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EVALUATION AND ACCEPTANCE CRITERIA SYNTHESIS FOR ASME SECTION VIII VESSELS AND B31.3 PIPING SUBJECTED TO SLUG LOADING

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ABSTRACT

A program has been underway to improve the tolerance for two-phase flow slug loading on separator pressure vessels and inlet piping systems at operating oil field processing facilities. Elements of the program plan focused on the tailoring of industry standards to establish realistic and reliable evaluation and design criteria which provide consistent margins of safety for all elements of the systems. This paper presents the bases and development of the evaluation and acceptance criteria for the coupled pressure vessel and piping systems. Central to the evolution of this criteria was the fatigue analyses undertaken to conform with the requirements of ASME Section VIII, Division 2 for the vessels and with the intent of ASME B31.3 for the piping. The paper demonstrates the compatibility between the design by analysis methods of Section VIII, Division 2 and the more empirical methods of ASME B31.3. Suggestions are offered that might enhance the future compatibility of pressure vessel and piping design rules.

INTRODUCTION AND BACKGROUND

Two-phase flow slug loading on the inlet systems of the oil field processing facilities at the Kuparuk River Unit, North Slope, Alaska has been the subject of recent, careful consideration by ARCO Alaska, Inc. This has been made necessary by higher than anticipated oil production rates from the field, which in turn has resulted in larger flow rates of oil, gas, and water mixtures.

Under stratified conditions the flow in a pipeline is well-behaved, with liquid flowing at one velocity in the bottom of the line and gas flowing at a higher velocity above the liquid. Elevation changes or other sources of flow disruption in the pipeline can cause liquid holdup, or blockage of the gas flow by liquid "plugs". The higher velocity gas behind the liquid accelerates the liquid plug and a "slug" results. Slugs have varying lengths, densities, and configurations. At each change in direction of flow, a resultant force must be resisted by the piping and pipe support system. Unrestrained elbows and branch connections in piping networks are subjected to varying loads causing cyclic stresses. Similarly, vessel nozzles are subjected to cyclic stresses and fatigue damage. Vessel internals can also be impacted by these dynamic fluid forces. To mitigate these effects the criteria described herein have been used to upgrade the Kuparuk inlet systems to meet the increased demands.

The upgrade program employed a variety of hardware modifications to enhance fatigue performance. They included reinforcements or replacement of vessel nozzles, sleeving or otherwise reinforcing highly stressed piping components (e.g., reinforcements to components such as elbows and branch connections), replacement of pipe spools, reinforcement or addition of pipe supports, and the reinforcement of existing or the addition of new pipe support structures. While it is true that each element of the inlet systems was initially designed and supplied in accordance with industry standards, the design conditions (e.g., the loads) specified did not include such severe slug loading as those which were being experienced as the result of the higher than anticipated production. The engineering approach used was to design modifications of the integrated systems using better loading definitions and synthesized acceptance criteria, particularly for the pressure boundary elements, in order to establish balanced designs.

With a realistic estimate of the systems' responses to slug loading of given magnitudes and frequencies of occurrence, the synthesized acceptance criteria can be invoked to establish whether or not modifications of a particular complexity would be prudent. However, reliable mechanistic means do not currently exist to predict, with certainty, realistic values of the slug loading magnitudes and their frequencies of occurrence. Hence, on-line monitoring systems have been installed and operating for nearly a year. The real-time monitoring of the facilities is the most logical and most cost-effective means of assembling key information needed as input for responsible decisions regarding hardware modifications. These systems are also being used to explore means to reduce the severity of the slug loading by changes in operational parameters and controls. The reader can gain additional insight as to how these on-line monitoring systems fit into the overall program from the paper titled "Program for Improving Multi-Phase Flow Slug Force Resistance at Kuparuk River Unit Processing Facilities" (Reference 1).

INTEGRATED SYSTEMS EVALUATION METHODS AND SLUG EQUIVALENT STATIC LOADS

The inlet systems consist of various diameter flow lines (12 to 24 inch diameter) tying into inlet manifolds (30 or 36 inch diameter), each of which flows into the inlet nozzle of a 15 foot diameter separator vessel. The vessels were designed for internal pressures of 150 psig. The inlet manifolds are housed in crude oil inlet modules (buildings). The separator vessel inlet nozzles are located outside and on top of the vessels, 30 to 50 feet above grade. The inlet lines vary from approximately 100 feet to more than 300 feet in length. Figure 1 illustrates a typical inlet system and Figure 2 provides photographs of that same system.

The flow lines from the drill sites are above grade and are supported on pipe rack structures. Some anchors and guides were incorporated in the original designs, but the flexible steel support structures allow considerable flow line movement when subjected to severe slug-induced forces. The inlet manifold systems are supported by the inlet module structures, but these have likewise proven to be less than adequate to limit slug-induced movements. The inlet manifolds terminate at the separator vessel inlet nozzles. Changes in direction of flow in the manifolds immediately upstream of the vessels and the lack of supports caused significant axial loads, shears, and bending moments to be introduced into the vessel shells at the inlet nozzles. Additional forces are introduced into the vessel shells by attached internals intended to divert the flow entering the vessel.

Figures 3(a) through 3(e) show an idealization of the problem. If flow conditions which could result in severe slug loading had been postulated during the original design, it is likely that a conservative value for the force **R** would have been specified. For steady flow conditions, a comparative force can be computed from momentum considerations and a load factor could have been applied to account for the transient nature of the flow. Design and operating pressures and the numbers of pressure cycles would have been appropriately specified for the piping systems and the vessels. The design of the piping and the locations of pipe supports would probably have been selected using traditional, proven piping design practices. Simple statics would have been applied to establish reaction loads for which the system support structures would have been designed. Likewise, nozzle reaction loads would have then been developed using similar methods and included as part of the vessels' design specifications.

Few would have questioned that the design of each individual element was conservative, even though no focused effort would have been undertaken to understand how conservative one class of elements (say, the piping) was relative to the other classes of elements (i.e., the vessel nozzles, the pipe supports, or the system support structures). Because the systems considered in this work were already built and in operation, more refined evaluation methods were employed. Also, since a major issue was related to the fatigue resulting from a range of dynamic loading magnitudes and frequencies, the concept of expressing the fatigue resistance of the various mechanical and structural elements in terms of slug equivalent static loads was introduced.

Why An Integrated Approach Is Needed

The inlet piping systems at each of the facilities are complicated. The pipe size is both large and varying. The piping layouts are different at each facility and they have numerous branches, bends, and other flow restrictions.

In addition to the variety of the pipe components and the routing, pipe supports and the associated support structures include a wide spectrum. Nearly all of the original supports were intended to act mainly as gravity supports. While most piping was supported to accommodate some lateral loads, the systems were not, in general, originally designed to withstand severe slug loading.

As noted, all inlet piping flows into the inlet manifolds. These inlet manifolds are routed to the separator vessels and they mate to inlet nozzles of the same diameter. They are large nozzles for vessels the size of these separator vessels (i.e., 30 inch or 36 inch diameter nozzles in 15 foot diameter vessels) and the slug loading in the piping systems can produce large reaction forces on the nozzles. The original designs of the separator vessel nozzles made no provision for reaction loads from severe slug loading on the attached piping.

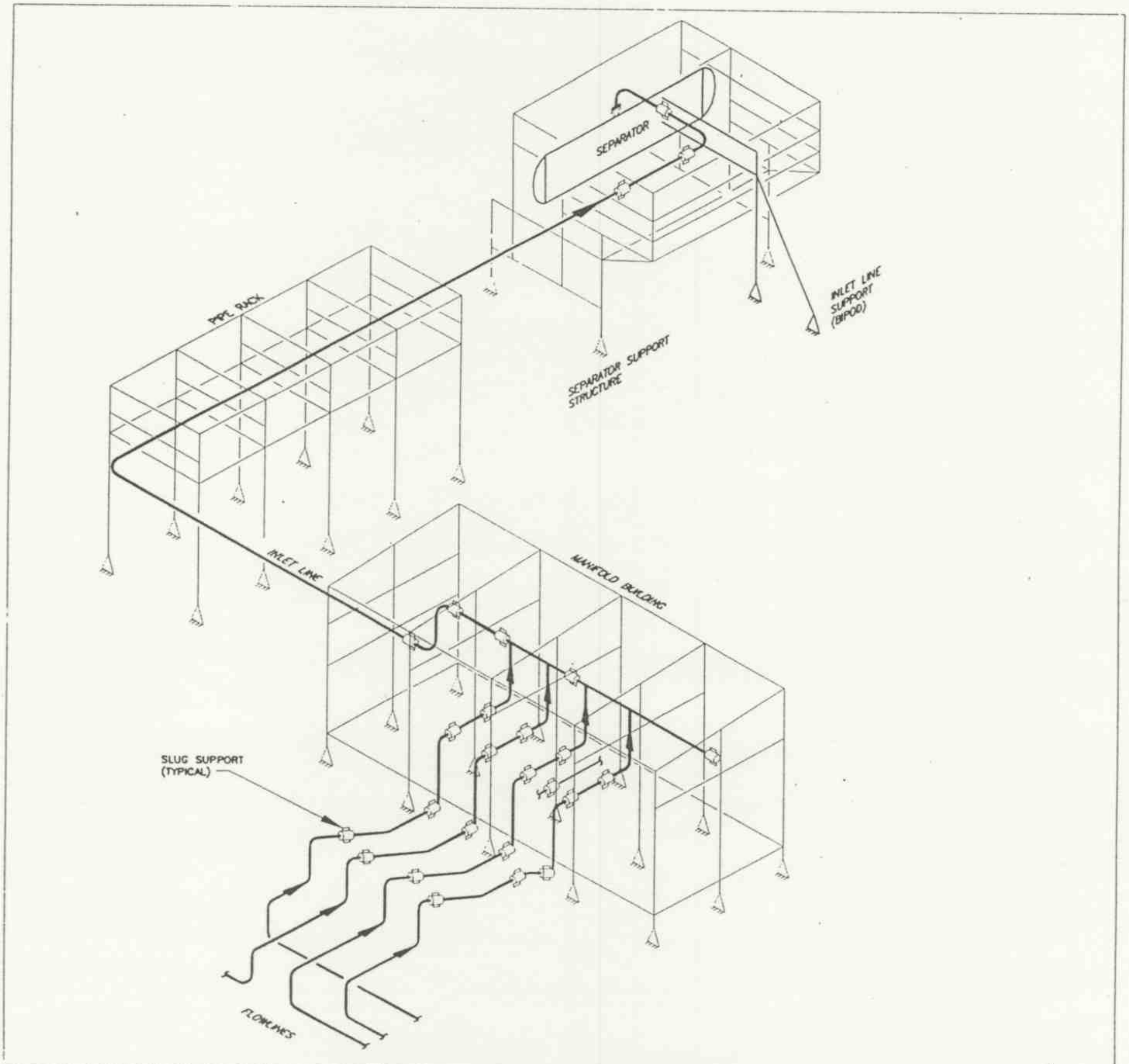

FIGURE 1 INLET SYSTEM PIPING

If completely new inlet piping systems were being designed for severe slug loading service, a much different layout of the piping and arrangement of the piping supports would probably result. Piping runs would be kept as straight as possible and as low to the ground as practical. Pipe supports would have been incorporated to protect the vessel nozzle from excessive pipe forces. Multiple inlet nozzles into the separators or, possibly, slug catcher vessels upstream of the separators, would have been used. The design of the individual elements (i.e., the piping, the pipe support structures, and the vessels with their more conventional nozzles) would have then proceeded in the more traditional, conservative, uncoupled manner.

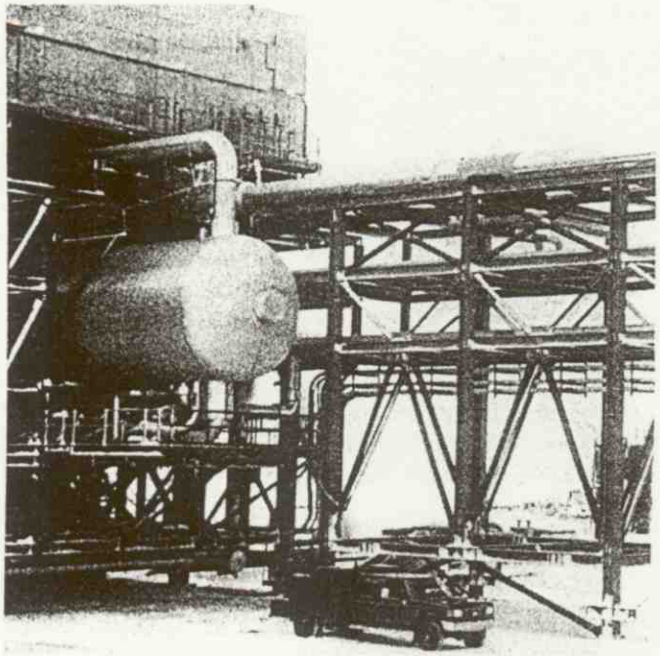 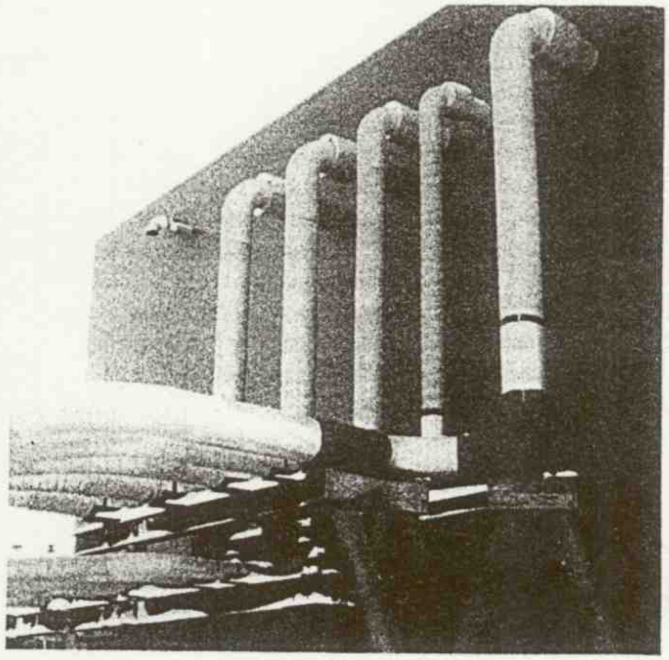

FIGURE 2 PHOTOGRAPHS OF THE PROCESSING FACILITY INLET SYSTEM

Modifying existing designs to increase a system's tolerance to withstand loadings requires a more integrated approach. The piping support structures act together with the piping, which is also a "structure". As the system deflects, force follows stiffness. The large diameter pipe runs with closely spaced supports are very stiff. Deflections can thus result in both high stresses in the pipe and large forces being transmitted through the pipe to the vessel nozzle. Treatment of the elastic interaction between the vessel nozzle and the attached piping is essential for a realistic understanding of the state of stress in the pipe in the vicinity of the nozzle and the stresses in the vessel as a result of pipe reactions on the nozzle.

Adding to the intricacy of the integrated system response is the fact that the vessels are supported on framed structural modules, a significant elevation above grade. The elevated vessel support framing also has an influence on the stiffnesses of the piping models' supports at the nozzles. This influence has been determined to be significant.

The need for integrated models was only acknowledged after the work began by using more traditional, approximate analysis methods. Those traditional methods were tested by applying the more rigorous understanding introduced above. The conclusion was clear. The interaction effects have a significant influence. In order to provide the maximum potential improvement by modifying the existing systems, and in order to more accurately understand the true capabilities of the systems (whether modified or existing), one must face up to the reality that all elements of the elastic piping, structural and vessel systems act together as an integrated system.

<u>How Integrated Evaluations Are Conducted</u>

When addressing the issue of slug loading on the integrated systems, it is only natural to focus initially on the piping. Some of the reasons for this tendency are (a) there is more piping than anything else, (b) it carries the product which is producing the phenomenon, and (c) it is the hardware element which first experiences a load and thus it loads the support structures and the vessel nozzle.

Since it would not be practical to assemble one large analytical model incorporating all elements of the integrated piping, piping support structures, and the vessel in a given system, a more workable modeling method is used. It centers around an analytical model (or models) of the piping system. At points of interaction between the piping and the piping support structures or the vessel nozzle, appropriate boundary conditions are specified for the piping analytical model. The selection of piping model support point boundary conditions is undertaken with care. When it is judged that the interaction effects discussed above are not important to the response of the piping system, large support stiffnesses are specified. On the other hand, when through a combination of experience and auxiliary calculations it is deemed likely that interaction effects are important, the piping model supports are represented as springs.

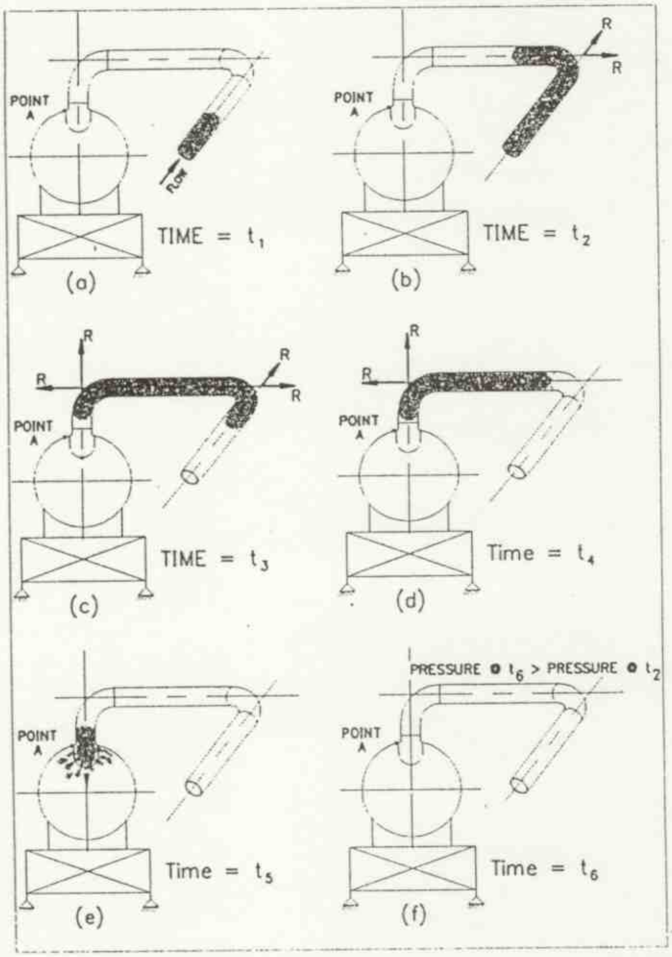

FIGURE 3 IDEALIZED SLUG LOADING SEQUENCE

The spring constants are determined from analyses of the support structures which they represent. Analytical modeling of those other system elements must proceed in parallel with the modeling of the piping.

After all of the piping models' support spring constants are known, the appropriate loading conditions are selected and the responses of the piping analytical models are computed. Part of the output from the piping analyses are reactions at the support points. These computed reactions are used to help establish the corresponding loading specifications necessary to complete the evaluations of the support structures. Particularly important are the nozzle reaction loadings. The realistic nozzle loads are essential to establish an understanding of the tolerance to slug loading which exists in the nozzles and in the vessel pressure boundaries local to the nozzles.

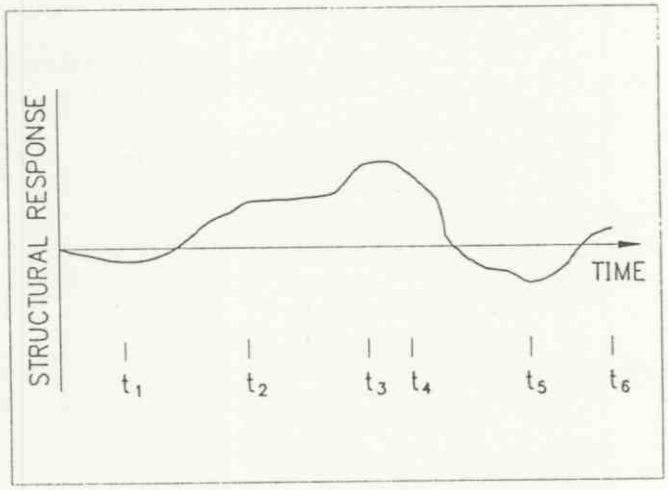

FIGURE 4 STRUCTURAL RESPONSE AT POINT A

Slug Equivalent Static Load Capability Reporting

Six different points in time during a single slug event have been illustrated in Figures 3(a) through 3(f). Time t_1 is before the slug produces any significant response of the portion of the system under study, t_2 corresponds to when the slug is loading the first elbow with momentum reaction forces **R**; and, so on. Time t_6 corresponds to a point in time when the slug event has passed but it is recognized that the gas behind the slug had most likely increased the pressure in the system to a value greater than it was in front of the slug.

Consider now the structural response at some point, say Point A on the pressure vessel's shell as identified in Figure 3, during this slug event. Without being specific about what quantity is being plotted, Figure 4 illustrates in concept what might occur. Depending on the element of response being monitored and the criteria being used to judge the acceptability of the design, one could understand the relevance of the loadings produced by that particular slug event, at least as controlled by that location on the pressure boundary. One could also estimate the points in time, during the slug event, which were most important relative to the different issues addressed by the acceptance criteria. For example, if the parameter under study was maximum primary stress, the data would suggest that somewhere around t_3 (Figure 3(c)) was the most important time. On the other hand, if stress range was the parameter of interest, the data would suggest that the range experienced between times t_3 (Figure 3(c)) and t_5 (Figure 3(e)) would be of most relevance.

The use of an integrated approach to compute the static response to slug loading representations, as illustrated in Figures 3(b) through 3(e), was discussed above. Working with those statically computed results for the locations where the previously mentioned slug monitoring systems collect data on the necessary elements of the systems' responses, it is, at least in principle, a straightforward task to establish what is defined as a "***slug equivalent static load***" magnitude for each such location and each recorded slug event. Although it is not the subject of this paper, that is being done at enough locations in the systems to provide calibration of the computed static responses.

With the ability to accurately compute the static responses of the systems to slug loading of variable magnitudes and postulated slug lengths, and with the ability to benchmark those statically computed responses, the only missing link is to establish the design lives of the different elements of the systems as controlled by slug loading magnitudes and frequencies. The numbers of loading cycles are being continuously monitored and various assumptions can be explored to forecast the future. With the selection of appropriate vessel and piping design criteria, the capabilities of the different pressure boundary elements for alternative design modifications can be represented by plots such as illustrated in Figure 5. The figure shows a component's capabilities in terms of the relationship between the magnitude of the slug equivalent static load and the allowable numbers of cycles for each load magnitude. An allowable value as controlled by vessel primary stress or piping sustained stress can also be shown for convenience on the same plot. The selection of criteria upon which these curves are based is discussed below.

CONSISTENT VESSEL AND PIPING CRITERIA

The fact that existing facilities were being evaluated and modified allowed some latitude in the development of acceptance criteria. Since both the existing vessel and piping industry standards are for new construction, use of their rules for guidance, rather than strict compliance, was possible. Actual operating pressures and temperatures, weights, and expected slug loading were the only loads judged to be significant. Also, since a well developed corrosion monitoring and mitigation program is employed at the Kuparuk facilities, it was not considered necessary to deduct material thicknesses to account for corrosion. Other issues are discussed below.

Baseline Criteria

The separator vessels were originally specified, designed, manufactured and Code Stamped to comply with the requirements of the ASME ***Boiler and Pressure Vessel Code, Section VIII, Rules for Construction of Pressure Vessels, Division 1*** (Reference 2). It was reasonable to continue reliance on that standard as the basis for conducting evaluations and designing modifications to the vessels and their inlet nozzles. Yet, because Division 1 lacks detailed guidance for fatigue loadings, reliance on certain provisions of the ASME ***Boiler and Pressure Vessel Code, Section VIII, Rules for Construction of***

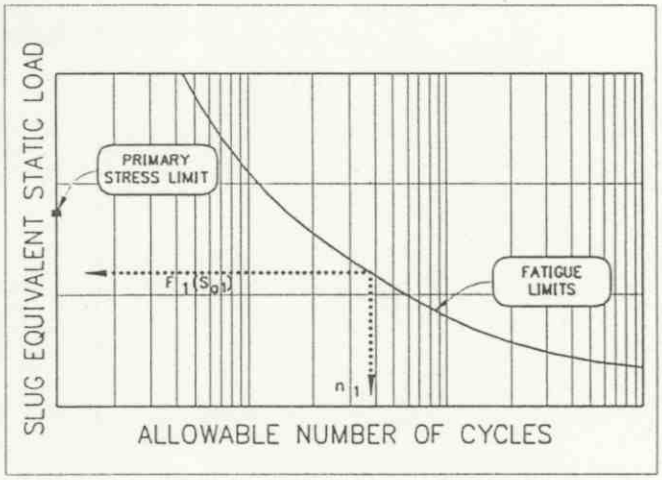

FIGURE 5 SLUG EQUIVALENT STATIC LOAD VERSUS ALLOWABLE NUMBER OF CYCLES

Pressure Vessels, Division 2, Alternate Rules (Reference 3) was considered necessary. This was judged to be the case, for some of the reasons discussed below, even though there are also some differences between Division 1's and Division 2's material, fabrication, examination and other requirements which may have influenced the original construction of the Division 1 vessels.

ASME Section VIII, Division 1, Paragraph U-2(a) requires —

> "The User or his designated agent shall establish the design requirements for pressure vessels, taking into consideration factors associated with normal operation, and such other ..."

Consistent with that requirement, mechanical loads on the nozzles to account for the expected piping reactions due to slug loading are needed. This is also suggested by ASME Section VIII, Division 1, Paragraph UG-22 which establishes the following requirements:

> "The loadings to be considered in designing a vessel shall include those from :
>
> ...(e) cyclic and dynamic reactions due to pressure or thermal variations, or from equipment mounted on a vessel, and mechanical loadings;..."

Because ASME Section VIII, Division 1 provides no specific criteria for designing to resist fatigue damage, more comprehensive design criteria to supplement the general design rules provided by Division 1 must be established. This need is communicated by footnote 21 to the heading "OPENINGS AND REINFORCEMENTS" (see UG-36) which says, in part —

> "... External loads such as those due to the thermal expansion or unsupported weight of connecting piping have not been evaluated. These factors should be given attention in unusual designs or under conditions of cyclic loading."

Such generally worded Code requirements are also included in Paragraph U-2 (g) which says —

> "(g) This Division of Section VIII does not contain rules to cover all details of design and construction. Where complete details are not given, it is intended that the manufacturer, subject to the acceptance of the Inspector, shall provide details of design and construction which will be as safe as those provided by the rules of this Division."

If new vessels were being designed today for current flow conditions, the specified requirements would include not only a set (or sets) of nozzle loads to account for slug primary type loading pipe reactions, but also, the specified requirements would set forth fatigue design criteria. Since Section VIII, Division 1 has no specific fatigue criteria, it is likely that Division 2's fatigue design criteria would have been specified. Thus, Section VIII, Division 2's criteria were used for this effort to judge the tolerance of the existing vessel nozzles and nozzle modifications to withstand slug loading in the inlet piping systems. It is argued this approach meets the intent of the above noted guidance of Division 1, at least as the goal here was to establish criteria for use in evaluating and modifying existing vessels.

The inlet piping was originally designed and constructed in accordance with the ASME ***Code for Pressure Piping***, either the section titled ***Chemical Plant and Petroleum Refinery Piping, B31.3*** (Reference 4), or the section titled ***Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia, and Alcohols, B31.4*** (Reference 5). The ASME B31.3 Code was written for new construction while the ASME B31.4 Code was written to include some operation and maintenance requirements for pressure piping. Severe slug loading was not considered in the original designs of either the ASME B31.3 or the ASME B31.4 portions of the piping systems. Since the purpose of this project was to evaluate and modify previously constructed systems, and since ASME B31.3 is only for new construction, strict application of the ASME B31.3 requirements did not seem appropriate. However, using the ASME B31.3 requirements as guidelines, based upon judgments similar to those described above for the vessels, was considered reasonable.

For piping originally designed in accordance with ASME B31.4, a different basis for judgment was required. While ASME B31.4 addresses some maintenance issues in a very general way (see Paragraph 450.2(d)), it does not provide the type of detailed evaluation and modification design criteria that were necessary for this project. Even the ASME B31.4 new design and construction requirements do not provide rules for high-cycle fatigue design. Therefore, all piping was evaluated using an ASME B31.3 fatigue analysis methodology, since the ASME B31.3 and B31.4 Codes' requirements have a common origin.

Internal Pressure, Primary Stress, Sustained Load and Occasional Load Criteria

The main body of ASME Section VIII, Division 2 contains the same 'design by formula' provisions as does Division 1. However, under a statement of scope, AD-100 (b) says in part —

> "(b) when complete rules are not provided for a vessel or vessel part, or when the vessel designer or user chooses, a complete stress analysis of the vessel or vessel part shall be performed considering all of the loadings specified in the User's Design Specification. This analysis shall be done in accordance with Appendix 4 for all applicable stress categories and in accordance with Appendix 5 when fatigue evaluation is required. ..."

This means two fundamental criteria ingredients are involved. First, a set of basic load carrying capabilities or strength criteria are expressed in terms of allowable values of *primary stress intensities*[1]. The second, a set of fatigue life criteria, are expressed in terms of the allowable range of *peak stress intensities*.

The vessel rules' *design loads* or strength criteria are controlled by *primary membrane* or *primary bending* stress intensities. The criteria apply to the maximum membrane stress intensity or primary bending stress intensity experienced at a point, as opposed to a stress intensity range. But since it is the maximum stress intensity which is being evaluated, the stresses due to other major loadings, such as the vessel internal operating pressure, must be added to those which result from slug loading. Further, it is the stress intensity resulting from stresses due to the controlling slug loading condition, which when added to the stresses due to the internal pressure loading condition, that must be limited. There may actually be other slug loading conditions which produce higher slug loading stresses; but it is the condition which produces the highest combined stress intensity that controls.

Another variable that enters into the search for the controlling primary stress intensity limit condition is the allowable stress intensity itself. The *general primary* membrane stress intensity, P_m, may not exceed $1.0*S_m$, where S_m is the basic allowable for the material being considered. Yet in local regions, such as at the intersection of the nozzle with the basic shell plates or reinforcing pad plates, a value of $1.5*S_m$ is permitted for what is defined as a *local primary* membrane stress intensity, P_L.

In order to implement these criteria, it is necessary to prepare a finite element model of each unique nozzle geometry. Such a model showing stress contours of the membrane stress intensity in the vicinity of a nozzle is shown in Figure 6. Then, working

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[1] The terms shown in italics when they first appear have specific meanings as defined by ASME Section VIII, Division 2. The reader who is not familiar with those terms may benefit from a review of the Code's definitions.

with (a) such a model, (b) an understanding of the basic concept of a slug equivalent static load as discussed earlier in this paper, (c) the ASME Section VIII, Division 2 definitions of primary general membrane and primary local membrane stress intensities, and (d) some good bookkeeping and linear interpolations, a realistic quantification for the allowable slug equivalent static load magnitude, as controlled by the vessel nozzle's primary stress intensity allowables, can be obtained.

In this same category of considerations, there are similar issues relative to the piping. An attribute of the slugging phenomena is locally increased pressures as the slug passes a point. The increased pressures continue at that point for a time after the slug has passed. The ASME B31.3 Code provides well defined limits for internal pressure in Paragraphs 302.3.5(a) and 304, with variations from these limits permitted by Paragraph 302.2.4. These limits were considered appropriate for dealing with this attribute of the slug loading phenomena. The limits in Paragraph 302.3.5(a) are referred to as limits for sustained loads. In addition, stress limits for piping subjected to occasional loads are established in Paragraph 302.3.6.

Both sustained load and occasional load limits are intended to protect the piping system from structural collapse. Sustained load stresses, which are to be evaluated in combination, are "the sum of longitudinal stresses...due to pressure, weight, and other sustained loadings". Occasional load stresses, which are also to be evaluated in combination, are "the sum of the longitudinal stresses due to pressure, weight, and other sustained loadings...and of the stresses produced by occasional loads, such as wind or earthquake". The longitudinal stresses mentioned are traditionally determined by adding a longitudinal pressure term to a bending moment stress term. The bending stress term usually represents weight and other sustained loadings and, when appropriate, various other occasional loadings. For the application of these criteria it is convenient to think of the pipe acting both as a long slender pressure vessel and as a continuous beam spanning between pipe supports.

Slug loading must also be resisted by the supporting structures to assure equilibrium of the piping system. At issue was whether slug loading should be considered as sustained loads or occasional loads. A higher allowable stress is permitted for occasional loads. The permitted increase for occasional loads in ASME B31.3 is 33 percent, so the issue could be significant.

Whether loads are classified as sustained or occasional is mainly based upon the duration of the loads. Since ASME B31.3, Paragraphs 302.3.5(c) and 302.3.6 do not provide specific guidance to distinguish between sustained and occasional loads, other B31.3 paragraphs must be relied upon for an approach. Paragraph 302.2.4 deals with whether or not a pressure or temperature increase can be permitted to exceed design (or normal) values. Increased stresses as a function of a

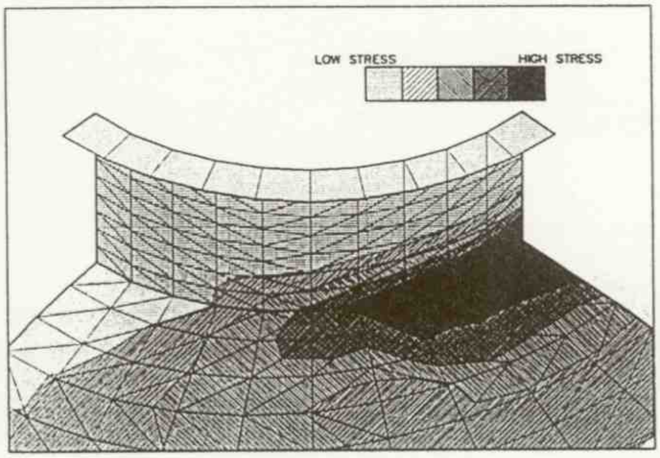

FIGURE 6 STRESS INTENSITY CONTOURS

percentage of the design allowable are permitted for varying durations.

Early observations, which were later confirmed by data from the on-line monitoring systems, suggested that slug loading can account for substantially increased pressures and stresses for as much as 5 to 10 percent of the time or between 400 and 800 hours per year. Given the limited durations permitted by ASME B31.3, Paragraph 302.2.4, it was concluded that the slug loading at these facilities should be classified as sustained loads. This means that pipe stresses due to slug pressure increases and slug momentum forces should be combined with other sustained loads to satisfy ASME B31.3, Paragraph 302.3.5(c).

Fatigue Criteria

The ASME Section VIII, Division 2 vessel fatigue criteria require an evaluation based upon peak stress intensities. Those peak stress intensities are calculated from the maximum range of cyclic stresses experienced. The basic allowable permitted for each peak stress intensity range is a function of the number of times the peak stress intensity range is expected to occur. Different terms with similar meanings are used by ASME B31.3 to establish the piping fatigue criteria rules. The characteristics of these two different sets of fatigue criteria are discussed in the following paragraphs.

First there is the matter of peak stresses versus primary stresses as defined by ASME Section VIII, Division 2. In many applications, such as the ones being considered in this paper, peak stress is an quantity used to characterize concentrations of strains that are not usually calculated directly with linear elastic analyses. In the application of ASME Section VIII, Division 2 fatigue criteria, peak stresses are most often approximated by multiplying calculated primary plus *secondary stresses* by a stress concentration factor (SCF). However, the establishment of numerical values for the SCFs is largely a matter of experience and judgment. The code defined secondary stress (i.e., category Q stress intensity) is intended to account for the effects of discontinuities in geometry. These would include the redistribution of shell stresses due to the nozzle opening and the intersection of a nozzle cylinder with the vessel and the nozzle reinforcing pad. When the concept of using a SCF to determine peak stress was first made a part of the ASME Code fatigue rules, it was seldom possible to calculate accurate values of the secondary stresses at all locations of interest. Thus it was, and often still is, common to see references to SCF multipliers greater than 3. With analysis methods such as those available today (see Figure 6), SCFs ranging from 1.1 to 2.0 are justified for many applications. But, it can not be over-emphasized that rational values must be used since conclusions regarding the reported fatigue life are so heavily influenced by the values selected.

For the application of fatigue criteria to the piping systems' response to slug loading, it is useful to review some history. The common fatigue analysis methods of the B31 Codes are based on testing and the work done by the B31 Task Force on Flexibility in the early 1950's and are summarized in a paper by A.R.C. Markl published in the ***Transactions of the ASME*** (Reference 6). The Markl paper states:

> "... the curves of failure stress versus numbers of cycles to failure parallel each other for straight pipe and other components. While this is not strictly true, test data conform reasonably well to a law expressed by
>
> $$iSN^{0.2} = C \quad \text{..........................(4)}$$
>
> where i designates the stress-intensification factor, S the...[reversing stress] (cyclic moment...divided by the section modulus of matching pipe, rather than fitting), N the number of stress reversals to failure, and C a materials constant."

It is important to distinguish between the piping "stress-intensification factor" (SIF) and the vessel terms "stress intensity" and "stress concentration factor" as discussed above. They are not the same. The piping stress-intensification factor is determined from fatigue <u>tests</u> of full-scale piping assemblies containing a piping "part under consideration". Markl defined the stress-intensification factor as "the ratio of the bending moment producing fatigue failure in a given number of cycles in a straight pipe of nominal dimensions to that producing failure in the same number of cycles in the part under consideration".[2]

Markl went on to point out that his Equation (4) is valid between 20 and 2,000,000 cycles.

Further, Markl's tests established a material constant for Grade B carbon steel of 245,000 psi for reversing stress (similar to the vessel alternating stress) or 490,000 psi for full stress-range (two times the reversing stress and similar to the vessel peak stress-range). Testing has also established material constants for other comparably ductile materials, but no conclusive evidence has been yet offered to the ASME codes and standards committees of any increased fatigue capacity for high tensile strength materials, e.g., pipeline grade materials. Consistent with a soon to be published Welding Research Council bulletin (Reference 7) which provides an industry standard for the development of piping SIFs, the Grade B carbon steel values for the material constant C were used in this program. Also, these points should be understood within the context that prevention of failure, as implied by the piping design rules, corresponds to an established factor of safety against a through-wall crack where the pipe would leak. On the other hand, the intent of the vessel design criteria is to prevent crack initiation, again within some specified, yet different factor of safety. This contrast adds to the difficulty of developing criteria which are consistent between vessels and piping.

Normally, the application of the Markl criteria in the ASME B31 Codes consists of a fatigue analysis to verify that piping has sufficient flexibility to withstand displacement limited strains (see ASME B31.3, Paragraph 319.2.1) and stresses (Paragraph 319.2.2) caused by thermal expansion cycles and other externally imposed displacement effects. The B31 Task Force on Flexibility of the early 1950's documented in the Markl paper the expectation that:

> "Piping systems are subject to a diversity of loadings creating stresses of different types and patterns, of which only the following more significant ones need generally be considered in piping stress analysis:
>
> 1. Pressure, internal or external.
> 2. Weight of pipe, fittings and valves, contained fluid and insulation.
> 3. Thermal expansion of the line."

Typically, an ASME B31.3 analysis would consist of only a pressure, weight, and thermal expansion analysis. But, the slug loading which motivated this work can only be characterized as the type of load referred to in Paragraph 301.5.1 as one of several possible dynamic effects "which should be considered when they occur". Since ASME B31.3 offers no specific guidance for the analysis of cyclic primary stresses, criteria needed to be developed.

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[2] Markl's definition is somewhat general and presumes the reader has reviewed several prior papers authored by him. A significant clarification occurs if the reader understands that stress-intensification factors are baselined against butt weld fatigue data. That is, the stress-intensification factor for a butt weld is assigned the value of 1.0; the stress-intensification factor for polished pipe, i.e., pipe free from stress concentrations, approaches 0.5.

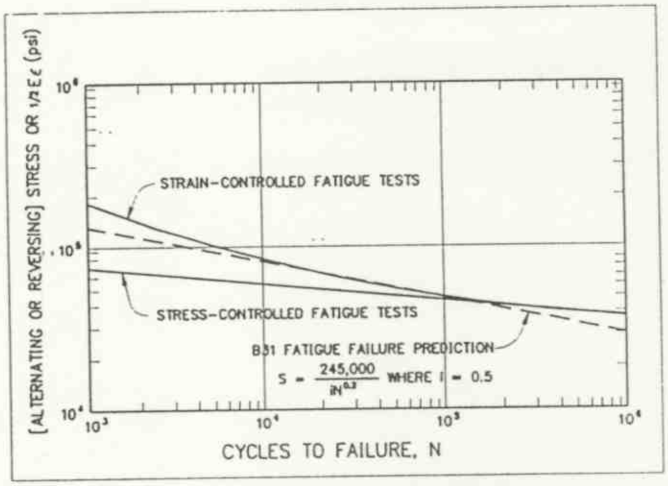

FIGURE 7 ASME CRITERIA DOCUMENT AND B31 CRITERIA FATIGUE PREDICTIONS

A figure from the document titled "Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2" (Reference 8), often referred to as the "ASME Criteria Document", has been edited and is shown here as Figure 7. The figure shows separate curves for strain-controlled and stress-controlled fatigue tests; strain-controlled being characteristic of displacement [limited] stresses as defined in ASME B31.3 and stress-controlled being characteristic of the primary stresses which result in the piping from slug loading. In the region above 70,000 cycles the strain-controlled and stress-controlled curves approach each other. The Kuparuk on-line monitoring systems have shown the number of slugs expected during the lives of the systems is well above 70,000, thereby justifying the use of the common curve.

A plot of the Markl developed B31 fatigue failure prediction curve described above is also shown on Figure 7. The generally favorable agreement, above approximately 70,000 cycles, is based upon two important values assigned to parameters in the original Markl equation. First, Figure 7 is based upon the ASME vessel commentary documentation which plots the number of cycles to failure versus alternating stress, S_a. Thus, a value of 245,000 psi is used for the material constant C in Markl's equation. The other point is that the vessel commentary data is from the testing of polished bars. Therefore, the developed B31 curve has been plotted in Figure 7 using a value of 0.5 for the stress-intensification factor, i. This approaches the value which would apply for "polished" pipe.[3]

Thus, to establish criteria with consistent margins of safety and to present the piping evaluation conclusions in a manner consistent with the vessel evaluation conclusions, "ASME B31.3 Code" fatigue criteria were developed based on the Markl, et al., work and the above demonstrated comparability with ASME Section VIII, Division 2 criteria. This was done by taking the B31 fatigue failure prediction curve (or equation) and dividing the full-range material constant for carbon steel (490,000) by 1.67, which is the safety factor for ASME B31.3 estimated in the Markl paper. The program's ASME B31.3 fundamental fatigue criterion was thus established as:

$$iSN^{0.2} \leq 290,000 \tag{1}$$

This requirement is felt to be appropriate so long as slug loading concerns are dominated by the need to design piping systems capable of withstanding large numbers of slug loading cycles. It is also argued that the resulting piping fatigue requirements are consistent with the fatigue evaluation methods and criteria presented earlier in this paper for the associated vessels.

The issue of stress range versus the maximum absolute value of the stresses influences the manner in which the search is conducted to find the controlling slug loading conditions. This is true when implementing both the vessel and the piping criteria. It is not a single slug loading condition (i.e., a single most critical location of the slug in the system), but rather a critical pair of slug loading conditions which causes the maximum range of the stresses. This makes the bookkeeping somewhat more involved. However, only the large numbers of cycles of stresses due to slug loading normally need to be considered. Then, referring to the terminology used by ASME Section VIII, Division 2, for any given number of stress range occurrences (i.e., the number of slugs that pass), say n_1, the material fatigue curves specify the permitted magnitude of the alternating stress intensity, say S_{a1}. Yet, one can determine the magnitude of the peak stress intensity range, and hence the alternating stress intensity, that corresponds to a slug of a given static equivalent load magnitude. With linear ratioing, it is possible to calculate the slug load magnitude which produces the alternating stress intensity, S_{a1}, which the adapted criteria permit for that particular number of cycles, n_1. And so with just a little more work, one can prepare a plot of not just alternating stress

[3] Further confirmation of the B31 approach is derived from the knowledge that a relationship of approximately 2 to 1, relating peak stress to intensified stress for elbows and branch connections, was observed by the nuclear piping committee in the early 1970's. Paragraph NC-3673.2(b) of Section III (Reference 9) quantified this by stating that the stress-intensification factor, i, may be taken as 1/2 the product, C_2K_2. The product C_2K_2 consists of a secondary stress index multiplied by a peak stress index forming a scalar by which nominal moment stress in pipe is multiplied to obtain peak bending stress. This 2 to 1 relationship appears to be the same relationship between the baselined butt weld fatigue data and polished pipe (see footnote 2).

intensity versus number of slug loading cycles, but a plot of static equivalent slug load magnitudes versus number of permitted loading cycles, as was suggested in Figure 5.

The ASME Section VIII, Division 2 and the ASME B31.3 Codes also provide guidance on how to evaluate histograms of slug load magnitudes. For vessels, a quantity called the *usage factor*, U, is introduced. Suppose one is dealing with slugs of three different magnitudes, say F_1, F_2 and F_3. And, further suppose that the loading histogram called for a design that could tolerate n_1 cycles of F_1, n_2 cycles of F_2 and n_3 cycles of F_3. To comply with the requirements of ASME Section VIII, Division 2 one would use a plot as illustrated in Figure 5 to read three corresponding values of the allowable number of loading occurrences, as if these three loads were completely independent. The three allowable numbers of loading cycles would be assigned to the variables N_1, N_2 and N_3. One would then compute the usage factor, U, from the relation:

$$U = n_1/N_1 + n_2/N_2 + n_3/N_3 \qquad (2)$$

As long as the usage factor, U, computed in the above manner, was less than or equal to 1.0, the design would be acceptable in accordance with ASME Section VIII, Division 2 criteria. The criteria presented in Paragraph 302.3.5(d) of ASME B31.3 address this issue of loads with different magnitudes in terms of a quantity called the "number of equivalent full displacement cycles". However, it can be shown that the two approaches give comparable results because of the form of the ASME B31.3 material fatigue curve.

SUGGESTIONS FOR IMPROVEMENTS TO ASME SECTION VIII VESSEL AND B31 PIPING CRITERIA STANDARDS

The work which lead the authors to write this paper was both challenging and rewarding. However, there were occasions where it was felt that the issues which needed to be addressed were obscured by omissions in and inconsistencies between the industry codes and standards which one would normally expect to apply to a project of this nature. In the spirit of providing the industry with some views held by practicing engineers, who also have some unique insights on the rationale behind criteria published in two of our most important pressure boundary component codes and standards, the following suggestions are offered:

1. ASME Section VIII, Division 1 and Division 2 should be combined into one document. The one book would allow both "design by formula", for simple cases, and "design by analysis" for any case. In the event that any specified loading would result in primary stress intensities higher than some threshold value, design by analysis would be clearly required. Similarly clear criteria would be included to specify when fatigue evaluations are required. Much current duplication and confusion could be eliminated.
2. When fatigue evaluations are required for ASME pressure vessels, much clearer requirements relating stress concentration factors (SCF) to component geometry and construction details should be specified in the Code. Currently, wide latitudes and significant differences in opinions exist on this subject. Yet, the conclusions regarding design fatigue life are extremely sensitive to small variations in the selected values.
3. Material allowable fatigue curves should be extended beyond 1,000,000 cycles, at least for the most widely used classes of materials.
4. Terms defined for conducting the ASME B31 piping code fatigue evaluations were established when those rules were empirically developed in the 1950s. They do not agree with the terms defined for ASME Section VIII, Division 2 fatigue evaluations. At piping to vessel nozzle interfaces as a minimum, and for integrated projects in general, this is confusing and can be dangerous. These and other inconsistencies between the vessel and the piping codes should be eliminated.
5. The fatigue rules for ASME B31.3 (and other B31) Codes are based upon displacement (or strain) limited stresses. Load limited or non-self limiting stresses follow a less conservative S-N curve below some threshold number of cycles (see Figure 7). Serious fatigue damage can be sustained from non-self limiting sources, e.g., liquid slug loading, fluid hammering, earthquake, and other cyclic inertia effects. At least a discussion of this source of fatigue damage should be introduced into the B31 Codes and perhaps even adjunct rules should be developed.
6. The ASME B31.4 pipeline (and ASME B31.8) Code rules for piping above grade do not provide the same protection for fatigue as does the process (B31.3) piping Code. This is evidenced by generally higher calculated fatigue stresses in the analyzed Kuparuk near-plant pipeline components (B31.4) than in comparable in-plant (B31.3) piping components in the same system subjected to similar loads and numbers of cycles. The pipeline Codes should consider comparable fatigue rules for piping above grade where hazards are similar in nature.

Again, these are some of the recommendations noted by the authors. They are offered with the understanding that such actions would require careful consideration by a broad spectrum of professionals from the industry. Of course, it is also recognized that many others in the industry would have equally constructive suggestions which should be evaluated before any action could be taken.

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