A CONUNDRUM – THE DIFFICULTIES OF PIPE STRESS ANALYSIS FOR COLD PIPES

Ross Sinclair (Jacobs NZ Limited) and Glenn McDougall (Jacobs NZ Limited)

ABSTRACT

It is often argued that ambient temperature piping systems do not require pipe stress analysis. This may be true up to a point – where good engineering practice is followed and the piping systems have a generous amount of flexibility. However, there are aspects of cold water piping design that do require an analysis of the stresses in the piping system and the loads generated by the pipework.

In particular, pump stations need to be properly engineered to manage pump nozzle loads. High nozzle loads can cause misalignment of pump and motor and cause wear and vibration. Also, in the New Zealand context, the significant loads arising from seismic action need to be controlled. It is essential that any pump can withstand these loads without significant damage.

This paper looks at the design of cold water pipework with a focus on pump station design. Case studies illustrate some of the issues that arise in pipe stress analysis of cold water systems and how to resolve these. The importance of developing an appropriate flexibility concept at the preliminary design stage is described. In particular, it discusses the difference between high temperature piping and cold water piping and why the latter often requires a more complex analysis.

KEYWORDS

pipe stress, analysis, cold water, pump station

PRESENTER PROFILE

Ross Sinclair is a Senior Mechanical Engineer at Jacobs in Auckland. He has over 25 years' experience including thermal, geothermal and hydro power generation, desalination, materials handling, forest industry, dairy, food, water and wastewater. His specialties include design and stress analysis of piping systems, pipe supports and associated pressure vessels.

1 INTRODUCTION

Is pipe stress analysis required for cold water piping systems? Many will argue that it is not required.

In practice it has become a much more common element of the design of piping systems in the water and wastewater industries. This has happened for a number of reasons:

- pressure to make designs more efficient – optimisation of pipe sizes and minimising of building sizes, resulting in a tighter piping layout
- a move to larger systems
- increasing emphasis on seismic loadings and behaviour of piping systems under seismic action
- owners are demanding an increased level of assurance that their plant has been properly engineered

For every piping system a high level assessment should be carried out to determine if pipe stress analysis is required.

Factors that can require the use of pipe stress analysis:

- an increase in pipe temperatures from ambient – a change in the fluid temperature, an empty pipe exposed to sunlight, a full pipe exposed to sunlight, hot fluids

- displacements – settlement of structures or pipe supports, seismic or wind action displacements
- limits on loadings for nozzles on pumps, compressors and vessels
- large diameter pipes, large valves, high loadings

1.1 LOOKING BEHIND THE SCENES IN PIPE STRESS ANALYSIS

Pipe stress analysis that is performed by programmes such as AutoPIPE and Caesar 2 is a powerful tool, but in many ways a very simple form of analysis. Whilst the program graphics show a 3D model of the piping, the actual representation inside the calculation engine is a series of beam elements along the pipe centreline that have properties added – wall thickness, density, elastic modulus, weight of contents etc.

The diagrams below illustrate this point. Figure 1 is the 3D representation of a simple system. Figure 2 is the same system shown in a single line view. High stresses in piping systems are normally encountered at branch tees, bends and reducers. The piping codes apply stress intensification factors (SIFs) to these features to determine the magnitude of the pipe stress.

Figure 1 - 3D Representation

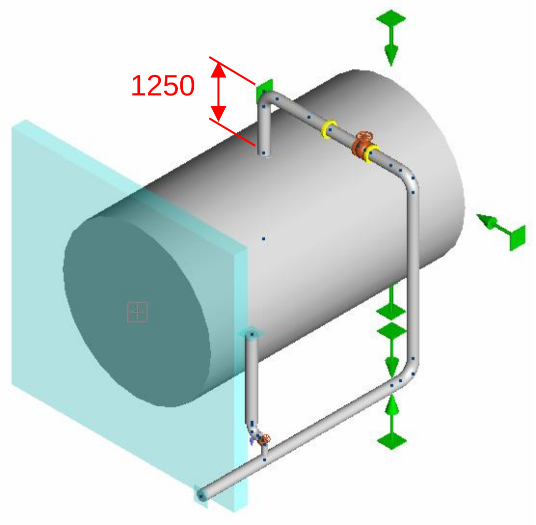
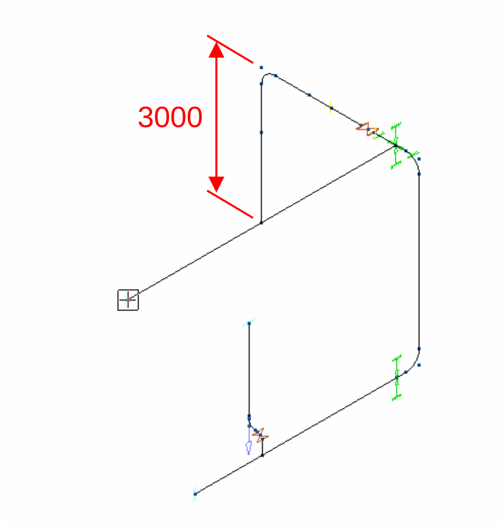


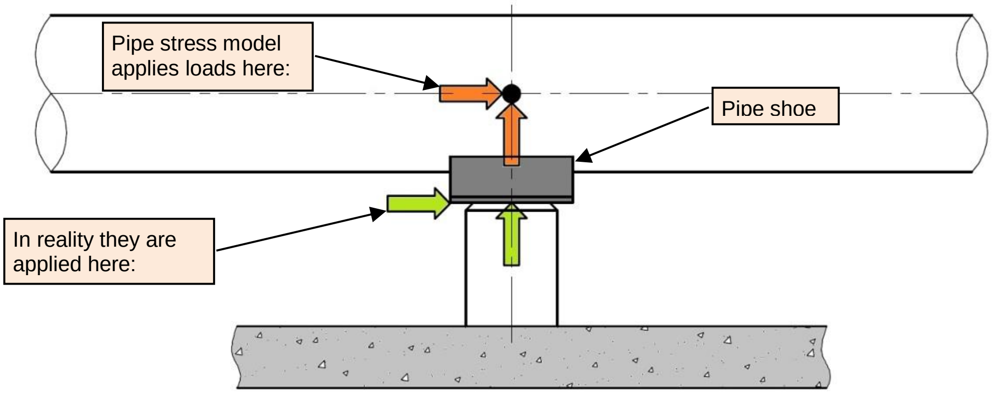
Figure 2 - Single Line Representation



The model above is a 3500mm diameter vessel with a DN200 line connected to a nozzle on the top. The expectation is that the DN200 pipe has a projection of 1250mm from the top of the vessel. However when modelled in a pipe stressing program, the connection point is on the vessel centreline, and so it is modelled as 3000mm long, giving a significantly different behaviour to the real world object. This can be corrected by modelling techniques, but users need to understand the significance of this characteristic of pipe stress analysis.

It can be particularly important when considering the action of pipe supports near pump nozzles. Because the pipe stress model is a centreline, the pipe supports act on the centreline of the pipe. In reality the reaction point for a pipe support is always somewhere outside the pipe – see the example in Figure 3 below.

Figure 3 Application of Load to Pipe from Pipe Supports

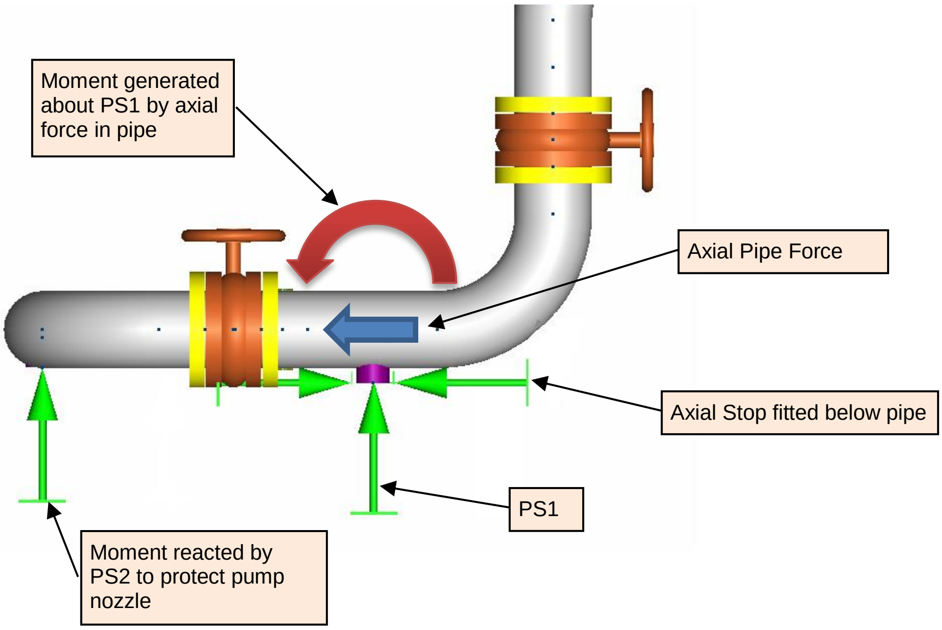


For a straight run of pipe located away from points of interest this approach gives perfectly adequate outcomes. However the same approach applied to a support near a pump nozzle can give a completely erroneous result. Figure 4 below shows the DN500 discharge pipe on a high pressure pump for desalination of seawater using reverse osmosis.

Figure 4 R.O. Desalination Discharge Pipe

Pipe support PS1 is fitted with an axial stop to protect the pump nozzle from being overloaded by the piping. When modelled on the pipe centreline, the axial stop has little effect on pump nozzle loads. However the physical arrangement of the pipe support shoe puts the axial stop reaction point 100mm below the bottom of pipe. When modelled correctly with an extension from the centreline to the bottom of the pipe shoe, the axial force creates a moment about the support (see Figure 5 below), which in turn causes increased loads at the pump nozzle. PS2 is used to react against this moment and avoid nozzle overloading.

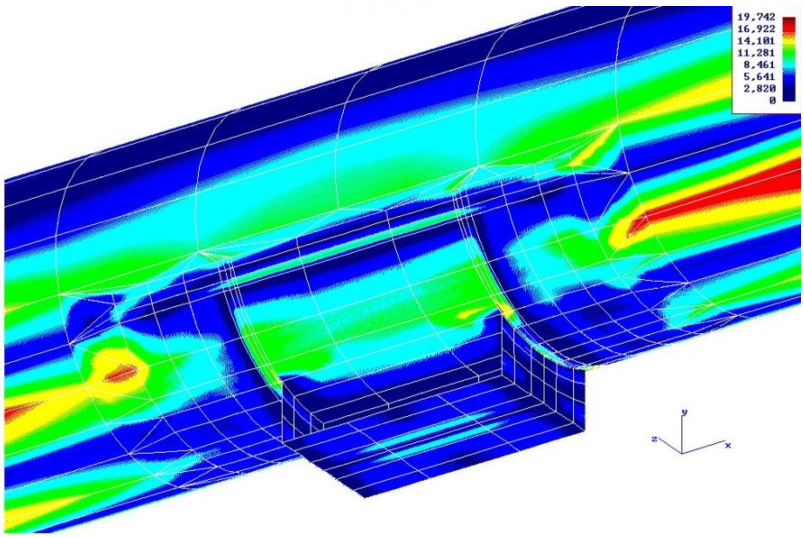
Figure 5 Axial Stop on Discharge Pipe



With pipes and pump stations being constructed in ever increasing sizes, there is a need to ensure that pipe supports are suitably engineered. One of the benefits of carrying out a pipe stress analysis is to obtain accurate loads for design of the pipe supports. But the beam type pipe stress analysis from AutoPIPE and Caesar 2 does not check for local stresses in the pipe wall. Rather it calculates stresses on the gross pipe section as if they are perfectly distributed into the entire cross-section.

So for high load supports and sensitive pipes, a local stress check is required. Thin walled pipes such as blower pipes can be susceptible to local stress failures. Supports next to valves carry high loads – for example a DN900 gate valve can weigh 3000 kg. Cement linings can deteriorate if cracked due to local deformation. Finite element analysis (FEA) techniques are often used to check local stresses. Figure 6 below shows the FEA analysis of a pipe shoe and reinforcing pad.

Figure 6 Local Stress check using finite element techniques

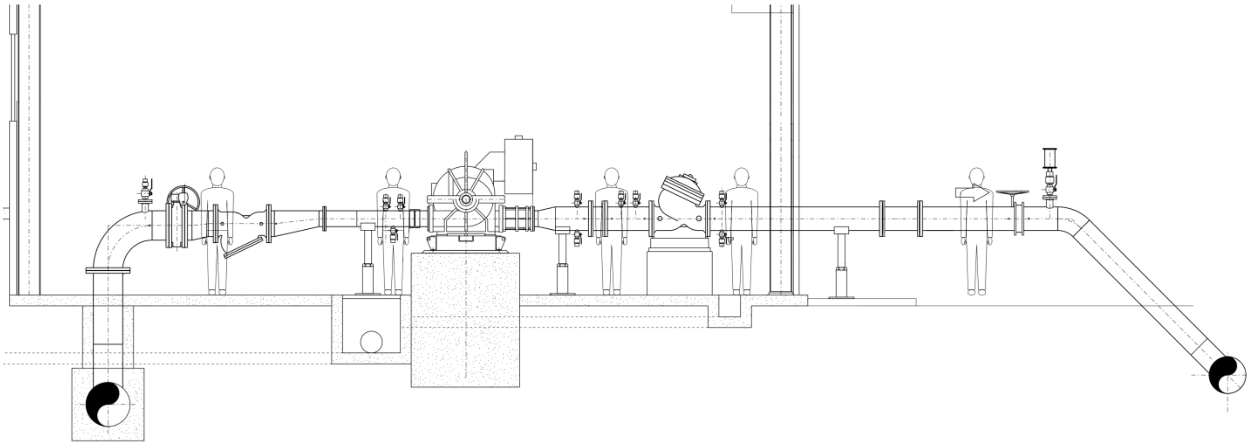


2 PIPELINE PUMP STATION EXAMPLE

Consider a pump station for an 87 km pipeline delivering water from a borefield to an industrial user. The pumping system includes four pumps operating in parallel plus four surge vessels to mitigate transient pressures.

This is an example of the initial piping layout being quite rigid (i.e. it had little inherent flexibility), and it could not be assessed as being suitable without performing some analysis.

Figure 7 Cold Water Pump Station Cross-Section

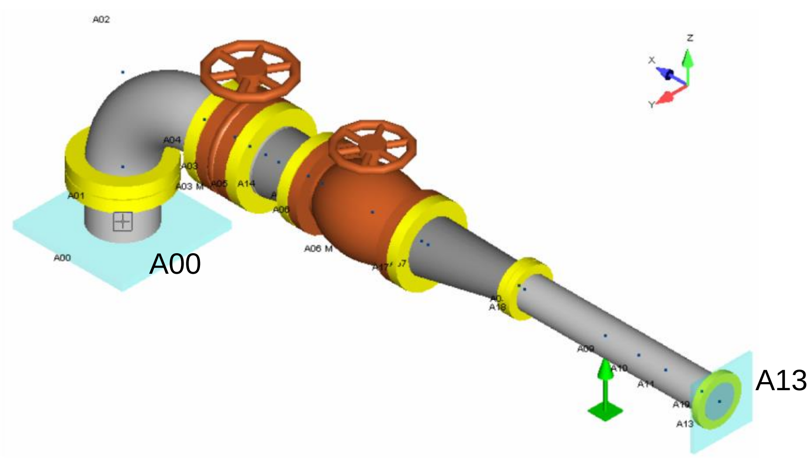


2.1 SUCTION PIPING

The suction piping consisted of a riser from a buried suction manifold, turning through 90°, then with isolating valve and y-strainer connecting through to the pump suction nozzle. The design conditions for this suction line were 25°C ambient and 40°C maximum, giving a temperature change of 15°C.

The pipe stress model for this system is shown in Figure 8 below.

Figure 8 Pump station suction piping



To produce a simple model, the floor penetration at A00 and the pump nozzle at A13 were treated as anchors in the model. The pipe stress analysis results showed that the stresses in the pipeline were acceptable in terms of the design code; however the combined effect of the two anchors, and the thermal expansion in the pipe was to put unacceptable loads on the pump nozzle. The axial force, Fx, on the suction nozzle was 54 kN, some 5 times greater than the allowable value of 10 kN in API 610 (Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries).

Now a critique of the piping arrangement and of the model would raise the arguments:

- *"the floor penetration isn't a true anchor – there will be some clearance around the pipe, and it will be able to move and rotate slightly, and this will relieve load on the pump"*

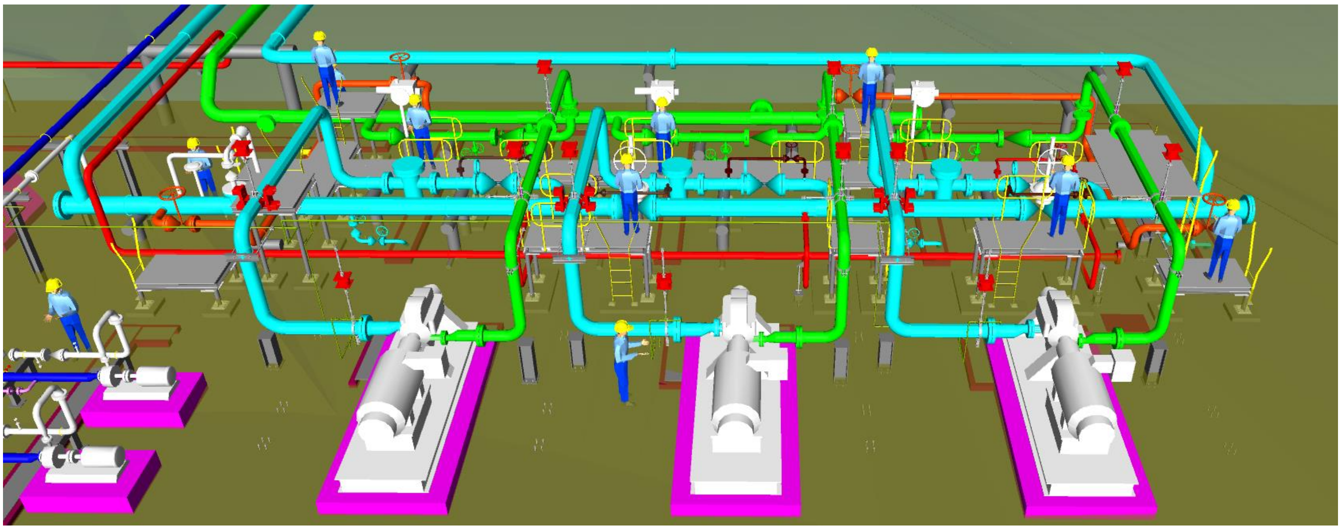
- *"the pump isn't perfectly rigid – it will flex a little bit, relieving the nozzle loading"*

These arguments highlight the difficulty with analysis of cold systems.

When modelling a hot system (for example a steam line), the large thermal expansion requirements mean that the designer always builds in a minimum level of flexibility by way of expansion loops or offsets in order to keep stresses and loads within the allowable limits. The thermal expansion requirements tend to dominate the design of the system. Therefore it is rare to need to consider the flexibility of the floor penetration or the stiffness of the pump body.

Consider the example below for a pump station for hot brine at 200°C – it has extensive piping loops to allow for the thermal expansion without generating excessive nozzle loadings.

Figure 9 High Temperature Pump Station



By contrast, on a cold system, designers rarely make allowance for flexibility and thermal expansion. So in the cold water pump station example in Figure 7 above, the designer now needs to consider the flexibility of each element.

Consider this tag line:

"***The world is a spring.***"

(from pipe stressing engineer, Edward Klein, S&B Engineers & Constructors, Houston, TX)

Essentially, Mr Klein is saying the world is made up of elastic elements, each with its own stiffness value. Some we would think are rigid – for example large blocks of concrete, heavy steel structures, but even these have a finite stiffness value.

Taking the floor penetration, the concrete will be cast around the pipe - see Figure 10. In some cases there may be a clearance. If so, how big is the gap between pipe and concrete to enter into the model? To put this into context, if the pipe at A00 is released to expand freely, the pipe moves 1mm axially. So a 1mm gap between the concrete and pipe means that the pipe can move freely as it expands - see Figure 11. However, this now means that the model has to be extended below the floor to the suction header, and a section of the buried pipe has to be modelled. Buried pipe introduces another level of complexity to the stress model in that it requires knowledge of the native soil, the backfill material and the level of compaction used for the installation. In this case the buried pipe is concrete encased and the concrete will be quite rigid, however the soil restraining the concrete is elastic – see Figure 12. How stiff is the backfill material around the concrete? To determine the inputs to the pipe stressing model requires a number of extra calculations on the stiffness values for the structures and soil properties, followed by additional modelling of the piping system. It also means that if the behaviour of the system relies on a gap or a specific stiffness, then the construction documents must ensure that requirement is clearly stated, and the erection supervisor must ensure that it is complied with.

Figure 10 Pipe through concrete with zero clearance

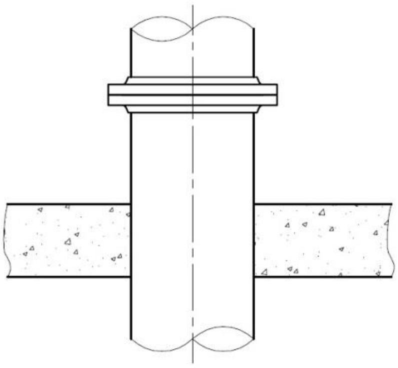


Figure 11 Gap between pipe & concrete

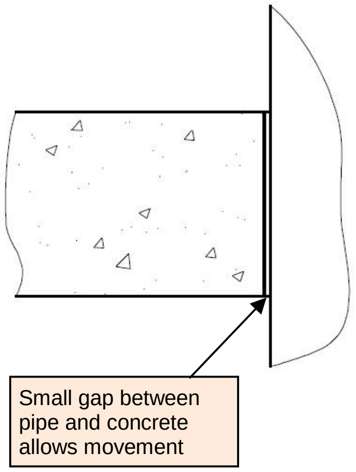
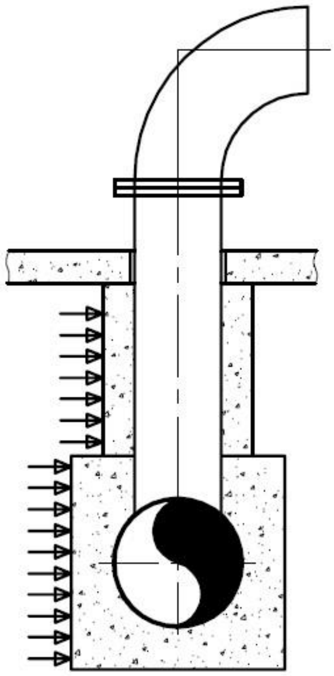
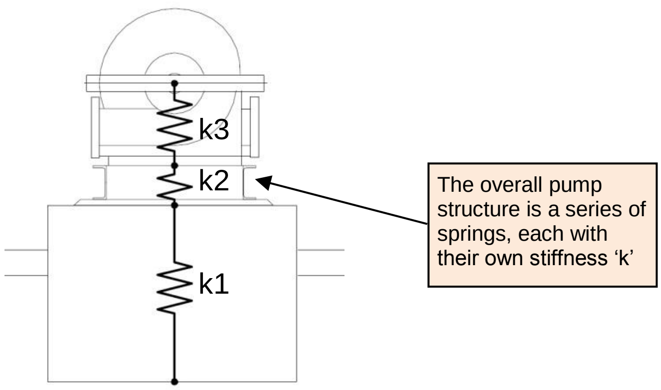


Figure 12 Backfill stiffness affects behaviour



Looking at the other end of the pipe (Node A13 in Figure 8), how stiff is the pump? It is a foot mounted unit on a concrete plinth, so the pipe effectively acts on a beam that is cantilevered from the ground – see Figure 13. The overall stiffness of the beam will be a function of the strength of the concrete plinth, the base plate and the pump casing. How do we determine this value? We can calculate the stiffness of the concrete pump plinth (k1). For the baseplate and pump (k2 and k3), the most appropriate source would be the pump vendor, but they are likely to be difficult pieces of information to obtain. So again we need to carry out additional engineering work to determine stiffness values, and then add these to the pipe stress model.

Figure 13 Pump assembly as a compound beam



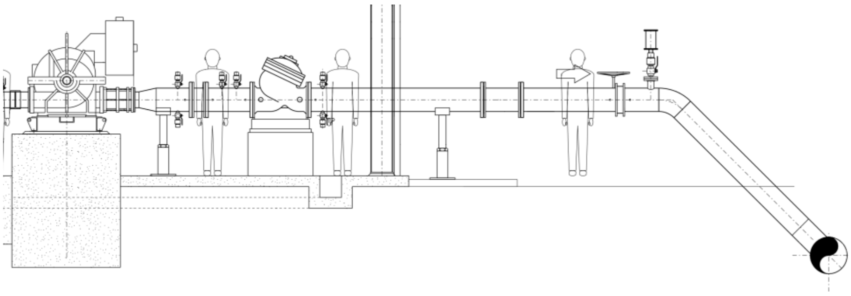
For the steam pipe example, the thermal expansion requirements dominate the analysis, and the stiffness of supports and anchors has a negligible contribution to the overall result, so they are rarely included. In comparison, for the cold water piping system, we are faced with a lot of additional work to determine the inputs for the model, and performing the actual modelling.

So we have just looked at a system without much inherent flexibility. The following example looks at ways of designing a pipe in order to avoid pipe stress analysis or to make the pipe stress analysis much simpler.

2.2 DISCHARGE PIPING

Let us consider the discharge piping for the pipeline pump station previously discussed above.

Figure 14 discharge piping



The original design is compared with two other options. The figure at left is the original concept design – this is the most direct pipe route. The middle option adds some flexibility, while the third option is the most flexible.

Figure 15 Option A

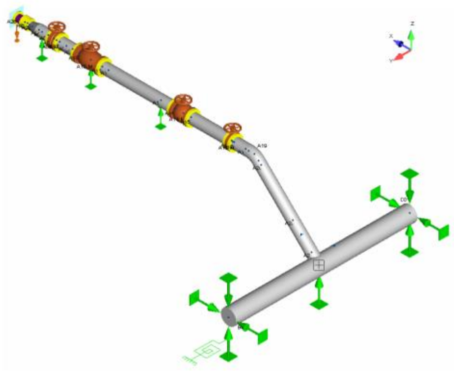


Figure 16 Option B

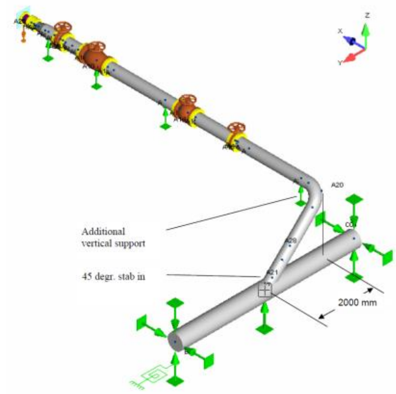
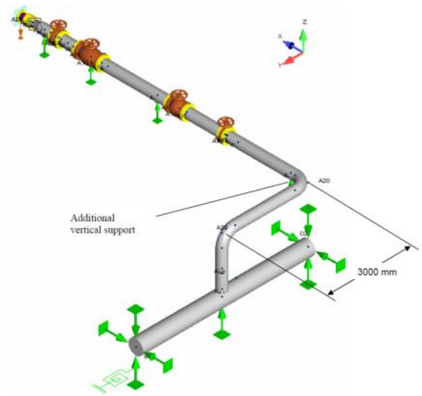


Figure 17 Option C



The analysis modelled three temperature cases:

- Normal Ambient – 25°C
- Operating – 40°C
- Pipe empty – low ambient – 0°C
- Pipe empty – high ambient – 70°C

The comparison between the results from these piping layout options is significant. The point of highest stress for all three cases was the intersection where the branch line connected to the header. The calculated stresses at this point are shown in Table 1 below:

Table 1 Code Compliance Results

Category	**Option**		
	A	**B**	**C**
Code Compliant	No	No	Yes
Sustained Stress Level (code compliance) (gravity + pressure)	205%	212%	97%
Thermal Stress Level (code compliance) (0°C to 70°C range)	249%	413%	93%

The stress analysis results are reflected in the magnitude of loads generated on the pump nozzle – Table 2 below shows loads for the operating case – which consists of Gravity + Pressure + Operating Temperature of 40°C. They are compared against the allowable values from API 610.

Table 2 Nozzle Load Results

	Option			API 610 Allowable
	A	**B**	**C**	
Fx (kN)	27	14	9	10
Fz (kN)	-8	-9	-5	6
My (kNm)	10	-3	-2	7

Option C has resulted in a pipe that has significantly lower stresses, is code compliant and has lower pump nozzle loads – all due to its natural flexibility. The conclusion reached from this is that the use of a flexible piping design can avoid the need for detailed and complex stress analysis. Or does it?

3 PUMP NOZZLE LOAD COMPLIANCE

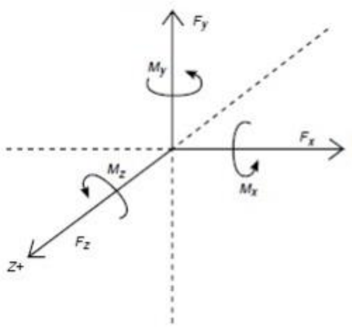
It would be nice to think that by always using a flexible piping arrangement, the need to build and analyse a pipe stress model can be avoided. What is more likely to happen is something like this:

> Client:
>
> "*We are happy with the piping design….provided the pump supplier agrees that the nozzle loads are acceptable.*"

How can the nozzle loads be checked without carrying out a pipe stress analysis? Determination of nozzle loads using manual methods is not a trivial task – it may take longer than using a computerised model. So it is often the nozzle load compliance requirement that dictates the need for a computerised pipe stress analysis.

3.1 WHAT ARE NOZZLE LOADS ON PUMPS ?

Figure 18 Nozzle Axes



Nozzle loads on pumps are the net forces and moments exerted on the equipment nozzles from the weight and thermal expansion of connected piping, and other loadings generated from seismic and wind actions. We consider three forces, Fx/Fy/Fz and three moments, Mx/My/Mz – see Figure 18.

Pumps are rotating machines that depend on good shaft alignment and proper clearances for smooth and reliable operation. If nozzle loads are higher than the acceptable values, they may cause coupling misalignment and casing deformation, which results in increased wear and vibration rates in pumps. Exceptionally high loads could result in the failure of pump supports or the pressure casing.

Pump manufacturers would prefer to have zero loadings on their nozzles, however this is not possible, nor is it practical. Industry standards have been developed which provide sets of loads for specific pump types and nozzle sizes. One such standard is *API 610 Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries*. Whilst ideally these allowable loadings should provide a middle ground between the pump manufacturer and the piping designer, typically these standards will protect the pump manufacturer's interests. This is illustrated by comparing the allowable loads under API 610 with stress due to bending in a STD wall thickness pipe.

Figure 19 (from Peng) plots pipe size against bending moment. A curve showing a constant 6,000 psi stress (41.4 MPa) in a STD weight pipe curves up markedly. The API 610 allowable values follow closely for sizes 2 & 4 inch (DN50 and DN100), however beyond this the API 610 values fall away sharply. This means that as pump and nozzle sizes increase, it becomes more difficult to comply with the manufacturer's requirements.

Figure 19 Allowable Piping Loads (Peng)

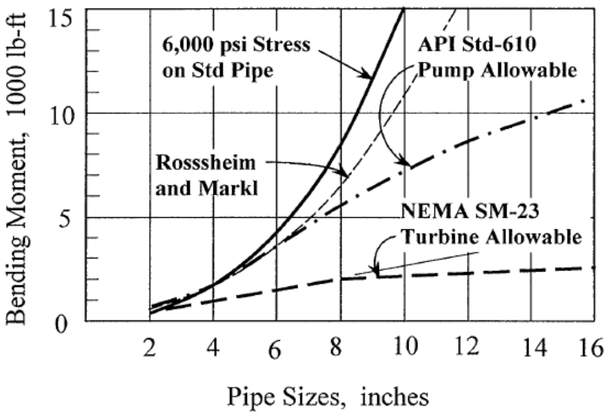
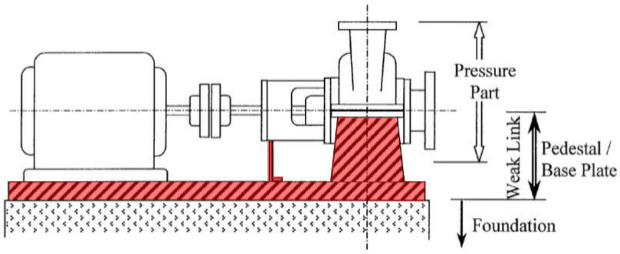


Figure 20 Pump Structure



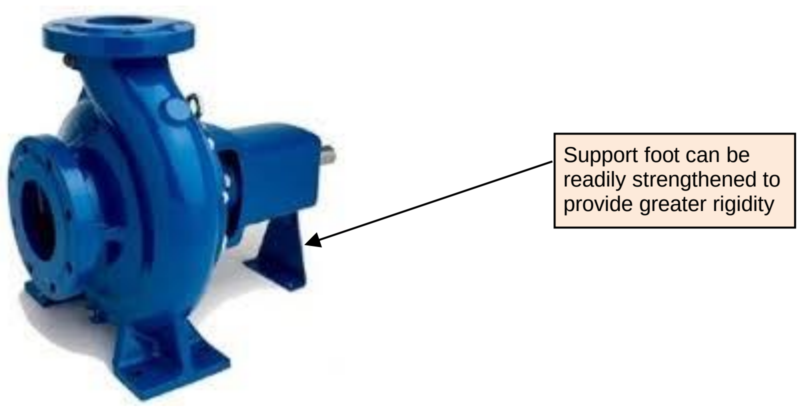
So why are the allowable nozzle loads on pumps so small? Consider the structure of a pump which consists of three elements:

- foundation
- mounting – baseplate, pedestal & foot
- pressure casing

Of these, the weak link is the mounting assembly (shown in red in Figure 20), and fortunately, because this is typically a carbon steel fabricated item, it is the cheapest and easiest part to modify.

Taking the pump in Figure 21 as an example, the support foot is relatively weak but can readily be upgraded to provide greater stiffness - which will limit the shaft misalignment when nozzle loads are applied.

Figure 21 Foot Mounted Centrifugal Pump



Note also that the pump configuration can have an influence on the rigidity and therefore its ability to withstand nozzle loadings. For example a centreline mounted pump is easier to strengthen than a foot mounted pump – consider the overhung type pumps in Figure 22 and Figure 23 below.

Figure 22 Type OH1 Foot Mounted Pump

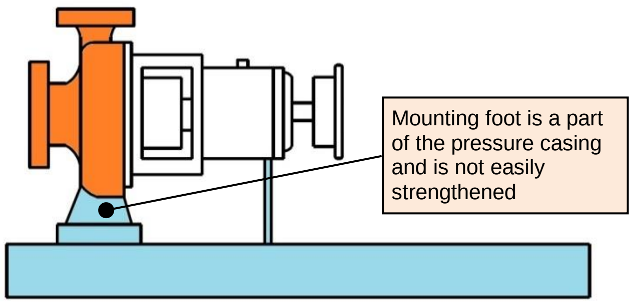
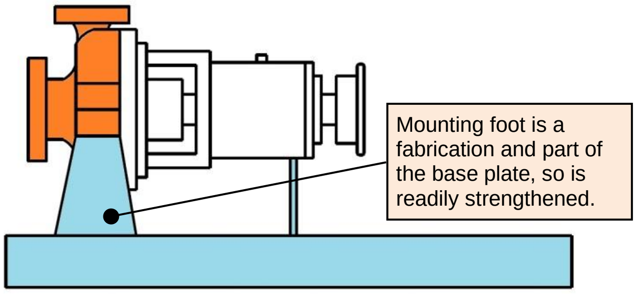


Figure 23 Type OH2 Centreline Mounted Pump



The allowable nozzle loads given in standards such as API 610 are based on compliance for the operating loads – i.e. the loads that act 24 hrs per day, 7 days per week. This raises the issue of what nozzle load values should be allowed under occasional events such as seismic action.

Consider the loading criteria for API 610. The allowable loads are based on meeting two criteria:

a. the pump casing operates without leakage or internal contact
b. the displacement of the pump shaft relative to the driver shaft is limited to 0.25mm

If it is considered that both these criteria should be met under seismic action, then both operating and occasional loads shall meet the tabulated values. However if it is acceptable to relax these criteria for an occasional event, then it should be possible to increase the allowable nozzle load values for seismic action.

This requires an examination of the criticality of the pump and its service. For example:

• is the pump required to operate continuously during and after a design seismic event ? • i.e. a rub between the impellor and casing would not be acceptable	⇒	• no change to the allowable loads for a seismic event • the piping design is likely to require special features and additional flexibility
or		
• is it likely that automatic or manual systems will shut the pump down during a seismic event and that it will be checked and restarted afterwards ? • i.e. that a rub between impellor and casing and/or short term vibration due to shaft misalignment would be acceptable	⇒	• an increase in the allowable loads for a seismic event would be permissible

Once the criteria is determined and agreed, the nozzle loading values should be discussed and agreed with the pump vendor. Note that pump vendors have little interest and incentive to engage in discussions on nozzle loading issues outside the standardised values once an order has been placed. It is therefore essential that this is done before any agreement to purchase the pumps is completed.

It is always worth engaging in dialog with a pump manufacturer to explore options and compliance for nozzle loadings.

4 USE OF BELLOWS OR EXPANSION JOINTS

When piping loads on a nozzle become an issue, often one of the first suggestions is to fit a flexible connection (i.e. an expansion joint or flexible bellows). These are commonly either rubber or metallic in construction.

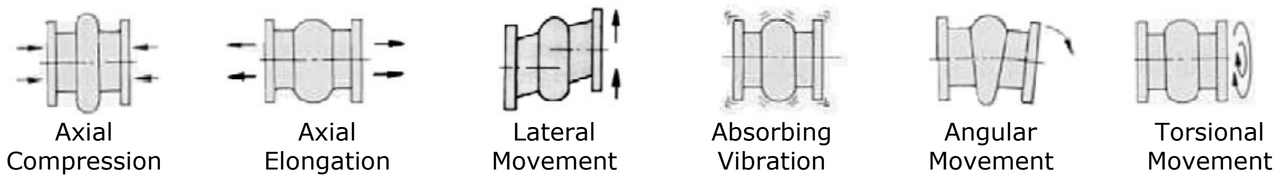
Figure 24 Rubber Bellows

Figure 25 Metallic Bellows

Bellows are one of the most misunderstood components in piping systems. They come in many configurations – they can be tied, un-tied, hinged, gimballed, compensated, and combined into multiple units.

A convoluted bellows is theoretically capable of the following movements:

Figure 26 Available movement from bellows

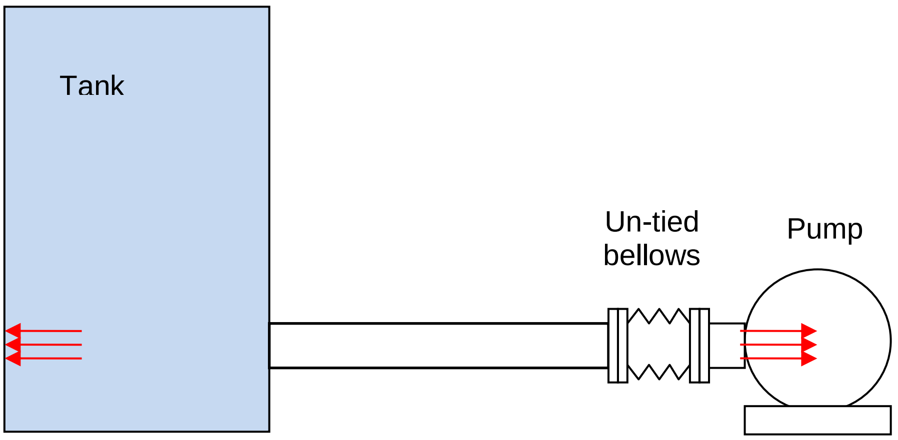


However some of these movements are limited by the configuration of the bellows.

4.1 UNTIED BELLOWS

The simplest implementation of a bellows is the untied unit – simply a flexible element between two flanges. A suitable location for this would be on the suction line for a pump running from a chest or tank with a low static head. Being untied, the bellows can absorb axial compression & elongation, lateral, angular and torsional movement. The characteristic that designers need to consider with an untied bellows is pressure thrust. Having nothing to tie the flanges together means that all of the internal pressure thrusts are passed through to the connected pump. Whilst technically they don't act on the nozzle, they do act on the internals of the pump and create a force which applies a moment about the pump's baseplate, and therefore need to be considered in the overall summation of forces acting on the pump.

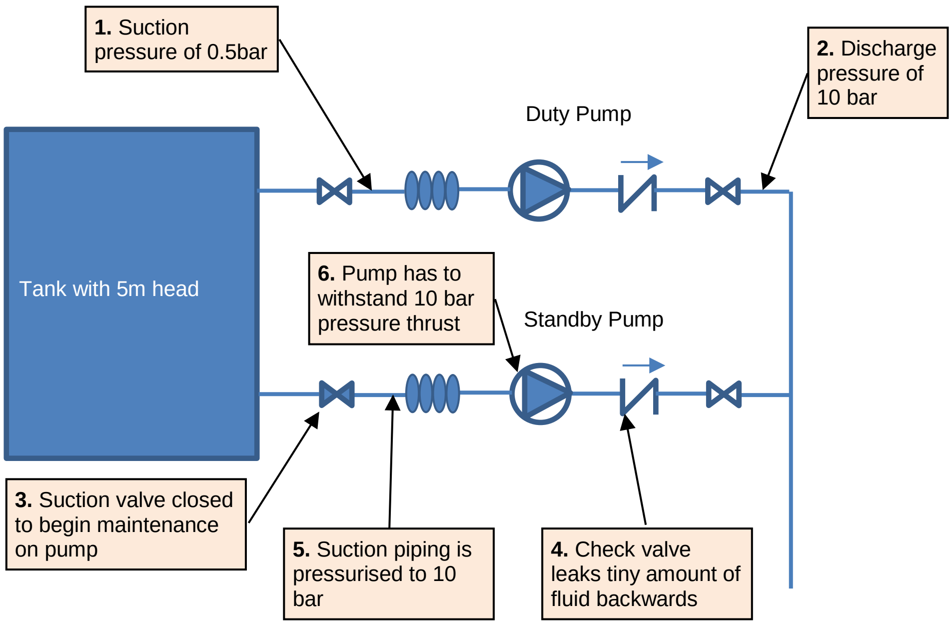
Figure 27 Pressure Thrust from Untied Bellows



With a single pump installation and low static head, the pressure thrust forces are likely to be manageable in terms of the pump design and the untied bellows is a sensible solution.

However if the pump is installed in parallel with other pumps, and it includes a suction isolation valve, then further care is necessary – refer to Figure 28 below.

Figure 28 Suction Pressure Due to Leaking Check Valve



In this example, if the suction isolation valve on the standby pump is closed, the suction line can be pressurised to the discharge header pressure. The pressure thrust on the pump from the untied bellows increases to 20 times its original design value. The designer needs to ensure that the pump can withstand this force, or provide pressure relief to ensure that the suction line cannot be exposed to discharge header pressure.

If the pump is in an application such as a booster station with high suction line pressures, then the untied bellows is unlikely to be a practical solution due to the large loads that will be put on to the pump body. For example a DN200 pipeline at 10 bar generates a force of 31.4 kN.

4.2 OTHER BELLOWS CONFIGURATIONS

For restrained bellows, there are three common configurations:

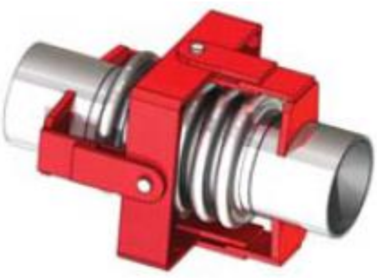
Figure 29 Tied Bellows

- resists pressure thrust
- allows for lateral movement only
- not designed for angular

Figure 30 Hinged Bellows

- resists pressure thrust
- accommodates angular movements in one plane only
- does not allow torsional

Figure 31 Gimballed Bellows

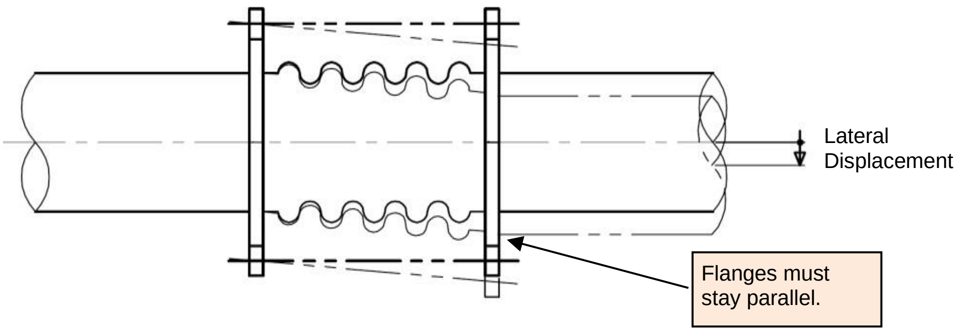


- resists pressure thrust
- accommodates angular movements in any plane
- does not allow torsional

movement • will allow torsional displacement • requires lock nuts both sides to withstand compressive and tensile loads	displacement • can carry loads from dead weight of piping and equipment and externally applied forces	displacement • can carry loads from dead weight of piping and equipment and externally applied forces

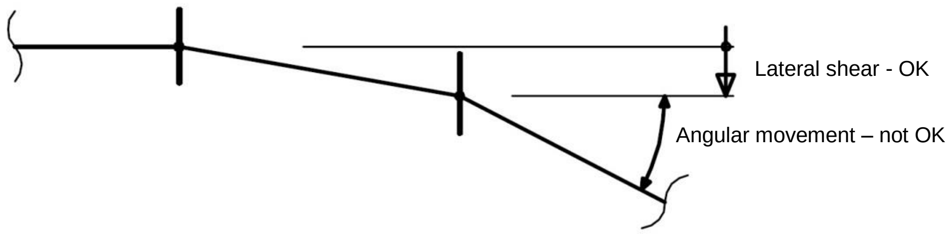
The most commonly used of these is the tied bellows. These should be used for lateral movement only. For example:

Figure 32 Lateral movement of bellows



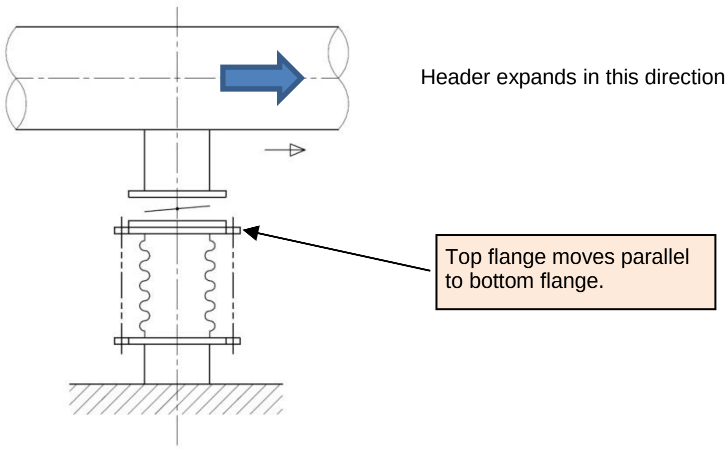
Note that this must be a parallel movement of one end with respect to the other. The lateral displacement cannot be combined with rotation of one pipe (angular movement).

Figure 33 Shear vs. angular movement



An application where tied bellows might be used is for the blower air supply to a filter for backwashing. In Figure 34 the header expands, while the connection to the structure is fixed.

Figure 34 Tied bellows in blower piping



A common misapplication is to fit a tied bellows to a pump nozzle with the expectation that it can absorb axial thermal expansion. As soon as any thermal expansion occurs, the tension in the tie-rods reduces and the pressure thrust load is transferred to the nozzle.

> "*But it's only 1mm of thermal expansion !*"

So is there enough of a spring effect in the bellows to absorb the thermal expansion? Essentially the components of the bellows are a compound spring. It is very complex task to calculate these loads with any accuracy because they rely on being able to model the stiffness of the flanges, plus the extension arms that hold the tie-rods and the tie-rods themselves. This becomes a wonderful (or perhaps indulgent) exercise for a pipe stress modeller and a terrible result for a project manager. It is only valid if it is modelled with the same initial setup of the nuts on the tie-rods as the actual installation. And it is unlikely that the setup would be repeatable unless a very detailed process is followed.

4.3 APPLICATION OF BELLOWS

So bellows should be used only after careful consideration of the manner in which they are being used. And to do this properly requires a concept – a flexibility concept.

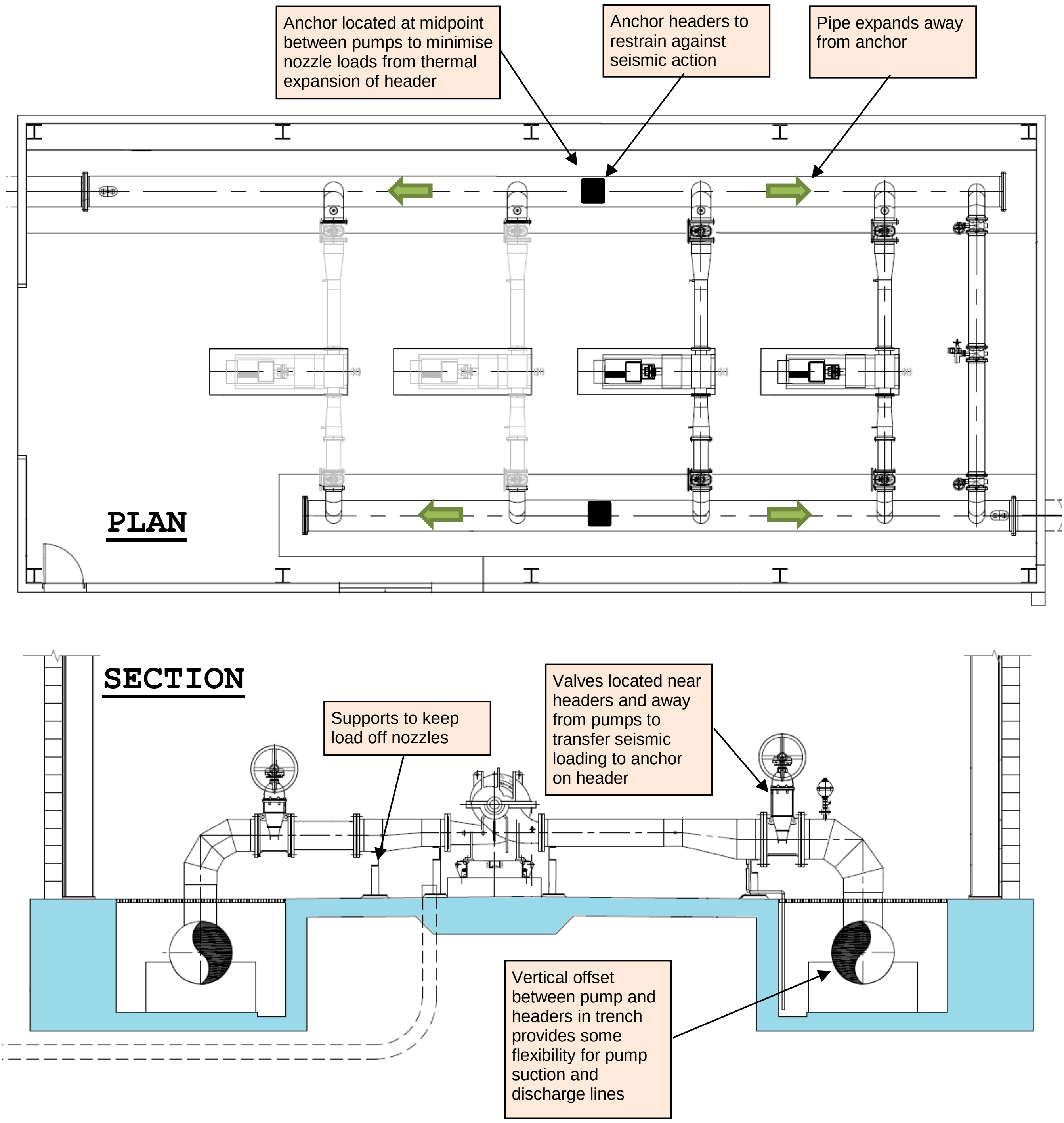
5 FLEXIBILITY CONCEPT

When preparing a design for a pump station, it is essential to determine all of the design inputs – the temperatures, pressures, seismic actions, hot/cold temperature cases, space limitations and any other constraints that may influence the layout.

Then develop an appropriate flexibility concept. This involves considering how each of the design inputs will be managed. If the pipe expands due to an increase in temperature, how will the expansion be absorbed? Which parts can flex or bend? And which are very stiff and will not bend? Where will the seismic loads be restrained? Where will the pipe supports be located and what function will they have?

Figure 35 below shows how the thermal expansion, seismic restraint and flexibility are to be managed in this concept design of a pump station.

Figure 35 Flexibility Concept for Pump Station



6 CONCLUSIONS

Whilst the low temperature of a water or wastewater pump station initially suggests that pipe stress analysis is not necessary, other factors can come in to play that require a formal analysis. Problems can occur when flexibility and restraint is not considered early in the design process. This can result in overloaded nozzles on sensitive equipment. Bellows should be used with care, and only after confirming that they will serve the intended purpose.

The design process should begin with the preparation of a well-considered flexibility concept.

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