Chemical Process Utilities
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Module No # 09
Lecture No # 42
Natural Gas Transmission-IV

Welcome to the pipeline mechanical design as you know that we were discussing about the natural gas transmission under the area of chemical process utilities. Let us have a look about that what we discussed previously? We discussed about the natural gas transmission with the pipeline gas velocity and erosional velocity. Then we discussed about the optimum pressure drop for design and pipeline packing.

Determination of different type of gas leakage wall thickness and pipe grades we establish the relationship between the wall thickness and design pressure and talked about the temperature profile.

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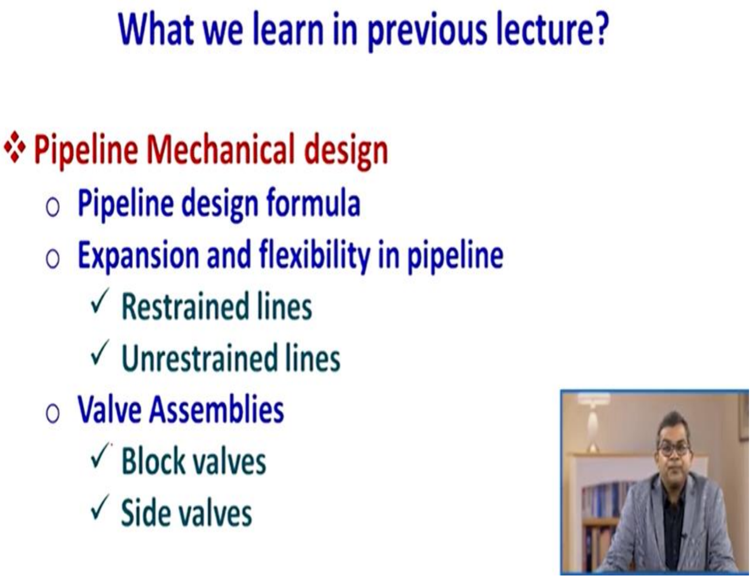

In this particular lecture we are going to discuss about the mechanical design aspects of pipeline. Now in which we will discuss about the pipeline design formula, expansion and flexibility in pipeline, we will discuss about the restrained lines then unrestrained lines. We will discuss about the valve assemblies including the block valves, side valves.

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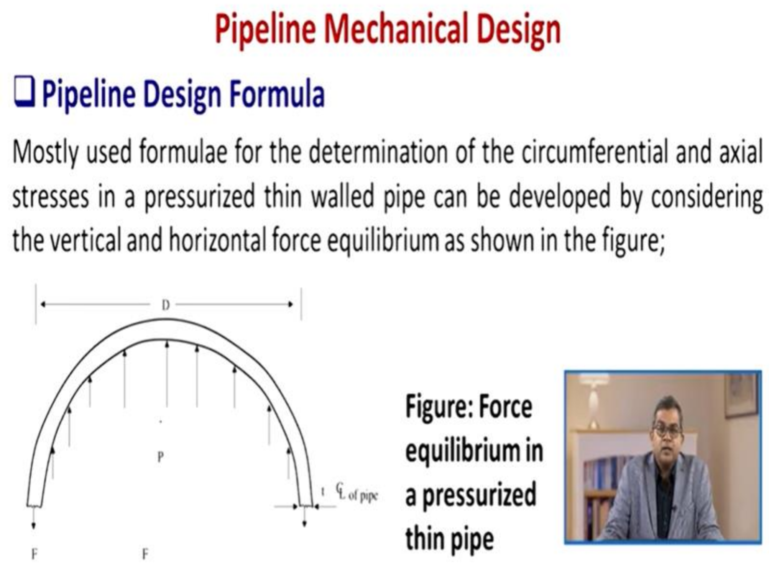

(refer time: 01:29) So when we talk about the pipeline mechanical design, so first thing is that we need to establish the pipeline design formulation. Now it is mostly used formulae for determination of circumferential x and axial stresses in the pressurized thin-walled pipe it can be developed by considering the vertical and horizontal force equilibrium. Which; we have shown in this figure, now here you see that this force equilibrium in a pressurized thin pipe.

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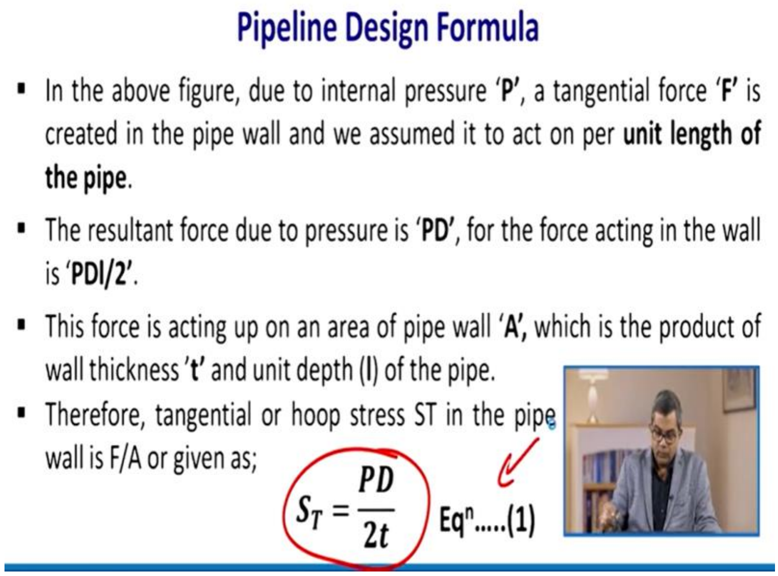

Now in this particular figure due to internal pressure P, a tangential force F, you see over here is created in the pipe wall and we assumed it to act on per unit length of the pipe. The resultant force due to pressure is say P D, and force acting in the wall is PDI by 2. Now this force is acting up on an area of pipe wall which is A, and this is the product of wall thickness t and the unit length l of the pipe.

So therefore, tangential or hoop stress S T in the pipe wall is F over A and it is given as; S T = P D over 2 t and that is equation number 1.

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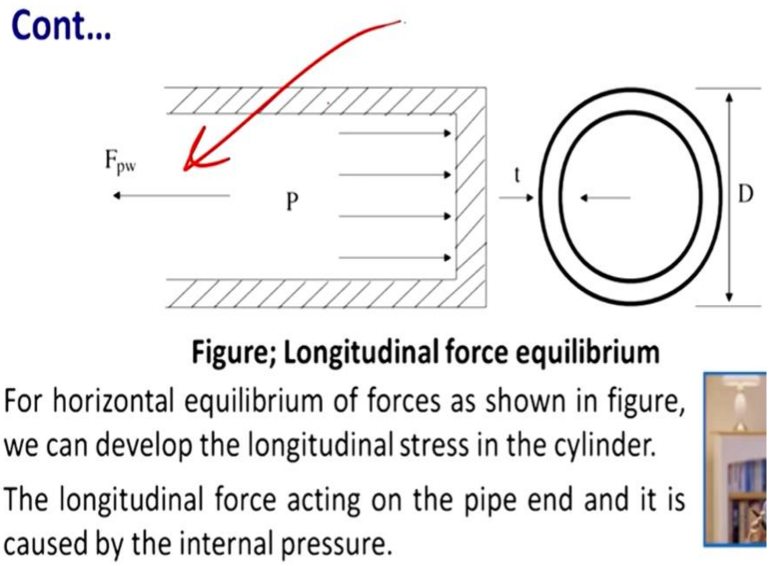
tangential or hoop stress ST in the pipe wall is F/A or given as;

$$S_T = \frac{PD}{2t} \qquad \text{Eq}^{n} \text{.....(1)}$$



Now for horizontal equilibrium the forces which are shown in this figure can develop the longitudinal stress in the cylinder. The longitudinal force acting on the pipe and it is caused by the internal pressure.

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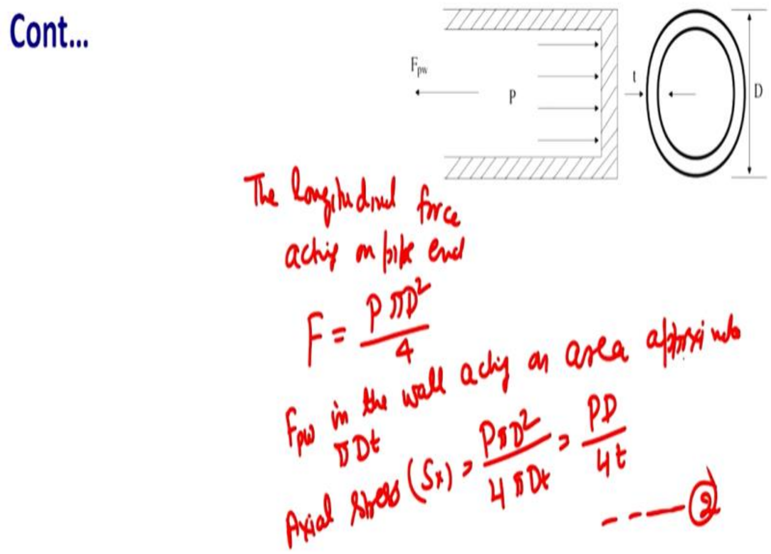

Now, the longitudinal force acting on pipe longitudinal force acting on pipe and this is caused by internal pressure that is F = P pi D square over 4. The longitudinal force is equilibrated by the force F p w in the wall acting on area approximated by pi D t. So axial stress S X = P pi D square upon 4 pi D t and that = P D 4 over 4 t that is equation number 2.

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The Longitudinal force acting on pipe end caused by internal pressure is;

$$\boldsymbol{F = \frac{P\pi D^2}{4}}$$

The longitudinal force is equilibrated by force (**F_{PW}**) in the wall acting on area approximated by **πDt**, so the axial stress **S_X** will be;

$$\boldsymbol{Axial\ Stress\ (S_X) = \frac{P\pi D^2}{4\pi Dt} = \frac{PD}{4t}} \qquad \textbf{Eq}^{\textbf{n}} \textbf{.....(2)}$$

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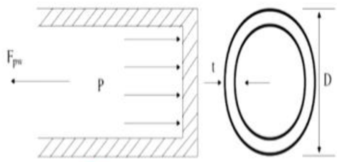

Axial Stress $(S_x) = \dfrac{F}{\frac{\pi(D^2-d^2)}{4}} = \dfrac{Pd^2}{D^2-d^2}$ ---- (3)

D & d
outer and inner pipe diam
Z of thin walled cylinder

$Z = \pi r^2 t$ $\quad Z = \dfrac{\pi}{32}\dfrac{(D^4-d^4)}{D}$

Now for most accurate formula of axial stress will be fined by using actual area of the pipe wall bearing the longitudinal pressure force. And that axial stress S X = F upon pi D square - small d square upon 4 and that = P d square upon D square - d square and that is equation number 3. Now where this capital D and d they are the outer and inner pipe diameter, now this is less conservative than using thin wall approximation.

The bending section modulus Z of a thin-walled cylinder this can be approximated as where r means if we take r as a mean radius then, Z = pi r square t or Z = pi over 32 D to the power 4 - small d to the power 4 upon D.

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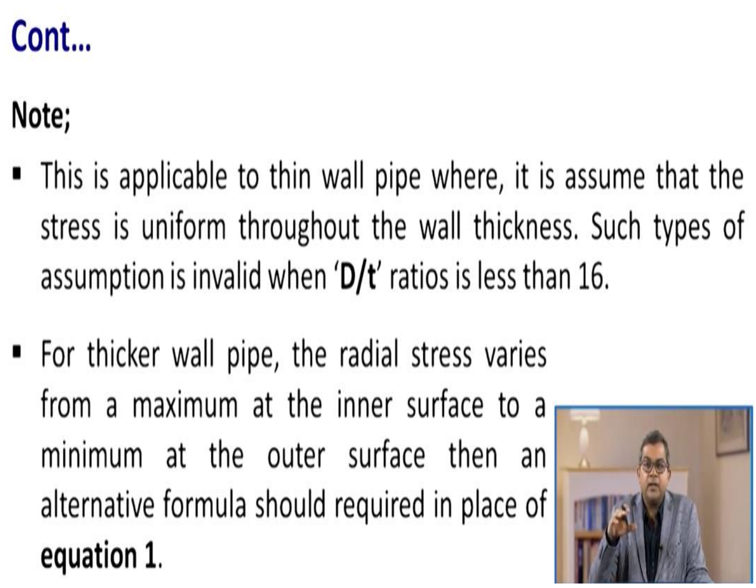
For most accurate formula of axial stress will be found by using actual are of the pipe wall bearing the longitudinal pressure force, is;

$$\boldsymbol{Axial\ Stress\ (S_X) = \frac{F}{\frac{\pi(D^2-d^2)}{4}} = \frac{Pd^2}{D^2-d^2}}$$

Where, D and d, are the outer and inner pipe diameter respectively. This is less conservative than by using thin-wall approximation. The bending section modulus Z of a thin-walled cylinder can be approximated as; where 'r' is mean radius;

$$\boldsymbol{Z = \pi r^2 t} \qquad \text{Exact,} \qquad \boldsymbol{Z = \frac{\pi}{32}(D^4-d^4)/D}$$



Now this is applicable to thin wall pipe where, it is assumed that the stress is uniform throughout the wall thickness. Now such type of assumption is invalid when D over t ratio is less than 16. For thicker wall pipe, the radial stress varies from a maximum at inner surface to a minimum at the outer surface then an alternative formula should require in place of equation 1 which we have discussed earlier.

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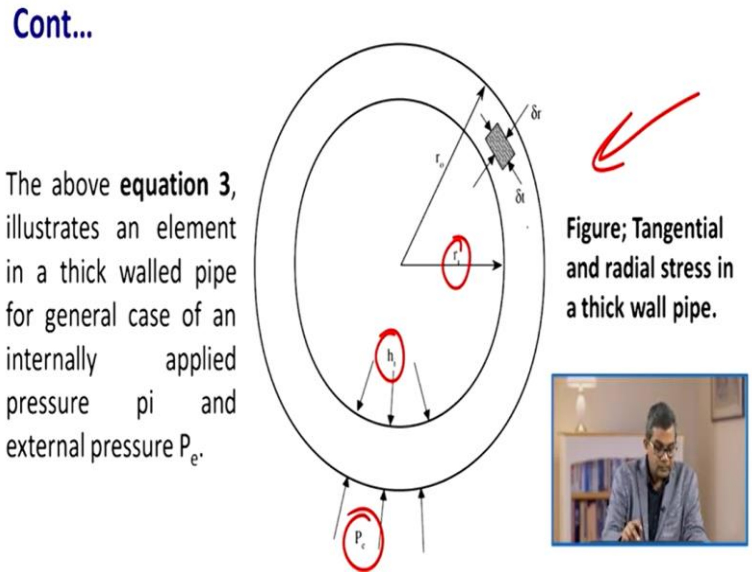

Now as we were discussing about the equation 3 this equation is 3, illustrates an element of thick-walled pipe for general case of an internally applied pressure p i and external pressure that is P e, now this figure represents the tangential and radial stress in a thick wall pipe. Now here you see P e and this is an internal diameter internal in radius.

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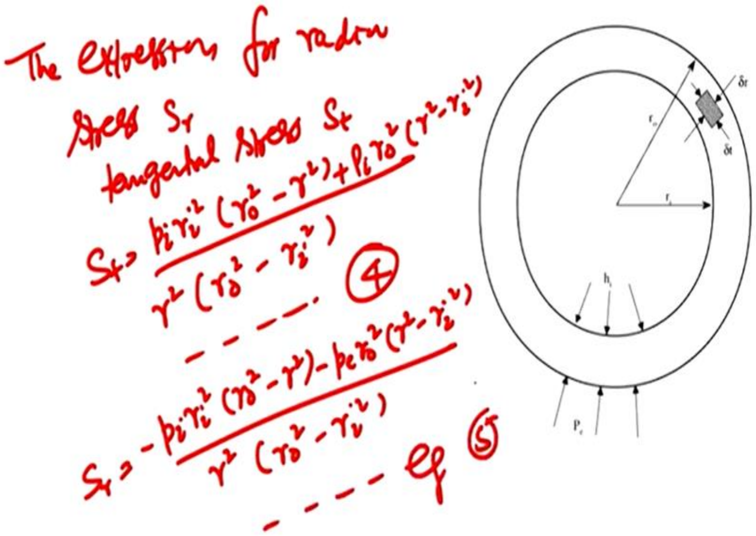

Now the expression for radial stress S r and tangential S t apply to the element of at any radius. Now S t = p i r i square r naught o square - r square + p i r o square r square - r i square over r square r o square - r i square and that is equation number 4. And S r = p i r i square r naught square - r square - p e r naught square r square - r i square upon r square r naught square - r i square and that is equation number 5.

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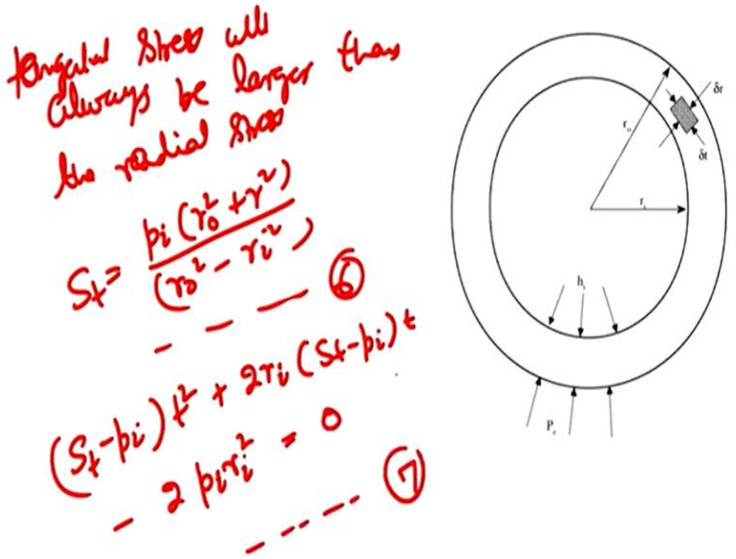
The expression for radial stress S_r and tangential stress S_t apply to the element at any radius r.

$$S_t = \frac{p_i r_i^2 (r_o^2 - r^2) + P_i r_o^2 (r^2 - r_i^2)}{r^2 (r_o^2 - r_i^2)}$$

And, $$S_r = \frac{-p_i r_i^2 (r_o^2 - r^2) - p_e r_o^2 (r^2 - r_i^2)}{r^2 (r_o^2 - r_i^2)}$$ **Eq^n …..(5)**



Now in this equation number four which we discussed in this particular slide; the numerator will always exceed that of equation number 5. Now that is the tangential stress will always be larger than the radial excess stress. So, when pipe is subjected only to internal pressure then, the previous equation becomes S t = p i r o square + r square upon r o square - r i square and that is equation number 6.

So, on substitution of say r o in terms of r i and t and if we rearrange the equation then it becomes S t - p i t square + 2 r i s t - p i t - 2 p i r i square = 0 and that is equation number 7.

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In the above equation 4, the numerator will always exceed that of equation 5, i.e., the tangential stress will always be larger than the radial stress.

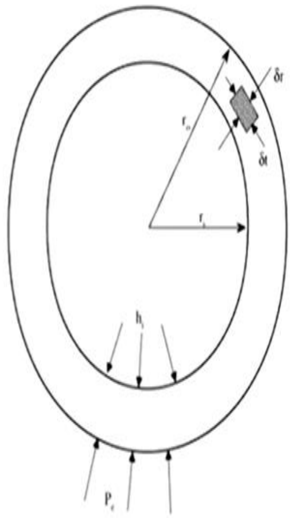
When the pipe is subjected only to internal pressure, then equation 4 becomes;

$$S_t = \frac{p_i(r_o^2 + r^2)}{(r_o^2 - r_i^2)} \qquad \textbf{Eq}^{\textbf{n}} \textbf{.....(6)}$$

On substitution of 'r_o' in terms of 'r_i' and 't' and rearranging the terms we have;

$$(S_t - p_i)t^2 + 2r_i(S_t - p_i)t - 2p_ir_i^2 = 0 \qquad \textbf{Eq}^{\textbf{n}} \textbf{.....(7)}$$

$$t = \frac{D}{2}\left(\left(\frac{S_t+p_i}{S_t+p_i}\right)^{0.5}-1\right) \quad (8)$$



Now if we solve these equations then t = D by 2 s t + p i upon s t + p i 0 to the power half - 1 and that has become the equation number 8. Now these equations may be arranged to determine the bursting pressure of the pipe which occurs when the tangential stress equal to ultimate tensile stress.

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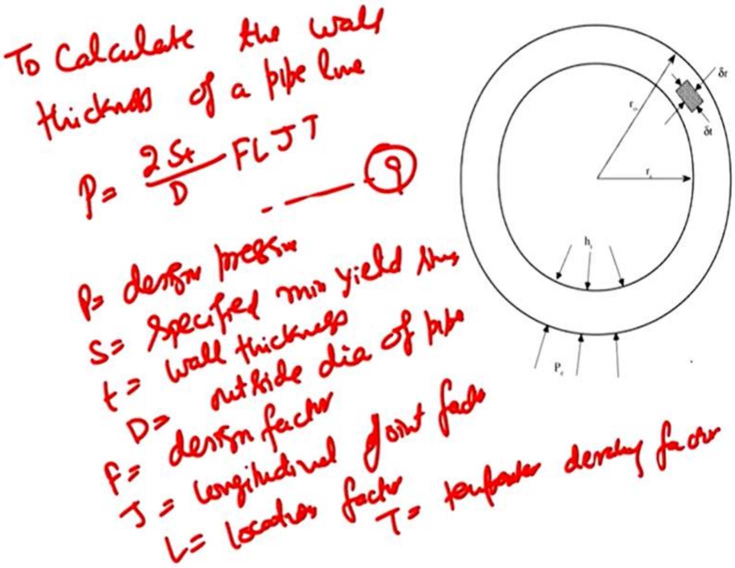
On solving equation 7 we have;

$$\boldsymbol{t = \frac{D}{2}\left(\left(\frac{S_t + p_i}{S_t + p_i}\right)^{0.5} - 1\right)} \qquad \textbf{Eq}^{\textbf{n}}\textbf{.....(8)}$$

Note;

The above equation 6 or 8 may be arranged to determine the bursting pressure for the pipe, which occurs when the tangential stress equals the ultimate tensile stress.



Now to calculate the wall thickness of a pipeline, now this design formula can be used that is P = 2 s t upon D F L J T and that is equation number nine. Now where P is the design pressure, S is a specified minimum yield strength, t is wall thickness, D is outside dia of pipe, F design factor, J longitudinal joint factor, L location factor and T is equal to temperature derating factor.

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To calculate the wall thickness of a pipeline the following design formulae are used;

$$P = \frac{2St}{D} FLJT \qquad \text{Eq}^{n} \ldots\ldots(9)$$

Where,

P= design pressure,

S= specified min yield strength

t= wall thickness

D= outside dia of pipe

F= design factor, J= Longitudinal Joint factor

L= location factor, T= temperature derating factor

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Table 1; Comparison of design factors in North American codes (Ref. CSA Z662, 2007 and ASME B31.8, 2003);

Application	CSA Z662-07 (Gas & liquid)			AMSE B31.8-03 (gas) and ASME B31.4-02 (liquid)		
	Design Factor x Location factor (FxL)			Design Factor (F)		
	Class 1	Class 2	Class 3	Class 1	Class 2	Class 3
Gas (nonsour)						
General & cased crossings	0.80	0.72	0.56	0.80	0.60	0.50
Roads	0.60	0.50	0.50	0.60	0.50	0.50
Railways	0.50	0.50	0.50	0.60	0.50	0.50
Stations	0.50	0.50	0.50	0.50	0.50	0.50

Now this in this table there is a comparison of design factor in North American codes. We have given the reference now, if you see that we are having enlisted various applications like gas and apart from this we have the Canadian standard association references and then ASME standards. So, if we take the design factor and location factor for CSA as well as the design factor for gas AMSE.

Then the general and seized crossing general and case crossing the design for class 1 0.80, and class 2 0.72, and for class 3 it is 0.56. Now if you compare with the AMSE or ASME codes course then the class 1 0.80, class 2 0.60, and class 3 0.50. Similarly, if you take the roads into consideration, it is 0.60 for class 1 under CSA and 0.50 in class 2 and 0.50 again in class 3.

Where ASME suggests 0.60 for class 1, class 2 0.50 and class 3 0.50 again. For railways in CSA, it is 0.50, 0.50, and 0.50 respectively for all 3 classes and 0.60, and 0.50, 0.50 respectively for all 3 classes under ASME. Now if we take the stations then it is uniform across all 3 classes in CSA and also in under ASME it is again uniform to 0.50 in all 3 classes.

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Table (Cont...)

Gas (sour services)						
General & cased crossings	0.80	0.72	0.56	0.80	0.60	0.50
Roads	0.60	0.50	0.50	0.60	0.50	0.50
Railways	0.50	0.50	0.50	0.60	0.50	0.50
Stations	0.50	0.50	0.50	0.50	0.50	0.50
High Vapor pressure Liquid General & cased crossings		0.64	0.64	0.72	0.72	0.72
Roads		0.64	0.64	0.72	0.72	0.72
Railways		0.50	0.50	0.72	0.72	0.72
stations		0.64	0.64	0.72	0.72	0.72
Low vapor pressure liquid (RR crossing)		0.80	0.80	0.72	0.72	0.72
Uncased railroad crossings		0.80	0.50	0.72	0.72	0.72

Now if you take the general and cased crossing with respect to the roads and railways are there is a slight variation between these 2 standards CSA and ASME. Another class is that high vapor pressure or liquid general cased crossing, in which the roads they are having the 0.64 and 0.64 in class 2 and class 3 in CSA whereas, 0.72 in all cases of class 3 under ASME. Similarly, if we talk about the railways the class 2 and class 3 are 0.50 each in CSA, and 0.72 each in ASME in all 3 different classes.

Similarly in stations 0.64, 0.64 in class 2 and class 3 and 0.72 is common for all 3 classes under ASME. Similarly, low vapor pressure liquid that is R R crossing then it is 0.80 and 0.80 for class 2 and class 3 in CSA and 0.72 is common for all 3 classes of ASME. Uncased railroad crossings it is 0.80 and 0.50 in class 2 and class 3 in CSA and 0.72 is common for all 3 classes of ASME

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Table 2; Comparison of longitudinal joint factors in North American codes

Type of pipe	Longitudinal joint factor, J (CSA Z662-07) or E (AMSE B31.4-02 and B31.8-03)	
	CSA Z662-07	AMSE B31.4-02 and B31.8-03
Seamless	1.00	1.00
Electric welded	1.00	-
Resistance welded		1.00
Induction/flash welded		1.00
Fusion arc welded		0.80
Fusion welded		0.80-1.00
Submerged arc welded	1.00	1.00
Spiral welded		0.80
Furnace butt welded	0.60	0.60

Now here you can see the comparison of longitudinal joint factors in North American courts, and again the seamless and again this is compared with the CSA and ASME and seamless for CSA is 1.00 and it is similar to ASME. The electric welded the applicable code is CSA that is 1.00 and similarly, if you see that submerged arc welding again both the values are similar. And the furnace but welded the both the values are similar apart from this all the values are pertaining to ASME.

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Table 3; Comparison of temperature Derating factors for North American codes (Steel pipe) Ref; CSA Z662, 2007 and AMSE B31.8, 2003)

Temperature (oC)	Temperature Derating Factor (T)	
	CSA Z662-07 &AMSE B31.8	AMSE B31.4
>120	1.00	-
150	0.97	-
180	0.93	-
200	0.91	-
130	0.87	-
>30	-	1.00
<120	-	1.00

Now if we compare the temperature derating factors in both the references then the temperature profiling from 120, to say 150, 180, 200, and 130 only CSA and ASME ratings are same. And similarly, if we take the 30 degrees Celsius and greater than 120 degrees Celsius in between 120 degrees Celsius the ASME reported 1.00.

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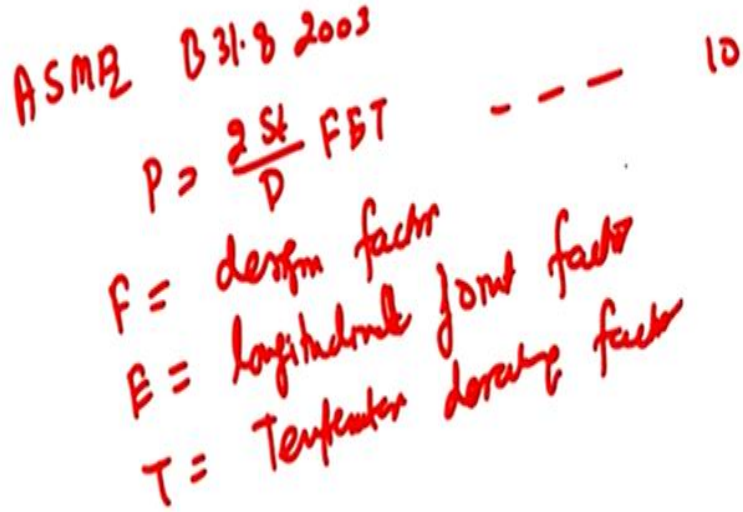

Now to calculate if we take this ASME B31.8 2003 to calculate the wall thickness of a pipeline now the design formula would be, P = 2 s t over D F E T and that is equation number 10. Now here F is the design factor based on location classification we can refer to the table, now E is the longitudinal joint factor, and T is the temperature derating factor.

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<u>ASME B31.8, 2003</u>; To calculate wall thickness of a pipeline the design formulae will be;

$$P = \frac{2St}{D} FET \qquad \text{Eq}^{n} \ldots..(10)$$

Where,

F= design factor based on the location classification; see table 1,

E= longitudinal joint factor; see table 2

T= Temperature derating factor; see table 3

ASME B31.4 2002 --- 11

$P = \frac{2St}{D} FE$

F= design factr 0.72

E=

T → temp deraty factr is 1.0

$-30°C \le T_{pipe} \le 120°C$

Now if we take ASME B31.4 2002 to calculate the wall thickness of a pipeline the design formula would be, P = 2 s t over D F E and this is equation number 11. Now again F is the design factor 0.72, E is the longitudinal joint factor, and temperature derating factor T is 1.0 provided it is in lying between - 30 degree Celsius to 120 degrees Celsius, now T pipe is the pipe temperature.

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<u>ASME B31.4, 2002</u>; To calculate wall thickness of a pipeline the design formulae will be;

$$P = \frac{2St}{D} FE \qquad \text{Eq}^n \text{.....(11)}$$

Where,

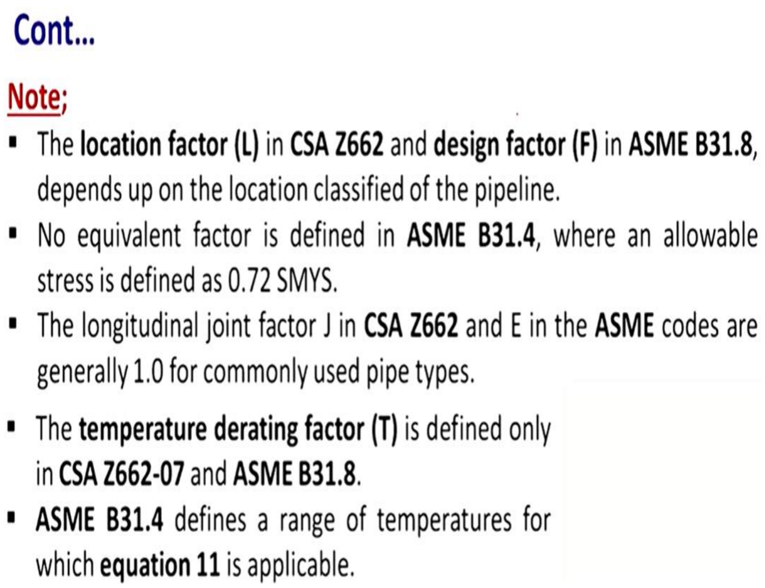
F= design factor =0.72; see table 1,

E= longitudinal joint factor; see table 2

Temperature derating factor (**T**) is 1.0 provided;

$$-30℃ \le T_{pipe} \le 120℃$$

Where, T_{pipe} is the pipe temperature



Now the location factor in CSA Z662 and the design factor F in ASME B31.8, depends on the location classification of the pipeline. No equivalent factor is defined in ASME codes where an allowable stress is defined as 0.72 SMYS. The longitudinal joint factor j in CSA Z662 code and E in the ASME code are generally one for commonly used pipe types. The temperature derating factor T is defined only in CSA Z662-7 and ASME 31.8. ASME B31.4 defines a range of temperature for which equation previous equation which we derived earlier is applicable.

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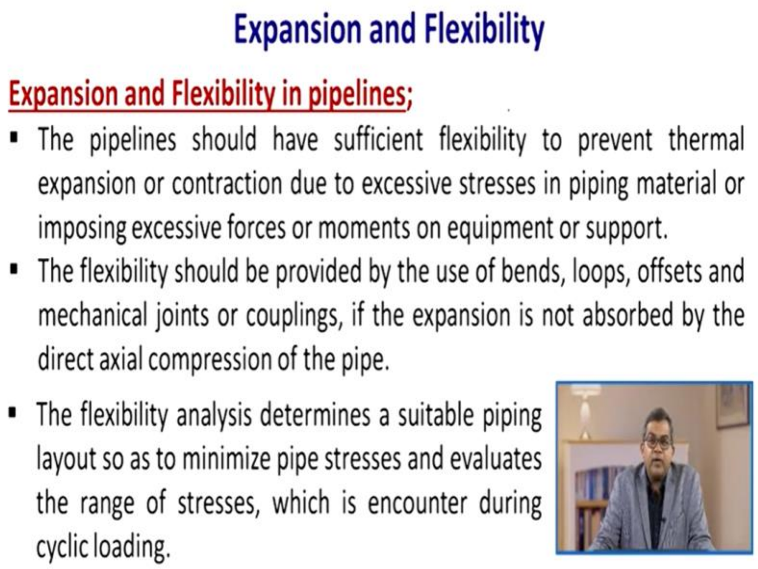

Now let us talk about the expansion and flexibility. The expansion and flexibility in pipeline; the pipeline should have sufficient flexibility to prevent the thermal expansion or contraction due to excessive stresses in piping material or imposing excessive forces or movements on equipment or support. The flexibility should be provided by the use of bends, loops, offsets, and mechanical joints or couplings, now if the expansion is not absorbed by the direct axial compression of the pipe.

The flexibility analysis this determines a suitable piping layout so as to minimize pipe stresses and evaluates the range of stresses which is encountered during the cyclic loading. The most common range is the thermal stresses range and this is caused by the system start up and shutdown conditions. There are some fundamental differences in loading conditions for buried or similarly restrained portion of piping and above ground portions not subjected to axial restraints. So, different limits on allowable expansion stresses are necessary.

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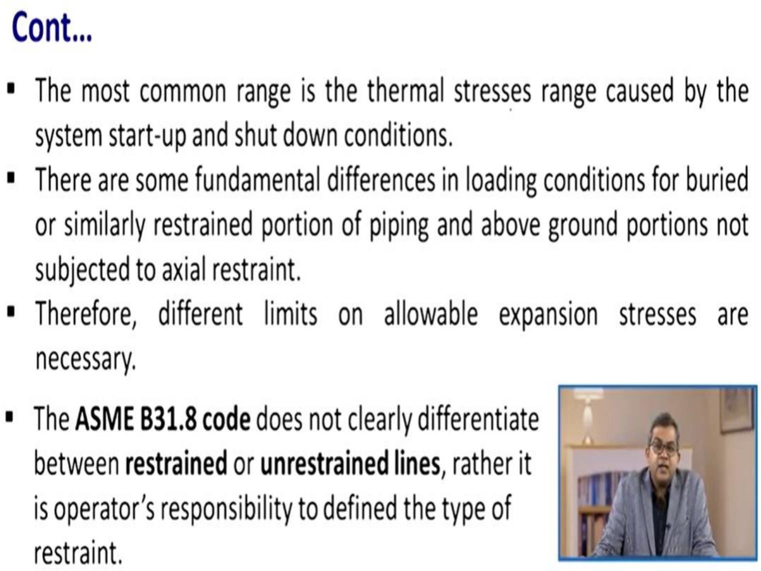

Now the ASME B31.8 code it does not clearly differentiate between the restrained or unrestrained lines, rather it is operator's responsibility to define the type of restraint.

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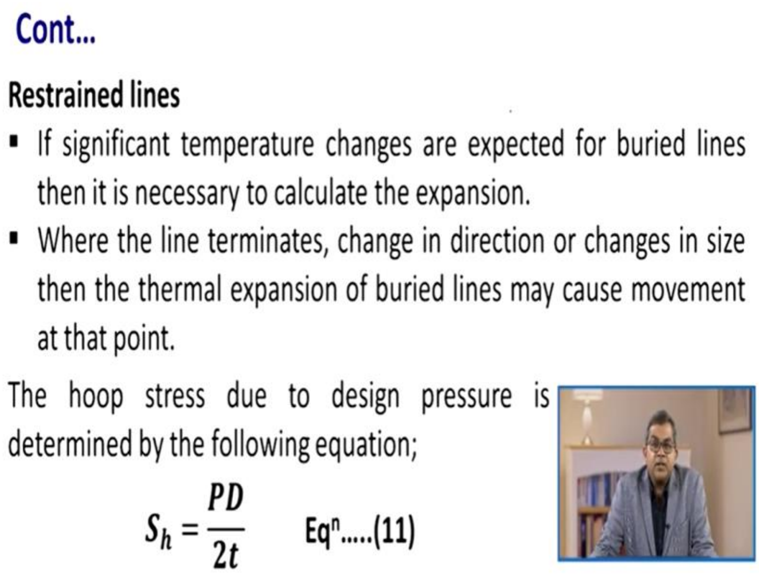

Now let us talk about the restrained lines, now if significant temperature changes are expected for buried lines, then it is necessary to calculate the expansion. Now where the lines terminate, change in direction or changes in size then that thermal expansion of buried lines may cause movement at that point. The hoop stress due to design pressure is determined by this particular equation that is S h = P D over 2 t and that is our equation number 11.

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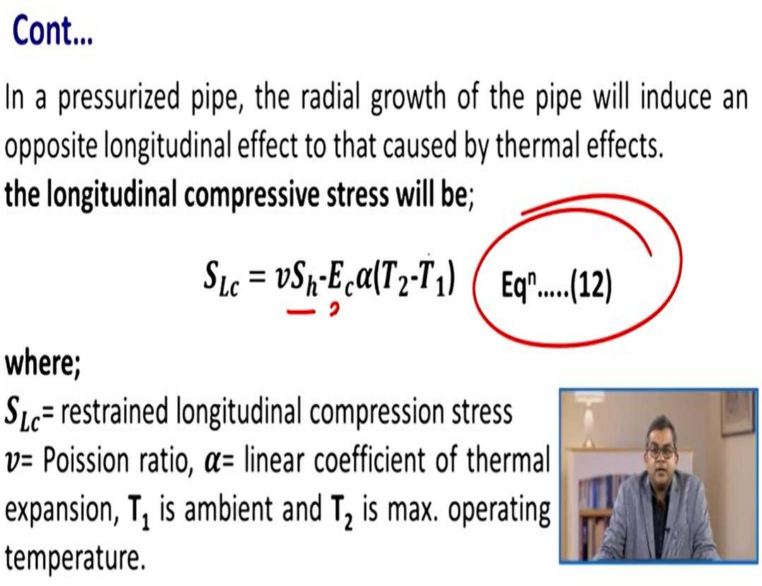
The hoop stress due to design pressure is determined by the following equation;

$$S_h = \frac{PD}{2t} \qquad \textbf{Eq}^{\textbf{n}}\textbf{.....(11)}$$



Now in the pressurized pipe, the radial growth of the pipe will induce an opposite longitudinal effect to that caused by the thermal effect. So, the longitudinal compressive stresses would be S L C = v S h – E C alpha T 2 – T 1 that is equation number 12. Where; S L C is the restrained longitudinal compression stress, v is the Poisson ratio, alpha is the linear coefficient of the thermal expansion, T 1 is the ambient and T 2 is the maximum operating temperature.

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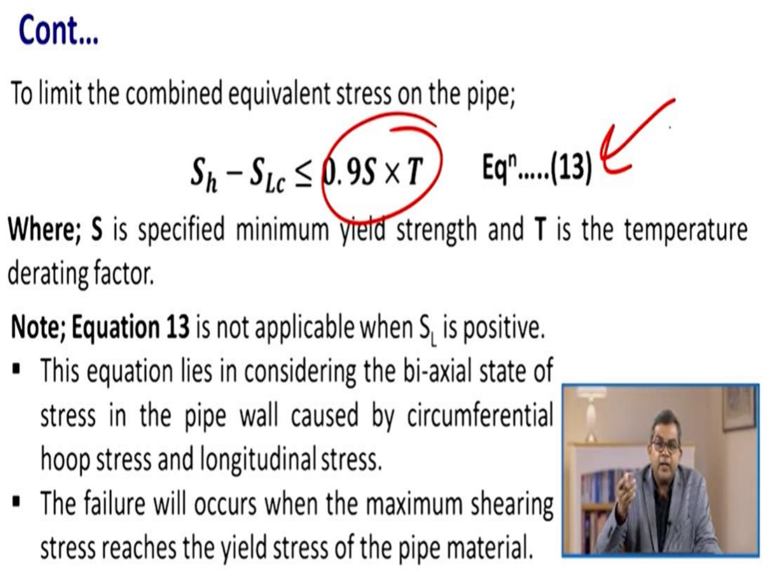
the longitudinal compressive stress will be;

$$S_{Lc} = vS_h - E_c\alpha(T_2 - T_1) \qquad \text{Eq}^n \ldots\ldots(12)$$

where;

S_{Lc}= restrained longitudinal compression stress

v= Poission ratio, α= linear coefficient of thermal expansion, T_1 is ambient and T_2 is max. operating temperature.



Now to limit the combined equivalent stress on the pipe; S h – S L C is less than equal to 0.9 S T and that is equation number 13. Now where; S is the specified minimum yield strength and T is the temperature derating factor. Now this equation is not applicable when S L is positive. Now this equation lies in considering the bi-axial state of stress in the pipe wall caused by the circumferential hoop stress and longitudinal stress.

The failure will occur when the maximum shearing stress reaches the yield stress of the pipe material.

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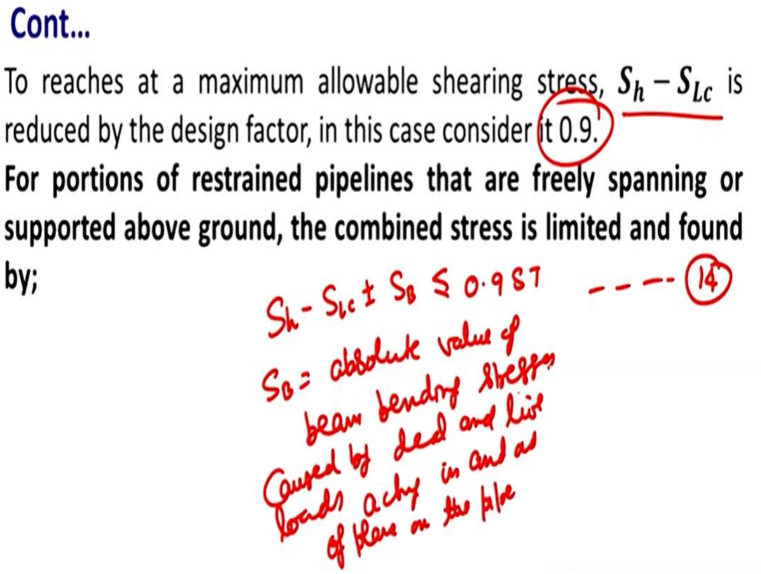
To limit the combined equivalent stress on the pipe;

$$S_h - S_{Lc} \leq 0.9S \times T \qquad \text{Eq}^{n} \text{.....(13)}$$



Now to reach at a maximum allowable shearing stress S h – S L C is reduced by the design factor, and in this case, it is 0.9. Now for portion of restrained pipeline that are freely spanning or supported above ground, the combined stress is limited and usually found by S h - S L C plus minus S B less than equal to 0.9 S T and that is equation number 14. Now where S B = absolute value of beam bending stresses caused by dead and live loads acting in and out of plane on the pipe.

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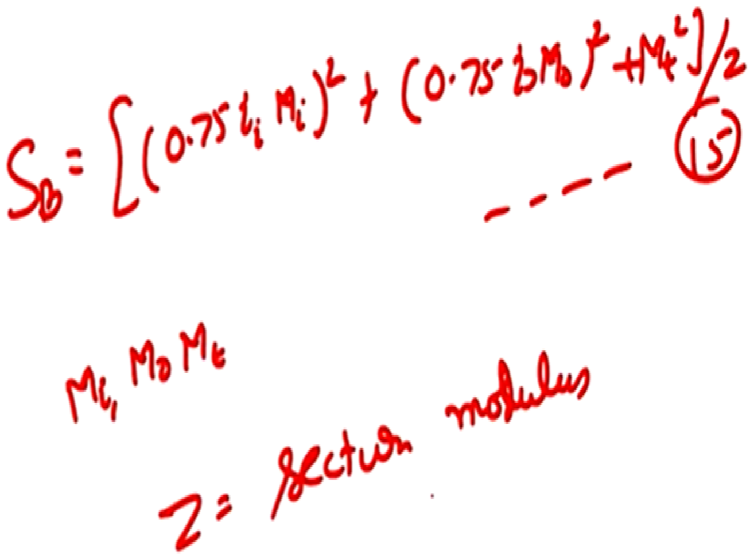
For portions of restrained pipelines that are freely spanning or supported above ground, the combined stress is limited and found by;

$$S_h - S_{Lc} \pm S_B \leq 0.9ST \qquad \text{Eq}^{n} \text{.....(14)}$$

Where, S_B= absolute value of beam bending stresses caused by dead and live loads acting in and out of plane on the pipe.



So, the value of S B can be found out by this equation that is S B = 0.75 i i M i square + 0.75 i o M o square + M t over Z and that is equation number 15. Now where this M i M o and M t are in plane out of plane and the torsional movement respectively acting on the pipe and Z is the section modulus i and i o is the in and out of plane stress intensification factors.

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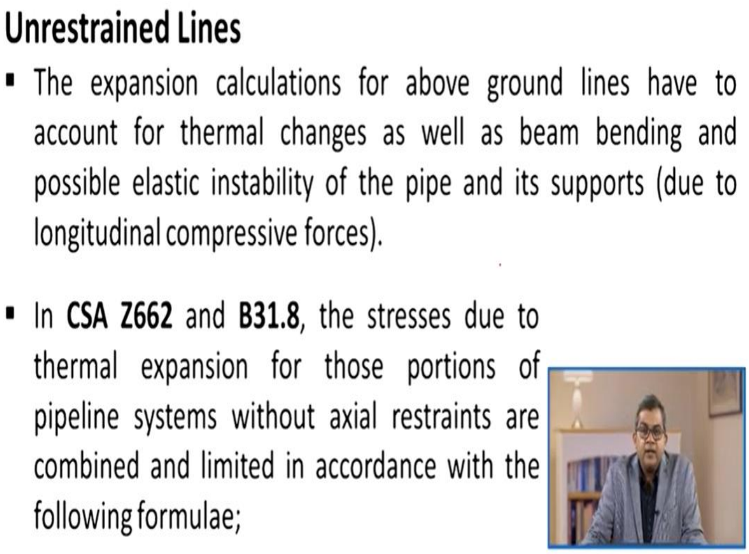
The value of S_B can be found by the following equation;

$$S_B = [(0.75 i_i M_i)^2 + (0.75 i_o M_o)^2 + M_t^2]/Z \qquad \textbf{Eq}^{\textbf{n}} \textbf{.....(15)}$$

Where, M_i, M_o, M_t are in plane, out of plane and torsional moments respectively, acting on the pipe and Z is the section modulus. i_i, i_o refer to the in and out of the plane stress intensification factors.



Now there are certain unrestrained lines now, the expansion calculations for above ground lines have to account for thermal changes as well as beam bending and the possible elastic in stability of the pipe and its support that due to the longitudinal compressive forces. Now in CSA and ASME, the stress due to thermal expansion for those portions of pipeline system without axial restraints are combined and limited in accordance with the with the formula.

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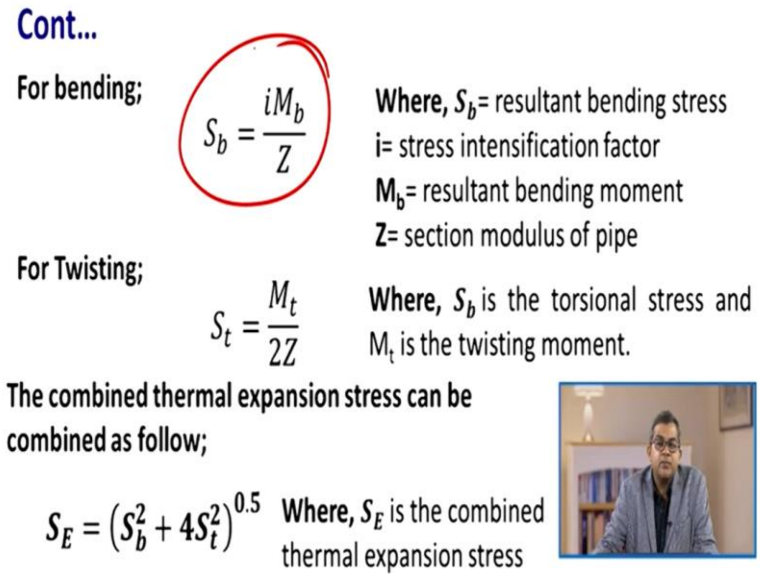


That is S b = i M b Z where, S b = resultant bending stress, i is a stress intensification factor, M b is the resultant bending moment, and Z is the section modulus of pipe. Now for twisting S t = M t over 2 Z where, S b is the torsional stress and M t is the twisting movement. The combined thermal expansion stress can be given like, S E = S b square + 4 S t square to the power 0.5 where, S E is the combined thermal expansion stress.

(Refer Slide Time: 27:04)

For bending;

$$S_b = \frac{iM_b}{Z}$$

Where, S_b= resultant bending stress

i= stress intensification factor

M_b= resultant bending moment

Z= section modulus of pipe

For Twisting;

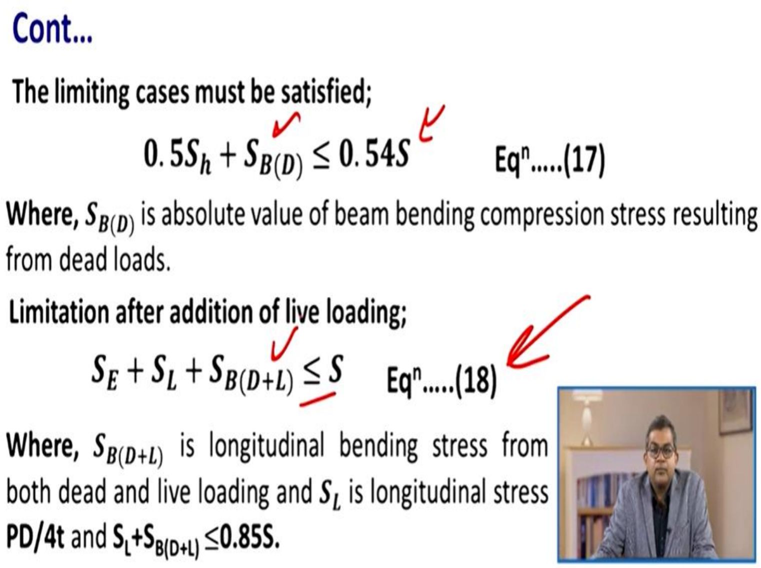
$$S_t = \frac{M_t}{2Z}$$

Where, S_b is the torsional stress and M_t is the twisting moment.

The combined thermal expansion stress can be combined as follow;

$$S_E = \left(S_b^2 + 4S_t^2\right)^{0.5}$$

Where, S_E is the combined thermal expansion stress



So, the limiting case must be satisfied like 0.5 S h + S B D is less than equal to 0.54 S and that is equation number 17. Where S B D this one is absolute value of beam bending compression stress resulting from dead loads. Now limitation after addition of live loading; that is S E + S L + S B D + L should be less than equal to S and that is equation number 18. Now where this S B D + L is a longitudinal bending stress from both dead and live loading and S L is the longitudinal stress P D over 4 t and S L + S B D + L is less than equal to 0.85 S.

(Refer Slide Time: 28:01)

The limiting cases must be satisfied;

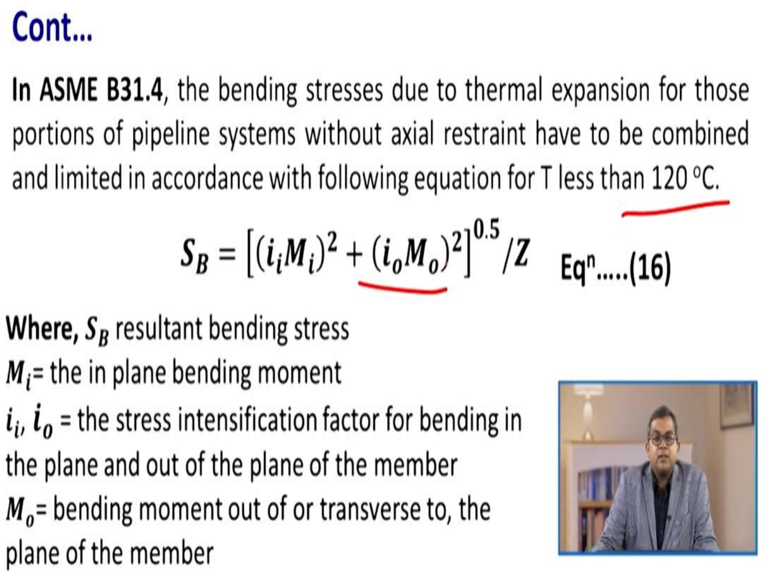
$$\mathbf{0.5S_h + S_{B(D)} \leq 0.54S} \qquad \textbf{Eq}^{\textbf{n}}\textbf{…..(17)}$$

Where, $S_{B(D)}$ is absolute value of beam bending compression stress resulting from dead loads.

Limitation after addition of live loading;

$$\mathbf{S_E + S_L + S_{B(D+L)} \leq S}$$

Where, $S_{B(D+L)}$ is longitudinal bending stress from both dead and live loading and S_L is longitudinal stress PD/4t and $S_L + S_{B(D+L)} \leq 0.85S$.



Now when we talk about the ASME B31.4, the bending stresses due to the thermal expansion of for those portion of pipeline systems without axial restraint have to be combined and limited in accordance with the equation for ah for T less than 120 degree Celsius. So, S B = i M i square + i o M o square to the power half over Z.

Where, S B is the resultant bending stress. M i is the in plane bending moment, i i and i o is the stress intensification factor for bending in the plane and out of the plane of the member, M o is the bending moment out of or transfers to the plane of the member.

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In ASME B31.4, the bending stresses due to thermal expansion for those portions of pipeline systems without axial restraint have to be combined and limited in accordance with following equation for T less than 120 ^{o}C.

$$S_B = [(i_i M_i)^2 + (i_o M_o)^2]^{0.5}/Z$$

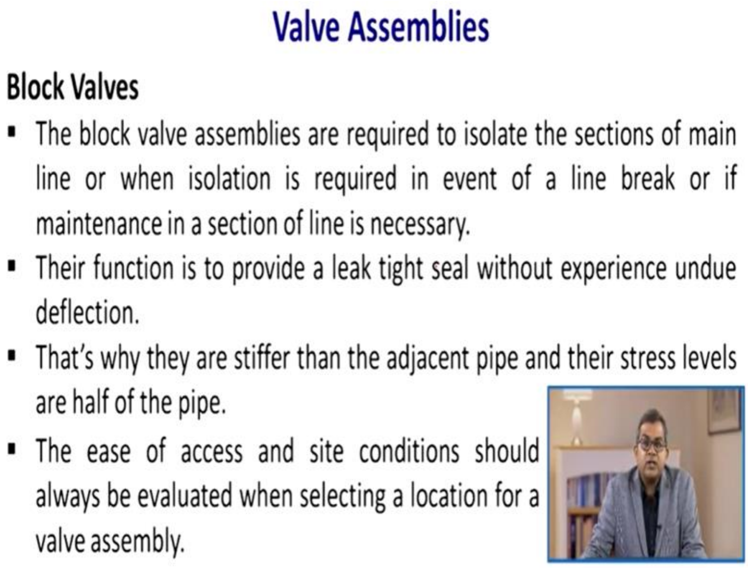
Where,

S_B resultant bending stress

M_i= the in plane bending moment

i_i, i_o = the stress intensification factor for bending in the plane and out of the plane of the member

M_o= bending moment out of or transverse to, the plane of the member



Now let us talk about the valve assemblies, now valve assemblies let us first take up the block valves. The block valve assemblies are required to isolate the sections of main line or when isolation is required in event of a line break or if maintenance in a section of line is necessary. Their function is to provide a leak tight seal without experience undue deflection, and that is why they are stiffer than the adjacent pipe and their stress levels are half of the pipe.

The ease of access and the site conditions should always be evaluated when selecting a location for a valve assembly. There may be certain required components and if required a gate or a ball valve or same size as to mainline to allow the passage of pigs.

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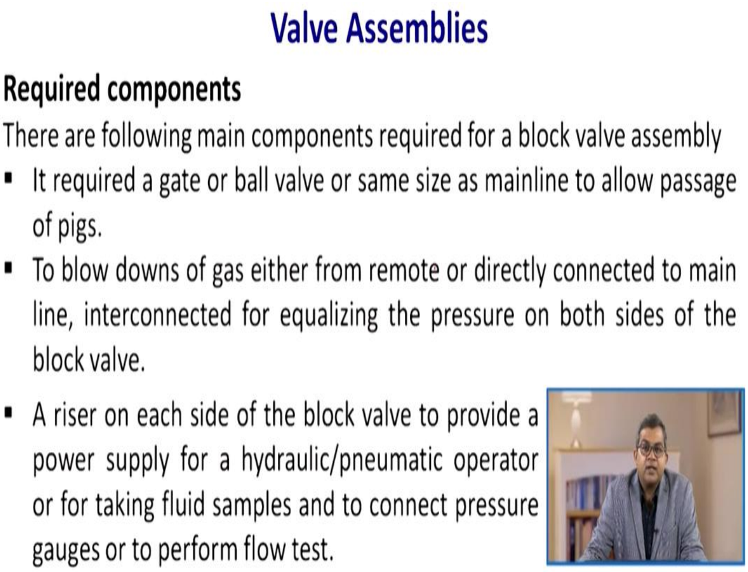

Now to blow down of gas either from remote or directly connected to main line interconnected for equalizing the pressure on both side of the block valve. Usually, a riser on each side of the block valve is to there to provide a power supply for hydraulic or pneumatic operator or for taking fluid samples and to connect the pressure gauges or to perform the flow test.

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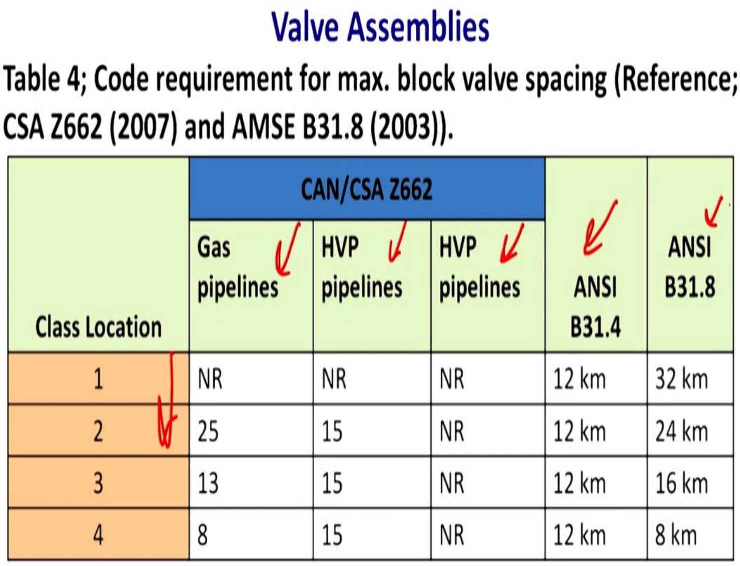
Valve Assemblies

Table 4; Code requirement for max. block valve spacing (Reference; CSA Z662 (2007) and AMSE B31.8 (2003)).

Class Location	CAN/CSA Z662			ANSI B31.4	ANSI B31.8
	Gas pipelines	HVP pipelines	HVP pipelines		
1	NR	NR	NR	12 km	32 km
2	25	15	NR	12 km	24 km
3	13	15	NR	12 km	16 km
4	8	15	NR	12 km	8 km

Now here you see that the various code requirement for the maximum block valve is spacing again the reference is CSA Z662 2007 and ASME B31.8 2003. Now here you see that different type of a class locations listed and the gas pipeline HVP pipeline for this and this ANSI code B31.4 and ANSI B31.8, now here see that different classes are enlisted with the different distances.

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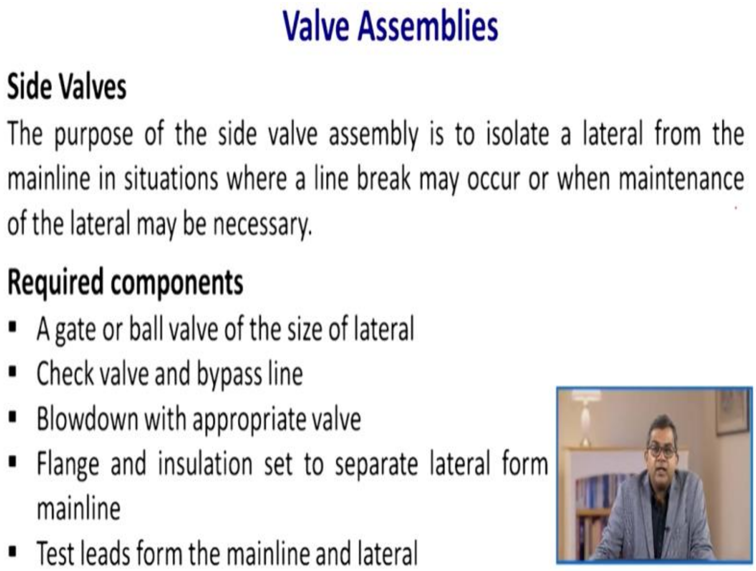

Now let us talk about the side valves, the purpose of side valve assembly is to isolate a lateral from the main line in situation where a line break may occur or when maintenance of a lateral main necessary. Now there are various component required for this one is a gate or a ball valve of the site of lateral, check valve and bypass line, blow down with appropriate valve.

Flanges and insulation set to separate the lateral form mainline, and test leads from the main line and lateral. Now the check valve in the assembly is to prevent reverse flow and also to prevent the flow from mainline into the inflowing lateral, when the pressure in lateral is less than that in the inline.

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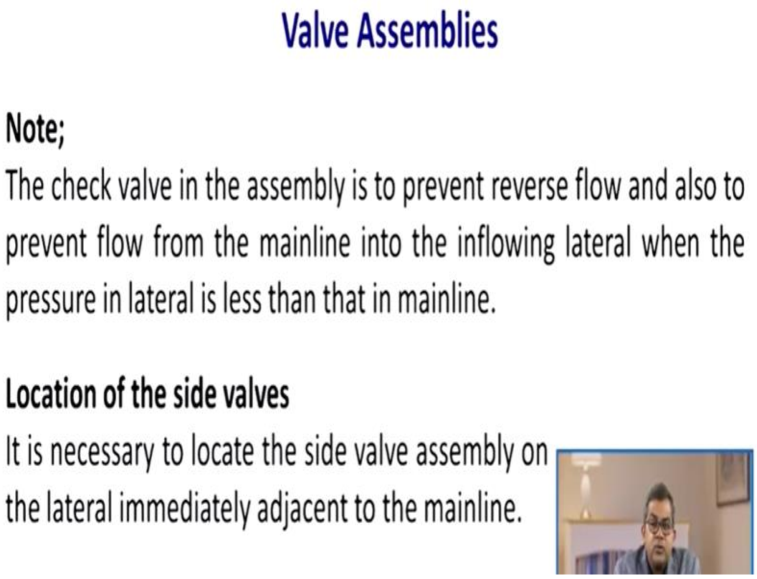

The location of side valves it is necessary to locate the side valve assembly on the lateral immediately adjacent to the mainline.

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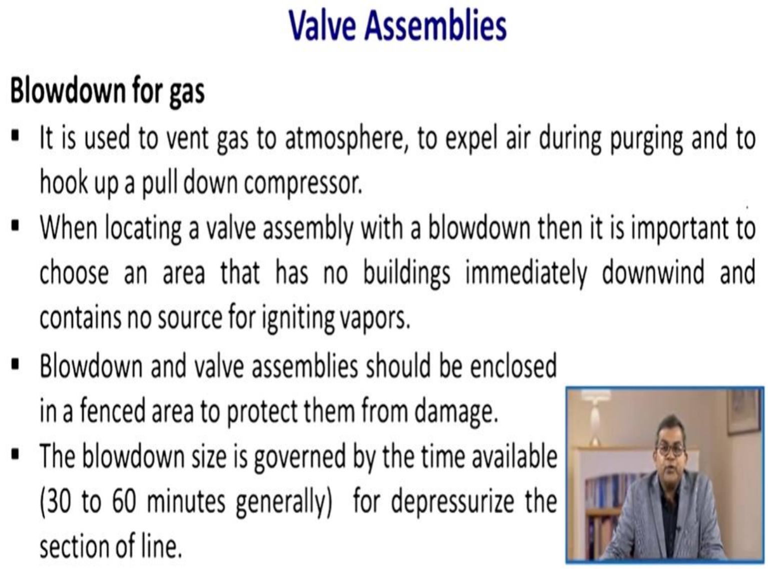

Now let us talk about the blowdown for gases now, it is used to vent gas to atmosphere to expel air during purging and to hook up a pull-down compressor. Now when locating a valve assembly with a blow down then it is important to choose an area that has no building immediately downwind and contains no source for igniting vapours. The blow down and wall assembly it should be enclosed in fenced area to protect them from any kind of damage.

The blow down size is governed by the time available that is 30 to 60 minutes generally for depressurize the section of line.

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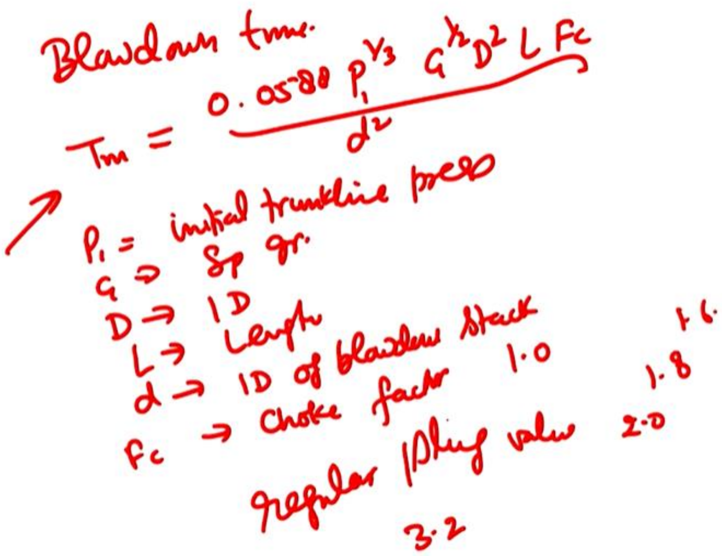

Now sometimes you need to know the formula for calculating the blowdown time so blowdown time that is referred as T m = 0.0588 P to the power 1 by 3 P 1 G to the power half D square L

F c upon D square. Where T m is the blow down time in minutes, P 1 is the initial trunk line pressure; G is specific gravity of the gas. D is the internal diameter of trunk line; L is the length of trunk line being blow down. d is the internal diameter of blow down stack, F c is the choke factor.

For ideal nozzle it is 1.0, for full port gate valve 1.6, reduced port gate valve it is 1.8, and regular plug valve it is 2.0, and venturi plug valve it is 3.2.

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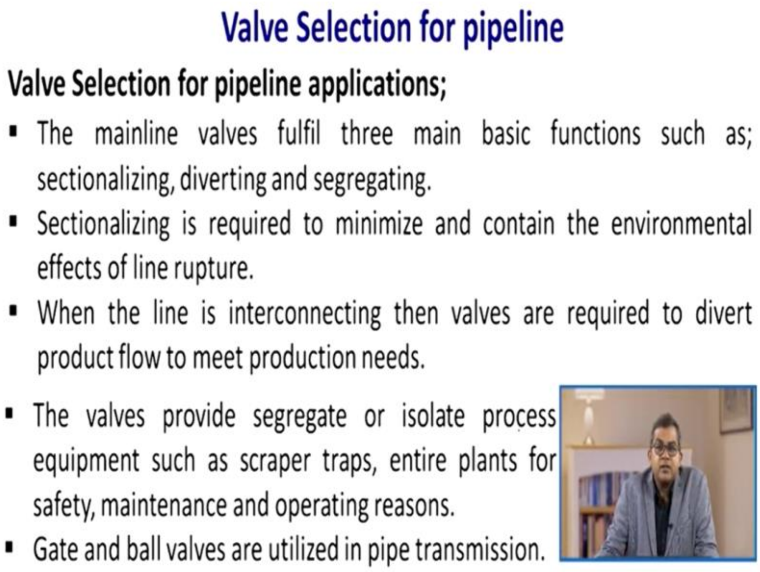
Formula for calculating blowdown time;

$$T_m = \frac{0.0588 P_1^{\frac{1}{3}} G^{\frac{1}{2}} D^2 L F_C}{d^2}$$

Where, Tm is blowdown time (minute), P_1; Initial trunkline pressure (Psi), G; specific gravity of gas, D; ID of trunkline (inches), L; length of trunkline being blown down (miles), d; ID of blowdown stack (inches):

F_c is choke factor:

For ideal nozzle= 1.0, full port gate valve= 1.6; reduced port gate valve= 1.8; regular plug valve= 2.0; venture plug valve=3.2;



Now valve selection for the pipeline application; the mainline valves fulfil 3 main basic function such as sectionalizing, diverting and segregating. Now sectionalizing is required to minimize and contain the environmental effect line rupture. Now when line is interconnecting then valves are required to divert the product flow to meet production line. The valves provide segregate or isolate process equipment such as scrapper traps, entire plant for safety maintenance and operating reasons. Now the gate and ball walls are utilized in the pipe transmission.

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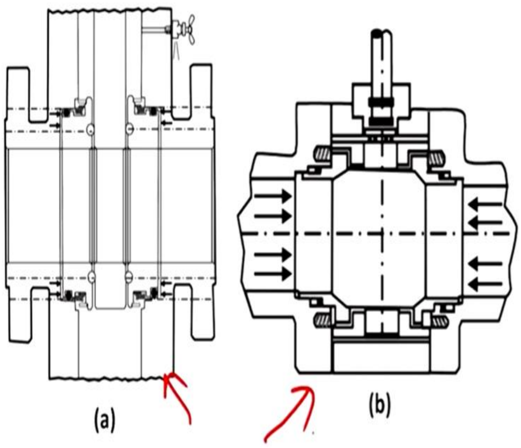
Valve Selection for pipeline



Figure; (a) Gate valve (b) ball valve

Here, you see the gate valve and ball valve which are used in various kind of a pipeline operation.

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Valve Selection for pipeline

The variables that must be considered for each valve installation are as follows;

- Operating characteristics
- Function and location
- Fluid service
- Materials options
- Space available
- Maintenance and repair capability
- Delivery schedule
- Costs etc.

Now there are certain variable that must be considered for each valve installation, and these are what, are the different operating characteristics one must know. what, is the function required and where they need to be located, then fluid services which kind of the fluid services need to be done, then materials option, space available, maintenance and repair capabilities, delivery schedule, cost etcetera.

So, all these factors are available must be considered while considering all these things. So, at last in this particular segment, we discussed a lot of about the gas transmission especially natural gas transmission.

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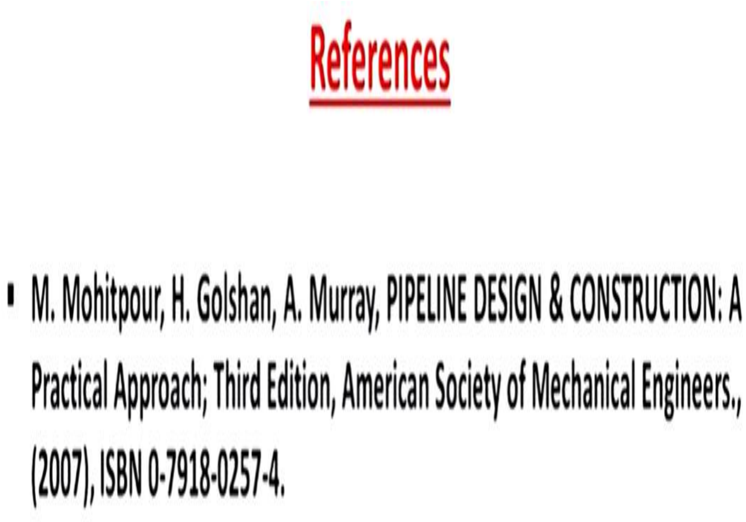

And for your convenience we have listed couple of references, if you wish to have a further reading you can seek the help of all these references thank you very much.