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FAILURE AND FACTORS OF SAFETY IN PIPING SYSTEM DESIGN (U)

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FAILURE THEORIES AND FACTORS OF SAFETY IN PIPING SYSTEM DESIGN

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INTRODUCTION

An important body of test and performance data on the behavior of piping systems has led to an ongoing reassessment of the code stress allowables and their safety margin [1,2].

The codes stress allowables, and their factors of safety, are developed from limits on the incipient yield (for ductile materials), or incipient rupture (for brittle materials), of a test specimen loaded in simple tension. In this paper, we examine the failure theories introduced in the B31 and ASME III codes for piping and their inherent approximations compared to textbook failure theories. We summarize the evolution of factors of safety in ASME and B31 and point out that, for piping systems, it is appropriate to reconsider the concept and definition of factors of safety.

FAILURE THEORIES FOR DUCTILE MATERIALS

The common failure theories for ductile materials are:

The maximum stress theory:

$$S_{max\ tensile} = S_y/(FS_y)$$

The maximum shear stress theory (Tresca):

$$t_{max} = S_y/2(FS_y)$$

The constant energy distortion theory (Von Mises):

$$(s_1 - s_2)^2 + (s_2 - s_3)^2 + (s_3 - s_1)^2 = 2S_y^2/(FS_y)^2$$

While the Von Mises theory correlates best with experimental failure predictions, the Tresca or the maximum stress theories are used in the B31 and ASME Boiler and Pressure Vessel Code [6], due to their simplicity, as listed in Table 1.

TABLE 1 - Failure Theories for Various Codes

Code (Section)	Limits	Failure Theory
B31.1 (102.3.3(A), 104.8.1)	limit on sum of longitudinal stress	maximum stress theory
B31.3 (302.3.5)	limit on sum of longitudinal stress	maximum stress theory
III NB-3200 (NB-3211 (a))	limit on stress intensities	maximum (Tresca) shear stress theory
III NX-3600 (NB-3611.1, NC/ND-3650)	"stress will not exceed limits described"	not explicitly defined
VIII Div. 1 (UG - 23(a))	limit on maximum tensile stress	maximum stress theory
VIII Div. 2 (AD-140)	limit or stress intensities	maximum (Tresca) shear stress theory

General State of Stress in a Pipe Cross Section

We will now investigate which failure theory is espoused in NB/NC/ND-3600. Consider a piping segment subject to internal pressure (P), a bending moment (M_b), a normal axial force (N), a torsional moment (T), and a shear load (V). To simplify the formulation of stress equations, we limit ourselves to primary stresses and exclude the thermal gradient effects. Under these general loads, the state of stress in a cross section of the piping segment is illustrated in Figure 1, and consists of:

At 0° and 180° of vertical	$s_n = PD/4t + M_b r/I + N/A$
At all points of cross section	$s_t = PD/2t$
At all points of cross section	$s_r = P/2$
At all points of circumference r	$t = Tr/2J + V/A$

In this formulation, the shear load stress is considered to be constant and equal to V/A. The actual shear load stress varies from zero at 0° to 180° to a maximum of 2V/A at ± 90°. The constant shear load approximation simplifies the formulation of stress equations and overestimates, somewhat, the stress at 0° and 180° where the bending moment stress $M_b r/I$ is maximum.

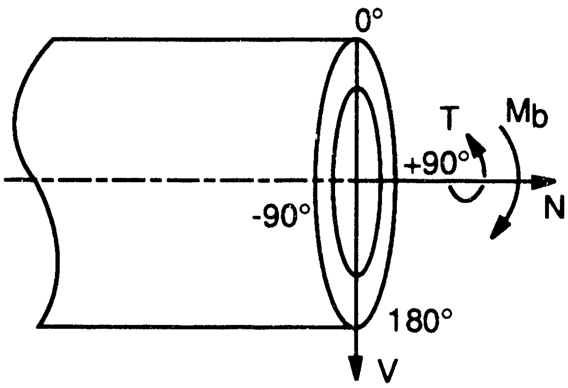

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Figure 1 - General Loading and Stress in a Cross Section of Pipe

In the above equations, the shear stress due to torsion and the stresses due to pressure are considered to be constant through the pipe cross section, as is the case for thin wall vessels. The bending moment stress is maximum at the outer fiber where it is equal to $M_b R_o/I = M_b/Z$. The maximum shear stress is obtained from the Mohr circle of Figure 2:

$$t_{max} = \frac{1}{2}\left[\frac{1}{2}(s_n + s_t) + \left(\frac{1}{4}(s_n - s_t)^2 + t^2\right)^{1/2} s_r\right]$$

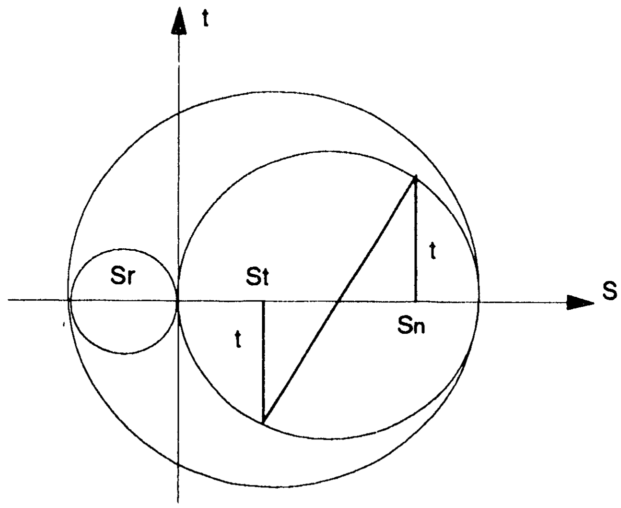

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Figure 2 - Mohr Circle for the General Stress State in a Pipe Cross Section

By substitution, the general form of twice the maximum shear stress (or stress intensity as defined in Sections III and VIII Division 2) is:

$$2t_{max} = \frac{1}{2}(M_b/Z + N/A + 3\,PD/4t) + \left(\frac{1}{4}(M_b/Z + N/A - PD/4t)^2 + (TD/2J + V/A)^2\right)^{1/2} + P/2$$

while the maximum axial stress is:

$$s_n = PD/4t + M_b/Z + N/A$$

Therefore, the stress computed in B31 and NB/NC/ND-3600 is not the maximum shear stress (or the stress intensity), but the maximum axial stress where N/A is ignored.

Stress in the Case of Large Bending Moments

In order to evolve towards a form of stress more consistent with NB/NC/ND-3600, it is necessary to limit ourselves to the particular case where the stresses due to the normal axial load (N/A), the torsional moment (TD/2J) and the shear load (V/A) can be neglected in comparison to the stresses due to bending (Mb/Z). We refer to this case as the large bending moment approximation. Furthermore, if D/t >>1 (the thin wall approximation), the stress intensity and the maximum stress become:

$$2t_{max} = PD/4t + M_b/Z$$
$$S_{max} = s_n = PD/4t + M_b/Z$$

Therefore, for piping systems where the thin wall and the large bending moment approximations apply, the maximum shear failure theory (limit on stress intensity) and the maximum stress failure theory (limit on longitudinal stress) are identical:

$$S_{max} = 2t_{max} = PD/4t + M_b/Z$$

Prior to investigating the failure theories, let us consider the two approximations introduced in the development of the above equations: the large bending moment approximation, and the thin wall approximation.

Large Bending Moment Approximation

To assess the relative effect of the shear load compared to the bending moment, consider a straight span of pipe, as shown in Figure 3.

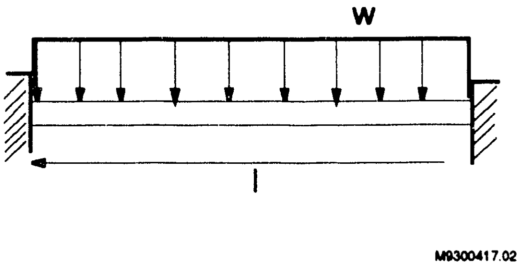

Figure 3 - Simplified Beam Approximation

Consider pressure, torsion, and normal forces to be nonexistent. The stress intensity in any cross section reduces to:

$$2t_{max} = M_b/Z + \left((M_b/Z)^2 + 4(V/A)^2\right)^{1/2}$$

The large bending moment approximation (stress due to V neglected relative to stress due to M_b) is valid if:

$$(2V/A)^2 << (M_b/Z)^2$$

with the thin wall approximation:

$$A = 3.14\ Dt \text{ and } Z = .8D^2t$$

and for a distributed load w, at the built-in ends (Figure 3):

$$M_{bmax} = wl^2/12 \text{ and } V_{max} = wl/2$$

The condition for large bending moment approximation becomes:

$$l/D >> 3$$

This means that, in the case of a simple beam, the large bending moment approximation is valid if the span length is much larger than the pipe diameter, which is practically always the case. Therefore, it would appear reasonable for the weight of a simple span pipe to neglect load components other than the bending moment; hence, the code approximation in ASME B31 and NB/NC/ND-3600: $s_n = M_b/Z$.

However, in actual piping layouts, with bends and eccentricities, it is not evident that the large bending moment approximation of ASME B31 and NB/NC/ND-3600 applies. To illustrate this point, an actual piping system was analyzed and the stress intensity $2t_{max}$ (which includes all components of load) was compared to the unintensified code stress S_{max}= PD/4t+ M/Z. The ratio $2t_{max}/S_{max}$ is shown in Figure 4 for all points in the piping system. It appears from Figure 4 that the large bending moment approximation (i.e., neglecting axial and shear loads and torsional moments relative to bending moment effects) is not accurate for the piping system analyzed: The unintensified code stress S_{max} is, in this case, clearly smaller than the stress intensity $2t_{max}$. In other words, the ASME B31 and NB/NC/ND-3600 primary pipe stress equations correspond to the maximum stress failure theory (S_{max}) and not to the maximum shear stress theory ($2t_{max}$).

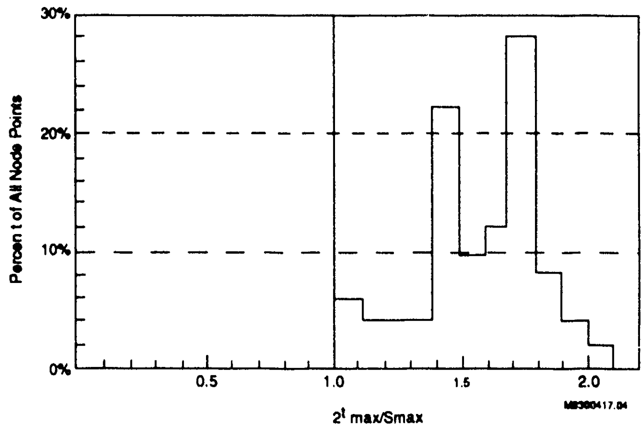

Figure 4 - Actual Stress Intensity $2t_{max}$ (contains all load terms) Compared to the Unintensified Code Stress S_{max} (PD/4t+M/Z) in an Actual Piping System Subject to Its Own Weight.

Thin Wall Approximation

The thin wall approximation is used to allow the shear stress due to torsion and the stresses due to pressure to be considered constant through a cross section. It is also used to simplify the expression of the shear and axial stresses due to pressure, as shown in Table 2.

TABLE 2 - Thin Wall Approximation

Stress Component	Actual pressure induced stress	Thin wall approximation
normal axial	$P[R^2_i/(R^2_o-R^2_i)]$	PD/4t
circumferential hoop	$P[(R^2_o + R^2_i)/(R^2_o - R^2_i)]$	PD/2t

The inaccuracy introduced by such approximations on the pressure stress can be quantified for certain common pipe sizes, as presented in Table 3. It can be seen from Table 3 that the thin wall approximation used in the codes overpredicts the actual pressure induced stress.

TABLE 3 - Inaccuracy of Thin Wall Approximation for Pressure Stress

		Inaccuracy (1)	
Pipe Schedule	**Nominal Size**	**Normal Axial Stress**	**Circumferential Hoop Stress**
160	1	112%	17%
160	4	51%	11%
160	8	43%	10%
160	12	42%	10%
40	1	41%	10%
40	4	19%	5%
40	8	12%	4%
40	12	10%	3%

NOTE: (1) Inaccuracy is defined as (approximate-actual)/actual pressure stress.

Factor of Safety for Normal Operating Loads

As illustrated in Table 4, the factors of safety (FS) for normal operating loads (typically static) have varied over the years and in today's codes, FS depends on the code section and safety class [3,4]. In addition to the limits and factors of safety associated with yield stress (FS_y), the code has also introduced limits and factors of safety associated with ultimate stress (FS_u) to prevent failures for less ductile materials.

$$S \text{ or } S_m = \min (S_y/(FS_y) \text{ or } S_u/(FS_u)$$

TABLE 4 - Evolution of Code Factors of Safety

1909	Massachusetts Rules [3, p. 8]	FSu = 5 to 6 (new boilers) FSu = 4 (existing boilers)
1934	API-ASME Code for Unfired Pressure Vessel [3, p. 97]	$FS_u = 4$ (all vessels)
1943	ASME VIII [3, p. 137; 4, p. 164]	$FS_u = 4$ (all vessels)
1955	B31.1	$FS_y = 3/2$* $FS_u = 4$
1955	B31.3	$FS_y = 3/2$* $FS_u = 3$
1963	ASME III	$FS_y = 3/2$* $FS_u = 3$ (class 1) $FS_y = 3/2$* $FS_u = 4$ (Class 2 and 3)
1968	ASME VIII Division 2	$FS_y = 3/2$* $FS_u = 3$

Note * $FS_y = 3/2$ for carbon steel and 90% for stainless steel, at temperature

The introduction of a limit on the ultimate strength remains valid for the theory of maximum stress (B31 and ASME VIII Div. 1). This also applies to brittle materials, but it can be questioned when applied to the maximum shear stress (ASME III and ASME VIII Div. 2), which is a failure theory for ductile metals. This point is significant since, as illustrated in Table 5, the limit on ultimate strength controls the code allowable in various cases for common carbon and stainless steel pipe materials.

		S_m (ASME III cl. 1, ASME VIII D IV 2, B31.3)				S (ASME III cl. 2 & 3, ASME VIII Div. 1, B31.1)			
		$S_u/3$	$S_{uT}/3$	$2S_y/3$	$2S_{yT}3$*	$S_u/4$	$S_{uT}/4$	$2S_y/3$	$2S_{yt}/3$*
Carton Steel (CS)	SA. 106 Gr. A Seamless	16 (100°F)			14.8 (600°F)	12 (100°F) (600°F)			
	SA. 106 Gr. B Seamless	20 (100°F)			17.3 (600°F)	15 (100°F) (600°F)			
Stainless Steel (SS)	SA376 TP 304 Seamless			20 (100°F)	16.4 (600°F)	17.5 (100°F)	14.8 (600°F)		
	SA376 TP 316 Seamless			20 (100°F)	17 (600°F)	18.8 (600°F)			17 (600°F)

TABLE 5
Code Allowables (ksi) at Ambiant and 600°F Based on S_u or S_y

Note: *$2S_y/3$ for carbon steel and 90% S_y for stainless, at temperature

The Redefinition of Factors of Safety

As introduced in the codes for S and S_m, the codes definition of factors of safety is a textbook application of limits against incipient yielding (FS_y) or incipient rupture (FS_u) of test specimens loaded in simple tension. There are, of course, essential differences between the mode of failure in a simple tension test and the mode of failure of a piping system.

Several papers have investigated various aspects of the failure of piping systems: the load redistribution, the geometrical and materials non-linearities, and the dynamic effects of materials and loadings. Because of the differences between tension tests (stress failure theories) and the failure of actual piping systems, it may be appropriate to consider in the piping codes a new definition of factor of safety other than $S_y/(FS_y)$ or $S_u/(FS_u)$[2].

It can be argued that this change in factor of safety has indirectly been introduced into the code. Over time, the code has increased the basic allowables by multipliers based on service limit and type of stress, as summarized in Table 6 [5]. However, these coefficients do not fully account for the capability of a piping system for load redistribution, its geometrical and materials non-linearities, and the dynamic effects of materials and applied loadings.

TABLE 6 - Code Allowable Multipliers

Code Section	Stress Allowable	
NB-3200	P_mS_m	S_m
	$P_m + P_b$	$1.5S_m$(1)
	$P_m + P_b + Q$	$3.0S_m$
NB-3600 and NC/ND-3600 post-1980	Normal (A)	S_m or $_h$ = .7 S_y (2, 3)
	Upset (B)	min(1.8S_m or $_h$, 1.5S_y) = 1.2 S_y
	Emergency (C)	min(2.25 S_m or $_h$, 1.8S_y) = 1.5S_y
	Falted (D) in	(3 S_m or $_h$, 2 S_y) = 2S_y
NC/ND-3600 pre-1980 and B31.1 nuclear practice	Normal	S_n = .7S_y
	Upset	1.2S_n = .8S_y
	Emergency	1.8S_n = 1.2S_y
	Faulted	2.4S_n = 1.6S_y

(1) 1.5 S_m is based on the shape factor for a rectangular cross section, appropriate for a through wall stress in a vessel wall. The shape factor for a pipe cross section varies from 1.27 (thin wall) to 1.70 (solid bar).

(2) Reference 5 provides a comprehensive study of the service limit coefficients.

(3) For simplicity, we consider $S_m = S = 2S_y/3$.

It is, therefore, suggested that based on the experimental evidence, the multipliers be revisited or the factors of safety (FS_y and FS_u, and therefore S and S_m) be redefined for piping systems differently than for vessels.

CONCLUSIONS

The code stress allowables are a textbook application of factors of safety against incipient yielding or rupture of test specimens loaded in simple tension.

Unlike the stress intensity and corresponding shear stress theory adopted for vessels in ASME III and ASME VIII Division 2, the pipe stress equations of ASME B31 and ASME III are simplified expressions of the maximum axial stress. The pipe stress allowables are accordingly based on the maximum stress theory. The maximum stress theory of failure for piping is identical to the shear stress theory for vessels only in cases where the large bending moment approximation applies. This is indeed the case for a beam under uniform loading, but is not necessarily the case for actual piping configurations subjected to distributed loads (such as weight, winds, or the inertial effects of earthquakes).

The codes' straightforward application of failure theories with simple tension allowables does not yet fully account for non-linear and dynamic aspects of the behavior of piping systems.

Nomenclature

Loads:

w	=	distributed load
P	=	internal pressure
M_b	=	bending moment
N	=	normal axial force
T	=	torsional moment
V	=	shear load

Stresses:

S	=	code allowable stress
S_{max}	=	maximum code stress
S_m	=	code allowable stress intensity
s_n	=	axial stress (normal to cross section of pipe)
s_t	=	tangential stress (hoop stress)
s_r	=	radial stress
t	=	shear stress
s_i	=	principal stress (i = 1, 2, 3)
S_y	=	yield stress of pipe material at ambiant temperature
S_u	=	ultimate stress of pipe material at ambiant temperature
S_{yT}	=	yield at temperature T
S_{uT}	=	ultimate stress at temperature T

Pipe properties:

D	=	outside diameter
t	=	nominal wall thickness
r	=	radial distance of a point
A	=	cross sectional area
J	=	polar moment of inertia
I	=	moment of inertia
Z	=	section modulus
Ro	=	outside radious

Factors of Safety:

FS_y	=	factor of safety relative to S_y
FS_u	=	factor of safety relative to S_u

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