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Mechanical, Stress and Flow Considerations for Piping Design of Centrifugal Compressors

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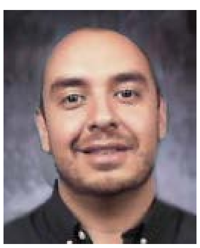

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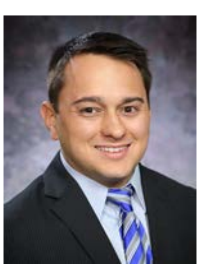

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ABSTRACT

This tutorial covers a range of factors that must be considered in the piping design associated with the installation of any new centrifugal compressor system. Multiple factors must be balanced in the piping design to have an overall successful final installation. The compressor piping must be configured and supported in a manner to safety contain the mechanical forces from the internal fluid pressure as well as the weight of the piping, fittings and valves. Additionally, the piping must not place any unusually high loads on the compressor itself or any piping supports due to thermal expansion, pressure elongation or weight loads. Finally, the piping layout should result in an even flow velocity profile that does not result in any detrimental impact to the aerodynamic performance of the centrifugal compressor.

This tutorial will cover good general design practice as well as some on-going research efforts related to optimizing loads and flow profiles. There is active research examining what design factors (such as piping diameter, piping layout, temperatures, piping offsets in each direction, etc.) have most impact on the loads acting on the compressor flanges. Understanding the key factors can help develop a better initial design and minimize any potential redesign costs.

Additionally, there is a separate on-going experimental testing effort evaluating the impact of piping design on the aerodynamic performance of a centrifugal compressor. This effort is also evaluating if historically used guidelines and rules of thumb for piping layout are still valid. The uniformity of the velocity profile and amount of swirl are being quantified by experimental and CFD techniques for different piping layouts and recommendations will ultimately be made to accept or improve current accepted practices.

This tutorial is of interest to professionals involved in the design of piping systems that require centrifugal compressors such as compression stations, hydrocarbon processing facilities, and research and development facilities as it will show which assumptions still yield acceptable results and which need to be updated.

INTRODUCTION

The design of a piping system for a new centrifugal compressor installation involves many different considerations and decisions. Some requirements are obvious and necessary to meet certain industry codes or safety standards. Other factors might not be as clear and could involve trade-offs between competing objectives with no obvious “better” design.

Although centrifugal compressors can be used in a wide variety of applications, this paper will focus on the design of the main process gas piping between the compressor and the headers for larger horsepower centrifugal compressors commonly used in the refinery or natural gas transmission industries and piping designed per the ASME B31.3 or B31.8 codes. However, the lessons presented here can be applied to a variety of applications. This paper is divided into three main sections. First is general piping design which discusses the classical considerations, second is optimizing thermal expansion loads on the a centrifugal compressor and third is flow distortion and aerodynamic performance.

GENERAL PIPING DESIGN

Basic Function, Layout and Components

A centrifugal compressor uses mechanical energy to create a pressure rise in a continuous flow of a process gas by first adding kinetic energy (velocity) through a rotating impeller. A pressure rise is then created by slowing the flow through a diffuser. Once the technical and commercial need for the installation of a new compressor has been established, and the new type compressor selected, one of the first activities is to determine the basic layout and components needed in the installation. This is the type of information that would typically appear on a Process Flow Diagram (PFD) or a Piping and Instrumentation Diagram (P&ID).

A centrifugal compressor will be installed at a facility with many other components including filters/separators/scrubbers, coolers, flow meters, pressure regulators, block and check valves, blowdown systems, recycle/bypass/anti-surge systems, and the piping and supports necessary to tie it all together. A typical compressor piping layout near the unit is shown in Figure 1.

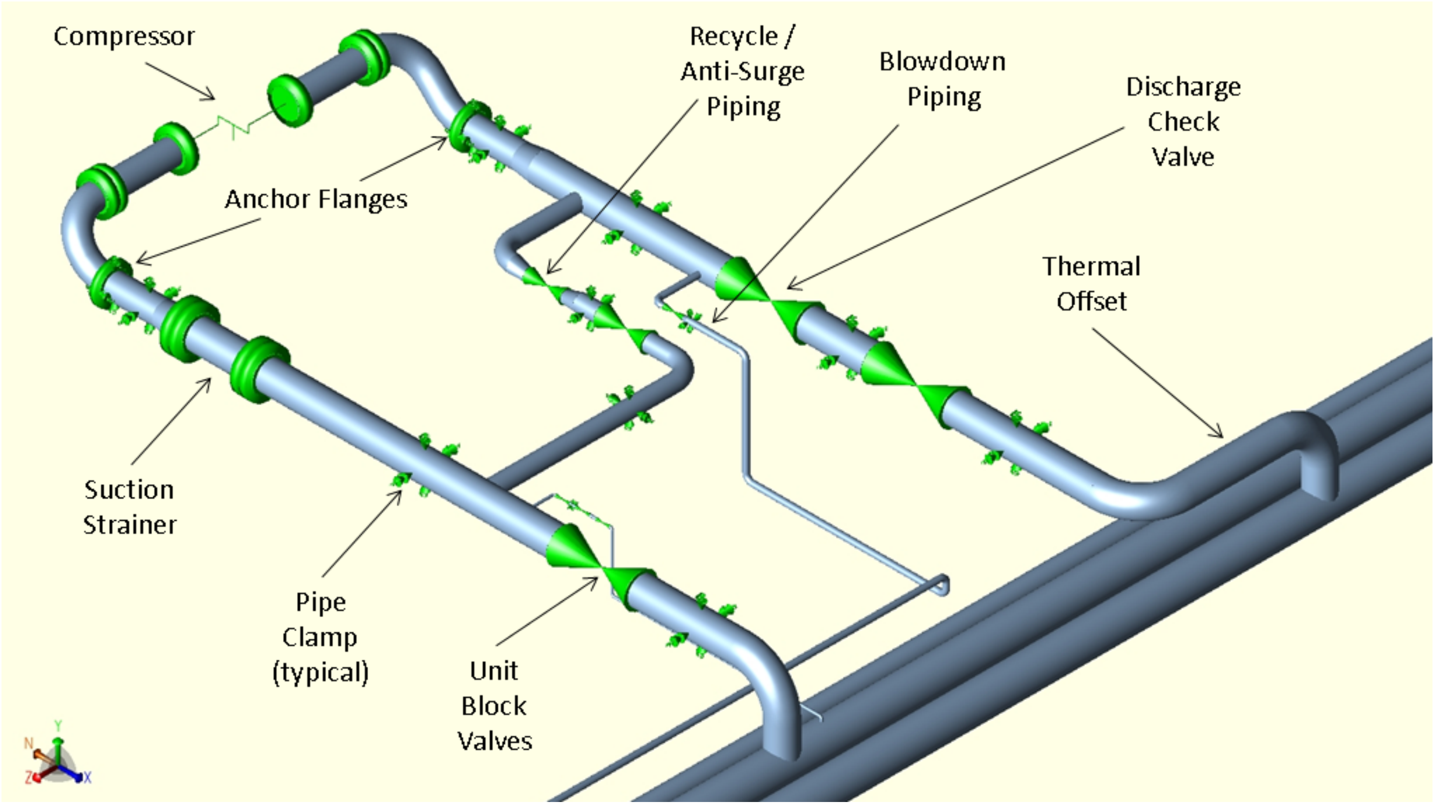


Figure 1. Typical Centrifugal Compressor System Piping Layout

Basic Pipe Sizing

The most important factors in sizing the piping system are the flow rates and the maximum design pressures. The necessary diameter of a piping segment will be based on the design flow rates, or more precisely, the maximum flow velocities of the gas in the piping. Different areas of the system will have different diameters depending on how the flow is split among a single or multiple compressors operating in parallel.

Piping diameters are selected to keep flow velocities within a preferred range. For typical natural gas pipeline pressures (≈500 to 1500 psig), flow velocities in the main piping should typically be kept between 20 and 65 ft/sec. Excessively low flow velocities could result in concerns related to liquid or settlement buildup in low points, improper check valve operation and higher installation costs. Excessively high flow velocities could result in concerns related to pressure drop, noise, flow induced pulsation, flow turbulence induced vibration, erosion, etc. Higher flow velocities up to approximately 120 ft/sec may be permissible in local piping runs subject to infrequent use such as recycle/anti-surge piping or blow down piping. Higher velocities may also be permissible in systems with lower pressures or lower fluid densities.

The most critical design load for a pipeline is internal pressure. Once a desired Maximum Allowable Operating Pressure or MAOP is determined, this will impact the required wall thickness and material selection for the piping. The hoop stress (Pressure * Diameter / (2 * wall thickness)) is typically limited to between 40 and 72% of the yield strength of the material (typically 35,000 - 60,000 psi). There may be considerations related to availability, economics, material temperature limits, material suitability for certain applications, corrosion, etc. that influence trade-offs between material grade and required minimum pipe wall thickness. Generally, it is more economical to use higher grade materials (with corresponding thinner wall thicknesses) as the piping diameters get larger.

Other system components, such as flanges, valves, coolers, vessels, etc. will also all have to be rated to the system MAOP or higher. The system will be limited by the weakest link.

Codes and Standards

There are a wide range of industry codes and standards that could potentially apply to the installation of a new centrifugal compressor system. In general, these codes and standards exist to provide design guidelines, requirements and definitions. Codes can serve to improve safety, reliability or performance. They can also serve to standardize parts or design to reduce costs, improve interchangeability or to serve as a purchasing specification.

For centrifugal compressors, the API 617 Standard covers the compressor itself, but not the piping system. There are many different piping design codes available depending on the application and geographical location of the installation. The purpose of these codes is to ensure the integrity of a piping system for the design conditions (which typically operate at elevated pressures and temperatures), including basic piping design, material selection, standard dimensions, fitting selection, stress analysis, fabrication, inspection and testing. Some companies also have their own internal standards.

In the United States, most industrial piping systems are designed based on one of several ANSI/ASME Standards such as B31.3 for Process Piping or B31.8 for Gas Transmission and Distribution Piping Systems. However, in general, these types of codes only provide very limited guidance on many of the topics discussed in this paper.

Weight, Pressure and Thermal Stress

Most compressor units have elevated discharge gas temperatures inherent to the compression process. These elevated gas temperatures cause thermal expansion of the piping, and as a result, loads and static stresses are generated. A computer model of the compressor and piping system is typically developed to calculate static stresses due to thermal expansion, pressure and weight loads and to comply with the various piping design codes such as ASME B31.3, B31.4, B31.8, Z662, etc.

A proper thermal analysis should review and ensure a number of parameters are acceptable for operation. These include the stress in the piping, compressor nozzle loads, cooler nozzle loads, and pipe support reaction loads. Additionally, all mechanical natural frequencies should be identified and designed to reduce the likelihood of vibration problems. An ideal thermal system is flexible to allow for thermal expansion without excessive stress. This is in conflict with the goals of the mechanical analysis. Therefore, concurrent engineering with the mechanical analysis is necessary to balance trade-offs.

There is no absolute criteria to when a thermal stress analysis needs to be performed, but they are typically done when:

- Operating temperatures above 130 °F (54 °C)
- Large diameter piping (12” and above)
- Design not similar to past installations
- There are critical equipment load allowables (cooler/compressors/etc.)
- Critical applications (offshore, etc.)
- When required by piping code or project specifications.

Codes such as ASME B31.8 often state that a formal flexibility analysis is not required if the system duplicates without significant change a system operating with a successful record or can be readily judged adequate by comparison with previously analyzed systems.

The amount of thermal expansion a straight segment of piping will experience is simply the product of multiplying the pipe length times the temperature differential times the thermal expansion coefficient of the material. Therefore, Growth = Expansion Coefficient * Length * ΔT. For a 100’ long carbon steel pipe installed at 70 °F and operating at 180 °F, the growth = 6.3e-6 in/(in °F) * (100 ft * 12 in/ft) * (180 °F -70 °F) = 0.832 inches.

The analysis should consider a number of operating conditions, with typical cases including:

- Maximum design temperatures
- Normal operating temperatures
- Maximum differential temperatures
- Minimum design temperatures (cold ambient)
- Recycle conditions
- One unit on, another unit off
- Ambient (installation) temperature
- Pressure elongation effects (Bourdon)

A number of load cases, commonly including the cases below, should also be considered:

- Operating Load Cases (Weight + Pressure + Temp.)
- Primarily for Nozzle and Restraint Loads
- Sustained Load Cases (Weight + Pressure)
- Expansion Load Cases (Temp. Only) – OPE-SUS "self-limiting loads" (Primarily for Pipe Stress)
- Total Temperature (Displacement) Range Expansion Load Case
- Occasional Loads (Wind, Seismic)
- Hot Sustained Load Case (restraint lift off)

The solution to a high load or high stress condition due to thermal expansion depends on the specifics of the problem, but some common solutions often include modifications such as changing restraint types, locations or quantity, or re-routing the piping to add an offset or expansion loop. Forty five degree elbows can be replaced with 90 degree elbows or tee types (where high stress intensification factors can exist) can be changed or enlarged.

Piping Supports

The supports or restraints installed on a piping system serve several different purposes and include many different types. First, it is necessary to support the weight of the piping system to prevent excessive sagging and stress. Secondly, it is necessary to locate special guides, stops or anchors at certain locations to direct the thermal expansion of the system to force growth (and resulting loads) away from load sensitive components. Thirdly, the system must have sufficient mechanical restraint to prevent excessive vibration. There are a wide variety of restraint types with which exist (such as simple weight supports, guides, stops, hold down clamps, U-bolts, spring cans, hangers, etc.) to achieve these objectives.

For example, special axial pipe stops or anchor flanges are often installed very near the compressor (as shown in Figure 1) to reduce loads generated by thermal expansion of the long suction and discharge piping laterals from placing excessively high loads on the flanges of the centrifugal compressor. This is discussed further in the later section of this paper on "Optimizing Thermal Loads".

A piping system designed exclusively for vibration control would include many rigid hold down clamps and anchors. However, this is competing with the needs of thermal expansion which require the system to have flexibility and to expand freely to reduce thermal induced stresses and loads. Support types such as spring cans, hangers and simple weight supports provide much less vibration control than more rigid hold down clamps. One objective of performing a thermal stress analysis is to develop a balanced system to simultaneously satisfy these competing requirements. The analysis will evaluate necessary support types and locations needed for weight support, vibration control and load reduction.

The balance between thermal expansion and vibration control can be achieved by careful selection of support types and locations. For example, the use of hold down clamps with low friction liner material and slotted bolt holes will allow the thermal expansion of the piping if sufficient force is generated for the load on the restraint to overcome the friction. In general, dynamic loads are much lower in amplitude than static loads (often by an order of magnitude) and will not have sufficient force to overcome friction, thus resulting in the restraint being "active" or effective for vibration control. Locating restraints on one side of an elbow or on the other side can also help balance these competing requirements in some cases.

Also see later discussion on this topic in the "Flow Induced Turbulence (FIT) and Piping Vibration" section of this paper.

As with any model, the accuracy of the results is a function of the accuracy of the input assumptions. Modeling restraints stiffnesses as rigid may be conservative in some cases (such as for calculating static stresses in localized regions), but is also likely to be non-conservative when considering the results of a modal analysis or if a high stiffness restraint is expected to limit loads acting on an equipment nozzle. In reality, all supports have finite stiffness values, which should be considered in the analysis (at least estimated to a rough order of magnitude) to obtain more realistic results. Calculated loads acting on the supports is another output of the thermal stress analysis that will be influenced by the assumed support stiffness values. Even with very small displacements, assuming rigid restraint stiffnesses will result in high calculated loads. These calculated support loads can be used for foundation design and in evaluating any local support stresses.

Flexibility of Piping Elbows

Oftentimes, the type of elbow used is determined by the station layout and the desired accessibility of certain areas to personnel. While 90° elbows are far more common than 45° bends, changing from one type of elbow to another can have significant thermal effects. Along with providing a lower pressure loss, 45° bends tend to provide a higher resistance to bending. Thermally speaking, additional stiffness in the piping run will usually lead to higher reaction loads at clamps and nozzles as well as higher stress levels in the pipe. Figure 2 shows a comparison of a piping run with both a 45° bend and a 90° elbow.

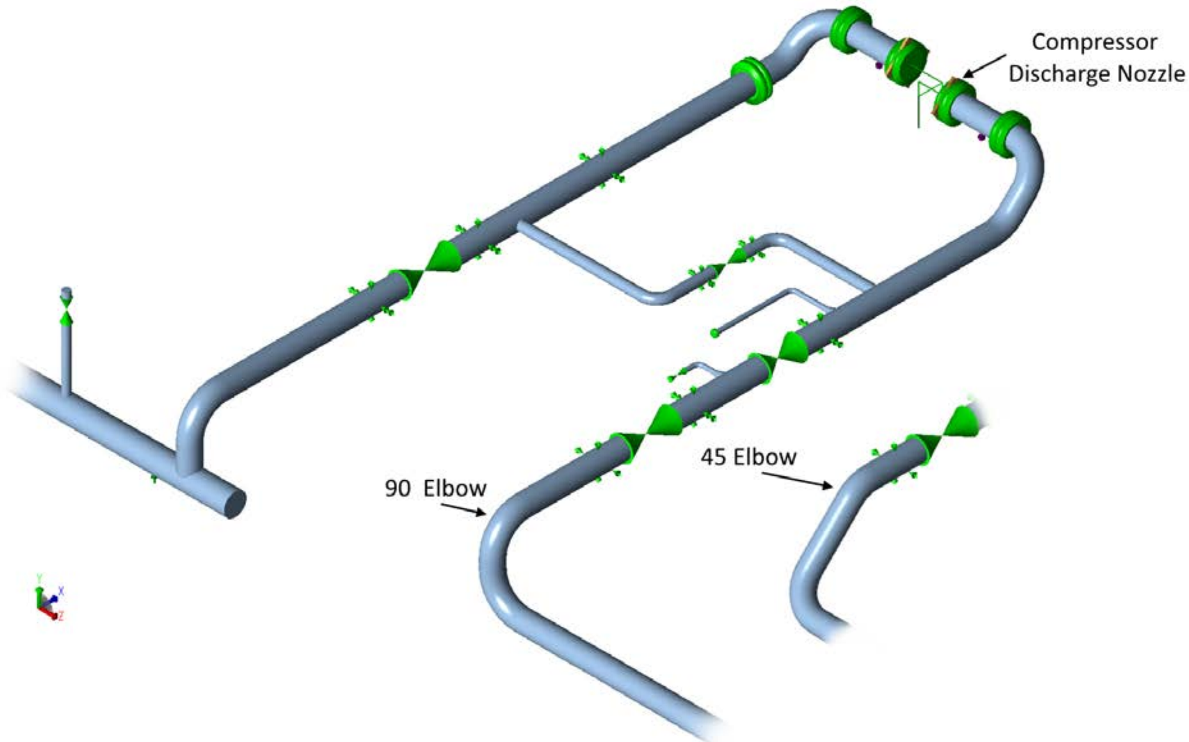


Figure 2. Example Compressor Layout using a 90° Elbow and 45° Elbow

From
Table 1, it can be seen that the 45° elbow nearly doubles the axial stiffness, which would lead to significantly increasing the stresses and loads on the compressor discharge nozzle in both the X and Z directions. Inversely, if a 45° elbow is installed and nozzle loads are excessive, replacing the elbow with a 90° bend will reduce the loads by the same stiffness ratio.

Table 1. Compressor Nozzle Loads for Different Configurations

Discharge Nozzle Loads	45° Elbow	90° Elbow
FX	4,420	3,059
FY	12,794	11,187
FZ	-8,692	-4,433
MX	-779	-321
MY	1,742	1,405
MZ	1,184	1,173
Code Allowable	94%	73%

Compressor Flange Loads

Because the piping is connected directly to the compressor, a thermal analysis is vital to ensure compressor nozzle loads remain below the manufacturer's specified limits. Exceeding these limits without approval from the manufacturer can lead to misalignment between the driver and compressor and can lead to coupling or bearing failures.

The piping, restraints, and soil stiffnesses have a direct relationship to the stress in the pipe as well as the loads on the compressor nozzle. If the nearby restraints are rigid, the piping grows thermally towards the compressor and increases the nozzle loads. A very flexible system would result in low compressor nozzle loads but would be very susceptible to vibration. This balance between thermal flexibility and mechanical stiffness is a critical factor which should be accounted for in a station piping design.

API 617 is commonly used to determine acceptability of the compressor nozzle loads on centrifugal compressors. The API 617 allowable values are equal to 1.85 * NEMA SM23. Nine (9) different allowable values are calculated based on the "equivalent" nozzle diameter. An allowable multiplier is typically applied (often 3.0 * API 617). The calculated moments are resolved about a common point which can and does affect the acceptability of the predicted loads. The summation of forces and moments are compared to the code allowable values along with the three combined loads shown below.

<u>Individual Nozzle Loads</u>

- Suction Flange 3*Fr + Mr
- Discharge Flange 3*Fr + Mr

<u>Nozzle Load Summations</u>

- SFx, SFy, SFz, SMx, SMy, SMz (6 different allowable values)
- 2 *Fc + Mc

Figure 3 below presents a sample calculation of the API 617 compressor nozzle allowable values.

```
Individual Nozzle Calculations
 ------------------------------------------------------------------------------
 Nozzle          Node    Components         Resultants        Values/Allowables
                       ( lb. &  ft.lb.) ( lb. &  ft.lb.)          (ENGLISH)
 ------------------------------------------------------------------------------
 SUCTION          290    FX=   -2314                       3F + M <2781*D(used)
                         FY=    1379
                         FZ=   -1720                       3F + M =     13955
                         Fr=    3196     F =    3196
                                                     2781*D(used) =     37080
                         MX=   -4316
                         MY=    -364                     % of ALLOW. =     37.64
                         MZ=     557
                         Mr=    4367     M =    4367
   Moments About Largest Discharge/Suction Nozzle
   MX=     4288   ft.lb.
   MY=    11363   ft.lb.
   MZ=    -1617   ft.lb.
 ------------------------------------------------------------------------------
 DISCHARGE        360    FX=   -1394                       3F + M <2781*D(used)
                         FY=   -1061
                         FZ=    -764                       3F + M =     16518
                         Fr=    1911     F =    1911
                                                     2781*D(used) =     37080
                         MX=   -4776
                         MY=   -9259                     % of ALLOW. =     44.55
                         MZ=   -2787
                         Mr=   10785     M =   10785
   Moments About Largest Discharge/Suction Nozzle
   MX=    -4776   ft.lb.
   MY=    -9259   ft.lb.
   MZ=    -2787   ft.lb.
 ------------------------------------------------------------------------------
 Diameter Due to Equivalent Nozzle Areas, DC =   17.31 (in.)
 ------------------------------------------------------------------------------
     Nozzle Loads & Summations          Allowables      % of ALLOW.   Status
     ( lb. &  ft.lb.)    (lb. & ft-lb)
 ------------------------------------------------------------------------------
 SFX          =     -3708      -3708    276*DC =      4779      77.60
 SFY          =       318        318    693*DC =     11998       2.65
 SFZ          =     -2484      -2484    555*DC =      9609      25.85
 FC(RSLT)     =      4474       4474
 SMX          =      -488       -488   1386*DC =     23997       2.03
 SMY          =      2104       2104    693*DC =     11998      17.53
 SMZ          =     -4404      -4404    693*DC =     11998      36.70
 MC(RSLT)     =      4905       4905
 2FC + MC     =                13854   1386*DC =     23997      57.73
 ------------------------------------------------------------------------------
                          Overall Status **PASSED**
```

Figure 3. API 617 Compressor Nozzle Calculation

Oftentimes, resolving the loads about a single point will result in the highest calculated (and presumably most conservative) value. This can be due to high thermal loads that should be lowered to ensure safe operation. For larger diameter piping and large compressor frames, however, resolving the load about the discharge nozzle often results in a large moment about the X-axis as shown in Figure 4. The weight of the suction flange (FY) is multiplied by the length between the compressor nozzles (DZ) and creates a high moment about the discharge nozzle. This is not due to any thermal growth of the piping or unit but rather it is due to the weight of the flange.

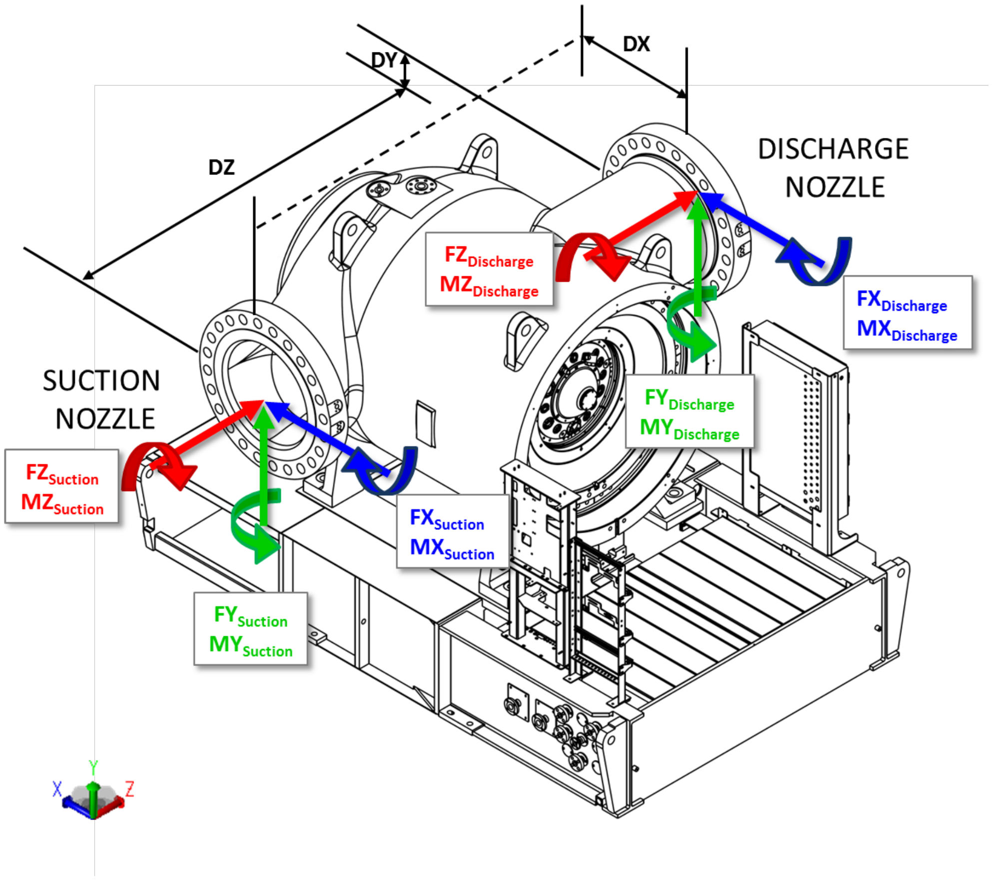


Figure 4. Compressor Nozzle Loads

The predicted loads and code check should be discussed with the OEM to determine if the loads are acceptable to the vendor. A good understanding of the loads and their sources is necessary to understand whether the loads are due to a thermal growth issue or if it is a result of the resolved loads calculation. Good design practice can often lower nozzle loads to acceptable levels during the design phase and is preferred to validating high loads and having a review by the OEM.

Pulsation and Vibration Considerations

Machinery Induced Pulsation

Although centrifugal compressors and their piping systems are not nearly as susceptible to vibration issues as most positive displacement (reciprocating and screw type) compressors, there are still a number of potential risks related to pulsation and vibration that should be considered in the design of their piping systems.

Centrifugal compressors generally produce a relatively continuous flow of compressed gas with typically very minor amplitudes of low frequency pressure pulses in the fluid flow. However, many compressors will produce high frequency excitations in the range of 1,000 to 5,000 Hz due interaction between the rotating impeller blades and stationary blades. This is often referred to as the "blade pass" frequency and occurs at a frequency equal to the number of impeller blades times the rpm/60. The excitation at the blade pass frequency is typically not a concern (in regards to piping design) unless the excitation occurs at the same frequency as a mechanical shell wall mode of the attached piping. If that occurs, significant high frequency pipe wall vibration can occur, which can result in noise concerns and fatigue failures of small bore piping attachments near the compressor, on both suction and discharge. Thin walled, large diameter piping is most susceptible to this. Potential solutions include using heavier pipe schedules on the main piping near the compressor, minimizing the number of small bore attachments near the compressor and possibly the use of a Helmholtz resonator array to absorb the pulsation energy within the compressor frame.

Mixed Compression

The mixed operation of centrifugal and reciprocating compressors in a single compression plant has become common design practice over the last 20 years as this arrangement is beneficial for operational flexibility or when needing additional capacity at an existing older station. Pressure and flow fluctuations can create an oscillating operating point for a centrifugal compressor. As the operating point varies in location on the compressor map, it may pass through regions of instability resulting in the compressor experiencing multiple transient surge events. This creates operational and safety concerns in mixed compression stations as pressure pulsations into the suction or discharge of a centrifugal compressor can move its operating point into operational instability regions such as surge, rotating stall, or choke [6].

To evaluate the effect of pressure pulsations on the operating point of a centrifugal compressor, the piping impedance of the entire system must be considered in conjunction with the compressor operating map to evaluate the attenuation or amplification of pulsations at the centrifugal compressor flanges. The piping impedance is a combination of resistive impedance (i.e., due to frictional losses), acoustic inertia (due to the mass of the gas in the pipe) and stiffness (due to the compressibility of the mass in the pipe). The impedance curve is overlaid on the compressor performance map to convert the pressure pulses to velocity fluctuations thus creating a transient compressor map. The resulting operating point oscillates when exposed to pulsating flow in a path described by an ellipsoidal shape on a head/flow centrifugal compressor map. Experimental testing showed that 30% or more of the area defined by the transient operating ellipse is across the surge line, the centrifugal compressor will experience the effects of surge [7]. Modifications to the piping impedance, compressor surge margin, or additional attenuation of the reciprocating compressor pulsations can be made to avoid the possibility of the centrifugal compressor experiencing surge.

Flow Induced Excitation Analyses: FIV and AIV

Flow induced excitation is used to describe an excitation source that is purely flow related rather than from a machinery source. Well-known piping sources of flow induced excitation in compressor systems are high velocity and/or high density flow, changes in the piping geometry such as a tee or protrusion and valves or orifice plates with high pressure differentials. The excitation can be of low amplitude over a wide range of frequencies, broadband excitation, or discrete excitation with periodic oscillations with high amplitudes at particular frequencies.

The three types of flow induced excitation that are analyzed in compressor systems are acoustic induced vibration (AIV), flow induced pulsation (FIP or flow induced excitation) and flow induced turbulence (FIT). The Energy Institute Guideline (EI) provides the only widely available reference when performing an analysis of these types of excitations [8]. However, this guideline is considered by some in the industry to be overly conservative; therefore, many engineering companies and operators have developed internal guidelines when analyzing piping for flow induced excitation.

Flow induced pulsation (also known as flow-induced excitation) occurs when fully developed turbulent flow, as defined by Reynold's number regimes, encounters a disturbance in the piping geometry. The most typical disruptions in compressor piping systems are pipe stub or side-branch connections, instrumentation inserted in the flow such as a thermowell, and orifice or valve internals. The fluid excitation takes the form of periodic vortex-shedding initiated by an unstable shear layer of the flow. If the frequency of the pressure fluctuations associated with the vortex shedding are close to the acoustic or mechanical natural frequency of the flow disturbance, energy from the fluid flow can create a response. The response in turn strengthens the fluid instability and the coherent vortex formation, resulting in a feedback loop between the fluid response and the acoustic response.

New centrifugal compressor piping system should be screened for the risk of flow induced pulsation by performing a vortex shedding induced pulsation analysis (also commonly called a “Strouhal” analysis).

Acoustic induced vibration is a vibration phenomenon due to excitation from valves with high pressure differentials combined with high mass flow rates. The flow is choked at the vena contracta creating a high energy turbulent jet that expands into the piping creating broadband turbulence. This creates pressure and velocity fluctuations in the piping system that can couple with piping acoustic and mechanical natural frequencies creating high cycle fatigue failures at high stress concentration points such as tee connections or welded supports. The excitation is typically low amplitude broadband excitation from approximately 200 Hz to 5,000 Hz, depending on the cutoff frequency [9].

Flow Induced Turbulence (FIT) and Piping Vibration

Low frequency vibrations in piping systems can be caused by broadband excitation forces that occur due to flow-induced turbulence. Flow Induced Turbulence (FIT) is an evaluation of risk of the turbulent energy generated by fluid flow exciting the mechanical natural frequencies (MNFs) of the piping. The amplitude of the turbulent energy is a function of the fluid energy (ρv2 and dynamic viscosity, μ). The goal of this analysis is to quantify the risk of piping vibration problems. It should be noted that this screening method applies to continuous service operation (and not transient operations).

In a centrifugal compressor installation, it is typically recommended that all mechanical natural frequencies of the main process gas piping be above 10 Hz to reduce the risk of a vibration problem. Lower frequencies may be permissible with lower flow rates and/or lower fluid densities. The Energy Institute Guideline provides methodologies in Technical Module T2.2 for additional screening at various fluid densities and flow velocities.

To achieve a 10 Hz (or other) minimum mechanical natural frequency, two general approaches are utilized by industry. The more detailed approach is to run a modal analysis on a model of the piping system using a commercially available finite element modeling software. This can often be done using the same software package as used for the thermal stress analysis calculations.

A simpler approach, which is also a well proven and commonly used industry approach, is to verify that the spacings between pipe supports (pipe clamps) do not exceed recommended maximum distances. This approach involves using classical beam theory to calculate a table of maximum span lengths between supports based on the desired minimum mechanical natural frequency, the pipe diameter and whether or not the pipe span includes any elbows. A typical span chart is shown in Table 2.

Note that this chart is based on a 10 Hz design frequency and assumes dynamically effective piping restraints (typically hold down type pipe clamps) mounted on solid, near grade elevation piers. Also refer to previous comments in the “Piping Supports” section of this paper. Additional considerations may be necessary for elevated piping on more flexible support structures. Piping with heavy insulation or high fluid densities may also need additional consideration.

It is considered good engineering practice to add restraints near valves and flanges, where practical, as concentrated masses are not considered in Table 2. Restraints should also be located near elbows and tees as these components add flexibility to the piping system and introduce coupling points where flow or pulsation energy can couple into the mechanical shaking forces.

Small bore piping (typically 2 inches or less in nominal pipe size) commonly used for various instrumentation and auxiliary systems can also be very susceptible to high vibration and fatigue failures due to any vibration on the main piping. In general, any small bore attachments should be kept as short and stiff as possible.

PIPE SIZE (inches)			TOTAL SPAN LENGTH (feet)		
				Hinged Support	
Outside Diameter	Inside Diameter	Wall Thickness	Straight Span	One Elbow	Two Elbows
1.050	0.824	0.113	8.44	6.11	6.03
1.315	1.049	0.133	9.47	6.86	6.77
1.900	1.500	0.200	11.36	8.23	8.12
2.375	2.067	0.154	12.96	9.39	9.26
2.375	1.939	0.218	12.79	9.26	9.13
2.875	2.469	0.203	14.22	10.30	10.15
3.500	3.068	0.216	15.75	11.42	11.25
3.500	2.900	0.300	15.57	11.28	11.12
4.500	4.026	0.237	17.94	13.00	12.82
4.500	3.826	0.337	17.75	12.86	12.68
6.625	6.065	0.280	21.88	15.86	15.63
6.625	5.761	0.432	21.64	15.68	15.46
8.625	7.981	0.322	25.03	18.14	17.88
8.625	7.625	0.500	24.78	17.95	17.70
10.750	10.020	0.365	27.99	20.28	20.00
10.750	9.750	0.500	27.82	20.16	19.87
12.750	12.000	0.375	30.55	22.14	21.83
12.750	11.750	0.500	30.41	22.03	21.72
14.000	13.250	0.375	32.06	23.23	22.90
16.000	15.250	0.375	34.33	24.88	24.52
18.000	17.000	0.500	36.33	26.33	25.95
24.000	23.000	0.500	42.10	30.51	30.07
30.000	29.000	0.500	47.17	34.18	33.69
36.000	35.000	0.500	51.74	37.49	36.96

Table 2. Maximum Recommended Span Lengths between Pipe Clamps (10 Hz Design Frequency)

System Performance and Dynamic Simulation Analyses

Another consideration during the initial design phase is to perform a dynamic simulation study of the proposed piping system. The purpose of a dynamic simulation study is to evaluate the adequacy of the anti-surge system to protect the centrifugal compressor(s) from surge (reverse flow) and flow instabilities during normal operation or during transient events such as the start-up or shut down of other compressors (reciprocating or centrifugal) that are running in parallel. This type of study can confirm the suitability of the gas compressor system operation with the proposed anti-surge system design, identifying any deficiencies and confirming the required sizing for the anti-surge control valves and any hot-bypasses.

Such an evaluation, which is usually done utilizing company specific or commercially available transient pipeline simulation software, can include the interaction of the other compressors when one unit trips by incorporating the control scheme of the entire centrifugal compression system. Thus, the effect of one machine experiencing an emergency shut down (ESD) can be predicted for the entire system. Other typical applications for this type of modeling could include evaluation of:

- Machine operating range and pipeline system interaction
- Power and fuel consumption (energy utilization)
- Surge conditions
- Load sharing and machinery configuration
- Help to identify pipeline networks capacity, bottlenecks, and harsh transient conditions
- System optimization
- System balance, gas inventory availability and demand
- Leak detection

OPTIMIZING THERMAL LOADS

As noted previously, the loads on the compressor flanges are directly related to the thermal growth of the piping as well as the unsupported weight near the compressor. The section of piping between the line stop and the compressor nozzle has the greatest impact on the nozzle loads. However, the design of the piping layout is often finalized before a thermal analysis is completed limiting the options available to reduce compressor loads. Additionally, any changes, particularly rerouting piping, becomes much more expensive once the design is finalized.

The single most significant factor in the amount of thermal expansion is the temperature of the piping, with expansion increasing approximately linearly with increasing temperature differentials. The piping between compressor and cooler is critical, and this piping is primarily routed above ground to allow for thermal expansion. Internal gas pressure can also cause elongation of the piping, particularly in larger diameter, high pressure piping, but usually temperature dominates. For long underground cross-country pipelines, with near ambient temperatures, the elongation effects of pressure can be more significant than those of temperature.

This section discusses design parameters of the piping layout which have significant effect on the thermal loads. A range of piping geometry parameters near the compressor nozzles were analyzed and using finite element analysis, sensitivity curves are presented for a range of piping sizes, operating conditions, and piping layouts. This section also presents a number of data plots that can be beneficial in checking a proposed layout early in the design stage. The primary focus of this section is on the piping geometry between the compressor flange and the line stop on the discharge piping. Surface plots are provided to help determine the optimized routing and line stop placement for a piping run based on the expected temperature range and desired piping configuration.

Piping Configurations

The effects of different elbow types on thermal growth and loads is not often considered. Flow requirements, pressure losses, and the general station layout usually control the type of elbows selected. The differences in thermal loads from an elbow change can be significant. While 45° elbows provide lower pressure losses, they also provide much higher in-plane bending stiffness. This higher stiffness directly leads to higher thermal loads on nearby pipe supports and equipment. Three different piping configurations were investigated as shown in Figure 5. The 0° configuration consists of a single 90° elbow and does not have any elevation change. The 45° configuration consists of a 45° bend near the line stop followed by a 90° elbow into the compressor. Finally, the 90° configuration consists of two 90° elbows.

Note that Figures 5 through 12 are reprinted with permission from [5].

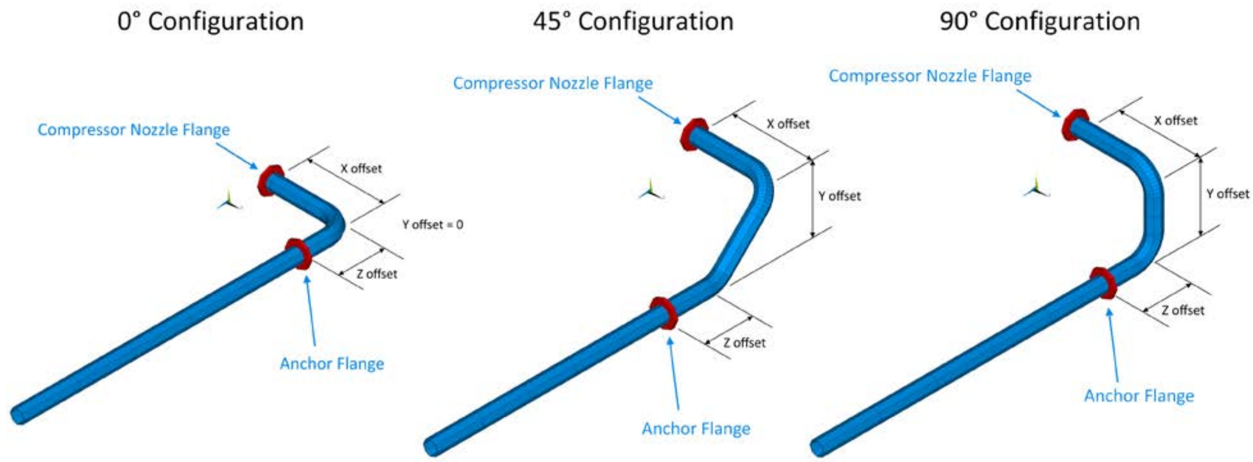


Figure 5. Piping Configurations – 0° Offset, 45° Offset, 90° Offset

A standard piping layout was chosen for the comparative analysis. A stiff boundary condition was modeled 31' in the Z-direction from the compressor nozzle. This is representative of having the piping turn downward into the soil at this location. The piping downstream of the line stop can have a significant effect on the piping stress and thermal loads but for this study, a realistic condition was chosen and kept constant. A stiff section of piping downstream of the line stop greatly increases the thermal loads while a flexible run with multiple elbows would allow the piping to grow freely.

Figure 6 presents the predicted compressor nozzle loads as a function of the x offset. For the 36" piping, the 45° configuration results in the highest loads while the 90° configuration provides the lowest nozzle loads. As discussed earlier this is due to the higher bending stiffness of the 45° bend as well as the location of the line stop. For the 45° configuration, the line stop is located further from the compressor due to the nature of the layout. The further the line stop from the compressor, the higher the nozzle loads. This is shown in Figure 8 and is discussed later.

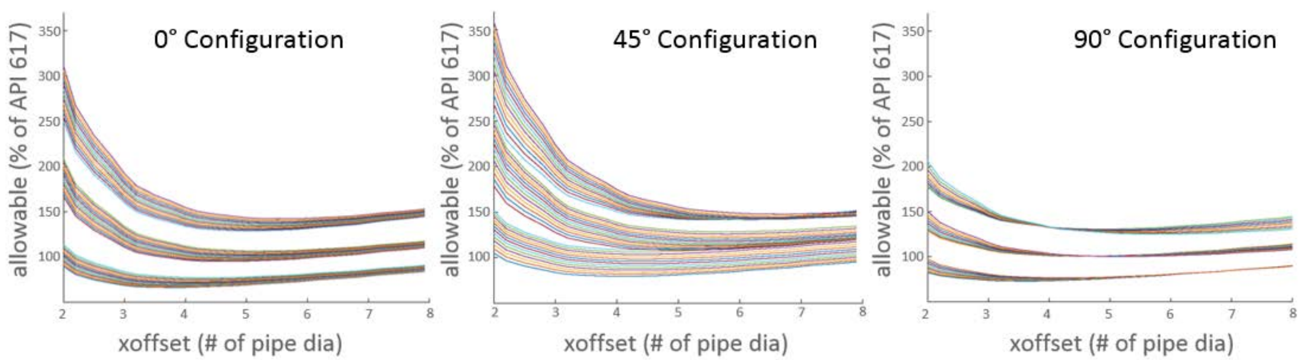


Figure 6. Configuration Comparison – Allowable (% of API 617) for 0°, 45°, 90° Configuration and 36" OD

The trend of 90° elbows lowering thermal loads is consistent for all configurations. When possible, it is recommended that 90° elbows be used to reduce loads around compressors and coolers; however, the added flexibility does result in lower mechanical natural frequencies which are more susceptible to vibration. Flow consideration should also be taken into account when developing a balanced piping layout.

Anchor Flange Stiffness and Location

Anchor flanges or line stops are a crucial element in controlling the thermal growth of the piping. Ideally, the stop should force the thermal expansion away from the compressor. The design of the line stop should allow for lateral growth of the piping while providing axial support to the anchor flange.

Line stops are typically modeled and assumed to be very stiff, if not completely rigid. While this assumption is valid for certain line stop designs, overestimating the stiffness and effectiveness of a line stop can significantly under-predict the compressor nozzle loads. The design of the line stop should be reviewed and analyzed to confirm the assumed stiffness is accurate. Figure 7 below shows the stiffness near the base of the lines top is nearly twice as high as the stiffness near the top.

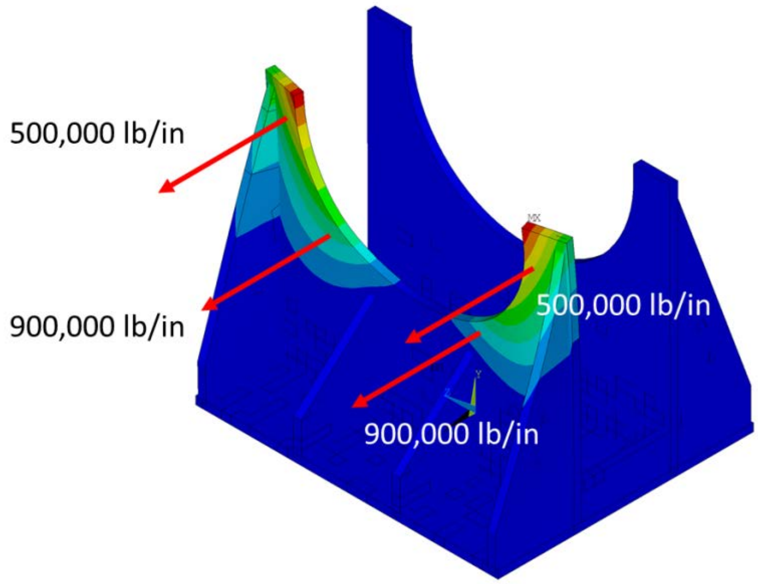


Figure 7. Finite Element Analysis of a Line Stop Presenting Calculated Stiffness at Jack Bolt Locations

The placement of the line stop can also have a significant impact on the compressor nozzle loads. The thermal loads are directly related to the length of the piping in question. The primarily function of the line stop is to direct the growth of the piping away from the compressor. However, the piping between the line stop and the compressor will grow thermally and impart loads on the compressor nozzle. The longer this section of piping is, the higher the compressor nozzle loads. As shown in the plots of Figure 8, the nozzle loads are linearly related to the placement of the line stop. It is recommended the distance between the line stops and the compressor nozzles be minimized as much as reasonably possible.

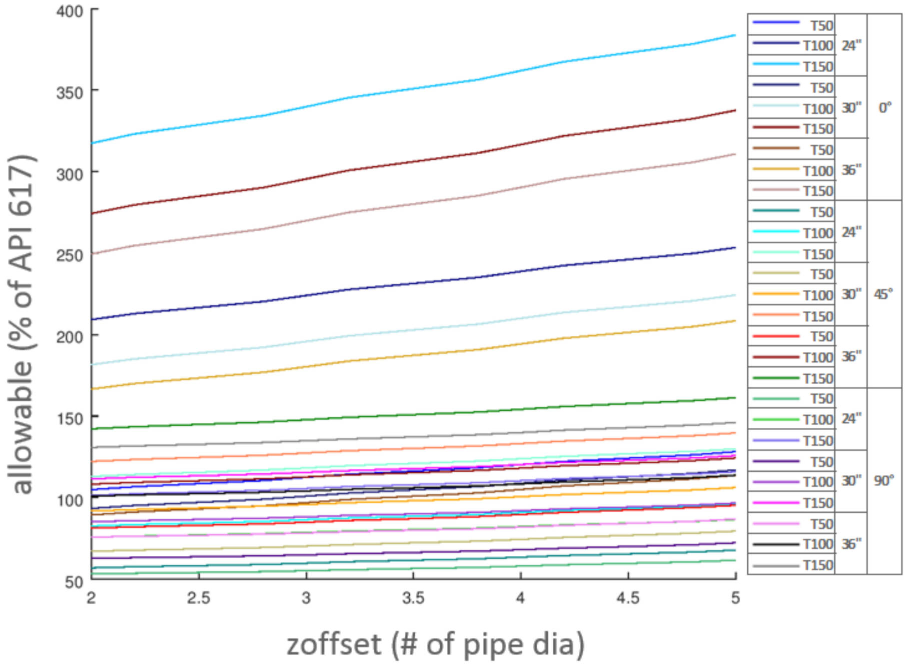


Figure 8. Allowable (% of API 617) vs. Z Offset – Overall

While it is recommended the length of pipe between the line stop and the compressor nozzle is kept to a minimum, reducing this length in turn lengthens the section of piping downstream of the line stop. The added length often results in high thermal loads on the anchor flange. With a properly designed line stop the loads may be acceptable however the design of the line stop should be evaluated to ensure the loads are not excessive. Figure 9 presents the effect of the line stop placement on the anchor flange loads.

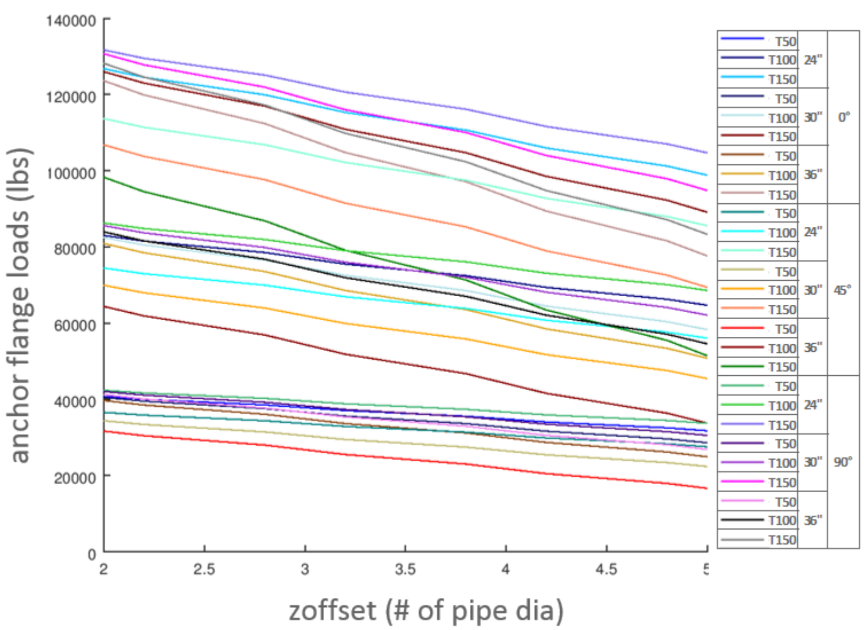


Figure 9. Anchor Flange Loads (lbs) vs. Z Offset – Overall

Pipe Stiffness as a Function of Pipe Diameter

Large diameter piping is becoming more common as larger units are being installed. While large diameter piping allows for high flow rates, they also provide higher stiffness in the system than smaller size piping. A 6' by 6' bend was modeled to quantify the effect larger diameter piping has on the system stiffness. This bend would be similar to a cooler riser or discharge piping turning downward and running underground. The two ends of the pipe remained in the same location for all runs and the piping size was swept from 4" to 48". A unit load was applied on one end of the model while the other was fixed. Figure 10 shows a stiffness curve for a wide range of pipe sizes.

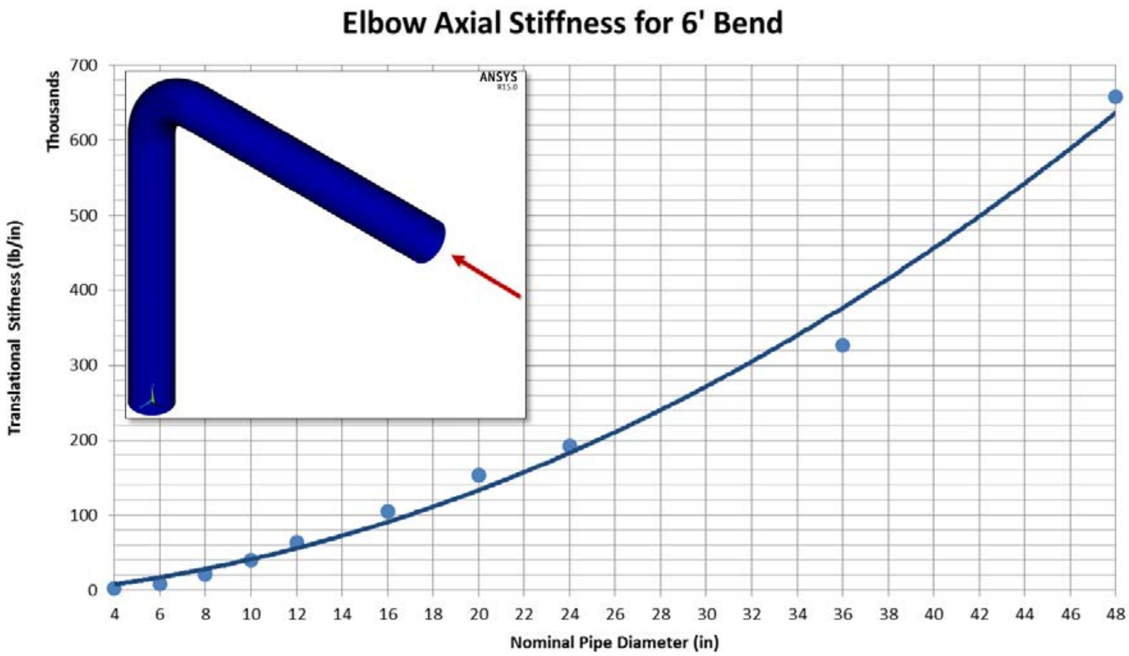


Figure 10. Pipe Run Bending Stiffness for Different Pipe Size

The curve shows the stiffness changes exponentially with an increase in piping size. Therefore, a similar layout with a larger pipe size would result in much higher equipment loads. On a smaller sized line, the elbow provides added flexibility which relieves thermal stress and loads. A larger diameter elbow does not provide the necessary flexibility and can result in overloaded pipe supports or equipment.

While decreasing pipe size in not a simple change, shifting reducers to incorporate smaller diameter elbows is recommended when possible. Figure 11 shows a comparison of the compressor nozzle loads as a percentage of API 617 for different pipe sizes. Note that increasing the x offset is much more effective in reducing nozzle loads for the 24" piping than the 36" piping.

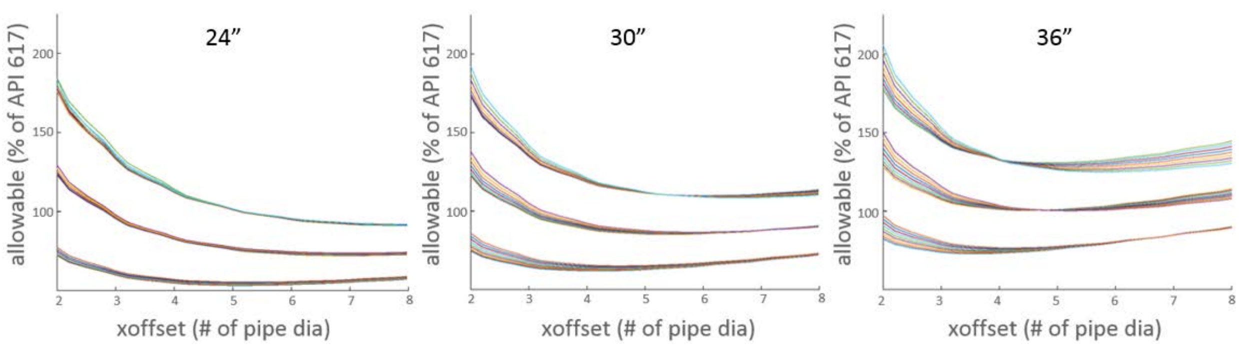


Figure 11. Pipe Diameter Comparison – Allowable (% of API 617) for 24", 30", 36" in 90° Configuration

X & Y Offset

The length of the run of pipe into, or out of the compressor (X offset), and the elevation change (Y offset) of the piping have a significant impact on the compressor nozzle loads. A short run and small elevation change results in a very stiff section of piping between the line stop and compressor nozzle. A stiff section of pipe directly translates the thermal growth of the pipe to thermal loads. A longer run of pipe allows the pipe to flex and reduces the loads. Understanding the effects of both of these offsets up front allows for a more efficient design and reduces costly modifications further in the design process.

Maximizing both these lengths results in low thermal loads, however it also increases the weight loads on the nozzle. There is an optimal range where the piping provides enough flexibility to absorb the thermal growth while minimizing the weight of the piping itself. Figure 12 presents a surface plot showing the effects of varying the X offset and the Y offset for a 36” 90° configuration.

For this case the optimal dimensions would be 6 pipe diameters in the X direction and 5 pipe diameters in the Y direction. The compressor nozzle loads are more sensitive in the X offset direction and not as sensitive to an elevation change of the piping. While this layout may not be feasible, the surface plot allows modifications to be developed quickly and allows the user to prevent high loads on the compressor.

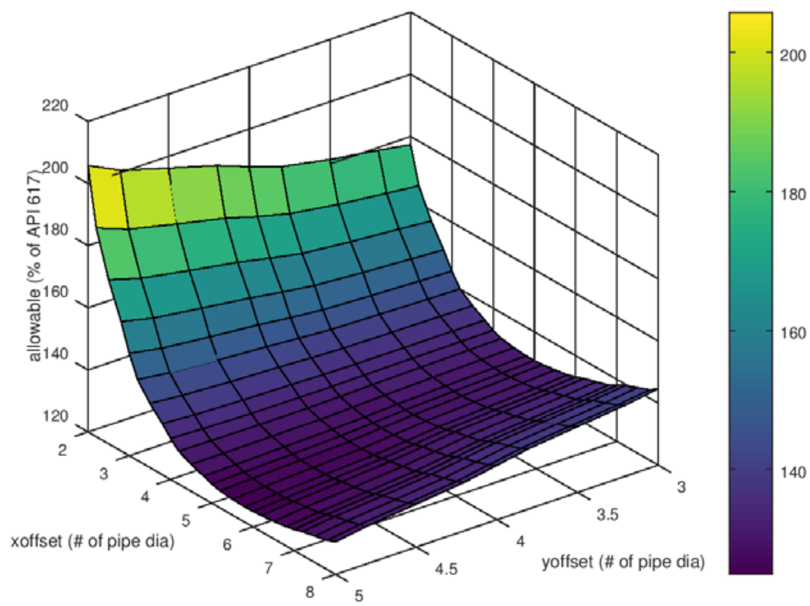


Figure 12. Surface Plot Example – Allowable (% of API 617) vs X offset vs Y offset for 90° Configuration, 36”, ΔT=150°F

FLOW DISTORTION AND AERODYNAMIC PERFORMANCE

Overview

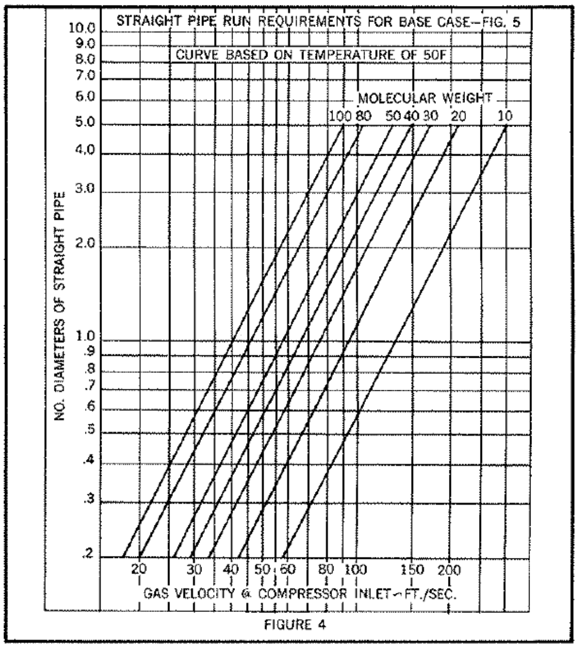
The aerodynamic performance of a centrifugal compressor depends strongly on the velocity and total pressure profile of the flow when it arrives at the inlet [10]. Thus, one of the primary goals of the design of the inlet piping is to ensure that the flow is as uniform as possible at the inlet plane. Many manufacturers and operators today still use design guidelines [11] that were developed in the 1960s and 70s and that have not been updated since then. The purpose of these guidelines is to calculate the length of the last run of straight pipe that leads to the compressor. However, their empirical and theoretical basis has not been revised recently and does not guarantee homogenous flow arriving at the impeller. Additionally, designs for double-inlet compressors that follow these guidelines may create additional problems. Double-inlet compressors have an additional requirement that both inlets receive an equal amount of flow [12]. Nevertheless, it has been shown that designs that rely only on these guidelines can potentially result in flow distribution problems that affect the performance of the machine [13].

Given the widespread use of these dated methods, recent research is studying the effects of asymmetric velocity profiles on the performance of centrifugal compressors. Because the performance of each machine is unique, there are not any general reduced order methods that can be used to predict the performance losses caused by improper inlet piping design. Thus, there is a need to predict, on a case-by-case basis, the shape of the velocity profile as it arrives at the impeller and how it will affect the performance of the compressor. CFD simulations can be used to quantify the uniformity of the velocity profile and amount of swirl at the compressor inlet plane for different piping layouts. The effect of those layouts on the performance of a generic impeller wheel is part of an on-going research effort. The data from this experimental study is being used to validate the CFD simulations and to develop an understanding of how compressor performance is influenced by the incoming flow.

Technical Background

The performance of a centrifugal compressor is strongly affected by the quality of the flow that arrives at the impeller [14]. In the case of these machines, flow quality may be assessed by looking at, among other parameters, the inlet velocity profile. In general, it should be as uniform as possible to guarantee homogenous flow and pressure distribution around the impeller. Obtaining a perfectly uniform inlet velocity profile is impossible by definition. Circular pipes are used to bring the flow to the impeller, which implies that, under the best circumstance, the flow will exhibit a parabolic velocity profile and additional disturbances may arise from other sources, such as surface roughness, turbulence, or swirl originating farther upstream in the piping. However, proper design of the inlet piping system can minimize these problems and ensure that compressor performance is not compromised.

Today, many compressor manufacturers and facility designers use guidelines that were based on recommendations for flow meters [11]. These procedures are still being referenced in recent books on operation of centrifugal compressors [10] [12] [14]. The purpose of these guidelines is to determine the minimum required length of straight pipe that leads to the compressor inlet to allow swirl created by upstream elbows to be dissipated. The typical procedure starts with the calculation of the flow velocity and, based on the fluid temperature, the required inlet length, in terms of pipe diameters, is determined. Additional allocations are made depending on the type and number of pipe elements in the piping layout directly upstream of the compressor. This method is very simple but may result in longer upstream lengths of straight pipe than are actually required or, conversely, may yield inlet lengths that are too short to ensure sufficient flow uniformity, since the design methodology does not consider the amount of swirl that the flow has from all the piping elements upstream of the compressor. The charts and correction factors used in this procedure are shown in Figure 13.



COMPRESSOR INLET CORRECTION FACTORS FOR VARIOUS PIPING ARRANGEMENTS

	Factor	Figure
1. One long radius elbow (plane parallel to rotor)	1.0	5*
2. One long radius elbow (plane normal to rotor)	1.50	6
3. Two elbows at 90° to each other with second elbow plane parallel to rotor	1.75	7*
4. Two elbows at 90° to each other with second elbow plane normal to rotor	2.0	8
5. Butterfly valve before an elbow		
a. valve axis normal to compressor inlet	1.5	9*
b. valve axis parallel to compressor inlet	2.0	—
6. Butterfly valve in straight run entering compressor inlet		
a. valve axis normal to rotor	1.5	—*
b. valve axis parallel to rotor	2.0	—
7. Two elbows in same plane (parallel to rotor)	1.15	10*
8. Two elbows in same plane (normal to rotor)	1.75	—
9. Gate valve (wide open)	1.0	—*
10. Swing check valve (balanced)	1.25	—*

*Factors also apply to single stage, axial inlet compressors.

(a) (b)

Figure 13. (a) Chart and (b) Correction Factors for the Determination of the Final Straight Section of a Centrifugal Compressor Inlet (taken from Reference [11])

The rationale behind these traditional design methods presents several problems. First, these guidelines were developed in the 1960s without much attention being paid to the physical phenomena that occurs in real pipe flows. Instead, they relied only on simple measurements of the velocity profile some distance downstream of the last flow-turning element ahead of the compressor inlet. For instance, it is now well known that the flow through an elbow results in secondary flows that can convect downstream for up to 100 pipe diameters. The most well-documented secondary flows produced through pipe bends are the so-called Dean vortices - a counter-rotating vortex pair that is, usually, symmetrical about the half-plane of the pipe. An example of the Dean vortices is shown in Figure 14 below. However, there are instances when there is only one dominating, larger vortex, or when the vortices are not symmetrical owing to turbulence effects upstream of the bend. In industrial applications involving centrifugal compressors, these vortices must be dissipated before the flow arrives at the impeller to ensure optimum aerodynamic performance.

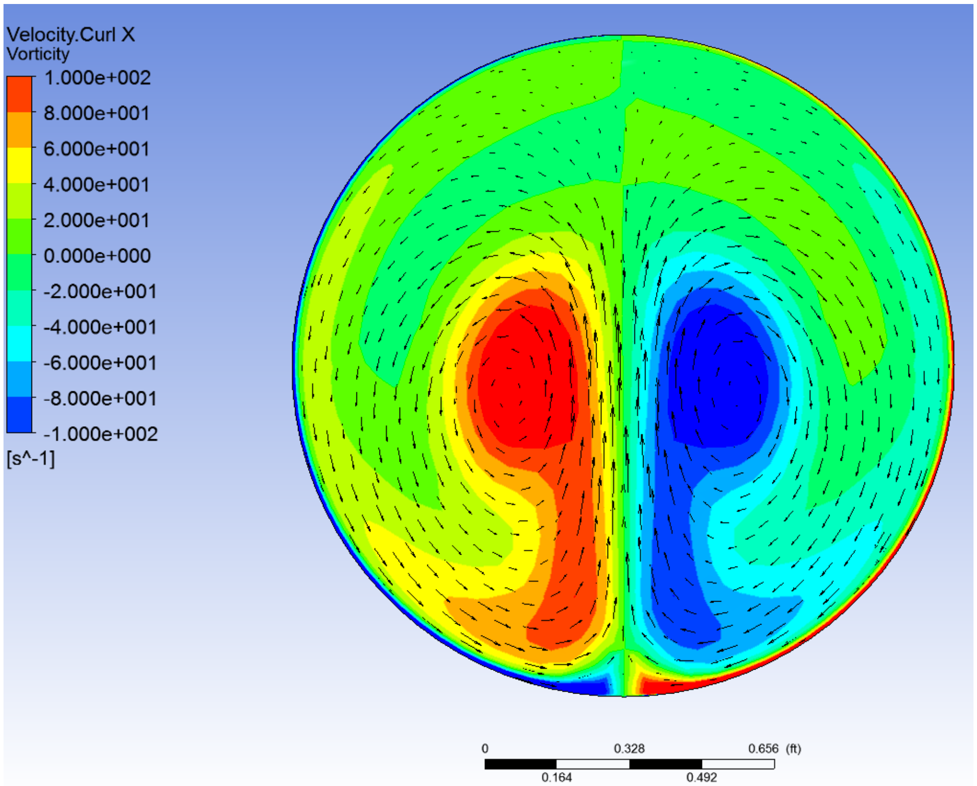


Figure 14. Dean Vortices Created by the Flow Through a 90° Bend

Second, some of the correction factors used to determine the number of required upstream straight pipe diameters are arbitrary. For instance, when discussing their base compressor case, Hackel and King [11] stated that "to compensate for the effect of the normal-oriented elbow, it is suggested that the straight run of inlet pipe be 50 percent longer than the base case or a correction factor of 1.5 should be used". For the case of two elbows oriented at 90° to each other, they suggest "that the straight run ahead of the compressor be 75% longer than the base case thus providing a little more time for the flow to equalize" [11]. These correction factors are presented without any apparent rationale other than extending the length of the pipe gives the flow more distance to dissipate the Dean structures that may arise from the upstream piping. In addition, the recommendations of Hackel and King [11] are based on the assumption that the distance between the turning elements under consideration and the previous element is at least 10D, where D is the pipe diameter. This is considered to be a best-case scenario, which may not always be found in industrial applications. As a result, the calculation procedure may result in lengths of pipe that are too long (with the associated increase in cost and pressure losses) or too short (which would not completely remove the flow non-uniformities).

Finally, they do not consider how the perturbed velocity profile interacts with other elements along the piping layout. An example of this problem would be how the kidney-shaped vortices coming off one elbow would interact with another piping element, such as another elbow or a valve, farther downstream. Figure 15 below shows the flow structure resulting from the flow past two out-of-plane 90° elbows. The Dean vortices have turned into two vortices that rotate about the centerline of the pipe.

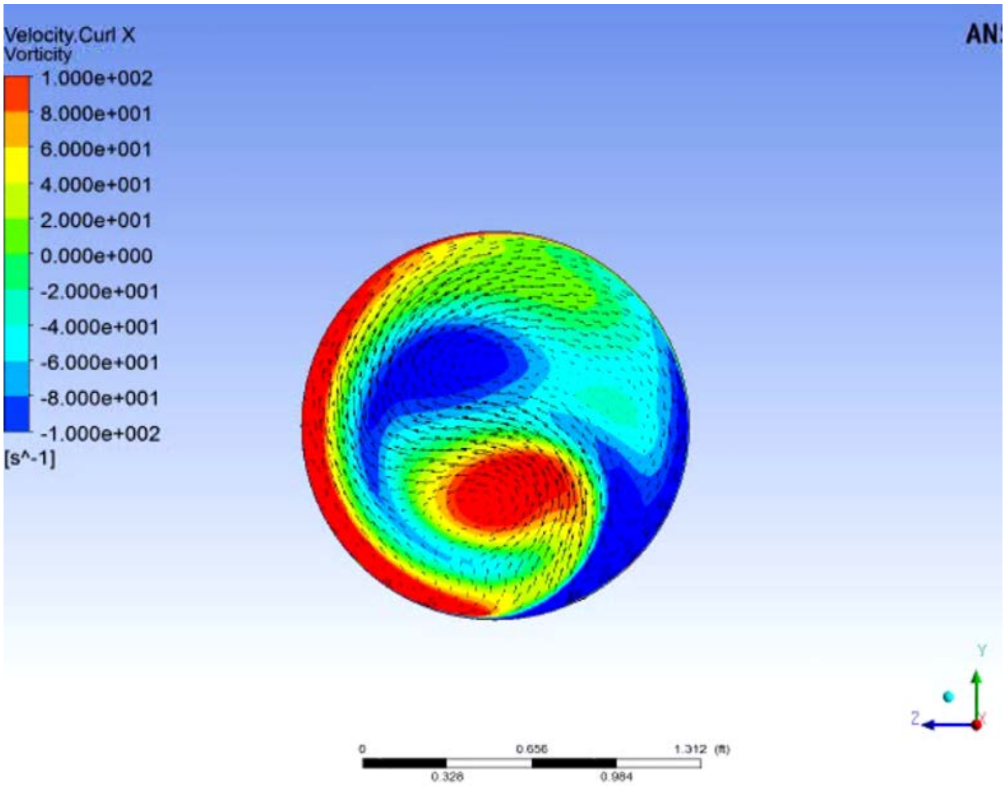


Figure 15. Vortices Produced by the Flow Through Two Out-of-plane 90° Bends

The effect of inlet flow non-uniformities, also known as flow distortion, on turbomachinery performance has long been a topic of research in the turbomachinery community. This work has historically been done both on axial and radial compressors. Significant reductions in both machine efficiency and operability have been reported due to inlet flow distortion of all types. Inlet distortion can also cause cyclic loading on the impeller and rotor, which may be significant especially at high pressures [15]. Some of the earliest work used artificial distortion generators as opposed to realistic inlets that produce distortion. This was done in an attempt to generalize the impact of different levels of flow non-uniformity [16] [17]. Much of the work in the literature is on the effect of distortion on axial fans with application to jet engines. In this space, several industry standards exist that describe methods to describe numerically any arbitrary inlet flow distortion profile [18] [19]. Even with a rigorous method for quantifying flow distortion, a generally applicable relationship between severity of flow distortion and machine response is not available. Even the most recent research is focused on experimentally or numerically testing the effect of a certain distortion on a specific turbomachine [20].

Regarding centrifugal compressors, the focus of much of the work in the literature has been on quantifying and proposing methods to reduce the effect of upstream piping features, such as elbows [21] [22] [23]. In recent years, with the advent of computational fluid dynamics, the flow through a distortion-causing inlet, as well as the effect on the compressor itself, can be simulated. Numerical simulations play a major role in most recent publications, even to the exclusion of experimental results in some papers.

Experimental Setup

An on-going research project aims to correlate the performance of the compressor with the length of the pipe immediately upstream of the inlet. This study is taking place in an open loop air compressor performance test rig shown in Figure 16. The inlet piping can readily be adjusted to vary the length of the last straight section between 1 and 10 pipe diameters.

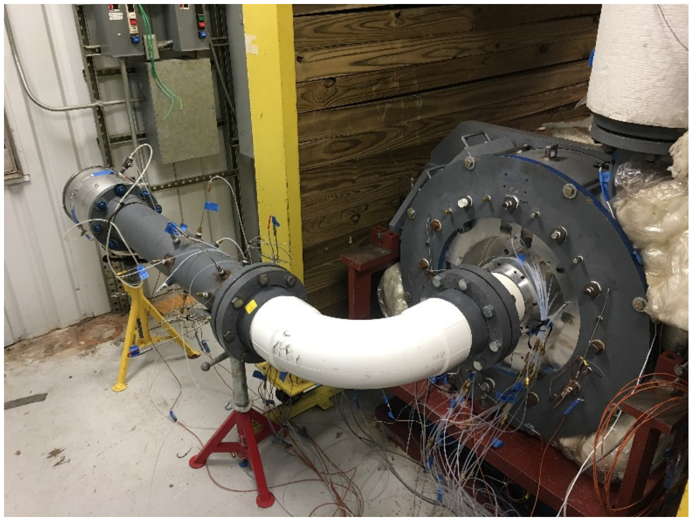


Figure 16. Single Stage Test Rig

Experimental Results

Using the experimental setup described previously, the compressor performance with an elbow one diameter, two diameters, four diameters, and ten diameters upstream of the compressor was tested. All of these tests were done at the same corrected speed as the baseline case which consisted of a purely axial inlet. During each of the tests, the compressor was tested at the maximum flow condition as well as reduced flow by using an outlet valve. The efficiency was calculated by measuring the total pressure and temperature upstream of the elbow as well as the compressor outlet pressure. This was used to calculate the isentropic headrise of the compressor stage, which was compared to the shaft power input into the impeller as measured by a torque meter.
Isentropic efficiency for the baseline and distorted cases is shown in Figure 17.

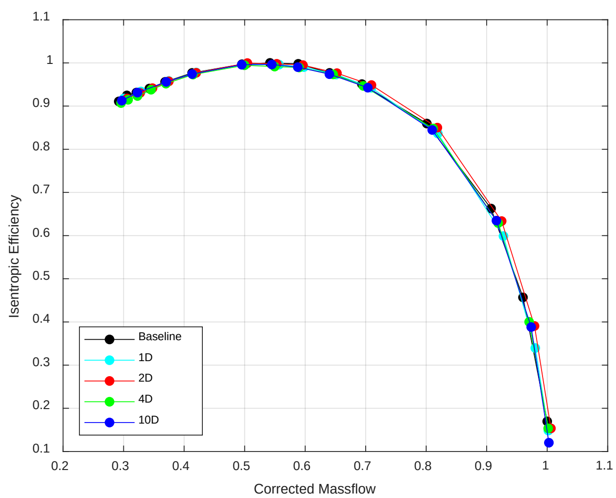


Figure 17: Normalized isentropic efficiency as a function of normalized corrected massflow for the baseline as well as the elbow configurations.

As can be seen in in the figure, the isentropic efficiency is essentially the same for all of the cases that were tested. This shows that the imposed distortion created by the elbow had no effect on the compressor efficiency regardless of where it was placed relative to the compressor inlet. Physical limitations prohibit putting the elbow much closer to the impeller than one diameter in the single stage test rig.

This result is in contrast with the recommendation by Hackel and King, which, for this configuration, would require 7.5 diameters of straight pipe after the elbow for the high flow condition, 2.5 diameters of straight pipe for the peak efficiency condition, and 0.75 diameters of straight pipe for the low flow condition [11].

Similar to the effect on isentropic efficiency, for the right hand side of the map, there was no change measured in the isentropic headrise as shown in Figure 18.

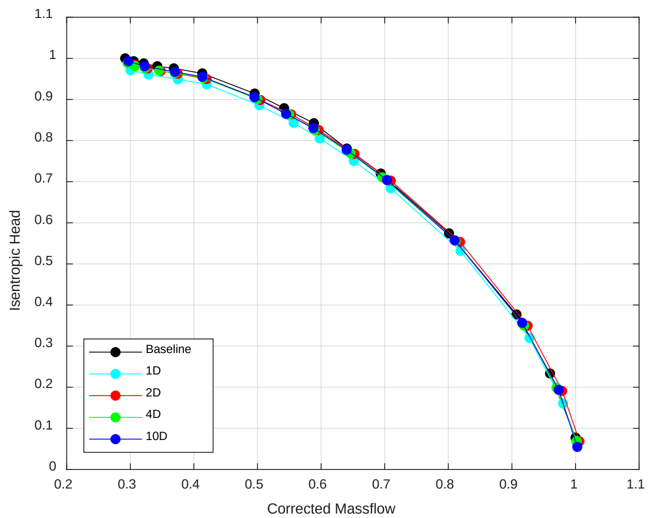


Figure 18: Normalized isentropic head as a function of normalized corrected massflow for the baseline as well as the elbow configurations.

Near the left hand side of the map however, the distorted inlet cases show slightly lower isentropic headrise with the one diameter case showing the greatest reduction. This slight reduction in isentropic headrise for the most aggressive swirl distortion case is similar to what would be seen if there were a bulk swirl pattern imposed on the impeller by IGVs. It should be noted that this reduction in isentropic headrise does not come with any appreciable reduction in isentropic efficiency as shown in Figure 17.

During the compressor testing, no appreciable difference in compressor range was detected. Each of the tested configurations entered compressor surge at roughly the same point on the compressor map. The furthest left point on the map was chosen during the testing of the baseline case as being a reasonable final stable operating point directly before surge. This same point was used for all of the elbow configurations and the impeller behavior at this point for all cases was similar to the baseline case.

Conclusions and Future Work

While swirl distortion certainly can have an effect on compressor performance, the swirl distortion created by the tested configurations in this study had minimal impact. This result shows that swirl distortion is a complicated phenomenon and that the referenced guidelines are too generalized to be broadly applicable. In this situation, these guidelines would have required an inlet piping configuration that would have been unnecessarily long which would add additional piping costs and complexity to a permanent installation of this compressor.

Future work in this area will include additional upstream elements such as elbows in and out of plane with each other as well as valves of different types. Along with this, future research will include the instrumentation necessary to directly measure the swirl at the impeller inlet instead of relying on CFD to predict the swirl distortion profile. Also, impellers of different types need to be tested in order to ensure that the results presented here, as well as those from future studies, are not unique to a particular impeller. Also, in the future, CFD simulations of the impeller itself will be done to verify the ability of numerical methods to predicted changes in compressor performance due to inlet distortion.

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