UNIVERSITY OF MUMBAI

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PROJECT REPORT

ON

PIPING STRESS ANALYSIS

BY

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CERTIFICATE

This is to certify that the dissertation entitled '**PIPING STRESS ANALYSIS**' is being submitted to the University of Mumbai by the following students:

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INTRODUCTION

Pipes are the most delicate components in any process plant. They are also the busiest entities. They are subjected to almost all kinds of loads, intentional or unintentional. It is very important to take note of all potential loads that a piping system would encounter during operation as well as during other stages in the life cycle of a process plant. Ignoring any such load while designing, erecting, hydro-testing, start-up shut-down, normal operation, maintenance etc. can lead to inadequate design and engineering of a piping system. The system may fail on the first occurrence of this overlooked load. Failure of a piping system may trigger a Domino effect and cause a major disaster.

Stress analysis and safe design normally require appreciation of several related concepts. An approximate list of the steps that would be involved is as follows.

- 1. Identify potential loads that would come on to the pipe or piping system during its entire life.
- 2. Relate each one of these loads to the stresses and strains that would be developed in the crystals/grains of the Material of Construction (MoC) of the piping system.
- 3. Decide the worst three dimensional stress state that the MoC can withstand without failure
- 4. Get the cumulative effect of all the potential, loads on the 3-D stress scenario in the piping system under consideration.
- 5. Alter piping system design to ensure that the stress pattern is within failure limits.

The goal of quantification and analysis of pipe stresses is to provide safe design through the above steps. There could be several designs that could be safe. A piping engineer would have a lot of scope to choose from such alternatives, the one which is most economical, or most suitable etc. Good piping system design is always a mixture of sound knowledge base in the basics and a lot of ingenuity.

OBJECTIVE AND SCOPE

With piping, as with other structures, the analysis of stresses may be carried to varying degrees of refinement. Manual systems allow for the analysis of simple systems, whereas there are methods like chart solutions (for three-dimensional routings) and rules of thumb (for number and placement of supports) etc. involving long and tedious computations and high expense. But these methods have a scope and value that cannot be defined as their accuracy and reliability depends upon the experience and skill of the user. All such methods may be classified as follows:

- Approximate methods dealing only with special piping configurations of two-, threeor four-member systems having two terminals with complete fixity and the piping layout usually restricted to square corners. Solutions are usually obtained from charts or tables. The approximate methods falling into this category are limited in scope of direct application, but they are sometimes usable as a rough guide on more complex problems by assuming subdivisions of the model into anchored sections fitting the contours of the previously solved cases.
- 2. Methods restricted to square-corner, single-plane systems with two fixed ends, but without limit as to the number of members.
- 3. Methods adaptable to space configurations with square corners and two fixed ends.
- 4. Extensions of the previous methods to provide for the special properties of curved pipe by indirect means, usually a virtual length correction factor.

The objective of this project is to check the adequacy of rules of thumb as well as simple solution approaches by comparison with comprehensive computer solutions of similar systems, based on FEM. It may also be possible to extend the existing chart solutions and rules of thumb to more complex systems with these comparative studies.

METHODOLOGY CLASSIFICATION OF LOADS AND FAILURE MODES

Pressure design of piping or equipment uses one criterion for design. Under a steady application of load (e.g., pressure), it ensures against failure of the system as perceived by one of the failure theories. If a pipe designed for a certain pressure experiences a much higher pressure, the pipe would rupture even if such load (pressure) is applied only once. The failure or rupture is sudden and complete. Such a failure is called catastrophic failure. It takes place only when the load exceeds far beyond the load for which design was carried out. Over the years, it has been realised that systems, especially piping, systems can fail even when the loads are always under the limits considered safe, but the load application is cyclic (e.g. high pressure, low pressure, high pressure, ...). Such a failure is not guarded against by conventional pressure design formula or compliance with failure theories. For piping system design, it is well established that these two types of loads must be treated separately and together guard against catastrophic and fatigue failure.

The loads the piping system (or for that matter any structural part) faces are broadly classified as primary loads and secondary loads.

Primary Loads

These are typically steady or sustained types of loads such as internal fluid pressure, external pressure, gravitational forces acting on the pipe such as weight of pipe and fluid, forces due to relief or blow down pressure waves generated due to water hammer effects. The last two loads are not necessarily sustained loads. All these loads occur because of forces created and acting on the pipe. In fact, primary loads have their origin in some force acting on the pipe causing tension, compression, torsion etc leading to normal and shear stresses. A large load of this type often leads to plastic deformation. The deformation is limited only if the material shows strain hardening characteristics. If it has no strain hardening property or if the load is so excessive that the plastic instability sets in, the system would continue to deform till rupture. Primary loads are not self-limiting. It means that the stresses continue to exist as long as the load persists and deformation does

not stop because the system has deformed into a no-stress condition but because strain hardening has come into play.

Secondary Loads

Just as the primary loads have their origin in some force, secondary loads are caused by displacement of some kind. For example, the pipe connected to a storage tank may be under load if the tank nozzle to which it is connected moves down due to tank settlement. Similarly, pipe connected to a vessel is pulled upwards because the vessel nozzle moves up due to vessel expansion. Also, a pipe may vibrate due to vibrations in the rotating equipment it is attached to. A pipe may experience expansion or contraction once it is subjected to temperatures higher or lower respectively as compared to temperature at which it was assembled.

The secondary loads are often cyclic but not always. For example load due to tank settlement is not cyclic. The load due to vessel nozzle movement during operation is cyclic because the displacement is withdrawn during shut-down and resurfaces again after fresh start-up. A pipe subjected to a cycle of hot and cold fluid similarly undergoes cyclic loads and deformation. Failure under such loads is often due to fatigue and not catastrophic in nature.

Broadly speaking, catastrophic failure is because individual crystals or grains were subjected to stresses which the chemistry and the physics of the solid could not withstand. Fatigue failure is often because the grains collectively failed because their collective characteristics (for example entanglement with each other etc.) changed due to cyclic load. Incremental damage done by each cycle to their collective texture accumulated to such levels that the system failed. In other words, catastrophic failure is more at microscopic level, whereas fatigue failure is at mesoscopic level if not at macroscopic level.

The Stresses

The MoC of any piping system is the most tortured non-living being right from its birth. Leaving the furnace in the molten state, the metal solidifies within seconds. It is a very hurried crystallization process. The crystals could be of various lattice structural patterns such as BCC, FCC, HCP etc. depending on the material and the process. The grains, crystals of the material have no time or chance to orient themselves in any particular fashion. They are thus frozen in all random orientations in the cold harmless pipe or structural member that we see.

When we calculate stresses, we choose a set of orthogonal directions and define the stresses in this co-ordinate system. For example, in a pipe subjected to internal pressure or any other load, the most used choice of co-ordinate system is the one comprising of axial or longitudinal direction (L), circumferential (or Hoope's) direction (H) and radial direction (R) as shown in figure. Stresses in the pipe wall are expressed as axial (S_L), Hoope's (S_H) and radial (S_R). These stresses which stretch or compress a grain/crystal are called normal stresses because they are normal to the surface of the crystal.

But, all grains are not oriented as the grain in the figure. In fact the grains would have been oriented in the pipe wall in all possible orientations. The above stresses would also have stress components in direction normal to the faces of such randomly oriented crystal. Each crystal thus does face normal stresses. One of these orientations must be such that it maximizes one of the normal stresses.



The mechanics of solids state that it would also be orientation which minimizes some other normal stress. Normal stresses for such orientation (maximum normal stress orientation) are called principal stresses, and are designated S_1 (maximum), S_2 and S_3

(minimum). Solid mechanics also states that the sum of the three normal stresses for all orientation is always the same for any given external load. That is

$$S_{L} + S_{H} + S_{R} = S_{1} + S_{2} + S_{3}$$

In addition to the normal stresses, a grain can be subjected to shear stresses as well. These act parallel to the crystal surfaces as against perpendicular direction applicable for normal stresses. Shear stresses occur if the pipe is subjected to torsion, bending etc. Just as there is an orientation for which normal stresses are maximum, there is an orientation which maximizes shear stress. The maximum shear stress in a 3-D state of stress can be shown to be

$$\tau_{\rm max} = (S_1 - S_3) / 2$$

i.e. half of the difference between the maximum and minimum principal stresses. The maximum shear stress is important to calculate because failure may occur or may be deemed to occur due to shear stress also. A failure perception may stipulate that maximum shear stress should not cross certain threshold value. It is therefore necessary to take the worst-case scenario for shear stresses also as above and ensure against failure.

It is easy to define stresses in the co-ordinate system such as axial-Hoope's-radial (L-H-R) that are defined for a pipe. The load bearing cross-section is then well defined and stress components are calculated as ratio of load to load bearing cross-section. Similarly, it is possible to calculate shear stress in a particular plane given the torsional or bending load. What are required for testing failure - safe nature of design are, however, principal stresses and maximum shear stress. These can be calculated from the normal stresses and shear stresses available in any convenient orthogonal co-ordinate system. In most pipe design cases of interest, the radial component of normal stresses (S_R) is negligible as compared to the other two components (S_H and S_L). The 3-D state of stress thus can be simplified to 2-D state of stress. Use of Mohr's circle then allows to calculate the two principle stresses and maximum shear stress as follows.

$$\begin{split} S_1 &= (S_L + S_H)/2 + [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5} \\ S_1 &= (S_L + S_H)/2 - [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5} \\ &= 0.5 \; [(S_L - S_H)^2 + 4 \; \tau^2]^{0.5} \end{split}$$

The third principle stress (minimum i.e. S_3) is zero.

All failure theories state that these principle or maximum shear stresses or some combination of them should be within allowable limits for the MoC under consideration. To check for compliance of the design would then involve relating the applied load to get the net S_H , S_L , τ and then calculate S_1 , S_2 and τ_{max} and some combination of them.

Normal And Shear Stresses From Applied Load

As said earlier, a pipe is subjected to all kinds of loads. These need to be identified. Each such load would induce in the pipe wall, normal and shear stresses. These need to be calculated from standard relations. The net normal and shear stresses resulting in actual and potential loads are then arrived at and principle and maximum shear stresses calculated. Some potential loads faced by a pipe and their relationships to stresses are summarized here in brief

Axial Load

A pipe may face an axial force (F_L) as shown in Figure. It could be tensile or compressive.



What is shown is a tensile load. It would lead to normal stress in the axial direction (S_L) . The load bearing cross-section is the cross-sectional area of the pipe wall normal to the load direction, A_m . The stress can then be calculated as

$$S_L = F_L / A_m$$

The load bearing cross-section may be calculated rigorously or approximately as follows.

$$Am = \pi (d_o^2 - d_i^2) / 4 \quad (rigorous)$$
$$= \pi (d_o + d_i) t / 2 \quad (based on average diameter)$$
$$= \pi d_o t \qquad (based on outer diameter)$$

The axial load may be caused due to several reasons. The simplest case is a tall column. The metal cross-section at the base of the column is under the weight of the column section above it including the weight of other column accessories such as insulation, trays, ladders etc. Another example is that of cold spring. Many times a pipeline is intentionally cut a little short than the end-to-end length required. It is then connected to the end nozzles by forcibly stretching it. The pipe, as assembled, is under axial tension. When the hot fluid starts moving through the pipe, the pipe expands and compressive stresses are generated. The cold tensile stresses are thus nullified. The thermal expansion stresses are thus taken care of through appropriate assembly-time measures.

Internal / External Pressure

A pipe used for transporting fluid would be under internal pressure load. A pipe such as a jacketed pipe core or tubes in a Shell & Tube exchanger etc. may be under net external pressure. Internal or external pressure induces stresses in the axial as well as circumferential (Hoope's) directions. The pressure also induces stresses in the radial direction, but as argued earlier, these are often neglected.

The internal pressure exerts an axial force equal to pressure times the internal cross-section of pipe.

$$F_{\rm L} = P [\pi d_i^2 / 4]$$

This then induces axial stress calculated as earlier. If outer pipe diameter is used for calculating approximate metal crossection as well as pipe cross- section, the axial stress can often be approximated as follows.

$$S_{L} = P d_{o} / (4t)$$

The internal pressure also induces stresses in the circumferential direction as shown in figure



The stresses are maximum for grains situated at the inner radius and minimum for those situated at the outer radius. The Hoope's stress at any in between radial position (r) is given as follows (Lame's equation)

$$S_{\rm H}$$
 at $r = P (r_i^2 + r_i^2 r_o^2 / r^2) / (r_o^2 - r_i^2)$

For thin walled pipes, the radial stress variation can be neglected. From membrane theory, S_H may then be approximated as follows.

$$S_{\rm H} = P d_{\rm o} / 2t$$
 or $P d_{\rm i} / 2t$

Radial stresses are also induced due to internal pressure as can be seen in figure



At the outer skin, the radial stress is compressive and equal to atmospheric pressure (P_{atm}) or external pressure (P_{ext}) on the pipe. At inner radius, it is also compressive but equal to absolute fluid pressure (P_{abs}). In between, it varies. As mentioned earlier, the radial component is often neglected.

Bending Load

A pipe can face sustained loads causing bending. The bending moment can be related to normal and shear stresses. Pipe bending is caused mainly due to two reasons: Uniform weight load and concentrated weight load. A pipe span supported at two ends would sag between these supports due to its own weight and the weight of insulation (if any) when not in operation. It may sag due to its weight and weight of hydrostatic test fluid it contains during hydrostatic test. It may sag due to its own weight, insulation weight and the weight of fluid it is carrying during operation

All these weights are distributed uniformly across the unsupported span, and lead to maximum bending moment either at the centre of the span or at the end points of the span (support location) depending upon the type of the support used.

Let the total weight of the pipe, insulation and fluid be W and the length of the unsupported span be L (see Figure).



The weight per unit length, w, is then calculated (w = W/L). The maximum bending moment, M_{max} , which occurs at the centre for the pinned support is then given by the beam theory as follows.

$$M_{max} = wL^2/8$$
 for pinned support

For Fixed Supports, the maximum bending moment occurs at the ends and is given by beam theory as follows

$$M_{max} = w L^2 / 12$$
 for fixed support.

The pipe configuration and support types used in process industry do not confirm to any of these ideal support types and can be best considered as somewhere in between. As a result, a common practice is to use the following average formula to calculate bending moment for practical pipe configurations, as follows.

$$M_{max} = w L^2 / 10.$$

Also, the maximum bending moment in the case of actual supports would occur somewhere between the ends and the middle of the span.

Another load that the pipe span would face is the concentrated load. A good example is a valve on a pipe run (see figure).



The load is then approximated as acting at the centre of gravity of the valve and the maximum bending moment occurs at the point of loading for pinned supports and is given as

$$M_{max} = W a b / L$$

For rigid supports, the maximum bending moment occurs at the end nearer to the pointed load and is given as

$$\mathbf{M}_{\max} = \mathbf{W} \mathbf{a}^2 \mathbf{b} / \mathbf{L}^2$$

a is to be taken as the longer of the two arms (a and b) in using the above formula.

As can be seen, the bending moment can be reduced to zero by making either a or b zero, i.e. by locating one of the supports right at the point where the load is acting. In actual practice, it would mean supporting the valve itself. As that is difficult, it is a common practice to locate one support as close to the valve (or any other pointed and significant load) as possible. With that done, the bending moment due to pointed load is minimal and can be neglected.

Whenever the pipe bends, the skin of the pipe wall experiences both tensile and compressive stresses in the axial direction as shown in Figure .



Max Compressive Stress

The axial stress changes from maximum tensile on one side of the pipe to maximum compressive on the other side. Obviously, there is a neutral axis along which the bending moment does not induce any axial stresses. This is also the axis of the pipe.

The axial tensile stress for a bending moment of M, at any location c as measured from the neutral axis is given as follows.

$$S_L = M_b c / I$$

I is the moment of inertia of the pipe cross-section. For a circular cross-section pipe, I is given as

$$I = \pi (d_0^4 - d_i^4) / 64$$

The maximum. tensile stress occurs where c is equal to the outer radius of the pipe and is given as follows.

 S_L at outer radius = $M_b r_o / I = M_b / Z$

where $Z (= I/r_o)$ is the section modulus of the pipe.

Shear Load

Shear load causes shear stresses. Shear load may be of different types. One common load is the shear force (V) acting on the cross-section of the pipe as shown in figure .



It causes shear stresses which are maximum along the pipe axis and minimum along the outer skin of the pipe. This being exactly opposite of the axial stress pattern caused by bending moment and also because these stresses are small in magnitude, these are often not taken in account in pipe stress analysis. If necessary, these are calculated as

$$\tau_{max} = V Q / A_m$$

where Q is the shear form factor and A_m is the metal cross-section.

Torsional Load

This load (see figure) also causes shear stresses.



The shear stress caused due to torsion is maximum at outer pipe radius. And is given there in terms of the torsional moment and pipe dimensions as follows.

$$\tau$$
 (at r = r_o) = M_T r_o / R_T = M_T r_o / (2I) = M_T / 2Z

 R_T is the torsional resistance (= twice the moment of inertia).

All known loads on the pipe should be used to calculate contributions to S_L , S_H and t. These then are used to calculate the principal stresses and maximum shear stress. These derived quantities are then used to check whether the pipe system design is adequate based on one or more theories of failure.

Theories Of Failure

A piping system in particular or a structural part in general is deemed to fail when a stipulated function of various stresses and strains in the system or structural part crosses a certain threshold value. It is a normal practice to define failure as occurring when this function in the actual system crosses the value of a similar function in a solid rod specimen at the point of yield. There are various theories of failure that have been put forth. These theories differ only in the way the above mentioned function is defined. Important theories in common use are considered here.

Maximum Stress Theory

This is also called Rankine Theory. According to this theory, failure occurs when the maximum principle stress in a system (S_1) is greater than the maximum tensile principle stress at yield in a specimen subjected to uni-axial tension test.

Uniaxial tension test is the most common test carried out for any MoC. The tensile stress in a constant cross-section specimen at yield is what is reported as yield stress (Sy) for any material and is normally available. In uni-axial test, the applied load gives rise only to axial stress (S_L) and S_H and S_R as well as shear stresses are absent. S_L is thus also the principle normal stress (i.e. S_1). That is, in a specimen under uni-axial tension test, at yield, the following holds.

$$S_L = S_Y, S_H = 0, S_R = 0$$

 $S_1 = S_Y, S_2 = 0$ and $S_3 = 0$.

The maximum tensile principle stress at yield is thus equal to the conventionally reported yield stress (load at yield/ cross-sectional area of specimen).

The Rankine theory thus just says that failure occurs when the maximum principle stress in a system (S_1) is more than the yield stress of the material (Sy).

The maximum principle stress in the system should be calculated as earlier. It is interesting to check the implication of this theory on the case when a cylinder (or pipe) is subjected to internal pressure.

As per the membrane theory for pressure design of cylinder, as long as the Hoope's stress is less than the yield stress of the MoC, the design is safe. It is also known that Hoope's stress (S_H) induced by external pressure is twice the axial stress (S_L). The stresses in the cylinder as per the earlier given formula would be

$$\begin{split} S_1 &= (S_L + S_H)/2 + [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5} \\ &= S_L \\ S_1 &= (S_L + S_H)/2 - [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5} \\ &= S_H \end{split}$$

The maximum principle stress in this case is S_2 (= S_H). The Rankine theory and the design criterion used in the membrane theory are thus compatible.

This theory is widely used for pressure thickness calculation for pressure vessels and piping design uses Rankine theory as a criterion for failure.

Maximum Shear Theory

This is also called Tresca theory. According to this theory, failure occurs when the maximum shear stress in a system τ_{max} is greater than the maximum shear stress at yield in a specimen subjected to uni-axial tension test. Note that it is similar in wording, to the statement of the earlier theory except that maximum shear stress is used as criterion for comparison as against maximum principle stress used in the Rankine theory.

In uniaxial test, the maximum shear stress at yield condition of maximum shear test given earlier is

$$\begin{split} \tau_{max} &= 0.5 \; [(S_L - S_H \;)^2 + 4 \; \tau^2]^{0.5} \\ &= S_L/2 = Sy/2 \end{split}$$

The Tresca theory thus just says that failure occurs when the maximum shear stress in a system is more than half the yield stress of the material (Sy). The maximum shear stress in the system should be calculated as earlier.

It should also be interesting to check the implication of this theory on the case when a cylinder (or pipe) is subjected to internal pressure.

As the Hoope's stress induced by internal pressure (S_H) is twice the axial stress (S_L) and the shear stress is not induced directly ($\tau = 0$) the maximum shear stress in the cylinder as per the earlier given formula would be

$$\begin{split} \tau_{max} &= 0.5 \; [(S_L - S_H \;)^2 + 4 \; \tau^2]^{0.5} \\ &= 0.25 \; S_H \end{split}$$

This should be less than 0.5Sy, as per Tresca theory for safe design. This leads to a different criterion that Hoope's stress in a cylinder should be less than twice the yield

stress. The Tresca theory and the design criterion used in the membrane theory for cylinder are thus incompatible.

Octahedral Shear Theory

This is also called Von Mises theory. According to this theory, failure occurs when the octahedral shear stress in a system is greater than the octahedral shear stress at yield in a specimen subjected to uniaxial tension test. It is similar in wording to the statement of the earlier two theories except that octahedral shear stress is used as criterion for comparison as against maximum principle stress used in the Rankine theory or maximum shear stress used in Tresca theory.

The octahedral shear stress is defined in terms of the three principle stresses as follows.

$$\tau_{\text{oct}} = 1/3 \left[(\mathbf{S}_1 - \mathbf{S}_2)^2 + (\mathbf{S}_2 - \mathbf{S}_3)^2 + (\mathbf{S}_3 - \mathbf{S}_1)^2 \right]^{0.5}$$

In view of the principle stresses defined for a specimen under uni-axial load earlier, the octahedral shear stress at yield in the specimen can be shown to be as follows.

$$\tau_{\rm oct} = \sqrt{2} \, \mathrm{Sy} / 3$$

The Von Mises theory thus states that failure occurs in a system when octahedral shear stress in the system exceeds $\sqrt{2}$ Sy / 3.

For stress analysis related calculations, most of the present day piping codes use a modified version of Tresca theory.

Design Under Secondary Load

As pointed earlier, a pipe designed to withstand primary loads and to avoid catastrophic failure may fall after a sufficient amount of time due to secondary cyclic load causing, fatigue failure. The secondary loads are often cyclic in nature. The number of cycles to failure is a property of the material of construction just as yield stress is. While yield stress is cardinal to the design under primary sustained loads, this number of cycles to failure is the corresponding material property important in design under cyclic loads aim at ensuring that the failure does not take place within a certain period for which the system is to be designed.

While yield stress is measured by subjecting a specimen to uni-axial tensile load, fatigue test is carried out on a similar specimen subjected to cycles of uniaxial tensile and compressive loads of certain amplitude, i.e. magnitude of the tensile and compressive loads. Normally the tests are carried out with zero mean load. This means, that the specimen is subjected to a gradually increasing load leading to a maximum tensile load of W, then the load is removed gradually till it passes through zero and becomes gradually a compressive load of W (i.e. a load of W), then a tensile load of W and so on. Time averaged load is thus zero. The cycles to failure are then measured, The experiments are repeated with different amplitudes of load.

Conclusion

Stresses in pipe or piping systems are generated due to loads experienced by the system. These loads can have origin in process requirement, the way pipes are supported, piping system's static properties such as own weight or simple transmitted loads due to problems in connecting equipments such as settlement or vibrations. Whatever may be the origin of load, these stresses the fabric of the MoC and failure may occur.

Fatigue failure is an important aspect in flexibility analysis of piping systems. Often cyclic stresses in piping systems subjected to thermal cycles get transferred to flexibility providing components such as elbows. These become the components susceptible to fatigue failure. Thermal stress analysis or flexibility analysis attempts to guard against such failure through very involved calculations.

FLEXIBILITY ANALYSIS

Flexibility analysis is done on the piping system to study its behaviour when its temperature changes from ambient to operating, so as to arrive at the most economical layout with adequate safety.

The following are the considerations that decide the minimum acceptable flexibility on a piping configuration.

- 1. The maximum allowable stress range in the system.
- 2. The limiting values of forces and moments that the piping system is permitted to impose on the equipment to which it is connected.
- 3. The displacements within the piping system.
- 4. The maximum allowable load on the supporting structure.

Methods Of Flexibility Analysis

There are two methods of flexibility analysis which involve manual calculations.

1. Check as per clause 119.7.1/319.4.1 of the piping code

This clause specifies that no formal analysis is required in systems which are of uniform size, have no more than two points of fixation, no intermediate restraints and fall within the empirical equation.

$$K \ge \frac{DY}{(L-U)^2}$$
 where,

D = the outside diameter of the pipe

Y = resultant of total displacement strains to be absorbed by the piping system.

L = developed length between anchors.

U = anchor distance, straight line between anchors.

K = 0.03 for FPS units.

= 208.3 for SI units.

2. Guided Cantilever Method

Guide cantilever is based on the simple concept of "minimum length".



When two vessels are connected by a straight pipe, the pipe may buckle or dent the sides of the vessel when operating at high temperature due to expansion. To overcome this difficulty a bend is provided as shown in figure above. So that the movement ' δ ' due to expansion will be absorbed and stresses are restricted to a given value. The minimum length for this configuration to absorb movement can be calculated as-

$$L = \sqrt{\frac{DE\delta}{48f}}$$

where,

L = minimum leg length.

- f = maximum bending stress.
- $\delta = movement.$
- E = Young's modulus.
- D = outer diameter of the pipe.

WORKING

CAD Packages like CAEPIPE and CAESAR II have been developed for the comprehensive analysis of complex systems. These software make use of Finite Element Methods to carry out stress analysis. However they require the pipe system to be modelled before carrying out stress analysis. Due to time constraints it is not possible to model the pipe systems always. Hence it becomes necessary to carryout elementary analysis before going in for the software analysis. Chart solutions, Rules of Thumb and Mathematical formulae are at our disposal. Our project is mainly concerned with the analysis of two anchor problems using the formula and modify it if needed. If possible we may also extend it to three anchor problems.

Clause 119.7.1/319.4.1 of the piping code suggests that for a pipe to be safe the value of critical coefficient K where $K = \frac{DY}{(L-U)^2}$ should be less than 208.3 (in SI units). If according to this formula a pipe is safe, then no further analysis is required.

The software, which we would be using to compare our studies, is CAEPIPE. In CAEPIPE the ratio of Maximum Stress Induced to Maximum Allowable Stress is calculated. If this ratio is below 1 then the pipe system is safe else redesigning is required.

CAEPIPE

Caepipe allows a comprehensive analysis of piping systems. The piping system has to be modelled on the computer as shown below. Various values are needed like loads on the pipe, material of the pipe, operating temperature, diameter of the pipe, types of bends etc.

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#	Node	Туре	DM (mm)	D [∨] (mm)	DZ (mm)	Mad	Sect	Load	Data
1	lille =								
2	lic 👘	From							Anchor
3	20	Renc			5000	MI	51	L L	
۷	3C	Benc		8000		M1	S1	L1	
5	4C	Benu			80CO	M1	S1	L1	
G	5C	Denc	11000			M1	S1	L1	
7	GC			6000		М1	51	L1	Anchor
8									
9									
10									
11									
12									
1.1									

When this is done caepipe produces a 3-D orientation of the pipe in space as shown below



After this the analyze command in the file menu is given and the analysis for safety is carried out. Various results are displayed in a tabular format as follows

-1-	Caepip	pe : As	ME CI	ass 2 [1992)	Code (Campli	ance	(Sarte	d Stres	*C*)	(finalpro	11.ica	(D:\!	502
File	<u>H</u> eat	ulte 🗹	e~ <u>L</u> p	tions (Wincow	• <u>⊢</u> ∈lp									
e	3	#			Ô			Ξ			S	%			
1		Suda	ined [5]		1. s	= «pansion (1U)				Expansion [11]					
#	Node	SL (MPa)	1.5011 (M구ai	<u>)_</u> 1.50	Node	SE (MPa)	GA (MPai	SE SA	Node	SL+SE (MPai	CHI-SA (MEa)	<u>3_+9E</u> 3 I+SA			
•	50E	283.7	1551	1.83	-0	0.000	1551	0.00	50B	122.9	258.6	0.48			
2	50/	262.5	1551	1.69	204	0.000	1551	0.00	534	115.4	258.6	0.45			
3	4UE	176.5	1551	1.14	208	U.UU.J	1551	U.UJ	IJ	11,1,1	258.6	J.43			
4	304	173.5	155 1	1.12	304	0.000	1551	0.00	40B	84 93	258.6	D.33			
5	404	170 9	1551	1.10	30B	0.000	1.56.1	ົກກາ	374	83.85	258.6	1.32			
6	30E	154.7	1551	1.00	404	0.000	1551	0.00	434	82.94	258.6	0.32			
?	204	116.5	1551	0.75	40B	0.000	1551	0.00	33B	77 2C	258.6	0.30			
0	-0	110.1	1551	0.71	504	0.000	1551	0.00	23A	CO 6C	250.0	0.25			
9	20E	92.32	1551	0.E0	50B	0.000	1551	0.00	23B	55.22	258.6	0.21			
.0	60	53.59	1551	0.35	60	0.000	155 1	0.00	60	53 59	258.6	3.21			
		_				-									
5-14											-				
Ť.															

Caepipe shows the precise point at which the pipe will fail. As in the above example we see that the pip will fail at node 50A, 50B, 40A, 40B, 30A, S/A ratio being the maximum at node 50B. The graphic of the analysis is shown in the following figure.



Let us consider the following ex	camples where we v	will find the value of	f critical coefficient
'K' and compare it with the $\frac{S}{A}$	ratio in CAEPIPE.		

Sr.No	X Y Z		Diameter	K	S/A	
	Length(m)	Length(m)	Length(m)	(mm)		
1.	4	7	12	457.2	207.02	0.56
2.	5	20	5	323.8	203.4	0.48
3.	4	15	3	219.07	206.74	0.38
4.	20	13	31	219.07	31.63	7.91
5.	23	11	65	114.3	21.42	34.68
6.	3	3	2	219.07	214.248	0.19
7.	3	11	2	168.27	227.49	0.27
8.	10	5	8	406.4	149.19	0.52

Thus it is clear that although the formula gives accurate results many a times, there are certain discrepancies. Therefore we have to modify the formula accordingly to cover a wide spectrum of problems

The Value of 'K' depends on the following factors :



1. Diameter of the Pipe :- $K \propto D$ so as K increases, D also increases but is actual case, for a fixed configuration as D increases pipe becomes more and more safe. Consider the following example





Sr	Х	Y	Z	Diameter	K	S/A
No	Length(m)	Length(m)	Length(m)	(mm)		
1.	4	15	6	114.3	62.27	1.25
2.	4	15	6	168.27	91.675	0.8
3.	4	15	6	219.07	119.352	0.63
4.	4	15	6	273.05	148.76	0.47
5.	4	15	6	323.85	176.43	0.42
6.	4	15	6	355.6	193.43	0.41

Thus we need to modify the formula in order to account for the above factor. After a lot of analysis and tedious calculation we were able to establish a homomorphic index of D which itself is a function of D. So the new formula is

$$K = \frac{Y}{(L - U)^2} D^{\frac{1 + 0.5D}{1 - D}}$$

2. Lengths of Pipe :- It is the total length in the X, Y and the Z direction. If any one of the length in the X,Y or the Z direction is very large as compared to the other two, it greatly affects the value of K. However it is already considered in the original formula where:

Total length in X,Y and Z directions L = |X| + |Y| + |Z| and Distance between the supports $U = \sqrt{X^2 + Y^2 + Z^2}$

Total length of the pipe 'L' has a considerable effect on the value of criticality constant 'k'. We multiply the formula with $2 + 0.05L^{2.5}$ to reduce the weight of the length.

We also consider the ratio of the longest length to the shortest length in the pipe system. If any one of the length is too long or too short then the pipe system may become unsafe. Here we multiply the formula with the ratio $\frac{\text{Longest Length}}{\text{Shortest Length}}$.

Also length of the leg attached to the anchor (anchor leg) and the leg attached to it plays an important role. The leg attached to the anchor leg induces a bending moment in the anchor leg due to its weight. This bending moment may make the pipe unsafe hence a proper combination of this two must be chosen. We again multiply the formula

with the factor

Length of the first Anchor leg – Length of the leg attached to anchor leg Length of the second Anchor leg – Length of the leg attached to anchor leg where first anchor is the starting anchor and second anchor is the concluding anchor.

3. Number of Bends :- As the number of bends increases, the pipe becomes more and more safe provided the initial and the final point is same. The original formula does not account for the number of bends.

Let us consider some examples :-



Sr	Х	Y	Z	Diameter	n	S/A
No	Length(m)	Length(m)	Length(m)	(mm)		
1.	4	15	6	406.4	2	0.81
2.	4	15	6	406.4	3	0.66
3.	4	15	6	406.4	4	0.61
4.	4 15		6	406.4	5	0.52

We established a factor $\left(\frac{2}{n}\right)^{0.3}$ by which the original formula is to be multiplied if the number of bends is less than 4 else we multiply by $\left(\frac{2}{n}\right)^{0.1}$

- **4.** Pipe Configuration :- Pipe configuration greatly affects the S/A ratio. More complex the configuration, more difficult is its analysis. Such systems are usually modelled on the computer for a comprehensive analysis so we don't consider such cases here.
- **5.** Internal Pressure :- As the internal pressure increases pipe becomes more and more unsafe. However this does not affect the critical constant much hence it is not considered in our analysis.

- 6. Dead Loads :- As the dead loads increase the pipe becomes more and more unsafe. Dead loads may include hangers, diaphragms, bellows etc. When the piping system is designed and the initial and final points are decided a preliminary analysis is carried out. This analysis doesn't consider the effects of dead loads however software like Caepipe takes into account these dead loads.
- 7. Material of the Pipe :- Material of the pipe affects the pipe system. Depending on various materials, the linear expansion will vary. This will change the value of Y and hence critical coefficient K. In our calculations we have assumed the material as ASTM A106 Grade B, which is widely used in the industry.
- **8. Pipe Temperature** :- The pipe may operate at various temperatures. If the temperature of the pipe is very high or very low then the pipe may fail. The values of the temperatures can be selected and the corresponding elongations can be taken from a standard databook.

Hence the final formula after considering the effect of diameter, lengths of pipe and no of bends is

$$K \ge \frac{2 + 0.05L^{2.5}}{10} \frac{Y}{(L - U)^2} D^{\frac{1 + 0.5D}{1 - D}} \left(\frac{2}{n}\right)^{0.3} \left(\frac{LL}{SL}\right) \left(\frac{A1 - AL1}{A2 - AL2}\right) \text{ for } n < 4$$

$$K \ge \frac{2 + 0.05L^{2.5}}{7} \frac{Y}{(L - U)^2} D^{\frac{1 + 0.5D}{1 - D}} \left(\frac{2}{n}\right)^{0.1} \left(\frac{LL}{SL}\right) \left(\frac{A1 - AL1}{A2 - AL2}\right) \text{ for } n \ge 4$$

where

L = Total Length = |X| + |Y| + |Z|

- $Y = e x U \dots e$ is the Expansion Coefficient
- $\mathbf{U} = \sqrt{\mathbf{X}^2 + \mathbf{Y}^2 + \mathbf{Z}^2}$
- n = Number of Bends
- LL = Length of the Longest leg
- SL = Length of the Shortest leg

- A1= Length of starting anchor leg
- A2 = Length of concluding anchor leg
- AL1 = Length of leg attached to starting anchor leg
- AL2 = Length of leg attached to concluding anchor leg

PROGRAM IN VISUAL BASIC

The Program consists of two forms as follows: -

FORM1



Pressing Enter Key loads form 2.

FORM2



Entering in the various values and pressing commandbutton gives the value of "K".

The coding of the Program is as follows: -

Option Explicit

Dim a, k, l, ll1, ll2, ll3, u, e1, b, e, g, g1, n2, n1, l1, l2, d, d1, d2, d3, x1, y1, z1, d4

As Double

{ "Dim " is a command used to declare variables." Double" is a type of a variable.}

Private Sub dd_Click()

{ Procedure declaration of the Combobox "dd" (drop down box) used to input value of a diameter. Variable " d" is for diameter which is assigned values according to the value of diameter selected in the Combobox "dd". The Combobox gives values in inches as well as in millimetres. }

If dd.Text = "6inch-168.27" Then d = 168.27 If dd.Text = "8inch-219.07" Then d = 219.07

{Every variable has a text field where its value is assigned. Thus according to the value assigned to the text field of "dd", "d" is given a value. The value in this text field is selected from the list in the drop down box (combobox).}

If dd.Text = "46inch-1168.4" Then d = 1168.4 If dd.Text = "48inch-1219.2" Then d = 1219.2

End Sub {End of procedure dd_click.}

Private Sub t_Click()

If t.Text = "-28.89" Then e2.Text = "0.00001055"

If t.Text = "21.11" Then e2.Text = "0.00001093"

{Declaration of procedure t_click."t" is a Combobox for selecting the value of temperature of pipe. "e2" is the Textbox where the elongation of the pipe is shown corresponding to the temperature selected in

Combobox "t". IF statement assigns various values for "e2" corresponding to the value of "t" that is selected from its list. The value of "e2" is displayed in its textbox automatically.}

. If t.Text = "565.6" Then e2.Text = "0.00001449" If t.Text = "593.3" Then e2.Text = "0.00001462" End Sub {End of procedure t_click}

Private Sub Command1_Click()

{ Declaration of commandbutton "Command1" in form2. The Command1 Commandbutton has the caption "Show Constant K" on it.}

If n < 4 Then e = Val(e2) * Val(t) * 1000 { value of e in mm per metre is found}

l = Val(x) + Val(y) + Val(z)l = x + y + z

```
x1 = sqr(x)
\{x1 = square of x\}
y1 = sqr(y)
\{y1 = square of y\}
z1 = sqr(z)
\{z1 = square of z\}
b = x1 + y1 + z1
\{b = sqr x + sqr y + sqr z\}
u = pow(Val(b), 0.5)
\{u = square root of b\}
g1 = 1 - u
g = sqr(Val(g1))
\{g = square of the difference between l and u\}
```

n1 = (2 / n) n2 = pow(Val(n1), 0.3) {n2 = (2 / n) whole raise to 0.3 } e1 = e * u l2 = pow(Val(1), 2.5) { l2 = l raised to 2.5 } l1 = (2 + (0.05 * l2)) { The value of l2 is = 2 + 0.05L^{2.5} } d1 = 1 + 0.5 * Val(d) d2 = 1 - (Val(d)) d3 = (d1 / d2) d4 = pow(Val(d), Val(d3)) { The value of d4 is = $D^{\frac{1+0.5D}{1-D}}$ } l11 = Abs(a1 - al1)

{The value ll1 is the difference between the length of starting anchor leg and the length of leg attached to starting anchor leg.}

If ll1 = 0 Then ll1 = 1 {*If* ll1=0 then it is assigned value of 1} ll2 = Abs(a2 - al2) {*The value ll2 is the difference between the length of concluding anchor leg and the length of leg attached to concluding anchor leg.*} If ll2 = 0 Then ll2 = 1 ll3 = ll1 / ll2

k = (n2 * d4 * e1 * l1 * ll3 * ll) / (10 * sl * g)

MsgBox k

{The value of K is displayed in a message box }

If k < 1 Then

{Depending on the value of K being lesser than or greater than one the respective message box is displayed.}

MsgBox "THE PIPE IS SAFE"

Else: MsgBox "THE PIPE IS UNSAFE"

```
End If
{ End of inner If statement }
End If
{ End of outer If statement }
If n \ge 4 Then
{We use two loops for different number of bends i.e. for no. of bends less than four and greater than four.}
e = Val(e2) * Val(t) * 1000
l = Val(x) + Val(y) + Val(z)
x1 = sqr(x)
y1 = sqr(y)
z1 = sqr(z)
b = x1 + y1 + z1
u = pow(Val(b), 0.5)
g1 = 1 - u
g = sqr(Val(g1))
n1 = (2 / n)
n2 = pow(Val(n1), 0.1)
e1 = e * u
l2 = pow(Val(1), 2.5)
11 = (2 + (0.05 * 12))
d1 = 1 + (0.5 * Val(d))
d2 = 1 - (Val(d))
d3 = (d1 / d2)
d4 = pow(Val(d), Val(d3))
ll1 = Abs(a1 - al1)
If 111 = 0 Then 111 = 1
ll2 = Abs(a2 - al2)
If 112 = 0 Then 112 = 1
113 = 111 / 112
k = (n2 * d4 * e1 * l1 * ll * ll3) / (6 * s1 * g)
MsgBox k
If k < 1 Then
```

MsgBox "THE PIPE IS SAFE" Else: MsgBox "THE PIPE IS UNSAFE" End If End If

End Sub { End of procedure } Function sqr(i As Double) As Double {User Defined Function for squaring a number. The function returns a value of type "Double".} sqr = Val(i) * Val(i) {"sqr" is assigned a value equal to the square of the number. This value is returned.} End Function {End of function declaration}

Function pow(i As Double, j As Double) As Double {User Defined Function for finding the value of a number raised to another number. The function returns a value of type "Double".} Dim i1, i2 As Double

i1 = (j * Val(Log(i)))
{we first find the Log of the number.}

i2 = Exp(i1)
{then we find the antilog or exponential.}

pow = Val(i2)

{pow is assigned the value of i2. This value is returned.}

End Function

{End of function declaration.}

Private Sub Form_Load()

{The values of various standard diameters and operating temperatures have to be stored in " respective lists of the comboboxes "dd" and "t". This is done with the help of the "Additem" function, which adds the values in the order in which the values are typed. The commands are given in a procedure "Form_load" which gets executed at start of runtime. } dd.AddItem "6inch-168.27" dd.AddItem "8inch-219.07" . . . dd.AddItem "46inch-1168.4" dd.AddItem "46inch-1168.4" dd.AddItem "48inch-1219.2" t.AddItem "28.89" t.AddItem "21.11" . . t.AddItem "565.6" t.AddItem "593.3"

End Sub
{ End of "Form_load" procedure.}

The entire coding is shown below:-

Coding for Form1 :-

Option Explicit Private Sub Command01_Click () Unload form1 form2. Show End Sub

Coding for Form2 :-

Option Explicit

Dim a, k, l, u, e1, b, e, g, g1, n2, n1, 11, 12, d, d1, d2, d3, x1, y1, z1, d4 As Double

- Private Sub dd_Click()
- If dd.Text = "6inch-168.27" Then d = 168.27
- If dd.Text = "8inch-219.07" Then d = 219.07
- If dd.Text = "10inch-273.05" Then d = 273.05
- If dd.Text = "12inch-323.85" Then d = 323.85
- If dd.Text = "14inch-355.60" Then d = 355.6
- If dd.Text = "16inch-406.4" Then d = 406.4
- If dd.Text = "18inch-457.2" Then d = 457.2
- If dd.Text = "20inch-508.0" Then d = 508
- If dd.Text = "22inch-558.8" Then d = 558.8
- If dd.Text = "24inch-609.6" Then d = 609.6
- If dd.Text = "26inch-660.4" Then d = 660.4
- If dd.Text = "28inch-711.2" Then d = 711.2
- If dd.Text = "30inch-762.0" Then d = 762
- If dd.Text = "32inch-812.8" Then d = 812.8
- If dd.Text = "34inch-863.3" Then d = 863.3
- If dd.Text = "36inch-914.4" Then d = 914.4

If dd.Text = "38inch-965.2" Then d = 965.2 If dd.Text = "40inch-1016.0" Then d = 1016 If dd.Text = "42inch-1066.8" Then d = 1066.8 If dd.Text = "44inch-1117.6" Then d = 1117.6 If dd.Text = "46inch-1168.4" Then d = 1168.4 If dd.Text = "48inch-1219.2" Then d = 1219.2 End Sub

Private Sub t_Click()

If t.Text = "-28.89" Then e2.Text = "0.00001055" If t.Text = "21.11" Then e2.Text = "0.00001093"If t.Text = "93.33" Then e2.Text = "0.00001148" If t.Text = "148.9" Then e2.Text = "0.00001188" If t.Text = "204.4" Then e2.Text = "0.00001228" If t.Text = "260" Then e2.Text = "0.00001264"If t.Text = "315.6" Then e2.Text = "0.00001301"If t.Text = "343.3" Then e2.Text = "0.00001319" If t.Text = "371.1" Then e2.Text = "0.00001339" If t.Text = "398.9" Then e2.Text = "0.00001357" If t.Text = "426.7" Then e2.Text = "0.00001377" If t.Text = "454.4" Then e2.Text = "0.00001395" If t.Text = "482.2" Then e2.Text = "0.00001411" If t.Text = "510" Then e2.Text = "0.00001424"If t.Text = "537.8" Then e2.Text = "0.00001435"If t.Text = "565.6" Then e2.Text = "0.00001449"If t.Text = "593.3" Then e2.Text = "0.00001462" End Sub

Private Sub Command1_Click() If n < 4 Then

```
e = Val(e2) * Val(t) * 1000
l = Val(x) + Val(y) + Val(z)
x1 = sqr(x)
y1 = sqr(y)
z1 = sqr(z)
b = x1 + y1 + z1
u = pow(Val(b), 0.5)
g1 = l - u
g = sqr(Val(g1))
n1 = (2 / n)
n2 = pow(Val(n1), 0.3)
e1 = e * u
l2 = pow(Val(1), 2.5)
11 = (2 + (0.05 * 12))
d1 = 1 + 0.5 * Val(d)
d2 = 1 - (Val(d))
d3 = (d1 / d2)
d4 = pow(Val(d), Val(d3))
```

```
ll1 = Abs(a1 - al1)
If ll1 = 0 Then ll1 = 1
ll2 = Abs(a2 - al2)
If ll2 = 0 Then ll2 = 1
ll3 = ll1 / ll2
```

k = (n2 * d4 * e1 * l1 * ll3 * ll) / (10 * sl * g)MsgBox k

If k < 1 Then MsgBox "THE PIPE IS SAFE"

```
Else: MsgBox "THE PIPE IS UNSAFE"
End If
End If
If n \ge 4 Then
e = Val(e2) * Val(t) * 1000
l = Val(x) + Val(y) + Val(z)
x1 = sqr(x)
y1 = sqr(y)
z1 = sqr(z)
b = x1 + y1 + z1
u = pow(Val(b), 0.5)
g1 = 1 - u
g = sqr(Val(g1))
n1 = (2 / n)
n2 = pow(Val(n1), 0.1)
e1 = e * u
l2 = pow(Val(1), 2.5)
11 = (2 + (0.05 * 12))
d1 = 1 + (0.5 * Val(d))
d2 = 1 - (Val(d))
d3 = (d1 / d2)
d4 = pow(Val(d), Val(d3))
ll1 = Abs(a1 / al1)
If 111 = 0 Then 111 = 1
ll2 = Abs(a2 / al2)
If 112 = 0 Then 112 = 1
113 = 111 / 112
k = (n2 * d4 * e1 * l1 * ll * ll3) / (6 * sl * g)
MsgBox k
```

If k < 1 Then MsgBox "THE PIPE IS SAFE" Else: MsgBox "THE PIPE IS UNSAFE" End If End If End Sub

Function sqr(i As Double) As Double sqr = Val(i) * Val(i) End Function

Function pow(i As Double, j As Double) As Double Dim i1, i2 As Double i1 = (j * Val(Log(i))) i2 = Exp(i1) pow = Val(i2) End Function

Private Sub Form_Load()

dd.AddItem "6inch-168.27"

dd.AddItem "8inch-219.07"

dd.AddItem "10inch-273.05"

dd.AddItem "12inch-323.85"

dd.AddItem "14inch-355.6"

dd.AddItem "16inch-406.4"

dd.AddItem "18inch-457.2"

dd.AddItem "20inch-508.0"

dd.AddItem "22inch-558.8"

dd.AddItem "24inch-609.6"

dd.AddItem "26inch-660.4"

dd.AddItem "28inch-711.2"

dd.AddItem "30inch-762.0"

dd.AddItem "32inch-812.8"

dd.AddItem "34inch-863.3"

dd.AddItem "36inch-914.4"

dd.AddItem "38inch-965.2"

dd.AddItem "40inch-1016.0"

dd.AddItem "42inch-1066.8"

dd.AddItem "44inch-1117.6"

dd.AddItem "46inch-1168.4"

dd.AddItem "48inch-1219.2"

t.AddItem "-28.89"

t.AddItem "21.11"

t.AddItem "93.33"

t.AddItem "148.9"

t.AddItem "204.4"

t.AddItem "260"

t.AddItem "315.6"

t.AddItem "343.3"

t.AddItem "371.1"

t.AddItem "398.9"

t.AddItem "426.7"

t.AddItem "454.4"

t.AddItem "482.2"

t.AddItem "510"

t.AddItem "537.8"

t.AddItem "565.6"

t.AddItem "593.3"

End Sub

A solved Example is shown below : --

🛋 pipingsoffware	PIPING STRESS	ANALYSIS	<u>_</u>
Length Ot⊃beln S≻daras intro	rr C	Thermol Deflection (an deal	0.00001000 mm/mm/ mm/ C
Length Ot⊃ bein Micketton L(v)	m C	Lergthotthe Iongestmember	D. m
n nemOi≓ aela Z director L(t)	m	Lergit otike sinch dire ando c	^s m
tu∺rmshirt). Tibe c	3inen 138 27 💌 mm	emple of the member proceed to first and to:	5 m
No of Bends	3	taqti dilu membercoljocen: lutur: –utu	3 n
Mcter of Of Pipe	_A10E 3PA ⊯ B Tygs: Cercon Steel	t a gli atha memberattashed h sastant encorr	m
C∋erating Len suevalue	2 cagree C	Langth of the member adjacant fineae und ancorr	τ. π
	Shue G re -n	X	

Values are input as shown above and commandbutton command1 with caption "Show constant K" is pressed which displays a message box.



On pressing "OK" another message box saying whether the pipe is safe or unsafe is displayed.



The values can be changed and any number of further computations can be carried out in the manner explained.

Sr	X	Y	Z	D	n	LL	SL	A1	A2	Al1	Al2	S/A	K
No	(m)	(m)	(m)	(mm)		(m)	(m)	(m)	(m)	(m)	(m)		
1.	6	12	12	323.85	5	7	2	2	5	5	7	0.53	2.62
2.	56	15	10	406.4	5	40	25	26	25	40	25	21.07	114.59
3.	50	40	55	508	4	40	20	25	20	40	25	13.9	23.04
4.	16	25	20	457.2	8	20	10	15	15	20	15	30.54	10.45
5.	8	3	13	323.85	5	8	3	5	5	8	8	0.76	1.34
6.	3	17	7	273.05	4	11	7	8	8	11	9	2.25	4.2
7.	6	7	8	219.07	2	8	6	8	7	6	6	0.56	0.64
8.	18	1.5	10	355.6	9	8	3	6	5	6	5	4.08	2.53
9.	8	8	5	168.25	3	8	3	5	3	8	5	0.67	1.02
10.	25	12	17	323.85	4	14	8	12	8	14	11	3.49	1.45
11.	11	14	13	355.6	4	11	5	5	6	8	11	1.81	0.8
12.	11	14	13	355.6	4	11	5	5	11	8	6	0.85	0.8
13.	16	13	6	355.6	6	8	5	5	7	8	7	2.42	3.1
14.	16	11	8	219.04	4	8	5	8	8	5	6	1.00	1.85
15.	5	-1	12	273.05	4	14	9	11	9	14	10	4.49	3.12
16.	13	-11	12	323.85	2	3	11	11	12	13	13	1.03	0.23
17.	-12	15	12	323.85	3	15	5	7	5	15	12	1.21	1.51
18.	9	8	0	273.05	3	9	8	9	9	8	9	1.05	0.4
19.	6	5	6	219.07	3	9	3	9	6	5	3	0.75	0.62
20.	3	4	10	273.05	3	5	3	5	5	4	3	0.34	0.2
21.	4	15	6	406.4	3	8	4	7	6	4	8	0.39	1.11
22.	10	5	8	406.4	2	10	5	10	5	8	8	0.47	0.3
23.	3	10	4	323.85	3	5	3	3	5	5	4	0.26	0.74
24.	5	20	5	323.85	3	10	5	5	10	10	5	0.48	1.43
25.	10	10	10	323.85	4	10	5	5	5	5	5	0.88	0.53
26.	3	8	2	168.27	3	6	2	3	5	6	2	0.27	0.74
27.	3	11	2	168.27	3	6	2	3	5	6	2	0.57	1.4

Comparison of the new formula with the results obtained from Caepipe

28.	20	30	10	168.27	3	20	10	20	15	15	10	7.85	3.05
29.	30	25	55	457.2	5	30	10	10	25	15	20	12.28	9.69
30.	23	11	65	558.8	6	25	5	13	25	5	10	8356	12.98
31.	2	2	10	114.3	2	10	2	2	2	10	10	0.70	3.1
32.	4	15	3	141.3	5	5	2	2	5	5	2	0.47	2.83
33.	4	7	12	219.07	5	8	2	2	4	3	2	0.85	0.99
34.	3	3	2	219.07	2	3	2	3	2	3	3	0.19	0.1
35.	20	13	31	219.07	5	16	6	10	7	6	16	6.91	2.38
36.	21	9	4	457.2	3	17	4	4	17	9	4	0.55	1
37.	4	28	2	101.6	3	14	2	14	14	4	2	0.32	32.98
38.	3	4	16	219.07	3	8	3	8	8	4	3	0.75	1.56
39.	33	5	6	273.05	4	11	5	11	11	5	6	2.28	10.12
40.	13	9	0	219.07	3	7	4	7	4	5	6	0.84	1.09

Thus we can see that the results are quite satisfactory except for some cases where our formula shows that the pipe is unsafe whereas in actual case it is safe. This type of difference can cause no harm in the calculation of the values. The pipe will have to be checked on Caepipe again after check by the normal formula. This gives double safety.

CONCLUSION

ADVANTAGES

- With increasing diameter the pipe becomes safer. This drawback that existed in the old formula has been dealt with.
- The old formula did not account for the number of bends. The formula derived does account for number bends. With increase in bends the pipe becomes safer.
- The old formula did not consider anchor lengths which have a considerable effect on the value of 'k' this has been considered.
- If one of the length is too short or too long then the pipe may become unsafe this was not included in the old formula. We have comsidered this in the new formula.
- The formula is a quick and easy check for pipe configurations. The formula gives results in conformance to CAEPIPE results in most of the simple standard configurations.

DRAWBACKS

- The formula gives very accurate results for simple two anchor with three orientations. It does not account for three anchor problems
- When the number of bends are high there are multiple combinations possible with other factors remaining same so the S/A ratio in Caepipe may change but the formula gives only one value which is the average of all such S/A ratios of Caepipe.
- The formula gives a bit higher values in some case. The piping engineer may have to carry out analysis once more on some software like Caepipe, Ceaser II etc. But this is justified as it ensures double safety.