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# TREATMENT OF SUPPORT FRICTION IN PIPE STRESS ANALYSIS

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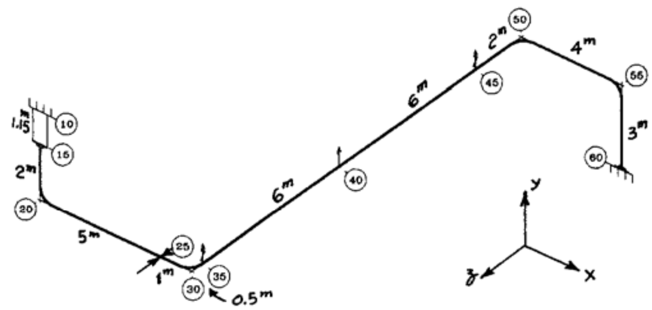
## ABSTRACT

The friction force at the pipe support has a significant effect on the behavior of a piping system. Just like an analysis without including the restraint effect, an analysis without including the support friction may be meaningless in some cases. The treatment of support friction in the pipe stress analysis is not yet well defined in practice. This paper will try to outline the procedure to be used in the analysis. The paper first presents a typical problem to show the significance of the support friction. It then discusses some techniques used in the inclusion of the friction in a computer program. Detailed discussion is given in the arrangement of the analysis to comply with the piping code requirements of separating the sustained stress from the self-limiting stress. Special treatment of wind and earthquake loads are also discussed. The paper deals only with the static aspect of the analysis.

## INTRODUCTION

Support friction in a piping system can prevent the pipe from free expansion thus creating a higher stress in the pipe and a higher load on the connecting equipment. However, in certain instances the friction can help stabilize the system and reduce damage. Even in dealing with pure thermal expansion, the friction can serve as guides thus preventing a large load from being transmitted to the rotating equipment. Therefore, there is no rule of thumb as to whether it is nonconservative to ignore the friction. In general, when dealing with the dynamic load, the friction tends to reduce the magnitude of both the pipe stress and the equipment load. In this case, the omission of the friction is conservative. However, there is no general rule governing the static load. In this case, the effect of the friction need to be investigated to simulate as closely as possible the real situation.

The effect of the friction is more important in some areas. In the analysis of the long transmission pipeline [1], it is entirely the balancing of the friction force against the potential expansion force. Without including the friction the analysis would have been meaningless. Another area of importance is the piping connected to the rotating equipment. The rotating equipment is notorious for its low allowable piping loads. Sometimes the friction at one support can completely change the acceptability of the piping system. Take the system shown in Figure 1 for instance. The restarint at 25 is installed to protect the compressor at 10. The effect of the friction at 25 is demonstrated by comparing the analysis results of the case with friction against the case without the friction. It is clear that the friction at restraint 25 is significant. By applying API STD-617 [7] criteria, only the load calculated with the restraint but without the friction is acceptable. The API criteria is evaluated separately and is not included in this paper.




Pipe Data : 323.9 mm O. D. (12" nominal), 9.5 mm tk (Std), 150°C, E=192360 MPa,
exp rate = 1.53 mm/m, wt=75.5 kg/m
friction factor at 25 = 0.4

Figure 1, Effect of Friction on Compressor Piping

Table 1, Pipe Load at Discharge Nozzle Flange 15

| Condition | Forces (N) | | | Moments (N-m) | | |
|---|---|---|---|---|---|---|
| | Fx | Fy | Fz | Mx | My | Mz |
| No Restraint | -1145 | -3727 | 3227 | -4519 | -10152 | -1829 |
| Restraint Without-Friction | -1485 | -3913 | -289 | 1143 | -17 | -2988 |
| Restraint With-Friction | -4735 | -3397 | -339 | 591 | 206 | -7308 |

By including the friction in the analysis, the designer will appreciate the requirement of using low friction type sliding plates or struts. Some might think that low friction sliding plates should have been used in the first place. The truth is that the friction is very often needed for the smooth operation of the machine. It stabilizes the piping and dampens out the potential vibration. Furthermore, the popular low friction sliding plate adds a considerable problem in the operation and maintenance of the plant.

## NON-LINEAR RESTRAINTS

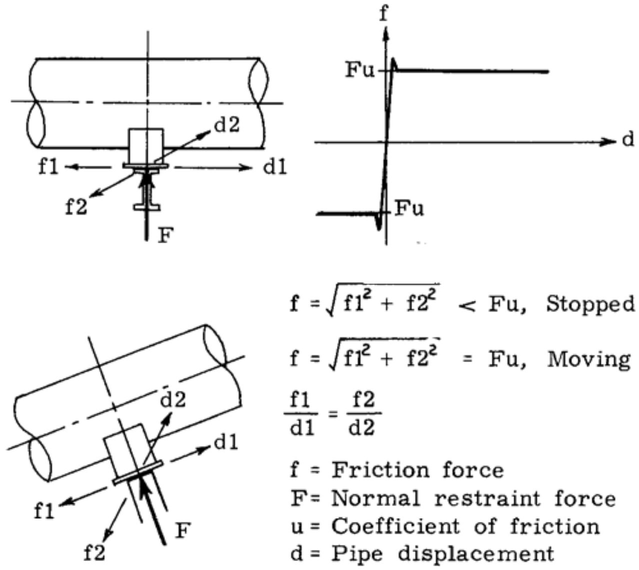
In a finite element computer program the friction is handled by the friction element. However, to make the input more efficient and the interaction more direct, the support element and the friction element are often combined into one three dimensional interface element [2]. In piping it is called by the general term, Non-Linear Restraint [3].

The non-linear restraint defines the restraint direction which is perpendicular to the sliding surface. For a non-linear restraint to be able to include the friction, it has to have the capability to perform the functions as shown in Figure 2, and as described in the following :

(1). Create a friction vector in the sliding surface. Normally this is defined by two local mutually perpendicular vectors which are perpendicular to the restraint direction. In a Y-direction restraint, the friction vector is to be determined by the X- and Z- vectors.

(2). If the potential friction force is sufficient to stop the pipe from moving along the restraint surface, the pipe will be stopped. The resultant friction force created is less than the potential friction force. It is equal to the force required to elastically stop the pipe from moving.

(3). If the potential friction force is not large enough to stop the pipe from moving along the restraint surface, the pipe will move. The resultant friction force created is the product of the normal restraint force and the coefficient of friction. The friction applied to the pipe is directly opposite to the pipe movement.




$f = \sqrt{f1^2 + f2^2} < Fu$, Stopped

$f = \sqrt{f1^2 + f2^2} = Fu$, Moving

$\frac{f1}{d1} = \frac{f2}{d2}$

f = Friction force
F= Normal restraint force
u = Coefficient of friction
d = Pipe displacement

Figure 2, Friction Restraints

The potential friction force is the product of the restraint normal force and the coefficient of friction. In the calculation, a small displacement is assumed to be required to develop the full potential friction. This displacement is taken as so small that its existence will not affect the result of the analysis.

Theoretically both static and sliding friction coefficients have to be used. However, even if the translational pipe motion is being stopped, the rotation and jerking of the pipe make it difficult to maintain the static coefficient. In practice only the sliding or dynamic coefficient is used.

In addition to the friction handling capability, a non-linear restraint can also handle gap, initial load, and plasticity of the restraint element.

## PIPING MOVEMENTS

When a piping system expands, its movement at a given support is not likely to be in a straight line. Therefore, the friction at the support does not maintain the same direction throughout the whole expansion process. The situation is even more pronounced in a system which is restrained by limit stops. The pipe can start out in one direction, then make a sharp change after reaching the stop. In this case, the direction of the friction would also have to be adjusted constantly throughout the expansion process. This type of analysis can be done with a series of analyses at incremental steps. At each step the expansion is increased by a certain increment with the friction force balanced at the end of each step. The force and moment at each step are recorded and enveloped to ensure that the most severe result is obtained.

Although it is preferable to perform the incremental analysis to ensure that no extreme load is overlooked, the current practice is to make a one step analysis. The pipe at the support location is assumed to move in a straight line from the initial position to the final operating position. In this way the friction is applied based on the final displacement. All the intermediate displacements are ignored, because their existence is temporary in nature. However, sound engineering judgement should be exercised to see if a more elaborated analysis is justified.

## COMPUTER IMPLEMENTATION

The concept of the friction element is clear, but the computer program implementation can be different from one software package to another. For the sake of explaining the implementation detail, a general discussion on the static problem solution procedure is in order. The static pipe stress problem is solved by first assembling the equilibrium equation (1).

$$[K]\ X = F \qquad (1)$$

Where, [K] = Stiffness matrix of the piping system
X = The unknown nodal displacement vector
F = The known nodal load vector

The load vector F includes weight, thermal initial load, pressure, external force, and so forth. The unknown displacement can be solved by the inversion of [K], or most likely by the decomposition of [K] as in Equation (2).

$$[K] = [L]\ [D]\ [L]^T \qquad (2)$$

Where, [D] = A diagonal matrix
$[L]$ and $[L]^T$ are unit triangular matrices being each the transpose of the other.

The equilibrium equation is then solved by letting

$$[L]^T\ X = Y \qquad (3)$$

$$\text{or} \quad [L]\ [D]\ Y = F \qquad (4)$$

Where Y is an intermediate solution vector which is being used as a bridge of the solution. A forward substitution is performed on Equation (4) to solve Y. This Y is then used in Equation (3) to solve the displacement X by back substitution. The decomposition step in Equation (2) takes a lot more computer time than the substitution steps in Equations (3) and (4). This makes the avoidance of the decomposition step highly desirable.

The computer program implementation of the support friction can be categorized into three groups being discussed in the following. They all use the interative approach, but each group has its strong and weak points. Some emphasize the saving of the computer time, while others are more concerned about the convergence and stability. In the long run, the idea originally intended to save computer time might end up using more computer time due to unexpected slowness in convergence. A scheme has very little practical value if it does not converge, or if it is not stable. The following are the detailed discussion on the chracteristics of each scheme.

### (1). Direct Substitution of Friction Force

The most simple method is the direct substitution of the forces expected from the friction. The analysis starts out with no friction to find out the potential movement of the piping. The friction forces corresponding to these movements are then included in the load vector, F, for a new analysis. The procedure continues iteratively until the convergence is reached when no significant change occurs between two consecutive analyses.

This method is straight forward. It requires no additional decomposition of the stiffness matrix at each analysis. Therefore, it appears to have the potential of saving some computer time. The method works fine in some rather rigid systems where the friction does not affect the direction of the movement, but works very poorly for most practical piping systems which always have considerable flexibility to absorb thermal expansion. This scheme does not have the capability to stop the pipe when the stopping force required is less than the potential friction force. Instead it keeps applying the same full friction force to the system resulting in a back and forth oscillatory but no ending iterations.

This method also quickly becomes divergent when applied to the system which has considerable movements in the flexible direction. In this flexible direction, if the friction force is applied against the movement, a very large displacement will be created opposite to the original movement. The situation reverses during the next iteration with the displacements randomly getting bigger and bigger in each subsequent iteration.

### (2). Fixed Stiffness Method

In this method, each frictional restraint is assigned two factitious orthogonal restraints laying on the plane perpendicular to the main supporting restraint as shown in Figure 3. The spring rate of these two restraints are taken to be the same as the initial slope given in Figure 2.

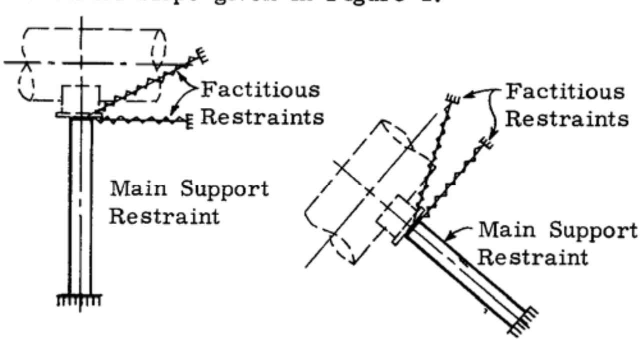



Figure 3, Factitious Restraints

When the pipe moves along the support plane, resistance forces will be generated on these two factitious restraints. These forces are the simulation of the friction effect. When the resultant force from these two restraints is less than the potential friction force, the pipe is being stopped by the friction. The forces generated are the actual friction forces acting in the corresponding directions. If the resultant force exceeds the potential friction force, then the pipe is moving. In this case, since the friction force generated is greater than the potential friction force, an adjustment needs to be made. The method applies a reverse external force back to the system to counter balance the excessive resistance force generated. The idea is to make the combined effort, from the resistance of the restraints and the reverse external force, equivalent to the friction expected. The iteration continues until the combined effect at each restraint matches the friction expected.

Each iteration in this fixed stiffness method changes only the load vector, F. No redecomposition of the stiffness matrix is required. Yet it has two factitious stiff springs which serve to stabilize the system and to readily stop the pipe from moving when it is called for. This method has the advantage of the first method in preserving the decomposed stiffness matrix, but has less tendency in getting into the domain of divergence. It is quite popular in the finite element analysis [4]. However, the method does have some undesirable behaviors. Again these undesirable behaviors are more pronounced in flexible systems. First, since the two factitious restraints are fairly stiff in most cases, it can take a large number of iterations to have the pipe moved to the final destination. The most disturbing part, however, is when the movement reverses at a certain point in the next iteration, the reverse external force derived from the previous iteration will tend to reinforce the reversal. This can often lead to an instable analysis.

(3). Variable Stiffness Method

The first two methods discussed are all based on the idea of preserving the most computer time intensive martix decomposition process. However, in piping stress analysis, the use of limit stops, single-acting restraints, and other non-linear features have become common place. To account for these nonlinear features, the revision and redecomposition of the stiffness matrix has become a necessity through each iteration. Based on this premise the saving of the decomposed stiffness matrix has become less important of a factor in pipe stress analysis.

Like the fixed stiffness method, the variable stiffness method also assigns two factitious restraints at each restraint location to simulate the friction. Only the stiffness or the spring rate of these factitious restraints are not fixed. Depending on the developers, some schemes start out with some stiffness, while others start out with zero resistance. The stiffness of these factitious restraints at each iteration is estimated from the previous iteration. The stiffness matrix is then revised for these updated spring rates and for the activity changes of other non-linear restraints. It is then redecomposed for the new iteration. A more detailed discussion on this method can be found in Reference [5].

The variable stiffness method normally converges to the required accuracy much quicker than the other two methods discussed. This makes the total computing effort required by this method not much different from those of the other methods, although the matrix decomposition is performed at every iteration. The convergence in this method can also get quite slow if there are multiple locations where the pipe is being stopped by the friction in the system. The solution, however, is always stable.

EXAMPLE

All the above three methods have been in use by different pipe stress computer programs. No matter which method is adopted, there are refinements that need to be made in the programming. These include the methods of increasing the convergence rate and the schemes for avoiding an insatble solution. These refinements are proprietary to the program developer and are not obvious to the users. Their effectiveness can only be measured by their actual performance. The example system shown in Figure 4 with the partial results tabulated in Table 2 can serve as a benchmark. This is a flexible off-site system with considerable movement in the flexible direction of the system. It is a typical system whose solution can easily become instable when using some of the less sophisticated iteration schemes.

Table 3 compares the analysis results of the case with friction against the case without friction.

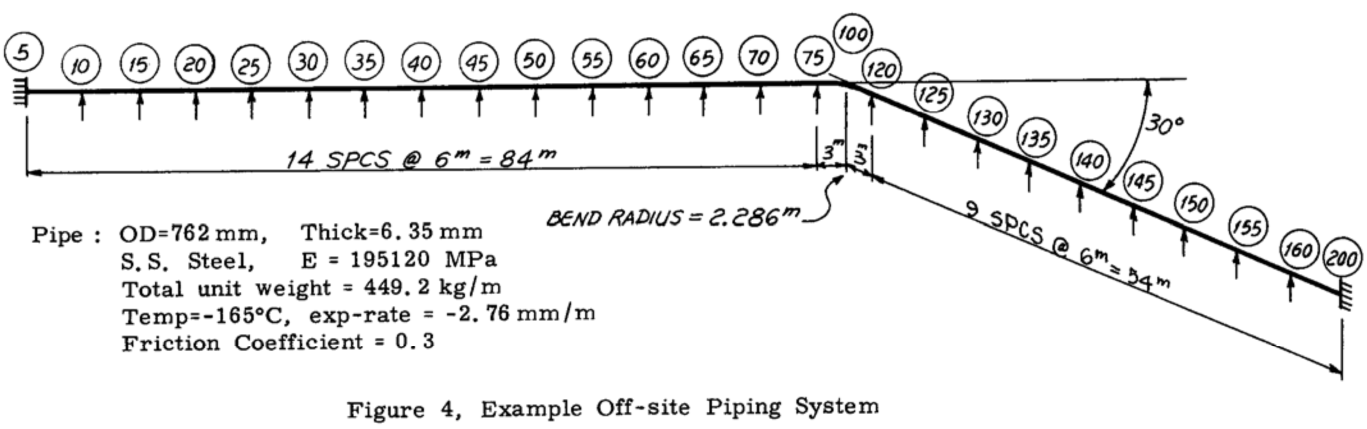



Figure 4, Example Off-site Piping System

Table 2, Partial Results of the Example Analysis

CASE 1 TH + WT RESULTS LOAD= THM,WGT,BWG,FOR,UFR,CSP,PRES

*** ANCHOR AND SUPPORT FORCES - INCLUDING FRICTION (ACTING ON SUPPORT) ***

| SUPT TYPE | DATA PT | SUPPORT FORCE AND MOMENT: FORCES ( N ) FX | FY | FZ | MOMENTS ( N-M ) MX | MY | MZ | FRICTION: FORCES ( N ) FFX | FFY | FFZ | T( N-M ) FMT | DEFLECTION (MM) DX | DY | DZ | NOTES |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ANCH | 5 | 222567 | -13214 | 198 | -304 | 9984 | -13213 | 0 | 0 | 0 | 0 | .0 | -.0 | .0 | |
| NLY | 10 | 0 | -26429 | 0 | 0 | 0 | 0 | -7914 | 0 | -436 | 0 | -16.1 | -.0 | -.9 | |
| NLY | 15 | 0 | -26429 | 0 | 0 | 0 | 0 | -7876 | 0 | -885 | 0 | -32.2 | -.0 | -3.6 | |
| NLY | 20 | 0 | -26429 | 0 | 0 | 0 | 0 | -7823 | 0 | -1275 | 0 | -48.3 | -.0 | -7.9 | |
| NLY | 25 | 0 | -26429 | 0 | 0 | 0 | 0 | -7785 | 0 | -1503 | 0 | -64.5 | -.0 | -12.5 | |
| NLY | 30 | 0 | -26429 | 0 | 0 | 0 | 0 | -7802 | 0 | -1434 | 0 | -80.6 | -.0 | -14.8 | |
| NLY | 35 | 0 | -26429 | 0 | 0 | 0 | 0 | -7882 | 0 | -895 | 0 | -96.8 | -.0 | -11.0 | |
| NLY | 40 | 0 | -26429 | 0 | 0 | 0 | 0 | -7916 | 0 | 310 | 0 | -113.0 | -.0 | 4.4 | |
| NLY | 45 | 0 | -26430 | 0 | 0 | 0 | 0 | -7578 | 0 | 2209 | 0 | -129.2 | -.0 | 37.7 | |
| NLY | 50 | 0 | -26425 | 0 | 0 | 0 | 0 | -6592 | 0 | 4285 | 0 | -145.5 | -.0 | 94.5 | |
| NLY | 55 | 0 | -26446 | 0 | 0 | 0 | 0 | -5279 | 0 | 5828 | 0 | -161.7 | -.0 | 178.5 | |
| NLY | 60 | 0 | -26359 | 0 | 0 | 0 | 0 | -4119 | 0 | 6686 | 0 | -177.9 | -.0 | 288.7 | |
| NLY | 65 | 0 | -26721 | 0 | 0 | 0 | 0 | -3359 | 0 | 7240 | 0 | -194.2 | -.0 | 418.5 | |
| NLY | 70 | 0 | -25215 | 0 | 0 | 0 | 0 | -2678 | 0 | 7057 | 0 | -210.5 | -.0 | 554.3 | |
| NLY | 75 | 0 | -27359 | 0 | 0 | 0 | 0 | -2610 | 0 | 7776 | 0 | -226.7 | -.0 | 675.2 | |
| NLY | 120 | 0 | -27337 | 0 | 0 | 0 | 0 | -2534 | 0 | 7795 | 0 | -217.8 | -.0 | 669.5 | |
| NLY | 125 | 0 | -25218 | 0 | 0 | 0 | 0 | -2178 | 0 | 7230 | 0 | -163.8 | -.0 | 543.6 | |
| NLY | 130 | 0 | -26720 | 0 | 0 | 0 | 0 | -1973 | 0 | 7734 | 0 | -103.9 | -.0 | 407.3 | |
| NLY | 135 | 0 | -26358 | 0 | 0 | 0 | 0 | -1358 | 0 | 7728 | 0 | -49.3 | -.0 | 280.1 | |
| NLY | 140 | 0 | -26445 | 0 | 0 | 0 | 0 | -317 | 0 | 7830 | 0 | -7.1 | -.0 | 174.5 | |
| NLY | 145 | 0 | -26424 | 0 | 0 | 0 | 0 | 1502 | 0 | 7656 | 0 | 19.0 | -.0 | 96.8 | |
| NLY | 150 | 0 | -26429 | 0 | 0 | 0 | 0 | 4114 | 0 | 6663 | 0 | 29.0 | -.0 | 47.0 | |
| NLY | 155 | 0 | -26428 | 0 | 0 | 0 | 0 | 6258 | 0 | 4825 | 0 | 25.9 | -.0 | 20.0 | |
| NLY | 160 | 0 | -26429 | 0 | 0 | 0 | 0 | 7048 | 0 | 3635 | 0 | 14.4 | -.0 | 7.4 | |
| ANCH | 200 | -145898 | -13214 | -96251 | -7008 | -28484 | 11211 | 0 | 0 | 0 | 0 | -.0 | -.0 | -.0 | |
| NET FORCES | | 76669 | -634174 | -96053 | | | | -76651 | 0 | 96059 | 0 | | | | |

Table 3, Comparison of the Example Analysis

| | Anchor Load at 5 | | Bend Moment at 100A |
|---|---|---|---|
| | Fx (N) | My (N-m) | My (N-m) |
| With Friction | 222600 | 10001 | 328480 |
| Without Friction | 16746 | 123942 | 124462 |


REFERENCES (cont.)

6. Building Code Requirements for Minimum Design Loads, ASNI A58.1, American Nat. Std, Inst., N.Y.C.

7. API Standard 617, Centrifugal Compressors for General Refinery Services, American Petroleum Inst. Washington, D.C.


It is obvious that without the friction the axial anchor load would have been very small. On the other hand, the friction tends to prevent the pipe from moving into the flexible lateral direction. This, in essence, serves as the guide preventing the anchor from getting the moment created by the lateral movement of the pipe. Because of the restriction on the lateral movement, the system becomes more stiff Thus resulting in higher bending moment at the bend.

## CODE COMPLIANCE ANALYSIS

In order to comply with the Piping Code requirements, the analysis has to separate the sustained stress from the self-limiting stress. The friction in a piping system is generally caused by the weight being pushed by thermal expansion. Weight is sustained loading but thermal expansion is self-limiting. The friction on the other hand is passive which by itself does not have the damaging potential. For the convenience of the analysis, the friction can be treated as self-limiting same as in the case of thermal expansion. To satisfy the Code requirement of separating the stresses, separate load cases for weight, thermal and occasional loads have to be performed. But without the weight the thermal expansion will hardly have any friction resistance. That is, if the straight weight or expansion is applied to the corresponding load cases, the friction force will completely disappear from the picture. Therefore, special arrangements have to be made so the friction effect can be accounted for properly.

One method to include the friction yet still be able to separate the sustained stress from the self-limiting stress is to apply the weight loading under the normal operating condition as the initial support load. With this method, the weight load under the normal operating condition is first determined at each support. This load is then used as the support initial load for the analysis of the thermal and occasional load cases. That is, if the normal operating weight load is 2000 N, and the thermal expansion load is 1000 N, then the support friction is included based on the support normal force of 3000 N. The single-acting restraint activity status will also be checked based on the premise that the support initial force is there.

The normal operating weight load is the balanced weight load under the operating condition. It is the weight load calculated by removing the inactive restraints at where the pipe is pushed off the support by the thermal expansion.

## OCCASIONAL LOAD ANALYSIS

The occasional load, by piping code criteria, is to be combined with the sustained load. In theory it can be directly added to the weight load for the analysis. However, because the sustained load has its own separate requirements and the occasional load is always considered as dual directional, the weight and occasional loads are normally analyzed with separate load cases in practice.

In the occasional load analysis, the initial weight load may also be included in the calculation of the friction effect. Although some may argue that the weight initial load is already included in thermal expansion analysis and should not be included again in the occasional load analysis. But the friction due to weight is still there to resist the occasional load motion whether the pipe has gone through the expansion process or not. Nevertheless, since the friction tends to help the system in resisting the occasional load, its inclusion to the occasional load analysis requires some justification. The practice varies due to the different beliefs in the availability of the initial weight load during the occasional load condition.

The earthquake and wind load analyses are normally done with the equivalent static method given by ANSI A58.1 [6]. In this method the piping is applied with the horizontal load appropriate for the location of the piping. The vertical load is not addressed in tacit recognition of the adequacy of the normal support structure in resisting the load. The vertical load may or may not be adequately supported by the normal support structure, but it does have a significant effect on the initial weight load. During the earthquake, because of the upward acceleration existing at a certain instant, the pipe may be lifted up fully or partially from the support leaving very little weight load on the support. Therefore, the inclusion of the initial load in estimating the friction will be nonconservative. The same, in a lesser degree, is also true during the hurricane condition. Based on the above consideration, some companies require that the initial weight load, if not the friction, cannot be included in earthquake and wind load analyses. Others have allowed the initial weight load in the wind load case, but not in the earthquake load case. In any case the magnitude of the occasional loads and their method of analysis shall be clearly defined in the Design Specification.

## CONCLUSION

The support friction can have a significant effect on the pipe load at the connecting equipment. It can also increase the thermal expansion stress by several folds in some cases. There are different methods which can be used to implement a computer program in handling the support friction. Some are more effective in certain cases, and some may not give a stable result under certain conditions. Owing to the inherent flexibility in a piping system, the simple direct substitution of the friction force scheme does not work well in the analysis of piping systems.

In order to comply with the ASME B&PV and ANSI B31 Piping Code requirements of separating the sustained stress from the self-limiting stress, separate load cases are performed for weight, thermal expansion, and occasional loads. The weight support load is included in the thermal expansion analysis for calculating the friction effect. However, the weight support load may or may not be included in the occasional load analysis depending on the individual design specification. While it is generally more conservative in including the friction in thermal expansion analysis, a separate expansion analysis without including the friction is also recommended to check the loading condition after the system has gone through a long period of operation with the friction effect shaken off. The inclusion of the initial weight in the occasional load analysis requires some serious consideration. The initial weight should not be included if the pipe is likely to be lifted off the support either fully or partially during the occurance of the event.

## RFERENCES

1. Peng, L. C., "Stress Analysis Methos for Underground Pipe Lines," Pipe Line Industry, Apr & May, 1978
2. ANSYS User's Manual, STIF52, Swanson Analysis Systems, Inc., Houston, Pennsylvania.
3. SIMFLEX-II, Peng Engineering, Houston, Texas.
4. Shah, V. N. and Gilmore, C. B., "Dynamic Analysis of a Structure with Coulomb Friction," ASME paper 82-PVP-18, July 1982.
5. Sobieszczanski, J., "Inclusion of Support Friction Into a Computerized Solution of a Self-Compensating PipeLine," ASME paper 71-WA/PVP-1, Nov, 1971.