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MOP, MAOP, DP AND MAWP – UNDERSTANDING THE DIFFERENCES TO AVOID UNNECESSARY COSTS

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ABSTRACT

The ASME pressure vessel and piping codes and standards provide excellent references for code writers in international jurisdictions when developing their own national codes and for safety authorities when developing regulatory acts. The inclination to customize this effort may add unnecessary complexity that unintentionally obscures the underlying engineering principles.

In developing the Canadian pipeline code, the authors use the notion of maximum operating pressure or MOP similar to the MOP found in the ASME codes for pipelines. While the ASME code definitions are explicit and articulate, the MOP defined in the Canadian code is less so and has led to inadvertent confusion by industry users. Misunderstanding of complementary terminology used in ancillary ASME standards has contributed to further complexities. The use of the term, maximum allowable operating pressure or, MAOP in the ASME pipeline codes has further reduced clarity when integrating this term into international codes and regulatory acts.

This paper examines, in detail, some aspects of the Canadian pipeline code and illustrates via a representative case study some of the aforementioned difficulties that have arisen. These difficulties resulted in unnecessary derating of assets by imposing operational limits that were well below actual capacity. A clear explanation of the engineering principles underlying the provisions for codes which use a "design by rules" philosophy will help operators set appropriate limits for both static and dynamic loads that may not be apparent in the specific codes considered and will be expository for regulators and code users in general.

INTRODUCTION

Pressure vessel and piping codes have provided protection for the public and environment with respect to catastrophic failures for nearly a century. By the 1880's, exploding boilers in the United States of America, had caused 50,000 deaths and 2 million people were being injured annually in a national population of 50 million. These dreadful statistics prompted development of a boiler test code in 1884 and subsequently, the ASME boiler and pressure vessel construction code in 1915. Piping code development was initiated in 1926 and the first piping code was published in 1935. This single code was later specialized along industry lines with ASME B31.8 Gas Transmission and Distribution Piping Systems published in 1955 and ASME B31.4 Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids in 1959. The success of these Codes is well recognized.

In Canada, until 1967, the two referenced ASME piping codes (collectively, the "Code") were used explicitly since the first editions of separate Canadian oil and gas pipeline standards referenced use of the ASME Codes without modification. Since 1994, the Canadian standards have been combined into a single document entitled Canadian Standards Association CSA Z662, Oil and Gas Pipeline Systems (the "Standard") [1]. As with its predecessor codes, CSA Z662 advises that it is a consensus document, providing requirements considered to be adequate under conditions normally encountered in the oil and natural gas pipeline industry but not prescribing requirements for abnormal or unusual conditions. Individual pipeline owners and contractors commonly have their own engineering standards that reference CSA Z662 as the base case, and then specify additional requirements that must be met considering the specifics of their particular situation, experience and preferences. The Standard appeals to good engineering practice in a number of instances and similar to the ASME Codes, it also declares that it is not a design handbook and competent engineering judgment should be employed with its use.

In developing a national standard, modifications were introduced to differentiate the Standard from its predecessors. However, in doing so, these modifications have led to reduced clarity in the overall objective of simplified treatment, adequacy and expressed safety.

DESIGN BASIS

The specific design requirements for the Codes and the Standard rely upon a simplified engineering approach and are often described as “rules based”. Although these rules are grounded in engineering principles (i.e., the strength of materials approach) implementation in these Codes and Standard, nominally, does not require detailed analysis and deep understanding of stress categorization and behavior of the materials of construction under various applied loadings. The Codes and Standard do not define nor require the designer to be an engineer. However, in Canada, the practice of engineering is restricted to those licensed to practice by regulatory jurisdictions.

The primary consideration of the rules based approach is whether there is adequate wall thickness in the pipe and piping system components to contain system pressure. Loadings from other sources are then evaluated to ensure that the pipe or component wall thickness remains adequate under application of these other loads. The evaluation of other applied loads is implemented by calculating stresses caused by these loads using prescribed methodology and then comparing them to allowable stress criteria. The focus in this paper is on the first step, determination of pipe wall thickness using the applied system pressure and the specified minimum yield strength (SMYS) of the selected material of construction. A review of the terminology and methodology of ASME B31.4 and ASME B31.8, the two predecessor Codes is given and focused on the various descriptions of pressure and pressure wall thickness. US federal government requirements are also introduced in order to understand their influences.

ASME B31.4 Methodology

ASME B31.4, the pipeline Code for liquid hydrocarbons, identifies and defines three (3) pressure designations in order to calculate a minimum required wall thickness that is adequate for the internal pressure [2]:

- Internal design gage pressure
- Maximum steady state operating pressure
- Transient overpressure

The internal design gage pressure, P_i shall not be less than the maximum steady state operating pressure at any point in the piping system. It is used in order to calculate the required pressure design nominal wall thickness of the pipe, t considering the pipe outside diameter, D. Pipe material specified minimum yield strength, Sy, weld joint factor, E and a design factor, F are used to determine the applicable allowable stress.

$$S = F \cdot E \cdot Sy \qquad (1)$$

The maximum value of F is based on nominal pipe wall thickness and considers potential underthickness tolerance and maximum permitted depth of imperfections that might be present in the pipe and still meet its materials specification requirements. The maximum value of F is prescribed as 0.72. The weld joint factor, E, is intended to account for the quality of the pipe longitudinal seam weld; E may be taken as 1.0 for seamless and qualified welded pipe. Therefore,

$$t = P_i \cdot D / (2 \cdot S) \qquad (2)$$

Allowance for pressure rise above maximum steady state operating pressure, (i.e., transient overpressure) is provided by limiting this variation to not exceed internal design pressure by more than 10%. Since a 10% overpressure is permitted, it is important not to use the transient overpressure value as P_i in calculating t since this could result in the pipe being 10% thicker than is otherwise necessary. Since pipelines are often many miles long, such overdesign would result in a large and unnecessary cost increase.

In the U.S.A., pipelines may be within the jurisdiction of the Pipeline and Hazardous Materials Safety Administration of the Department of Transportation and fall under the design requirements of the Code of Federal Regulations, Title 49 Transportation, Part 195 Transportation of Hazardous Liquids by Pipeline (CFR 49 Part 195). This federal code prescribes a design requirement whereby the internal design pressure for the pipe in a pipeline is to be calculated as [3],

$$P = (2 \cdot S \cdot t / D) \cdot E \cdot F \qquad (3)$$

However, S in this expression is the SMYS value. CFR 49 Part 195 also identifies specific cases where the value of F must be less than 0.72 (e.g., pipe on offshore platforms or on platforms located in navigable waters).

This legislation defines the term maximum operating pressure, MOP as the maximum pressure at which a pipeline or segment of a pipeline may be normally operated. Other design requirements are not addressed but must be provided for in the design of the pipeline system. Presumably, the designer would choose to apply ASME B31.4 if not otherwise locally legislated. Steel flanges may be used in compliance with the pressure-temperature ratings of ASME B16.5 and ASME B16.47 [4] [5].

ASME B31.8 Methodology

ASME B31.8, the pipeline Code for gas transmission and distribution systems, identifies and defines five (5) pressure designations in order to calculate a design pressure rather than a pressure design wall thickness as is done in ASME B31.4 [6]:

- Design pressure
- Internal design pressure
- Maximum operating pressure (MOP)
- Maximum actual operating pressure
- Maximum allowable operating pressure (MAOP)

The first two terms are stated to be equivalent per the Code and are defined as the maximum pressure permitted by the Code. The next two terms are also considered as equivalent and defined as the highest pressure at which a piping system is operated during a normal operating cycle. The final term, maximum allowable operating pressure, is defined to be *the maximum pressure at which a pipeline system may be operated in accordance with Code* provisions.

The Code determines the design pressure for steel gas transmission and distribution piping systems as

$$P = (2 \cdot S \cdot t / D) \cdot F \cdot E \cdot T \qquad (4)$$

where S is the SMYS value, T is a temperature derating factor and F is the basic design factor.

The value for T is a function of the design temperature of the pipeline and varies from a maximum of 1.000 for design temperatures up to 250 °F [121 °C] and down to 0.867 for a design temperature of 450 °F [232 °C]. This factor accounts for the reduction in the strength of steel as temperature increases. This derating factor is not used in ASME B31.4 because the maximum permitted design temperature allowed by that Code is 250 °F [121 °C].

The basic design factor F, is a function of the physical location of the pipeline considering the potential consequences of a pipeline failure. It has a maximum value of 0.80 and a minimum value of 0.40, with the value of F decreasing as the potential consequences of failure increases. As can be seen in the above equation, the permitted value of P is directly proportional to F, and thus decreases as F decreases.

The minimum required nominal pipe thickness for a specified design temperature can be determined by rearranging the terms of equation (4). This rearrangement is more commonly used in design (i.e., determine the required nominal pipe thickness for a specified design pressure).

The MAOP is established as the least of four quantities:

- Design pressure of the weakest element comprising the pipeline
- Test pressure divided by a test factor ranging from 1.10 to 1.40 (steel pipelines)
- Maximum safe pressure to which a pipeline should be subjected
- Consideration of connected lines

The test pressure is limited to a pressure not higher than the pressure to produce a hoop stress equal to the yield strength as determined by testing, (i.e., actual material yield strength). Hence, this value could exceed the SMYS value of the material of construction.

There is no further elaboration of transient conditions in ASME B31.8 and any transients are captured by the definition of MOP already given (i.e., it is the maximum operating pressure experienced during a normal operating cycle). Flanges shall conform to ASME B16.5 or MSS SP-44 for steel materials and have adequate ratings to qualify the pipeline to the intended MAOP. Large diameter steel flanges to ASME B16.47 are referenced in mandatory Appendix A of the Code.

The Code of Federal Regulations, Title 49 Transportation, Part 192 Transportation of Natural and Other Gas by Pipeline: Minimum Federal Standards (CFR 49 Part 192) uses the MAOP and design pressure concepts. However, it defines the maximum actual operating pressure as the maximum pressure that occurs during normal operations over a period of 1 year [7].

As can be seen, these Codes carry the notion that either the calculated nominal thickness is limited for a given pressure or if a thickness is given, the calculated pressure is limited such that yield strength of the piping material is not exceeded.

CSA Z662 Methodology

The Standard addresses both oil and gas pipeline systems and uses the ASME Code rules-based approaches as well as explicitly allowing more detailed methods for loads not directly addressed. The Standard uses both the MOP concept and design pressure calculation of ASME B31.8. However, the precision of both terms is diminished in use in the CSA standard. The MOP is defined as the maximum pressure at which piping is qualified to be operated. Design pressure is defined in four ways, two of which are actually contradictory. One definition gives the design pressure as being specified by the designer, the second calculates a design pressure based on SMYS and selected design wall thickness similar to the design pressure calculation of ASME B31.8:

$$P = (2 \cdot S \cdot t / D) \cdot F \cdot L \cdot J \cdot T \qquad (5)$$

The variable “J” is the weld joint factor, comparable to “E” in the ASME codes. The design factor, “F”, is equal to 0.800 here so that “L” provides the comparable location class factor given in ASME B31.8. The obvious point of confusion in the Standard is the use of the design pressure as both an input variable (i.e., a value specified by the designer) and as a calculated quantity (i.e., based on selected pipeline component thickness), thereby, rendering the concept circular.

In any case, the design pressure cannot be less than the MOP and the MOP is further defined as being the strength test pressure of the pipeline divided by 1.25. Hence, the hoop stress during the pressure test may reach 100% SMYS by the fact that $1.25 \cdot S \cdot 0.800$ (i.e., $1.25 \cdot S \cdot F$) $= 1.00 \cdot S$ (recall that $S \equiv$ SMYS per CSA Z662). Flanges are selected on the basis of suitability with the grade of pipe to which they are joined and ability to withstand operating pressures.

Therefore, selected flanges must possess pressure ratings compatible with the calculated design pressure, P. Flange standards are provided by CSA Z245.12 which uses the concept of nominal pressure class ranging from PN 20 to PN 420 rating which corresponds to working pressure ratings from 1,900 kPa to 41,370 kPa [275 psig to 6,000 psig] [8]. These nominal pressure classes are further declared to be maximum cold working pressure ratings. A rating of PN 20 nominally matches the cold working pressure of ASME B16.5 /ASME B16.47 Class 150 flanges. However, the ASME Class 150 flange is designated on the basis of maximum allowable working gage pressure at an elevated temperature. This reflects the intended wider range of utility for these flanges such as process plant applications. For example, ASTM A 105 flanges are rated at a maximum allowable working pressure of 150 psig at 565 °F [1,034 kPa at 296 °C]. In contrast, flanges rated to CSA Z245.12 are limited for use to a temperature of 248 °F [120 °C]. The CSA Z662 pipeline standard extends the temperature range for steel piping to 446 °F [230 °C]. For flanges, no derating need be taken above the CSA Z245.12 standard limit of 248 °F [120 °C]. The practice by CSA Z662 code users has been to limit flanges to a maximum of 10% over MOP during pressure variations.

The Standard limits stress design requirements to design conditions for operating pressure, thermal expansion ranges, temperature differential and sustained force and wind loadings. Hydraulic shock, such as pump startup and shutdown and mechanical vibrations are not addressed and the designer is advised to provide assessment of these additional loadings.

CODES AND STANDARD ASSESSMENT

The summary shows two major deficiencies:

- Differences in what is considered operating pressure
- Imprecision in language carried from ASME B31.4 and B31.8 into the other subject codes

Table 1 provides a summary of the above discussion. ASME B31.4 provides an allowance for transient overpressures of 10% over internal design pressure. The design pressure is defined to be not less than the maximum steady state operating pressure; hence, the design pressure is an operating characteristic of the system. In contrast, ASME B31.8 does not provide an explicit allowance for overpressure but, rather, includes this in its definition of maximum operating pressure (MOP) by defining it to be the highest pressure at which a piping system is operated during a normal operating cycle (compressor startup and shutdowns are expected). In other words, ASME B31.8 treats the MOP as an operating characteristic and pressure variations are included in the quantification of MOP.

However, ASME B31.8 then deviates from ASME B31.4 by defining the design pressure as a permitted quantity based on calculation of the pressure load capacity of the designed system. It then adds the supplementary concept of maximum allowable operating pressure (MAOP) which is established by taking the test pressure divided by the location class factor. The basis is that the test pressure subjects the piping to a stress equal to the yield strength and, this pressure is divided by the location class factor. This MAOP then defines an upper limit to the mechanical capacity of the piping system.

Perhaps because the CSA Z662 standard represents a combined influence of the two ASME pipeline Codes, it confusingly defines design pressure as simultaneously an input operating characteristic and a mechanical capacity limit. It also makes ambiguous the concept of maximum operating pressure (MOP) by defining it is as a mechanical capacity limit, when it is really an operating characteristic.

To help reconcile the confusion of CSA Z662, it may help to look at both ASME B31.3 (used for process plant piping systems) and ASME Section VIII Divisions 1 and 2 (used for unfired pressure vessels) [9][10][11]. In these Codes, two concepts are simply and consistently defined: design pressure, as a characteristic of the operating system; and, maximum allowable working pressure (MAWP), as an upper limit to the pressure capacity of respective piping and pressure vessel systems. The two concepts are necessary to address the two basic aspects of the design process; determination of the minimum component thickness required for a specified design pressure, and the maximum steady state operating pressure allowed for the actual thickness of pressure components installed in the system. As a point of explanation, once the minimum required thickness of the pressure containment equipment is calculated, the selection process may result in specification of a thicker member (e.g., to permit use of a standard and readily available pipe or plate thickness). The mechanical capacity of the system, using the thicker member actually used, leads to determination of the MAWP.

Both ASME B31.3 and ASME VIII Divisions 1 and 2 deal with non-steady state conditions. The ASME B31.3 Code deals with it explicitly by evaluation of the cyclical conditions due to temperature or pressure variations. Pressure variations (our particular interest for this paper), may exceed design pressure by 20% or 33% for prescribed time durations and cumulative totals. For example, a 33% overpressure allowance provides for an increase in hoop stress to 100% of SMYS during the event provided the cumulative total is no more than 100 hours per year. This means that 6,000 cycles could be tolerated if each event is of one minute duration. ASME VIII Division 1 requires the designer to consider cyclic, dynamic and impact loadings; the assessment methods in Division 2 may be used.

Division 2 methods vary in detail but do provide screening criteria which can be effectively used to limit the design effort. Method A for fatigue analysis screening is limited to materials with a specified minimum tensile strength (SMTS) that is less than or equal to 80 ksi [552 MPa]. The example given later uses API 5L B pipe material having a SMTS of 60 ksi at 100 ˚F [414 MPa at 38 ˚F] and ASTM A 105 flange material with a SMTS of 70 ksi at 100 ˚F [482 MPa at 38 ˚F] allowing application of the Method A screening criteria.

The current Code edition allows for any of the following

- combination of full-range pressure variations plus 20% of full-range pressure variations limited to 1,000 cycles for integral construction or
- combination of full-range pressure variations plus 15% of full-range pressure variations limited to 400 cycles for non-integral construction

Note that temperature cycling has not been considered in the example since most pipelines operate at temperatures near ambient conditions and temperature changes are sufficiently gradual to preclude a significant temperature differential.

It is noteworthy that CSA Z662 allows compressor and pump station piping to be designed to ASME B31.3 in its entirety.

IMPACTS ON PIPELINE USER DESIGNS

Since there are very distinguishable differences among the US industry, US federal government and Canadian industry codes as outlined above, it is worthwhile to examine how these impact a typical installation located in Canada. In this case, an oil pipeline terminal is studied; the owner has operations in both the USA and Canada and the system of interest is designed to CSA Z662.

The User's piping design specification defines the following:

- maximum operating pressure (MOP) as the maximum pressure at which the pipeline may be operated under steady state conditions
- design pressure is defined to be an input variable provided by process conditions
- Further into the specification, the design pressure is defined in abbreviated form and is, in part, inconsistent with CSA Z662 but providing a more conservative outcome

The design pressure is calculated as 438 psig [3,020 kPag] based on the selected wall thickness of 0.375 inch [9.525 mm] for the piping material. Normal pump discharge operating pressure is 110 psig [760 kPag]. Class 150 flanges are selected that have a nominal design pressure rating of 275 psig [1,896 kPag] at ambient temperature. The station is constructed and pressure transients occur in service and are estimated to range from 320 to 625 psig [2,240 to 4,310 kPag] using software and experiential estimation methods. Unexplained, actual overpressure values are not measured by the facility owner. A relief system is devised to limit maximum pressures to 110% of the declared MOP, 302 psig [2,085 kPag].

It is seen that the Standard mandated calculation, using (5), results in a code compliant design pressure of 580 psig [4,035 kPag] for 36 NPS STD wall thickness API 5L B pipe. The Standard would require installation of Class 300 flanges with a pressure-temperature rating of 740 psig [5,100 kPag] in conformance to CSA Z245.12, the supporting standard for steel flanges [10]. Since the Standard explicitly provides no rules for evaluation of pressure variations, consideration of ASME B31.4 rules appears reasonable and would provide for a pressure variation of 10% and transient pressures to 638 psig [4,400 kPag] should be considered acceptable. Also, since the CSA Z662 standard allows pump station piping to be designed in accordance with ASME B31.3, overpressure to 1.33 times design pressure can be considered acceptable (provided the cumulative time during this overpressure does not exceed 100 hours per year). The ASME B31.3 approach provides for an upper bound transient design pressure limit of 476 psig [3,280 kPag] for piping and 1.33 x 275 psig = 366 psig [1.33 x 1,896 = 2,522 kPag] for ASME B16.47 Class 150 flanges. A close reading of CSA Z662 provisions would allow use of PN 20 flanges to 1.5 x 275 / 1.25 = 330 psig [1.5 x 1,896 / 1.25 = 2,275 kPag] and an overpressure limit, per practice, to 1.10 x MOP or 363 psig [2,503 kPag] which is the same as the B31.3 overpressure limit.

Since the actual transient pressure was not definitively established, a prudent and reasonable operating step would be to measure the true transient overpressure. Unfortunately, the facility owner decided to throttle production in order to limit pump shutdown transient pressure costing a reported $100M per year in throughput and to authorize installation of a pressure relieving system in order to limit transient pressures to 110% of stated MOP, i.e. 302 psig [2,086 MPag]. The efficacy of flow throttling is not discussed in this paper. It should be noted however, that, after installation of the pressure relieving system, the pipeline is still not in Code compliance.

An additional point to remember is that care should be taken before “mixing” the requirements of different Codes and applying these mixed requirements to a specific system. In this case, CSA Z662 allows compressor and pump station piping to be designed to ASME B31.3 in its entirety. Thus, using the permitted overpressure rule would be acceptable if the station piping was actually designed to ASME B31.3 requirements. However, if, for some reason, it was designed to ASME B31.4 requirements, use of the ASME B31.3 permitted overpressure limits is not appropriate because the allowable stress criteria are different between the two Codes.

Code Non-Compliance Post Relief System Installation

The CSA Z662 standard does not address transient conditions such as the surge pressure from pump shutdown and the use of ASME B31.4 could be considered, in this instance. On first pass, the Code provides for an overpressure limit of 302 psig [2,086 kPag] and this was the basis for the owner’s decision to use a pressure relieving device set at 110% MOP. Unfortunately, this is specious; the declared MOP of 275 psig [1,896 kPag] upon which the limit is based is not consistent with either the Standard or Code.

In both documents, the MOP is intended to represent the maximum steady state operating pressure, which, in this instance is 110 psig [760 kPag] since this is the actual pressure in the line segment. Therefore, even if the maximum pressure is limited to 302 psig [2,086 kPag], the pressure difference of 192 psid [1,324 kPad] amounts to 70% of the design pressure and a fatigue analysis would be required (based on competent engineering judgment and ASME VIII Division 2 rules). For a fatigue evaluation, the stress range and cycles of exposure are under consideration, not the amount by which the overstress exceeds the incorrectly declared MOP of 275 psig [1,896 kPag]. The expected number of these operating pressure cycles needs to be estimated in order to conduct the fatigue analysis.

Since many pipelines are constructed with non-fatigue compatible features, such as fillet welded encirclement sleeves at branch connections, knowledge of the anticipated number of pressure variations is critical for the correct evaluation of fatigue loading and long term reliability assessment.

CONCLUSIONS

Based on the above, the following conclusions may be drawn:

1. The ASME oil and gas pipeline codes are well recognized in the transportation industries and have been used to develop other national codes such as Canada’s CSA Z662 standard code for oil and gas pipelines.

2. In adapting the ASME Codes, the CSA Z662 standard and ancillary standards have lost precision in defining the concepts of design pressure and maximum operating pressure.

3. In excluding consideration of fatigue loadings in CSA Z662, users of the Standard have accessed ASME B31.4 but have not understood the impact of the more precise definition of MOP contained therein.

4. Users of the CSA Z662 standard are vulnerable to incorrect assessment of fatigue loadings, such as pump surge and may potentially suffer both economic loss and loss in long term facility reliability.

Table 1 Comparison of ASME and CSA Piping Codes

	ASME			CSA
	B31.3	B31.4	B31.8	Z662
Strength limits	SMYS, SMTS	SMYS	SMYS, SY	SMYS
Stress design limits, S	⅔ SMYS, ⅓ SMTS	S	S	S
Basic design factor, F	-	0.72	0.80	0.800
Joint factor	E	E	E	J
Location factor	-	(1)	(2)	L
Temperature factor	included	-	T	T
Weld joint FSRF	W	-	-	-
Material coefficient	Y	-	-	-
Basic expression	$t = \frac{PD}{2(SEW + PY)}$	$t = \frac{P_i D}{2(SMYS \cdot F \cdot E)}$	$P = \frac{2St}{D} FET$	$P = \frac{2St}{D} FLJT$
Overpressure allowance	Yes	Yes	Included	No
Fatigue (3)	Yes	No	No	No
P, design pressure	input	input	calculated	calculated
t, thickness	calculated	calculated	input	input

Notes
1. A value of F < 0.72 may be used to account for location or service consideration
2. A value of F < 0.80 may be used according to location class, but is designated as the basis design factor in B31.8
3. Fatigue associated with cyclic and transient pressure loadings.

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