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Design of Pump Casings: Guidelines for a Systematic Evaluation of Centrifugal Pump Pressure Boundary Failure Modes and their Mechanisms

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ABSTRACT

This document contains recommendations of rigorous pump casing design techniques. A comparison of different design methods is made. The most sophisticated of these is then described in detail. Finally, a list of design checks by which most centrifugal pump casings might be evaluated is given.

INTRODUCTION

Currently, there is not an existing code or standard particular to the design of centrifugal pump casings. Because they must hold internal pressure, it is common for pump users to specify that a pressure vessel code be used as the basis of a manufacturer's casing design for high pressure pumps. In fact, many of the methods which are employed within two of the most well-known pressure vessel codes – American Society of Mechanical Engineer's (ASME) Boiler and Pressure Vessel Code (2011), and European Standard DIN EN13445 (2012) – are useful for evaluating pump casing designs. However, even the publishers of these codes indicate that pumps are not pressure vessels, but rather 'mechanical devices'. Consequently, it is not surprising to encounter places within these pressure vessel codes which are not applicable to pump design. The intent of this manuscript is not to provide a list of arguments for ignoring pressure vessel codes, but rather to show what portions of existing codes are most appropriate for use in the design of a centrifugal pump case. This is done by examining first the general scope of necessary pump casing design work, and then identifying where application of a pressure vessel code is most fitting. For those items which have no appropriate design methodology published in a pressure vessel code, a set of principles are given which might be used stand-alone.

A BRIEF HISTORY OF ENGINEERING METHODS

We can consider the approach to design which is laid out in the body of this paper to be only that which is the most-recently available by modern understanding and technology. It is unlikely that this method will prove to be the final end of the spectrum of engineering methods. However, by glancing to the past, we can gain insight into the usefulness of the modern approach as well as see where we are still tied to schemes of antiquity.

Design-by-Precedence (DbP)

Modern engineering has its roots not only in the awe-inspiring construction efforts of the Pyramids of Egypt, China's Great Wall, and the Roman Aqueducts, but also in the countless projects of minor scale which are scattered about even the not-too-ancient history. Long before the engineering phenomena of creep and fatigue were identified, people have been building things. Whether constructed of wood, stone, or metal, things which were built sufficiently robust would last and could serve as templates for future construction. Ancient engineering efforts were not purely trial and error, but in many circumstances a study of a system or structure was considered unnecessary because a suitably similar 'qualified' design had already been observed.

Advantages of DbP are relative inexpensiveness and the lack of a need for deep understanding of the system being designed. A major disadvantage is the risk of copying a design from outside the limits of applicability. One noteworthy example the authors had the opportunity to investigate involved a centrifugal pump which was dimensionally identical to a machine previously built by the same manufacturer. The copied design had been performing quite successfully for years; however, the new machine began leaking virtually immediately after installation. It was ultimately discovered that the materials of construction of the new pump were different from the copied design. Even though the new strength properties of the changed materials were considered in a set of repeated design calculations, no examination of the effects of changed *thermal expansion rates* had been made, and higher temperature operation resulted in reduced preload on the main bolting. Only a more rigorous evaluation method revealed the problem and allowed a recommended solution to be provided.

Clearly when applying DbP, care must be taken to ensure that critical details are not missed. This is particularly important when considering any proposed modifications (e.g. new materials, scaled up dimensions, etc.) Also, if a copied design hasn't failed, it might only be because it hasn't failed *yet*. Progression of designs as a whole can be very slow, as the boundary of possibility must be expanded gradually with this method. Some failures are inherently unavoidable when using only DbP as the real boundary is discovered and occasionally exceeded.

Design by Rule (DbR)

As understanding of physical phenomena increased, DbR methods became practical. These allowed quantification of a design's suitability using rudimentary equations, the forms of which were adapted from known physical laws. While a bridge produced using DbP might have an efficient shape, one using DbR could be substantially scaled in size or load capacity while still having confidence in the integrity of the design. DbR has proved highly valuable in the development of various standards, including the ASME Boiler and Pressure Vessel Code, portions of which remain DbR.

A major advantage of DbR methods is their simplicity. The individual calculation steps performed – while sometimes numerous and tedious – are often not very complex. For basic components and uncomplicated structures, this method may be perfectly suitable. It might also be used as a basis for initial sizing of a design before refinement is performed using the more comprehensive method described below. However, while it is at times conservative, the DbR method can actually hinder understanding of a system because the principles upon which the rules are based may not be clearly identified. Indeed, as is the case with some codes and standards using DbR, the underlying principles are no more identifiable even by those publishing them. This becomes especially troublesome when seeking to perform an evaluation which is simply not supported by the scope of available DbR methods. Subjective interpretations and assumptions may then be needed.

Design by Analysis (DbA)

The DbA methodology represents a thorough and precise effort to describe what is physically happening to the system under investigation. The process requires the development of a conceptual model which is an abstraction of the physical reality. This model must include all relevant details that produce effects of significant magnitude on the results of interest. A mathematical model is then formed from this, and the ensuing mathematics produce an end result which is checked against an allowable value or acceptance criteria.

A successfully executed DbA process requires strong knowledge of the phenomena at work in the system. Experienced engineers are needed to ensure that the abstract models are sufficiently complete. The mathematics produced from such models is often of a scale requiring powerful computing hardware and software. On the positive side, with its precise nature and having a scope limited only by the knowledge of the engineer using it, DbA gives the best opportunity for creating maximally-optimized designs.

A brief note concerning the design methodology endorsed by ISO 13709 (2009, formerly API 610) is relevant here, as many centrifugal pumps are produced using this specification. Historically, this standard pointed to the use of the ASME code and its DbR methods. Even as the ASME code expanded to include DbA, ISO 13709 listed the DbR sections to be the basis of design. The most recent edition of ISO 13709 now has this particular reference removed, and in its place remains only a defined material design stress. Some have interpreted this to mean that these design stress values are to be incorporated into any modern design process, including the DbA method. In principle this is possible; however, the authors recommend against this. A true DbA process includes a rigorous abstract physical model with all accompanying details to support evaluation of the item of interest. Conservatism is applied where necessary, usually in multiple stages of the analysis. Overriding a DbA method by specifying only the design stress value actually provides incentive for a manufacturer to use a process with the greatest conservatism weighted in the material definition, and little elsewhere. Some pump users may be unsure of the loading applied to their equipment, perhaps because of the level of control or monitoring applied in the process. Consequently, they might seek what is often seen as a more conservative overall approach in the DbR methodology. Where necessary, such an approach may be used as a fall-back, but it is much more desirable to perform complete DbA evaluations. This provides detailed answers to specific concerns that the DbR method cannot. As further described in this manuscript, evaluation strategies are available for handling large variability in design input, which may mitigate the apprehension of moving away from historical DbR methods.

The primary tool of DbA of mechanical structures is the finite-element method. The principles hereafter described are intended for use in developing rigorous finite-element models. It is assumed that the reader is already familiar with the basics of good finite-element practice. Ultimately, the appropriate method of design must be chosen for the task at hand. Constraints of time or resources in workflow tend to push engineers toward the use of DbR or even DbP; however, the benefits of DbA become most apparent when it is fully embraced and the engineering workflow has been streamlined to support it. In particular, the evaluations which have the greatest impact on the overall design should be placed at the front of the process and performed as early as possible to ensure the highest-quality overall outcome, using less development time and fewer design iterations. The table described in the next section is recommended as the framework for organizing a product development / design workflow using the DbA methodology.

THE FAILURE MODE TABLE

The DbA method involves the creation of a conceptual model to support evaluation of a particular result or set of results. It is therefore advantageous to have the desired results arranged in a way that they can be seen at an overview, and to understand them in their appropriate context. This is accomplished by having each result of interest described as a failure mode in one row of a table, hereafter referred to as a Cause-Effect-Consequence (CEC) table. The CEC table has three major columns for describing the failure mode, with each column having a particular use to the people examining the integrity of the machine.

Cause

This is the underlying physical mechanism which triggers the failure. It is the task of the pump's designer to prove that the machine will not succumb to this failure mechanism by means of an engineering evaluation. Typically, there will be a one-to-one correspondence between failure mechanisms and evaluations, though an evaluation might be composed of multiple steps.

Effect

This is the directly-observable result of a failure. This item will perhaps be of greatest interest to those investigating a failed machine, since it provides a link from the failure's apparent origin right into the already-executed design methodology. All evaluated causes in the CEC table which point to the observed failure effect can be checked for errors or oversights, while unlisted causes having this effect can immediately be recognized as not having been formally evaluated during the design process.

Consequence(s)

This is the immediate downstream outcome (still related to the machine) produced by the failure. This is useful as a jumping-off point for deeper analyses of the entire system in which the pump operates.

As an example, consider the failure mode with its identified cause and a consequence shown in Table 1.

Table 1: Example Failure Mode

Cause	Effect	Consequence(s)
Fatigue stress	Cracked case wall	Leakage of working fluid

The machine's design engineer would be responsible for the evaluation of fatigue stress, using an appropriate methodology and acceptance criteria. A field engineer who observes a crack in a customer's pump casing could consult the pump's design documentation to see that a fatigue evaluation was performed; they could then investigate whether it was performed correctly or whether the actual loads the pump experienced exceeded those used in the theoretical analysis. A HAZOP team could follow the consequence of product leakage through possible failure cascades such as fluid overflow, oxidation, electrical shortage, and so forth, all the while having knowledge of the underlying cause and perhaps its likelihood.

A comprehensive DbA of a centrifugal pump can therefore be broken down as a list of failure modes and mechanisms requiring evaluation in a Cause-Effect-Consequence table. Having such an overview of all items to be investigated allows for planning the scope of the abstract models used to perform the evaluations. For instance, a single finite-element model might be capable of producing stresses for evaluating fatigue life as well as deflections for checking the fit of assembled parts.

EVALUATION STRATEGIES

Every engineering analysis must be conducted according to a governing strategy. This ensures relevance of the evaluation in a real-world context. The following are three of the most common evaluation strategies applied in general mechanical engineering analysis.

Nominal with Safety Factor

The simplest approach uses nominal or average values to describe the system's model. Such an approach is ideal when variability is tight and when the inputs have the most proportional correlation to the resulting output. With this strategy, the applied safety factor is used not only to provide margin between the real loaded system conditions and failure, but also to cover things which are not explicitly quantified in the analysis. Of course, 'non-quantified' does not necessarily mean 'unknown'.

Compound Worst-Case

Where system variability cannot be ignored or assumed to be adequately covered by an applied safety factor, it is appropriate to have the variability taken into account in the values describing the system model itself. For example, the manufacturing tolerances involved in producing a casting will result in parts with wall thicknesses spread over a particular range. With the 'compound worst-case' strategy, one of the two extremes of this range of thicknesses will be used, depending on which produces the most conservative result. For a strength evaluation with internal pressure, the minimum wall thickness would likely be used. For an evaluation of casing distortion through a thermal transient event, the maximum wall thickness might be required.

A safety factor is still used with this approach, but the nature of the system's model means that the factor must primarily cover real margin from failure and not design variability. The major disadvantage to this strategy is the difficulty in determining the worst-case assumptions, which are not necessarily obvious. If the system contains a set of opposite-acting effects, then in some instances the worst-case could even be produced from values in the middle of the tolerance range. Where this is discovered, it is common to simultaneously apply both extreme values of the same variable depending on its effect in the calculation. For systems composed of several dimensions, the worst-case might therefore represent an extremely unlikely or even impossible scenario, and designs produced in this way might be substantially overbuilt.

Probabilistic

One possible method of reducing the number of engineering assumptions is the use of probabilistic calculations. The system's inputs are applied as random variables of an appropriate probability distribution. In the most common approach, referred to as Monte-Carlo simulation, each input is randomly assigned from its corresponding probability distribution and the overall result is evaluated. The process is repeated a large number of times using new random values for each input. Ultimately, the evaluation is based on whether a sufficient number of successful calculations were produced.

Such an outcome can generally be used to make a statement regarding the likelihood of the failure occurring, which is useful when failure risk needs to be quantified. In addition, any potential worst-case scenarios which arise from intermediate values of variable inputs are captured. Since the calculation must be performed many times in order for the total results to closely approximate the input probabilities, this strategy is impractical for models which are already computationally expensive.

FAILURE MODES OF PUMP CASINGS

A distinction can be made between two classifications of failure: execution failures, and design failures. Execution failures are those which are caused by 'not doing what was said would be done'. Considering a design as simply a set of conceptual instructions, design failures are those which would occur even when all instructions and procedures have been followed. Integrated within a manufacturer's process, the full table of failure modes for a product could include both design and execution failures. The scope of the following listed failure modes will be restricted in the following ways:

- Only design failures
- Only failures directly related to the pressurized pump casing
- Only failures which when evaluated lead predominantly to a change in part shape or dimension (e.g. Chemical suitability of materials is not discussed.)

It is not practical to reproduce here every detail of each recommended evaluation method, particularly when they are found in a published standard. The most important aspects of the analysis models are discussed, and it is the hope of the authors that the principles listed here will be sufficient in allowing the methods to be understood and followed.

Frame Structural Failures

The majority of centrifugal pump casing mechanical design is evaluation of its structural integrity. Careful examination of the defined load cases is needed to ensure that all appropriate effects have been included. For example, some pump casings experience an interesting phenomenon described as 'fluid-pressure-penetration' when loaded with internal pressure. Consider an axially-split pump as shown in Figure 1.

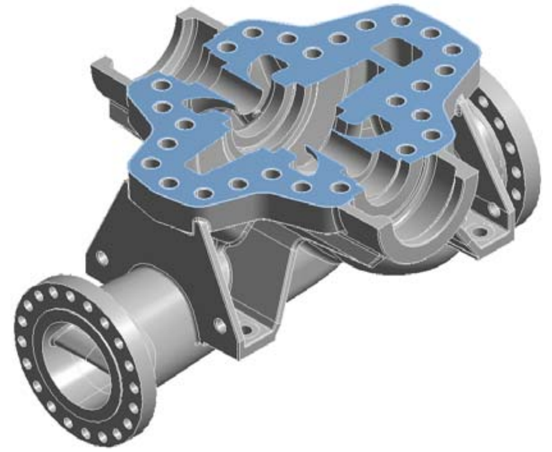

Figure 1: Axially-Split Pump with a Top Case-Half Removed Showing Gasket Subject to Fluid-Pressure-Penetration

Distortion of the case halves when pressurized will cause the inner edges of the gasket to become less loaded and expose a portion of the gasket to the working fluid. While the gasket continues to perform its intended function well (preventing both internal and external leakage) the area of the hydraulic load on the casing parts may have increased by a considerable amount (see Figure 2), affecting both the casing stress and bolt loads. These types of effects are important to consider when performing casing analyses.

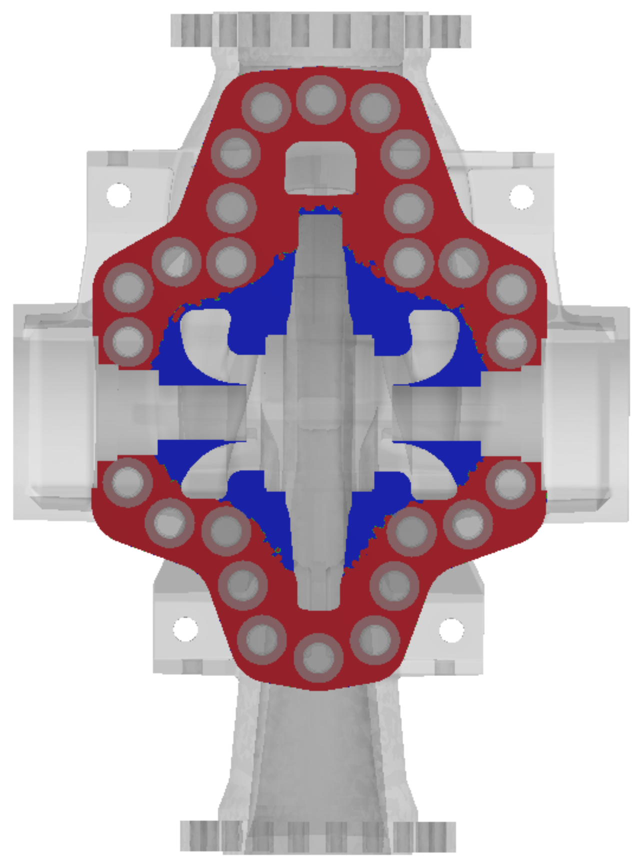

Figure 2: Pressurized Gasket Condition (Red Indicates Dry Contact, Blue Indicates a Possible Wetted Gasket Surface)

While probabilistic methods are not recommended for these frame structural failure modes, the models to be used contain a surprising number of input values with large variability (e.g. bolt pretension) which must be considered. Such effects are even larger when considered in combination. In the previous example, a low bolt pretension will contribute to an even larger hydraulically-loaded area when fluid-pressure-penetration is applied.

Cause: Static Plastic Collapse
Effect: Pump frame rupture
Possible consequences: Leakage of working fluid; toppling of supported equipment; detachment of pump frame / foundation
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Limit load (ASME 2011 VIII-2 5.2.3, or EN13445-3 2012 B.8.2)

Plastic collapse is one of the most significant failure modes evaluated in all of general machine design. It refers to the condition of a component being unable to sustain an applied load, resulting in gross distortion. It is important to note that safety against plastic collapse does not mean safety against plastic deformation. With pressure as an applied load, tank-like structures often develop high stress in local regions, especially near locations of geometric discontinuity.

When yielding takes place at regions of high stress, the load path will be forced to re-distribute itself from the highly-stressed material to that which is surrounding it. Plastic collapse will not occur as long as a structure has sufficient supporting material, though some residual plastic deformation is expected because yielding has occurred. The images shown in Figure 3 through Figure 6 illustrate the internal stress condition of a part up to plastic collapse. Up to the stress condition shown in Figure 4, the part remains elastic and will return to its original shape if the load were removed. The intermediate condition in Figure 5 is likely allowable when considering safety against plastic collapse, as the structure remains capable of sustaining the load; however, due to the significant yielded areas the part is expected to be permanently deformed once the load is removed. The condition in Figure 6 is just prior to failure, with the zones of yielded material extending from both top and bottom surfaces to nearly reach in the middle. Obviously, it is desired to have some margin against the actual load which will cause failure by plastic collapse.

Figure 3: Notched Cantilevered Beam

Figure 4: Onset of Yielding with an Applied End Force

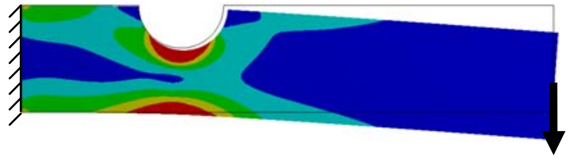

Figure 5: Yielded Region Expands with Increased Load

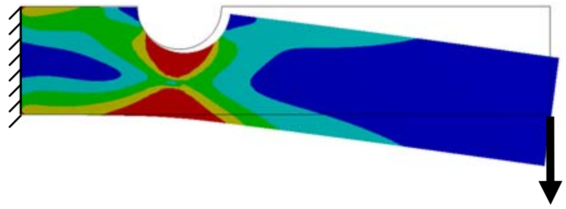

Figure 6: Imminent Plastic Collapse

Multiple options exist for evaluating plastic collapse. One commonly-encountered method in pressure vessel design involves the use of a linear-elastic material model and stress linearization and classification. Paths are drawn through the structure walls at locations of interest, and an idealized representation of the stress distribution is mathematically generated from the actual one calculated using finite-elements. The evaluation is then performed primarily on the idealized distributions themselves. By considering the idealized stress through the entire thickness, local regions with stress higher than yield (which can exist in a model using pure elastic material) might be allowed. For simply-shaped structures, the stress classification method can be an effective technique.

Unfortunately, this method generally also requires an accompanying set of rules regarding the allowable proximity of high stress resulting from structural discontinuities, which can often be impractical to fulfill for centrifugal pumps made of hydraulically-optimized shapes and efficient features like structural ribs. Also, as each evaluation path must be deliberately selected, the process involves some subjectivity, and it becomes increasingly cumbersome as the shape of the evaluated part becomes less like a thin-wall general pressure vessel and more like a centrifugal pump casing. Instead, the authors recommend using an evaluation method known as 'Limit-Load' analysis; both the ASME B&PVC and EN13445 recognize this method as satisfactory proof of safety against plastic collapse, and the codes differ only in the applied details.

The limit-load analysis method uses an elastic-plastic material model with an artificial yield point and no work hardening (i.e. ideal plasticity). When using ASME as the basis of the evaluation method, the geometry is to be modeled using the minimum material condition (including corrosion allowance and manufacturing tolerances). This is usually a simple matter when designing general pressure vessels of relatively thin walls, as such finite-element models can be created using shell elements which can quickly have their thickness redefined. When designing centrifugal pumps, however, complicated 3D CAD geometries are generally used to describe the casing. In practice, analyzing minimum material thickness parts would then mean maintaining multiple CAD geometry files for every design iteration: at least one 'thin' one for analysis, and another of 'real' thickness to support design documentation and other evaluations. The risk of divergence of such models in a design process is high enough to warrant the use of an alternative strategy.

One possible solution is to account for the lost material though the use of an amplified set of loads. (Note that the loading in a limit load analysis is already modified by factors according to the corresponding standard, to account for other variability.) Consider the factor shown in Equation (1). When loads are multiplied by this factor, the modeled system will produce an equivalent membrane stress to the original loads with minimum material condition.

$$f_t = \frac{t}{t - c - m} \qquad \text{Equation 1}$$

where f_t is the load amplification factor
t is the modeled design wall thickness
c is the corrosion allowance
m is the manufacturing tolerance

Because the model used in a limit load analysis has been constructed with a set of artificially modified parameters (yield strength, loads, etc.) not all results are representative of reality. In fact, the ASME method uses mere numerical convergence as an acceptance criterion, with all other results to be ignored. For the EN code, the criterion is an allowable absolute principle total mechanical (elastic + plastic) strain based on the particular load case being investigated.

The major advantage of the limit load analysis method is the unambiguousness of a successful result which can be seen in a single result plot. Unlike stress linearization, no subjective judgment is needed in choosing how or where to draw evaluation paths. This translates both to faster analyses and safer designs. It is therefore an effective evaluation method for proving protection against plastic collapse of all major pump casing parts, with only the bolting considered separately.

Cause: Local Failure
Effect: Pump frame rupture
Possible consequences: Leakage of working fluid; toppling of supported equipment; detachment of pump frame / foundation
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Local (sudden rupture) criteria (ASME 2011 VIII-2 5.3.2 or EN13445-3 2012 B.8.2)

Most material yielding or failure criteria have an equivalent stress as their basis. An example would be von Mises stress, which can be found from the local principal stresses as shown in Equation (2).

$$\sigma_{eqv} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \qquad \text{Equation 2}$$

where σ_{eqv} is von Mises stress
σ_1 is the first principal stress
σ_2 is the second principal stress
σ_3 is the third principal stress

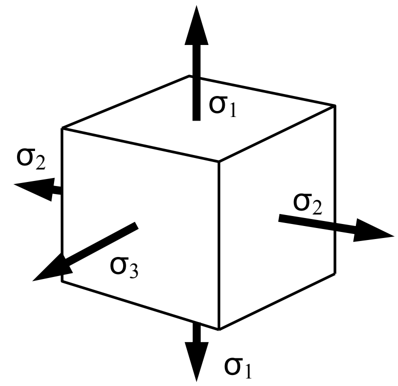

Figure 7: A Stress Element Loaded Uniformly in all Directions with Zero Equivalent (von Mises) Stress

As can be seen from the equation, when a material is uniformly loaded in all principal directions simultaneously (see Figure 7), the equivalent stress is zero. However, a fracture of the part would cause an immediate loss of load-carrying in the direction normal to the fracture plane, with the load in the remaining directions producing a non-zero equivalent stress. Thus, the underlying mechanism in an analysis of local failure is a sudden rupture due to what could be considered an instable stress state. Safety against this failure can be shown in multiple ways; the simplest of which is by limiting the value of the sum of all principal stresses throughout the part being investigated. The use of either ASME or EN13445 is recommended for evaluation of this failure mode.

Cause: Ratcheting
Effect: Pump frame rupture
Possible consequences: Leakage of working fluid; toppling of supported equipment; detachment of pump frame / foundation
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Shakedown criteria (ASME 2011 VIII-2 5.5.6 or EN13445-3 2012 B.8.3)

As seen in the discussion of plastic collapse, pressure-retaining parts may experience some amount of plastic deformation while still being capable of functioning satisfactorily. In locations of high stress, this plastic behavior generally results in some residual stresses being present once the applied loads have been removed. Such residual stresses, of sign opposite those which caused the yielding, reduce the capacity of the part to handle reversed-loading. A part subject to cyclic loading may thus be at risk for a sort of continuous plastic deformation, a phenomenon known as 'ratcheting'. In extreme cases the stress present during initial loading causes such great plastic deformation that the residual stresses developed upon relaxation of the load are themselves enough to result in yielding in the opposite direction.

This scenario can be contrasted with one where after the initial plastic deformation occurs, all subsequent load cases (including various states of partial-load or non-load) do not cause any stresses above yield in any direction. Such a machine would thereafter operate with its materials of construction entirely in an elastic range.

In centrifugal pump casings, the performed factory hydrostatic testing is generally the primary source of this initial induced plasticity. Following this, ratcheting behavior might more commonly result as a consequence of thermal effects or loading not related to the application of internal pressure. Both ASME and EN13445 provide evaluation methods for determining safety against ratcheting.

Cause: Fatigue
Effect: Case rupture
Possible Consequences: Leakage of working fluid; toppling of supported equipment; detachment of pump frame / foundation
Typical modeling strategy: Compound worst-case geometry, nominal loading
Recommended evaluation method: Cumulative damage (ASME 2011 VIII-2 5.5.3; 5.5.5 or EN13445-3 2012 B.8.5; 18)

Cyclic loading can cause progressive damage and ultimately failure of a part, even if its effects are not clearly visible throughout the process as might be the case with the plastic deformation which occurs during ratcheting. Centrifugal pumps are commonly exposed to cyclic loading, particularly of pressure and temperature.

Thermal fatigue is significant in centrifugal pump casings, with warm-up cycles generally desired as short as possible, and the thick walls of high-pressure cases being subject to steep temperature gradients as a result. The highest thermally-induced stresses will typically be found on the inner surfaces of the casing. The high velocities of the working fluid will develop large effective convective heat transfer coefficients at these surfaces. In most examinations of rapid warm-up or upset transient loadings, the highest stress occurs within a few minutes of reaching the new conditions. The thermal gradient is typically a shallow slope through the majority of the wall thickness, transitioning to a much steeper slope upon nearing the surface exposed to the transient thermal load. The thickness of this steep-gradient portion is typically about 10 percent of the full wall thickness when the worst-case stress is developed.

An example of the thermal distributions seen in a rapid warm-up scenario of a barrel-type pump casing is shown in Figure 8 through Figure 11.

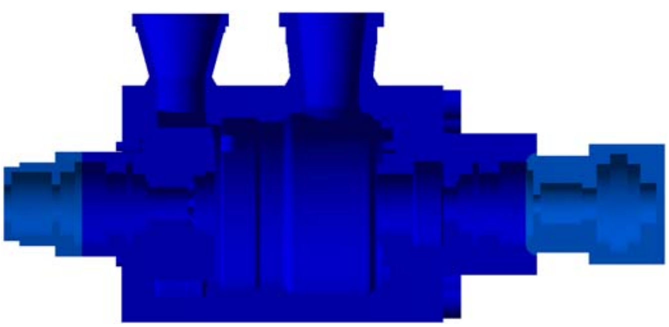

Figure 8: Temperature Distribution of a Pump Barrel Casing Subjected to Thermal Shock (0 seconds, warm stand-by)

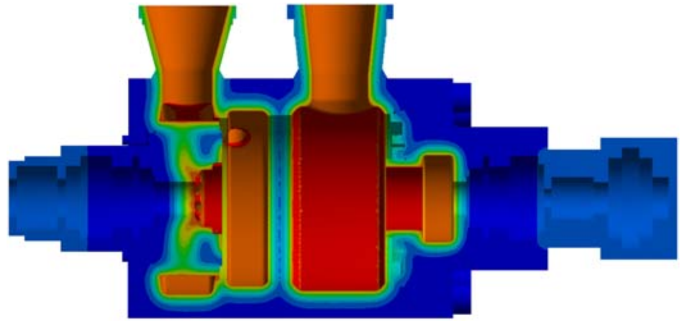

Figure 9: Temperature Distribution of a Pump Barrel Casing Subjected to Thermal Shock (30 seconds)

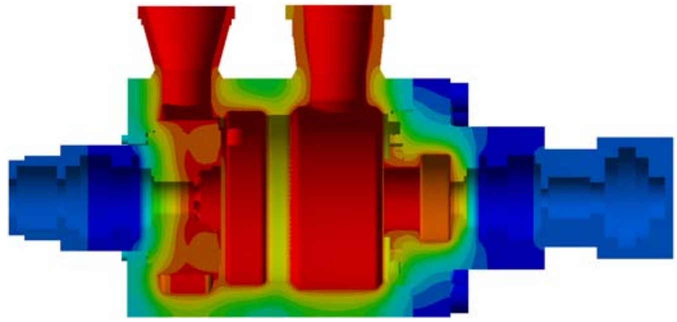

Figure 10: Temperature Distribution of a Pump Barrel Casing Subjected to Thermal Shock (360 seconds)

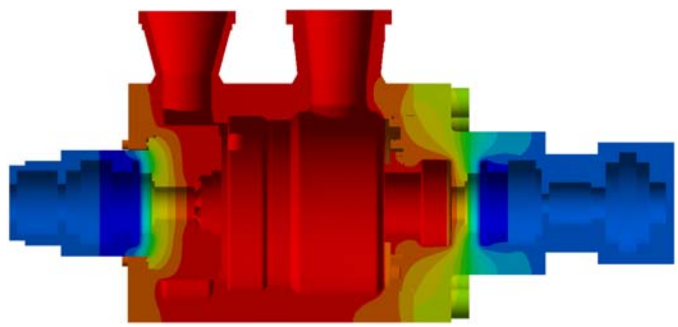

Figure 11: Temperature Distribution of a Pump Barrel Casing Subjected to Thermal Shock (3600 seconds, near steady-state)

Some pump users have processes which at times subject the system to a downward thermal transient event. During these events, equipment operating at steady-state high temperatures is exposed to working fluid of decreasing temperature. For centrifugal pumps this can be particularly troublesome as the casing is generally already in a state of tensile stress due to internal pressure. The worst stress caused by thermal strain is now found in the cool layer of shrinking material on the inner surface of the case, while the bulk of the case is still warm or hot. This means that this high tensile thermal stress is directly combined with the tensile pressure-induced stress. Also, since these downward transient events typically occur in addition to normal upward transients, the range of stress experienced by the parts is substantially increased. For casings clad with internal weld overlays, the fatigue mechanism is also a concern for de-lamination of the materials.

The recommended approach to evaluate fatigue is through the use of cumulative damage, with a linear-elastic material model. For most other structural failure modes, the entire set of design load cases can be considered independently; as some load cases can be recognized as subsets of stronger load cases, the weaker ones can even be conservatively ignored. This is not so for fatigue, where all load cases have the possibility to contribute to failure.

A fatigue evaluation begins by clearing the minor mental hurdle of switching from a defined set of load cases to a set of events. Each event represents a transition from one beginning state or condition of the machine through to a final state. (If the event actually describes a complete cycle, the initial state would also be the final one.) An example would be the start-up event which begins with the equipment at rest, and ends with the machine loaded to the normal operating load case. If the loads are known to be monotonically increasing or decreasing, it might be assumed that the extreme stress values throughout the event are found only at the beginning and end, and therefore only these two must be calculated. Otherwise, as is the case with thermal transient events, a discrete set of intermediate structural load cases are calculated using any corresponding thermal distributions.

The cumulative damage method is based on the notion that the peaks of the stress range throughout the event allows determination of how many such events the machine could withstand until the onset of failure. It is then assumed that the inverse of this number gives the effective portion of fatigue life consumed by that event. Multiplying this by the number of expected occurrences of this event gives the damage for that event, and this can be summed with the damage for all other events to find the total damage. Parts which have total damage values above one are thus expected to fail at some point through this set of events over the course of their life.

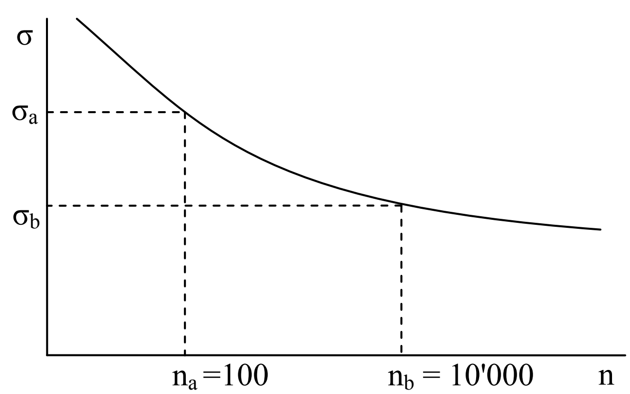

Figure 12: Example Cumulative Damage Fatigue Evaluation of Two Events ('a' and 'b') Using a Fatigue Strength Curve

An example of this can be seen in Figure 12. There, event 'a' produces a large stress with relatively few corresponding allowable cycles, and event 'b' produces a smaller stress with a higher number of allowable cycles. If event 'a' were actually expected to occur 20 times through the life of the machine, and event 'b' were expected 5'000 times, the total damage would be calculated as shown in Equation (3).

$$D = \sum \frac{E_i}{n_i} = \frac{E_a}{n_a} + \frac{E_b}{n_b} = \frac{20}{100} + \frac{5'000}{10'000} = 0.2 + 0.5 = 0.7 \qquad \text{Equation 3}$$

where D is total damage (< 1.0 for a successful design)
E_i is the number of expected occurrences of event 'i'
n_i is the allowable number of cycles before fatigue failure corresponding to the stress experienced during event 'i'

Note that while each occurrence of event 'a' produces the highest stress, the most damage is actually done by the large accumulation of occurrences of event 'b'. In this example, the total damage is 0.7, and thus the design would be considered safe from fatigue failure.

While the method of cumulative damage has proven very effective, there are two items which make its implementation challenging. First, when considering an event composed of several discrete load steps, it is not normally obvious which two-step combination will produce the worst-case stress range, as the direction of the local stress tensor is important. Secondly, the two steps appropriate for one location in a part may not be representative of the worst-case stress range throughout the entire part. While the worst-case total damage of all events can be conservatively covered by the sum of all maximum event damage values (even if these change location), a more accurate appraisal can be made by summing the individual event damage values together based on location. To use the cumulative damage evaluation method with a large finite-element model therefore requires some programmatic assistance.

It is recommended that the evaluation methodology (and all associated mathematics) of the chosen cumulative damage method (whether EN13445, ASME, or another) be used in the form of a validated software module so that the choosing of load step combinations and summing of event damage is performed rigorously. In case the fatigue assessment is instead done manually, strict attention must be given to the assumptions applied during result evaluation.

Additional evaluation work must be done to confirm the acceptability of any weld joints within the structure. Both major codes recognize this in their DbA requirements.

Cause: Creep
Effect: Structural collapse or fracture
Possible Consequences: Leakage of working fluid, toppling of supported equipment, detachment of pump frame / foundation.
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Set of creep analyses (EN13445-3 2012 B.9)

Creep, which is the phenomena of time- and temperature-dependent material flow behavior, is not often encountered in centrifugal pump designs, as most machines operate well below the temperatures required for creep to have a measurable effect. A general rule-of-thumb is to ignore creep below temperatures of roughly 720°F (380°C) for ferritic steels and 930°F (500°C) for austenitic stainless steels.

For machines determined to be susceptible to it, creep is not evaluated entirely on its own. Rather, creep is used as a modeled phenomenon within the evaluations of creep failure modes of which the previously-described 'plastic collapse' and 'fatigue' failure modes are analogues. In addition, creep may be a necessary effect in deformation evaluations described later. Where evaluation of creep behavior is necessary, the EN13445 standard is the recommended basis, as it is more complete than other codes. Even so, usually the most difficult part of performing creep analyses is obtaining an appropriate set of material creep properties at the required temperatures. The magnitude of assumptions and effort needed to produce such values may be considerable.

Cause: Buckling
Effect: Structural collapse
Possible Consequences: Leakage of working fluid, toppling of supported equipment, complete distortion of the structure
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Buckling analysis (ASME 2011 VIII-2 5.4 or EN13445-3 2012 B.8.4)

Buckling refers to structural instability under compressive loading. Because most centrifugal pump casings are subjected to loads which produce almost all tensile stresses, relatively few are susceptible to buckling as a failure mode. Some notable exceptions include externally-pressurized containers such as those used in subsea applications, or those which may have to support other equipment as with a motor atop a vertical pump structure (see Figure 13).

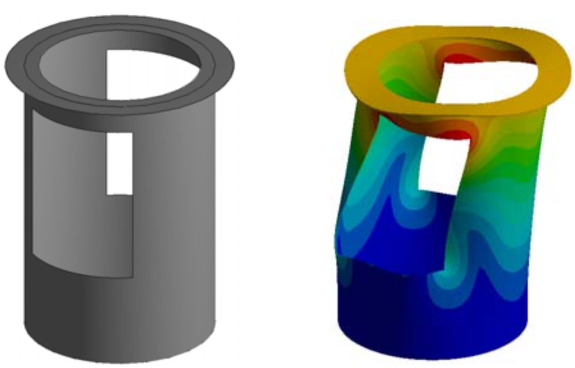

Figure 13: Vertical Pump Motor Support and Deformation Plot of a Buckled Shape

When buckling cannot clearly be excluded as a possible failure mode for a centrifugal pump, it is recommended to be checked using the methods of either ASME or EN13445. As Figure 13 shows, the buckling load can be influenced by the presence of local features such as corners and flanged connections; thus, a greatly simplified approach using equivalent cross-sections of basic geometry and hand equations might require substantial conservatism to avoid exceeding the actual buckling limit. Instead, the use of DbA methods with a full finite-element model produces the most accurate evaluation of the structure.

Flange Joint Failures

Most flanged connections do not need to be evaluated using DbA. Where standard flange designs are encountered, these are often considered proven by experience (DbP). For example, the ASME code considers the ANSI B16.5 flanges acceptable in this way. Interestingly, if such standard flanges are checked against their design limits using ASME DbR methods, they may not successfully pass! Discussion over whether the standard flanges are actually inadequate or the DbR methods are too conservative are irrelevant. If clear, specific details of the flange's robustness are needed, DbA should instead be applied.

When DbA is used, the flange can simply be considered an extension of the rest of the casing itself, subject to all other failure mechanisms listed in this manuscript. In case the flanges are standard and it is chosen to assume they are proven by DbP, then the casing evaluation criteria only need be applied up to the flange hubs.

Bolting Failures

Most pressure vessel codes have very little to say regarding the various failure mechanisms which can occur in association with bolted joints. Instead, focus is largely given to the normal stresses in the cross-section of the fastener, and for good reason. For the most-encountered 'vanilla' joints, the bolt cross-section is a major factor in determining the dimensions of a design. In the past, evaluation of this stress while establishing the rest of a part's size helped to produce an efficient workflow. In addition, bolted joints designed with standard hardware (e.g. heavy-hex nuts made of a companion material to the bolt's material) can have multiple failure mechanisms conservatively covered by the evaluation of the bolt cross-section itself, since this is the limiting safety factor. A design for which a working load will cause the bolt to fail in normal stress before any other locations do is desirable not only for the effort it saves in evaluation, but because a static overload failure will most likely be detected at the time of assembly and factory testing, lessening the risk of encountering an issue in the field.

Bolt stress evaluations come in many flavors. Not all of them necessarily advance the collective understanding of the physics in a bolted joint. For example, many DbR methods can be performed without ever encountering a value for the bolt preload. Instead, the joint geometry is used to scale and transfer any environment loads to the bolting, where they are used with the bolt cross-section area to calculate a stress value which is then compared to a code allowable. Because the method involves the use of an 'allowable stress', many designers may assume that the bolt will never be subjected to stress levels above these, but this is not true. The result of such a DbR method is not 'real', even though the results have units of stress. By ignoring preload and simply comparing to a small fraction of the bolt strength as an allowable, the routine becomes easier to implement. A full analysis of the joint takes considerable effort, but it produces more results, and they are of a type which is directly comparable to the actual allowable stresses of the materials.

A finite-element simulation of a bolted joint normally involves these steps:

- Create all the joint parts (bolts or studs, nuts, and clamped members) as separate pieces.
- Apply contact as needed between the parts. Attention must be given to the treatment of threaded components; as the direction of interaction forces on thread flanks can have a significant impact on calculation results, including bending stresses of the parts (see Figure 14).
- Introduce an initial load case where the applied pretension forces are used to determine what amount of bolt strain is expected during assembly.
- Hold the initial bolt strains constant throughout the application of any subsequent load cases.
- Where the results can be assumed to be sufficiently smeared out, (e.g. bearing pressure under a washer) the calculated forces can be extracted from the finite-element model and used with classic stress equations for evaluation.

In this way all results of interest can be investigated, and the joint can be proven secure by a thorough evaluation of the following items.

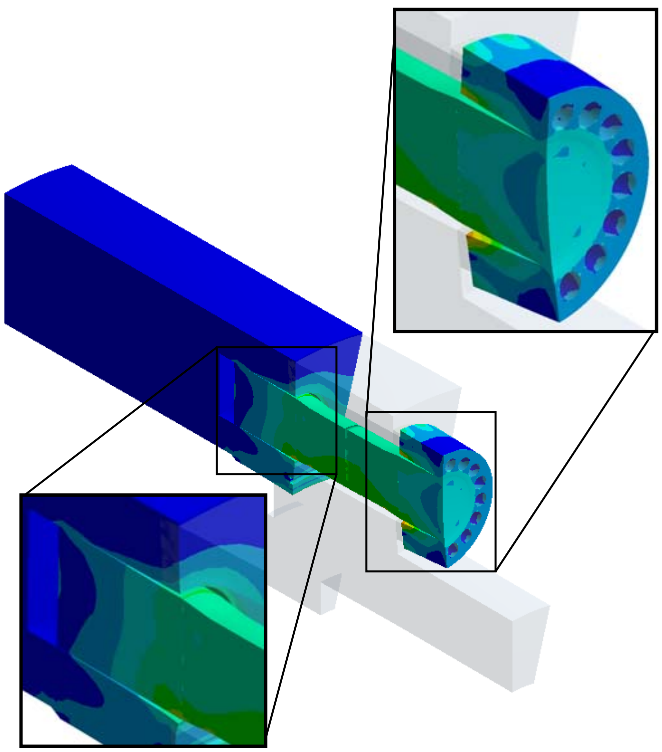

Figure 14: Barrel Cover Joint Deflection Plot Showing Radial Expansion of Nut and Case at Threaded Interface (Interaction Behavior Accurately Modeled without Individual Threads)

Cause: Plastic Bolt Elongation
Effect: Bolt yielding
Possible Consequences: Broken bolting, loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, producing extracted forces for evaluation
Recommended evaluation method: 'Smeared' equivalent stress

In general pressure vessel design, bolts are often subjected to conditions which are difficult to define but which clearly put the parts at risk of damage. For instance, time-sensitive maintenance operations are sometimes performed with whatever tools are close at hand. Add a few years worth of rust, and it is not uncommon to discover in practice some loosening method involving wrenches, large hammers, and possibly a torch. While such actions are never intended by a manufacturer, it is appropriate to recognize their possibility. Particularly for smaller fasteners such things can be a significant concern, because the applied forces are so large relative to the design loads.

It is not surprising then, that some codes apply more stringent allowables for the nominal bolting loads. An example would be the ASME 2011 VIII-2 part 5 limits, which reduce to the following for an application at room temperature:

- Maximum bolt stress (*neglecting* bending) of 66.6 percent of bolt yield strength
- Maximum bolt stress (*including* bending) of 100 percent of bolt yield strength

While bending can be a considerable portion of the bolt stress, the restriction on the bolt stress without bending is a substantial one. For many years, common industrial machine design practice for bolted joints has involved *minimum* preload values of 75 percent of the fastener proof load, and the maximum might be as high as 100 percent. For typical bolt materials, with proof strength approximately 90 percent of yield strength, this means that standard machine design practice for initial preload is already outside the allowable ASME limit for operation service stress values!

It may well be that many bolted joints used in general pressure vessel design are of a type that requires substantial load margin to withstand some form of abuse outside the scope of what it was designed for. Application of this limit might also be a trivial matter given the size of the fasteners. For example: if a joint normally designed using an allowable service bolt stress of 85 percent of yield strength must now be made to meet the 66.6 percent limit given by ASME, the result is a 13 percent increase in the size of the fastener nominal diameter. In small bolts, this likely corresponds to a step from one standard size to the next standard size. Such a change is likely easily fit to the current design without significant geometry modifications or much additional material used.

However, what about highly-engineered connections in locations of limited space? These often utilize sophisticated tightening methods and custom-built components in order to achieve their design goals. At typical sizes of over 2 inches (50 millimeters) they are not at risk of damage from an overly-zealous mechanic with a hand-wrench. The space afforded by smaller bolts and any accompanying smaller washers and nuts can often result in a substantially more efficient design. And because the DbA methodology is applied with the opportunity to rigorously check all major failure modes related to the bolted joint (not only the cross-section stress), such a design can be confidently built essentially without compromise.

The bolt stress limits of ASME or EN13445 might therefore be used generally to support evaluation of plastic bolt elongation. In locations where the overall design might be improved by the application of a higher allowable bolt stress and there is no recognizable reason against doing so, it is recommended to apply one which is appropriate for the load case under evaluation using DbA.

Cause: Bolt Fatigue
Effect: Bolt fracture
Possible Consequences: Loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: Cumulative damage

Fatigue of bolts is best evaluated using the same underlying philosophy of the cumulative damage method described previously for casing fatigue. As efficient finite-element models for centrifugal pump design often do not include sufficiently-dense meshes for direct stress evaluation of bolts, however, an alternative strategy must be used. One option is to combine the effect of forces and moments over the time-history using basic stress equations. Just as with fatigue analysis of the casing parts, this combination must be done carefully, or with the assistance of a programmed algorithm.

Cause: Shear of Bolt Thread
Effect: Bolt thread stripping
Possible Consequences: Loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: 'Smeared' equivalent stress

Designs using threaded fasteners tend to use stronger (perhaps significantly so) bolt material than is used for the clamped joint. Consequently, while the shear area associated with the bolt threads is actually less than that for the mating female threads, gross shearing of the bolt threads is typically unlikely to occur. Even with very low bolt forces, some highly localized plastic deformation on the threads will be unavoidable, but the deformed shape of either threaded part will not be so much to prevent assembly / disassembly.

Consequently, a conservative allowable can be easily applied for this failure mode. It should be compared to a 'smeared' thread shear stress obtained from the same bolting forces extracted for plastic bolt elongation evaluation.

Three additional items are important to note:

- The flank angle of threads creates significant radial forces when the interface between threaded components is loaded in the axial direction. This can produce opposing radial deflection of the parts which acts toward disengagement of the threads, and causing a reduction in the effective shear area.. (i.e. the threads become more loaded toward their tips. See Figure 14 and Figure 15) This radial deformation must therefore be limited for the applied shear criterion to be relevant.

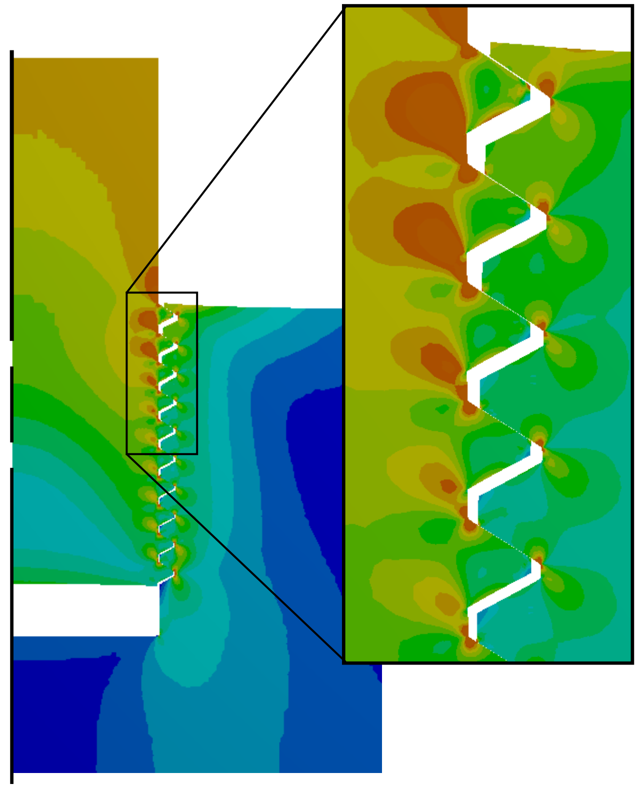

Figure 15: Fastener Thread Deformation under Load

- The distribution of the axial load along an engaged thread length is known to be biased toward the 'first' threads (i.e. those nearest the tensioned fastener body midsection.) Thus, while the shear capacity of a threaded connection can be generally improved by increasing the engaged length, there is a practical limit after which the deeper threads are not adding enough support to prevent the first threads from shearing out. Thus, an independent thread depth limit is also needed. A value of one-and-one-half times the bolt diameter is a commonly cited maximum. When a calculation shows that a thread depth greater than this is needed to prevent shearing, it indicates that something else must be changed in the design.

- Application of compound worst-case conditions typically includes the highest bolt load expected for the defined tightening method, as well as minimum thread engagement length based on stack-up of manufacturing tolerances.

Cause: Shear of Casing Thread
Effect: Case thread stripping
Possible Consequences: Loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: 'Smeared' equivalent stress

As described previously, the shear of casing threads is usually more critical than that of the mating bolt threads. It should be evaluated in the same way, using basic stress equations with the internal member forces extracted from the finite element model.

Cause: Shear of Nut Thread
Effect: Nut thread stripping
Possible Consequences: Loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: 'Smeared' equivalent stress

The shear of nut threads is evaluated in the same way as the bolt or casing threads. Due to its geometry, the nut is generally softer in resistance to radial expansion than the casing is. The radial deformation limit is therefore more applicable for this part. Otherwise, the evaluation is virtually identical.

Cause: Nut Body Fracture
Effect: Nut rupture
Possible Consequences: Loose parts, leakage of working fluid, toppling of supported equipment
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: 'Smeared' equivalent stress

As a result of standard dimensioned hardware, failure of the bodies of nuts is less common than that of thread stripping. However, some centrifugal pump designs require the use of substantial bolt loads to remain tight and structurally sound. The use of heavy-hex nuts would not allow for the close proximity of bolts needed to produce such high loads. Often, capnuts with cylindrical bodies and a torque-tightening feature on top are used to decrease the distance between bolt centers and increase the number of bolts applied in a limited space. If the body of this nut is too thin, it might fail from a combination of axial compression and hoop stress caused by radial expansion. Of course, investigation of this item is outside the scope of general pressure vessel codes, but it must be checked for a thorough evaluation of a properly engineered bolted joint.

An equivalent stress can be found from the extracted axial force and its corresponding radial force. An appropriate safety factor for each load case then ensures margin against rupture.

Cause: Bearing Pressure
Effect: Embedment of joint parts
Possible Consequences: Reduction of preload, leakage of working fluid
Typical modeling strategy: Compound worst-case, forces extracted for separate evaluation
Recommended evaluation method: 'Smeared' stress

High-tension bolted joints have parts heavily loaded in compression as well. The clamped members might have a narrow contact interface (e.g. a 'raised-face' flange) which results in large compressive stresses even with only moderate bolt loads. The material under the nuts or washers is often also subjected to high compression loads. The evaluation of these areas is performed like most items related to bolted joints: forces extracted from the finite-element model are used with basic equations to determine the contact stress. However, the safety factor used at these locations is often more relaxed from that used in evaluating bolt cross-sections or thread shear.

While a tensile-test specimen experiences necking as it is loaded beyond the elastic limit of the material, most locations of high bolting bearing pressure do not show a change to the loaded cross-section, even as the weaker material yields. The contact area may even increase. Some local plastic deformation is expected, of course, but the redistribution of the load and any work-hardening of underlying material means that loads at least up to yield are generally sustainable for this failure mode. It is not uncommon to see centrifugal pumps in practice have exactly these high bearing pressures allowed.

Tightness Failures

Confirming tightness is an important part of centrifugal pump casing design. Gaskets sealing to ambient environment prevent exposure of working fluid. Internal seals must also remain tight if machine efficiency is to be maintained.

Cause: Insufficient Gasket Seating
Effect: Gasket leakage
Possible Consequences: Loss of working fluid, reduction of efficiency, reduction in developed head
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Gasket seating stress criteria

Gaskets are not strictly necessary to avoid leakage between two objects. If two mating parts are sufficiently flat and smooth, they might be pressed together tightly enough so as to prevent fluid from escaping between them. In practice, however, the finished forms of real manufactured parts are never so ideal, especially under load. Even the small imperfections represented by surface roughness are enough to allow most pressurized fluid a way out. The job of the gasket, then, is to match and fill all imperfections by being plastically deformed under some initial load. This initial load, producing what is called 'seating stress', must be large enough to cause the gasket to deform and take the joint surface's imperfect shape.

Common generic seating stress values can be found in both ASME and EN13445. When in doubt, the gasket manufacturer can supply the required seating stress. The evaluation is performed by extracting gasket pressure from the model under the initial assembly load case.

Cause: Insufficient Gasket Working Load
Effect: Gasket leakage
Possible Consequences: Loss of working fluid, reduction of efficiency, reduction in developed head
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Gasket working factor

Non-self-energizing gaskets have surface loads resulting from the forced assembly of the joint. During operation, with applied external loads and deformation of the joint, this gasket stress will change. Generally, if the pressure on the sealing face of a gasket is less than the pressure of the fluid which is being sealed, then the gasket will leak. This is easily visualized as the fluid pressure itself (being larger than the pressure produced by the clamping parts) forcing apart the joint and preventing the gasket from regaining contact with the other parts.

To provide margin against this failure mechanism, the gasket working factor is used. This is the ratio of gasket pressure to sealed fluid pressure, and is found from an examination of all load cases following the initial assembly. An appropriate gasket working factor can be obtained from the gasket manufacturer. For narrow width gaskets (e.g. spiral wound) the working factor can be calculated as an average across the entire gasket face. In the case of flat sheet gaskets (as seen in Figure 2) the allowable working factor defines a boundary of tightness on the surface of the gasket. The DbA method thus proves useful in optimizing the design of such a joint, including features such as "crowned" surfaces intended to provide higher pressure at the inner gasket edges.

Cause: O-ring Extrusion
Effect: O-ring damage
Possible Consequences: Loss of working fluid, reduction of efficiency, reduction in developed head
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Extrusion gap check

For joints using O-rings, an initial seating analysis is not needed. The self-energizing nature of the O-ring creates a proper seal if normal gasket manufacturer recommendations regarding groove dimensioning is followed. During pressurized load cases, the O-ring behaves like a fluid with very high surface tension. Thus, it can safely span gaps between parts up to a limit, as shown in Figure 16. Above this threshold, portions of the O-ring will be forced into the gap, causing permanent damage to the gasket. Tightness may thereafter be compromised.

Within the DbA methodology, protection against the failure of O-ring joints is shown by displacement results which allow the O-ring gap to be determined. Care should be taken to ensure that the allowable gap values provided by the O-ring manufacturer are applicable to the pressure, temperature, and required loading cycles of the design.

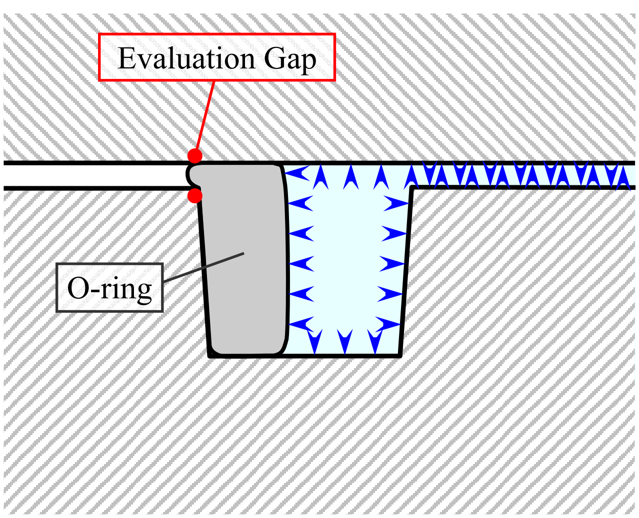

Figure 16: Pressurized O-ring Joint with Gap to be Evaluated (O-ring Distortion Exaggerated)

It is not the intention of the hydrotest load case to prove the long-term integrity of the O-rings. Analysis might be used to estimate the O-ring gap during factory testing, but, it is not necessary for it to fulfill the same criteria as applied to the design load case. Ultimately, the casing must simply remain leak-proof during the actual physical hydrotest.

Fit Failures

Cause: Plastic Deformation
Effect: Case distortion
Possible Consequences: Leakage of working fluid; inability to assemble machine
Typical modeling strategy: Compound worst-case
Recommended evaluation method: Elastic-plastic deformation analysis of load history including unload step

As previously described, plastic deformation of structures exposed to large internal pressures is expected, particularly for geometries which contain discontinuities that act to increase local stress. While most general pressure vessel designs are not sensitive to such plastic deformation, a centrifugal pump often has tight registration fits between parts. One such example would be a fit between a barrel and cover of nominal diameter 24 inches (600 millimeters), where there might be 0.003 inches (75 microns) radial clearance. While the Limit Load evaluation method described above provides assurance against plastic collapse, there is no guarantee by that method that the tight radial clearance will be maintained. Should it become reduced, the parts might not be capable of assembly. Should the clearance increase, the alignment required of the parts may be compromised.

In evaluating for plastic deformation, the pump manufacturer must identify those regions where limiting plastic deformation is critical to providing proper fits. An elastic-plastic material behavior is then introduced in the model. After each load case of interest, an unloaded relaxation load case could be calculated to give results which might be compared to physical measurements. A simulation matching the complete load history is needed in order to make a proper evaluation.

Careful thought must go into the application of material properties. Parts which take a permanent set have had their proof strength exceeded. The amount of deformation is dependent on the amount of stress *exceeding* the proof strength and not on the stress directly. Thus, the use of traditional safety factors on a residual deformation result may be misleading. (i.e. for a particular design, a ten percent increase in applied load might result in plastic deformation of twice the original magnitude.) Further, the yield and ultimate strength of steel materials are expected to vary from batch to batch, whether in cast or wrought form. Published values are typically minimums intended for use with deterministic analysis methods, and thus they correspond to the low end of statistical likelihood, perhaps in the range of the fifth percentile. In order for a manufacturer to validate their DbA method of evaluating plastic deformation failure modes they must also ensure this hidden margin is not simply covering a deficiency elsewhere in the calculation.

Rigorous guidelines are needed by the manufacturer to ensure that parts are not only initially fabricated to their proper dimensions, but that they continue to meet the design intent after experiencing any load case causing plastic deformation.

Cause: Tolerance Stack-up
Effect: Casing misalignment
Possible Consequences: Leakage of working fluid; rotor-stator contact; vibration
Typical modeling strategy: Probabilistic
Recommended evaluation method: Elastic deflection plus tolerances

Centrifugal pump casings not only have regions of close-tolerance dimensions, but many times these must be tightly restricted in their position relative to one another. An example would be the simultaneous compression of multiple gaskets between an inner and outer case. Because each dimension can be considered independently created, there is a wide space of possible manufactured conditions within the tolerance ranges. In some of them, a gasket might receive too much initial pressure; in others, not enough.

Worst-case tolerance stack-up evaluations of more than five independent dimensions approach the limit of reason. At such a point, it becomes more appropriate to consider them as true random variables in a probabilistic design. If six normally-distributed variables all had to simultaneously be in the 90th percentile in order for the design to fail, this represents a risk of failure equal to one in one-million. If the risk of this same failure mode by all other corresponding mechanisms (design or execution) is expected to be orders of magnitude higher, then allowing such a design means accepting a virtually negligible increase in the risk of failure. Since this is true even though the design might not pass a compound worst-case analysis, switching to a probabilistic design methodology sometimes allows manufacturing tolerances to be relaxed, without compromising the quality of the machine.

When critical locations are influenced by deflection due to load, the finite-element analysis can be used to determine relative movement, which is then superimposed on such a probabilistic evaluation of manufactured dimensions.

Cause: Thermal Distortion
Effect: Casing misalignment
Possible Consequences: Leakage of working fluid; rotor-stator contact; vibration
Typical modeling strategy: Nominal
Recommended evaluation method: Elastic deflection plus tolerances

Considering the tight dimensional tolerances used in centrifugal pump design, distortion of parts by thermal strain is an important effect to consider. Parts with dissimilar thermal expansion behavior will obviously show growth relative to one another. Even if a machine is composed of parts with equal thermal growth properties, the non-uniform temperature distribution expected for virtually all machines will create differential thermal expansion.

Evaluation of this failure mechanism involves a thermal simulation to determine the temperature distribution throughout all affected parts. This thermal calculation could be either steady-state or transient, based on the condition investigated. The thermal distributions are then used as a load in a set of structural calculations. With multiple thermal and structural load cases, such calculations are expected to be moderately computationally expensive, especially for 3D structures. As there are many model inputs with substantial uncertainty in such a coupled thermal-structural analysis, a nominal calculation approach with sufficient margin on the resulting output is recommended.

The designer is responsible for developing acceptable safety margins for each item of interest, based on practical experience. When investigating an new or unfamiliar failure mechanism, a set of analyses using arbitrary values could be used to determine of the sensitivity of the result to the calculation input. This is helpful in defining acceptable limits.

CONCLUSIONS

Design-by-Analysis is a powerful engineering method. It has the potential to produce safe designs of high quality which are well-suited to their intended service. When applied in a consistent and rigorous manner, the engineering work is more easily documented, reviewed, and interpreted. The ASME and EN13445 pressure vessel codes can provide useful evaluation procedures for many common failure mechanisms of pump casings. Minor adaptations are possible to make such pressure vessel codes more practical and appropriate when applied to centrifugal pumps. Not all important failure mechanisms and modes are explicitly covered by such codes, and thus the manufacturer's experience is important in determining the full set of failure mechanisms to be examined for each product.

NOMENCLATURE

ANSI	= American National Standards Institute	
ASME	= American Society of Mechanical Engineers	
HAZOP	= Hazard and Operability Analysis	
DbP	= Design-by-Precedence	
DbR	= Design-by-Rule	
DbA	= Design-by-Analysis	
Proof Load	= The maximum load a part can withstand before experiencing a permanent set.	
D	= Total damage factor	(-)
E_i	= Number of expected occurrences of event 'i'	(-)
f_t	= Load amplification factor	(-)
t	= Modeled design wall thickness	(L)
c	= Corrosion allowance	(L)
m	= Manufacturing allowance	(L)
n_i	= Allowable number of cycles before fatigue failure corresponding to the stress experienced during event 'i'	(-)
σ_{eqv}	= Von Mises stress	(F/L^2)
σ_1	= First principal stress	(F/L^2)
σ_2	= Second principal stress	(F/L^2)
σ_3	= Third principal stress	(F/L^2)

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