Modelling of Viking Johnson Couplings in CAESAR II

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Introduction

Viking Johnson (VJ, www.vikingjohnson.com) couplings and flange adaptors are cost effective solution for low pressure, low temperature applications where line flexibility is required. This coupling technology is not new but I have found that they are not fully understood. As a result they are either not used or the lines are incorrectly designed or erected.

Often low pressure lines where couplings are used do not require very detailed design calculations. Same applies for VJ couplings and flange adaptors. Reading and following the supplier's technical literature most applications can be easily designed. However there are applications where more detailed analysis is beneficial. Typical such application is PF Pipes. These are pipes that connect coal mill to burners at the boiler. Lines are low pressure lines but one end of the line has large forced movement. Typically some 150 to 200 mm downwards and 40 to 100 mm horizontal. Operating temperature is both sides of 100° C. Flange adaptors or couplings can be used to compensate these movements.

In a low pressure low temperature line stresses are normally low and they are easy to calculate or estimate. Using VJ couplings and adaptors with correct design the stresses and also pipe end forces can be substantially reduced. Traditional stress analysis is largely stress oriented. For a pipeline where VJ are used this is not the case. Far more important are angular movements and axial forces.

CAESAR II documentation or any known publications that I know of do not describe how to model VJ couplings. COADE newsletter June 1998 describes Victaulic Coupling modelling but this cannot be used for VJ couplings because of the different coupling design philosophy.

This document describes some basic steps and methods how to model VJ couplings and flange adaptors and also gives some ideas for line design. More advanced options are also discussed.

Description of the couplings

Flange adaptor can be considered like half of a VJ coupling. The following paragraphs refer to flange adaptors. By reading the supplier's technical literature it is easy to understand couplings once you figure out the adaptors.

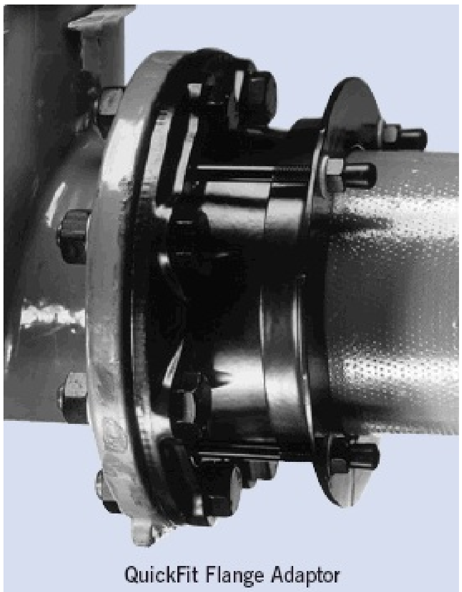

Figure 1 *QuickFit Flange Adaptor by Viking and Johnson*

Flange adaptor looks like a flange with enlarged hub and some extra bolts.

Flange adaptor cross sections shows the most important design features.
These are:

- There is no metal to metal connection between the pipe and the adaptor
- There is a gap between the flange face and the pipe end
- Connection between the flange adaptor and the pipe is by a rubber seal, which is compressed by bolts and a loose ring.

Because of these design features flange adaptor can rotate and allows small axial movement. Rotation is limited to 3° and axial movement to 5 mm. Both are from neutral position. Larger axial

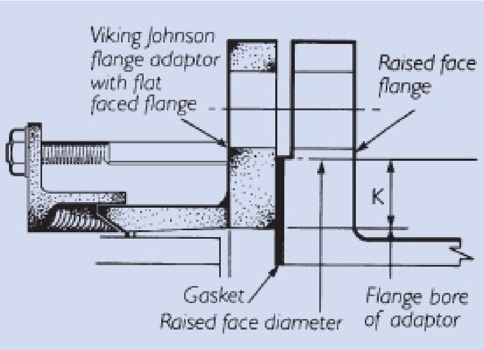

Figure 2 *Cross section of a flange adaptor.*

movement is possible but should not be allowed. Axial movement up to 5 mm is achieved by deformation of the rubber seal. After that seal will slide and most probably will get damaged.

No lateral movement is possible using one flange adaptor. Torsion is not allowed.

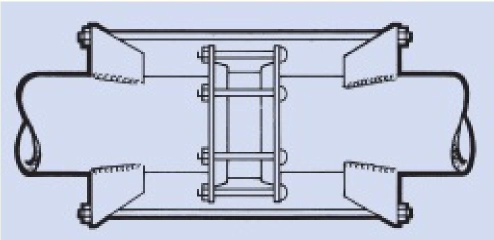

Figure 3 *Typical solution to carry pressure thrust*

It is important to understand that flange adaptor will not carry pressure thrust. It has to be carried by additional devices or anchors in the line. Anchors have to be in regular intervals. Manufacturer's technical literature gives exact details. Pressure thrust can be calculated using formula:

$$F = \frac{PipeOD^2 * \pi}{4}$$

Instead of solid rods chains can be used also. For a pipe designer it is important to understand that it is not only pressure thrust that has to be looked at. Compression forces from external sources or from thermal expansion have to be looked at also. Calculated axial movement in either direction must not exceed the 5 mm limit!

Line design with large movements

VJ couplings and adaptors have limited axial movement capacity. It is not sufficient to compensate large movements. It is a good solution for instance for overland pipelines where either ambient temperature or fluid temperature changes require some compensation in axial direction of the pipe. Such design can be calculated manually using manufacturer's literature.

For large movements different approach is required. Instead of the axial compensation the line design is using the rotation possibility of the flange adaptors. Connecting two flange adaptors with a pipe between we have a cost effective "universal compensator". For this to work we need change of direction in our line.

In the image you can see four flange adaptors. The high end of the line moves down and left. In addition there is normal pipe expansion but it is

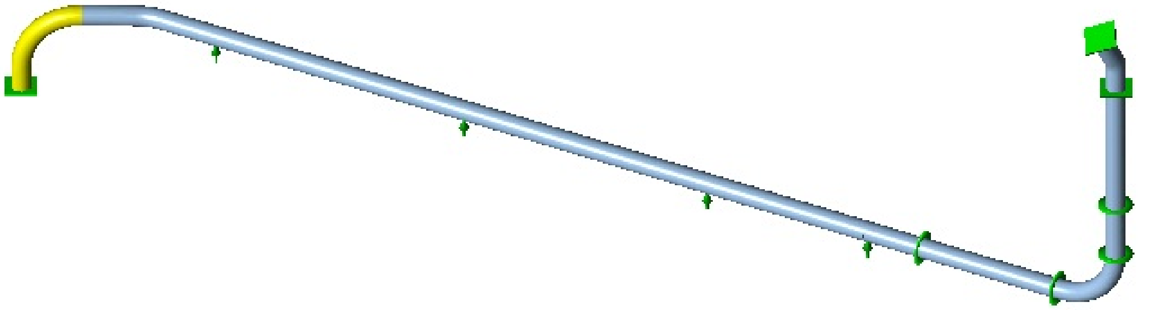

Figure 4 *Typical PF line for a small power station. Left end of the line is the mill end. Flange adaptors are the round rings. Note that there are no supports between the first and the fourth adaptor. These items must be free to move.*

less than the forced movements. Downwards movement is compensated by the horizontal pair of adaptors and the pipe between. Horizontal movements are similarly compensated by the flange adaptors and pipe between in the vertical section. How this works? Very simple. Flange adaptors are rotating and the pipe between is doing the same. Rotation is limited to 3° but increasing the length of the pipe between large movements are possible.

Face to Face	Maximum movement
1000	50
2000	100
3000	160
4000	210
5000	260
6000	310
7000	370
8000	420
9000	470

1 *Maximum movement in mm that pipe spool with two flange adapters can compensate without cold pull.*

Force required to rotate is low compared to axial compression of a flange adaptor. As a result minimal axial compensation in the adaptors will happen.

You need to remember that flange adaptors are not designed to carry axial load. Adaptors in the vertical line must carry the pipe mass below plus the pressure thrust. These adaptors require chains or rods to carry this load.

To increase the compensation capacity cold pulls can be used. It is generally beneficial that during the operation load case flange adapters are near their neutral position. Erection tolerances and temperature differences can be substantial. Having the adaptors near their neutral position is good as they have their maximum movement capacity still available.

Modelling in CAESAR II

So how to model the flange adaptor in CAESAR II? Simple, use expansion joint modelling.

The most simple approach is to use gimbal compensator modelling. Referring to CAESAR II document Application Guide Chapter 5 you model each flange adaptor individually. For this modelling CAESAR II requires five entries.

In the most simplified case you enter high axial stiffness. This will ignore any axial compression or extension of the adaptor. The error is minimal as the possible axial movement is small compared to other movements. If you do this you have to check the axial forces at each adapter after calculation to make sure that you do not exceed the maximum allowed force, which is the force when the seal slips. You do this by checking the global (or local) forces at the fitting. We discuss later the maximum allowed force and how to get it.

As a translational stiffness you enter high value. Lateral movement within single adaptor is not possible. If you like to use scientific approach you can calculate radial stiffness of the pipe, increase the calculated value by about 30% for the reinforcing effect of the adaptor and use this value. After the calculation you should check the lateral force and compare it to manufacturer's technical literature. They do not give maximum allowed force in Newton but you can calculate this from their comments.

Bending stiffness entry is the most important value. We discuss later how to get this value. After the calculation you need to check that in your design none of the adaptors have larger than 3° rotation in any of the load cases including one after the cold pull.

As torsional stiffness you enter relatively high value. Your design has to be such that the torsion is eliminated as much as possible and therefore this value is not critical. How high torsion moment is actually allowed is not published by the manufacturer.

Effective ID is the outside diameter of the pipe.

This simple modelling is fast to model but requires substantial evaluation of the results to make sure that the axial force or angular movement limits are not exceeded.

Complex modelling in CAESAR II

More complex modelling than the one described above is possible in CAESAR II. I will discuss some of these possibilities using the pipe in the figure as a example.

Firstly we will use the finite length modelling. Is this better is highly debatable. I discuss the problems a little bit later. However there are some critical issues, which have to be noted if this modelling is used.

First issue is what length to use. I have seen 50 to 60 mm used. I use 10 mm.

Secondly I will discuss the expansion joint entry values.

In the simple modelling we used high axial stiffness. We can improve this. It is important to remember the maximum axial movement capacity. Our design has to include suitable protection. Referring to our example line in the vertical section we would use chains or rods to carry the axial force. We do not need to worry compression here because there is no force to lift the pipes. Fittings have to be free to rotate so we will use two chains. For practical reasons we can assume that they are not tight when they are erected.

To model our chains and adaptor stiffness we use CAESAR II expansion joint modeller together with restraints. Firstly we enter axial stiffness of the adaptor into the expansion joint axial stiffness entry. How to get this value we will discuss later.

Modelling of the chains is easy using the restraint. We use Y or +Y connected to the lower end node of the finite length expansion joint and Cnode to the upper node. Chain not being tight we can enter 4 mm as a gap and then we can give some stiffness value also. It is difficult to design fully rigid chain with 2 pipe attachment points. We will use a generic 10 kN/mm stiffness.

In the horizontal pipe the modelling is very similar to vertical. There are two issues you need to consider. First is do you have protection for axial compression in your design. If you use chains the answer is no. In this case you need to check after calculation your global forces. Second is the direction of the restraint. You can give direction to restraint if necessary, e.g. pipe axis is not in X, Y or Z.

If you use rods with nuts on both sides of the fixing point you can model these same way as control rods. You may use gaps for the nuts. Remember to orient the rods correctly and do not use any RX, RZ or RZ restraints.

You have to enter translational stiffness. Use the same value as you would use for simple modelling. If you do not enter the value CAESAR II tries to calculate it and it will be wrong.

CAESAR II documentation indicates that if you enter translational stiffness you should not enter the bending stiffness. Not in this case. If you do not enter value CAESAR II will calculate the value using formula:

$$K_b = \frac{1}{3} * \frac{K_{tr} * L^2 * \pi}{180}$$

This gives a correct answer for metal bellows but not in this case. You have to enter correct bending stiffness.

For the metal bellows above formula calculates bending stiffness. Normally you have value for bending flexibility. You can find explanation for these two terms in COADE documentation. The difference between the two values is that the bending stiffness is four times the bending flexibility. Calculation tests I have done (Version 5.1 initial release) indicate that we need to enter bending flexibility. This is commonly known as bending spring rate. COADE is busy investigating this issue.

To further improve the model you could use rotational restraint with 3° gap and stiffness. However I am not recommending this. You have to limit the maximum movement to 3°. Exceeding this limit may damage the flange adaptor. After consultation with the manufacturer you may reconsider. If you enter stiffness for the restraint the value will be less than your pipe bending stiffness, see cross section of the adapter. Rotating pipe will connect inside the adaptor and the ring at the end.

Torsional stiffness and effective diameter entries are same as for the simple modelling.

Now we have still a problem what are the correct axial and bending stiffness. We look into this next.

VJ axial and bending spring rates

As you can see I use terms spring rates. Such terms are commonly used for flexible items.

Manufacturer doesn’t publish these values. They depend on pipe diameter, how tight the rubber seal tightening bolts are and possibly into some extend what seal material is used.

The best method to get these values is to test them. Time consuming and expensive.

Viking and Johnson was very kind and gave me force that will result in seal slippage. Their tests show that on average 0.2T per pipe diameter inch is the value to use. Using this value and NB 400 pipe (16") we can calculate: 16 * 0.2*1000*9.81 = 31 392 N or about 32 kN. Considering that the slippage happens after 5 mm movement we will get axial spring rate of 6400 N/mm. These values are maximum values that can be achieved in test conditions. To be conservative I would not use the maximum values. To be conservative we could use for instance as maximum allowed axial force for NB400 pipe 16 kN and as axial spring rate of 3200 N/mm. This will result in larger axial movement, which we have to check. It must be below 5 mm!

Bending spring rate is more difficult. There are no test results available. Such tests have to be done sooner or later.

For the calculations we need a value, which has correct magnitude. Formula given for metal bellows seems to give this:

$$K_b = \frac{1}{8} * \frac{K_{ax} * D^2 * \pi}{180}$$

Using NB400 pipe OD and axial spring rate 3200 N/mm we can calculate the bending spring rate (bending flexibility) as 1150 Nm/degr.

To check if the magnitude is correct I used the following:

Assuming a pipeline is erected using 6 m long NB400 pipes with 4.5 mm wall thickness would pipe of its own mass rotate after connected to fixed flange adapter. The answer is yes. Now we can calculate the maximum bending moment at the end of the pipe as 7870 Nm. Using 3° movement our maximum spring rate would be 2630 Nm/degr, which is same magnitude as we calculated before.

Not the most scientific method of establishing the important bending spring rate but it is best what I can do without the tests.

Discussion

Low temperature low pressure lines where you would use flange adaptors and couplings are not the critical lines to ASME B31.1 or ASME B31.3. The most important with these lines is to get the erection correct. Wrongly erected lines will leak and will not work. Erection must not be underestimated. Especially if you specify cold pulls the erection becomes complicated. Your design calculations will help to establish correct dimensioning and if cold pulls are required.

Calculation method as described is accurate as long as you can establish reasonably accurate bending stiffness.

What is the error and risk if the bending spring rate is wrong? It has small or no impact on calculated movements and generally very small impact on pipe stresses. Line stresses are generally low and some increase would not be an issue. So what is left? Pipe connection forces and moments to equipment.

If you do not have tested bending spring rate the first is to use the above method and then make test calculations to establish what error in the value would result in too high risk at the end connections. Based on the test calculations you can then make a decision if tests are required.

Finite length modelling

Finite length expansion joint modelling system in CAESAR II is designed for metal bellows. It has built in formulas, which do not necessarily work with VJ couplings or any other compensating element that works differently to normal metal bellows.

First issue is that the program was designed so that you enter only bending or translational stiffness. The one that is left out is calculated using formula applicable for metal bellows. Only in very rare cases you get correct results if you enter only one stiffness. Length of the compensator has impact in these calculations. For define length compensator you should normally enter bending stiffness and not bending flexibility (spring rate). In my test calculations some results have been horribly wrong using define length. Results include forces in wrong direction, 180 degree rotation etc. The following finite length combinations have worked:

- short 10 mm length and entering bending flexibility
- 100 mm length, translational entry left out and bending stiffness entered

In one specific case first option gave within 0.5% same results as zero length option and the second within 5%.

Zero length modelling has always worked so I recommend to use it instead of finite length.

Conclusion

Using VJ flange adaptors is an effective method to compensate large movements in low pressure low temperature lines. CAESAR II gives excellent possibilities to model them. You can use very simple modelling, which requires detailed manual evaluation of the results but you can use also comprehensive modelling, which reduces the manual checking.