

## **More BOP Equipment HPHT Considerations**

### **Presented at the Aberdeen, May 1996 IADC Conference**

#### **Abstract**

In 1994, WEST completed a study for a consortium of seven Norwegian operators regarding BOP equipment requirements for HPHT (high pressure, high temperature) operations. A paper was written and presented to the November 1994 IADC Well Control Conference of the Americas in Houston covering key findings of that study. It was also published in the Oil and Gas Journal. Since that time, additional information or clarifications have arisen. Also, a number of inspections have been completed in preparation for HPHT drilling programs. This paper focuses on both:

- what has been learned since the presentation of the prior paper, and
- areas of recommended future technical investigation.

#### **Discussion**

Knowing when to use HPHT BOP equipment and the limits of “standard” equipment, practices, and procedures is critical in this type of drilling program. However, because this is a new area of the drilling envelope for many people, there are many details that must be thought through and checked. Assurance that all involved parties comprehensively understand equipment capabilities will improve the safety and cost effectiveness of your HPHT drilling program.

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References and illustrations at end of paper

The most significant contribution of the Norwegian study was to assemble experiences and design considerations from manufacturers, operators, and contractors, resulting in a single document with comprehensive, organized data.

It is clear that the ramifications of equipment failures are even greater when drilling in the HPHT arena. Accordingly, this paper includes a component of “back to the basics” – areas that are particularly critical considering the increased consequences of HPHT activity.

Two years after this study, the practical distribution and application of this body of knowledge is now the highest priority. Based on inspections completed in preparation for this type of drilling program, this information has not been universally applied.

#### **The First Step – HPHT Philosophy**

Spending a few minutes to define the issues and parameters is a good way to begin this type of discussion. First, HPHT is defined as drilling where one anticipates reservoir conditions of either high pressures (over 10,000 psi) or high temperatures (over 250°F). Although much has been learned about drilling in this HPHT envelope, there remain many areas of valid technical debate concerning equipment limits and required safety factors in individual risk analyses.

Clearly, drilling operations are not begun without risk evaluations and contingency planning if there are known equipment weakness or deficiencies. Accidents benefit no one. However, each company has different risk

tolerance, perhaps even differences depending on the circumstances of a specific well or program.

A key conclusion of the original study was that if equipment was delivered from the manufacturer with a 15,000 psi pressure rating, one need not apply additional safety factors for HPHT operation. However, this subject to the contractor complying with the manufacturers' inspection, testing and maintenance directives.

On the temperature side, the list of parts to be replaced in preparation for a given program to achieve high temperature rating vary. Strictly speaking, there are three general methods utilized to determine this list, listed in decreasing order of technical analysis required:

1. Development of a stack temperature profile during most severe anticipated conditions. This is generally a two step computer modeling process. First, maximum kick volumes and temperature must be determined. Then, this data is input to a program which calculates a temperature profile, taking into account cooling effects with time and distance due to gas expansion and heat transfer to the well control equipment surroundings. Proponents of this level of technical detail use in this analysis to enable them to more accurately assess the need for upgrading specific parts. Thus, they reduce their risk and often reduce their total upgrading cost. One company that can provide this service is Well Control and Systems Design in Katy, Texas.
2. Use of industry experience for "commonly replaced" parts. This is generally rather informal, and reflects an empirical approach relative to other successful programs similar to the anticipated one.
3. Replacement of every available high temperature rated part. Obviously, although this is the most expensive alternative, it is most conservative, providing equipment with the highest possible temperature rating currently available.

Therefore, you can see that there is no single best answer to what parts should be replaced to achieve "high temperature" ratings. The operator should specify the temperature requirements of each elastomeric part for the contractor's installation on his drilling program.

Documentation and parts inventory on the rig must also be considered. Because of its price and availability of HT spare parts, they are utilized only as needed. Accordingly, rigs with these capabilities must institute procedural controls and training to distinguish whether the stack is HT capable currently, what must be done to make it so if it is not, how to differentiate HT replacement parts from normally rated parts, and how to separately manage dual sets of parts with different ratings.

Clearly, preparation for HPHT drilling includes qualifying not only the well control equipment, but also other drilling equipment and support systems. Although not directly related to the "BOP Equipment" theme of this paper, a list of items to consider has been prepared and included for your reference as table 1.

## **Other Technical Information**

Since the publication of "BOP Equipment Consideration for HPHT Drilling" in November of 1994, consulting experience and ongoing feedback from the equipment manufacturers have resulted in an increased understanding of the equipment used in this application. Two main areas include an improved understanding of some details of the major specifications under which BOP equipment is manufactured: API Specs 16A and 6A. Again, an improved understanding of these areas is warranted because of the more severe conditions of HPHT drilling

### **API Specification 16A – "Drill Through Equipment"**

Comprehensive as API Spec. 16A is, there are still several areas that are not addressed by the standard that relate to this topic.

Notably, the prior paper reported wide variations in hang-off capabilities both between

different manufacturers as well as with different equipment and testing techniques. Obviously, hangoff capability is more important when drilling deeper wells with higher string weight. This variation was somewhat confusing why this was possible if the manufacturers reported Spec. 16A compliance.

At the top of appendix C, title “BOP Operational Characteristics Test Procedures Used to Define the Operating Characteristics of Blowout Preventers”, is the heading **Not a Requirement – For Information Only**. However, even if a manufacturer does comply with this appendix, it does not give objective criteria for tests. To illustrate the point more specifically, section 4.5 describes the hang off test. After a description of how to conduct the test, section 4.5.6 states, “increase the load (hung off) incrementally and repeat 4.5.5 (pressure test) for each load increment either until the rams leak or until 600,000 lb. load is reached...”. Note that no acceptance criteria is specified. This accounts for the wide range of load ratings between manufacturers, models, etc., even when all comply with Spec 16A. The revised version of 16A currently in the approval process does not change this section.

The original findings reported several facts about temperature ratings, as well as temperature effects on elastomers. Key among these facts is that there are neither consistent, objective industry standards nor testing methods for high temperature qualification. This may be a primary reason why the specification is silent regarding 350°F elastomeric material rings.

## **Failsafe Valves and Related API Specifications**

The key to an understanding of the topic of “failsafe valves” is to know what failsafe valve design you have, and how it works. Then, the design operating envelope can let you know the limits of your valves.

It may be timely to think through the intended purpose of failsafe valves. The author’s opinion is that the original intention of these valves was

to provide the capability to shut off choke and kill lines remotely and “automatically” (e.g., in the absence of closing hydraulic pressure) in the event of operating emergencies. **Agreement on the fundamental definitions of operational capabilities of failsafe valves is a critical first step to specifying design requirements.** This is particularly important for HPHT operations which extend the drilling operating envelope past the standard temperature ratings of equipment.

An operating scenario to consider is loss, or partial loss, of a control pod. This could happen from:

- a) the “garden hose” effect of the choke/kill hose whipping upon failure, bursting the 1” main hydraulic supply line or the pilot lines to failsafe valve open functions.
- b) the valve closing function of the control system could be inoperative, or the close pilot hose could fail. This former condition could occur and go undetected during function/pressure tests due to spring closure of the valve and successful wellbore pressure tests being achieved. Due to the small fluid requirements of the valve operator and poor sensitivity of some flow meters, it is conceivable that the abnormality is also not detected during function testing. In either case, the gate valves would begin to close from spring force and erosion could impair the sealing characteristics of the valve before the opposite pod was selected and operating pressure applied to the close side of the gate valve operators.

Prudent operating procedures suggest that the outer valve be closed first in the event full closure is not achieved or erosion occurs preventing a seal. The outer gate will then act as a choke and increase the likelihood of the inner valve successfully closing and obtaining a seal.

Few accidents occur as a result of a single failure. Considering the deficiencies identified on choke and kill hoses and the non-compliance of inspection and testing of these hoses according to manufacturer’s recommendations, along with the above example, failsafe valve operating capabilities become extremely important.

First, the industry does **not** have a manufacturing specification for valves installed on a subsea BOP stack or hydraulically operated valves installed on a surface stack. The most common specification used for stack mounted valve is API Specification 6A, “Specification for Wellhead and Christmas Tree Equipment”. This specification was written for surface equipment, which is predominately manually operated. When a hydraulically operated valve is designed to this standard for use as wing valves on a surface stack or as choke and kill valves on a subsea BOP stack, little guidance is provided to the manufacturer with regard to actuator force. API Specification 6A, 17<sup>th</sup> Edition, 10.16.3.4 “Actuation Forces”, states:

“Actuator output forces must meet or exceed the operating requirements specified by the valve or choke manufacturer”.

Considering that, most often, a single manufacturer supplies both the valve body as well as the operator, the actuator must meet their own stated requirements. However, note that no operating conditions are listed. Thus, what was operating envelope to which the valve (and actuator) were designed, and how do they relate to the conditions within a rig’s possible operating envelope?

Perhaps a good place to start this discussion is to review design principles of both balanced and unbalanced valves.

#### **Case 1: The unbalanced valve** (see figure 1)

The unbalanced design takes advantage of wellbore pressure to exert a force on the valve’s operating stem, assisting the spring cartridge in closing the gate. Therefore, the need for spring cartridges as big as your desk is eliminated. In this case, the spring force is designed only to close the valve in the absence of wellbore pressure when operating pressure on the open side of the valve operator is lost.

#### **Case 2: The balanced valve** (see figure 2)

The balancing stem was added to this design to reduce the depth sensitivity of valve operation.

Accordingly, if the areas of the balancing stem and the gate operating stem are equal, wellbore pressure does not contribute any closing force. Manufacturers report that, if the control system is intact after a choke or kill hose fails, the normal 1,500 psi operating pressure will close the “failsafe” valve.

Without specific guidelines, what other options does the purchaser or manufacturer have when specifying or designing a gate valve for these drilling applications? Two options are available, but realize neither was written with drilling valves in mind:

- API Specification 14D, “Specification for Wellhead Surface Safety Valves and Underwater Safety Valves for Offshore Service”.
- API Specification 17D, “Specification for Subsea Wellhead and Christmas Tree Equipment”, and

#### **API Specification 14D**

API Specification 14D provides the user with only a marginally better specification for actuator design. Paragraph 2.3.1. b states:

“SSV and USV actuator closing force must be sufficient to close the SSV/USV valve when the SSV/USV valve is at the most severe closing condition.”

Not significantly different from Spec. 6A, it does however add the requirement to be able to operate under the most taxing operating condition. Since neither this condition nor the thought process for determining it is specified, the manufacturer has the responsibility to determine the most severe condition and provide a design that will operate under it. Alternately, the manufacturer could provide a design operating envelope.

#### **API Specification 17D**

On the other hand, API Specification 17D most closely fits the application of BOP failsafe valves and does provide more specific actuator guidelines.

In section 908.2c (4), the following specific actuator requirements are listed which include capability to return the gate to the safe position during various conditions. Observations and comments that relate to these requirements follow each individual line item and are in italics. Because the primary application on BOP stacks is of the failsafe closed design, our comments are restricted to this single application for clarity.

“...Manufacturers shall specify the maximum water depth rating of the valve/actuator assemblies designated as fail closed (open) shall be designed and fabricated to be capable of fully closing (opening) the valve at the maximum rated water depth under all of the following conditions:”

*Manufacturers surveyed agreed that all of the hydraulically operated valves in use today on BOP stacks, both the balanced as well as the unbalanced design, would fail to satisfy at least one of the following criteria.*

“a) From 14.7 psia to maximum working pressure of the valve in the valve bore.”

*When only atmospheric pressure is in the valve bore, wellbore pressure will not assist in closing the unbalanced valve. Because the larger piston area is provided for opening the valve, as compared to closing, hydrostatic pressure will have the tendency to open the valve, overcoming spring force. This is common application for the term “depth rating” of a valve which all unbalanced valves functionally have. This was the main impetus for the balanced valve design.*

“b) Differential pressure equal to the bore pressure across the valve bore sealing mechanism at the time of operation.”

*In this case, the unbalanced gate design would again have the advantage compared to the balance design. Full wellbore pressure would assist in closure.*

“c) External pressure on the valve/actuator assembly at the maximum rated depth using sea water specific gravity of 1/03.”

*Because external pressure assists in closing the balanced valve in this case, this design would be preferred to meet requirement c.*

“d) No hydraulic assistance in the closing (opening) direction of the actuator other than hydrostatic pressure at the operating depth.”

*Here is another example of the importance of precise definitions. Manufacturers surveyed agreed that as little as 300 – 500 psi operating pressure will close balanced gate valves under flowing conditions. Therefore, to comply with various regulatory authorities as well as apply a prudent engineering safety factor, some drilling contractors have upgraded the balanced gate valve design on BOP stacks with the dedicated stack mounted “assist close” hydraulic circuits shown in figure 3. This provides the hydraulic energy required to close the valve under dynamic flowing conditions without assistance from the drilling rig pod umbilical hoses. However, depending on one’s definition of “hydraulic assistance”, requirement d) might not allow the use of such circuits. That is, if one considers “hydraulic assistance” to include **any** hydraulic force external to the valve, assist close circuits would be precluded from meeting this requirement. However, if “hydraulic assistance” is defined to mean force applied from the rig’s surface system, this circuit could satisfy operational requirements.*

“e) 114.7 psia plus sea water ambient hydrostatic pressure at the maximum rated depth of the assembly acting on the actuator piston in the opening (closing) direction.”

*One manufacturer stated their unbalanced gate design with an 8” piston would not satisfy this requirement. The force exerted by 114.7 psia on the 8” piston is 5,765 pounds. This is in excess of 5,000 pounds maximum force exerted by their standard spring cartridge. To use this valve for 17D production and applications, the valve was redesigned with a 5” piston, using 3,000 psi, as opposed to 1,500 psi, normal operating pressure.*

## Actuator Spring Cartridges

So what contribution can you expect your spring cartridge to make towards the operation of your valve?

Generally speaking, the spring cartridges offered by BOP valve manufacturers are similar. On BOP stacks, it is impractical to design a valve to close using a spring cartridge only due to size limitations. The operators required would be as big as your desk. Several industry experts were surveyed, and they agreed with the following approximations:

- The static gate drag due to maximum rated wellbore pressure when the gate is approaching the closed position varies greatly depending on the valve size. Also, another significant factor is the coefficient of friction assumed between the mating surfaces. The author's own experiences with friction coefficients are dubious at best compounded with the fact that they can change during the life cycle of the equipment.

The two well respected gate valve experts<sup>1</sup> consulted provided the following estimates for static gate drag.

- 3 1/16" – 10,000 psi wp (working pressure) valve: one estimate was 25,000 lbs. force. Needless to say, both estimates are many times greater than the force provided by a spring cartridge.
- 4 1/16" – 10,000 psi wp valve: 40,000 lbs. force was estimate.
- As an example of the limitations of springs, one manufacturer stated that the largest spring operator his company had ever manufactured was for a 4 1/16" 10,000 psi working pressure valve developed as part of a joint industry project to be used in 7,500 feet of water. The maximum spring force generated was 13,000 pounds.
- Spring forces are not constant. When a typical actuator spring is compressed it has a force of approximately 5,000 lbs. but only 2,000 lbs. when in the fully extended

position (the gate being in the design safe position on a BOP stack). The spring generates the least force when the most force is required for complete closure.

- One industry expert expects the spring force to act in the following manner on 3 1/16" 15K BOP valves. Consider wellbore fluids flowing through an open gate valve. If hydraulic operating pressure is lost, and the valve is of the balanced design, the spring cartridge has the force to begin the motion of the gate will continue until the wellbore forces impose a 500 psi differential across the gate. The 500 psi differential pressure will create a downstream friction force (between the gate and the seat) to overcome the spring force and halt the motion of the gate to the closed position. Clearly, this is only a fraction of that required to close the valve at rated conditions.

## CX-18 Connections

The CX-18 joint is a common drill through equipment connection. The purpose of its development was to ensure face to face contact, which was not assured with the BX design. Face to face contact between the flanges or hubs allows the transmission of external forces, such as bending, through the flange/hub faces as opposed to through the gasket. This reduces gasket creep and the resulting loss of bolt preload. See figure 4.

Leaking connections are not uncommon occurrences during rig inspections. Most often, acceptable tests can be obtained by breaking the connection, visually inspecting the sealing surfaces and blending in any minor defects, if required, and replacing the gasket.

However, there are no inspection criteria for any connection design. The original HPHT study included a Cameron (a premiere equipment manufacturer) submission answering the in service ring groove inspection query with the following:

- There are no after test (meaning proof test) criteria for inspection of ring grooves.
- A prep is judged too large when a new gasket will no longer seal against it. And
- We do not have visual inspection criteria for ring grooves.

Therefore, manufacturers simply apply a performance criteria – if a connection holds pressure without any detectable leaks, it is fit for purpose. While this standard is adequate in most instances, a small number of CX-18 testing problems have arisen when the standard solution does not work.

While inspection criteria for the connection, how can a leaking joint problem be solved? In the CX-18 cases of interest, the manufacturer was consulted for assistance after numerous unsuccessful tests. Although the manufacturer's field service representative did not have acceptance criteria either, he did have and apply the original, pre-proof testing design measurements. Note that this manufacturing criteria (properly held only by the original equipment manufacturer as a result of the design and testing process) is the only available tool to help solve the problem.

The most recent CX-18 incident involved a new stack. WEST provided inspection services for the operator at the manufacturer's facility prior to shipping. After skim cutting one CX-18 sealing surface, successful tests were obtained on all connections. Because of prior, albeit infrequent, difficulties with CX-18 joints, WEST recommended that the manufacturer re-measure the groove dimensions after proof testing. This would document any plastic deformation caused by the proof testing. However, timing was critical to meet delivery commitments, and neither the manufacturer nor the contractor (the manufacturer's customer) had prior knowledge of CX-18 problems. Thus, the recommendation was not acted upon.

WEST was also commissioned to conduct final acceptance testing on the rig, which was expected to be primarily focused on control

system hookup due to the prior work. However, once on the rig, seven unsuccessful pressure tests were conducted prior to the arrival of the manufacturer's repair supervisor and detailed groove measurements. Only after six more pressure tests, with new gaskets, and machine work to correct 0.030" ovality and 1/2° excess groove angle was a successful test obtained.

In conclusion, for CX-18 connections, the following observations should be noted:

- Cadmium plated mild steel gaskets are recommended. This softer material seals more groove inconsistencies than the harder stainless steel.
- Particularly in the case of new or remachined joints, a program should be implemented to measure groove dimensions after proof testing to assemble inspection criteria. Measurements of connections with good service histories should also be assembled.

## Wellhead and Riser Connectors

Backdriving, or the loss of preload between the connector and the wellhead, which results in loss of wellbore integrity due to movement of the cam ring, has long been recognized as a potential problem with connectors. As wellbore pressures increase, the backdriving forces increase as well. For years, several manufacturers' standards precaution was to ensure the application of operating pressure prior to wellbore testing. Figure 5 illustrates the backdriving mechanics.

However, while recognizing the potential, this avoids adequately addressing the backdriving question. The not uncommon situation of disconnecting the LMRP places the equipment in exactly the situation explicitly avoided in most testing procedures; i.e. exposing the wellhead connector to wellbore pressure without any applied operating pressure. Accordingly, because of persistent WEST recommendations to the API Specification 16A (for new equipment) committee, a backdriving testing requirement has been added to the proposed 2<sup>nd</sup> edition

which has yet to be published.

At the same time, such a requirement does not exist for equipment in service. The HPHT study resulted in each manufacturer recommending different testing procedures. At the same time, improved testing procedures were developed which has resulted in WEST documenting three cases of backdriving in a six month period in 1995. To our knowledge, these are the only known documented cases of backdriving during a field inspection process, validating a prior technical concern.

In conclusion, backdriving is a real concern for all driving operations. In order to detect connector deficiencies that allow backdriving, the improved testing procedures only recently developed must be utilized as the only known documented cases have been identified with these field inspection procedures.

## **HT Equipment Testing and Inspection**

Equipment manufacturers have supplied the drilling contractor with field wear tolerances to allow the contractor to anticipate and pursue an active preventative maintenance program. The most useful and widely available field wear tolerances concern wear between the ram block and the ram cavity in a ram type BOP.

Field wear tolerances were established prior to high temperature operations being a consideration. A preventer and block were machined as part of the original testing procedure, increasing the block to cavity clearance until leakage occurred. A safety factor was applied to the clearance and the resulting tolerance thus established as the maximum allowable field wear.

With the advent of high temperature elastomers, questions occur whether these same wear criteria are still valid. Obviously, because the higher temperatures allow an increased amount of rubber flow, clearance between sealing

surfaces have increased importance. As an example, some high temperature ram rubber designs utilize a metal extrusion plate as part of the top seal to reduce rubber flow into the clearance between the ram cavity and the top of the ram block. Because there have been no published test results for HPHT application, the suitability of previously acceptable field wear tolerances is unknown with these new elastomers at elevated temperatures.

In the SINTEF study "Reliability of Subsea BOP Systems" which analyzed Norwegian downtime statistics from 1987 to 1989, the equipment type contributing the largest amount of downtime was riser choke and kill line pressure integrity. Some manufacturers have developed field acceptance criteria for dimensional inspection of choke and kill line boxes and pins. Just as different HPHT sealing surface clearances for ram preventers have not been established, revised dimensional acceptance criteria for choke and kill connections have not been established for the more demanding high temperature applications in which we are now working.

One frequently finds that equipment with the lowest temperature ratings in the well control system is the high pressure choke and kill hoses. One major hose manufacturer rates its hose at 100°C (212°F) continuous. At temperatures above 100°C but below 130°C (266°F), this hose manufacturer rates the hose at up to one month *cumulative* service. One must realistically question if the rig has documented and recorded the hose use at elevated temperatures to determine the cumulative exposure. This places increased importance on the annual hose internal inspection and pressure test above working pressure per the manufacturers' recommendations. It is interesting to note that such inspection and testing to the manufacturers' recommendations is almost exclusively limited to North Sea operations.

## Summary and Conclusions

Because an ever larger number of wells drilled worldwide involve either high pressures (over 100,000 psi) or high temperatures (over 250°F), the distribution of the technical information available is important to minimize problems. The following points should be kept in mind when planning your next HPHT program.

1. Particular attention should be paid to choke and kill hoses, subsea valves and elastomeric parts on rigs preparing for HPHT drilling.
2. Currently, subsea stack valves are manufactured to Specification 6A requirements, which did not anticipate remote actuation. To the author's knowledge, no actuated valves have been manufactured to either Specification 17D or 14D for use on BOP stacks. None of these specifications were written with subsea stack valves in mind.
3. Manufacturers should specify operating envelopes of their failsafe valves.
4. Consider developing an industry specification for subsea stack valves.
5. A review of CX-18 connection problems are appropriate, particularly with new or newly machined grooves.
6. Wellbore connectors should routinely be tested with the best available procedures to identify backdriving potential.
7. Field wear tolerances should be reviewed to determine if they remain suitable for HPHT applications.
8. Hoses are not normally inspected to OEM recommendations outside the North Sea. Because of ratings of some hoses relative to cumulative high temperature exposure, record keeping of hose use can be critical.

<sup>1</sup> Mr. John Fowler, President of On-Line Resources, Houston, Texas, provided valuable assistance on the failsafe valve sections. Mr. Fowler is a holder of 28 US patents, and was formerly the Vice President of Engineering for McEvoy. He also served the industry as Chairman of the API Specification 6A design committee, and president of the Association of Wellhead Equipment Manufacturers.

# Drilling Systems Considerations for HPHT Operation

The following items should be afforded particular attention for vessels audited in expectation of establishing High Pressure High Temperature (HPHT) drilling operations:

## Safety Considerations

- Access fire suppression systems in mud pit room and solids control area.
- Determine adequacy of available PPE and posted warnings for HPHT operation.

## Environmental Survey

- Assess ability for containment and recapture of HT synthetic or Oil Base Mud (OBM)
- Assess liquid mud transfer systems and storage procedures
- Confirm MSDS file is current and readily available

## Drilling Equipment

- Confirm manufacturer's product alerts and safety bulletins have been addressed on TDS.
- Confirm TDS components and elastomers are compatible for HPHT and synthetic or OBM

## Mud System

Mud Pumps (pump pressures greater than 3500 psi is assumed)

- Determine if mud pump fluid and modules have been reconditioned and confirm acceptability. Reconditioned modules are not recommended for maximum pumping pressures.
- Recommend NDE of module, valve caps, all rod ends and clamps.
- Confirm condition of wear plates and seal areas.
- Examine valve seating areas in modules.
- Assess fan and ducting capacity for cooling DC motors. DE motor power is directly related to its ability to stay cool.
- Review cooling capabilities for fluid and power ends.
- Confirm elastomers suitability for HPHT, synthetic or OBM.
- Test mud system HP piping and valves to MWP.
- Conduct ultrasonic examination on HP piping at points subject to hydraulic erosion. Review piping materials to allow calculation of maximum working pressures.
- Test run pumps at maximum expected SPM and pressure.
- Test relief valves and set as close as possible to maximum liner rating.
- Confirm adequate spares on board for mud pumps. Increased spares are recommended for HPHT programs.

## Mud Pits, Mixing and Transfer

- Determine if consideration has been given to condition of mud pit structure and ability to handle elevated mud densities.
- Confirm by testing (and recording results) that mud mixing and transfer systems will perform as required.
- Test medium must have properties equal to maximum expected mud density.

## Solids Control

- Confirm by testing (and recording results) that solids control will handle maximum expected flow rates.
- Test medium must have properties equal to drilling mud that will be used.

## Ventilation

- Determine if existing ventilation air flow has been calculated to ensure system will be adequate to handle fumes from high temperature, synthetic or OBM in pit room.

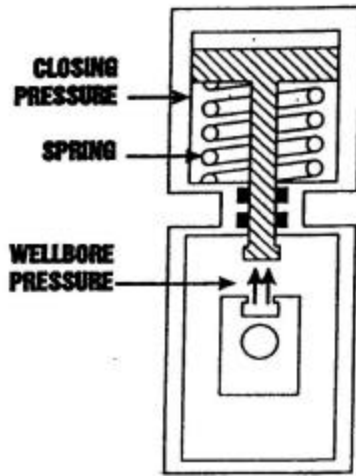
- Determine if ventilation will be adequate in pump room for HT operations.
- Determine if ventilation will be adequate in solids control area from for HT operations.
- Confirm operation and condition of all associated valves, fans and filters for above ventilation.
- Determine how oil extract product from ventilation system be handled.

### **Engine Room and Associated Equipment**

- Determine power generation and distribution ability to meet high pumping pressure requirements

### **Electrical Systems**

- Confirm integrity of all components in classified zones.

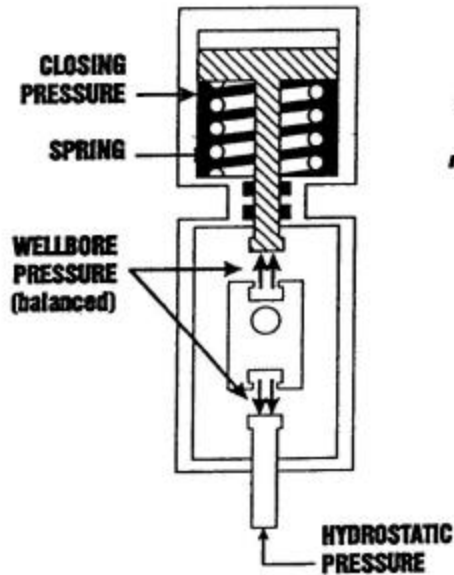


## Failsafe Valve FAILSAFE TECHNIQUES

Wellbore Pressure  
Close Operating Pressure  
Spring

**WEST**  
*CSI*

FIGURE 1



## 'Balanced' Gate Valve "FAILSAFE" TECHNIQUES

Close Operating Pressure  
Spring

**WEST**  
*CSI*

FIGURE 2

STACK MOUNTED "ASSIST CLOSE"  
HYDRAULIC CIRCUIT DIAGRAM

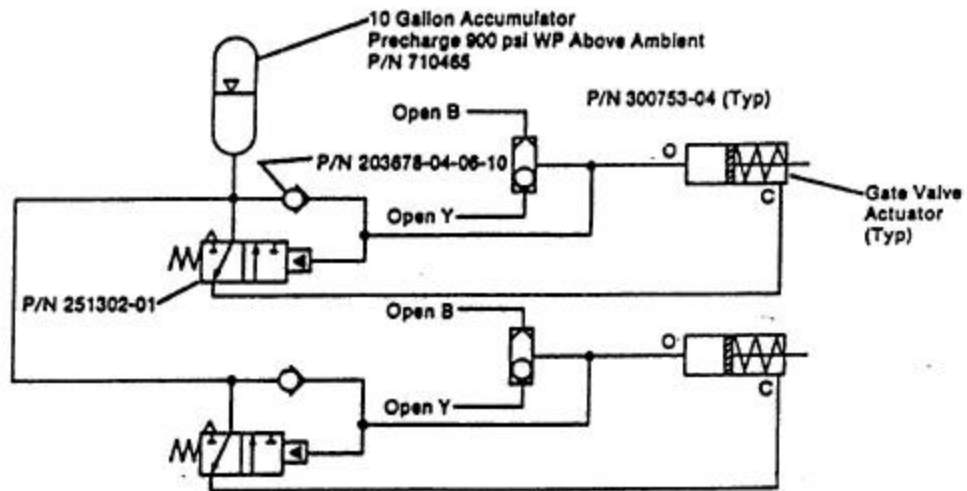
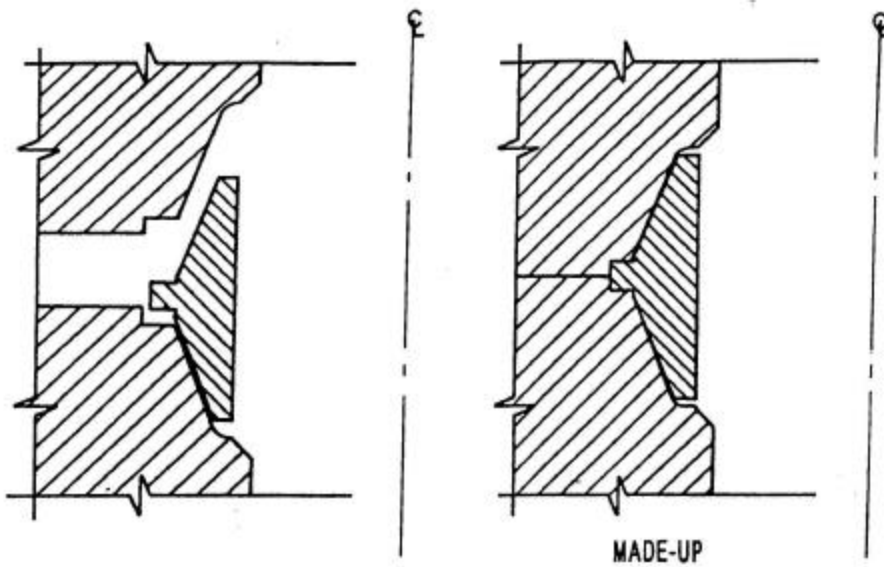


FIGURE 3

COURTESY OF CAMERON

"CIW" TYPE "CX" PRESSURE-ENERGISED RING JOINT GASKET



CX  
FIGURE 4

COURTESY OF CAMERON

Back Driving

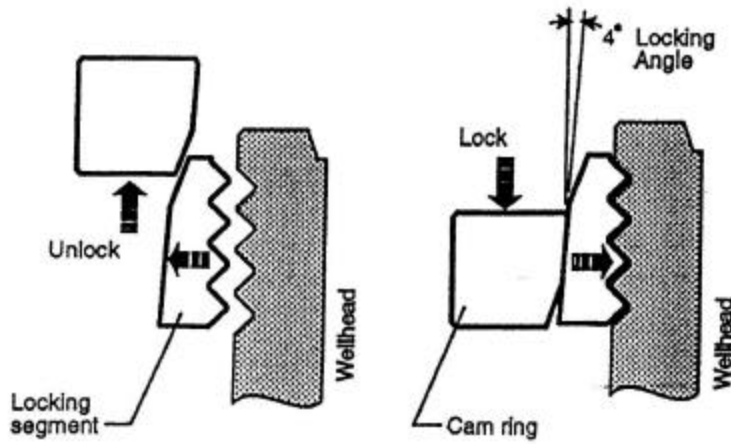


FIGURE 5