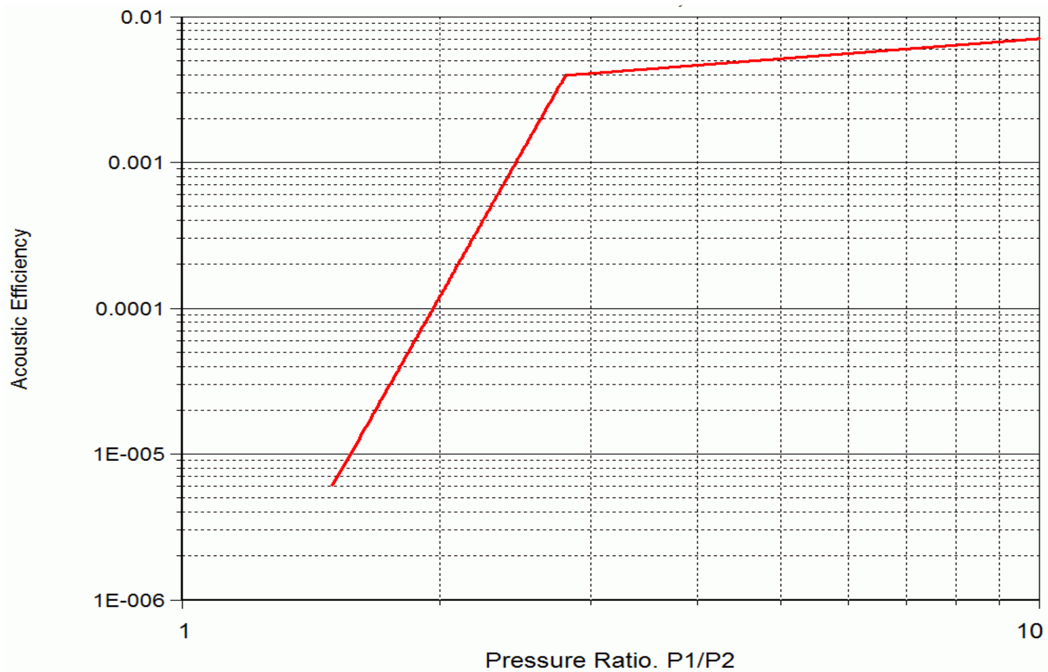
where $P_1 - P_2$ is the pressure differential across the valve and P_{vc} is the pressure at the vena contracta (note that P_2 is larger than P_{vc} since the pressure at the exit of the valve recovers):

$$P_{vc} = P_1 - \frac{P_1 - P_2}{F_l^2} \tag{42}$$

For liquid flow, P_{vc} can reach the vapor pressure of the liquid and cause choked flow.

$$P_{vc} = F_f P_v \simeq \left[0.96 - 0.25\sqrt{\frac{P_v}{P_c}}\right] P_v \tag{43}$$

Figure 7: Acoustic efficiency of shock noise generated by choked jets, η vs. jet pressure ratio P_1/P_2 [1]



where F_f is the liquid critical pressure factor, P_v is the liquid vapor pressure, and P_c is the liquid critical pressure.

The lowest possible value of P_{vc} in a valve flowing liquid, would be vacuum or 0. For this particular limiting case, Equation 41 can be solved for the pressure ratio P_1/P_2:

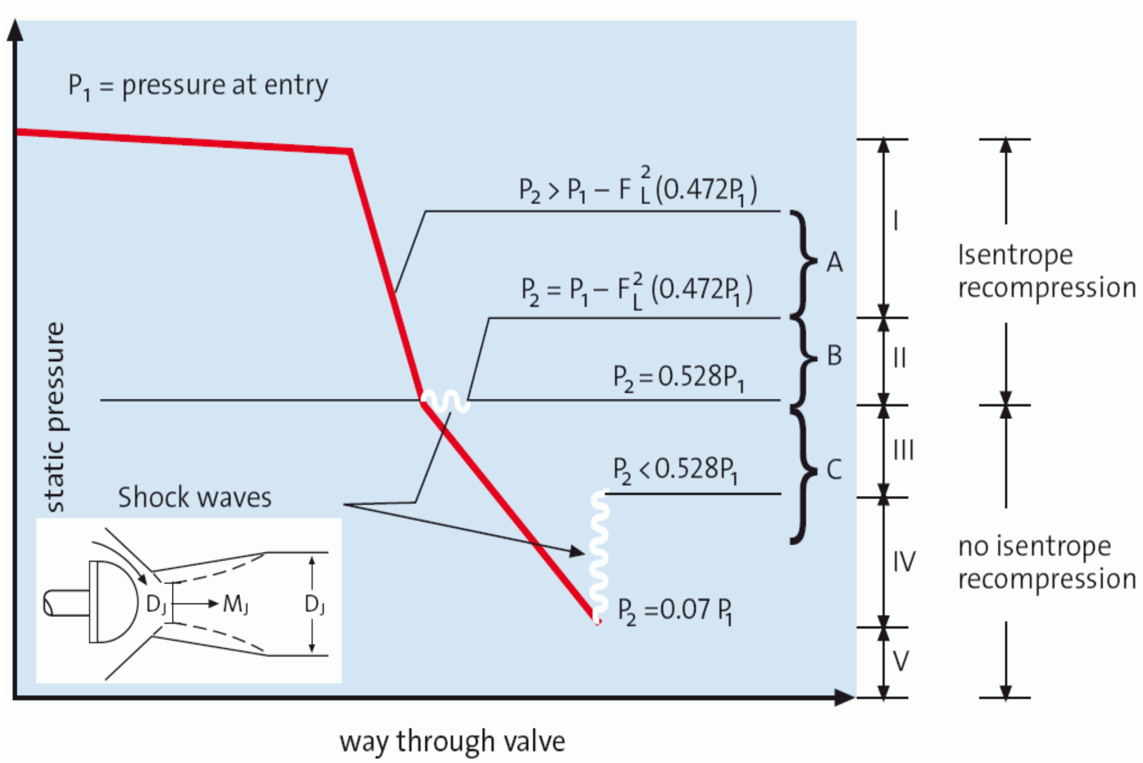
$$P_d = \frac{P_1}{P_2} = \frac{1}{1 - F_l^2} \tag{44}$$

where P_d is the damaging pressure ratio. If the valve is operated at a pressure ratio exceeding P_d, the flow will be choked, noisy, and subject to excessive vibration. Table 2 shows typical valve values for F_l and P_d.

Within every flowing pipe there will also be a standing wave moving axially back and forth in the pipe. The frequency of this wave depends on the effective acoustic length of the pipe and the sonic velocity of the fluid in the pipe. The effective acoustic length of the pipe is the distance between obstructions or acoustic barriers. Examples of obstructions would be valves, pumps, and orifices. An acoustic barrier would be an opening into a larger pipe, a reservoir, the end of a pipe run such as a T intersection where the branch of interest requires a right angle turn. Piping components such as expanders or reducers could be an obstruction. Any analysis should look at the frequencies with and without the expanders as obstructions.

The frequency of the standing wave can be calculated as shown below and then compared to the natural frequencies of valve components and the piping system to determine if there is a potential for vibration and/or resonance.

Figure 8: Flow regimes considered for the estimation of acoustical efficiency



Closed End Pipe $f = \frac{i * u_{ac}}{4L}$

Open End Pipe $f = \frac{i * u_{ac}}{2L}$

where $i = 1, 2, \ldots$ is the wave number, L is the length between acoustic barriers, and u_{ac} is a characteristic acoustic speed throughout the pipe contents defined as follows:

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1 + \frac{dK}{\delta E}}} \tag{45}$$

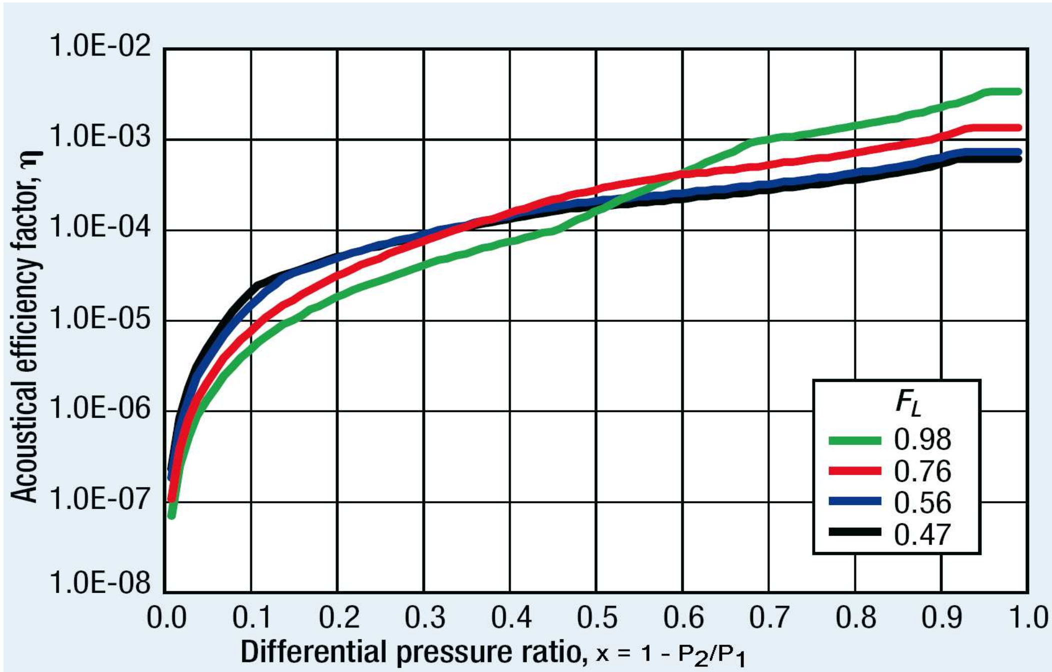
where u_{sonic} is the speed of sound in the fluid, K is the bulk modulus of elasticity in the fluid, d is the pipe diameter, δ is the pipe thickness and E is the pipe material of construction modulus of elasticity.

The above equation is derived from a more general form that depends on the elastic properties of the pipe:

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1 + \frac{K}{E}\psi}} \tag{46}$$

where ψ is a function of the elastic properties of the pipe. Typical expressions of ψ are shown in Table 4. Typical material properties for ψ are shown in Table 3. Note that materials properties

Figure 9: IEC Flow Acoustical Efficiency as a Function of F_L and Differential Pressure Ratio



change with temperature as shown in Figure 10 The speed of sound will change as a function of pressure, temperature, and composition. The presence of dissolved gas nuclei (such as air or other gases) in compressed liquids can significantly reduce the sonic velocity following a pressure drop which leads to the formation of gas bubbles. Up to 40 % reduction in sonic velocity has been observed (see Streeter and Wylie [12]).

To control the vibration caused by a standing wave it is necessary to change the magnitude and/or the frequency of the standing wave or to change the natural frequency of the pipe or components being excited by the wave. The best approach is to address the magnitude of the standing wave. The magnitude is related to the fluid turbulent energy that is enforcing the wave. The most dominant source of this turbulence is the kinetic energy generated by the fluid jet exiting the valve trim. Thus a valve change with a trim that reduces this jet energy will eliminate this wave influence. Trying to change the frequencies is usually not beneficial. There is such a wide range of frequencies present in the turbulent flow that excitation can continue to establish a strong wave at the new frequency and continue the piping vibration.

13 The Singing Safety Relief Valve Problem

As a result of increasing steam flow rates, several boiling water reactor (BWR) nuclear power plants have recently experienced the excitation of acoustic standing waves in closed side branches, e.g., safety relief valves (SRVs), due to vortex shedding generated by steam flow in the main steam lines (see Figure 11). Flow past a valve entrance cavity excites a standing wave, resulting in noise and vibration [13]. A similar tone is produced when air is blown across the mouth of a glass bottle.

Table 2: Typical valve values for F_l and P_d

Body	Trim	Flow Direction	F_l	$P_d = \frac{P_1}{P_2}$
Single Seat Globe	Cage	Open	0.90	5.3
	Cage	Closed	0.80	2.8
	V Plug	Open	0.90	5.3
	V Plug	Closed	0.90	5.3
	Contoured	Open	0.90	5.3
	Contoured	Closed	0.80	2.8
Double Seat Globe	V Plug		0.90	5.3
	Contoured		0.85	3.6
	Standard Bore		0.55	1.4
	Characterized		0.57	1.5
Angle	Cage	Open	0.85	3.6
	Cage	Closed	0.80	2.8
Ball	0.8 dia. Orifice		0.55	1.4
Butterfly	60°	Open	0.68	1.8
Butterfly	90°	Open	0.55	1.4

The amplitude of the acoustic pressure waves can be several times higher than the dynamic pressure present in the system (see Figure 12). The acoustic waves propagate in the steam lines, eventually reaching sensitive components such as steam dryers and turbine stop valves. In addition, the acoustic waves generated in the side branches may generate vibration problems locally and may lead to complications such as valve-seat wear. Therefore, the structural components are subjected to high-cycle fatigue loads, which over time may severely impact those components functionality and safety.

Resonance occurs when the vortex shedding frequency coincides with the acoustic frequency of the standpipe or the valve components. The natural frequency of the standpipe/valve combination for a closed end pipe is given by the following equation:

$$f_a = \left(\frac{2n-1}{4}\right)\frac{u_{ac}}{L+L_e} = \left(\frac{2n-1}{4}\right)\frac{u_{ac}}{L+0.425d} \tag{47}$$

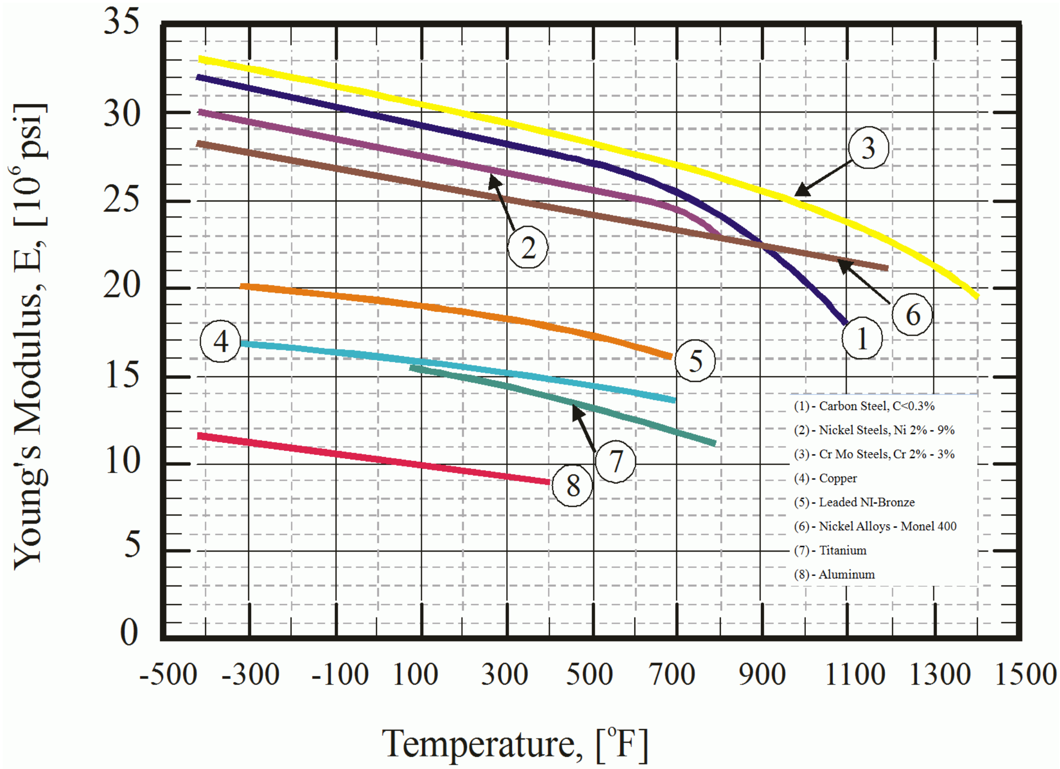
where n is the mode number (1 for 1st mode, 2 for 3rd mode, etc.), u_{ac} is the acoustic speed through the pipe contents as defined earlier, and L_e is an end correction corresponding to Rayleigh's upper limit.

The frequency of pressure oscillations (sound) created by vortex shedding, the energy source for the standing waves, is given by the following equation:

$$f_s = N_{str}\frac{u}{d+r} \simeq 0.33\,(n-0.25)\,\frac{u}{d+r} \tag{48}$$

Typically peak oscillations occur at a Strouhal Number around 0.4 as shown in Figure 12. Note that the root mean square pressure amplitude shown in Figure 12 is the ratio of pressure oscillations

Figure 10: Temperature effects on material of construction



divided by dynamic pressure ($\frac{1}{2}\rho u^2$). RMS begins increasing at a specific onset Strouhal Number and flow velocity depending on acoustic speed, pipe diameter, and pipe length, reaches a peak value and then decreases.

There are many similar installations of pressure relief valves in the process industries where the valves are mounted on large process lines such as overhead lines for distillation columns. In order to avoid fatigue failure from resonance caused by the coupling of normal flow vortex shedding frequency and the acoustic frequency of the standpipe ($f_s = f_a$), the normal flow velocity in the main line has to be limited to less than this critical value:

$$\begin{aligned} u_{max} &< \frac{f_s}{N_{st}}(d+r) < \frac{1}{4}\left(\frac{u_{ac}}{L+L_e}\right)\left(\frac{d+r}{N_{str}}\right) \\ &< \frac{1}{4}\left(\frac{u_{ac}}{L+0.425d}\right)\left(\frac{d+r}{0.6}\right) \end{aligned} \tag{49}$$

As shown in Figure 12 the pressure fluctuations start to increase at a Strouhal number of 0.6 and then decrease after they reach a peak value around a Strouhal number of 0.4.

The same approach can be applied to the flow through the inlet and/or discharge line of a pressure relief valve:

$$u_{max} < \frac{1}{2}\frac{u_{ac}}{L}\frac{D}{0.6} \tag{50}$$

Figure 11: The Singing Safety Relief Valve [2]

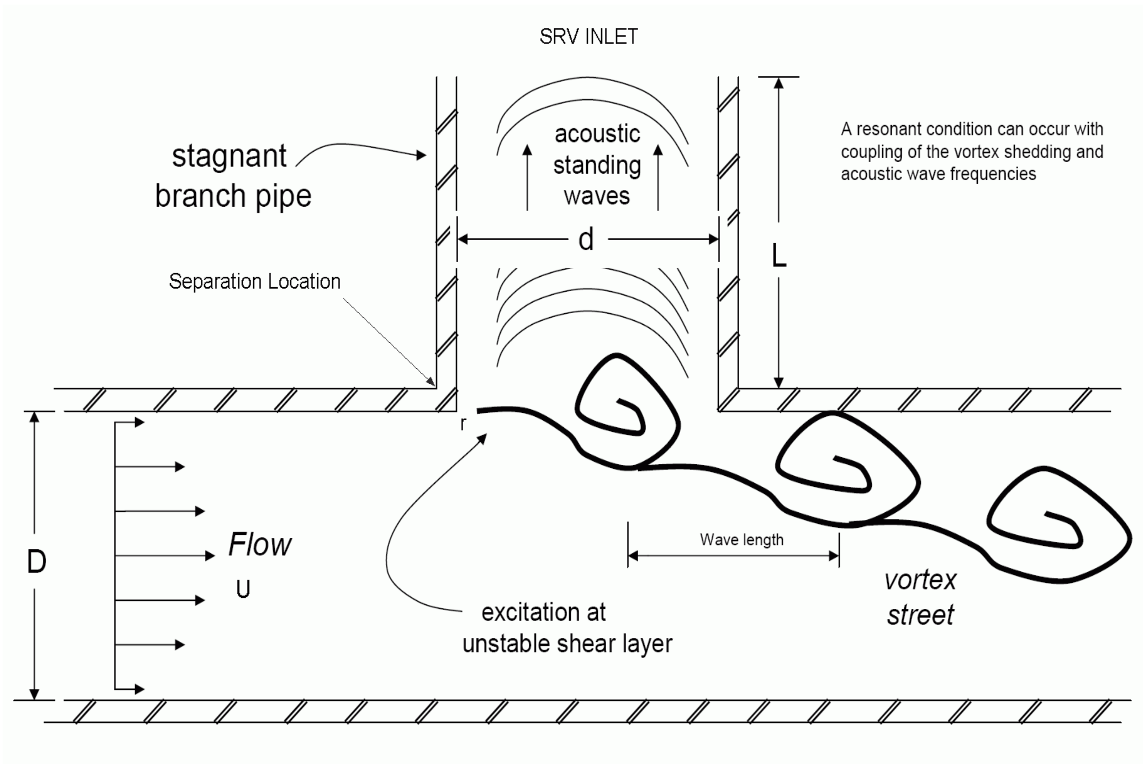


Figure 12: Comparison of Calculated and Measured Pressure Fluctuations as a Function of Strouhal Number [3]

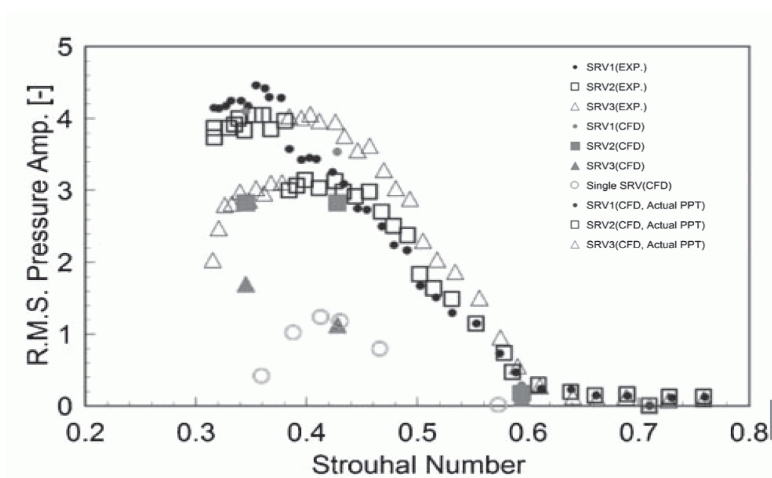


Table 3: Typical data used in the estimation of sonic velocity in pipelines

Material	E (GPa)	Poisson's Ratio ν	$K = \frac{1}{\kappa}$ (GPa)	ρ in $\mathrm{kg/m3}$
Aluminum	69	0.33		
Brass	78-110	0.36		
Carbon steel	202	0.303		
Cast iron	90-160	0.25		
Concrete	20-30	0.15		
Copper	117	0.36		
Ductile iron	172	0.30		
Fibre cement	24	0.17		
High carbon steel	210	0.295		
Inconel	214	0.29		
Mild steel	200-212	0.27		
Nickel steel	213	0.31		
Plastic / Perspex	6.0	0.33		
Plastic / Polyethylene	0.8	0.46		
Plastic / PVC rigid	2.4-2.75			
Stainless steel 18-8	201	0.30		
Water - fresh			2.19	999 at 20 C
Water - sea			2.27	1025 at 15 C

E is typically referred to as Young's modulus of elasticity
G is typically referred to as modulus of torsion, $G = \frac{1}{2}\frac{E}{1+\nu}$

where L is the acoustic length of the inlet or discharge line. Resonance can also be checked by comparing an open pipe/contents frequency with the natural frequency of the pressure relief valve, f_n:

$$f_n = \frac{1}{\tau_n} = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{K_s}{m_D}} \tag{51}$$

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_s}} \tag{52}$$

where τ_n is the undamped natural period in s, and f_n is the undamped natural frequency in Hz where one Hz equals 1 cycle/second, K_s is the spring constant in $\mathrm{N/m}$, m_D is the mass of the valve disc and moving parts in kg, and ω_n is the undamped circular natural frequency in $\mathrm{radians/s}$.

Table 4: Typical expressions for ψ

Pipe condition	ψ
Rigid	0
Anchored against longitudinal movement through its length	$\frac{d}{\delta}\left(1-\nu^2\right)$
Anchored against longitudinal movement at the upper end	$\frac{d}{\delta}\left(1.25-\nu\right)$
Frequent expansion joints present	$\frac{d}{\delta}$

14 Conclusions

This papers demonstrates that the Carucci and Mueller equation can produce unrealistic values of acoustic efficiency, well in excess of 1 percent for high pressure systems and/or systems with large mechanical flow energy. Thus, it is recommended that the Carucci and Mueller acoustic efficiency value be limited to a maximum of 2 percent if the calculated value exceeds 2 percent. The revised experience based failure criteria by Melhem (see Figure 5) based on the IEC acoustic efficiency is now recommended for single and multiphase flow.

The piping vibration risk assessment tools incorporated in SuperChems Expert are versatile and incorporate the methods outlined by the Energy Institute and those originally proposed by Carucci and Mueller. However, the SuperChems Expert methods can accurately calculate the absolute value of the sound power level at every axial piping location for single and multiphase flow using a more fundamental representation of sound power level and more realistic acoustic efficiencies. The SuperChems Expert solution can inherently show the impact of using pipe expansions and limiting flow orifice plates for example to reduce the sound power level and to alter the noise peak frequency. The SuperChems Expert implementation is immediately applicable to headers and flare networks where the flow mechanical energy is automatically calculated at the various flow/flare network nodes.

15 Appendix A: Internal Pipe Noise

The flow mechanical energy converted to noise can be related to sound pressure for reflection free planar waves inside the pipe:

$$W = \frac{P^2 A_i}{\rho u_{sonic}} = \frac{\pi D_i^2 P^2}{4 \rho u_{sonic}} \tag{53}$$

where u_{sonic} is the downstream fluid speed of sound, ρ is the downstream fluid density, A_i is the pipe cross sectional flow area, and D_i is the inside pipe diameter. The sound power level can be calculated in decibels refereced to 2×10^{-5} Pascals:

$$L_{P,i} = 10 \log_{10} \left[\frac{P^2}{(2 \times 10^{-5})^2} \right] \tag{54}$$

$L_{P,i}$ can be expressed as a function of W:

$$L_{P,i} = 10 \log_{10} \left[\frac{\rho u_{sonic} W}{\pi D_i^2 10^{-10}} \right] = 10 \log_{10} \left[3.183 \times 10^9 \frac{\rho u_{sonic} W}{D_i^2} \right] \tag{55}$$

The above equation can be used to calculate to the total internal sound pressure assuming 100 % of the noise is transmitted downstream. If the flow exits at an angle, only a portion of the total sound pressure level is transmitted downstream:

$$L_{P,i} = 10 \log_{10} \left[3.183 \times 10^9 \frac{\zeta \rho u_{sonic} W}{D_i^2} \right] \tag{56}$$

The value of ζ is 0.25.

A frequency dependent internal sound pressure level can be calculated as a function of the peak frequency f_p established earlier and a specific center frequency f_i:

$$L_{P,i}(f_i) = L_{P,i} + L_{f,i} = L_{P,i} - c - 10 \log_{10} \left[\left(1 + \left[\frac{f_i}{2 f_p} \right]^2 \right) \left(1 + \left[\frac{f_p}{2 f_i} \right]^4 \right) \right] \tag{57}$$

where the constant c is 7.9 for one-third octave center frequencies and 3 for octave center frequencies. Summing $L_{f,i}$ over the entire frequency spectrum should yield a 0:

$$0 = 10 \log_{10} \Sigma_{i=1}^{n=33} 10^{\frac{L_{f,i}}{10}} \tag{58}$$

Table 5: Weighting factor for one-third octave frequencies

f_i in Hz	$W(f_i)$	f_i in Hz	$W(f_i)$
10	-70.4	500	-3.2
12.5	-63.4	630	-1.9
16	-56.7	800	-0.8
20	-50.5	1000	0
25	-44.7	1250	0.6
31.5	-39.4	1600	1
40	-34.6	2000	1.2
50	-30.2	2500	1.3
63	-26.2	3150	1.2
80	-22.5	4000	1
100	-19.1	5000	0.5
125	-16.1	6300	-1.0
160	-13.4	8000	-1.1
200	-10.9	10000	-2.5
250	-8.6	12500	-4.3
315	-6.6	16000	-6.6
400	-4.8	20000	-9.3

16 Appendix B: External Pipe Noise

The pipe absorbs some of the noise and only a portion of the total internal noise escapes to the atmosphere. The sound pressure level at a specific distance from the outer surface of the pipe is given by the following equation:

$$L_{P,e}(f_i) = L_{P,i}(f_i) + TL(f_i) + W(f_i) - 10\log_{10}\left[\frac{2x + 2\delta + D_i}{D_i + 2\delta}\right] \tag{59}$$

where x is the distance from the outer pipe wall, $TL(f_i)$ is the transmission loss at frequency f_i, $W(f_i)$ is the A-weighting factor (see Table 5) for the one-third octave band center frequency f_i, and $L_{P,i}(f_i)$ was defined earlier.

The total sound pressure level received at a location x away from the wall of the pipe can be calculating by summing all the frequency contributions:

$$L_{P,e} = 10\log_{10}\Sigma_{i=1}^{n=33} 10^{\frac{L_{P,e}(f_i)}{10}} \tag{60}$$

The transmission loss can either be ignored (set to 0) or calculated using the method outline by Kiesbauer and Vnucec in 2008. The tranmission loss depends on pipe wall thickness and the downstream fluid properties:

$$TL(f_i) = 10\log_{10}\left[7.6\times 10^7\left(\frac{u_{sonic}}{\delta f_i}\right)^2 \frac{G_x(f_i)}{1+\frac{\rho u_{sonic}}{415 G_y(f_i)}}\right] - \Delta TL(D_i) \tag{61}$$

where $\Delta TL(D_i)$ is 0 for $D_i > 0.15$ and 9.7 for $D_i < 0.05$. Otherwise it is give by:

$$\Delta TL(D_i) = \frac{16}{(1000D_i - 46)^{0.36}} \tag{62}$$

The values for G_x and G_y are given below:

$$\begin{aligned}
f_r &= \frac{5000}{\pi D_i} && (63)\\
f_o &= \frac{f_r}{4}\frac{u_{sonic}}{343} && (64)\\
f_g &= \frac{\sqrt{3}}{\pi\delta}\frac{343^2}{5000} && (65)\\
G_x(f_i) &= \left(\frac{f_o}{f_r}\right)^{2/3}\left(\frac{f_i}{f_o}\right)^4 \text{ for } f_i < f_o && (66)\\
&= 1 \text{ for } f_i \geq f_o \text{ and } f_i \geq f_r && (67)\\
&= \sqrt{\frac{f_i}{f_r}} \text{ for } f_i \geq f_o \text{ and } f_i < f_r && (68)\\
G_y(f_i) &= 1 \text{ for } f_i < f_o \text{ and } f_o \geq f_g && (69)\\
&= \frac{f_o}{f_g} \text{ for } f_i < f_o \text{ and } f_o < f_g && (70)\\
&= 1 \text{ for } f_i \geq f_o \text{ and } f_i \geq f_g && (71)\\
&= \frac{f_i}{f_g} \text{ for } f_i \geq f_o \text{ and } f_i < f_g && (72)
\end{aligned}$$

where f_r is the ring frequency and f_o and f_g are the coincidence pipe frequencies.

References

[1] I. Heitner. How to estimate plant noises. *Hydrocarbon Processing*, 47:67–74, 1968.

[2] P. Sekerak. Potential adverse flow effects at nuclear power plants. In *16th International Conference on Nuclear Engineering*. ICONE16-48900, 2008.

[3] K. Okuyama F. Inada Y. Ogawa R. Morita, S. Takahashi and K. Yoshikawa. Evaluation of acoustic and flow induced vibration of the bwr main steam lines and dryer. *Journal of NUCLEAR SCIENCE and TECHNOLOGY*, 48(5):759–776, 2011.

[4] W. A. Skipwith. Acoustics technology. Survey SP-5093, NASA National Aeronautics and Space Administration, 1970.

[5] C. M. Harris. *Handbook of Noise Control*. cGraw-Hill Book Co., Inc., 1957.

[6] N. R. Miller G. M. Singh, E. Rodarte and P. S. Hrnjak. Modification of a standard aeroacoustic valve noise model to account for friction and two-phase flow. In *ACRC TR-162*, pages 1–11. Mechanical and Industrial Engineering Department, University of Illinois, 2000.

[7] N. R. Miller G. M. Singh, E. Rodarte and P. S. Hrnjak. Prediction of noise generated by expansion devices throttling refrigerant. In *ACRC TR-163*, pages 1–13. Mechanical and Industrial Engineering Department, University of Illinois, 2000.

[8] MTD. *Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework*. Marine Technology Directorate Limited (MTD), 1999.

[9] Energy Institute. *Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework*. The Energy Institute, London, England, 2008.

[10] V. A. Carucci and R. T. Mueller. Acoustically induced piping vibration in high capcity pressure reducing systems. In *82-WA/PVP-8*, pages 1–13. American Society of Mechanical Engineers, ASME, 1982.

[11] F. L. Eisinger. Designing piping systems against acoustically-induced structural fatigue. In *PVP-VOL 328, Flow-Induced Vibration*, pages 397–404. American Society of Mechanical Engineers, ASME, 1982.

[12] V. L. Streeter and E. B. Wylie. Water hammer and surge control. *Annual Reviews in Fluid Mechanics*, pages 57–74, 1974.

[13] T. M. Mulcahy S. A. Hambric and V. N. Shah. Flow-induced vibration effects on nuclear power plant components due to main steam line valve singing. In *Ninth NRC/ASME Symposium on Valves, Pumps, and Inservice Testing, NUREG/CP-0152*, volume 6, pages 3B:49–3B:69. NRC/ASME, 2006.

ioMosaic

Business Confidential Document

Assess Piping Vibration Risk with Experience Based Methods

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

DIERS Users Group Meeting

May 2012

Kansas City

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

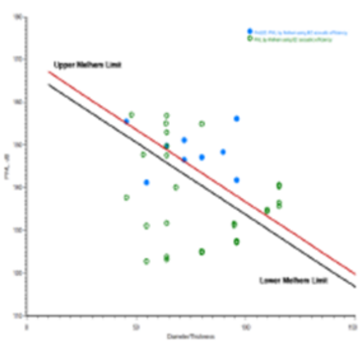
ioMosaic

This presentation is also available as a white paper

ioMosaic

Estimate Vibration Risk for Relief and Process Piping

May 2, 2012



Notice:

This report was prepared by ioMosaic for public disclosure. This report represents ioMosaic's best judgment in light of information made available to us. Opinions in this report are based in part upon data and information obtained from the open literature. The reader is advised that ioMosaic has not independently verified the data or the information contained therein. This report must be read in its entirety. The reader understands that no assurances can be made that all liabilities have been identified. This report does not constitute a legal opinion.

No person has been authorized by ioMosaic to provide any information or make any representations not contained in this report. Any use the reader makes of this report, or any reliance upon or decisions to be made based upon this report are the responsibility of the reader. ioMosaic does not accept any responsibility for damages, if any, suffered by the reader based upon this report.

ioMosaic Corporation
93 Stiles Road
Salem, New Hampshire
03079 U.S.A.

diated to the downstream piping, which becomes the transmitting medium. As the noise travels downstream along the inside of the pipe in the fluid it radiates through the pipe wall along its entire length.

We can calculate the sound power level in a fundamental way by assigning an efficiency factor to provide the fraction of flow mechanical energy that is converted to noise:

$$W = \eta \frac{1}{2} \dot{m} u^2 \qquad (1)$$

$$L_W = 10 \log_{10}\left[\frac{W}{10^{-12}}\right] = 10 \log_{10} W + 120 = 10 \log_{10}\left[\eta \frac{1}{2} \dot{m} u^2\right] + 120 \qquad (2)$$

where W is the flow mechanical power or energy, L_W is the sound power level in dB, and η is defined as the acoustical efficiency factor. If the flow is choked, then W becomes:

$$W = \eta \frac{1}{2} \dot{m} u^2_{sonic} \qquad (3)$$

If a value of η can be estimated for single and multi-phase flow, then the sound power level can be easily calculated not only for pressure reducing devices but also for pipe flow. Attenuation due to friction and temperature changes can be calculated from pipe flow equations in a more detailed manner. Computer codes such as SuperChems can then calculate the sound power level at every axial location for flow piping for single and multi-phase flow.

For incompressible flow, the value of flow velocity for an ideal nozzle can be calculated from the mechanical energy balance:

$$u = \sqrt{\frac{2}{\rho_l} \Delta P} \qquad (4)$$

Substituting the above equation for u in Equation 1 yields the following Equation for W for liquid flow through an ideal nozzle:

$$W = \eta \dot{m} \frac{\Delta P}{\rho_l} \qquad (5)$$

For liquid flow, a typical value of η is approximately 10^{-8}.

All gas flow acoustic efficiencies have been reported to approach 1 % of the total flow mechanical energy for rockets [4]. This is illustrated in Figure 1. When the acoustic efficiency is plotted against the flow mechanical power Figure 1 is obtained. The measured curve indicates that the acoustic efficiency falls off as the flow mechanical energy gets larger while the calculated curve [5] indicates increasing acoustic efficiencies.

Two-phase flow sound power level can also be calculated using Equation 1. Recent work by Singh et al. ([6], [7]) demonstrated that more attenuation of noise is exhibited by two-phase flow. Therefore, using the gas acoustic efficiency values that typically range from 10^{-6} to 10^{-4}, will overpredict noise levels for two-phase flow.

3

Failure of relief and process piping caused by vibration can develop due to flow energy

- Flow induced turbulence
  - Caused by flow discontinuities in piping, valves, tees, etc.
- High frequency excitations (acoustic energy)
  - Caused by choked flow from a relief device or control valve
- Other causes include
  - Mechanical excitations by compressors, pumps, and other rotating machinery
  - Vortex shedding
  - Water hammer
  - Cavitation
  - Etc.

ioMosaic

Factors that have led to an increasing incidence of vibration related fatigue failures in piping systems include but are not limited to:

- Increasing flow rates as a result of debottlenecking which contributes to higher flow velocities with a correspondingly greater level of turbulent energy
- Frequent use of thin-walled piping which results in higher stress concentrations, particularly at small bore connections
- Design of process piping systems on the basis of a static analysis with little attention paid to vibration induced fatigue
- Lack of emphasis of the issue of vibration in piping design codes
- Piping vibration is often considered on an ad-hoc or reactive basis

According to the UK HSE, 21 % of all piping failures offshore are caused by fatigue/vibration

- Large compressor recycle systems
- Steam de-superheater systems
- High capacity pressure relief depressuring systems
- Other?

The current ISO/API 521 standard does not formally address vibration risk

- It does not offer specific guidance on velocity limitations other than backpressure calculations
- It does not offer guidance on acoustic fatigue or vibration induced fatigue
- Many operating companies have established their own internal guidance for evaluating and minimizing piping vibration risk
- Although these criteria vary from company to company, they all in general include a limit of flow velocity in some form

ioMosaic

Typical criteria used by companies and other organizations to minimize vibration risk

- Limit Mach Number $$0.3 \leq M = \frac{u}{u_{sonic}} \leq 0.8$$

- Limit dynamic pressure for gas flow $$\rho u^2 \leq 1 \times 10^5 \ \ \mathrm{kg/m/s^2}$$

- Limit dynamic pressure for two-phase flow $$\rho u^2 \leq \frac{1}{2} \times 10^5 \ \ \mathrm{kg/m/s^2}$$

- Some companies use different limits for process piping and relief piping (See NORSOK Process Design Standard P-001, 2006, Fifth Edition)

ioMosaic

Sound power level and sound pressure level are all about flow energy

- A small portion of the flow energy is transferred to the pipe wall as vibration energy and a portion of that is radiated as noise

- We can calculate both sound power and sound pressure level in a more fundamental way that is appropriate for complex piping and multi-phase flow

$$W = \eta \frac{1}{2} \dot{m} u^2$$

$$L_W = 10 \log_{10} \left[\frac{W}{10^{-12}} \right] = 10 \log_{10} W + 120 = 10 \log_{10} \left[\eta \frac{1}{2} \dot{m} u^2 \right] + 120$$

- For choked flow

$$W = \eta \frac{1}{2} \dot{m} u^2_{sonic}$$

ioMosaic

ASME has published guidance for the estimation of noise levels for control valves and pressure reducing devices (Based on work by Exxon Research – Carruci and Mueller)

- This guidance is not readily applicable to piping
- This guidance is not readily applicable to multi-phase flow

$$
\begin{aligned}
L_W &= 10 \times \log_{10}\left[\left(\frac{P_1 - P_2}{P_1}\right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w}\right)^{1.2}\right] + 126.1 \\
&= 36\log_{10}\left(\frac{P_1 - P_2}{P_1}\right) + 20\log_{10}(\dot{m}) + 12\log_{10}\left(\frac{T_1}{M_w}\right) + 126.1
\end{aligned}
$$

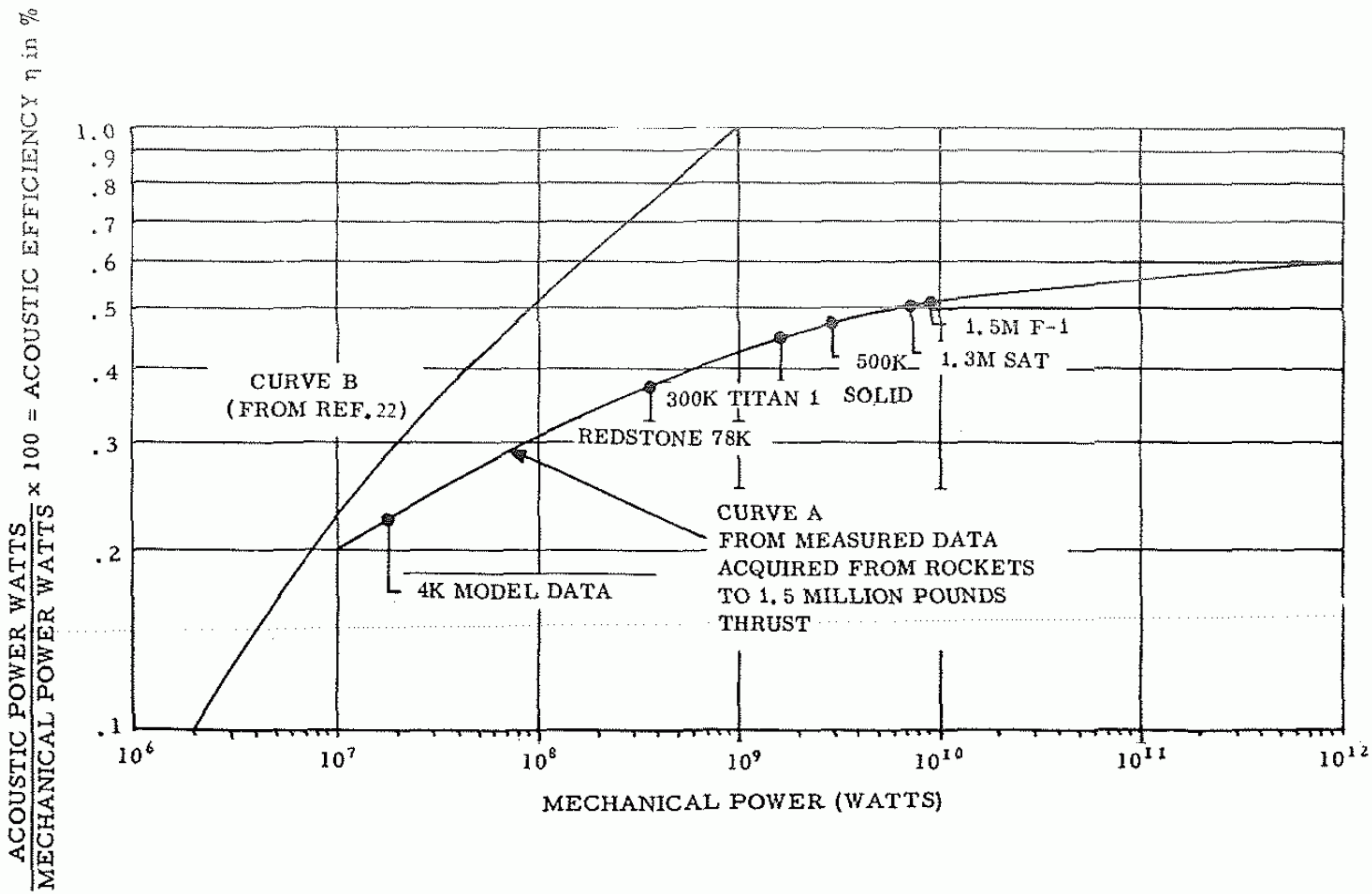
- Noise attenuation (3 dB for every 50 L/D) and for piping expansions

$$L_{W,At} = 0.06\frac{L}{D_i} \qquad L_{W,Ex} = 2\left(\frac{D_2}{D_1} - 1\right)$$

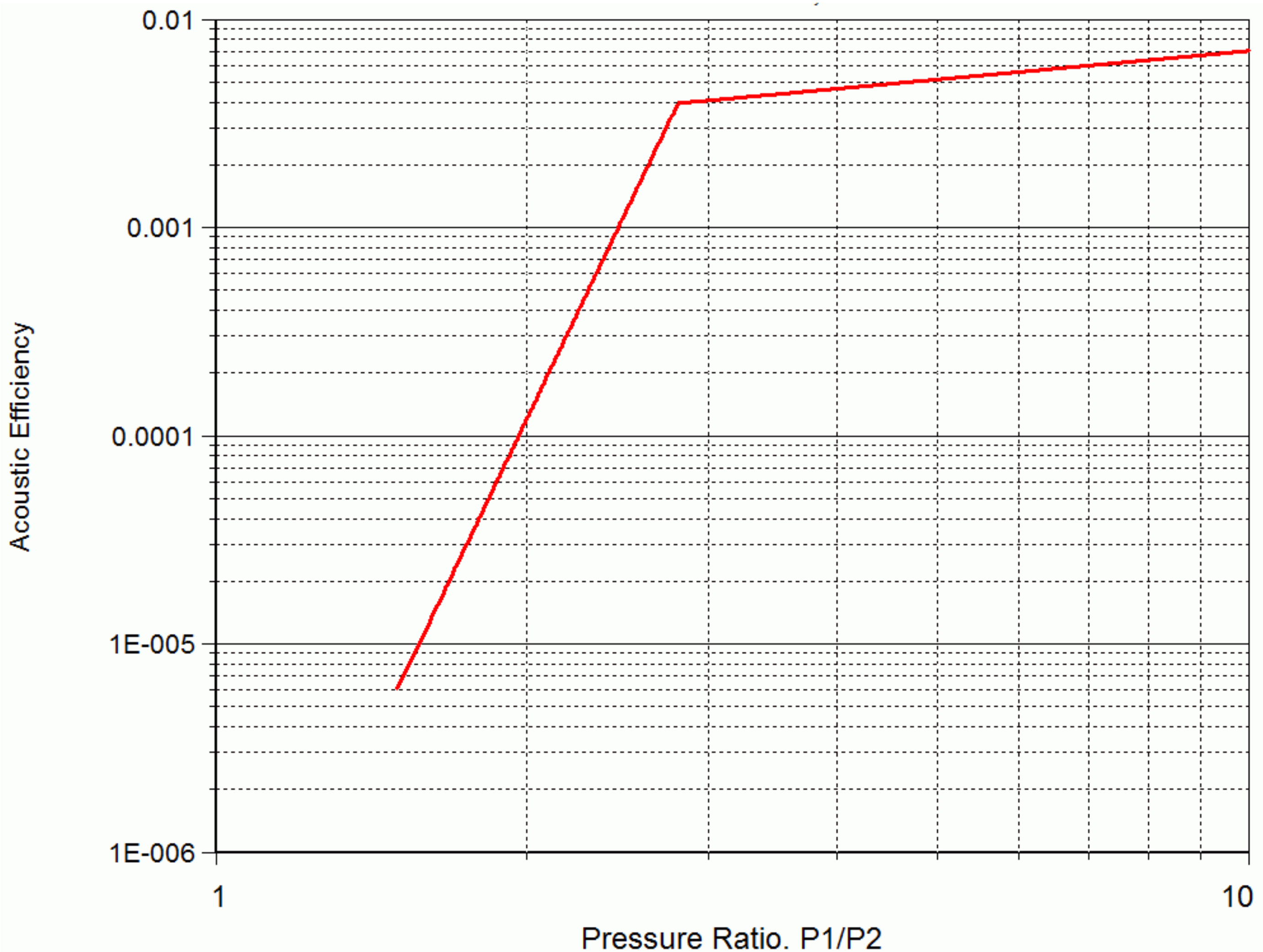
- Add 6 dB to the sound power level estimate when sonic flow exists at a branch connection to account for amplified dynamic response

ioMosaic

Noise efficiency for rockets is shown experimentally to asymptote to about 1 %

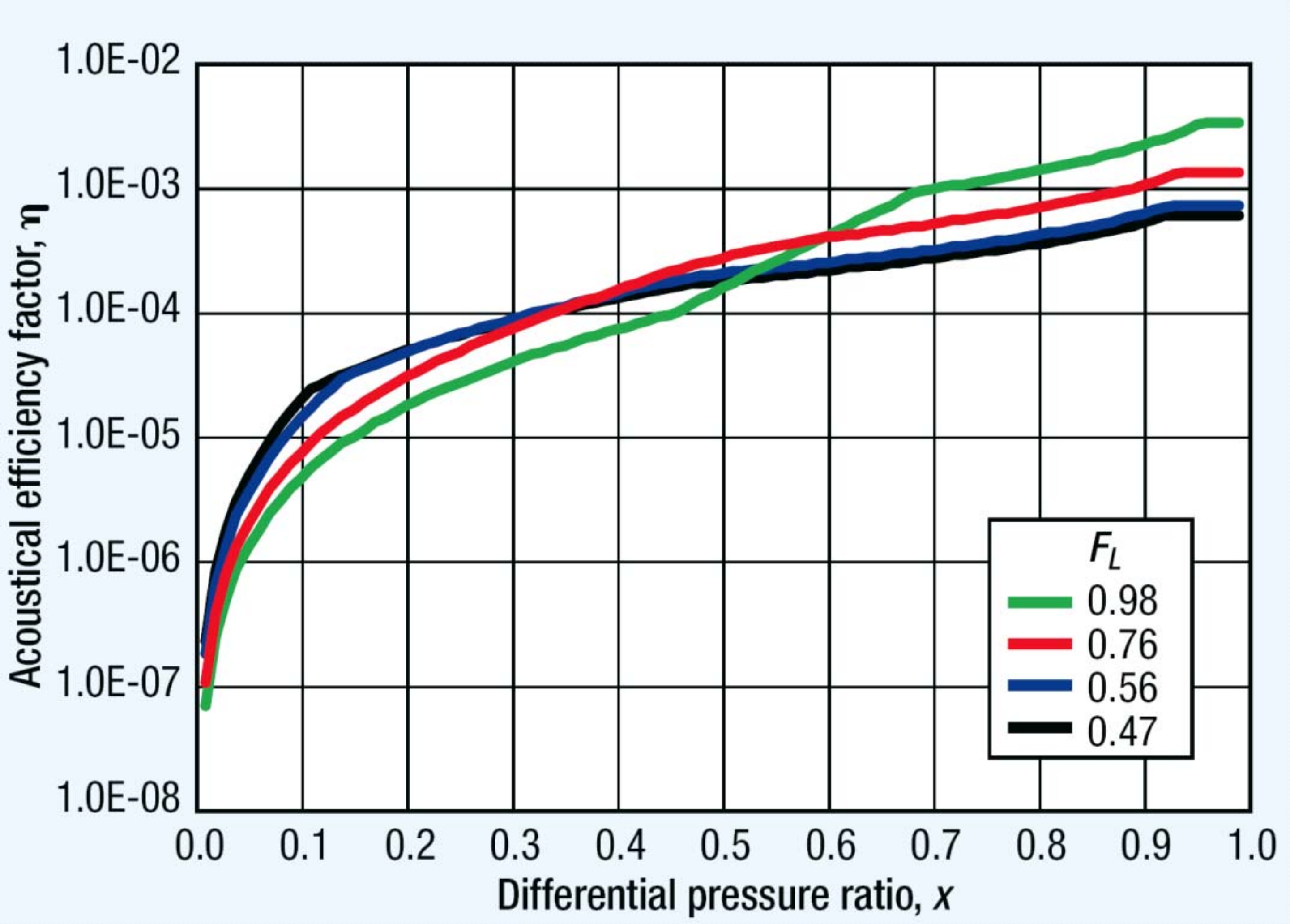


The efficiency represented below is the same used by API for calculating flare noise. Note $P_1/P_2 = 1 / (1-x)$ – Choked free jets

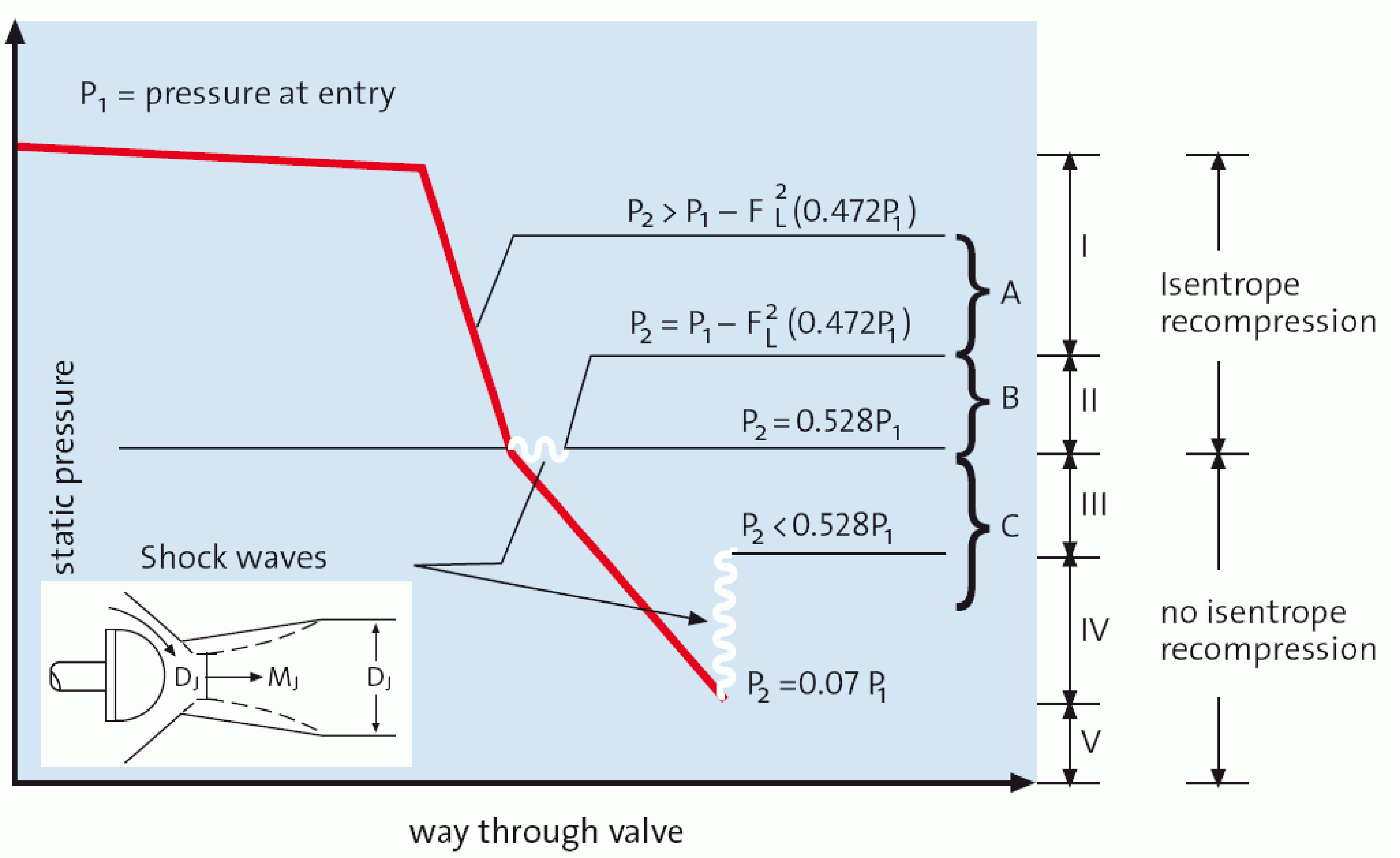


ioMosaic

IEC provides similar efficiency values to the curve used by API for control valves. Note $x = 1 - P_2/P_1$



How do we calculate the "acoustic efficiency factor", η



The acoustic efficiency implied by the Carucci and Mueller method is suspect!

$$W = \eta \frac{1}{2} \dot{m} u^2$$

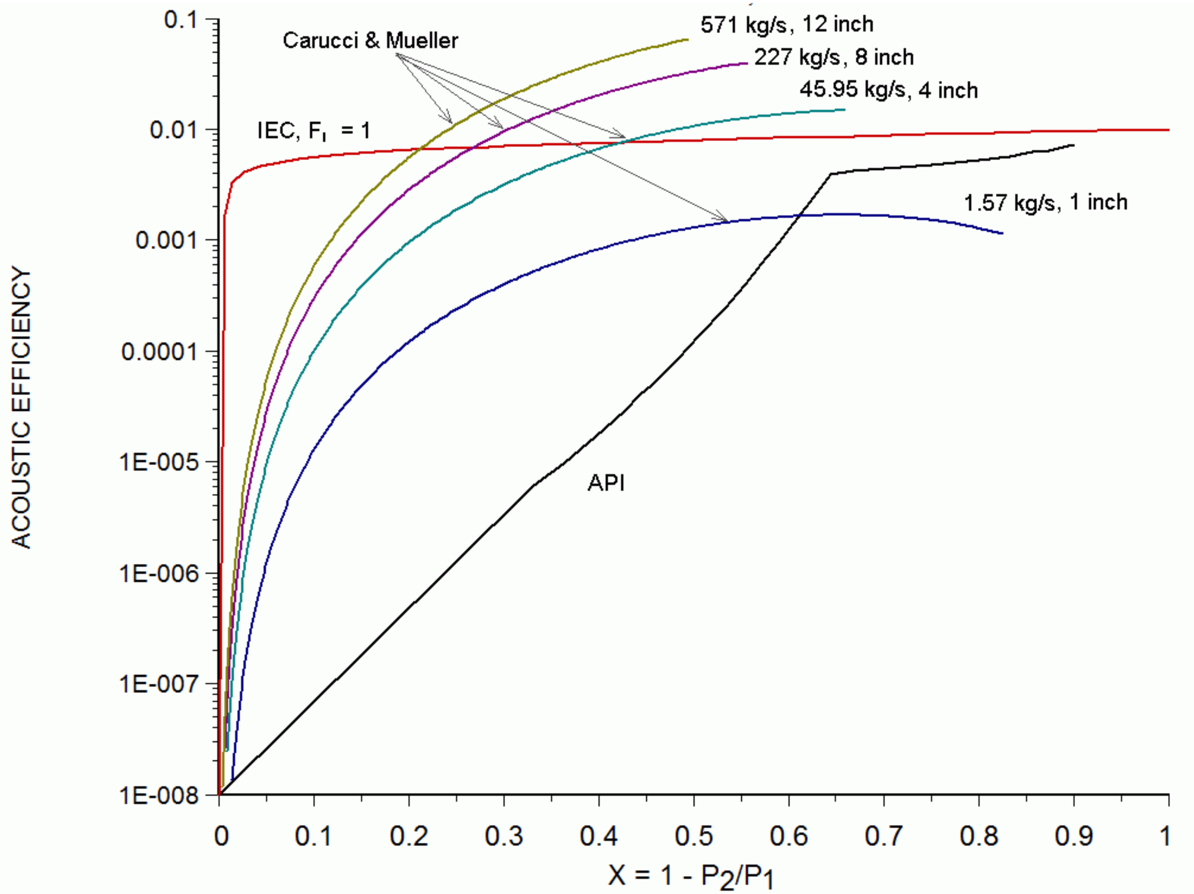
$$L_W = 10 \log_{10} \left[\frac{W}{10^{-12}} \right] = 10 \log_{10} W + 120 = 10 \log_{10} \left[\eta \frac{1}{2} \dot{m} u^2 \right] + 120$$

$$\begin{aligned} L_W &= 10 \times \log_{10} \left[\left(\frac{P_1 - P_2}{P_1} \right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w} \right)^{1.2} \right] + 126.1 \\ &= 10 \times \log_{10} \left[4 \times \left(\frac{P_1 - P_2}{P_1} \right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w} \right)^{1.2} \right] + 120 \\ &= 36 \log_{10} \left(\frac{P_1 - P_2}{P_1} \right) + 20 \log_{10} (\dot{m}) + 12 \log_{10} \left(\frac{T_1}{M_w} \right) + 126.1 \end{aligned}$$

$$\eta = 8 \left(1 - \frac{P_2}{P_1} \right)^{3.6} \left(\frac{T_1}{M_w} \right)^{1.2} \left(\frac{\dot{m}}{u^2} \right)$$

The Carucci and Mueller method is used by most of the major oil companies and the MTD guidelines

- The Carucci and Mueller method is not consistent with acoustic efficiencies reported by IEC and the literature
- The Carucci and Mueller method should not be used to calculate the absolute value of sound power level
- The Carucci and Mueller method overestimates the sound power level and does not extrapolate beyond flow mechanical power of 600,000 Watts or a Mach Number > 0.65



Now that we know how to calculate sound power and sound pressure levels in piping, how do we determine if the piping is going to fail or is likely to fail due to vibration risk?

- Experience based – D/t method (SuperChems for DIERS and Expert)
- Experience based – Mach Number method
- MTD / Energy Institute Guidelines (SuperChems Expert ioVIPER)
- Detailed structural dynamics

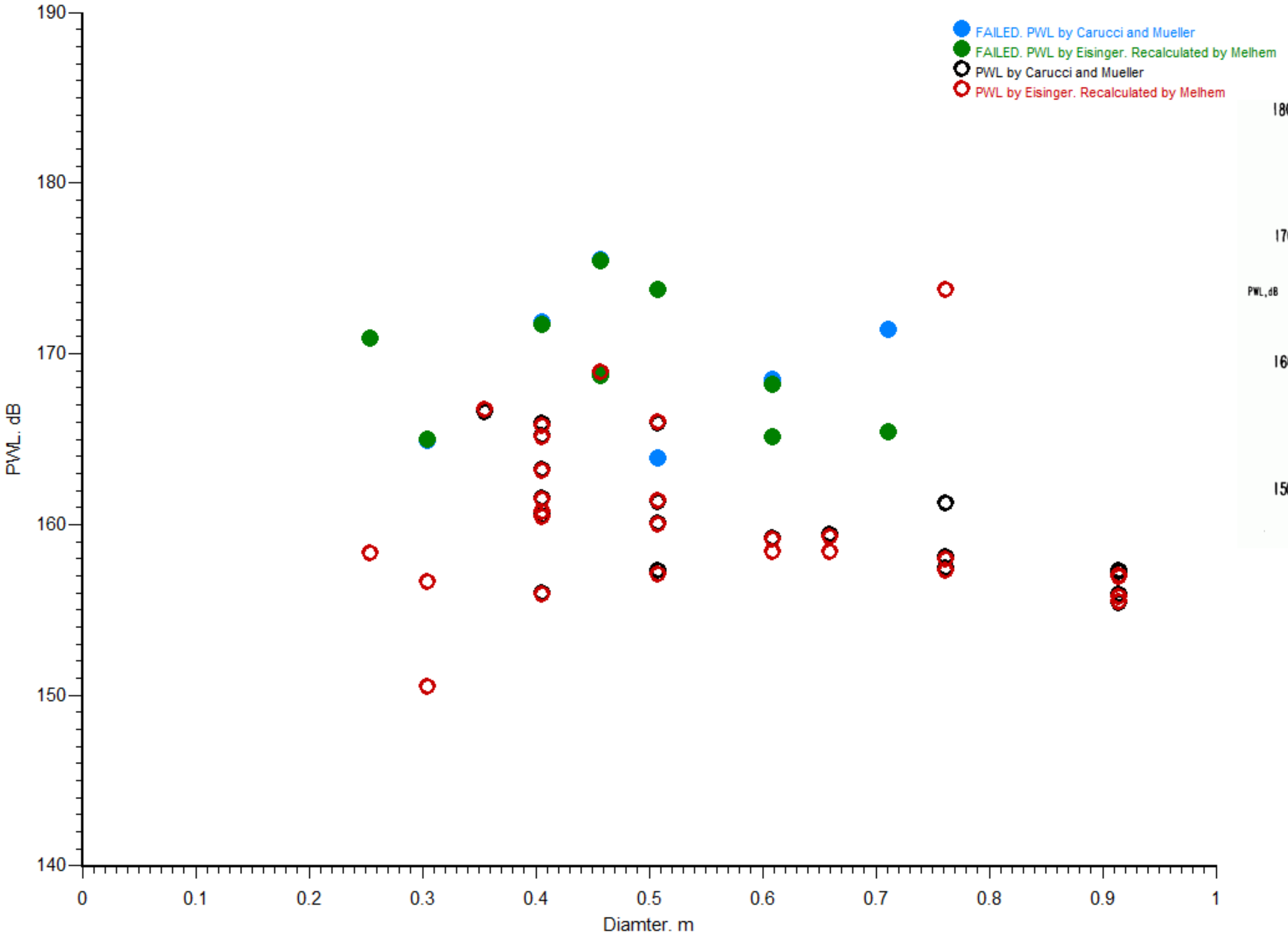
ioMosaic

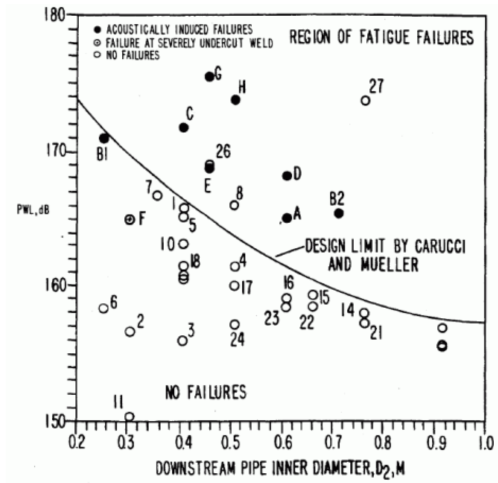
Carucci & Mueller used a small data set to derive their failure criteria

CASE	SERVICE	T_1. K	P_1. bara	T_2. K	P_2. bara	$X = 1 - P_2/P_1$	D_2. m	u2. m/s	Exit Mach Number, M_2	Mass Flow Rate, kg/s	Heat Capcity Ratio, Cp/Cv	Flow Mechanical Enery. MegaWatts	Carucci and Mueller Accustic Efficiency	IEC Accustic Efficiency	PWL Reported by Carucci and Mueller. dB	PWL Reported by Eisinger and Recalculated by Melhem. dB	Difference Between Reported PWL by Carucci and Mueller and Recalculated Eisinger PWL Values by Melhem. dB	PWL Calculated by Melhem Using IEC Acoustic Efficiency
A	Propylene Compressor Recycle	319.26	20.55	238.91	1.59	0.923	0.610	32.500	0.187	29.736	1.128	0.016	2.052	0.010	168.5	165.1	3.4	141.8
B1	Natural Gas Pressure	318.71	42.75	226.00	6.27	0.853	0.254	125.600	0.399	48.258	1.218	0.381	0.322	0.009	170.9	170.9	0.0	155.5
B2	Natural Gas Letdown to Flare	318.71	5.17	270.38	2.07	0.600	0.711	58.200	0.169	48.258	1.219	0.082	0.423	0.008	171.4	165.4	6.0	148.4
C	Natural Gas Letdown to Flare	308.15	47.57	195.58	4.14	0.913	0.406	64.400	0.218	47.628	1.228	0.099	1.498	0.010	171.8	171.7	0.1	149.8
D	Nitrogen Compressor Recycle	310.93	6.96	280.31	3.72	0.465	0.610	93.800	0.297	119.700	1.198	0.527	0.125	0.008	168.3	168.2	0.1	156.1
E	Nitrogen Compressor Recycle	310.93	16.82	267.59	7.03	0.582	0.457	60.100	0.194	84.798	1.208	0.153	0.481	0.008	168.8	168.7	0.1	151.0
F	Natural Gas Compressor Recycle	330.37	51.09	220.53	13.86	0.729	0.305	28.800	0.093	35.280	1.449	0.015	2.133	0.009	164.9	164.9	0.0	141.2
G	Desuperheater Steam	802.59	99.97	480.32	10.34	0.897	0.457	49.800	0.093	36.893	1.292	0.046	7.654	0.010	175.5	175.4	0.1	146.4
H	Desuperheater Steam	658.15	43.09	476.11	10.34	0.760	0.508	49.600	0.093	45.990	1.293	0.057	4.182	0.009	163.9	173.7	-9.8	147.1
1	Process Gas Compressor Recycle	308.15	13.17	204.98	1.03	0.921	0.406	122.800	0.461	27.090	1.191	0.204	0.186	0.010	165.9	165.8	0.1	153.0
2	Process Gas Compressor Recycle	311.48	38.75	265.38	10.00	0.742	0.305	14.200	0.048	13.860	1.132	0.001	3.304	0.009	156.6	156.7	-0.1	131.0
3	Propylene Compressor Recycle	316.48	20.55	262.14	3.24	0.842	0.406	15.700	0.065	12.600	1.114	0.002	2.517	0.009	156.0	155.9	0.1	131.6
4	Process Gas Compressor Recycle	300.37	2.48	257.91	1.03	0.583	0.508	145.600	0.450	34.776	1.211	0.369	0.038	0.008	161.3	161.4	-0.1	154.9
5	Process Gas Compressor Recycle	310.93	15.86	215.12	2.21	0.861	0.406	63.700	0.226	25.326	1.230	0.060	0.547	0.009	165.2	165.1	0.1	147.5
6	Process Gas Compressor Recycle	300.93	39.44	250.57	15.51	0.607	0.254	25.100	0.075	20.664	1.244	0.007	0.958	0.008	158.3	158.3	0.0	137.7
7	Propylene Compressor Recycle	310.93	16.69	233.38	1.38	0.917	0.356	131.300	0.575	38.052	1.129	0.328	0.143	0.010	166.6	166.7	-0.1	155.0
8	Propylene Compressor Recycle	310.93	16.69	234.72	1.38	0.917	0.508	59.200	0.259	35.028	1.127	0.061	0.646	0.010	165.9	166.0	-0.1	147.7
9	Waste Steam Dump	403.71	2.76	370.32	1.03	0.625	0.406	245.100	0.566	18.900	1.095	0.568	0.019	0.008	160.5	160.4	0.1	156.8
10	Process Gas Compressor Recycle	302.59	10.69	207.90	1.03	0.903	0.406	95.800	0.357	20.916	1.191	0.096	0.214	0.010	163.2	163.1	0.1	149.7
11	Process Gas Compressor Recycle	308.15	54.81	263.43	27.51	0.498	0.305	7.200	0.016	9.148	1.294	0.000	4.691	0.008	150.5	150.5	0.0	122.8
12	Natural Gas Compressor Recycle	308.15	3.79	282.54	2.48	0.345	0.914	28.400	0.075	39.690	1.257	0.016	0.220	0.007	155.4	155.5	-0.1	140.6
13	Natural Gas Compressor Recycle	308.15	6.48	279.42	4.00	0.383	0.914	17.300	0.046	39.438	1.254	0.006	0.855	0.007	157.2	157.0	0.2	136.4
14	Natural Gas Compressor Recycle	308.15	11.45	282.82	6.76	0.410	0.762	14.400	0.039	38.934	1.194	0.004	1.553	0.008	158.1	158.0	0.1	134.8
15	Natural Gas Compressor Recycle	308.15	21.79	266.72	11.65	0.465	0.660	10.100	0.027	36.036	1.300	0.002	4.619	0.008	159.4	159.3	0.1	131.6
16	Natural Gas Compressor Recycle	308.15	40.75	272.56	21.99	0.460	0.610	6.300	0.017	35.910	1.248	0.001	11.382	0.008	159.2	159.1	0.1	127.4
17	Natural Gas Compressor Recycle	308.15	80.12	271.44	40.89	0.490	0.508	4.800	0.013	35.406	1.232	0.000	24.451	0.008	160.1	160.0	0.1	125.1
18	Natural Gas Compressor Recycle	308.15	172.99	240.53	79.98	0.538	0.406	3.800	0.010	35.406	1.473	0.000	54.625	0.008	161.5	161.5	0.1	123.2
19	Natural Gas Compressor Recycle	308.15	3.86	280.10	2.34	0.393	0.914	27.600	0.074	37.170	1.237	0.014	0.345	0.007	157.3	156.9	0.4	140.2
20	Natural Gas Compressor Recycle	308.15	6.21	281.91	3.93	0.367	0.914	15.500	0.044	36.792	1.242	0.005	0.746	0.007	155.9	155.7	0.2	135.6
21	Natural Gas Compressor Recycle	308.15	10.62	274.09	6.27	0.409	0.762	14.300	0.038	36.036	1.286	0.004	1.442	0.008	157.4	157.3	0.1	134.4
22	Natural Gas Compressor Recycle	308.15	19.72	266.35	10.62	0.462	0.660	10.100	0.027	33.012	1.308	0.002	4.089	0.008	158.4	158.4	0.0	131.2
23	Natural Gas Compressor Recycle	308.15	36.82	263.14	19.79	0.463	0.610	6.400	0.017	32.886	1.341	0.001	10.225	0.008	158.4	158.4	0.0	127.2
24	Natural Gas Compressor Recycle	308.15	64.47	270.22	36.96	0.427	0.508	4.900	0.013	32.823	1.309	0.000	13.026	0.008	157.3	157.1	0.2	124.8
25	Natural Gas Compressor Recycle	308.15	139.48	320.30	64.54	0.537	0.406	4.200	0.012	32.760	0.952	0.000	40.562	0.008	160.7	160.7	0.0	123.7
26	Propane Compressor Recycle	308.15	12.07	234.24	0.97	0.920	0.457	143.500	0.644	50.400	1.122	0.519	0.150	0.010	168.9	168.9	0.0	157.0
27	Steam Desuperheater	658.15	43.09	522.78	10.34	0.760	0.762	22.000	0.041	45.990	1.192	0.011	21.257	0.009	161.2	173.7	-12.5	140.0

ioMosaic

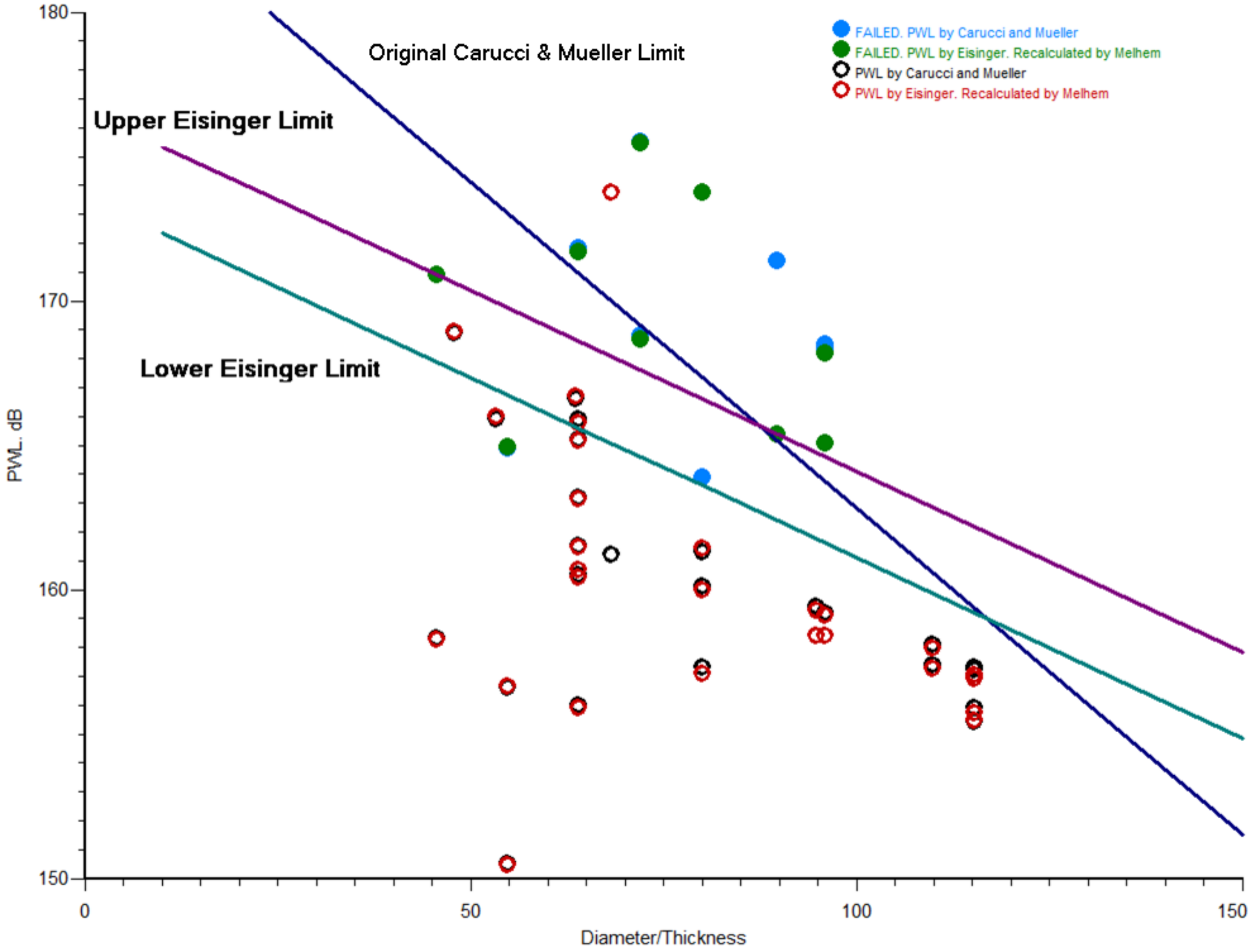
We have analyzed and recreated all the actual data points used by Carucci and Mueller in their original publications to develop their experience based failure limits

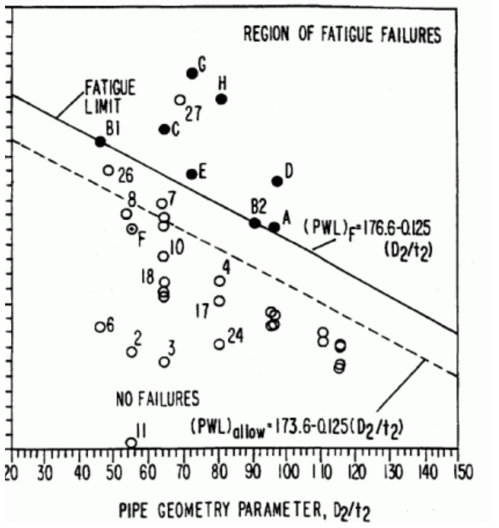




ioMosaic

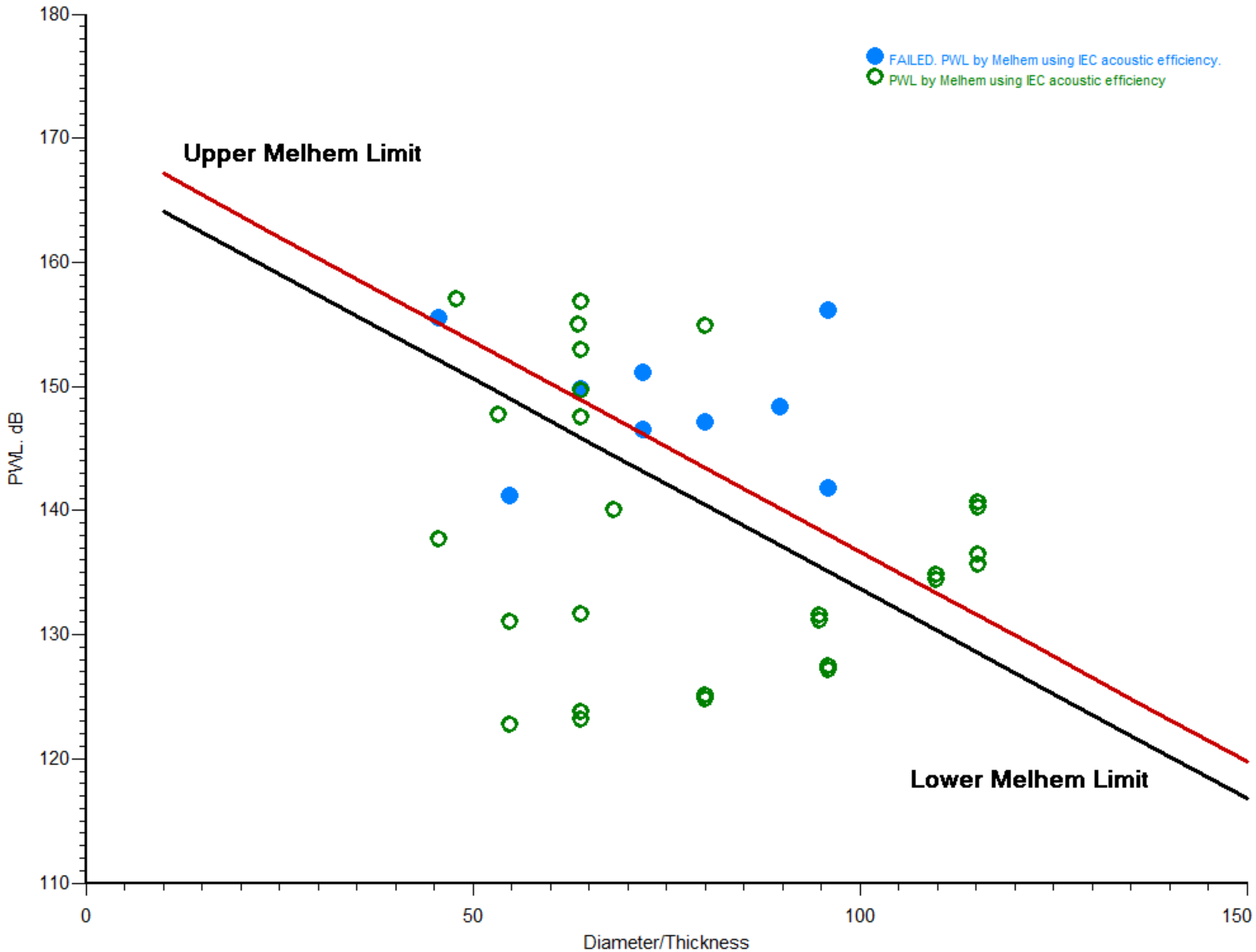
Both Carucci and Mueller as well as Eisinger have proposed failure limits based on the original PWL equation proposed by Carucci and Mueller





ioMosaic

We have recalculate the original data using the IEC efficiency and re-established a similar failure criterion based on more credible estimates of acoustic efficiency



ioMosaic

Since SuperChems calculates detailed piping solutions, it was easy to integrate both the experience based methods (see below) and the MTD methods (see ioVIPER)

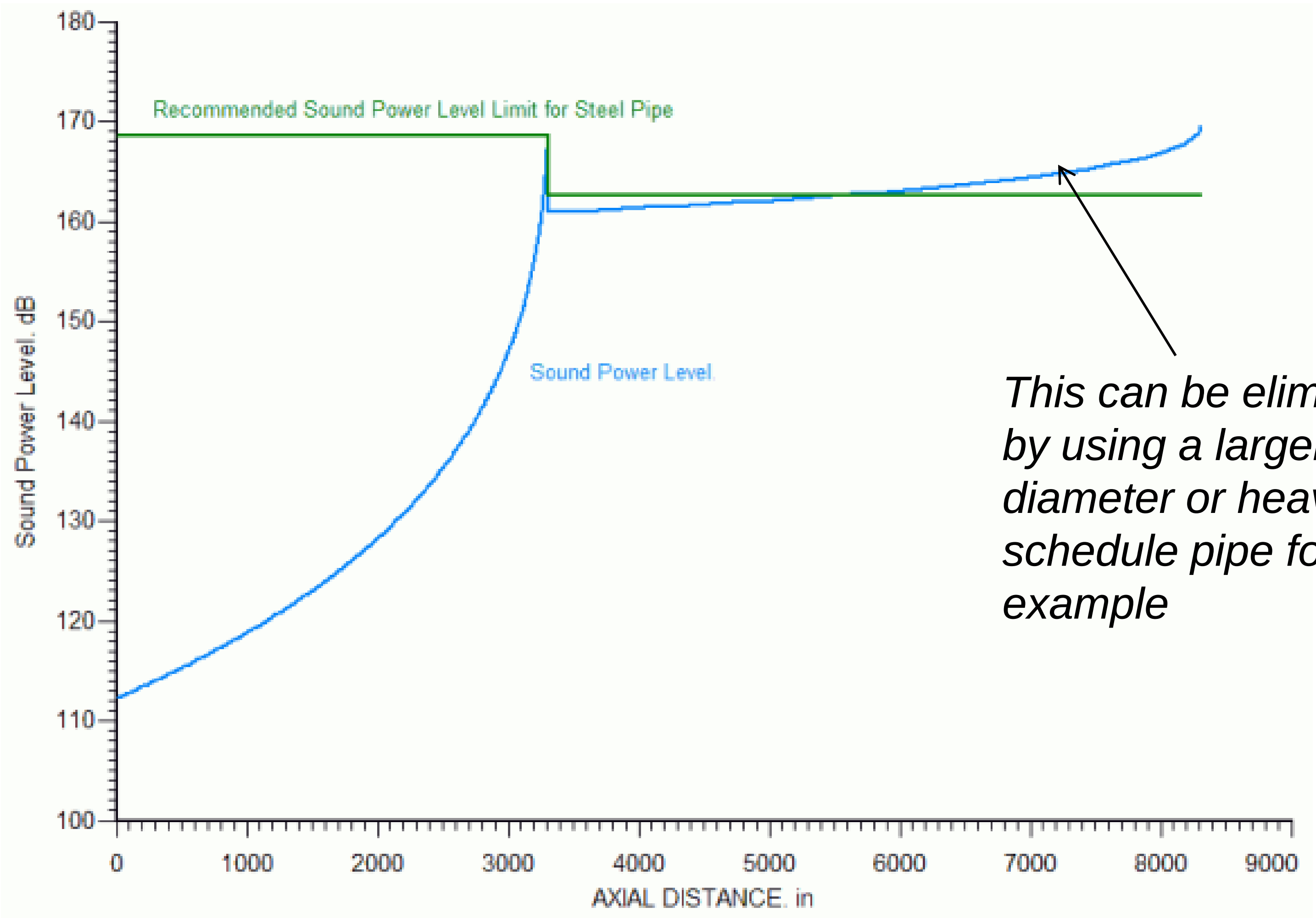
Pipes Containing Vapor. SuperChems Expert.

Gas Pipe Summary | Gas Pipe Axial Profiles | Notes

	A	B	C	D	E	F	G	H
92	Segment #, Type, Name	D/t	Maximum SPL. dB	Limit SPL. dB	SPL Difference. dB	Vibration Risk?		
93	001, Piping Segment, 14H-P3-HC-41X001-01-001	32.29	112.85	156.59	-43.74	Not Likely		
94	002, Piping Segment, 14H-P3-HC-41X001-01-002	32.29	115.79	156.59	-40.79	Not Likely		
95	003, Piping Segment, 14H-P3-HC-41X001-01-003	32.29	115.79	156.59	-40.79	Not Likely		
96	004, Piping Segment, 14H-P3-HC-41X001-02-001	32.29	114.40	156.59	-42.19	Not Likely		
97	005, Expander, 14H-P3-HC-41X001-02-002	34.00	50.30	156.01	-105.70	Not Likely		
98	006, Piping Segment, 14H-P3-HC-41X001-03-001	34.00	41.57	156.01	-114.44	Not Likely		
99	007, Piping Segment, 14H-P3-HC-41X001-03-002	34.00	41.56	156.01	-114.44	Not Likely		
100	008, Piping Segment, 14H-P3-HC-41X001-03-003	27.45	148.09	158.22	-10.13	Not Likely		
101	009, Piping Segment, 14H-P3-HC-41X134-01-001	27.45	148.35	158.22	-9.87	Not Likely		
102	010, Piping Segment, 14H-P3-HC-41X134-01-002	27.45	148.61	158.22	-9.61	Not Likely		
103	011, Piping Segment, 14H-P3-HC-41X134-01-003	27.45	148.84	158.22	-9.39	Not Likely		
104	012, Piping Segment, 14H-P3-HC-41X134-01-004	27.45	149.13	158.22	-9.09	Not Likely		
105	013, Piping Segment, 14H-P3-HC-41X134-01-005	27.45	149.26	158.22	-8.96	Not Likely		
106	014, Piping Segment, 14H-P3-HC-41X134-01-006	27.45	149.27	158.22	-8.95	Not Likely		
107	015, Piping Segment, 14R-P3-HC-41X134-01-001	27.45	150.64	158.22	-7.58	Not Likely		
108	016, Piping Segment, 14R-P3-HC-41X134-01-002	27.45	150.72	158.22	-7.50	Not Likely		
109	017, Piping Segment, 14R-P3-HC-41X134-01-003	27.45	150.79	158.22	-7.43	Not Likely		
110	018, Piping Segment, 14R-P3-HC-41X134-01-004	27.45	150.80	158.22	-7.42	Not Likely		
111	019, Piping Segment, 14D-P3-HC-41X134-01-001	27.45	151.02	158.22	-7.20	Not Likely		
112	020, Piping Segment, 14D-P3-HC-41X134-01-002	27.45	151.03	158.22	-7.19	Not Likely		
113	021, Reducer, 14D-P3-HC-41X134-01-003	27.45	155.36	158.22	-2.86	Not Likely		
114	022, Piping Segment, 14D-P3-HC-41X134-01-004	24.79	155.59	159.12	-3.53	Not Likely		
115	023, Piping Segment, HV-41X016	24.79	160.24	159.12	1.12	Yes		
116	024, Piping Segment, 14D-P3-HC-41X134-01-005	24.79	160.79	159.12	1.67	Yes		
117	025, Piping Segment, 14D-P3-FD-41X136-01-001	56.28	160.24	148.48	11.76	Yes		
118	026, Piping Segment, 14D-P3-FD-41X136-01-002	56.28	160.77	148.48	12.30	Yes		
119	027, Piping Segment, 14D-P3-FD-41X136-01-003	56.28	160.86	148.48	12.38	Yes		
120	028, Piping Segment, 14D-P3-HC-41X135-02-001	56.28	162.53	148.48	14.05	Yes		
121	029, Piping Segment, 14D-P3-HC-41X135-02-002	56.28	162.59	148.48	14.11	Yes		
122	030, Piping Segment, 14D-P3-FD-419068-01-001	56.28	167.93	148.48	19.45	Yes		
123								
124	The IEC acoustic efficiency was used to calculate sound power levels							

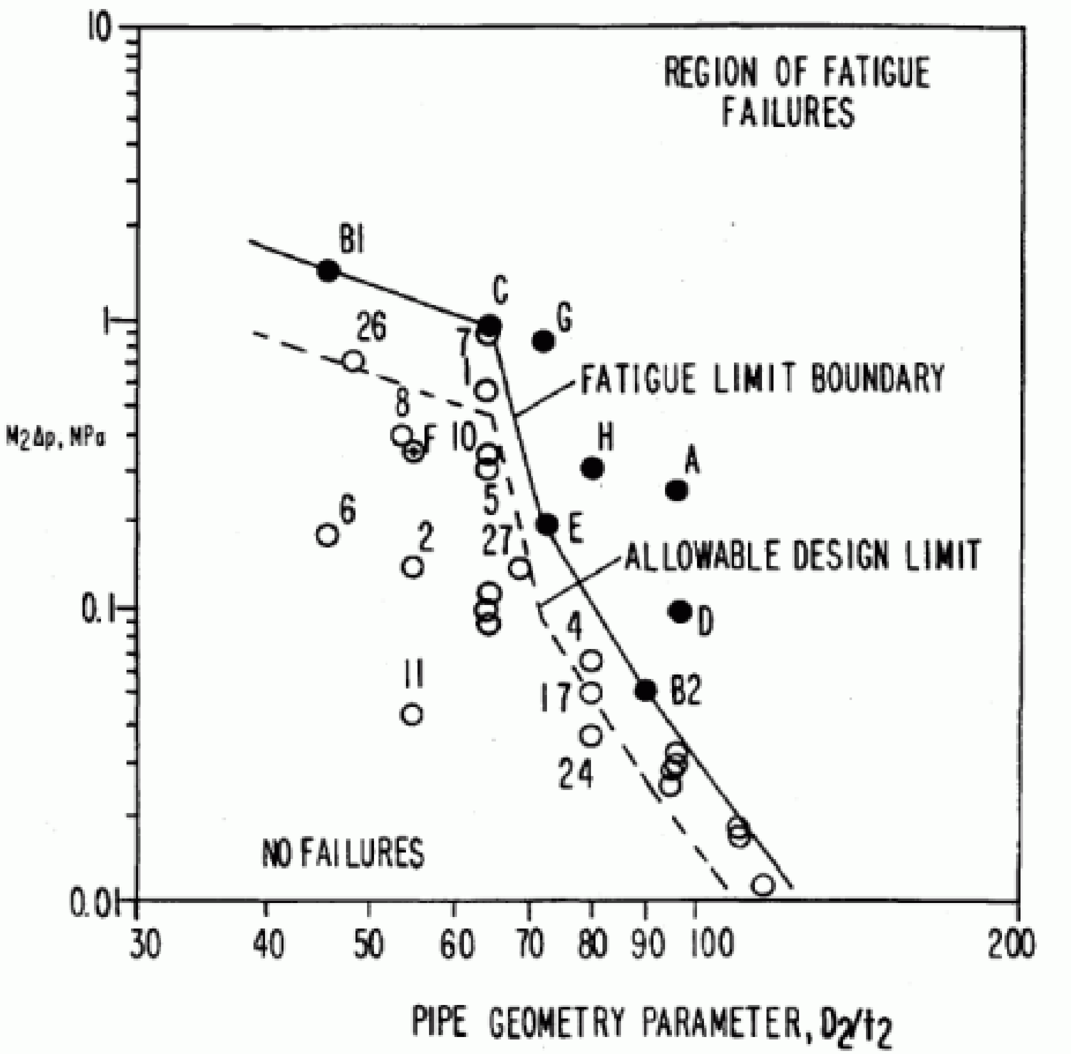
Update Format Print Save As Cut Copy Paste Cancel OK

Since SuperChems calculates detailed piping solutions, it was easy to integrate both the experience based methods (see below) and the MTD methods (see ioVIPER)



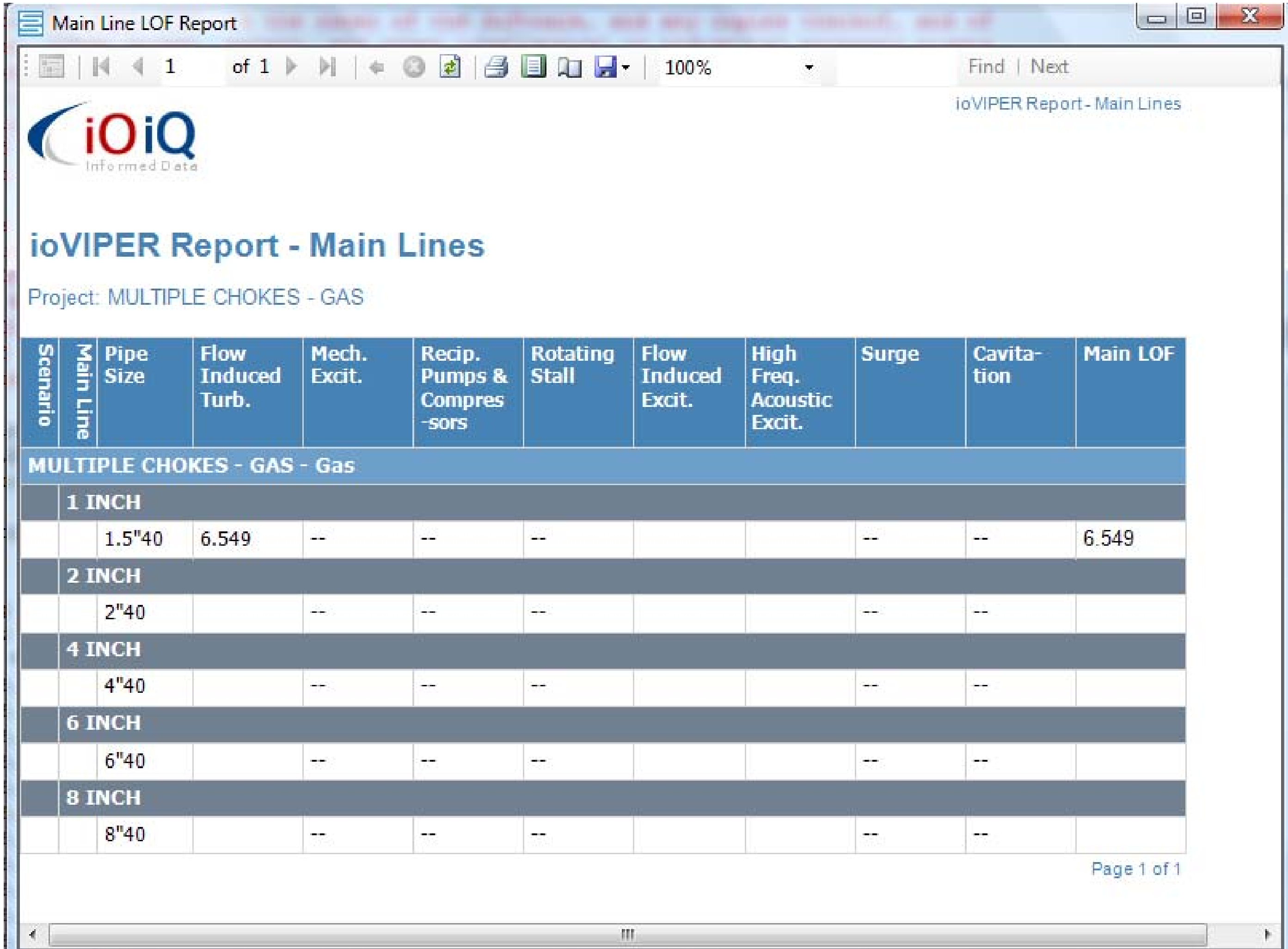
This can be eliminated by using a larger pipe diameter or heavier schedule pipe for example

The Mach number method is not as popular and difficult to apply for multi-phase flow



ioMosaic

SuperChems Expert also implements the methods outlined by the Energy Institute (MTD Guidelines) for both qualitative and quantitative vibration risk


Main Line LOF Report

1 of 1 100% Find | Next

ioVIPER Report - Main Lines

iOiQ Informed Data

ioVIPER Report - Main Lines

Project: MULTIPLE CHOKES - GAS

Scenario	Main Line	Pipe Size	Flow Induced Turb.	Mech. Excit.	Recip. Pumps & Compres-sors	Rotating Stall	Flow Induced Excit.	High Freq. Acoustic Excit.	Surge	Cavita-tion	Main LOF
MULTIPLE CHOKES - GAS - Gas											
	1 INCH										
		1.5"40	6.549	--	--	--			--	--	6.549
	2 INCH										
		2"40		--	--	--			--	--	
	4 INCH										
		4"40		--	--	--			--	--	
	6 INCH										
		6"40		--	--	--			--	--	
	8 INCH										
		8"40		--	--	--			--	--	

Page 1 of 1

ioMosaic

As the fluid moves through piping components, vortices are formed and swept into the main stream

- Vortex shedding can create standing waves
- Vortex shedding frequency (Mach No < 1.4) $f_p = \frac{N_{Str} u}{D} = 0.2 \frac{u}{D}$
- Frequency of standing waves
  - Open End Pipe $f = \frac{i * u_{ac}}{2L}$
  - Closed End Pipe $f = \frac{i * u_{ac}}{4L}$

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1 + \frac{K}{E}\psi}}$$

- If the vortex shedding frequency couples with piping components frequencies, the potential for damage can become substantial
- Note that L is the effective “acoustic length” of the pipe and depends on the presence of acoustic barriers such as valves, pumps, change of flow area, etc.

It is common to install relief devices on column overhead and process lines

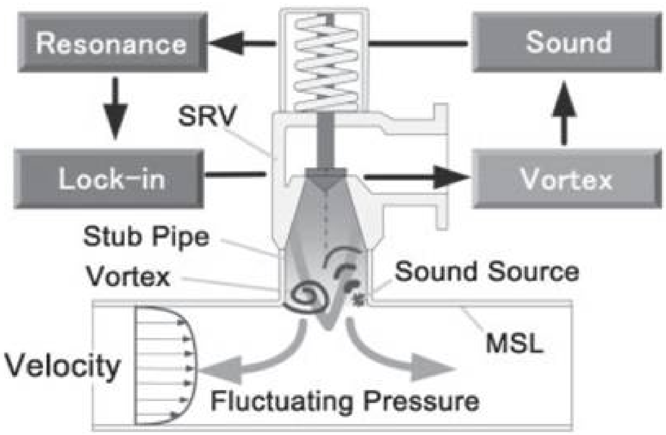
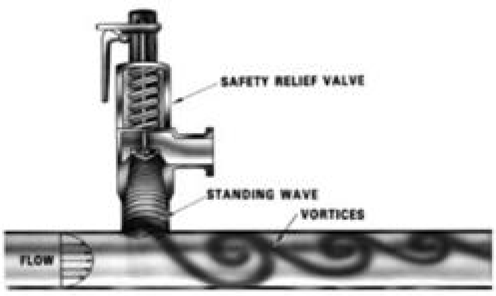
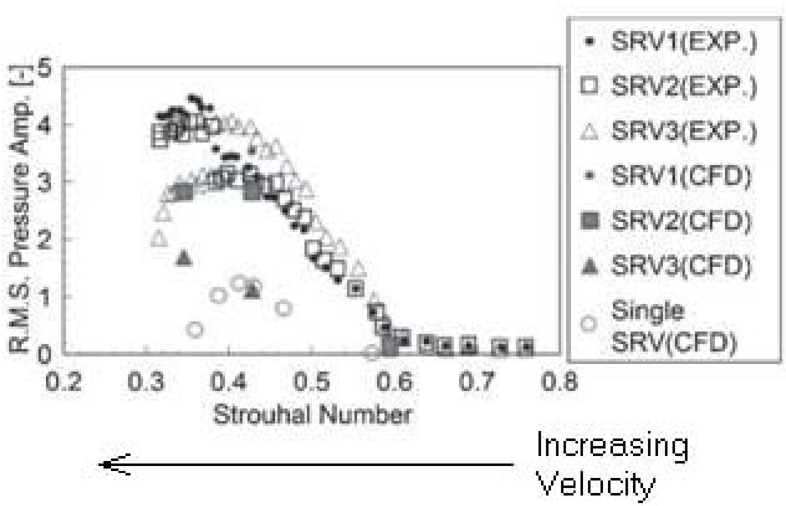


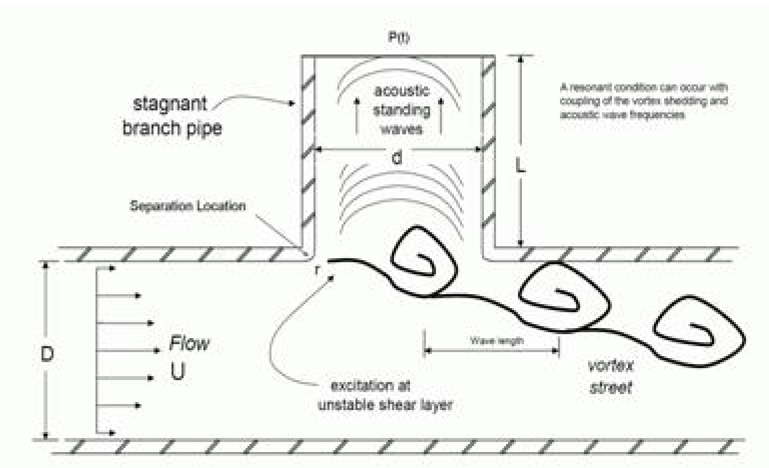
Fig. 4 Flow-induced acoustic resonance of SRVs in MSL

"Singing" Safety Relief Valve



Reference: Hambric, S.A., Mulcahy, T.M., Shah, V.N., et. al., "Flow-Induced Vibration Effects on Nuclear Power Plant Components Due to Main Steam Line Valve Singing," Proceedings of the Ninth NRC/ASME Symposium on Valves, Pumps, and Inservice Testing, NUREG/CP-0152, Vol. 6, pp. 3B 49-3B 69, July 2006





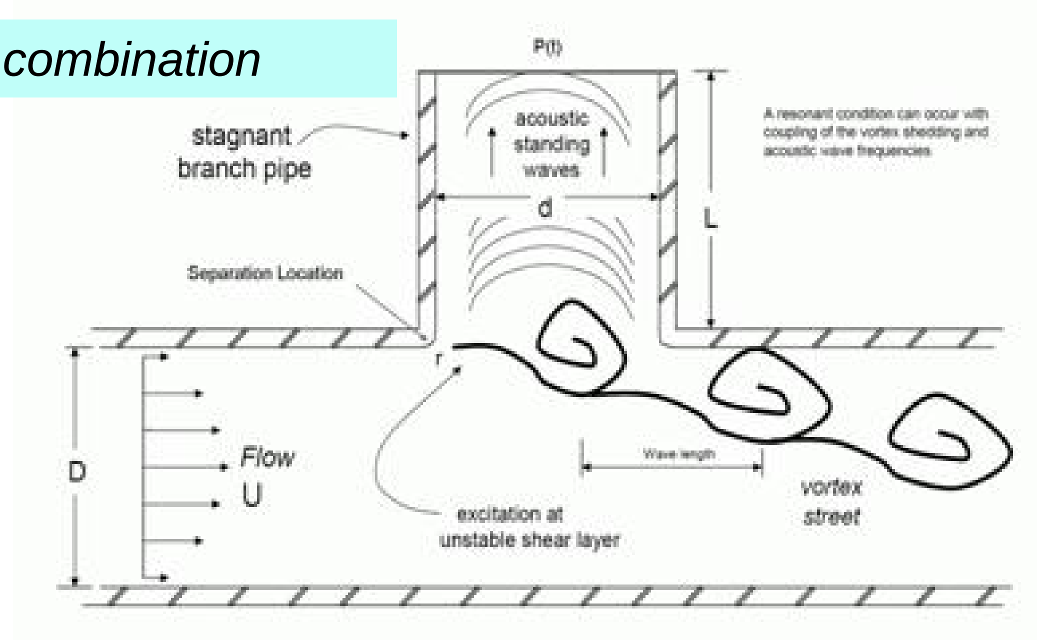
Resonance occurs when the vortex shedding frequency coincides with the acoustic frequency of the standpipe

Natural frequency of standpipe / valve combination

$$f_a = \left(\frac{2n-1}{4}\right)\frac{c_0}{L+L_e}$$

n = 1 for 1st mode, 2 for 3rd mode, etc., c_0 = fluid speed of sound, and r = radius of inlet chamfer

L_e = End correction corresponding to Rayleigh's upper limit = 0.425 d



Frequency of pressure oscillations (sound) created by vortex shedding

Vortex shedding creates pressure oscillations – the energy source for standing waves

$$N_{St} = \frac{f_s}{U}(d+r)$$

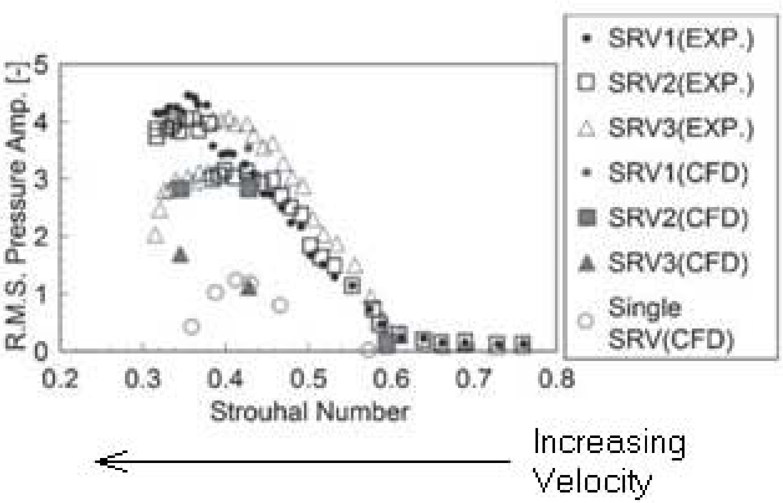
N_{St} = Strouhal Number where 0.63 >= N_{St} >= 0.3

Resonance can cause fatigue failure from cyclic loads and can cause leaking and chatter of the valve

$$f_s = N_{St}\left(U/D\right) \approx 0.33\left(n-0.25\right)\left(U/D\right)$$

For n = 1

$$U = \frac{f_s}{N_{St}}\left(d+r\right) = \frac{1}{4}\left(\frac{c_0}{L+L_e}\right)\left(\frac{d+r}{N_{St}}\right)$$



N_{St} = Strouhal Number where 0.63 $\geq N_{St} \geq$ 0.3

Peak oscillations occur around N_{St} = 0.4

RMS is the ratio of pressure oscillations divided by dynamic pressure = ½ ρu^2

RMS begins increasing at a specific onset Strouhal Number and flow velocity depending on acoustic speed, pipe diameter, and pipe length, reaches a peak value and then decreases

ioMosaic

The stability and adequacy of a relief system requires an assessment of vibration risk

- Use the experience based method as a screening tool with the IEC efficiency
- Use the Energy Institute / MTD guideline for more comprehensive analysis of additional causes of vibration
- Use finite element methods if the above two methods do not provide sufficient guidance to reduce and/or eliminate the risk

ioMosaic

Related Presentations

- G. A. Melhem, “Vibration risk for relief and process piping”, ioMosaic white paper, May 2012
- G. A. Melhem, “Pressure relief valve stability”, Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany
- G. A. Melhem, “Understand Flare Noise”, DIERS Users Group Presentation, D38-150-1, 2006

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

Business Confidential Document

Estimate Acoustically Induced Vibration Risk in Relief and Flare Piping

Joint US and European DIERS Users Group Meeting

June 2011

Hamburg, Germany

by

G. A. Melhem, Ph.D.
melhem@iomosaic.com

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

Failure of relief and process piping caused by vibration can develop due to flow energy

- Flow induced turbulence
  - Caused by flow discontinuities in piping, valves, tees, etc.
- High frequency excitations (acoustic energy)
  - Caused by choked flow from a relief device or control valve
- Other causes include
  - Mechanical excitations by compressors, pumps, and other rotating machinery
  - Vortex shedding
  - Water hammer
  - Cavitation
  - Etc.

Factors that have led to an increasing incidence of vibration related fatigue failures in piping systems include but are not limited to:

- Increasing flow rates as a result of debottlenecking which contributes to higher flow velocities with a correspondingly greater level of turbulent energy
- Frequent use of thin-walled piping which results in higher stress concentrations, particularly at small bore connections
- Design of process piping systems on the basis of a static analysis with little attention paid to vibration induced fatigue
- Lack of emphasis of the issue of vibration in piping design codes
- Piping vibration is often considered on an ad-hoc or reactive basis

ioMosaic

According to the UK HSE, 21 % of all piping failures offshore are caused by fatigue/vibration

- Large compressor recycle systems
- Steam de-superheater systems
- High capacity pressure relief depressuring systems
- Other?

ioMosaic

The current ISO/API 521 standard does not formally address vibration risk

- It does not offer specific guidance on velocity limitations other than backpressure calculations
- It does not offer guidance on acoustic fatigue or vibration induced fatigue
- Many operating companies have established their own internal guidance for evaluating and minimizing piping vibration risk
- Although these criteria vary from company to company, they all in general include a limit of flow velocity in some form

Typical criteria used by companies and other organizations to minimize vibration risk

- Limit Mach Number $$0.3 \leq M = \frac{u}{u_{sonic}} \leq 0.8$$

- Limit dynamic pressure for gas flow $$\rho u^2 \leq 1 \times 10^5 \ \ \mathrm{kg/m/s^2}$$

- Limit dynamic pressure for two-phase flow $$\rho u^2 \leq \frac{1}{2} \times 10^5 \ \ \mathrm{kg/m/s^2}$$

- Some companies use different limits for process piping and relief piping (See NORSOK Process Design Standard P-001, 2006, Fifth Edition)

ioMosaic

ASME and IEC have published guidance for the estimation of noise levels for control valves and pressure reducing devices

- This guidance is not readily applicable to piping
- This guidance is not readily applicable to multi-phase flow

$$\begin{aligned} L_W &= 10 \times \log_{10}\left[\left(\frac{P_1 - P_2}{P_1}\right)^{3.6} \times \dot{m}^2 \times \left(\frac{T_1}{M_w}\right)^{1.2}\right] + 126.1 \\ &= 36\log_{10}\left(\frac{P_1 - P_2}{P_1}\right) + 20\log_{10}(\dot{m}) + 12\log_{10}\left(\frac{T_1}{M_w}\right) + 126.1 \end{aligned}$$

- Noise attenuation (3 dB for every 50 L/D)

$$L_{W,At} = 0.06\frac{L}{D_i}$$

- Add 6 dB to the sound power level estimate when sonice flow exists at a branch connection to account for amplified dynamic response

Sound power level and sound pressure level are all about flow energy

- A small portion of the flow energy is transferred to the pipe wall as vibration energy and a portion of that is radiated as noise

- We can calculate both sound power and sound pressure level in a more fundamental way that is appropriate for complex piping and multi-phase flow

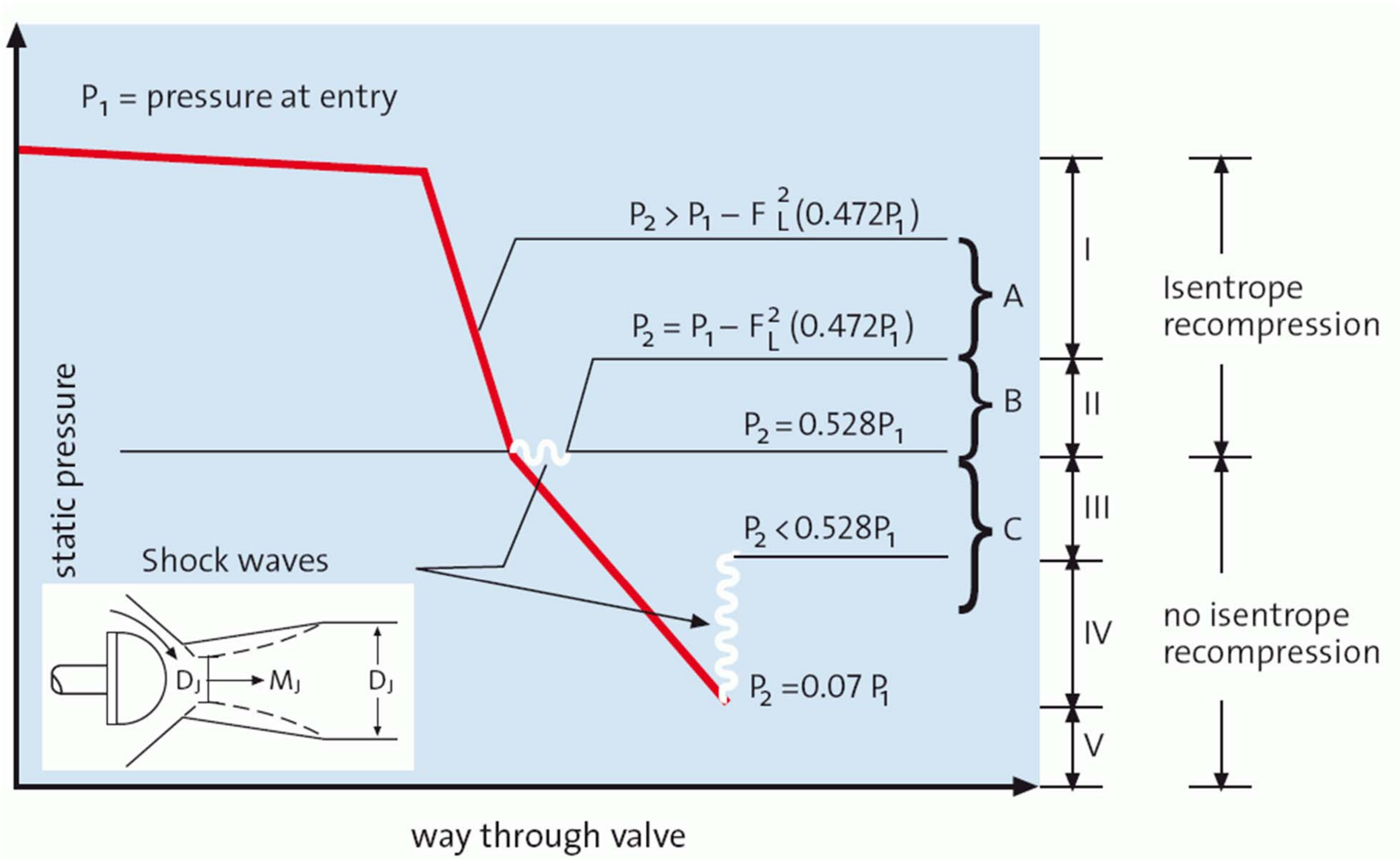
$$
\begin{aligned}
W &= \eta \frac{1}{2} \dot{m} u^2 \\
L_W &= 10 \log_{10} \left[\frac{W}{10^{-12}} \right] = 10 \log_{10} W + 120 = 10 \log_{10} \left[\eta \frac{1}{2} \dot{m} u^2 \right] + 120
\end{aligned}
$$

- For choked flow

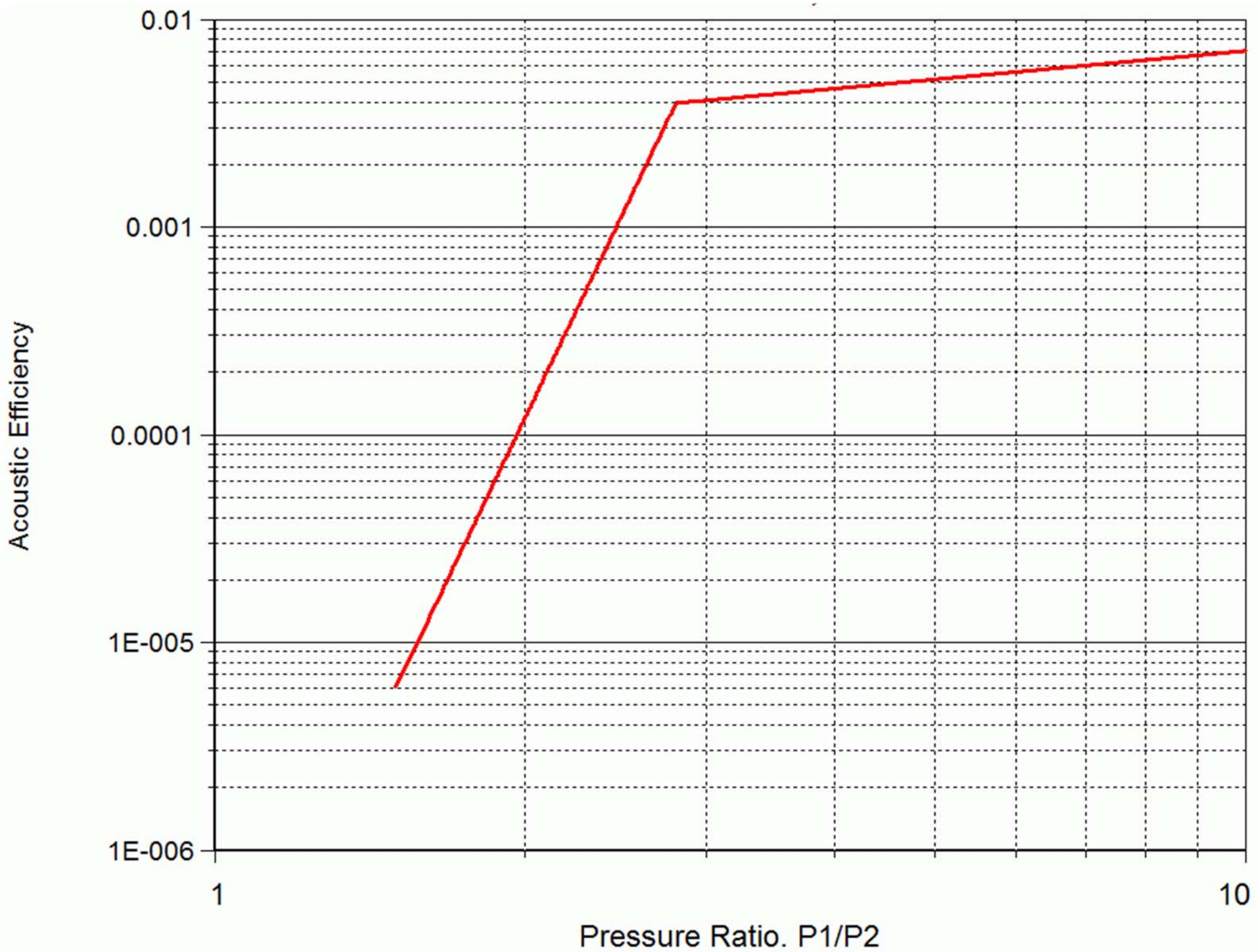
$$
W = \eta \frac{1}{2} \dot{m} u_{sonic}^2
$$

ioMosaic

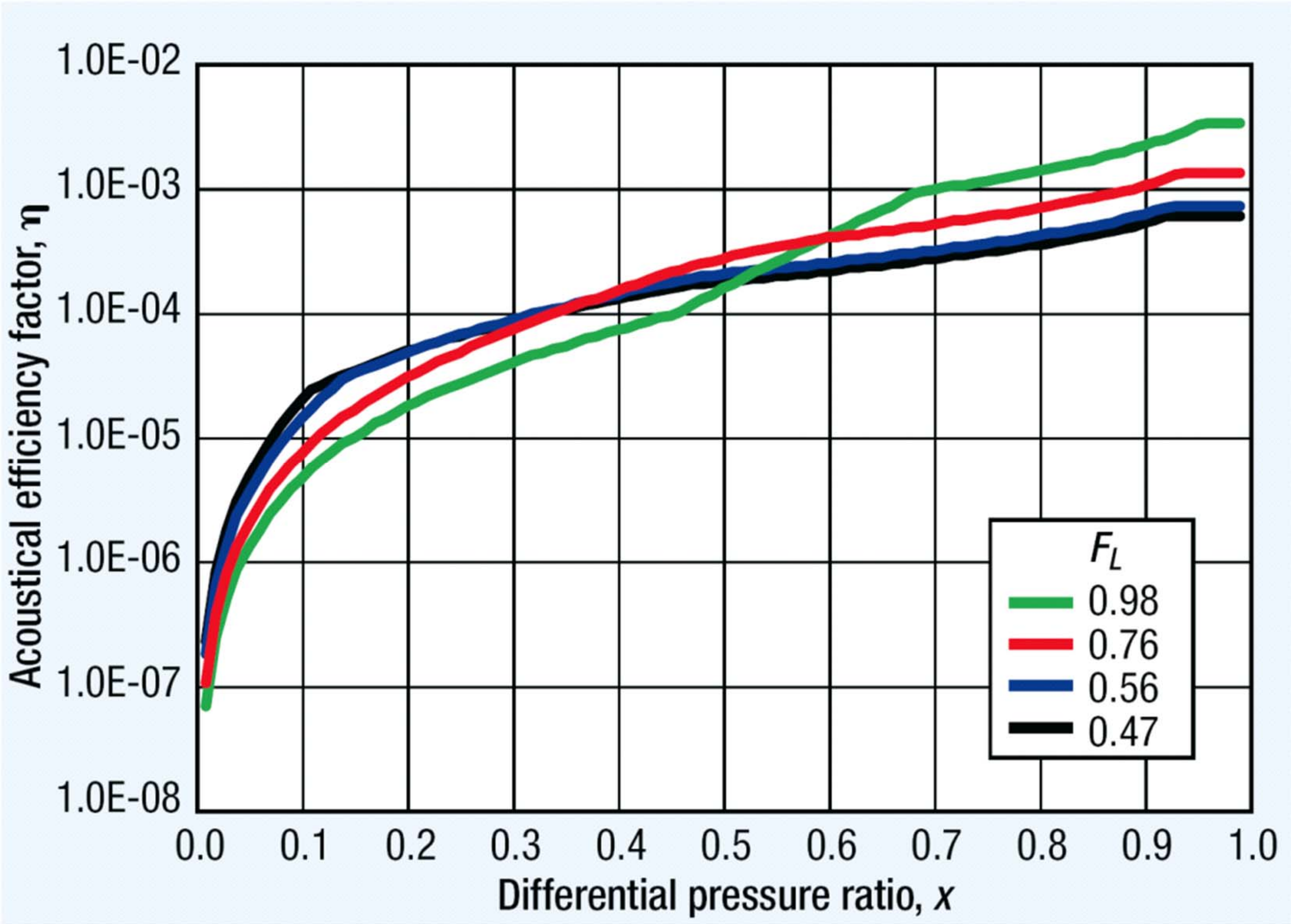
How do we calculate the "acoustic efficiency factor", η



The efficiency represented below is the same used by API for calculating flare noise. Note $P_1/P_2 = 1 / (1-x)$



IEC provides similar efficiency values to the curve used by API. Note $x = 1 - P_2/P_1$



As the fluid moves through piping components, vortices are formed and swept into the main stream

- Vortex shedding can create standing waves
- Vortex shedding frequency (Mach No < 1.4) $f_p = \frac{N_{Str} u}{D} = 0.2 \frac{u}{D}$
- Frequency of standing waves
  - Open End Pipe $f = \frac{i * u_{ac}}{2L}$
  - Closed End Pipe $f = \frac{i * u_{ac}}{4L}$

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1 + \frac{K}{E}\psi}}$$

- If the vortex shedding frequency couples with piping components frequencies, the potential for damage can become substantial
- Note that L is the effective "acoustic length" of the pipe and depends on the presence of acoustic barriers such as valves, pumps, change of flow area, etc.

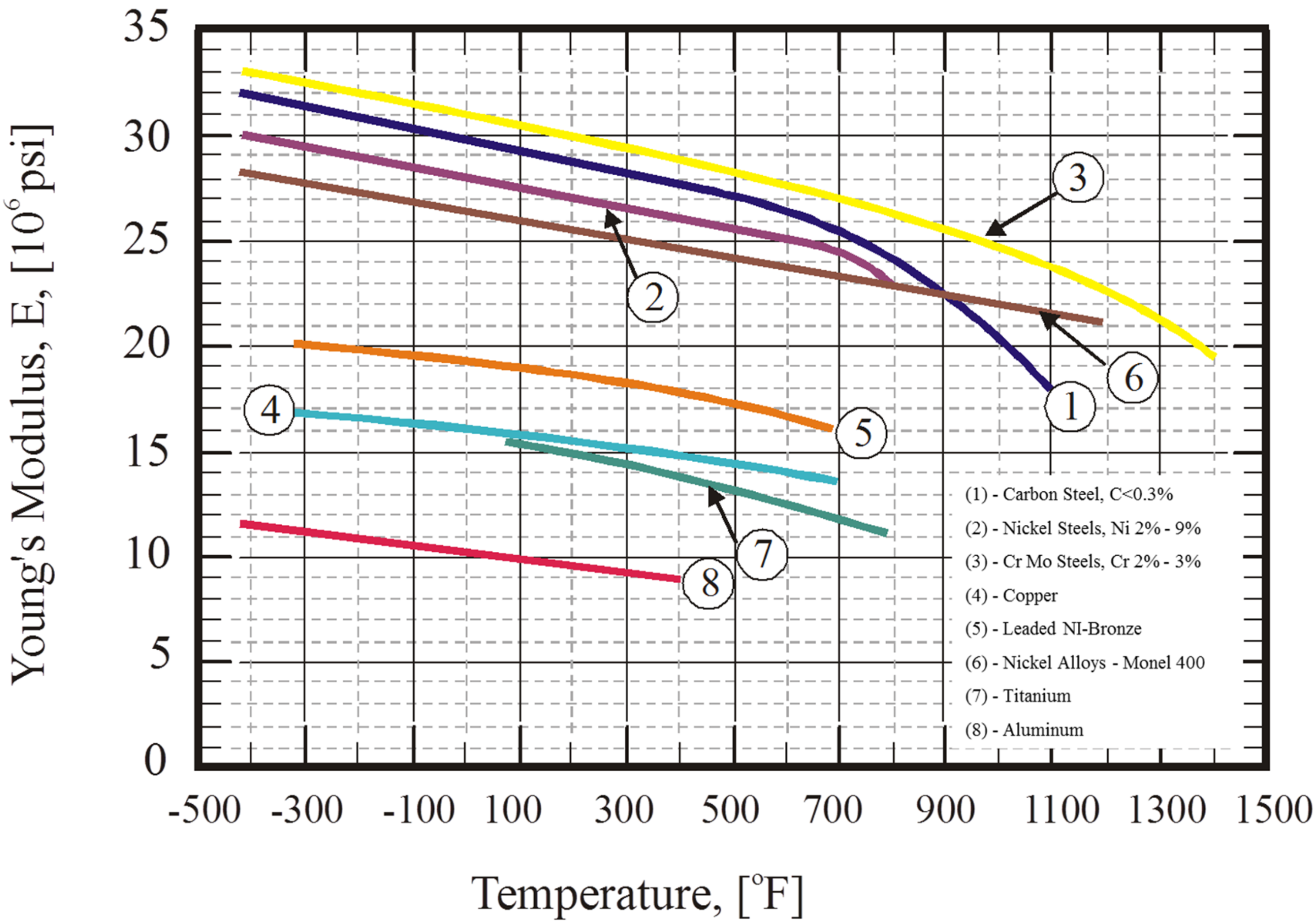
ioMosaic

How do we calculate u_{ac}?

Material	E (GPa)	Poisson's Ratio ν	$K = \frac{1}{\kappa}$ (GPa)	ρ in kg/m3
Aluminum	69	0.33		
Brass	78-110	0.36		
Carbon steel	202	0.303		
Cast iron	90-160	0.25		
Concrete	20-30	0.15		
Copper	117	0.36		
Ductile iron	172	0.30		
Fibre cement	24	0.17		
High carbon steel	210	0.295		
Inconel	214	0.29		
Mild steel	200-212	0.27		
Nickel steel	213	0.31		
Plastic / Perspex	6.0	0.33		
Plastic / Polyethylene	0.8	0.46		
Plastic / PVC rigid	2.4-2.75			
Stainless steel 18-8	201	0.30		
Water - fresh			2.19	999 at 20 C
Water - sea			2.27	1025 at 15 C

ioMosaic

How do we calculate u_{ac}?



ioMosaic

How do we calculate u_{ac}?

Pipe condition	ψ
Rigid	0
Anchored against longitudinal movement through its length	$\frac{d}{\delta}\left(1-\nu^2\right)$
Anchored against longitudinal movement at the upper end	$\frac{d}{\delta}\left(1.25-\nu\right)$
Frequent expansion joints present	$\frac{d}{\delta}$

$$u_{ac} = \frac{u_{sonic}}{\sqrt{1+\frac{K}{E}\psi}}$$

ioMosaic

How do we calculate the internal pipe noise, sound pressure level, from sound power level – as a function of axial distance?

- We can relate sound pressure level to sound power level for reflection free planar waves inside the pipe

$$W = \frac{P^2 A_i}{\rho u_{sonic}} = \frac{\pi D_i^2 P^2}{4\rho u_{sonic}}$$

$$L_{P,i} = 10\log_{10}\left[\frac{\rho u_{sonic} W}{\pi D_i^2 10^{-10}}\right] = 10\log_{10}\left[3.183\times 10^9 \frac{\rho u_{sonic} W}{D_i^2}\right]$$

- A frequency dependent internal sound pressure can then be calculated

$$L_{P,i}(f_i) = L_{P,i} + L_{f,i} = L_{P,i} - c - 10\log_{10}\left[\left(1+\left[\frac{f_i}{2f_p}\right]^2\right)\left(1+\left[\frac{f_p}{2f_i}\right]^4\right)\right]$$

ioMosaic

Weighting factors for one third octave frequencies are used to calculate the frequency dependent sound pressure level inside the pipe

f_i in Hz	$W(f_i)$	f_i in Hz	$W(f_i)$
10	-70.4	500	-3.2
12.5	-63.4	630	-1.9
16	-56.7	800	-0.8
20	-50.5	1000	0
25	-44.7	1250	0.6
31.5	-39.4	1600	1
40	-34.6	2000	1.2
50	-30.2	2500	1.3
63	-26.2	3150	1.2
80	-22.5	4000	1
100	-19.1	5000	0.5
125	-16.1	6300	-1.0
160	-13.4	8000	-1.1
200	-10.9	10000	-2.5
250	-8.6	12500	-4.3
315	-6.6	16000	-6.6
400	-4.8	20000	-9.3

$$0 = 10\log_{10} \Sigma_{i=1}^{n=33} 10^{\frac{L_{f,i}}{10}}$$

ioMosaic

External pipe noise can now be calculated accounting for transmission losses through the pipe wall

- Frequency dependent sound pressure level

$$L_{P,e}(f_i) = L_{P,i}(f_i) + TL(f_i) + W(f_i) - 10\log_{10}\left[\frac{2x + 2\delta + D_i}{D_i + 2\delta}\right]$$

- Total sound pressure level

$$L_{P,e} = 10\log_{10}\sum_{i=1}^{n=33} 10^{\frac{L_{P,e}(f_i)}{10}}$$

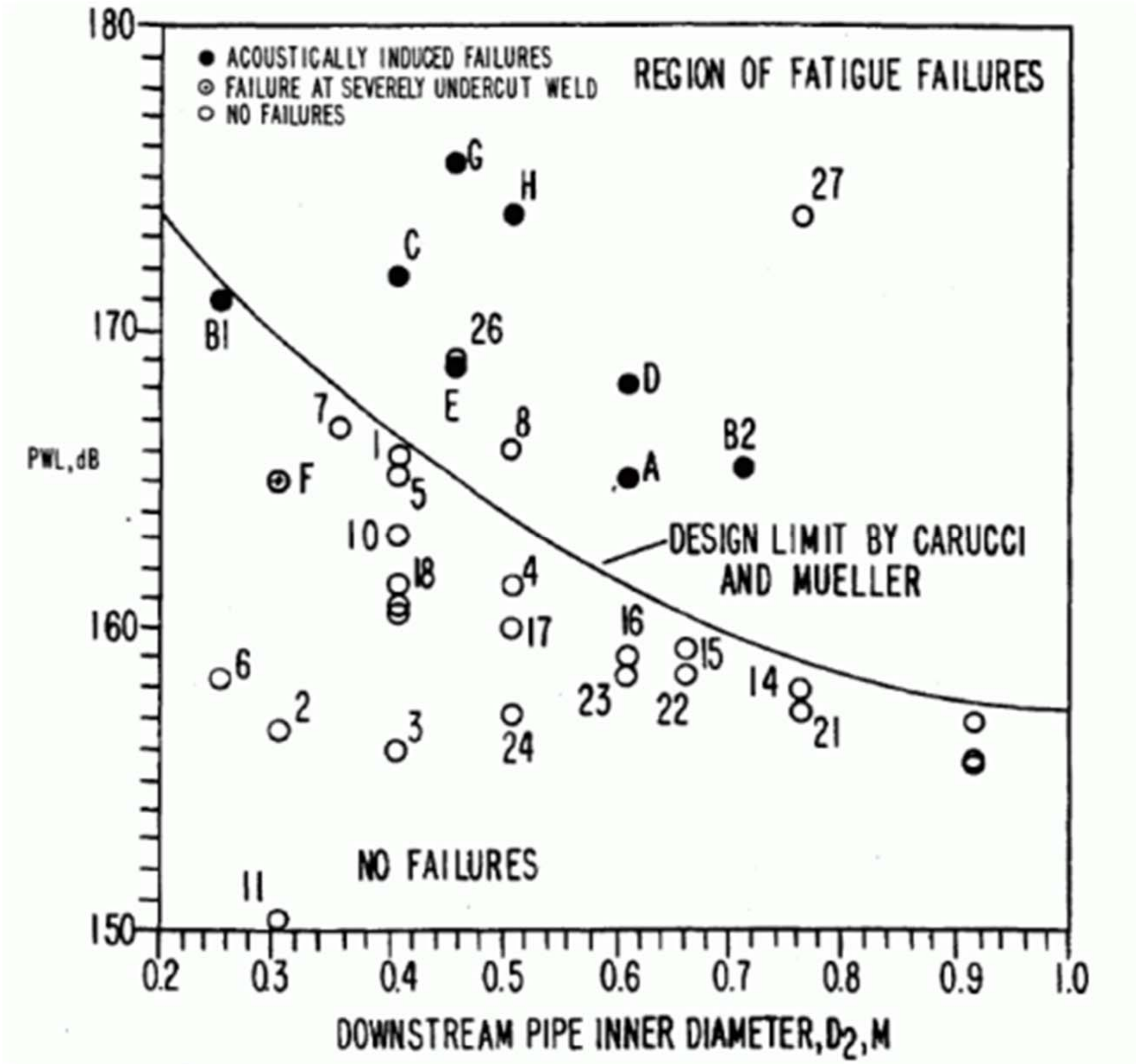
ioMosaic

Now that we know how to calculate sound power and sound pressure levels in piping, how do we determine if the piping is going to fail or is likely to fail due to vibration risk?

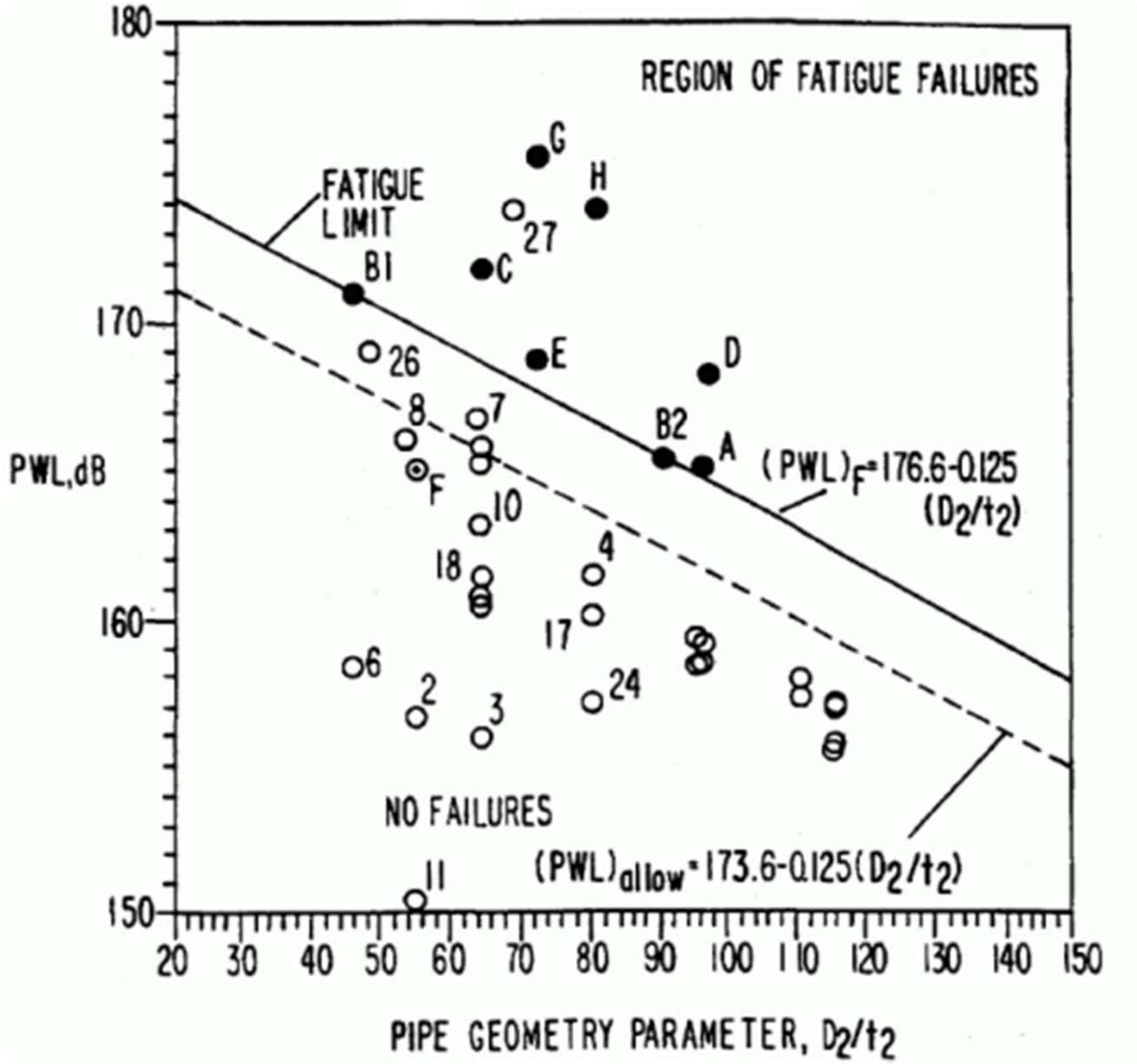
- Experience based – D/t method (SuperChems for DIERS and Expert)
- Experience based – Mach Number method
- MTD / Energy Institute Guidelines (SuperChems Expert ioVIPER)
- Detailed structural dynamics

ioMosaic

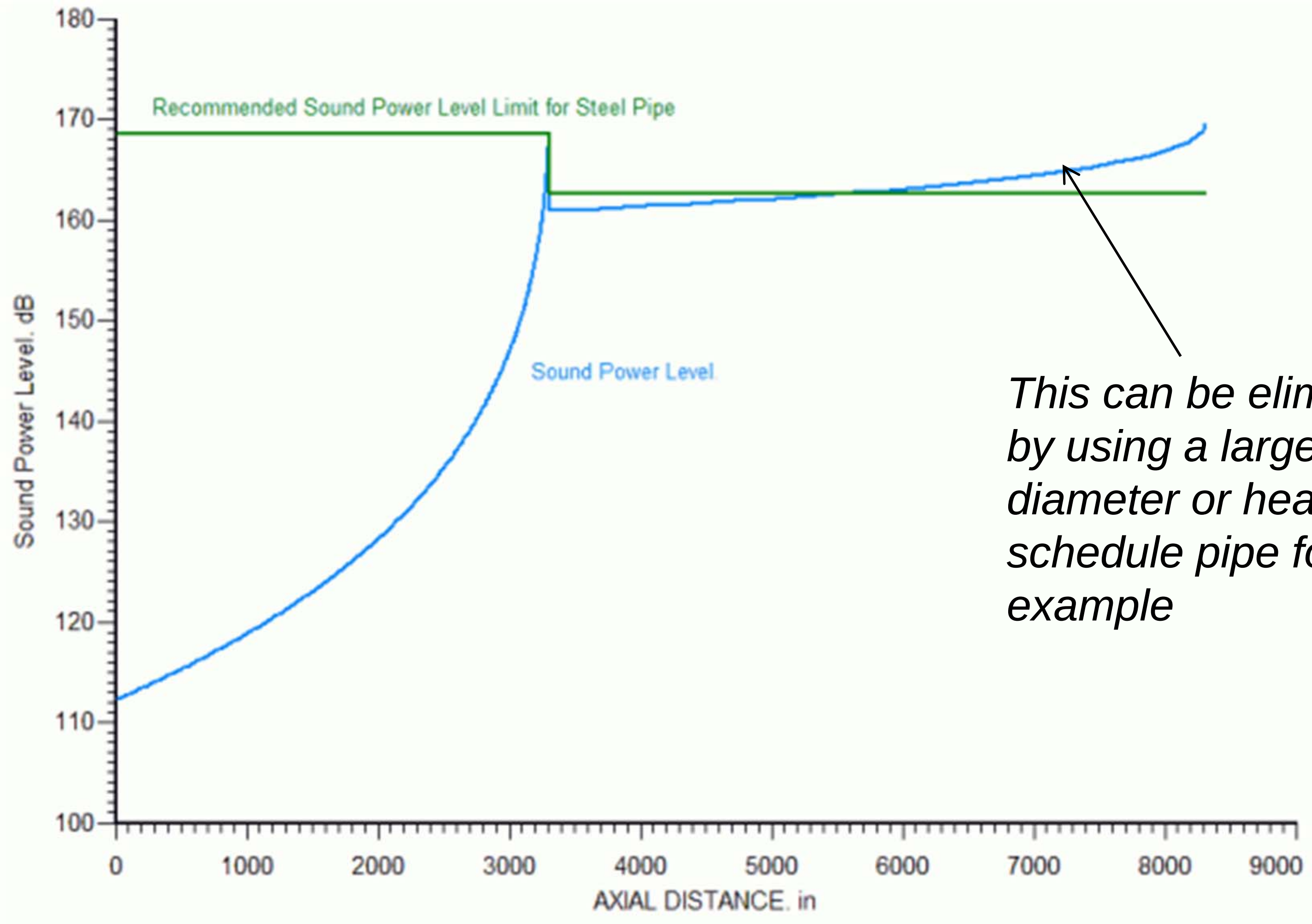
The experience based method is based on actual failure data with carbon steel pipe



Several companies figured out how to recast this data as a function of D/t



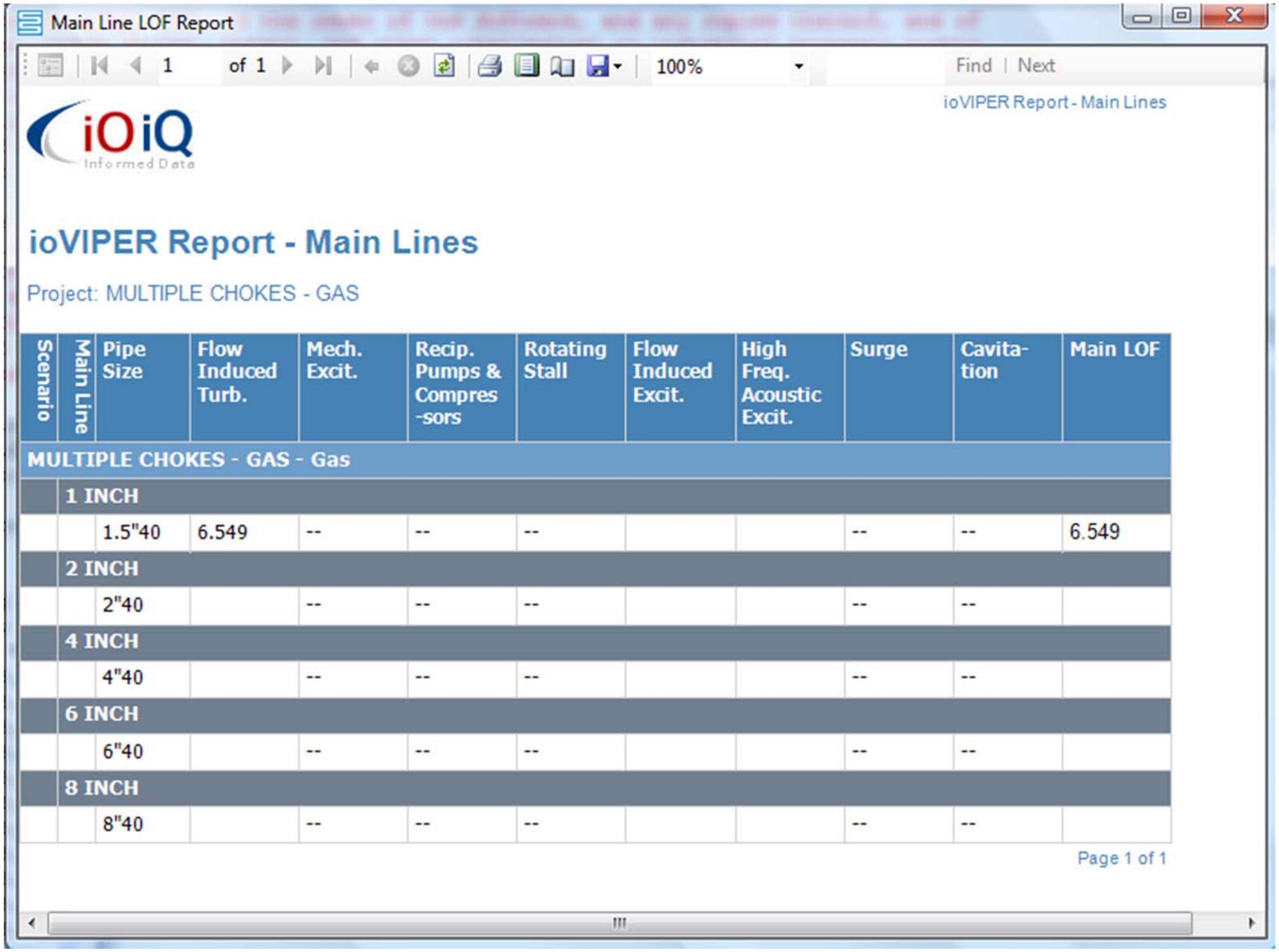
Since SuperChems calculates detailed piping solutions, it was easy to integrate both the experience based methods (see below) and the MTD methods (see ioVIPER)



This can be eliminated by using a larger pipe diameter or heavier schedule pipe for example

ioMosaic

SuperChems Expert also implements the methods outlined by the Energy Institute (MTD Guidelines) for both qualitative and quantitative vibration risk



Main Line LOF Report

ioVIPER Report - Main Lines

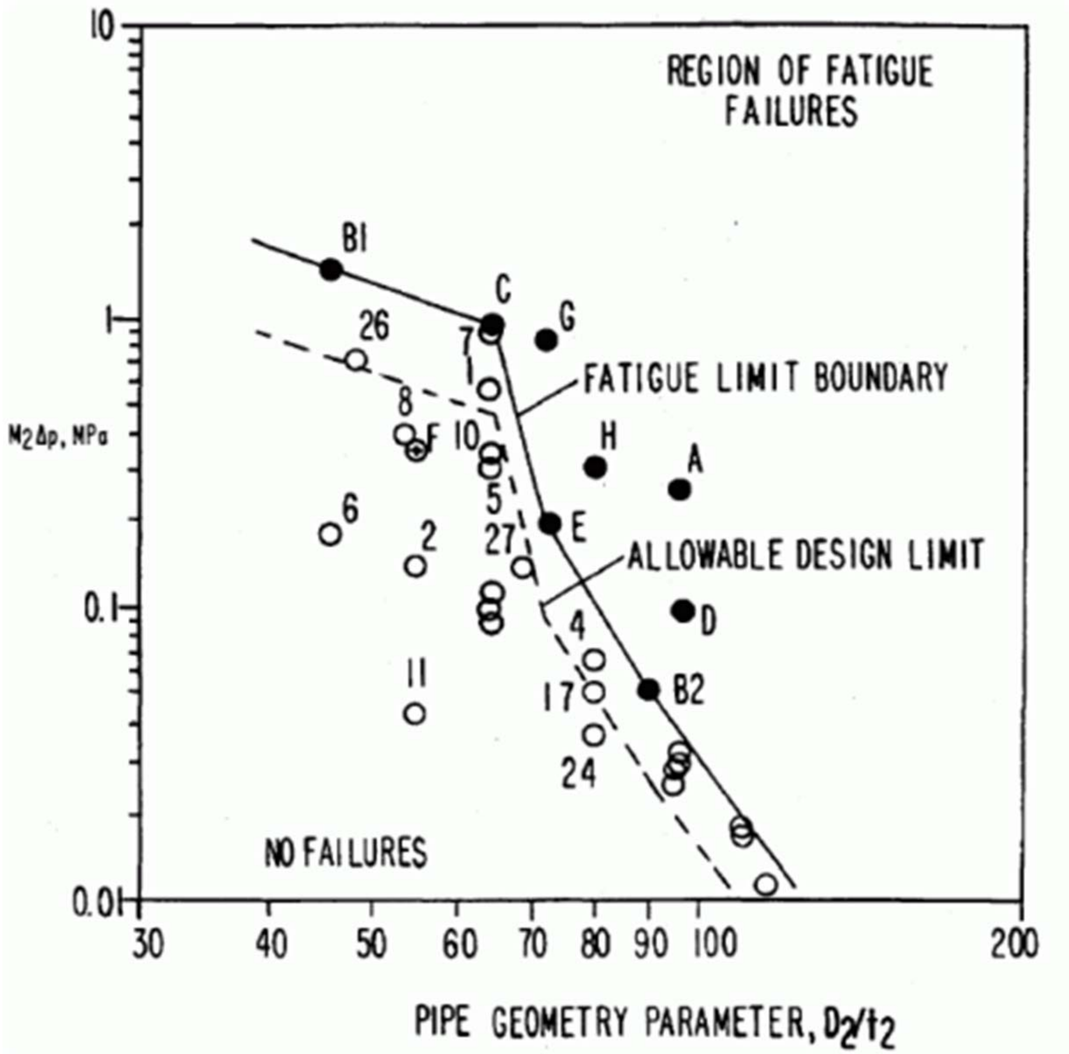
ioVIPER Report - Main Lines

Project: MULTIPLE CHOKES - GAS

Scenario	Main Line	Pipe Size	Flow Induced Turb.	Mech. Excit.	Recip. Pumps & Compres-sors	Rotating Stall	Flow Induced Excit.	High Freq. Acoustic Excit.	Surge	Cavita-tion	Main LOF
MULTIPLE CHOKES - GAS - Gas											
	1 INCH										
		1.5"40	6.549	--	--	--			--	--	6.549
	2 INCH										
		2"40		--	--	--			--	--	
	4 INCH										
		4"40		--	--	--			--	--	
	6 INCH										
		6"40		--	--	--			--	--	
	8 INCH										
		8"40		--	--	--			--	--	

Page 1 of 1

The Mach number method is not as popular and difficult to apply for multi-phase flow



ioMosaic

The stability and adequacy of a relief system requires an assessment of vibration risk

- Use the experience based method as a screening tool
- Use the Energy Institute / MTD guideline for more comprehensive analysis of additional causes of vibration
- Use finite element methods if the above two methods do not provide sufficient guidance to reduce and/or eliminate the risk

Related Presentations

- G. A. Melhem, “Vibration risk for relief and process piping”, ioMosaic white paper, June 2011
- G. A. Melhem, “Understand Flare Noise”, DIERS Users Group Presentation, D38-150-1, 2006
- G. A. Melhem, “Pressure relief valve stability”, Joint European/US DIERS Users Group Meeting, June 2011, Hamburg, Germany

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

An Overview of ioVIPER – a SuperChems Component for the Assessment of Piping Vibration Risk

Paper Presented at the Spring DIERS Users Group Meeting, April 2010

By

G. A. Melhem, Ph.D.

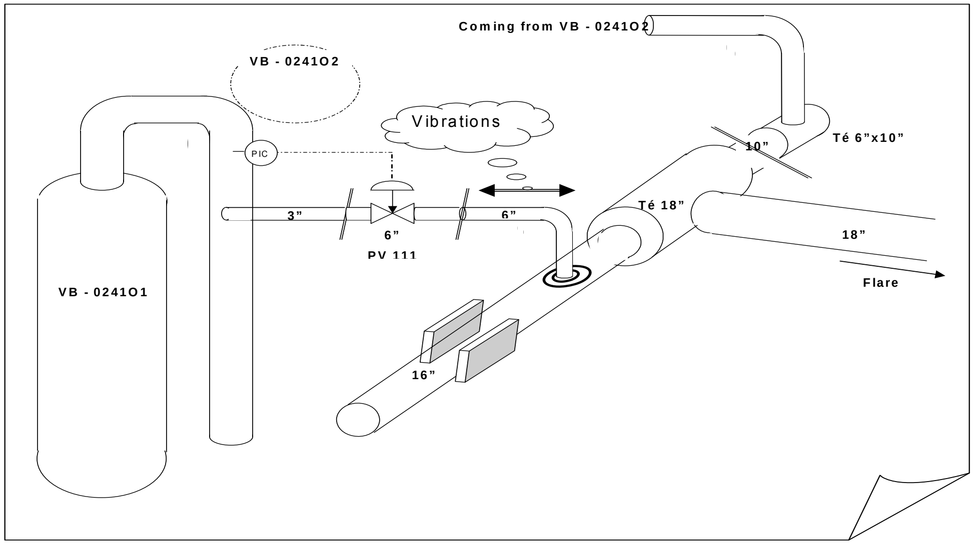
SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

ioMosaic

It is believed that flow induced turbulence caused the failure of this line at header tie-in (Salah Gas Incident)



Source: Ed Zamejc, Spring 2006 API Refining Meeting

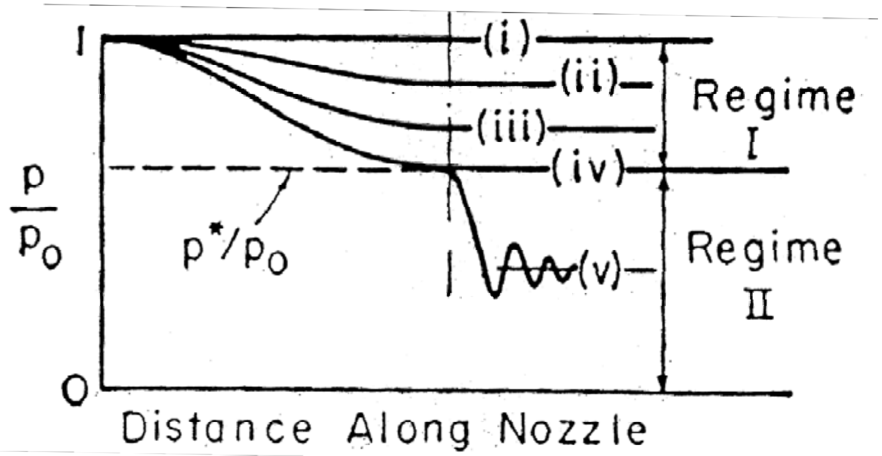
According to the UK HSE, 21 % of all piping failures offshore are caused by fatigue/vibration

- Large compressor recycle systems
- Steam de-superheater systems
- High capacity pressure relief depressuring systems

Source: HSE (2002), “Report on the Hydrocarbon Release Incident Investigation Project 1/4/2001 to 31/3/2002

Depressuring systems are often subjected to acoustic energy (rapidly fluctuating pressure forces) generated by flow turbulence which is accentuated by flow restricting devices within the flow path

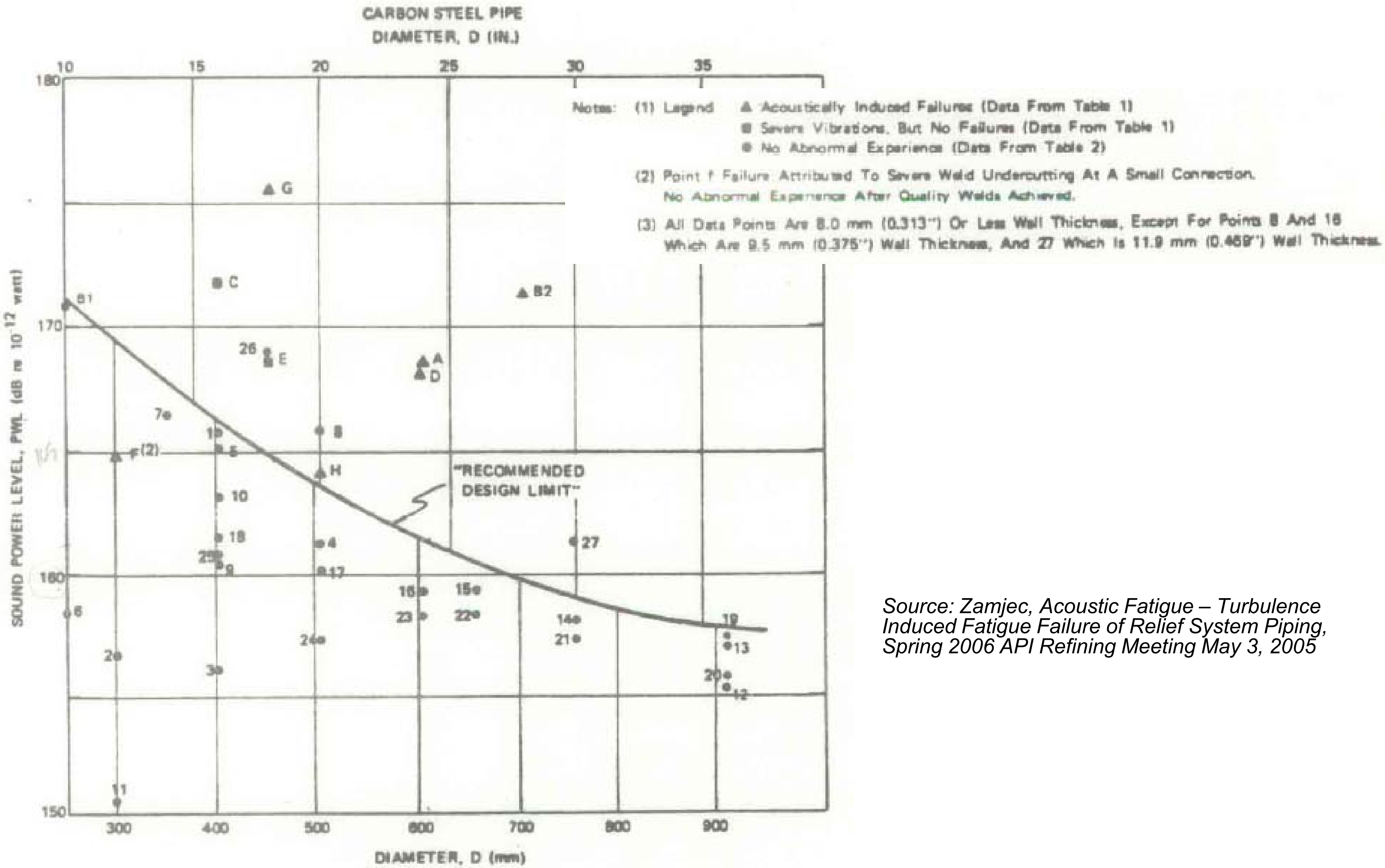
- The magnitude depends on the mass flow rate, speed of sound, and density
- For choked flow, intense noise due to large pressure fluctuations is generated
- The generated noise is non-periodic due to the randomness of the pressure fluctuations
- Choked flow typically leads to a wide frequency spectrum which with peak values than can exceed 1000 Hz
- Vibrations associated with small fittings are of special concern because they introduce discontinuities and stress concentration points



- In many situations resonance can onset which can lead to magnification of static piping loads up to a factor of 50 times!
- The presence of discontinuities (Tees, welded pipe supports, etc.) can further increase these loads

ioMosaic

There are references in the literature for experience based safe design limits



Source: Zamjec, Acoustic Fatigue – Turbulence Induced Fatigue Failure of Relief System Piping, Spring 2006 API Refining Meeting May 3, 2005

The vibration induced fatigue chain is well developed and quantified in the most recent version of the MTD Guidelines

Excitation → **Vibration of main line** → **Vibration of small bore connections**

Excitation:
- flow turbulence
- mechanical
- pulsation
- acoustic
- transient

Vibration of main line:
- pipe diameter
- wall thickness
- support

Vibration of small bore connections:
- length of connection
- type of connection
- no. of valves
- diameter
- parent pipe wall thickness
- location

Source: R. Swindell, "Vibration Fatigue in Process Pipework – A Risk Based Assessment Methodology", 2003?

ioMosaic

Currently, API does not offer specific guidance on piping vibration risk

- Factors that have led to an increasing incidence of vibration related fatigue failures in piping systems:
  - Increasing flow rates as a result of debottlenecking contributes to higher flow velocities with a correspondingly greater level of turbulent energy
  - Frequent use of thin-walled piping which results in higher stress concentrations, particularly at small bore connections
- Process piping systems are often designed on the basis of a static analysis with little attention paid to vibration induced fatigue
- Most piping design codes do not emphasize the issue of vibration proactively
- Take our most commonly used standard, ISO/API 521:
  - No specific guidance on velocity limitations other than backpressure calculations
  - No guidance on acoustic fatigue or vibration induced fatigue

ioMosaic

Piping vibration is often considered on an ad-hoc or reactive basis

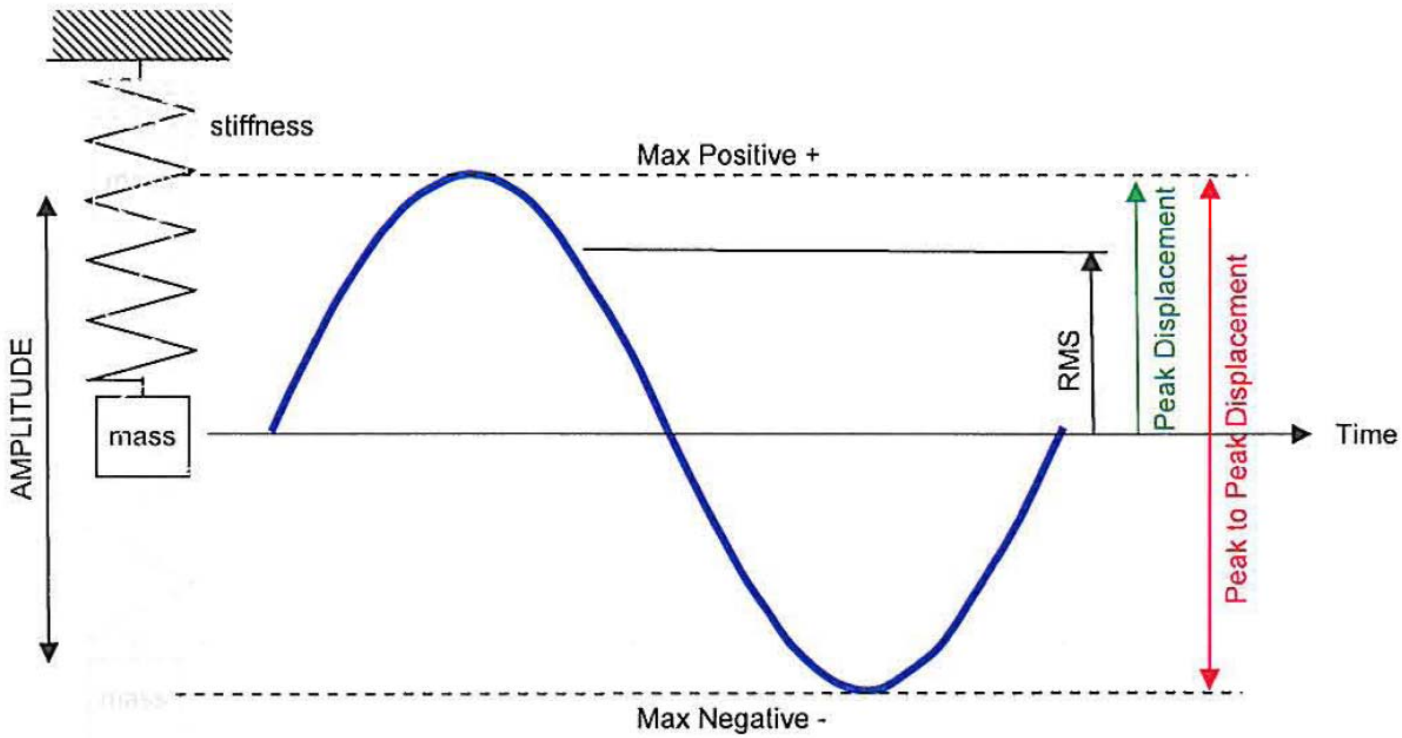
- Many operating companies have established their own internal standards and those standards vary from company to company

- Currently available public guidance documents include:
  - NORSK P-001 Process Systems, Rev. 4, October 1999
  - Harris Shock and Vibration Handbook, Chapter 29, “VIBRATION OF STRUCTURES INDUCED BY FLUID FLOW” BY R. D. Blevins
  - Energy Institute’s “Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework” (2nd Edition, 2008)

ioMosaic

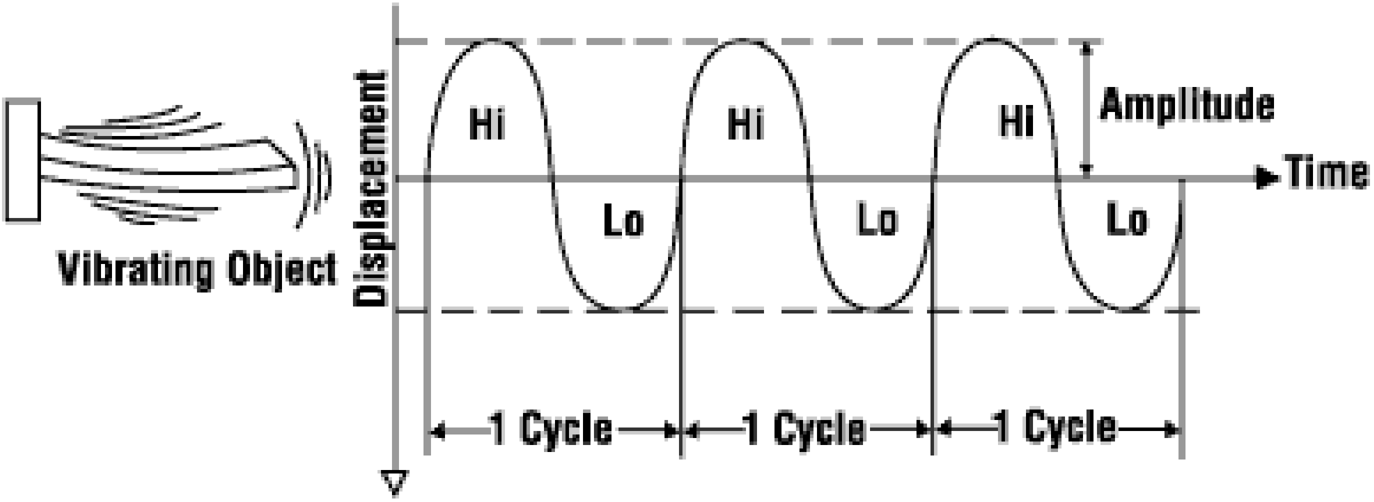
Vibration is the mechanical oscillation or repetitive motion of an object around an equilibrium position

- Vibration of an object is always caused by an excitation force, which can be exerted externally or originally from within the object



ioMosaic

The terms used to describe vibration are frequency, amplitude and acceleration



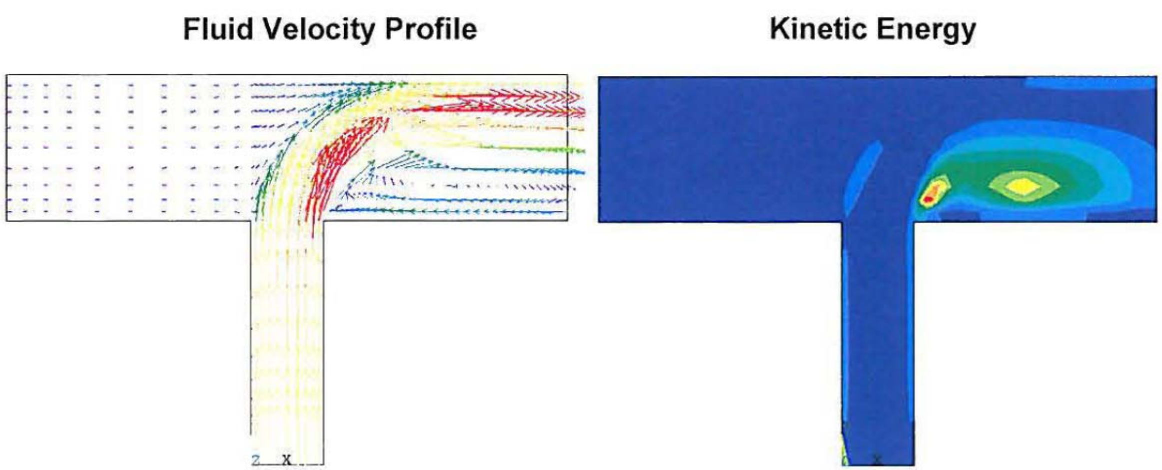
- **Frequency** - The number of cycles that a vibrating object completes in one second is called frequency. The unit of frequency is hertz (Hz).
- **Amplitude** - Amplitude is the distance from the stationary position to the extreme position on either side. The intensity of vibration depends on amplitude.
- **Acceleration** - The speed of a vibrating object varies from zero to a maximum during each cycle of vibration. It moves fastest as it passes through its stationary position to an extreme position. The vibrating object slows down as it approaches the extreme, where it stops and then moves in the opposite direction through the stationary position toward the other extreme.

Common causes of piping vibration include flow induced turbulence

- Fluid flow in pipes generates turbulent energy

- Dominant sources of turbulence are from flow discontinuities in the piping systems (e.g., partially closed valves, short radius, mitered bends, tees or expanders)

- The level of turbulence intensity is a function of pipe size, fluid density, viscosity, velocity, and structural support



Example of the distribution of kinetic energy due to turbulence generated by flow into a tee

Source: The Energy Institute Guidelines (2008)

Common causes of piping vibration also include high frequency excitation

- High frequency acoustic energy are often generated by a pressure reducing device (e.g., relief valve, control valve, or orifice plate)
- Acoustic failure is of a particular concern for safety related (e.g. relief and blowdown) systems
- High noise levels are generated by high velocity fluid impingement on the pipe wall, turbulent mixing, and if flow is choked, shockwaves downstream of flow restriction, which leads to high frequency vibration
- The severity of high frequency acoustic excitation is primarily a function of the pressure upstream and downstream of the valve, pipe diameter, and the fluid volumetric flow

Other causes of piping vibration include mechanical excitation and pulsation and vortex shedding

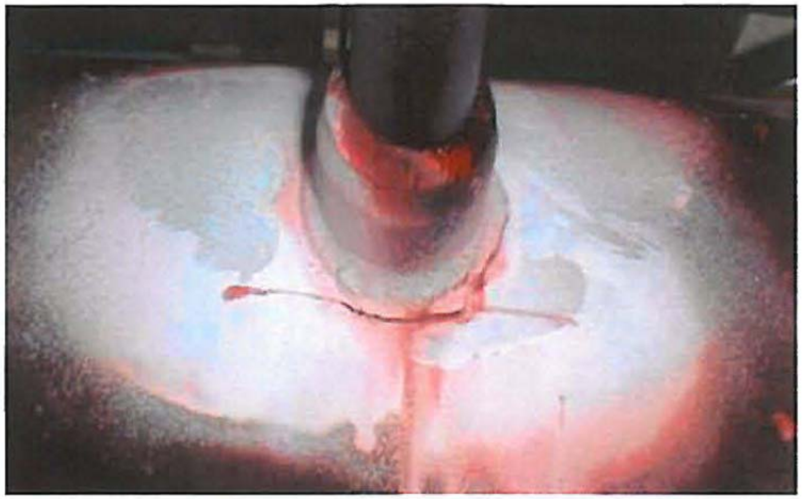
- Mechanical excitation and pulsation
  - These excitations are often associated with pipes connected with reciprocating compressors, pumps, or rotating machinery
  - Such connection machines cause vibration of the pipe and its support structure
- Vertex shedding from thermowells
  - Thermowells are intrusive fittings and are subject to static and dynamic fluid forces.
  - Vortex shedding is the dominant concern as it is capable of forcing the thermowell into flow-induced resonance and consequent fatigue failure
  - The latter is particularly significant at high fluid velocities

Other causes of piping vibration include surge/momentum changes associated with valves, cavitation, and flashing

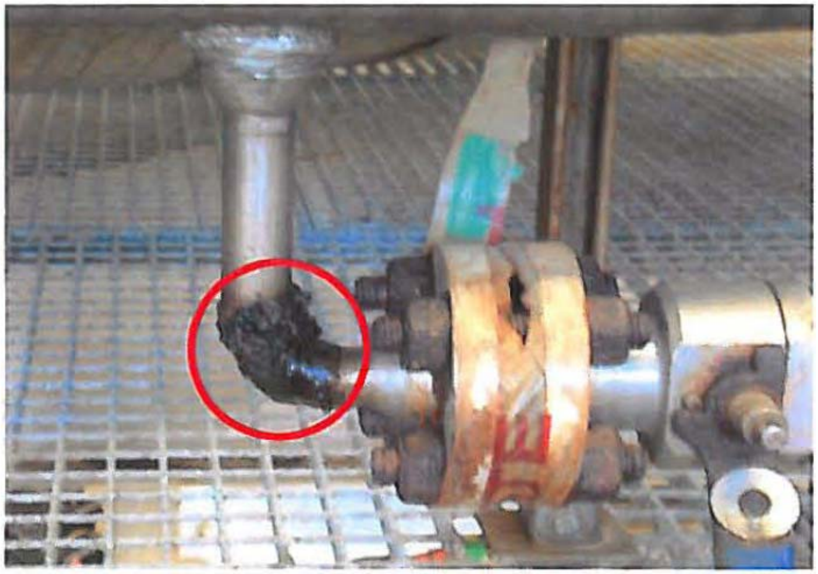
- For relief piping, flow induced turbulence and high frequency acoustic excitations are of the major concerns

ioMosaic

Samples of common vibration related failure risks



- *The most fatigue sensitive locations are welded joints associated with main lines and small bore connections*
- *Typically, fatigue failure of small bore connections occurs at the connection with the parent pipe*
- *However, depending on the local configuration fatigue failures can occur at other weld locations*
- *Example of a fatigue crack which did not occur at the connection to main line, resulting in a clear leak*

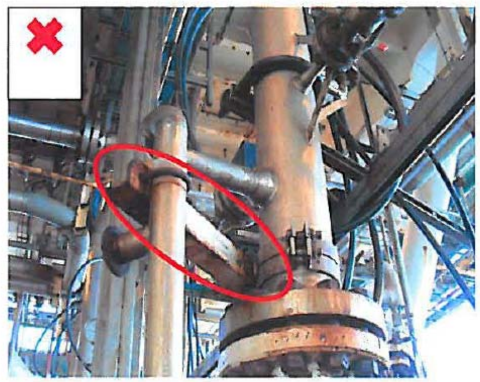


Source: Energy Institute Guidelines (2008)

ioMosaic

Samples of main line support related risks

U-Bolt providing little or no restraint to vibration; no lining between U-Bolt and pipe



Attempt to restrain main pipework by use of U-Bolt and strut (little/no lateral support)

Rope used to support pipework. In addition pipe suspended from another pipe.

Pipework clear of resting support, due to thermal growth (air gap)

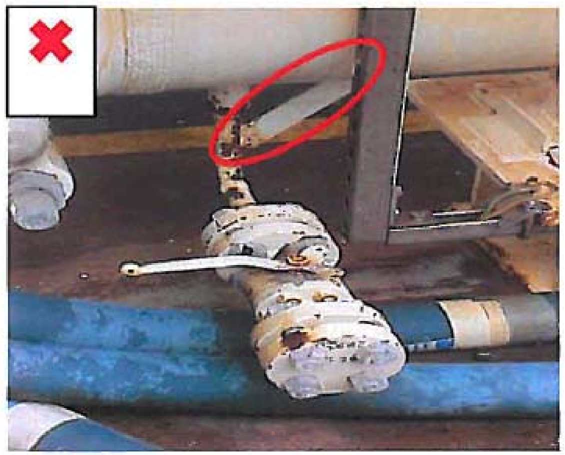
Pipework clear of resting support (air gap)

Source: Energy Institute Guidelines (2008)

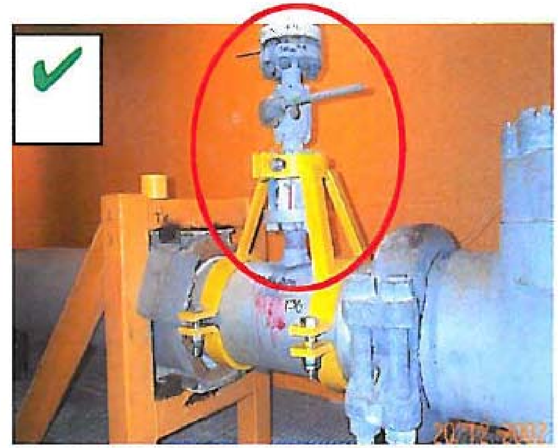
ioMosaic

Samples of small bore connections related risks

Large cantilevered mass with poor geometry



Connection braced at small bore pipe using flat bar, no support provided to the valve and potential punch through threat.

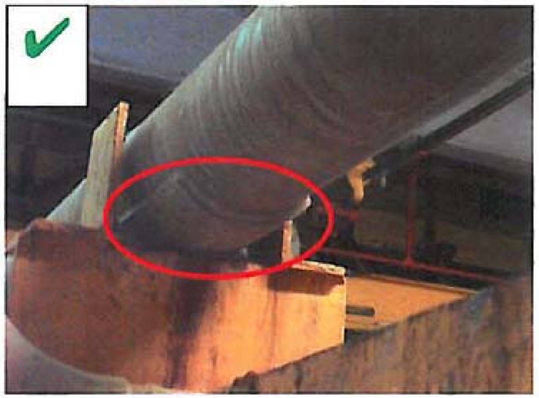


Good support of cantilevered mass

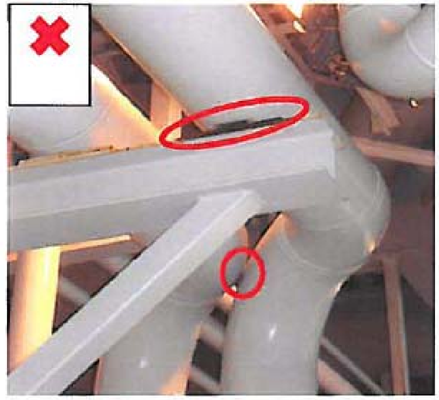
Source: Energy Institute Guidelines (2008)

ioMosaic

Samples of common fretting related risks

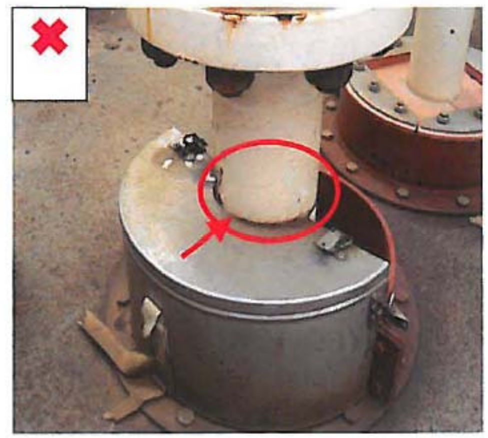


Reinforcement plate at rest support to resist fretting damage on pipe.



Fretting damage to main pipe; no resilient pad between support and pipe; also pipe clash

Resilient pad between support and pipe protects against fretting damage



Fretting damage to pipe caused by pipework vibrating relative to deck penetration cover

Source: Energy Institute Guidelines (2008)

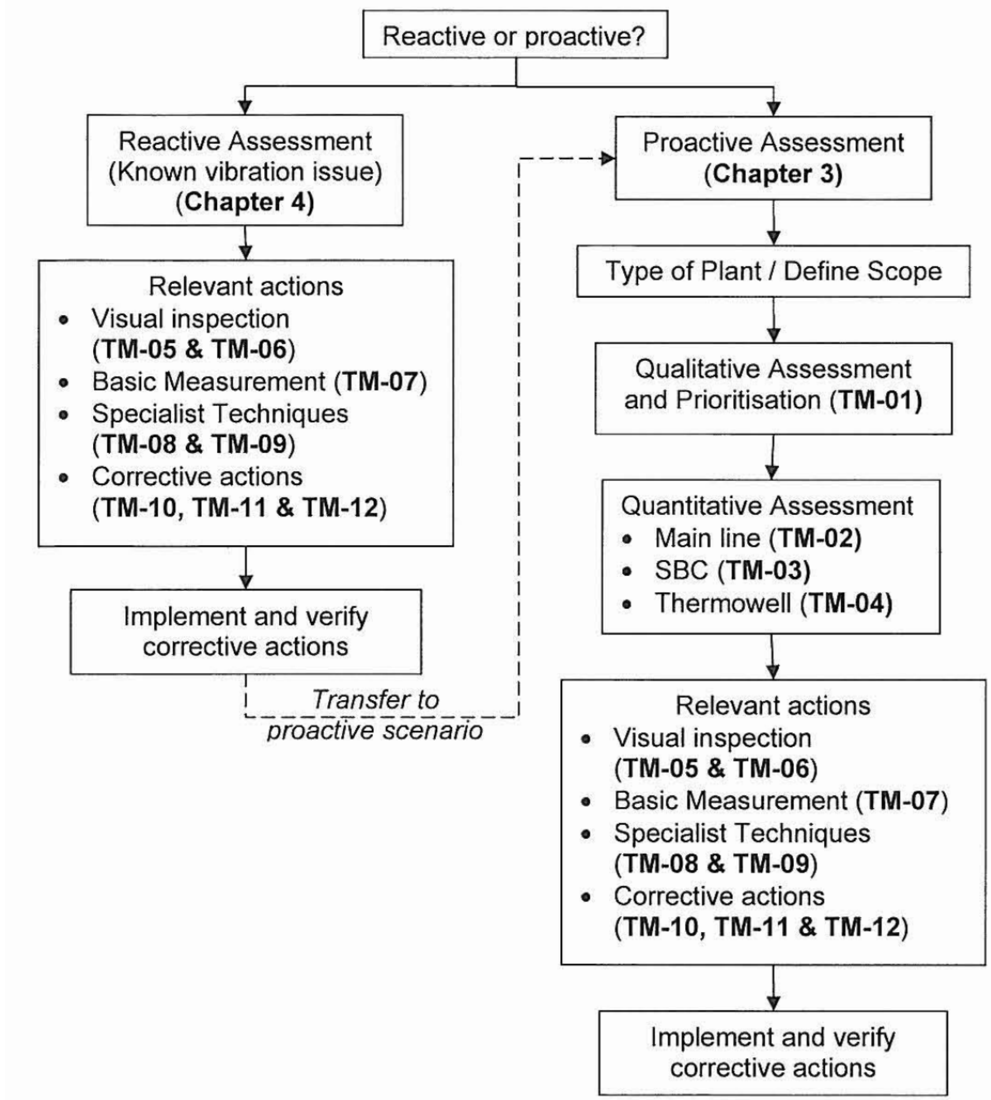
ioMosaic

We are a member of the Energy Institute and encourage you to consider membership

- The Energy Institute's "Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework" (2nd Edition, 2008) (the Guidelines)
- Copyright for the Guidelines (originally published by the Marine Technology Directorate) was transferred to the Energy Institute, in the U.K.
- Rearranged and significantly improved from the 1st Edition
- The Guidelines are used to conservatively estimate the likelihood of failure (LOF). The LOF can also be combined with a consequence of failure to determine the overall risk of a system or component

ioMosaic

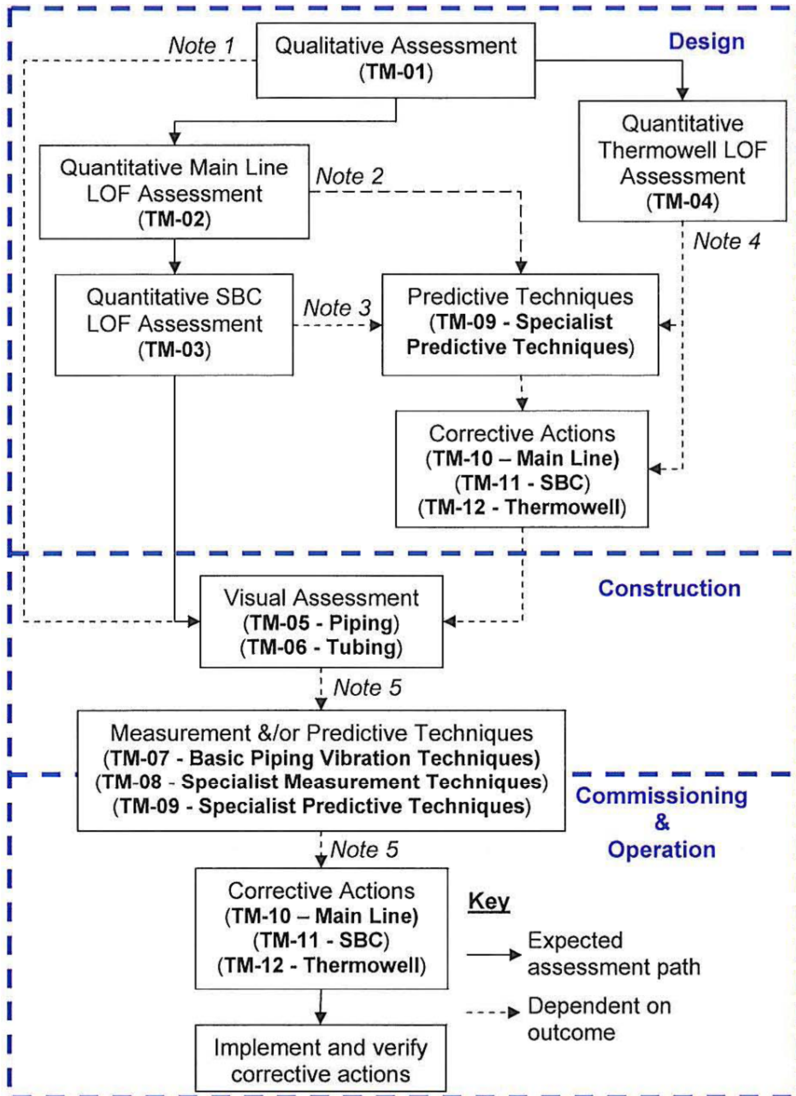
The MTD Guidelines follow a systematic approach which includes both qualitative and quantitative methods



Source: Energy Institute Guidelines (2008)

ioMosaic

The following proactive methodology for new design is proposed in the MTD Guideline



***Note 1:** If the qualitative assessment does not indicate any high or medium scores*

***Note 2:** If the main line qualitative assessment results in a LOF score greater than 0.5*

***Note 3:** If the SBC qualitative assessment results in a LOF score greater than 0.4*

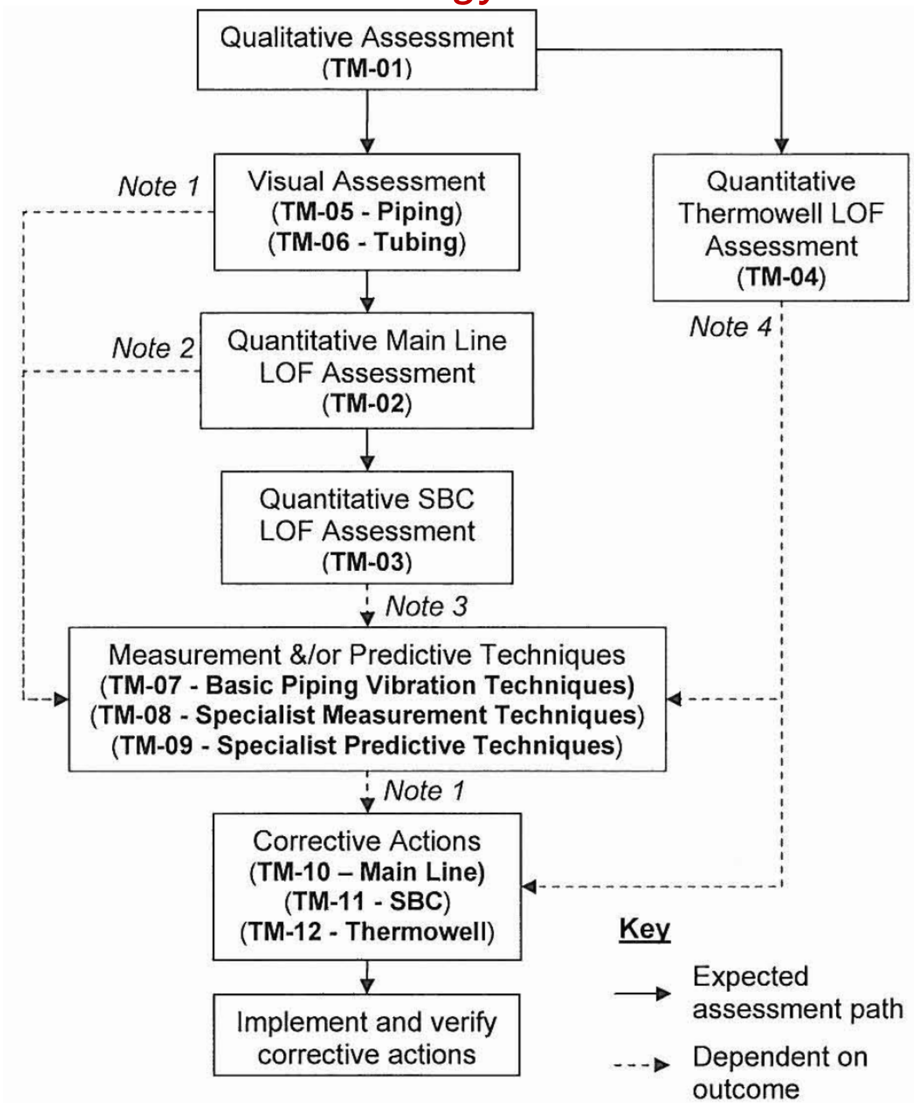
***Note 4:** If the thermowell qualitative assessment results in a LOF score of 1.0*

***Note 5:** If the location is identified to be of concern*

Source: Energy Institute Guidelines (2008)

ioMosaic

A different methodology is used for an existing plant



Note 1: *If the location is identified to be of concern*

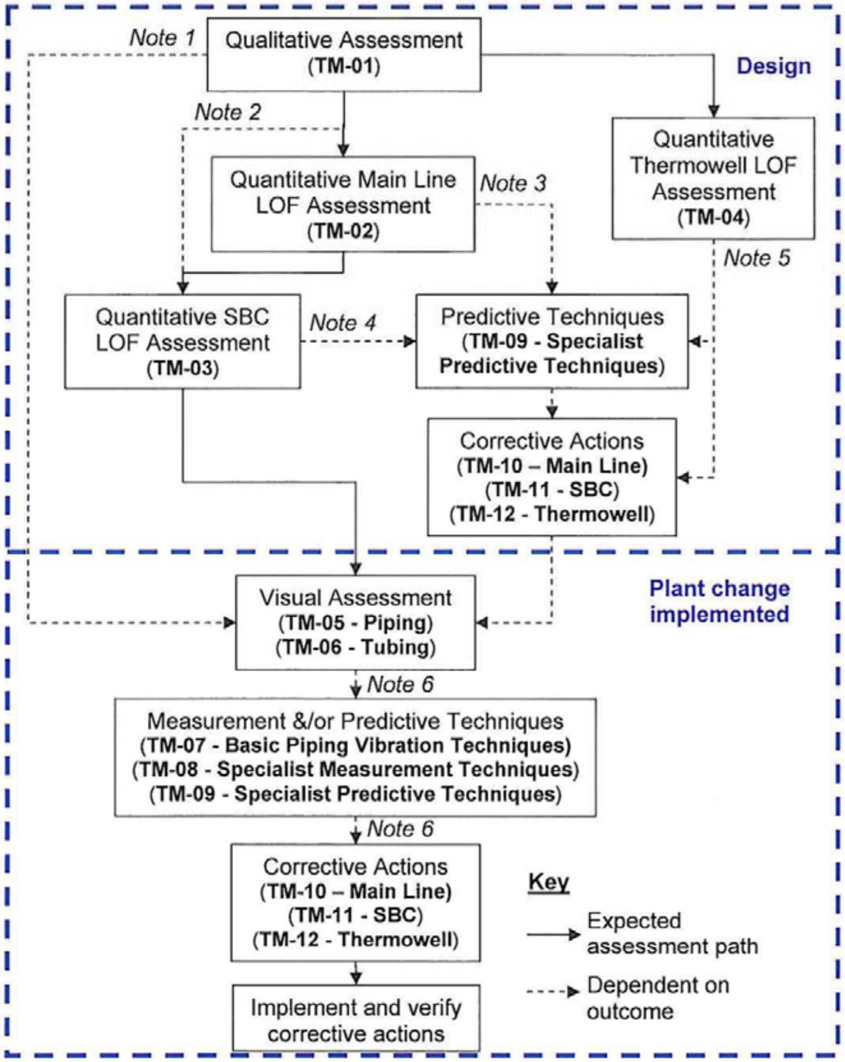
Note 2: *If the main line qualitative assessment results in a LOF score greater than 0.5*

Note 3: *If the SSC qualitative assessment results in a LOF score greater than 0.4*

Note 4: *If the thermowell qualitative assessment results in a LOF score of 1.0*

Source: Energy Institute Guidelines (2008)

The MTD guideline also provides a methodology to assess the vibration risks from changes to an existing plant



Note 1: *If the location is identified to be of concern*

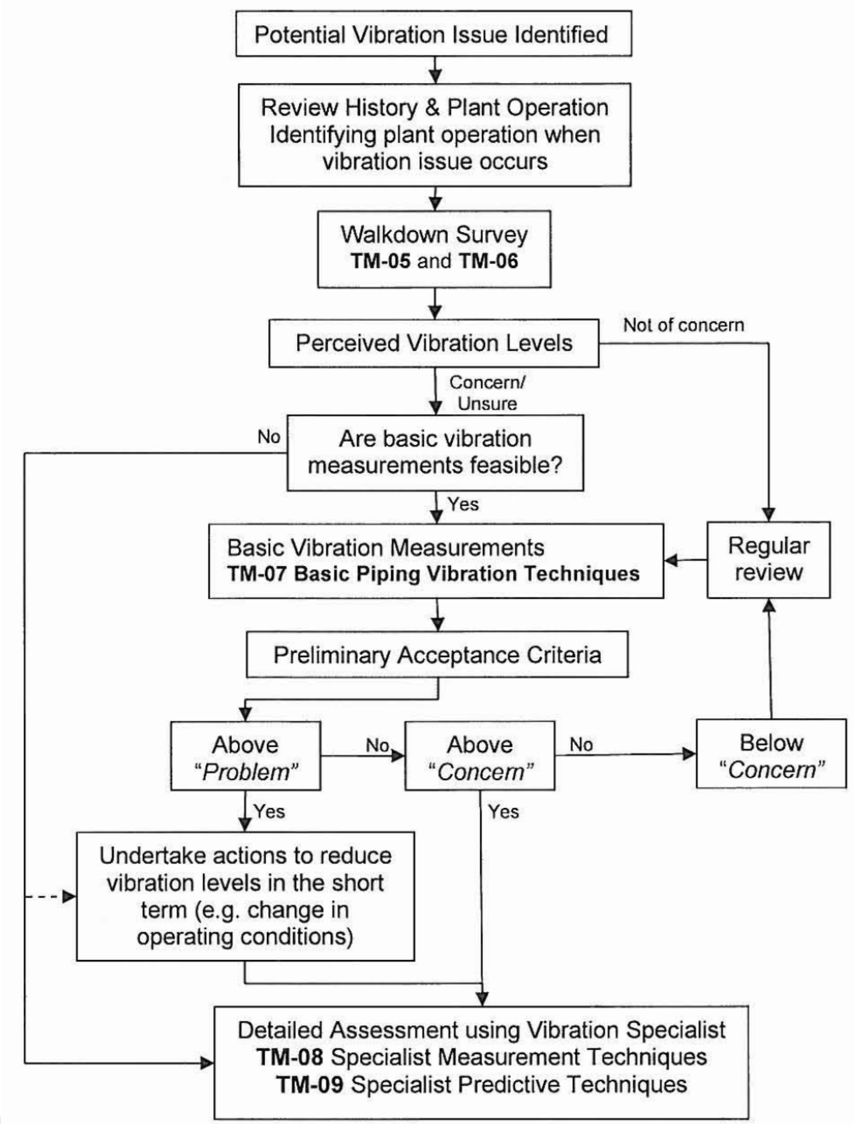
Note 2: *If the main line qualitative assessment results in a LOF score greater than 0.5*

Note 3: *If the SSC qualitative assessment results in a LOF score greater than 0.4*

Note 4: *If the thermowell qualitative assessment results in a LOF score of 1.0*

Source: Energy Institute Guidelines (2008)

If there are known vibration issues, the following methodology is proposed



Address case of an existing plant where there are know vibration issues

Once the issues are addressed, a proactive strategy should be implemented

Visual inspection and a verification process to implement and verify corrective actions are key

Source: Energy Institute Guidelines (2008)

The Guidelines' assessment methodologies are divided into several stages

- Qualitative Assessment
- Quantitative Main Line Assessment
- Quantitative Small Bore Connections (SBC) Assessment
- Quantitative Thermowell Assessment

ioMosaic

The information required is already available in relief systems studies

- Key information required include:
  - P&IDs, PFDs
  - General knowledge of the plant operation
  - Plant history, maintenance (e.g. corrosion management)
- Identify potential problem systems and their potential vibration excitation mechanisms
- Qualitative assessment questionnaire are dependent on the types of assessments
- Provide a means of risk ranking in order to prioritize the subsequent assessments

ioMosaic

*Qualitative assessment excitation factors use rho*u*u*

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	**Medium**	**High**	
1	What is the maximum value of kinetic energy (ρv^2) of the process fluid within the system under consideration?	All	$\rho v^2 < 5{,}000$ kg/m s^2	between $5{,}000 \leq \rho v^2 < 20{,}000$ kg/m s^2	$\rho v^2 \geq 20{,}000$ kg/m s^2	Flow induced turbulence (All fluids) refer to **Section T2.2** Flow induced pulsation (Gases only) refer to **Section T2.6**
2	Is choked flow possible or are sonic flow velocities likely to be encountered?	Gas	No		Yes	High frequency acoustic excitation refer to **Section T2.7**
3	Is there any rotating or reciprocating machinery?	All	No	rotating equipment only	reciprocating equipment	Mechanical excitation refer to **Section T2.3**
4	Are there any positive displacement pumps or compressors?	All	No	Screw/gear type positive displacement machine	reciprocating type positive displacement machine	Pulsation - reciprocating refer to **Section T2.4**
5	Are there any centrifugal compressors which have the potential to operate under rotating stall conditions?	Gas	No	Compressor has stall characteristics but operational restraints in place to ensure that rotating stall is not encountered	Stall rotating condition unknown. Compressor has rotating stall characteristics and may operate at conditions that will give rise to stall conditions	Pulsation - rotating stall refer to **Section T2.5**

Source: Energy Institute Guidelines (2008)

ioMosaic

*Qualitative assessment excitation factors use rho*u*u (continued)*

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Potential excitation mechanism(s)
			Low	Medium	High	
6	Are there any systems which may exhibit flashing or cavitation?	Liquid / Multiphase	No		Yes	Cavitation and Flashing refer to **Section T2.9**
7	Are there any systems with fast acting opening or closing valves?	All	No		Yes	Surge/ Momentum changes (refer to **Section T2.8**
8	Are there intrusive elements in the process stream?	Gas/Liquid	No		Yes	Vortex shedding from intrusive elements to refer to **TM-04**
9	Is there a possibility of slug flow?	Multiphase	No		Yes	Slug flow - seek specialist advice
10	Is there a history of pipework vibration issues, or are there any systems which are similar to those on another plant which have a known history of pipework vibration issues?	All	No	Yes: however, suitable corrective action in place and validated for the complete operating envelope.	Yes	Known vibration refer to **Chapter 4**

Source: Energy Institute Guidelines (2008)

ioMosaic

The qualitative methodology also includes condition and operating factors

Item	Aspect	Applicable process fluid(s)	Likelihood Classification			Contributory factor
			Low	Medium	High	
A	What is the quality of construction?	All	Better than industry standards	At industry standard	Below industry standards	Build quality
B	What is the effectiveness of the plant maintenance programme (including corrosion management)?	All	Better than industry standards	At industry standard	Below industry standards	Corrosion/ maintenance management
C	Are there any cyclical operations (e.g. batch operation)?	All	No		Yes	Cyclical loading
D	What is the number of unplanned process interruptions in an average year? (this is intended for normal continuous process operations)	All	0-1	2-8	9 or more	Process upsets

Source: Energy Institute Guidelines (2008)

ioMosaic

Qualitative Assessment Issues for Changes to Plant

Item	Description	If *Yes* - Potential Issues
1	Will the modification result in one or more of the following: • An increase in flow velocities by more than 5% over previous operational experience? • An increase in fluid density by more than 10% over previous operational experience?	• Flow induced turbulence (all fluids),refer to **Section T2.2** • Flow induced pulsation (gases systems only), refer to **Section T2.6** • Vortex shedding from intrusive elements (all fluids), refer to **TM-04** • Surge/Momentum Change refer to **Section T2.8**
2	For a gas system, will the modification result in one or more of the following: • A change in the molecular weight of the gas by more than ± 5% from previous maximum/minimum operational experience? • A change to the temperature of the gas by more than ± 5% from previous maximum/minimum operational experience? • A change to the ratio of specific heats (C_p/C_v) of the gas by more than ± 5% from previous maximum/minimum operational experience?	For all systems: • Pulsation - Flow induced excitation, refer to **Section T2.6** If there is a centrifugal compressor: • Pulsation - rotating stall (gas systems only) refer to **Section T2.5** If there is a reciprocating compressor: • Pulsation – reciprocating compressor (gas systems only) refer to **Section T2.4**
3	For a liquid system incorporating a reciprocating / positive displacement pump, will the modification result in one or more of the following: • A change in the density of the liquid by more than ± 5% from previous maximum/minimum operational experience? • A change to the bulk modulus of the liquid by more than ± 5% from previous maximum/minimum operational experience?	• Pulsation – reciprocating /positive displacement pump (liquid systems only) refer to **Section T2.4**

Source: Energy Institute Guidelines (2008)

Qualitative Assessment Issues for Changes to Plant (Continued)

Item	Description	If *Yes* - Potential Issues
4	Will the modification result in a change to the operational configuration of a positive displacement compressor or pump which is outside existing operational experience e.g.: • The use of a second compressor/pump in tandem? • The use of compressor/pump recycle or partial unloading of the compressor?	• Pulsation – reciprocating /positive displacement compressor or pump (liquid and gas systems only) refer to **Section T2.4**
5	Will the modification result in a centrifugal compressor being operated at low flow conditions?	• Pulsation - rotating stall (gas systems only) refer to **Section T2.5**
6	Will the modification result in choked flow and/or sonic velocities in the pipework?	• High frequency acoustic excitation (gas systems only) refer to **Section T2.7**
7	Will the modification result in flashing or cavitation?	• Cavitation and Flashing refer to **Section T2.9**
8	Will the modification result in a change or addition to the existing pipework or associated equipment (valves, machinery or intrusive elements such as thermowells) which is not a like-for-like replacement?	For changes to valves (including change of valve type or changes to valve closing timings) check for: • Surge/Momentum Change refer to **Section T2.8** For changes to machinery check for: • Mechanical excitation refer to **Section T2.3** For changes to thermowells check for: • Vortex shedding from intrusive elements refer to **TM-04** For changes to pipework, supports, small bore connections and tubing check for: • Poor geometry refer to **TM-05** and **TM-06**
9	Will the modification result in slug flow?	• Slug flow - seek specialist advice

Source: Energy Institute Guidelines (2008)

Quantitative Main Line Assessment

- Key information required include:
  - P&IDs, PFDs
  - General knowledge of the plant operation
  - Selected piping isometrics
  - Detailed equipment and process information (e.g. valve datasheets, flow rates, fluid densities)
- Determine the likelihood of failure from various vibration excitation mechanisms as identified from the Qualitative Assessment stage
- List corresponding corrective action items based on the severity indicated by the calculated likelihood of failure (LOF)

ioMosaic

Quantitative Small Bore Connections Assessment

- Key information required include:
  - Main line LOF
  - SBC geometry
  - SBC location
- For each main pipe under consideration, assess all SBCs as “flagged” from the Qualitative Assessment and Main Line Assessment
- Calculate the likelihood of failure
- List corresponding corrective action items based on the severity indicated by the calculated LOF

ioMosaic

Quantitative Thermowell Assessment

- Key information required include:
  - Process data
  - Thermowell geometry
  - Main line schedule
- For each main pipe under consideration, assess all applicable thermowells as “flagged” from the Qualitative Assessment
- Calculate the likelihood of failure
- List corresponding corrective action items based on the severity indicated by the calculated LOF

ioMosaic

Procedure for flow induced turbulence assessment

- Determine ρv^2
  - For single phase

    ρv^2 = *(actual density) x (actual velocity)*^{2}
  - For multi-phase

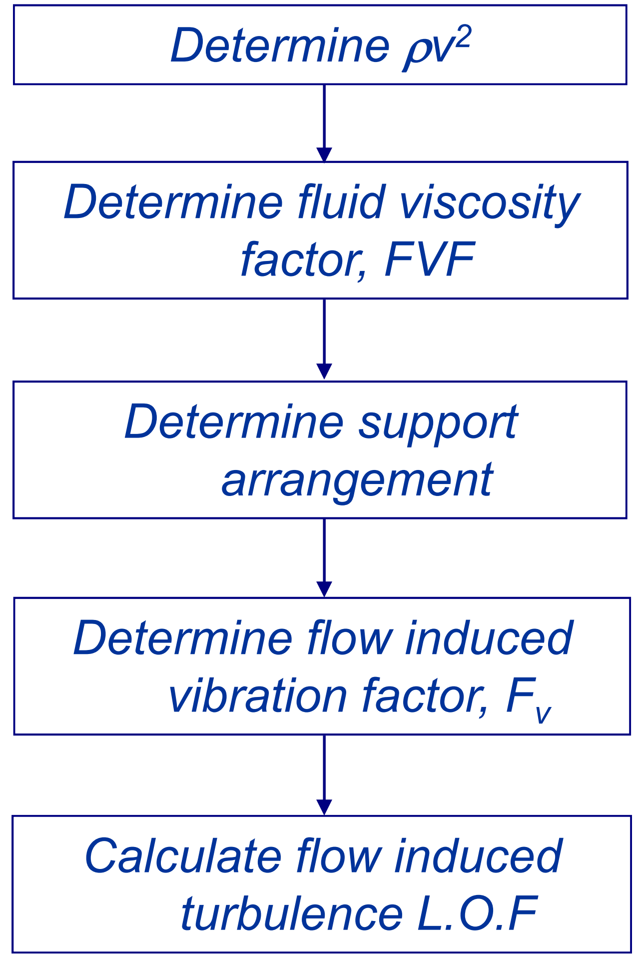
    ρv^2 = *(effective density) x (effective velocity)*^{2}

- Determine fluid viscosity factor, FVF
  - For single phase

$$FVF = \frac{\sqrt{\mu_{gas}}}{\sqrt{1x10^{-3}}}$$

    where μ_{gas} is the dynamic viscosity
  - For multi-phase

    FVF = 1



Source: MTD Guidelines, 1999. All units are in SI.

ioMosaic

Procedure for flow induced turbulence assessment

- Determine support arrangement

Support Arrangement	Span Length Criteria	Typical Fundamental Natural Frequency
Stiff	$L_{span} \leq -1.2346 * 10^{-5} D_{ext}{}^{2} + 0.02 D_{ext} + 2.0563$	14 to 16 Hz
Medium Stiff	$L_{span} > -1.2346 * 10^{-5} D_{ext}{}^{2} + 0.02 D_{ext} + 2.0563$ $L_{span} \leq -1.1886 * 10^{-5} D_{ext}{}^{2} + 0.025262 D_{ext} + 3.3601$	7 Hz
Medium	$L_{span} > -1.1886 * 10^{-5} D_{ext}{}^{2} + 0.025262 D_{ext} + 3.3601$ $L_{span} \leq -1.5968 * 10^{-5} D_{ext}{}^{2} + 0.033583 D_{ext} + 4.429$	4 Hz
Flexible	$L_{span} > -1.5968 * 10^{-5} D_{ext}{}^{2} + 0.033583 D_{ext} + 4.429$	1 Hz

Different support arrangements as a function of span length and outside diameter

Source: Energy Institute Guidelines (2008)

ioMosaic

Procedure for flow induced turbulence assessment

- Determine the flow induced vibration factor, Fv

Support Arrange-ment	Range of Outside Diameter	F_v	α	β
Stiff	60 mm to 762 mm	$\alpha\left(D_{ext}/T\right)^{\beta}$	$446187+646\,D_{ext}+9.17\cdot10^{-4}D_{ext}^{3}$	$0.1\ln(D_{ext})-1.3739$
Medium Stiff	60 mm to 762 mm	$\alpha\left(D_{ext}/T\right)^{\beta}$	$283921+370D_{ext}$	$0.1106\ln(D_{ext})-1.501$
Medium	273. mm to 762 mm	$\alpha\left(D_{ext}/T\right)^{\beta}$	$150412+209\,D_{ext}$	$0.0815\ln(D_{ext})-1.3269$
Medium	60 mm to 219 mm	$\exp\left[\alpha\left(D_{ext}/T\right)^{\beta}\right]$	$13.1-4.75\cdot10^{-3}\,D_{ext}+1.41\cdot10^{-5}\,D_{ext}^{2}$	$-0.132+2.28\cdot10^{-4}\,D_{ext}-3.72\cdot10^{-7}\,D_{ext}^{2}$
Flexible	273 mm to 762 mm	$\alpha\left(D_{ext}/T\right)^{\beta}$	$41.21\,D_{ext}+49397$	$0.0815\ln(D_{ext})-1.3842$
Flexible	60 mm to 219 mm	$\exp\left[\alpha\left(D_{ext}/T\right)^{\beta}\right]$	$1.32\cdot10^{-5}\,D_{ext}^{2}-4.42\cdot10^{-3}\,D_{ext}+12.22$	$2.84\cdot10^{-4}\,D_{ext}-4.62\cdot10^{-7}\,D_{ext}^{2}-0.164$

Source: Energy Institute Guidelines (2008)

ioMosaic

Procedure for flow induced turbulence assessment

- Calculate likelihood of failure (LOF) for flow induced turbulence

$$LOF = \frac{\rho \upsilon^2}{F_v} FVF$$

- For flexible pipes, LOF is very sensitive to the fundamental natural frequency (particularly having natural frequency between 1 and 3 Hz) and further detailed method is needed to estimate the LOF.

ioMosaic

Main Line Actions

Score	Action
LOF >= 1.0	➢ The main line shall be redesigned, re-supported or a detailed analysis of the main line shall be conducted, and vibration monitoring of the main line shall be undertaken (TM-07,08,09) ➢ Corrective actions shall be examined and applied as necessary (TM-10) ➢ Small bore connections on the main line shall be assessed. (TM-03) ➢ A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighboring pipework. (TM-05, 06)
1 > LOF >= 0.5	➢ The main line should be redesigned, re-supported or a detailed analysis of the main line should be conducted, or vibration monitoring of the main line should be undertaken (TM-07,08,09) ➢ Corrective actions should be examined and applied as necessary (TM-10) ➢ Small bore connections on the main line shall be assessed. (TM-03) ➢ A visual survey shall be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission to neighboring pipework. (TM-05, 06)
0.5 > LOF >= 0.3	➢ Small bore connections on the main line should be assessed. (TM-03) ➢ A visual survey should be undertaken to check for poor construction and/or geometry and/or support for the main line and/or potential vibration transmission from other sources. (TM-05, 06)
LOF < 0.3	➢ A visual survey should be undertaken to check for poor construction and/or geometry and/or support or the main line and/or potential vibration transmission from other sources. (TM-05, 06)

Source: Energy Institute Guidelines (2008)

ioMosaic

Small Bore Connections Actions

Score	Action
LOF >= 0.7	➢ The SBC shall be redesigned, re-supported or a detailed analysis shall be conducted, and vibration monitoring of the SBC shall be undertaken (TM-07,08,11) ➢ A visual survey shall be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing. (TM-05,06)
0.7> LOF >= 0.4	➢ Vibration monitoring of the SBC should be undertaken. Alternatively the SBC may be redesigned, re-supported or a detailed analysis conducted. (TM-07,08,11) ➢ A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing. (TM-05,06)
LOF < 0.4	➢ A visual survey should be undertaken to check for poor construction and/or geometry for the SBC's and instrument tubing. (TM-05,06)

Source: Energy Institute Guidelines (2008)

Thermowell Actions

Score	Action
LOF = 1.0	➢ Modify the thermowell or a detailed analysis shall be conducted.
LOF = 0.29	➢ No action required

Source: Energy Institute Guidelines (2008)

ioMosaic

Sample of Corrective Actions for Main Line

- **For flow induced turbulence:**
  - Decrease the flow velocity by increasing the diameter of the main pipe
  - Run a second pipe in parallel
  - Increase the pipe wall thickness
  - Minimize the number of bends in a system, the use of long radius bends, etc.
  - Stiffen the main line and its supporting structure
- **For high frequency acoustic excitation:**
  - Use of low noise trim, particularly for control valve
  - Increasing the pipe wall thickness
  - Reduce the flow rate

Sample of Corrective Actions for Small Bore Connections

- The fitting and overall unsupported length should be as short as possible
- The mass of valves and instrumentation should be minimized
- Any bracing supports should be from the main pipe, thus ensuring that the small bore connection moves with the main pipe

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire, Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief and flare systems design services and solutions. Its pressure relief system applications are used by over 300 users worldwide. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009

HOUSTON OFFICE
2401 Fountain View, Suite 850
Houston, Texas 77057
Tel: 713-490-5220

MINNEAPOLIS OFFICE
401 North 3rd Street, Suite 410
Minneapolis, Minnesota 55401
Tel: 612-338-1669

discovering solutions

Screening for Vibration Induced Fatigue in Process and Relief Piping

By Daniel Nguyen

2007 Fall DIERS Users Group Meeting

Manchester, New Hampshire

ioMosaic

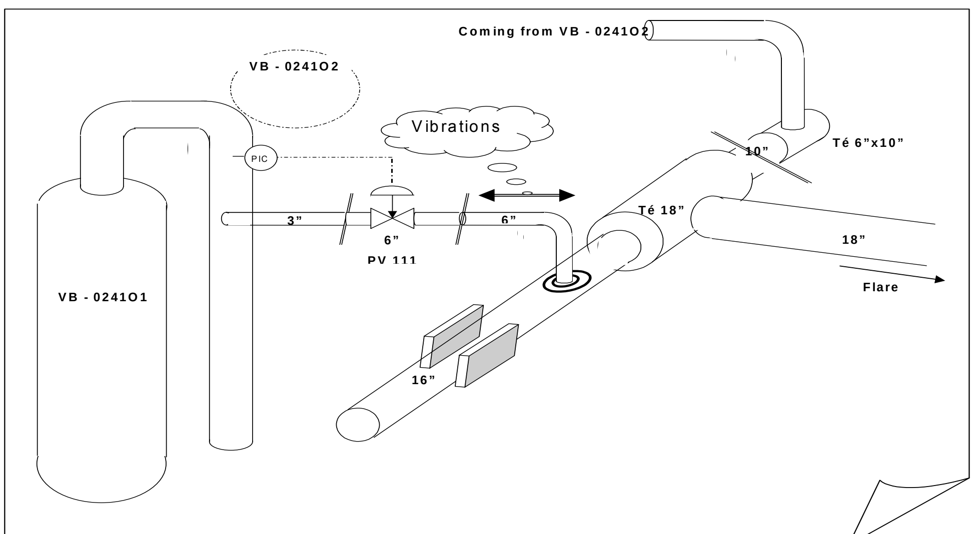
SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009
Fax:603-893-7885
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220
Fax:713-490-5222
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

MINNEAPOLIS OFFICE
333 Washington Avenue North
Minneapolis, Minnesota 55401
Tel: 612-373-7037
Fax:832-553-7283
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

Salah Gas Incident

It is believed that flow induced turbulence caused the failure of this line at header tie-in



Source: Ed Zamejc, Spring 2006 API Refining Meeting

ioMosaic

Overview

- Factors that have led to an increasing incidence of vibration related fatigue failures in piping systems:
  - Increasing flow rates as a result of debottlenecking contributes to higher flow velocities with a correspondingly greater level of turbulent energy in process systems;
  - Frequent use of thin-walled piping which results in higher stress concentrations, particularly at small bore connections
- Process and relief piping systems are often designed with little attention paid to vibration induced fatigue
- Most piping design codes do not emphasize the issue of vibration proactively
- Take our most commonly referenced standard, ISO/API 521:
  - No specific guidance on velocity limitations other than backpressure calculations
  - No guidance on acoustic fatigue or vibration induced fatigue

ioMosaic

Overview

- As a result, piping vibration is often considered on an ad-hoc or reactive basis

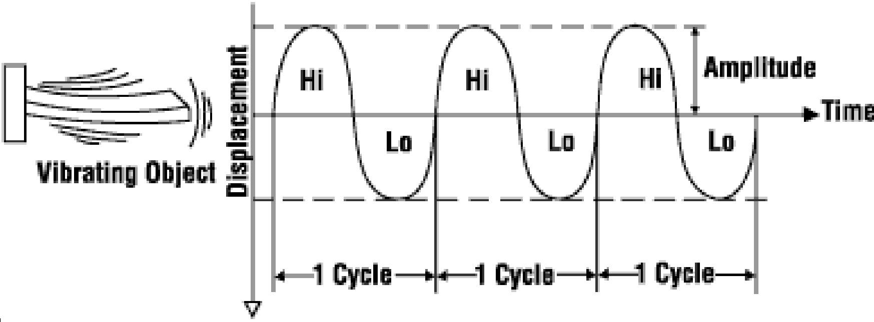
- Many operating companies have established their own internal standards and those standards vary from company to company

- Currently available public guidance documents include:
  - NORSK P-001 Process Systems, Rev. 4, October 1999
  - **MTD "Guidelines for Avoidance of Vibration Induced Fatigue in Process Pipework" ISBN 1870553 37 3, 1999 [There is a second revision pending]**
  - Harris Shock and Vibration Handbook, Chapter 29, "VIBRATION OF STRUCTURES INDUCED BY FLUID FLOW" BY R. D. Blevins

- SuperChems offers state of the art computational flow models that provide detailed estimates of kinetic energy and sound pressure/power levels required for the assessment of the risk of vibration induced failure of process and relief piping.

Vibration in brief

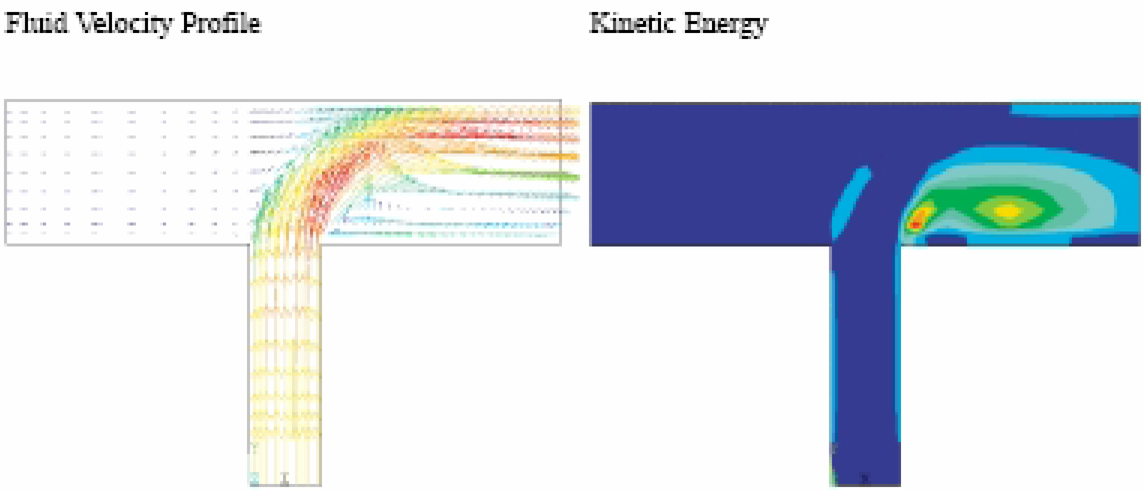
- Mechanical oscillation or repetitive motion of an object around an equilibrium position.



- Vibration of an object is always caused by an excitation force, which can be exerted externally or originally from within the object.
  - **Frequency** - A vibrating object moves back and forth from its normal stationary position. A complete cycle of vibration occurs when the object moves from one extreme position to the other extreme, and back again. The number of cycles that a vibrating object completes in one second is called frequency. The unit of frequency is hertz (Hz).
  - **Amplitude** - A vibrating object moves to a certain maximum distance on either side of its stationary position. Amplitude is the distance from the stationary position to the extreme position on either side. The intensity of vibration depends on amplitude.
  - **Acceleration** - The speed of a vibrating object varies from zero to a maximum during each cycle of vibration. It moves fastest as it passes through its stationary position to an extreme position. The vibrating object slows down as it approaches the extreme, where it stops and then moves in the opposite direction through the stationary position toward the other extreme.

Common causes of piping vibration

- Flow induced turbulence
  - Fluid flow in pipes generates turbulent energy
  - Dominant sources of turbulence are from flow discontinuities in the piping systems (e.g., partially closed valves, short radius, mitered bends, tees or expanders)
  - The level of turbulence intensity is a function of pipe size, fluid density, viscosity, velocity, and structural support



An example of the distribution of kinetic energy due to turbulence generated by flow into a tee (MTD Guidelines, 1999)

Common causes of piping vibration

- High frequency acoustic excitation
  - High frequency acoustic energy are often generated by a pressure reducing device (e.g., relief valve, control valve, or orifice plate)
  - Acoustic failure is of a particular concern for safety related (e.g. relief and blowdown) systems
  - High noise levels are generated by high velocity fluid impingement on the pipe wall, turbulent mixing, and if flow is choked, shockwaves downstream of flow restriction, which leads to high frequency vibration
  - The severity of high frequency acoustic excitation is primarily a function of the pressure upstream and downstream of the valve, pipe diameter, and the fluid volumetric flow

Common causes of piping vibration

- Mechanical excitation and pulsation
  - These excitations are often associated with pipes connected with reciprocating compressors, pumps, or rotating machinery
  - Such connection machines cause vibration of the pipe and its support structure
- Other causes of piping vibration include surge/momentum changes associated with valves, cavitation, and flashing
- For relief piping, flow induced turbulence and high frequency acoustic excitations are of the major concerns and therefore are the central discussion in this presentation

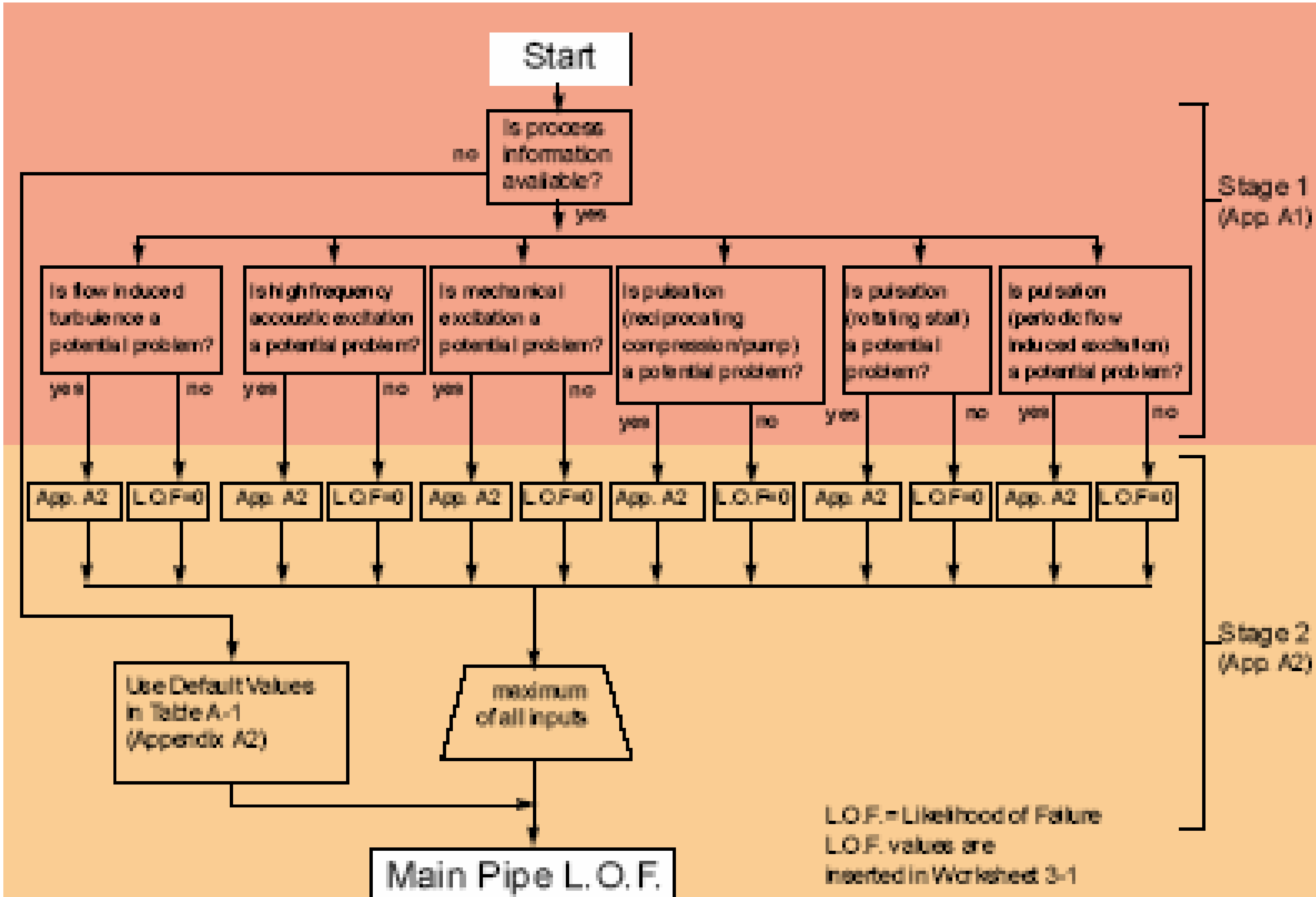
MTD Guidelines Screening Methodologies

- The “Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework” published by Marine Technology Directorate (MTD, 1999), is now administered by the Energy Institute (2nd Revision to be released soon)

- The Guidelines are used to conservatively estimate the likelihood of failure (LOF). The LOF can also be combined with a consequence of failure to determine the overall risk of a system or component.

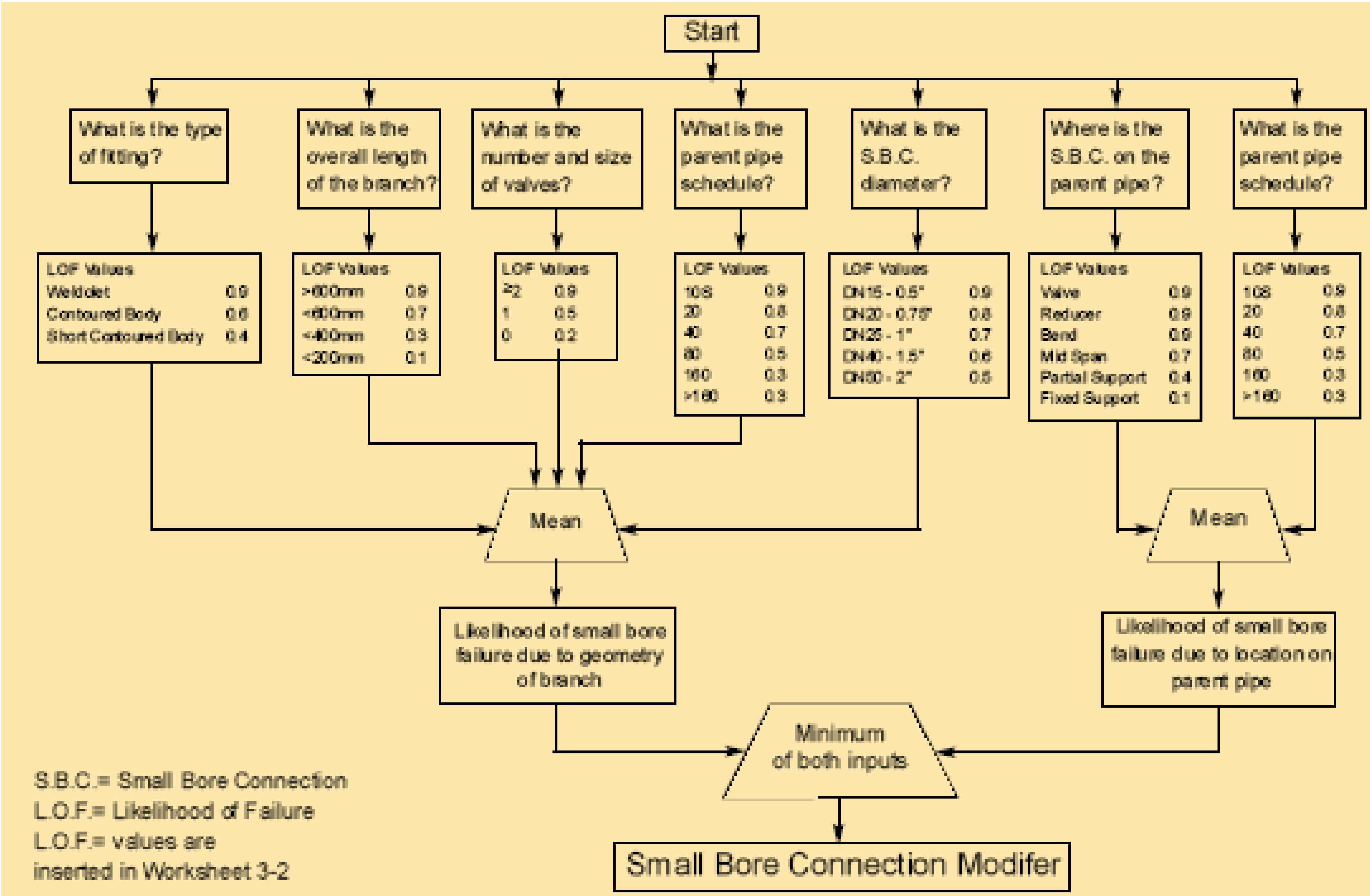
- The Guidelines’ assessment methodologies are mainly divided into three stages:
  - Stage 1: Identification of excitation mechanisms – Identify potential problem systems and their potential vibration excitation mechanisms by survey questionnaire
  - Stage 2: Detailed screening of the main pipe – Assess all main pipe systems identified in Stage 1 and determine the LOF. Appropriate action items are recommended based on the severity indicated by the calculated LOF.
  - Stage 3: Detailed screening of small bore connections (SBC) – Excitation energy that causes vibration comes from the main pipe. For each main pipe under consideration, assess all SBCs as “flagged” from Stage 1 and 2, and then determine the LOF. Recommended action items are proposed based the calculated LOF.

Screening methodologies – Stages 1 and 2



Reference: MTD Guidelines, 1999.

Screening methodologies – Stage 3



Reference: MTD Guidelines, 1999.

ioMosaic

Procedure for flow induced turbulence assessment

- Determine ρv^2
  - For single phase

    ρv^2 = (actual density) x (actual velocity)^2
  - For multi-phase

    ρv^2 = (effective density) x (effective velocity)^2

- Determine fluid viscosity factor, FVF
  - For single phase

    $$FVF = \frac{\sqrt{\mu_{gas}}}{\sqrt{1x10^{-3}}}$$

    where μ_{gas} is the dynamic viscosity
  - For multi-phase

    FVF = 1

Determine rv^2

↓

Determine fluid viscosity factor, FVF

↓

Determine support arrangement

↓

Determine flow induced vibration factor, F_v

↓

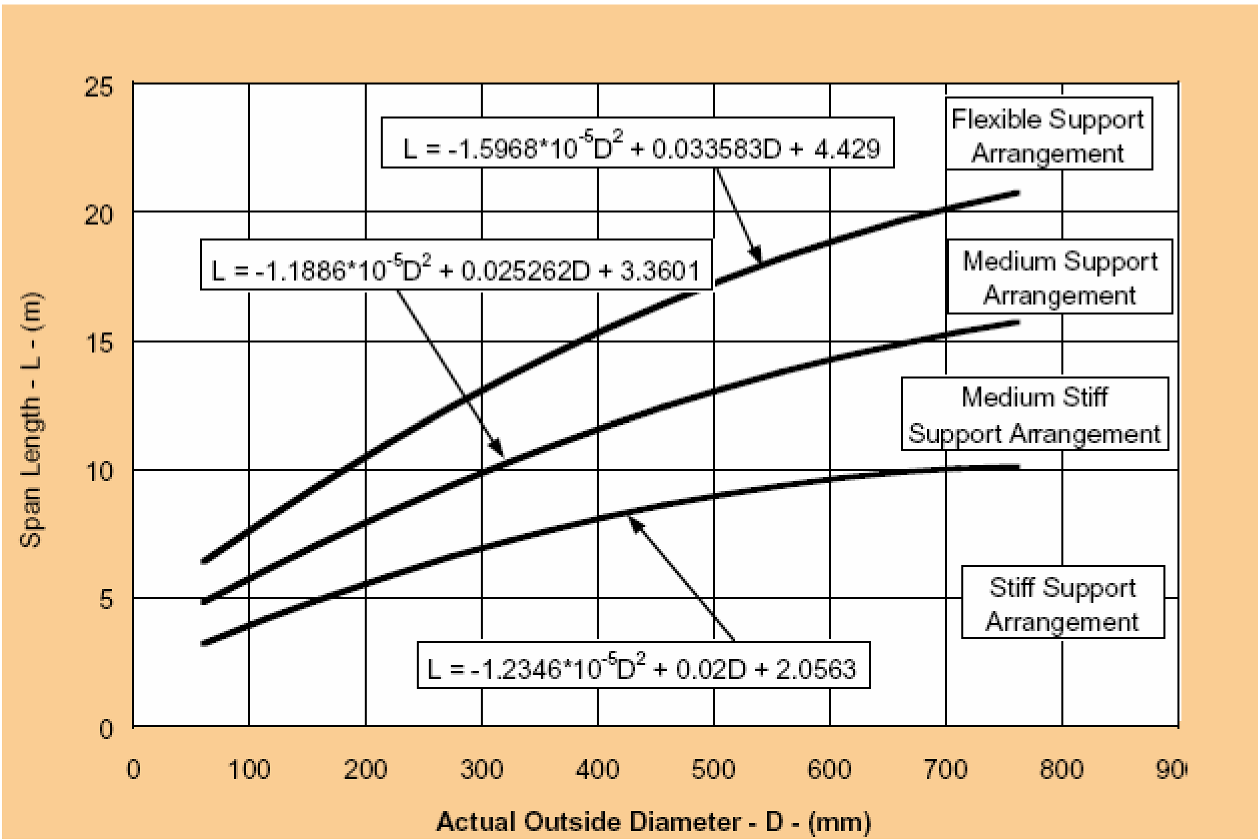
Calculate flow induced turbulence L.O.F

Reference: MTD Guidelines, 1999. All units are in SI.

ioMosaic

Procedure for flow induced turbulence assessment

- Determine support arrangement



Different support arrangements as a function of span length and outside diameter

Reference: MTD Guidelines, 1999.

Procedure for flow induced turbulence assessment

- Determine the flow induced vibration factor, Fv

	Range of Outside Diameter	F_v	α	β	Figure No
Stiff	60 mm to 762 mm	$\alpha(D/T)^{\beta}$	$446187.1 + 645.51D + 9.166*10^{-4}D^{3}$	$0.1\ln(D) - 1.3739$	A-3
Medium Stiff	60 mm to 762 mm	$\alpha(D/T)^{\beta}$	$283921 + 370.24D$	$0.1106\ln(D) - 1.501$	A-4
Medium	273. mm to 762 mm	$\alpha(D/T)^{\beta}$	$150412 + 208.93D$	$0.0815\ln(D) - 1.3269$	A-5
Medium	60 mm to 219 mm	$\exp\left[\alpha(D/T)^{\beta}\right]$	$13.1129 - 4.7455*10^{-3}D + 1.4141*10^{-5}D^{2}$	$-0.132 + 2.282*10^{-4}D - 3.7245*10^{-7}D^{2}$	A-6
Flexible	273 mm to 762 mm	$\alpha(D/T)^{\beta}$	$41.21D + 49397$	$0.0815\ln(D) - 1.3842$	A-7
Flexible	60 mm to 219 mm	$\exp\left[\alpha(D/T)^{\beta}\right]$	$1.3175*10^{-5}D^{2} - 4.4213*10^{-2}D + 12.217$	$-4.622*10^{-7}D^{2} + 2.835*10^{-4}D - 0.164$	A-8

Methods for calculating Fv

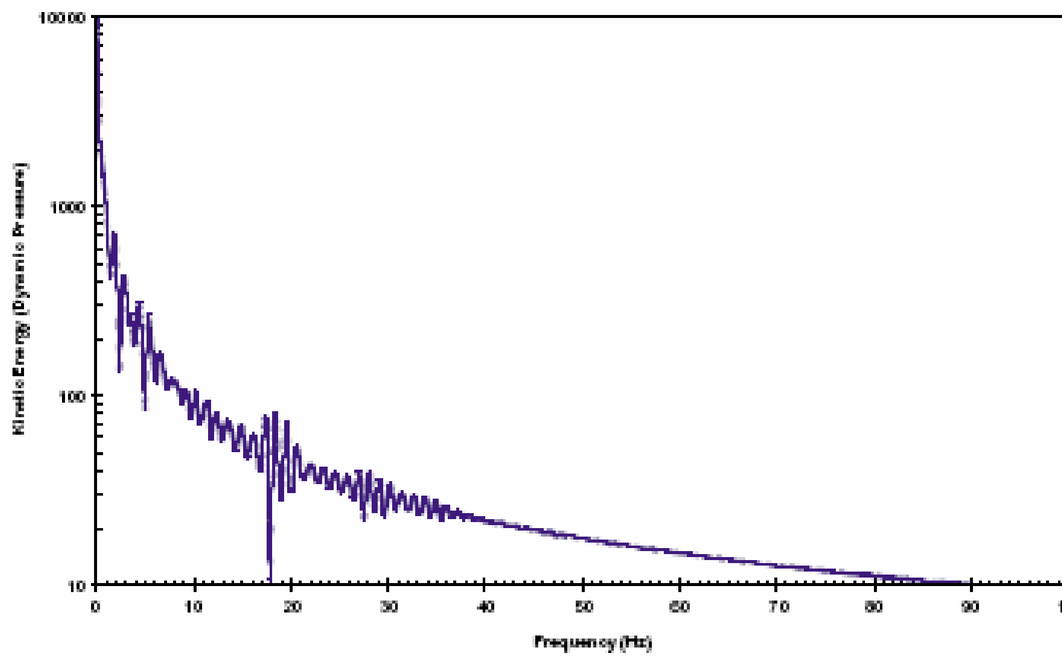
Reference: MTD Guidelines, 1999.

Procedure for flow induced turbulence assessment

- Calculate likelihood of failure (LOF) for flow induced turbulence

$$LOF = \frac{\rho \upsilon^2}{F_v} FVF$$

- For flexible pipes, LOF is very sensitive to the fundamental natural frequency (particularly having natural frequency between 1 and 3 Hz) and further detailed method is needed to estimate the LOF.



Turbulent energy as a function of frequency

Reference: MTD Guidelines, 1999.

ioMosaic

Procedure for high frequency acoustic excitation assessment

➢ Calculate source sound power level (PWL)

$$PWL(source) = 10\log_{10}\left[\left(\frac{p_1 - p_2}{p_1}\right)^{3.6} W^2 \left(\frac{t_1}{M}\right)^{1.2}\right] + 126.1$$

where:
p_1 = pressure upstream of valve (kPa absolute)
p_2 = pressure downstream of valve (kPa absolute)
W = mass flow rate (kg/s)
t_1 = upstream temperature (deg K)
M = molecular weight of gas

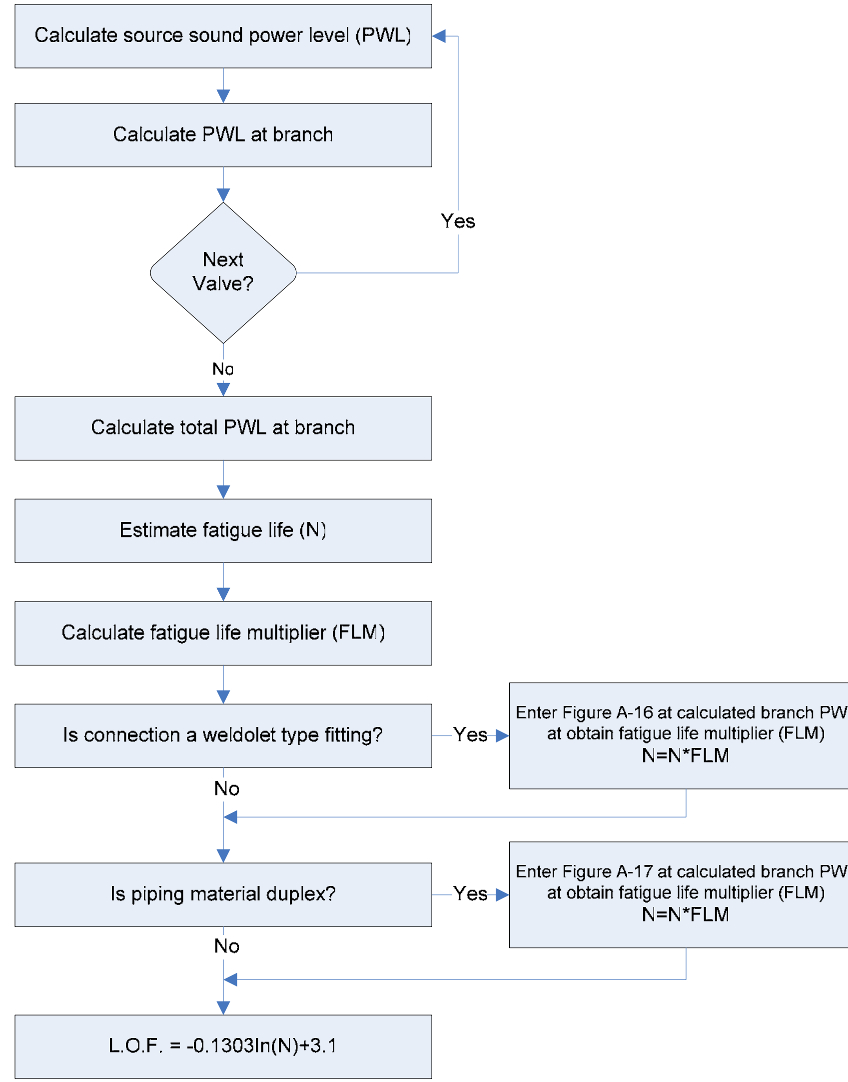
➢ Calculate PWL at branch

$$PWL(branch) = PWL(source) - 60*L/D_m$$

where:

L = distance downstream between source and branch (m)
D_{int} = internal diameter (mm) of the main line

Reference: MTD Guidelines, 1999.



Procedure for high frequency acoustic excitation assessment

- Calculate source sound power level (PWL)

$$PWL(branch,total) = 10\log_{10}\left[10^{\frac{PWL1(branch)}{10}} + 10^{\frac{PWL2(branch}{10}} + \ldots\right]$$

where:
PWL1 (branch) = PWL in the main pipe of valve 1 at the branch
PWL2 (branch) = PWL in the main pipe of valve 2 at the branch

- Estimate the fatigue life (N)

$$\log_{10} N = 470711.5155 - 63075.1242(\log_{10} B) + \frac{183685.4368}{\sqrt{B}} - \frac{575094.3273}{B^{0.1}}$$

$$B = a[PWL(branch) - 0.112762(s) - 0.001812(s)^2 + 0.00004307277(s)^3]$$

$$s = 91.9 - \frac{D}{T}$$

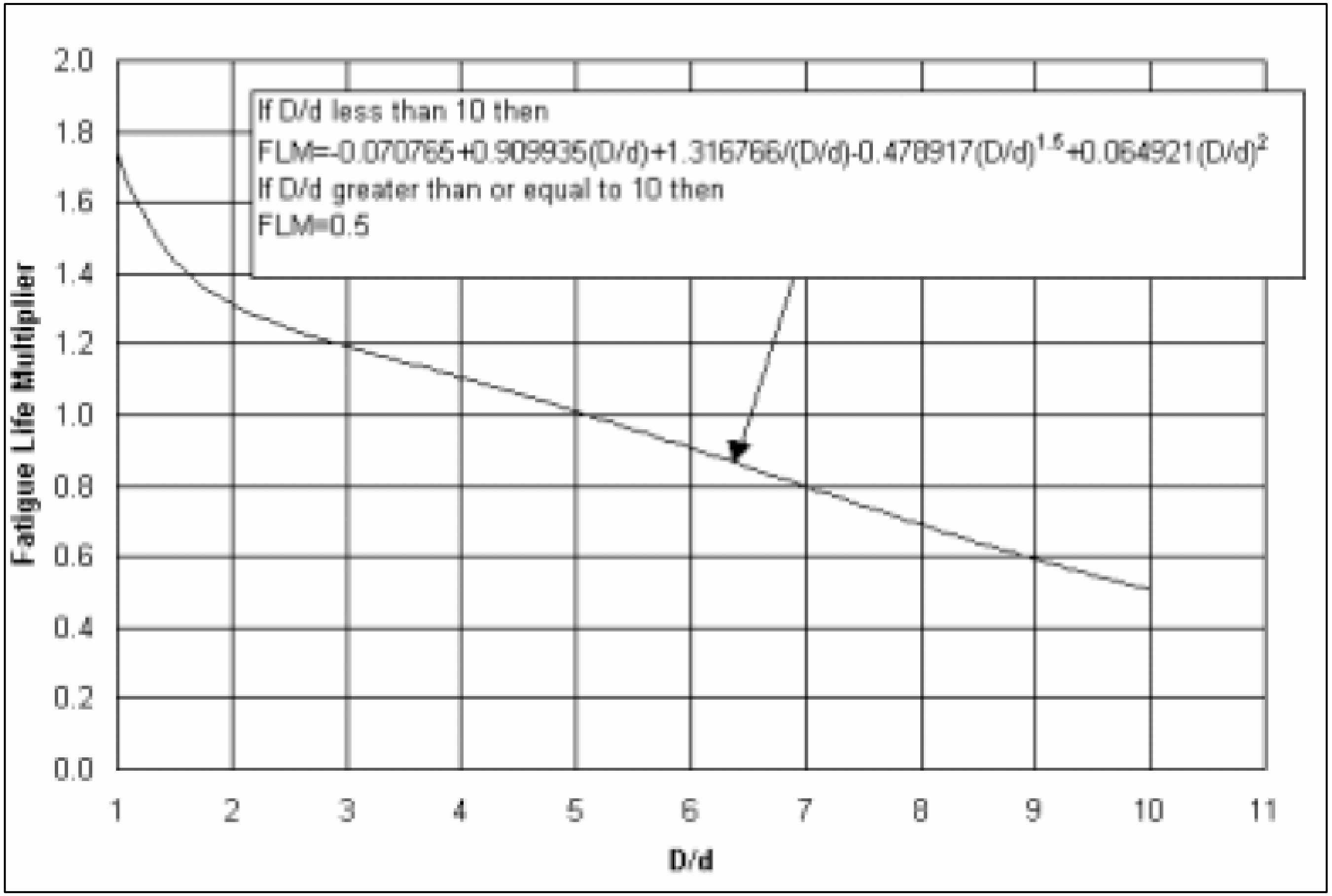
$$a = 3.28*10^{-7}\left(\frac{D}{T}\right)^3 - 8.503*10^{-5}\left(\frac{D}{T}\right)^2 + 7.063*10^{-3}\left(\frac{D}{T}\right) + 0.816$$

where:
D = external diameter (mm) of the main line
T = wall thickness (mm) of the main line

Reference: MTD Guidelines, 1999.

Procedure for high frequency acoustic excitation assessment

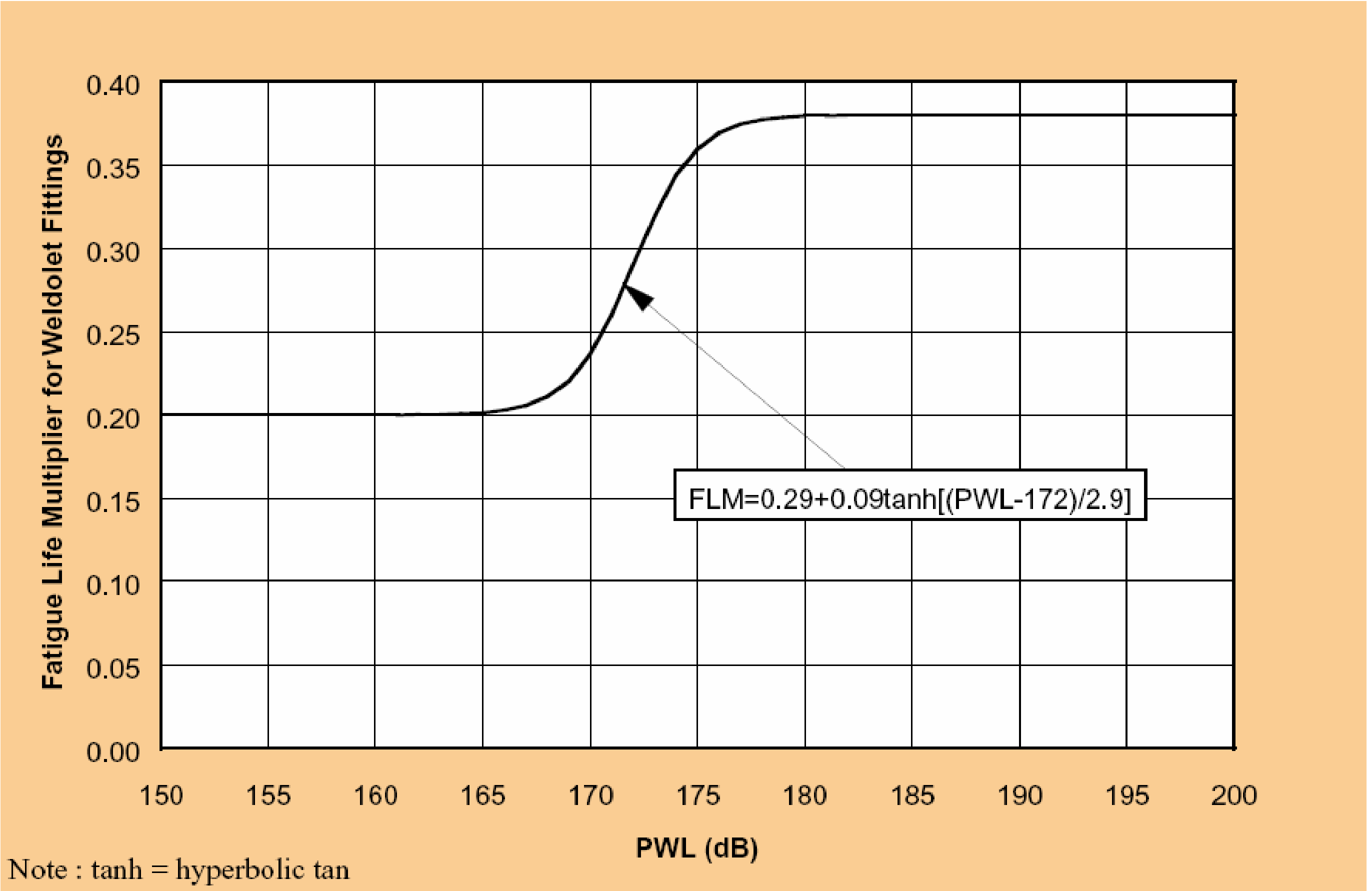
- Calculate the fatigue life multiplier (FLM), corrected for D/d



Reference: MTD Guidelines, 1999.

Procedure for high frequency acoustic excitation assessment

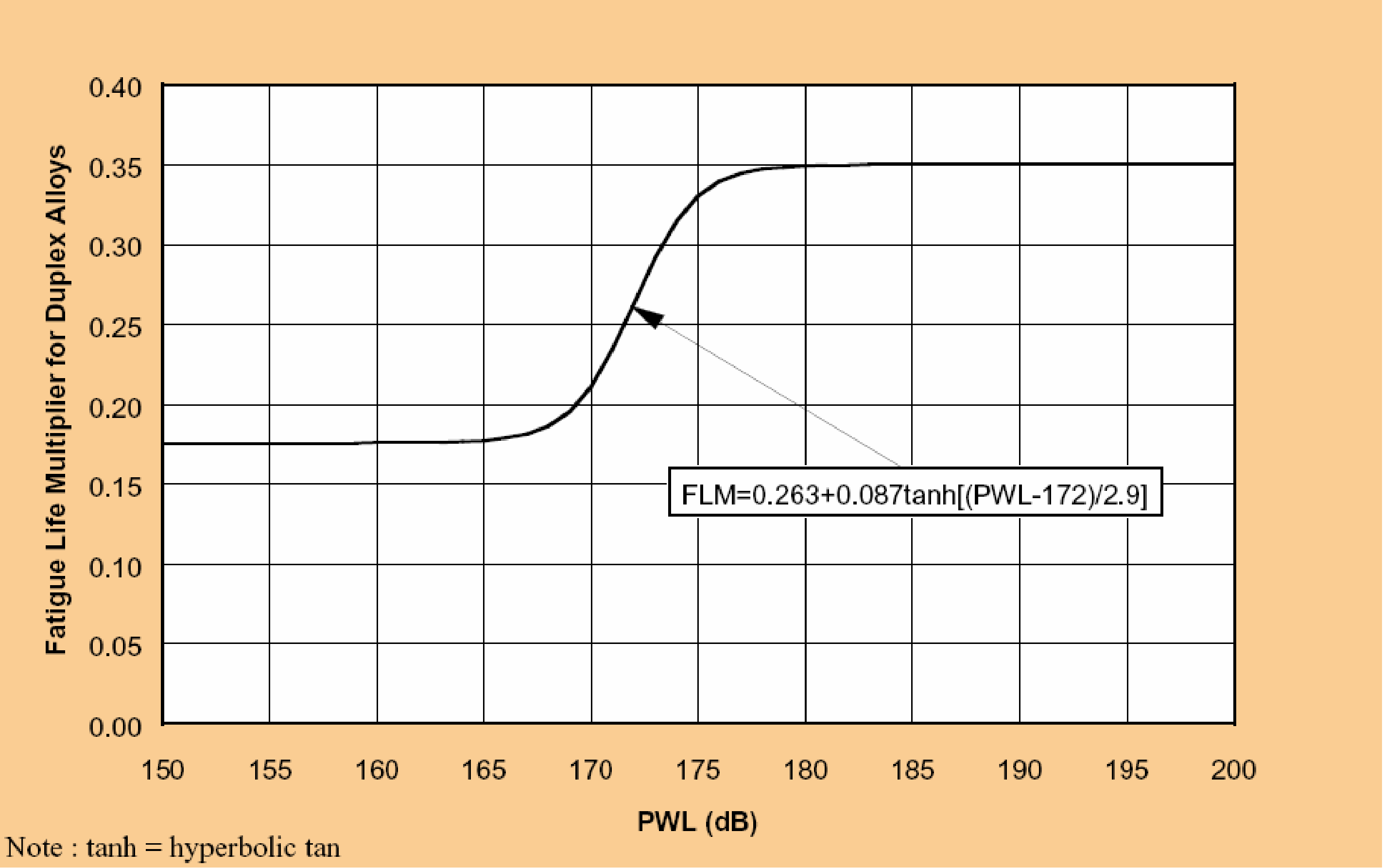
- Calculate fatigue life multiplier (FLM), corrected for weldolet connection



Reference: MTD Guidelines, 1999.

Procedure for high frequency acoustic excitation assessment

- Calculate fatigue life multiplier (FLM), corrected for duplex alloys piping material



Reference: MTD Guidelines, 1999.

Procedure for high frequency acoustic excitation assessment

- Calculate the LOF for high frequency acoustic excitation

LOF = -0.1303ln(N)+3.1

where:
N = number of cycles of failures, corrected for the D/d ratio, weldolet connections, and duplex alloys where applicable.

Reference: MTD Guidelines, 1999.

ioMosaic

Possible Design Solutions for Main Pipe

- For flow induced turbulence:
  - Decrease the flow velocity by increasing the diameter of the main pipe
  - Run a second pipe in parallel
  - Increase the pipe wall thickness
  - Minimize the number of bends in a system, the use of long radius bends, etc..
  - Stiffen the main line and its supporting structure

- For high frequency acoustic excitation:
  - Use of low noise trim, particularly for control valve. Uncertain success and durability.
  - Increasing the pipe wall thickness
  - Reduce the flow rate

ioMosaic

Possible Design Solutions for Small Bore Connections

- The fitting and overall unsupported length should be as short as possible
- The mass of valves and instrumentation should be minimized
- Any bracing supports should be from the main pipe, thus ensuring that the small bore connection moves with the main pipe

ioMosaic

Pipe and process data readily available from SuperChems

- SuperChems has piping, fittings, and stream properties readily available for assessing vibration induced fatigue
- The gas pipe summary shows a rating calculation for a 4P6 relief system set at 357 barg (~ 5200 psig)
- SuperChems provides appropriate warnings to flag potential vibration risks for relief piping

Gas Pipe Summary — Pipes Containing Vapor. SuperChems Expert, Sc=PSV-1001 - MIXTU.. — 9/3/2007 : 2:18 PM Page: 1

Piping layout — PSV-1001 LAYOUT
Liquid Density Method = Databanks
VLE/PVT Method = Equation of State

** INLET CONDITIONS		Ignore chemical reaction in piping segments
Inlet temperature. K	317.00	
Inlet pressure. barg	357.00	

** EXIT CONDITIONS		
Temperature. K	211.58	
Pressure. barg	33.86	
Surroundings/back pressure. barg	5.00	
Barometric pressure at site elevation. barg	0.00	DEFAULT site
Speed of sound. m/s	247.15	
Mach Number	0.96	** WARNING: Mach Number is > 0.7
rho*u*u in kg/m/s2 or in Pascals	4526179.53	** WARNING: Detailed estimates of vibration and line supports should be performed; > 100000 kg/m/s2
Reynold's Number	275706083.59	
Fraction of piping layout solved	1.00	
Mass flow rate. kg/s	353.86	
SCFH	52882784.61	

Component	Mw	Input	Output
METHANE	16.04	201.66	201.66
ETHANE	30.07	152.13	152.13

Relief device found at segment 2	PSV-1001: GENERIC API VALVE. Other. Service= Gas
Last iteration Kb correction	1.00
Pressure at device inlet. barg	356.47
% inlet pressure drop relative to effective set point	0.15
% irreversible inlet pressure drop relative to effective set point	0.14
% back pressure relative to effective set point	12.08

Segment #, Type, Name	Start Elevation. m	End Elevation. m	Length. m	Exit Diameter. m	Exit/Flow Area. m2	Inlet Temperature. K	Inlet Pressure. barg	Irreversible dP. bara	Total dP. bara
1, Piping Segment, PSV-1001 - INLET	0.0000	0.1000	0.1000	0.1023	0.0082	317.0000	357.0000	0.4905	0.5336
2, Pressure Relief Valve, (orifice) PSV-1001	0.1000	0.6048	0.5048	0.0724	0.0041	316.9782	356.4664	235.0765	235.0765
3, Piping Segment, PSV-1001 - OUTLET	0.6048	0.6048	1.0000	0.1541	0.0186	225.6561	43.1258	1.9560	9.2703

** NOTE : Backpressure calculation is enabled

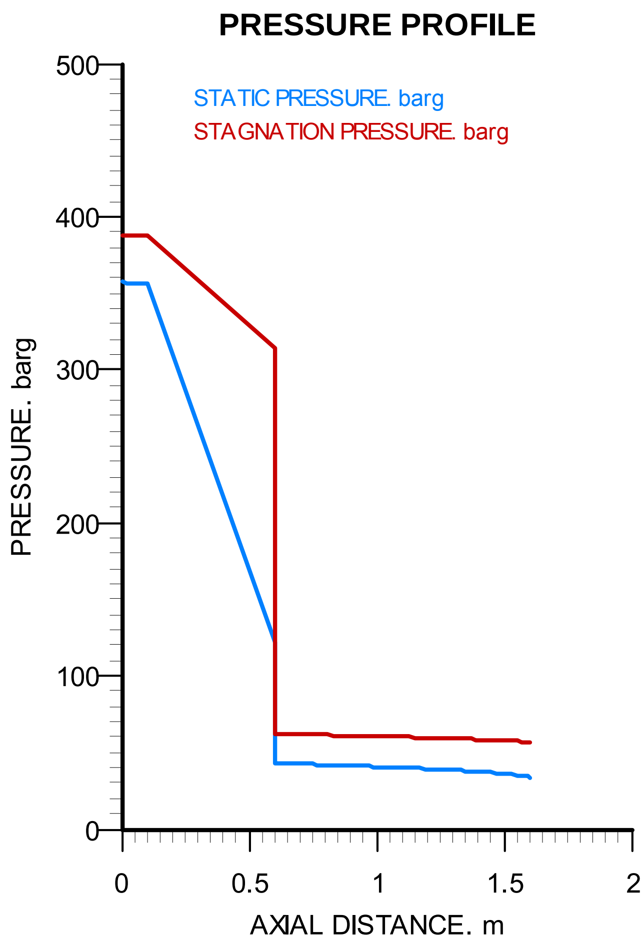
Last Specified: 02:00:09 PM, Fri Aug 31 2007
Last Executed: 02:16:05 PM, Mon Sep 03 2007

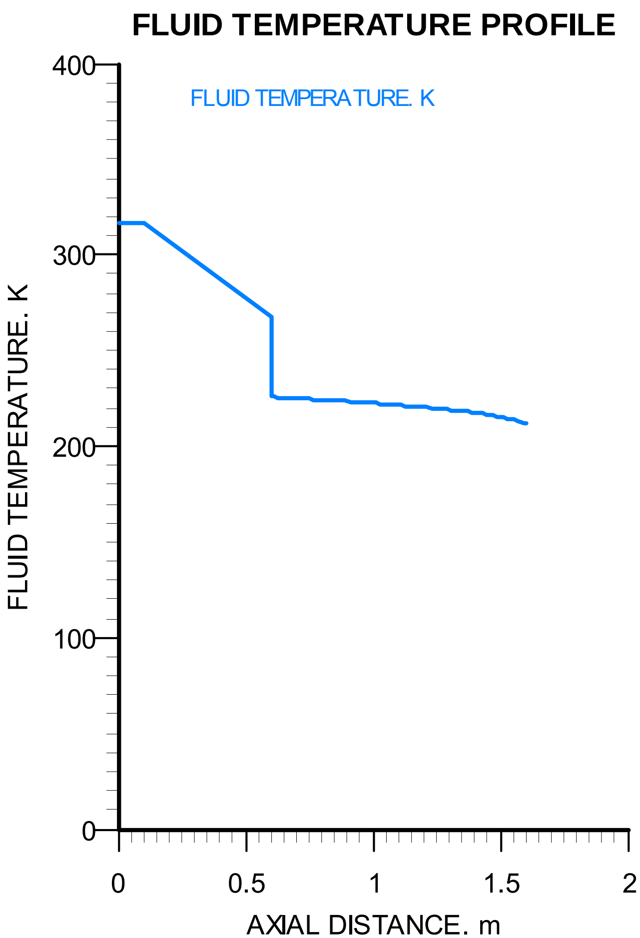
SuperChems Expert v5.9.6 (2-23-2007) — (C) ioMosaic Corporation — .. CEDAR BAYOU PLANTVIBRATION.COR

ioMosaic

Pipe and process data readily available from SuperChems

- Temperature/pressure can be extracted at anywhere along the axial distance along the relief system

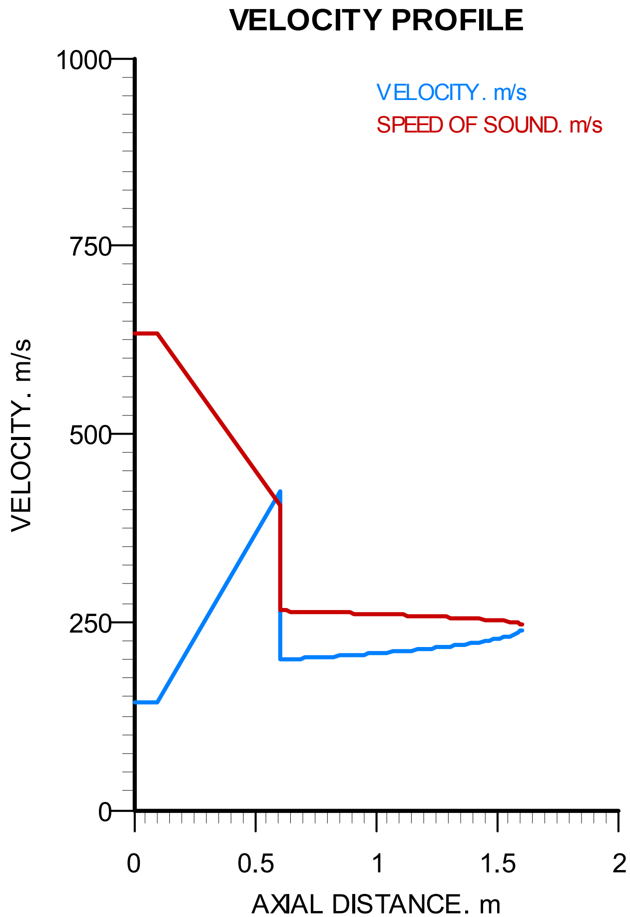


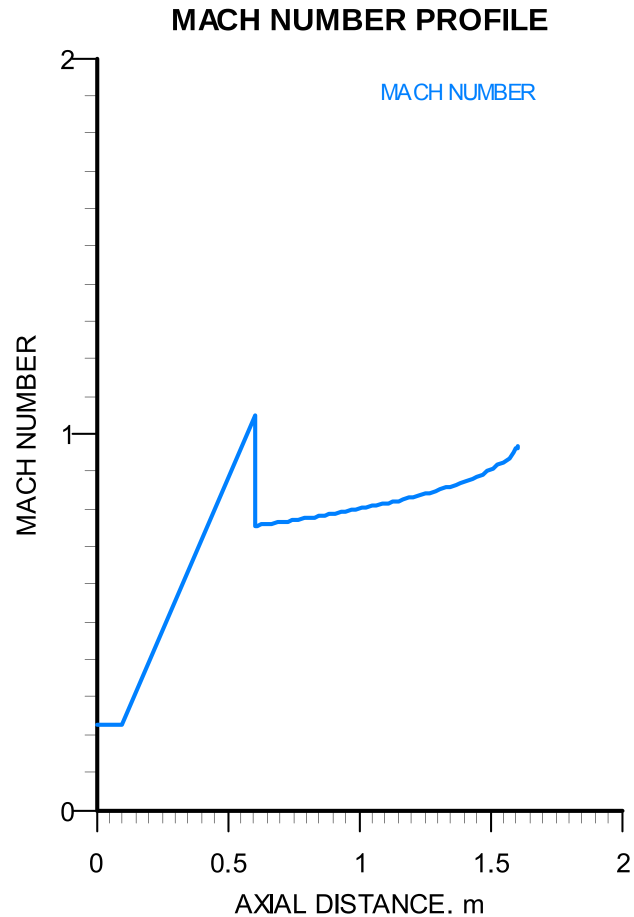


ioMosaic

Pipe and process data readily available from SuperChems

- Indication of choked points

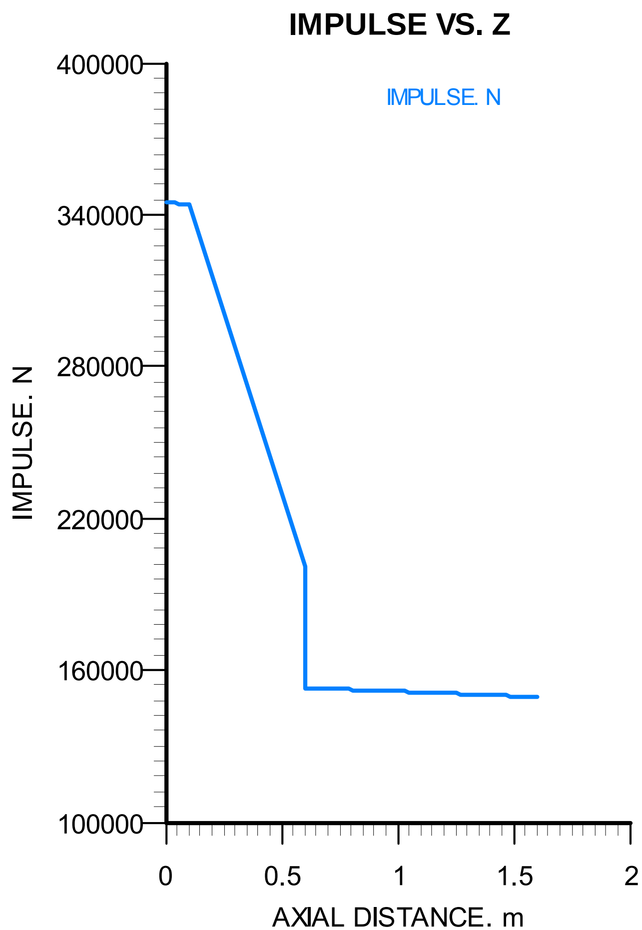


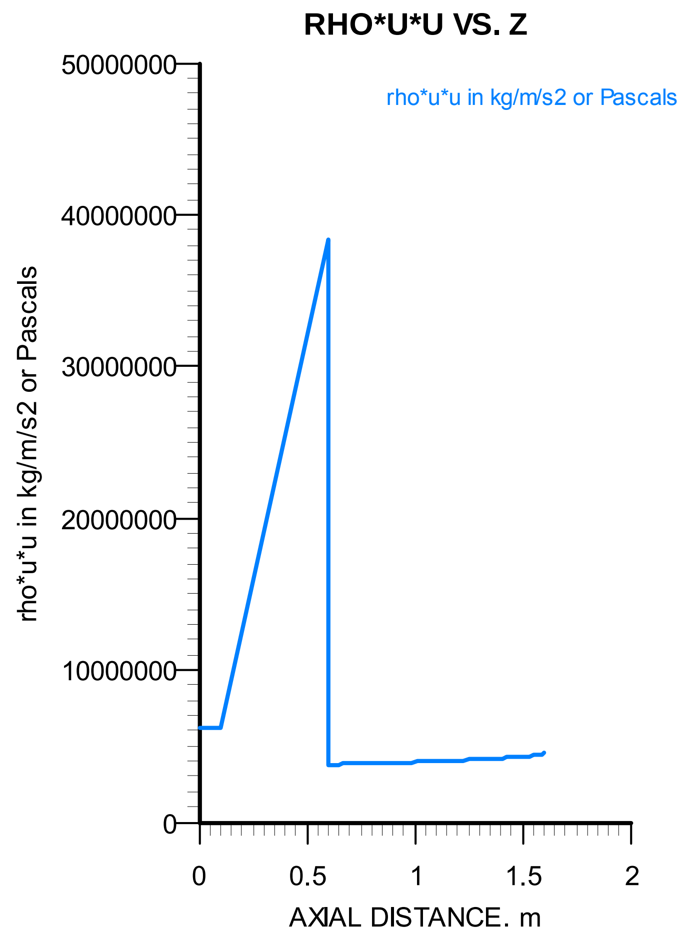


ioMosaic

Pipe and process data readily available from SuperChems

- Indications of reaction force and potential vibration

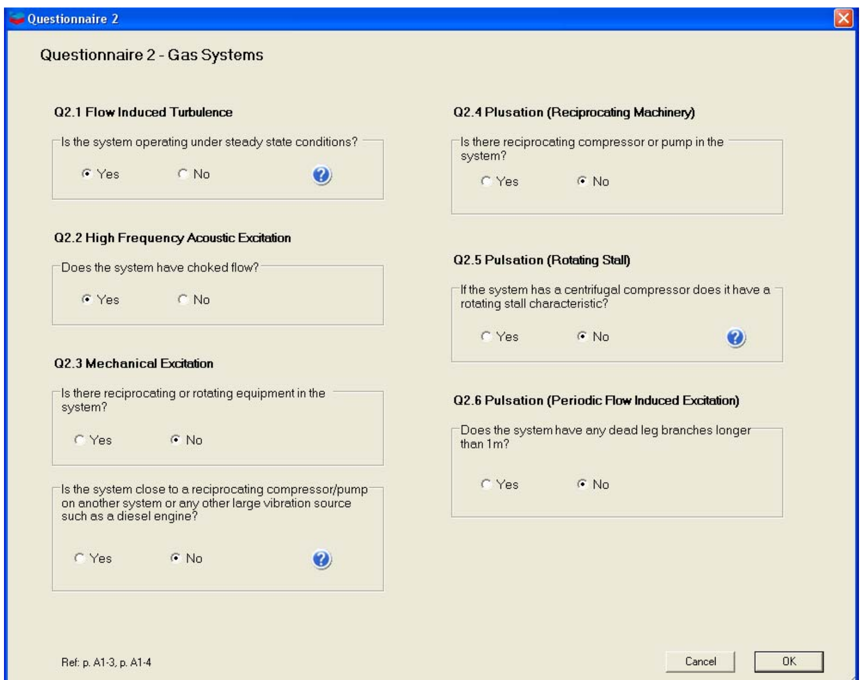


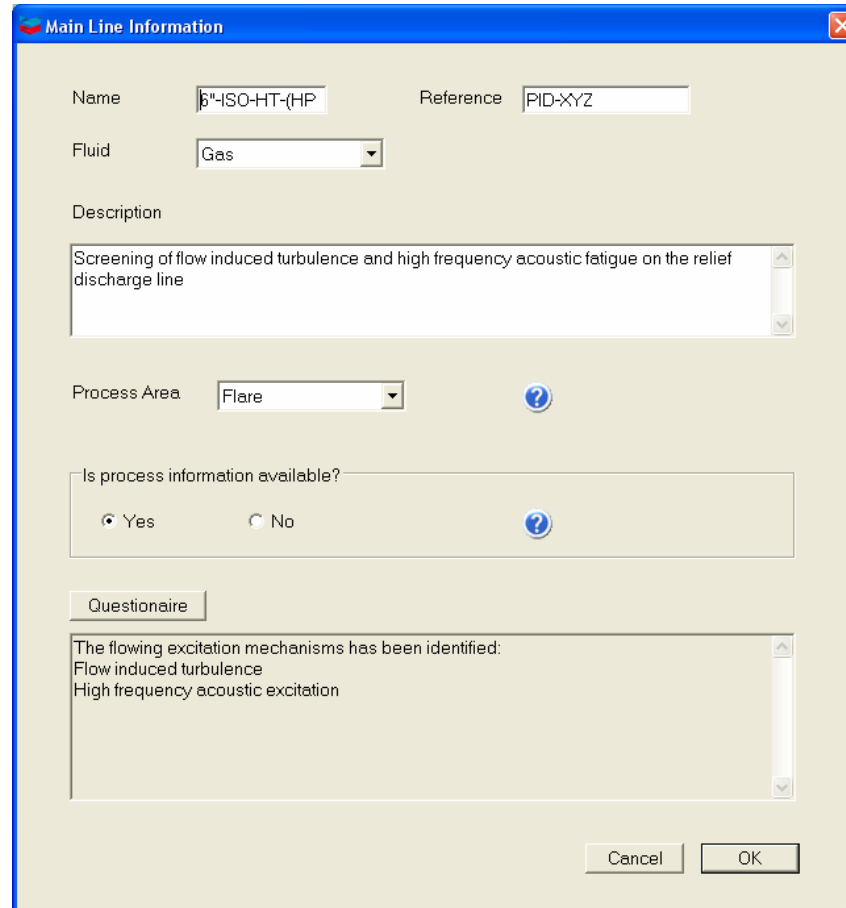


Piping Vibration Evaluation Tool (PVET)

ioMosaic has also developed general piping vibration risk evaluation tools for a large energy company

- PVET is based on the MTD's Guidelines
- Currently in the beta testing phase



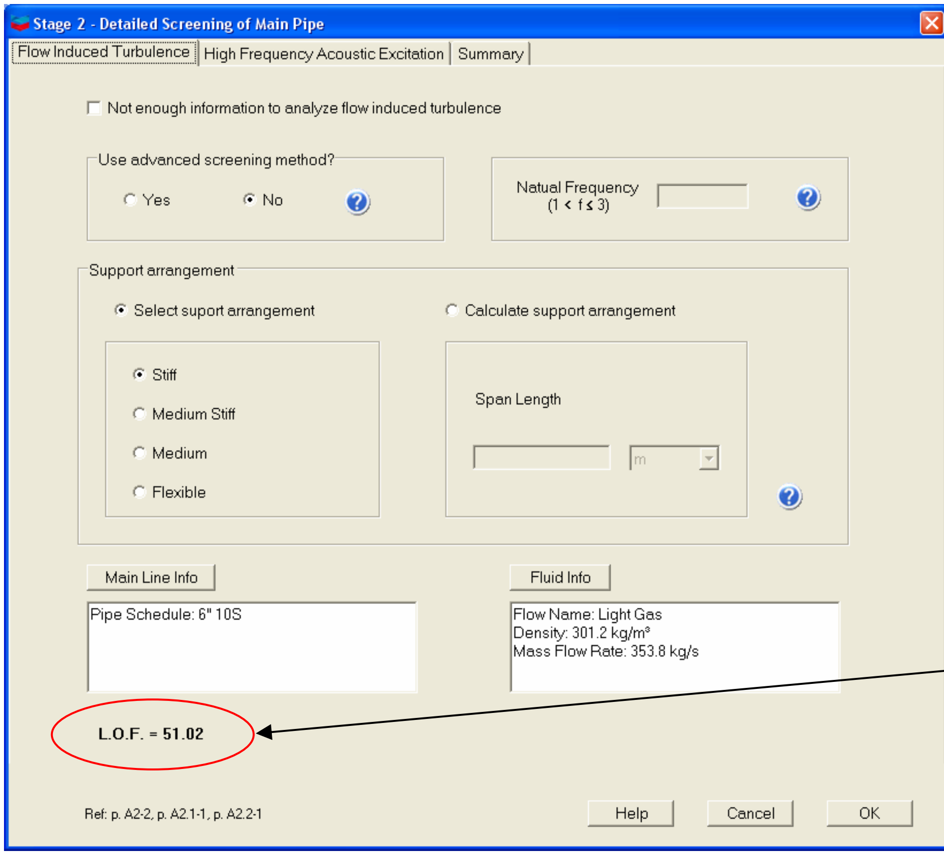


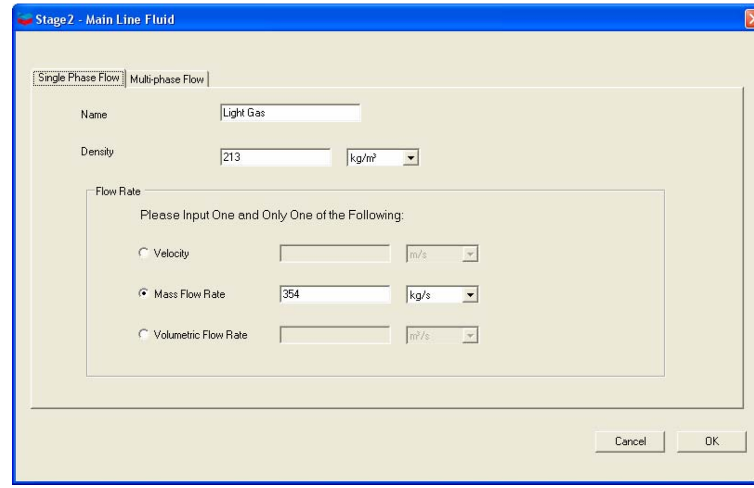
Sample of data entries for the 4P6 relief system set at 357 barg (~ 5200 psig)

ioMosaic

Piping Vibration Evaluation Tool (PVET)

- Data entries to determine LOF for flow induced turbulence



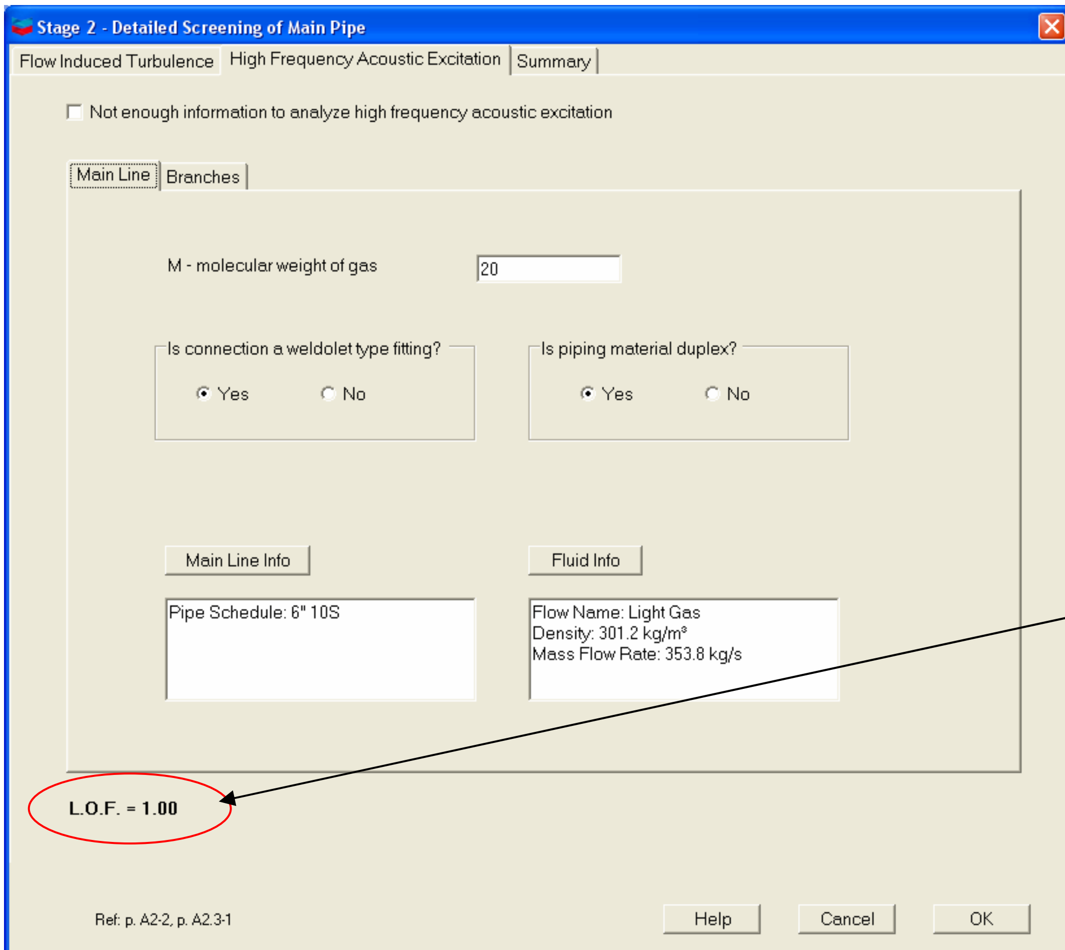


- Calculated likelihood of failure, LOF => 1 (classified as intolerable)
- Recommended actions:
  - ❑ Main pipe must be redesigned or re-supported
  - ❑ Consult Section 4.2 for design solutions
  - ❑ SBCs should be assessed per Stage 3
- Confirms that piping has flow induced turbulence concerns as flagged by SC

ioMosaic

Piping Vibration Evaluation Tool (PVET)

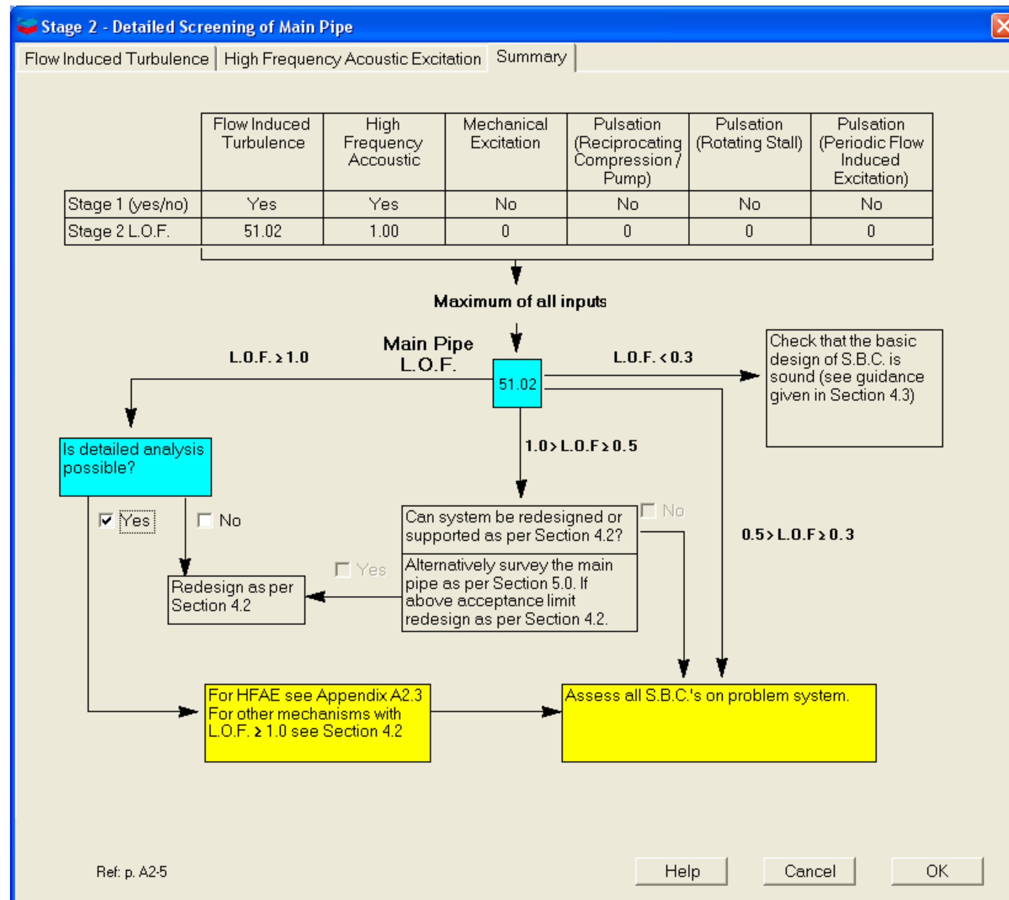
- Data entries to determine LOF for high frequency acoustic excitation using 1” weldolet downstream of the valve



- Calculated likelihood of failure, LOF => 1 (classified as intolerable)
- Recommended actions:
  - Main pipe must be redesigned or re-supported
  - Consult Section 4.2 for design solutions
  - SBCs should be assessed per Stage 3
- LOF estimated from high frequency acoustic excitation confirms that the main pipe has serious piping fatigue concerns

Piping Vibration Evaluation Tool (PVET)

- Summary of the main pipe findings
- The highest individual mechanism's estimated value of LOF will be the main pipe's LOF
- For design and redesign guidelines, consult Section 4.2 (Design Solutions for Pipe Main)
- Assess all SBCs on the indicated main pipe



ioMosaic

Summary

- MTD Guidelines provides a comprehensive methodology for screening vibration induced fatigue in process and relief piping systems
- SuperChems Expert offers state of the art computational flow models for single and multiphase reacting flows
- Flow models provide detailed estimates of kinetic energy and sound pressure/power levels required for the assessment of the risk of vibration induced failure of the relief piping
- The models can be coupled with MTD Guidelines to provide the user with a clear indication of potential vibration risks
- Screening results can be also used to develop mitigation measures or used as inputs to a more detailed piping structural analysis
- For high pressure piping system, design engineer should evaluate the risk of vibration induced fatigue

ioMosaic

About ioMosaic Corporation

Founded by former Arthur D. Little Inc. executives and senior staff, ioMosaic Corporation is the leading provider of safety and risk management consulting services. ioMosaic has offices in Salem, New Hampshire and Houston, Texas, and Minneapolis, Minnesota.

Since the early 1970's, ioMosaic senior staff and consultants have conducted many landmark studies including an audit of the Trans-Alaska pipeline brought about by congressional whistle blowers, investigation of the Bhopal disaster, and the safety of CNG powered vehicles in tunnels. Our senior staff and consultants have authored more than ten industry guidelines and effective practices for managing process safety and chemical reactivity and are recognized industry experts in LNG facility and transportation safety.

ioMosaic Corporation is also the leading provider of pressure relief systems design services and solutions. Its pressure relief system applications are used by over 250 users at the world's largest operating companies. It holds key leadership positions in the process industries' most influential and active pressure relief system design, and chemical reactivity forums, and plays a pivotal role in defining relief system design, selection, and management best practices.

SALEM OFFICE
93 Stiles Road
Salem, New Hampshire 03079
Tel: 603-893-7009
Fax:603-893-7885
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

HOUSTON OFFICE
2650 Fountain View, Suite 410
Houston, Texas 77057
Tel: 713-490-5220
Fax:713-490-5222
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

MINNEAPOLIS OFFICE
333 Washington Avenue North
Minneapolis, Minnesota 55401
Tel: 612-373-7037
Fax:832-553-7283
Email: trainingondemand@iomosaic.com
Web: www.iomosaic.com

Member Companies | DiERS

Abbott
ABS Consulting
Air Products and Chemicals
Aker Process Systems
AKZO Nobel
Albemarle
Alliant Techsystems
AMEC
ARKEMA
Ashland
AspenTech
Aspen Technologies
AUXITEC Industrie
Basell Polyolefins
BASF
Bayer
Baykey
Bechtel
BioPharma Engineering
Blackhawk of Louisiana
Boulder Scientific
British Petroleum (BP)
BPE Design & Support
BS & B Safety Systems
Cantwell Keogh & Associates
Catalonia Academic Developments Center
CB&I Lummus
Celanese
Centaurus Technology
Central Leather Research Institute
Cheminform Saint Petersburg (CISP)
Chemstations
Chemtura
Chevron Phillips Chemical Company
Chilworth Technology
CH2MHill Lockwood Greene
Chiyoda
CITGO
Compliance Engineering
ConocoPhillips
Consilab Gesellshaft fur Anlagensicherheit
Continental Disc
Covidien
Cray Valley
CRB Consulting Engineers
CTCI Corporation
CWFC-Farris Engineering
CYTEC Industries
Daelim Industrial Company
Day and Zimmermann International
Digital Solutions Technology
DNV Software
Doris Engineering
Dorlastan Fibers
Dow
DSM
DuPont
Eastman Chemical
Eastman Kodak
Eli Lilly
ENGLOBAL
EPCON International
Evonik Degussa
ExxonMobil Research & Engineering
EZ Relief Systems Consulting
Farris Engineering Services
Fauske & Associates
FB Engineering
Fike Corporation
Fike Europe
FisherInc.
Flint Hills Resources
Fluor
FM Global
FMC
Foster Wheeler Energy
Fukui Seisakusho
GE
General Physics
Georgia-Pacific
GL Nobel Denton
Golder Associates
Goodyear
Groth
Hargrove Engineers & Constructors
Hatch Associates Consultants
Health and Safety Executive
Henkel Corporation
Hexion Specialty Chemical
Holley Refining & Marketing
Honeywell International
Huff Consulting Services
ICA Fluor Daniel
INEOS
Inglenook Engineering
INGEMO Impianti
INNOVENE
Institute for Chemical Physics Research
Institute for the Protection and Security of the Citizen (IPSC – JRC)
Institute of Occupational Safety and Health
Institute of Safety Research
Instituto Mexicano del Petroleo
International Flavors & Fragrances, Inc.
Intertek Expert Services
Invensys Process Systems
ioMosaic
ioKinetic
Jacobs Engineering Group
Javan & Walter
KBC Advanced Technologies
KBR
Kemira Pulp and Paper Chemicals
KH Engineering
Kinetica
Kraton Polymers
Lanxess
LESER
LeungInc.
Lloyd's Register
LyondellBasell
Mark V
Marranca Engineering
Matrix Laboratories
MeadWestvaco
Merck
3M
Monsanto
Moorgate Consulting
M. P. Kenes
Muller & Associates
National Casein
Netzsch Instruments
NOVA Chemicals
Occidental Chemical
Orbit Controls
Orica Adhesives & Resins
OSECO,
PENTAIR Valves & Controls
Petromech Projects
Pfizer
Polimeri Europa
PPG Industries
Process Plus
Process Safety & Design
Process Safety, Simulation, Synthesis by Sheppard, LLC (PS4)
PROCHEM Engineering
Procter & Gamble
Project Management Limited
PROSAF
Protego
PSRE
Rawlings Mechanical
REC Silicon
Reliance Industries
REMBE
REMBRE, GmbH Safety And Controls
Resolution Performance Products
Rhodia
Roche
Russian Scientific Center Applied Chemistry
Samuel Engineering
S and B Engineers & Contractors
Sanofi-Aventia Deutschland
Saudi Aramco
Scientific Process Development Services, Ltd
SembCorp Simon-Carves
Setaram
Sasum Pharma Solutions
Shell
Siegfried
Siemens
Silver Eagle Refining
Smith & Burgess
SNC-Lavalin
Societe Artesienne de Vinyle
Solutia
SSOE
Stazione Sperimentale per i Combustibili
Stepan Chemical
Stockhausen
G.R. Stucker & Associates
Sumitomo
SWISSI
Syngenta
Technip
TG Engineering
TNO
Toyo Engineering
Transeparation
TriTex Technology
UNWIN
URS
Veolia Environmental Services
Wacker Chemie
WorleyParsons