REALISTIC EXPECTATIONS FROM A UNIQUE PLAN 66A BUSHING DESIGN

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ABSTRACT

As crude oil remains the dominant fuel and raw material source for transportation and industry the crude oil pipeline continues to be the preferred crude transportation method across large land masses. Along with this continued reliance on high pressure pipelines comes an increasing demand for safety and containment. Though various mechanical seal arrangements exist to ensure safety and spill avoidance including dual and safety containment seals, many pipeline pumps providing the power for fluid transportation rely on

single mechanical seals to contain the crude and prevent leakage. With the continued use of single mechanical seals comes the need for leakage detection, containment and management in the event of an operational upset.

In this paper we look at the capability of a unique dual bushing arrangement commonly referred to as API piping plan 66A for use with a single mechanical seal. We demonstrate the capacity of the arrangement to provide a warning signal in the form of a detectable pressure increase that can be used to trigger pump shut down and isolation activities. We examine the limitation of relying on the pressure signal to provide an accurate assessment of primary seal condition. Also shown are the capabilities of the arrangement to limit the volume of crude escaping the pump during operational upsets, shut down and isolation activities mitigating or eliminating the need for clean-up operations and reducing environmental impact.

INTRODUCTION

Crude oil pipeline pump applications are unique in the hydrocarbon processing industry. The application, equipment and operating environment are very different from the typical hydrocarbon process pump in a refinery or petrochemical plant. The crude oil pipeline network is huge, and continues to grow as old lines are upgraded and new capacity is added [1]. Figure 1 illustrates the market percentages of pipelines and their regional locations. This shows that 65% of crude oil lines operate in North America & Canada, where pipeline sealing practices tend to be different from other geographic locations.

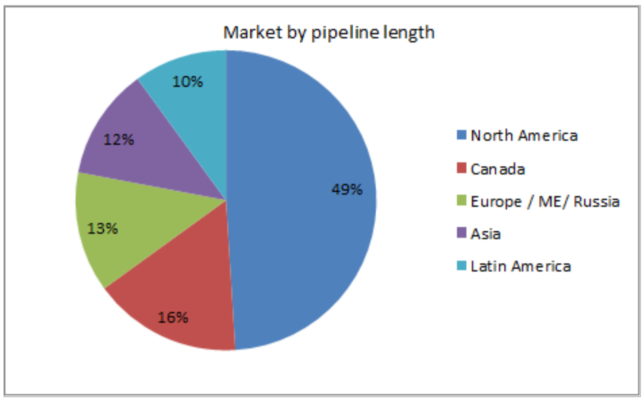

Figure 1 Regional Locations of Crude Oil Pipelines. [2]

The network of Crude oil transmission lines across the world continues to grow. Crude oil pipelines make up about 1/3rd of the liquid hydrocarbon lines in the continental U.S. This network is growing with significant investment in NA [3] involving an ever increasing population of pumping stations and fluid handling equipment. Pipelines delivered 8.3 billion barrels of crude oil and 6.6 billion barrels of petroleum products (gasoline, diesel, jet fuel, etc) and natural gas liquids (propane, ethane, etc) within the US in 2013. This equated to an 11.3% increase over 2012 deliveries [3] and this figure continues to grow. Pumping stations that serve transmission and gathering lines can be very remote and often unmanned for long periods of time. This necessitates remote monitoring and control without the luxury of walk around inspections and on the ground human intervention in the event of a problem manifesting.

There are therefore operational and process differences with respect to how these assets are operated and managed as a result of the layout and location of these stations as well as their position in the transmission network. Simple and safe tend to be the operating philosophy as reliability problems in remote locations present a number of challenges. For example, in the event of equipment malfunction sudden and rapid discharge could lead to significant environmental impact. Remote locations add to the complexity and cost of repair and preventive maintenance. Providing safe operation and effective management of operational leakage and containment in the event of a catastrophic seal or pump problem are high priorities for all operators. This prioritization has financial motivation as cost involved in containment or clean up can be orders of magnitude greater than in plant or refinery systems.

A pipeline is made up of a series of pumping stations to maintain process flow. Many factors may dictate the spacing of these stations and the pump design / capacity. Pumping stations in a crude oil network will typically be about 50-70 miles apart depending on piped product and terrain, and contain between 2 and 4 pumps in series. The pumps at each location form a pumping cycle, normally resulting in pumps within the same stations seeing suction pressures of around 100 psig (7 barg) and discharge of 500 psig (34.5 barg). Some pumps within the station seeing suction pressures as high as 900 psig (62 barg) and discharge pressures up to 1440 psig (100 barg). These pressures need to be contained during operation by the primary mechanical sealing device and in the event of equipment failure while the shutdown operation is instigated when the primary sealing device may have become breached.

Typically, though not exclusively, pipeline station design is symmetrical featuring equipment that is proven and preferred by the operating company. Redundancy and standardization is capacity driven and enables the company to continue transmission even in the event of a partial outage. Environmental regulations imposed on the operators in regard to leaks and spills are aggressive [4] so the provision of a reliable pumping and containment solution is critical to an operators P&L. Lost production due to outages can quickly eclipse $150k / hour [5], and have non favorable market implications [6] plus excessive clean up and regulatory costs.

Pipelines require seal solutions that deliver extended reliability in clean, dirty or mixed crudes, with a desire to run in excess of 3 years. Mechanical seals in this environment are well understood, but containment sealing – when a secondary (API plan 52 or 53 or 54) [7] mechanical seal and seal support system are not preferred due to the potential complexity of maintenance, - is evolving as new / better simple solutions are sort. API plan 66A is an attempt to improve containment reliability whilst maintaining the simplicity of operation favored in remote unmanned locations.

The 66A solution [7] is designed primarily for pipeline operators using single seals, to limit and control leakage to atmosphere. The design uses two throttle bushings in series. During normal operation seal leakage can freely flow through the drain port located past the first bushing. When leakage is excessive the inner bushing will restrict the flow to the drain and the second bushing will limit the amount of leakage leaving the gland. Located between the mechanical seal and the first bushing is a pressure indicator or alarm. With increasing seal leakage rates the pressure on the upstream side of the inner bushing will increase. This increase will be sensed by the pressure transmitter and set off an alarm indicating a seal failure. Leakage exiting the drain port can be collected and piped to a sump or liquid recovery system. Leakage that potentially overwhelms the drain, can be contained by the secondary bushing until equipment shut down and isolation has occurred. The concept that leakage could be monitored through a pressure increase would only be valid with significant seal leakage occurring, the ability to measure flow at the drain connection will give an earlier and more robust indication of inboard seal problems.

The paper will discuss the challenges associated with containment sealing of crude oil applications, will discuss the different technology and measurement strategies that can be employed and will make recommendations on best practice for a safe and simple deployment.

CURRENT PUMP TECHNOLOGIES FOR PIPELINE APPLICATIONS [8]

Pipeline operating conditions vary greatly, but typical operating conditions for crude oil pipeline applications located in North America are:

- Fluid: Crude Oil
- Temperature vs. viscosities (from the Canadian border to the Gulf Coast)
    - -2.5 C @ 3169Cst
    - 15C @ 650Cst
    - 30C @ 215Cst
- Temperature: -4 to 400 F (-20 to 204 C)
- Size range: 2.625” to 6.130” (67-156mm)
- Speed: Up to 5000 fpm (25.4 m/s)
- Pump: Double-ended between bearings configuration
- Seal chamber pressure: Up to 1440 psig (100 barg) dynamic / 2200 psig (150 barg) static

Typical equipment used on crude oil pipelines.

Pumps used in crude oil pipeline service are almost always described and defined by the API 610 pump standard [9]. API 610 groups pumps types using a designation code. The following pump types are typically used in crude oil pipeline service:

- OH2 is one horizontal overhung impeller, centerline mounted and very common in refineries and offshore; sometimes called the “API 610 pump” or “process pump”
- OH3 is one vertical impeller, in-line mounted, very common in refineries and offshore
- BB1 is one impeller between bearings, axially split case, used in refineries and pipelines
- BB3 is multi-stage, impeller between bearings, axially split case, used in pipeline service
- BB5 is multi-stage, impeller between bearings, used offshore, called a “barrel pump”
- VS1 is a vertical, multi-stage pump, sometimes called a vertical turbine pump

Types BB1 and BB3 are the most common pump used in crude oil pipelines in North America.

SINGLE SEALS AND SECONDARY CONTAINMENT

Crude oil pumps are most often sealed by single seals with a secondary containment device. For those applications allowing single mechanical seals several options are available as the secondary containment device. One option is to back up the primary mechanical seal with a single contacting bushing or close clearance throttle bushing and a drain. This arrangement would typically have the drain piped to a reservoir where a level transmitter would indicate when the reservoir is full or excess leakage is detected. This is identified as a Plan 65A or 65B. It is likely the most cost effective secondary containment option and is intended to provide for liquid leakage detection. This option will likely result in leakage from the pump case in the event of primary mechanical seal failure. A typical single seal with a bushing as a secondary containment device for use in Plan 65A or 65B is illustrated in Figure 2.

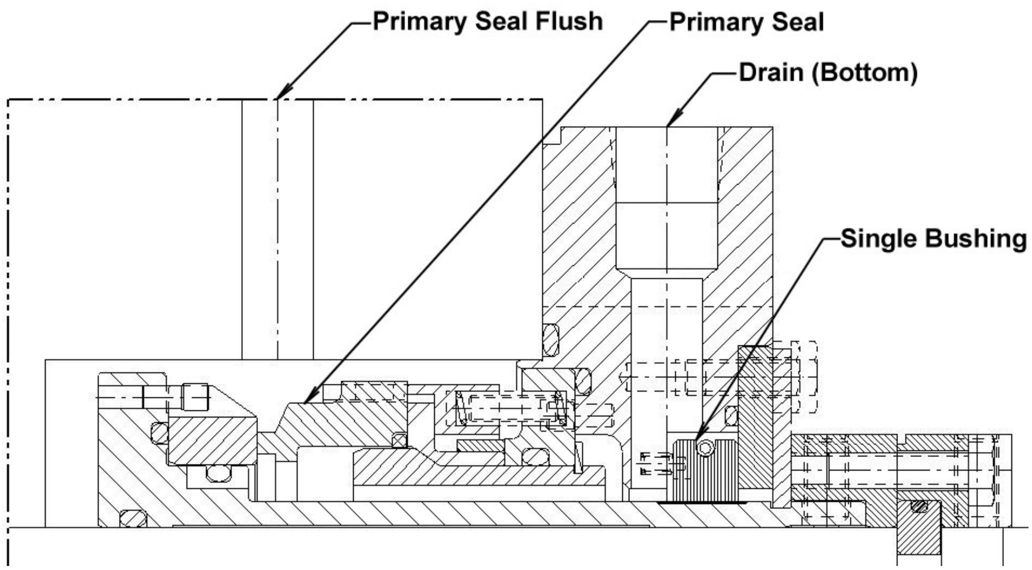

Figure 2 Typical single pipeline mechanical seal with bushing for secondary containment

Another option is to back up the single primary mechanical seal with a specially designed secondary mechanical seal. The secondary containment mechanical seal runs unpressurized during normal operation. It also keeps the vast majority seal leakage from exiting the pump while directing it to drain. The secondary containment seal has the capacity to operate at the same conditions as the primary seal. It is usually intended to operate for a short duration while equipment shut down and maintenance commences. This secondary containment option offers the greatest protection against seal leakage, costs more and takes on the characteristics of a dual mechanical seal. A typical single seal with a secondary containment seal is illustrated in Figure 3.

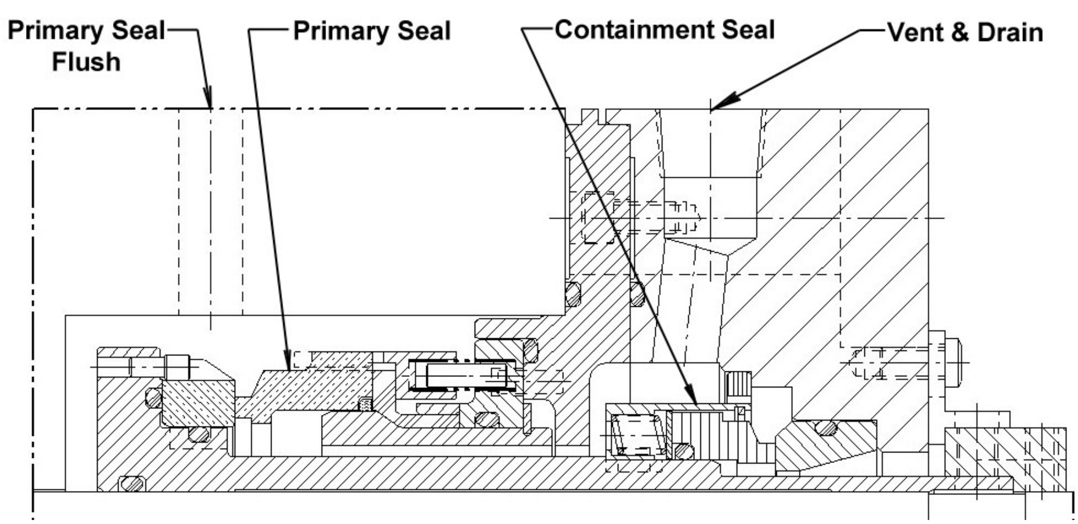

Figure 3 Typical single pipeline mechanical seal with secondary containment seal

The American Petroleum Institute (API) 66A piping plan was introduced in the fourth edition of API 682. It is described as a leakage detection plan. With crude oil pumps relying on single mechanical seals the 66A is sometimes specified to provide an additional element of safety. The main function of the plan is to provide an alert in the event of a mechanical seal malfunction and to reduce process loss from the pump. The plan is identified as a dual bushing arrangement downstream of the primary mechanical seal with a pressure transducer between the primary bushing and the primary seal. The cavity between the two bushings is piped to drain. In the case of the pipeline seal this drain is usually piped to a large tank or holding vessel where the fluid is captured and can be returned to the process stream. The typical plan 66A is shown in Figure 4. The typical pipeline mechanical seal with the dual bushing used with the 66A piping arrangement is illustrated in Figure 5.

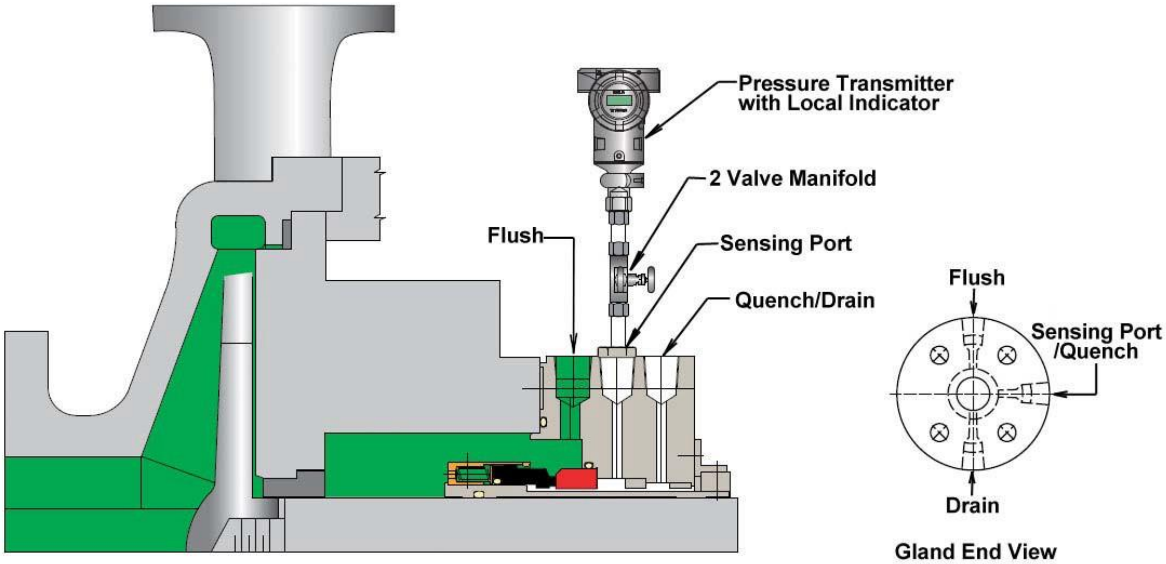

Figure 4 API 66A mechanical seal piping plan

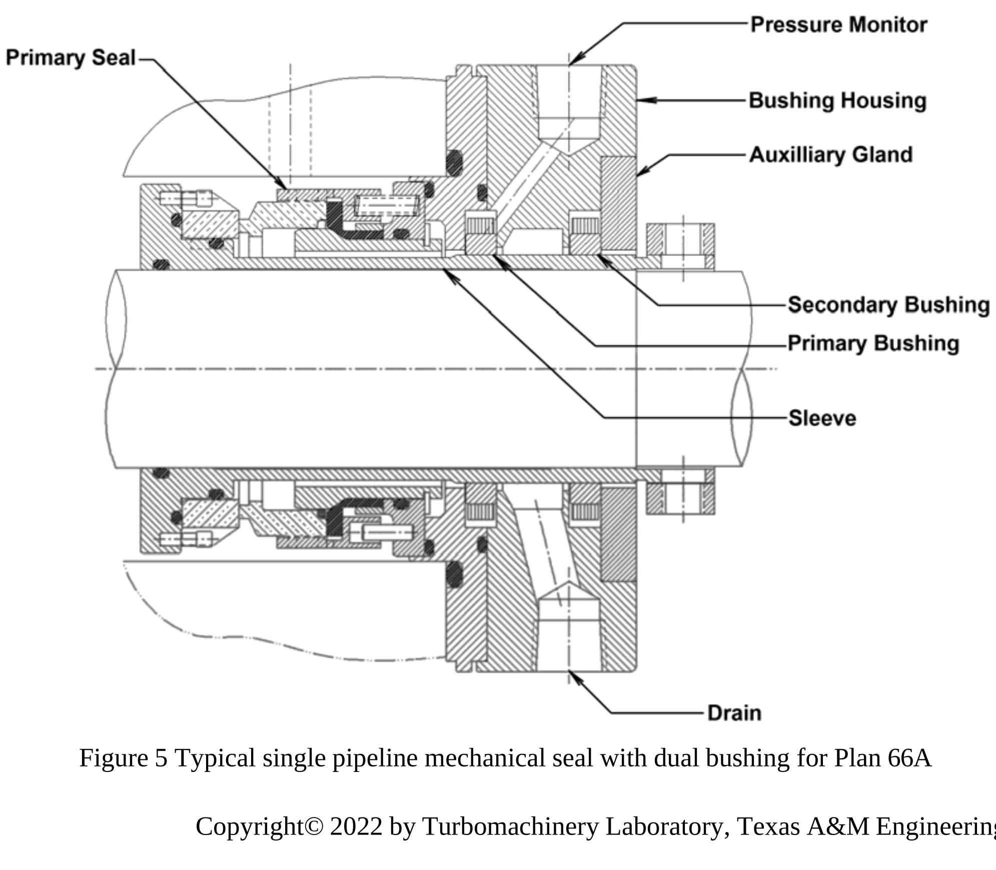

Figure 5 Typical single pipeline mechanical seal with dual bushing for Plan 66A

TESTING AND DEVELOPMENT

In this paper we focus on the design and performance of the dual bushing arrangement. Performance characteristics and test results are provided. Testing was performed on a 6.625 inch (168 mm) diameter dual bushing arrangement. Royal Purple™ 910 was used as the test fluid to approximate the properties of crude oil. We also look at the performance limitations of the 66A bushing arrangement in order to establish operational guidelines. Though the 66A arrangement by design may seem simple, listing all of the specific performance requirements reveals a deviation from simplicity and extends its function beyond merely leakage detection. The requirements were established through continued dialogue among field sales and service personnel, customers and the engineering development team. [10] These requirements provided focus for the design and testing and are used to establish success. The specific performance targets for the 66A arrangement are itemized.

The dual bushing arrangement should meet the following performance criteria which will be referenced by number throughout the paper.

1) Allow all normal seal leakage past the primary bushing without generating pressure or triggering an alarm.
2) Keep normal seal leakage from exiting the pump case and directed to drain.
3) Provide a signal in the form of a pressure increase when the primary seal is compromised.
4) Restrict leakage past the primary bushing at full pressure in the event of primary seal catastrophic failure during dynamic operation allowing time to detect the upset and permit shaft coast down.
5) Restrict leakage past the primary bushing at full pressure in the event of primary seal catastrophic failure during static operation after pump shut down and during pump isolation.
6) Restrict leakage past the secondary bushing in the event of primary seal failure such that the vast majority of fluid is directed towards drain during static and dynamic operation.
7) Operate for 3 years under normal conditions and maintain the capacity to meet requirements 1 through 6 above.
8) Maintain structural and operational integrity up to of 150% of maximum process pressure during static pressurization.
9) Optionally restrict leakage past the secondary bushing in the event of primary seal failure and primary bushing failure during full pressure dynamic and static operation.

In order to closely match the operating parameters expected to occur in pipeline pumps the 66A arrangement was tested on a rig in conjunction with a mechanical seal. Since drainage and the effect of gravity on the fluid proved to have a significant impact on the design's capacity to meet some of the performance criteria, the horizontal test position was essential to validate success under conditions similar to those expected in horizontal pipeline pump field installations. During testing shaft speed, temperatures and pressures were electronically monitored and digitally recorded. Since flow detection and measurement both at the drain and at the atmospheric collection point proved extremely variable, flow data was collected, measured and recorded manually using graduated collection vessels and a stopwatch. The test rig used for full pressure dynamic testing is shown in Figure 6 and Figure 7.

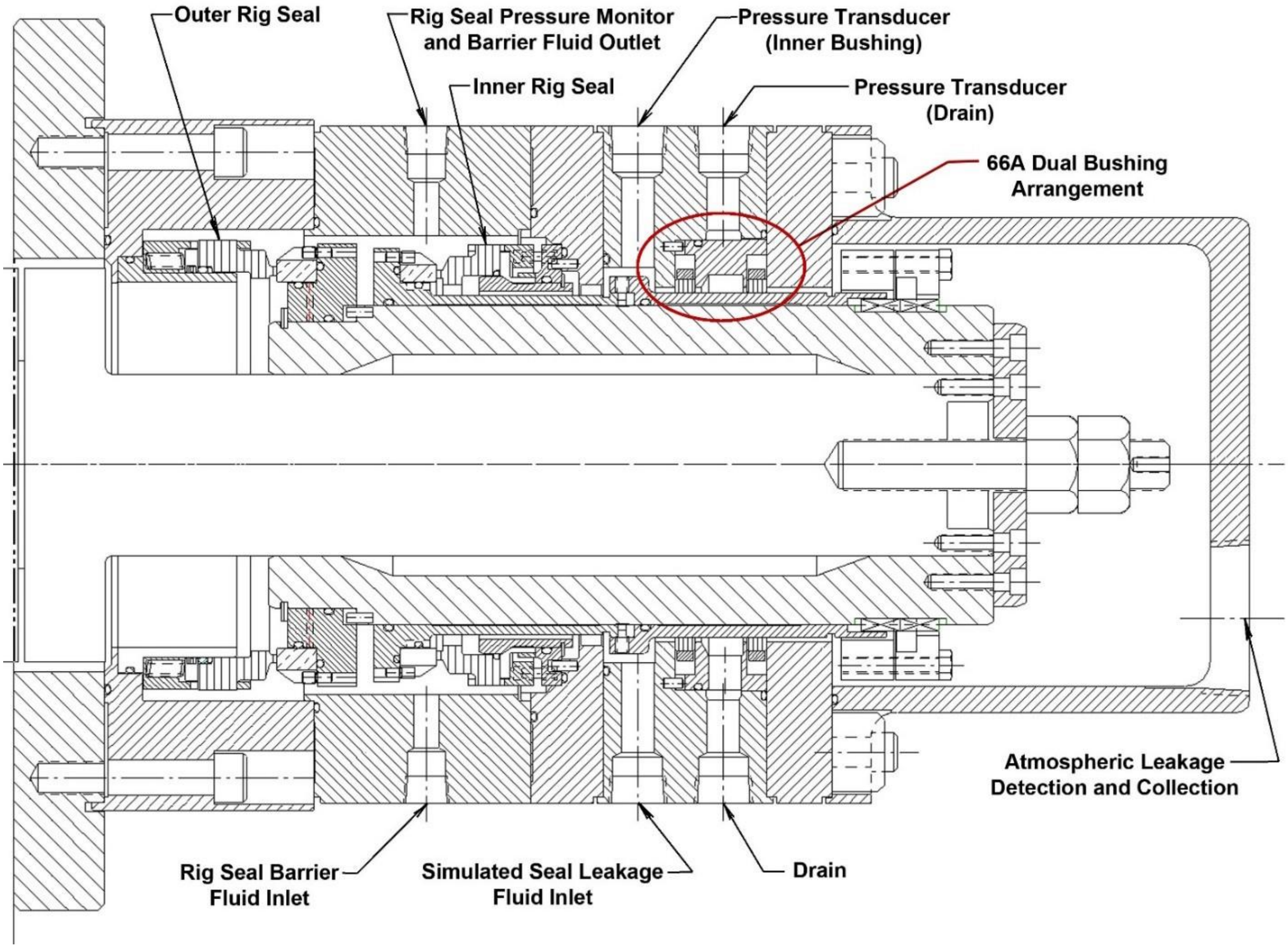

Figure 6 Test rig for full pressure dynamic testing of the 66A bushing arrangement

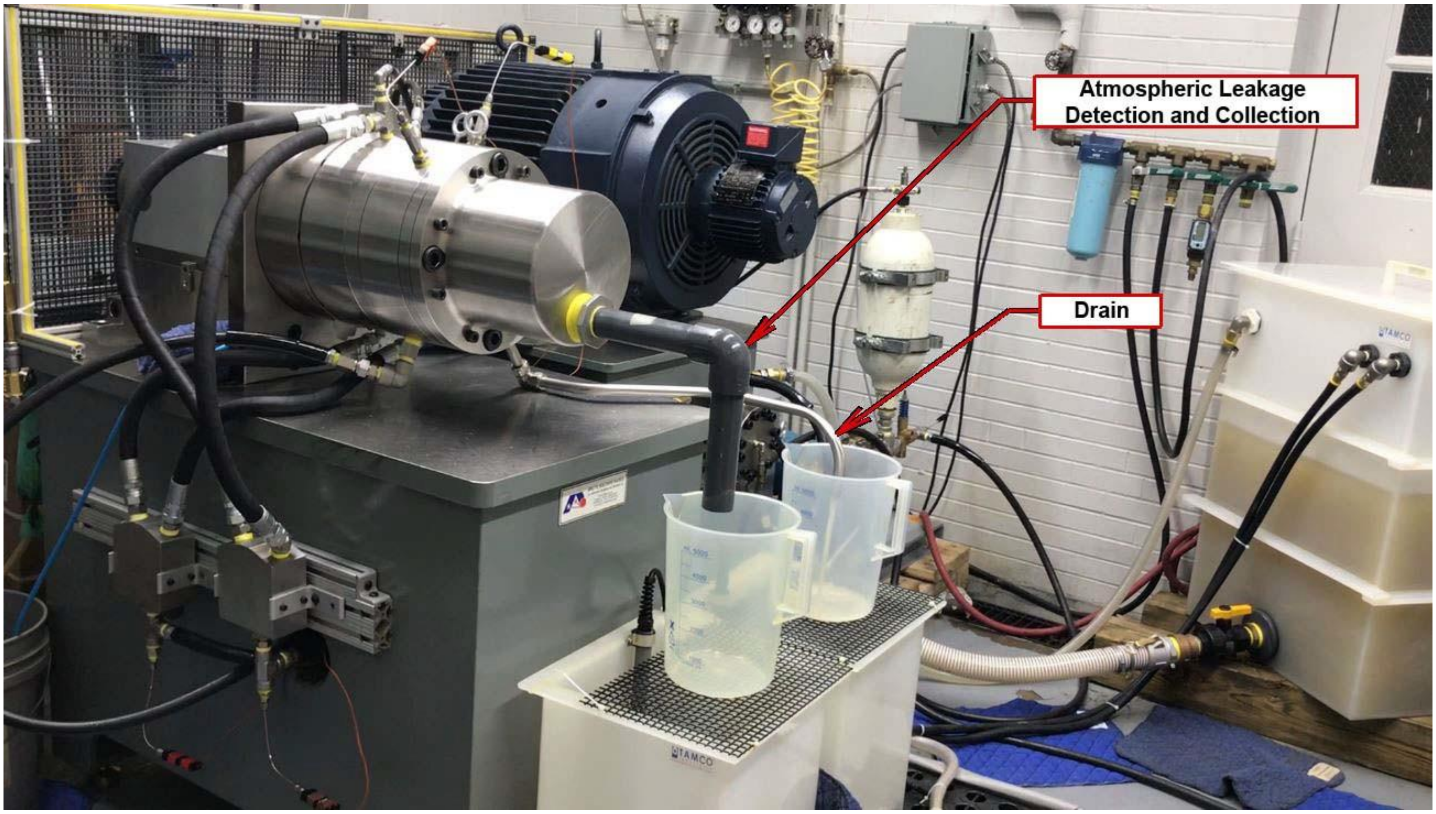

Figure 7 Photo of Test rig for full pressure dynamic testing of the 66A bushing arrangement

Requirement 1), allowing all normal leakage past the primary bushing without generating an alarm, is readily met by most if not all bushing arrangements given that the bushing is at best a flow restrictor and does not have capacity to perform as a mechanical seal. The extreme difference in flow vs pressure between a mechanical seal and a bushing is depicted in Figure 8 where bushing flow at various pressures is shown along with typical seal leakage. The bushing flows are estimates through a close clearance throttle bushing having a 0.002 inch (0.05 mm) radial clearance over a 6.625 inch (168 mm) diameter shaft for 70 °F (21 °C) oil having a viscosity of 148 centipoise. The seal leakage estimates are high-end approximations based on recent test data for a mechanical seal in oil. In most instances the measured seal leakage was lower and sometimes undetectable. The high-end values are listed as the extreme worst case conditions since higher flows would be more likely to generate back pressure. Data in Figure 8 demonstrates the significant difference between high-end estimated seal leakage and bushing flow.

The data provides assurance that the bushing will easily allow all normal seal leakage to pass without an alarm. It also provides a comparison between the flow magnitudes where seal leakage is measured in units of milliliters per hour (ml/hour) while bushing flow is routinely estimated in gallons per minute (gpm) or liters per minute and is typically thousands of times greater than seal leakage.

During validation testing of the arrangement it was repeatedly shown that the inboard bushing would allow a flow of 30 ml/hour to pass through to drain without generating any detectable back pressure. 30 ml/hour was chosen as the test point since it is considered to be a case of very high leakage for a large pipeline mechanical seal similar to the seal illustrated in Figure 5 operating in crude oil.

Under normal operating conditions the flow past the primary bushing will be equal to seal leakage. Since this flow is handled without generating any detectable back pressure at the bushing, the pressure between the primary seal and the primary bushing will remain essentially zero, even when the bushing flow far exceeds normal seal leakage. Depending on the bushing design and type it will take a considerable amount of flow to generate a detectable pressure increase.

This disparity between the bushing flow and the seal leakage is the reason why the signal in the form of a pressure increase provided by the arrangement is relegated to the service of indicating catastrophic seal failure or at best extreme seal distress. This distress would still warrant pump shut down and would be accompanied by flow many times greater than normal seal leakage.

Pressure (psig)	Pressure (Barg)	Bushing Flow (gpm)	Bushing Flow (ml/hour)	Seal Leakage (ml/hour)	Seal Leakage (gpm)
0	0	0	0	0	0.00E+00
100	6.9	0.0672	15263	2	8.66E-06
200	13.8	0.1342	30480	3.2	1.40E-05
400	27.6	0.2684	60960	6.2	2.74E-05
600	41.4	0.4027	91463	10.1	4.43E-05
800	55.2	0.5368	121921	14.7	6.48E-05
1000	69	0.671	152401	20.1	8.87E-05
1200	82.8	0.8052	182881	26.4	1.16E-04
1440	99.3	0.9662	219448	34.9	1.54E-04
1800	124.1	1.2076	274276	49.9	2.20E-04
2200	151.7	1.4759	335213	69.6	3.06E-04

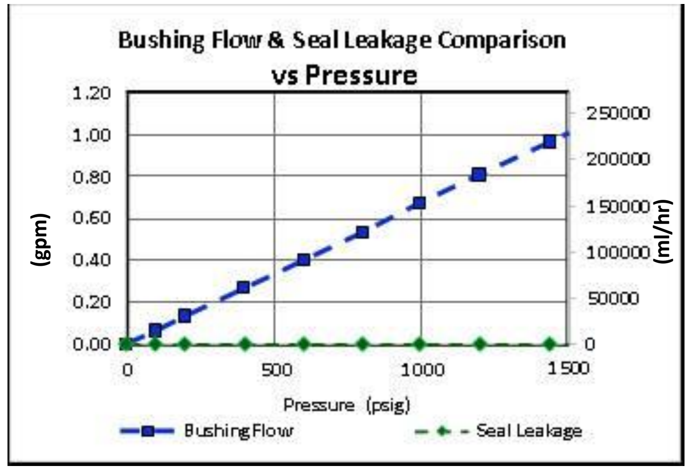

Figure 8 Estimates comparing close clearance throttle bushing flow and seal leakage

Some effort has been put forth focused on examining the capacity of the bushing to provide a back pressure which would give an indication of seal health with the aim of providing an early warning to seal failure. This effort focused on examining the relationship between pressure and flow for a typical bushing. The results of this investigation showed that flows which could be considered a precursor to seal performance issues would not generate a readily detectable pressure signal as primary bushing back pressure. Even if a transducer sensitive enough to detect the extremely low pressure were employed, the resulting flow would likely be too variable due to tolerances and operational variances to provide value in the form of useable data. Figure 9 & 10 show the test rig for this investigation. Figure 11 shows data for the relationship between pressure and flow collected during this investigation.

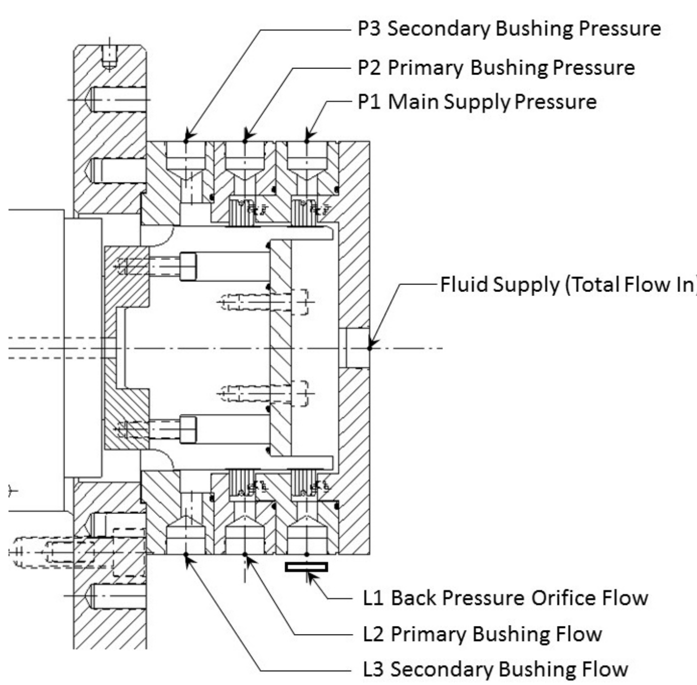

Figure 9 Test Rig used for low pressure versus flow investigation of typical bushing

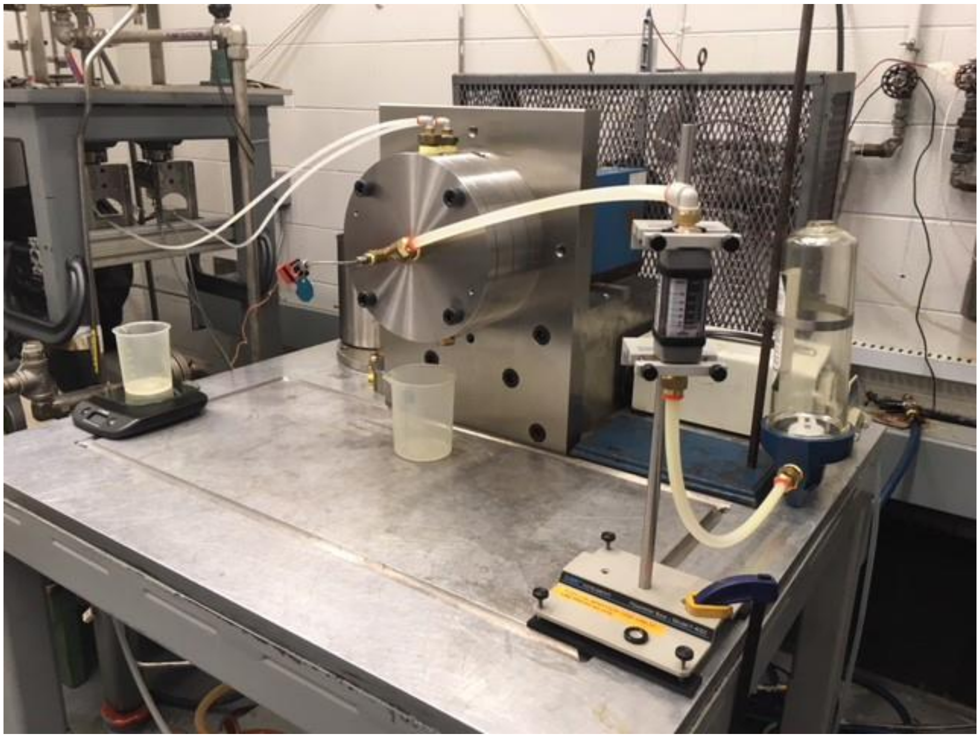

Figure 10 Photo of test rig used for low pressure versus flow investigation of typical bushing

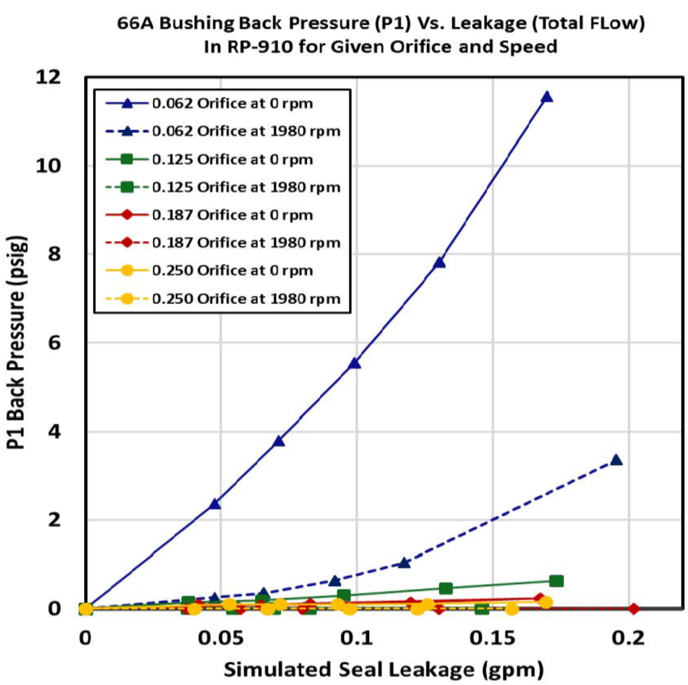

Figure 11 Test data showing low pressure versus flow for a typical bushing

If an early warning to seal performance issues is a functional requirement then an alternate seal or piping strategy may be recommended. Given that seal performance is more suitably monitored by leakage trends, a system capable of monitoring seal leakage rates would seem more appropriate. One scenario would involve a reservoir of a known volume which would trigger a drain to sump when filled to a known volume. API piping plans 65A and 65B could be used for this very purpose since they specify the use of a level transmitter on a reservoir to trigger alarm and/or reservoir drainage. The frequency of triggering drainage to sump over a given duration could provide data on the seal leakage trends which could be evaluated with respect to pump operational and environmental conditions in order to provide a more accurate indicator of seal health. Alternatively newer more advanced methods incorporating developed sensor technology may be available to monitor seal health and support a planned shut down and maintenance schedule.

Requirement 2), keeping normal leakage from exiting the pump and directed to drain, was shown to be readily achievable. During both the initial investigation into the pressure flow characteristics and during the subsequent bushing development tests, normal seal leakage was directed to drain with no flow bypassing the secondary bushing. Here we mention that flow rates past the secondary bushing were expected to be of a magnitude best measured in drops per minute. In order to expedite the measurement of low flows and promote ease of detection the outboard rig volumes were “primed” by circulating oil through to the atmospheric collection port thus filling any cavities and voids where oil might pool internally. This helped to ensure that any new leakage introduced during testing would be detected and not just settle into an internal pool giving a false measurement of no leakage.

The ability to keep leakage in the pump is critically linked not only to the performance and design of the secondary bushing, but also to the design and drainage capacity of the bushing housing and surrounding hardware. Though not conceptually novel the importance of hardware design utilizing a gravity fed drain is very important to keep the liquid from exiting the pump case and directed to drain. A hardware configuration with a large annulus and maximized bottom drain port was tested and shown to mitigate leakage past the secondary bushing under normal operating conditions simulating a high-end primary seal leakage of 30 ml/hr. The arrangement was also shown to successfully direct all flow to drain when simulating the operation of a distressed seal leaking 30 ml/minute.

Requirement 3), providing a signal in the form of a pressure increase when the primary seal is compromised, is readily demonstrated with various bushing designs. The magnitude of the pressure increase with respect to the flow, in this case flow past the primary seal, can be extremely variable as previously noted. In order to achieve the goal of providing a signal in the event of seal failure the pressure alarm need only be set to a few psig. In some cases a pre-start test can reveal the pressure flow characteristics of the bushing. During testing a low pressure fluid from 0 to 10 psig (0 to 0.7 barg) was introduced to the cavity between the primary seal and the primary bushing while monitoring the flow to drain. This test revealed that significant flow, equivalent to that bypassing an extremely distressed seal, would be realized at a few psi under dynamic conditions. Though pressure versus flow data was obtained during testing significant variance was demonstrated among tests. This variance, to be expected across different installed component sets, limits the value of the data and restricts its significance to reinforcing the low pressure alarm setting. Data for flow as a function of low pressure collected during various tests of the same design is shown in Figure 12.

Low Pressure Flow to Drain (ml/hr) for Various Tests						
Pressure PSIG	**Test 1**	**Test 2**	**Test 3**	**Test 4**	**Test 5**	**Test 6**
0	**0**	**0**	**0**	**0**	**0**	**0**
1	**8700**	**3600**	**6300**	**4500**	**1320**	**7200**
2	**13200**	**5040**	**8700**	**6000**	**2400**	**15240**
3	**17040**	**7680**	**11400**	**8700**	**3840**	**20400**
4	**23400**	**9600**	**15000**	**12000**	**5520**	**26400**
5	**26400**	**12000**	**17700**	**13200**	**7200**	**29400**
6	**30000**	**14400**	**19500**	**14100**	**7800**	**34200**
7	**33000**	**14880**	**21900**	**16500**	**8400**	**38040**
8	**35220**	**16320**	**23700**	**18300**	**10800**	**40800**
9	**39300**	**19200**	**25500**	**20100**	**11400**	**43200**
10	**42300**	**21120**	**27720**	**21600**	**12600**	**46800**

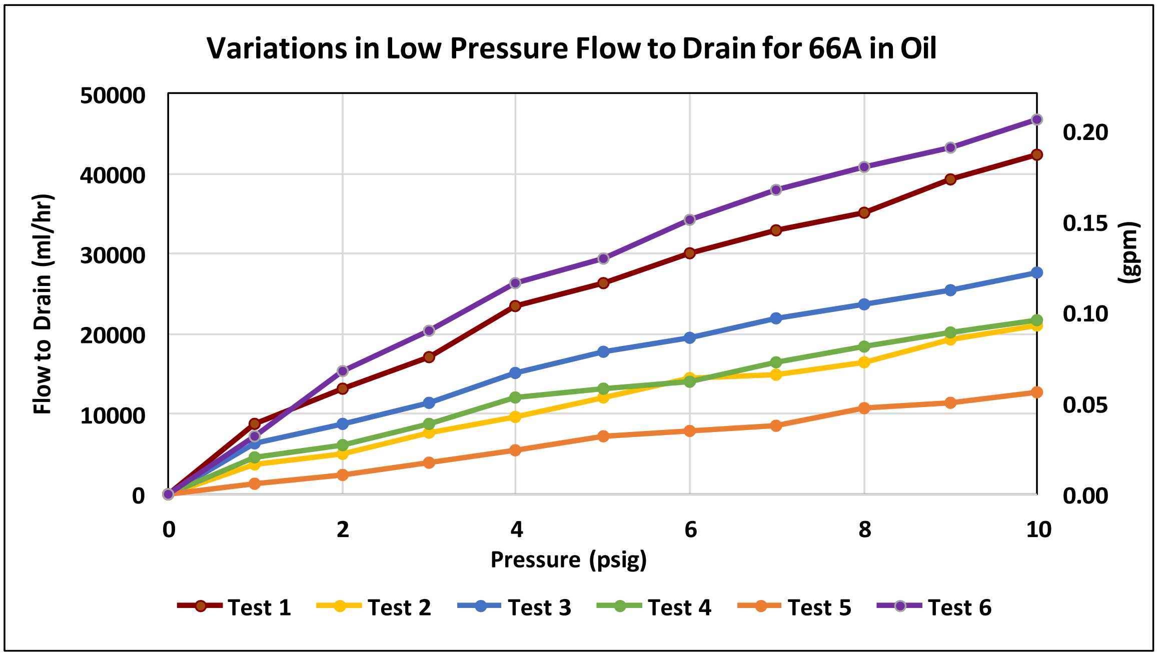

Figure 12 Test data and graph of low pressure flow to drain for the 6.625 inch bushing at 1800 RPM in oil

Requirement 4), restricting leakage past the primary bushing in the event of catastrophic seal failure during full pressure dynamic operation can prove to be challenging for various bushing designs. Full pressure test conditions included a pressure of 1500 psig (103 barg) and shaft speed of 1800 rpm for the 6.625 inch (168 mm) test diameter. The concept of providing restriction to flow would seem quite simple but in this instance complications arise challenging several different aspects of design.

If we look at the applied pressure in conjunction with the surface speed we see that, similar to an unbalanced shaft seal, the Pressure-Velocity (PV) on the primary bushing can exceed 3 million psi-ft/min (1050 bar-m/sec). By contrast a typical hydraulically balanced mechanical seal of a similar size operating at 1800 rpm would have to be pressurized to 3500 psig (241 barg) to experience a similar PV value. The operational element permitting the bushing to function at this PV is the minimal duration for dynamic performance. Dynamic performance at the maximum PV is only experienced between the time of primary seal upset and shut down commencement. This is expected to be a less than a minute. The PV would then gradually decrease as the shaft rotation slows and stops. During validation testing the primary bushing was subject to 1500 psig (103 barg) at a shaft speed of 1800 rpm for a period of two minutes. The shaft speed was then gradually reduced from 1800 to 0 rpm over a period of 3 minutes while maintaining full pressure on the bushing. These extreme PV values were shown to cause damage to some bushing designs. Damage was compounded when bushings were subject to extended dynamic runs under variable operating conditions. Extreme PV values also resulted in high heat generation and elevated temperatures which placed some restriction on the duration of dynamic testing.

When the primary bushing is subject to process pressure it will deform in a manner which will cause it to collapse onto the sleeve. This pressure collapse and the resulting contact force result in an increase in friction between the bushing and rotating sleeve. Similarly the applied pressure will load the bushing axially against the adjacent stationary housing resulting in an increase in friction at this location. The relative motion between the shaft and housing with the bushing loaded between them results in a braking affect with the bushing subject to torque related stress loading. Some less robust bushing designs were shown to fracture under this load condition.

The ideal design would survive full pressure dynamic operation and shutdown while minimizing flow with no loss of structural integrity. The final configuration successfully survived full pressure dynamic validation testing while restricting flow with virtually no damage. Post test component photos are shown in Figure 13.

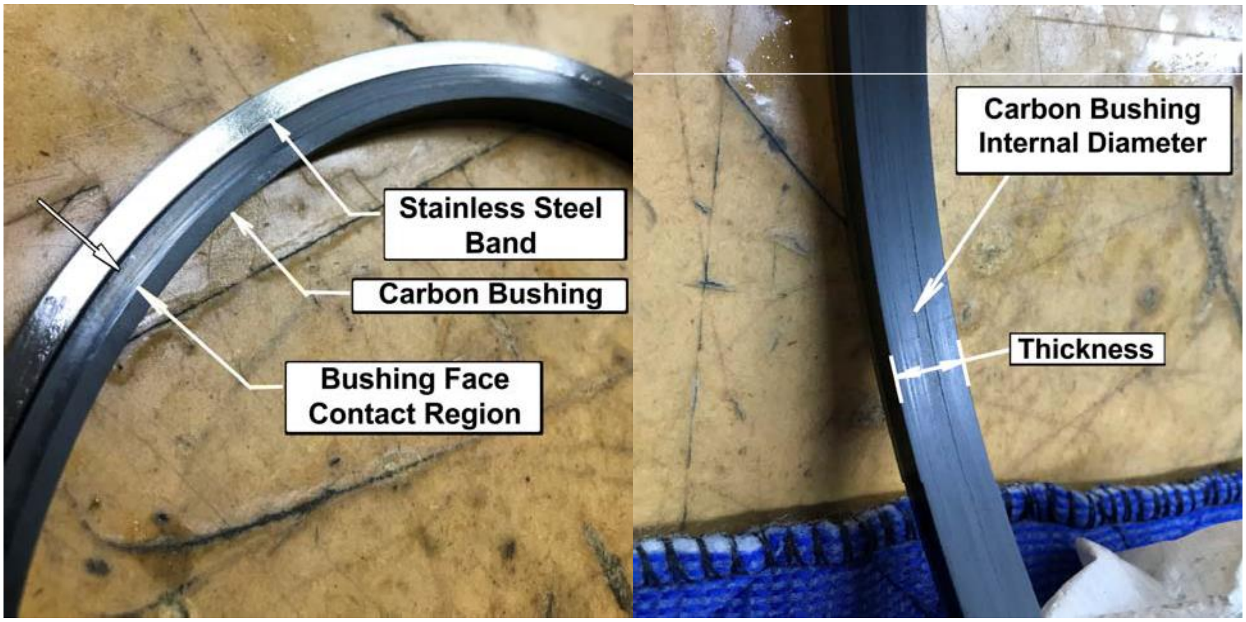

Figure 13 Post catastrophic containment test photos of the primary bushing face and internal diameter

The successful design will not only restrict flow past the primary bushing, it will also limit the magnitude of this flow to a low enough level as to prevent any pressurization at the drain cavity. It is noted that the capacity to handle increased flow past the primary bushing while not increasing pressure at the drain is linked to the external piping sizes. For this reason maximum pipe diameters are recommended. ¾ inch (19 mm) diameter stainless tubing was used on the rig drain during testing. Several bushing designs and modifications were tested and all exhibited some observable flow when subject to very low pressure. Though the capacity to restrict flow and contain process during full pressure dynamic operation is demonstrated, the 66A bushing arrangement should not be expected to function as a mechanical seal.

Bushing flow rate limitations help to ensure that the outboard bushing in conjunction with hardware design will contain the majority if not all of the fluid within the pump case and direct it to drain. It is noted and was demonstrated during testing that the effectiveness of the arrangement in meeting performance target 4) will directly affect its ability to meet target 6), restricting leakage past the secondary bushing. Since no restriction was placed on the amount of flow permitted to drain, the determination of success in meeting target 4) is determined by successful completion of target 6). Ultimately as flow to drain increases the chance of experiencing flow past the secondary bushing increases. Reference Figure 14 for primary and secondary bushing flows during test. Reference Figure 15 for digitally recorded test conditions.

Requirement 5), restricting leakage past the primary bushing in the event of primary seal catastrophic failure during full pressure static operation after pump shut down and during pump isolation was demonstrated during testing. Once again it is noted and was demonstrated during testing that the effectiveness of the arrangement in meeting performance target 5) will directly affect its ability to meet requirement 6). Post catastrophe static flow rates were shown to vary with time. This likely resulted from thermal affects where post catastrophe temperature changes caused dimensional variations which affected clearances and the ability to restrict flow. Reference Figure 14 for flow during post catastrophe simulation.

Requirement 6), restricting leakage past the secondary bushing in the event of primary seal failure such that the vast majority of fluid is directed towards drain during static and dynamic operation, was demonstrated in conjunction with requirements 4 & 5. The effectiveness of the design in meeting this requirement is also shown in Figure 14 as Secondary Bushing data during Catastrophic Failure Simulation and Post Catastrophe Simulation.

66A Oil Test									
Distressed Seal Simulation									
		Primary Bushing				Secondary Bushing			
Time	Shaft Speed (rpm)	Pressure (psig)	Flow to Drain (ml)	Duration (seconds)	Rate to Drain (ml/hr)	Pressure (psig)	Flow to Atm. (drops)	Duration (seconds)	Rate **(ml/hr)
13:20	1800	0	110	300	1320	0	0	300	0
13:25	1800	0	110	300	1320	0	0	300	0
13:30	1800	0	110	300	1320	0	0	300	0
Low Pressure Dynamic Operation									
		Primary Bushing				Secondary Bushing			
Time	Shaft Speed (rpm)	Pressure (psig)	Flow to Drain (ml)	Duration (seconds)	Rate to Drain (ml/hr)	Pressure (psig)	Flow to Atm. (drops)	Duration (seconds)	Rate **(ml/hr)
13:40	1800	1	145	60	8700	0	0	60	0
13:43	1800	2	220	60	13200	0	0	60	0
13:46	1800	3	284	60	17040	0	0	60	0
13:49	1800	4	390	60	23400	0	0	60	0
13:52	1800	5	440	60	26400	0	0	60	0
13:55	1800	6	500	60	30000	0	0	60	0
13:58	1800	7	550	60	33000	0	0	60	0
14:01	1800	8	587	60	35220	0	0	60	0
14:04	1800	9	655	60	39300	0	0	60	0
14:07	1800	10	705	60	42300	0	0	60	0
Catastrophic Failure Simulation									
		Primary Bushing				Secondary Bushing			
Time	Shaft Speed (rpm)	Pressure (psig)	Flow to Drain (ml)	Duration (seconds)	Rate to Drain (ml/hr)	Pressure (psig)	Flow to Atm. (drops)	Duration (seconds)	Rate **(ml/hr)
14:17	1800	1500	5000	107	168224	0	0	120	0
14:19	1800-0	1500	7500	198	136364	0	0	180	0
Post Catastrophe Simulation									
		Primary Bushing				Secondary Bushing			
Time	Shaft Speed (rpm)	Pressure (psig)	Flow to Drain (ml)	Duration (seconds)	Rate to Drain (ml/hr)	Pressure (psig)	Flow to Atm. (drops)	Duration (seconds)	Rate **(ml/hr)
14:23	0	1500	6850	300	82200	0	0	30	0
14:28	0	1500	9340	300	112080	0	0	30	0
14:33	0	1500	10550	300	126600	0	0	30	0
**Approximate conversion is 20 drops per ml									

Figure 14 Bushing flow rates measured during testing

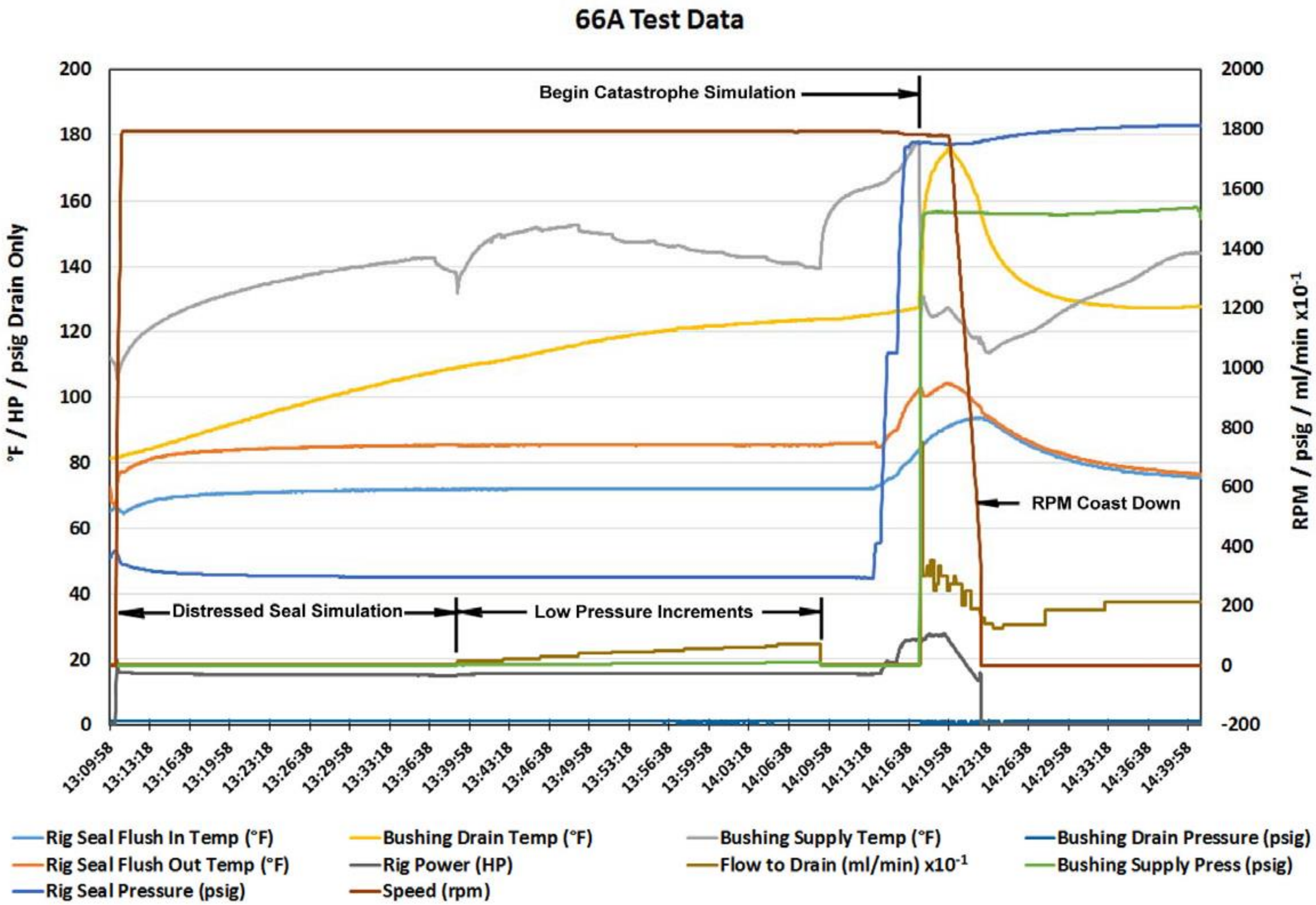

Figure 15 Operating conditions recorded during testing

The ability to meet requirement 7), operate for 3 years while maintaining functional capacity was demonstrated by subjecting the bushing to an extended period of normal operation with periodic dimensional checks to evaluate wear. The measured wear rate could then be extrapolated out to a 3 year time period to verify that critical dimensions linked to functional capacity would be maintained. The time period for the test was 500 hours. During the 500 hour test 20 to 30 ml/hour of simulated seal leakage was introduced between the rig seal and the primary bushing. At no time was pressure detectable between the seal and the bushing. The drain pressure was maintained at 0 psig (0 barg). The rig shaft rotation was stopped and restarted 1200 times using a 30 second speed ramp down and a restart to full speed in under 3 seconds. These restart cycles were intended to simulate a typical number of pump stops and restarts during 3 years of operation. The restarts are expected to exacerbate wear on the carbon bushings. Since wear could potentially be exhibited as a non-linear function of time the primary and secondary bushing condition and dimensions were inspected after 100 hours of operation and after 250 hours and again after 500 hours. Finally, after the 500 hour run the dual arrangement was subject to a full pressure, full speed catastrophe simulation. During catastrophe simulation the acceptable containment capacity was demonstrated with flow rates similar to results shown in Figure 14. The bushings exhibited virtually no wear even after 500 hours of normal operation. The lack of wear provides assurance that three years of service under normal operating conditions is feasible and likely. Measured wear is shown in Figure 16 where variance is likely the result of measurement accuracy. Photos of the internal diameter and flat outboard surface after the 500 hour extended run are shown in Figures 17 and 18.

Change to Inboard and Outboard Bushing Heights and Internal Diameters at Different Stages of the Endurance Test			
	100 Hour	250 Hour	500 Hour
Measurement Position	**Inboard Bushing Height Change (inches)**		
1	0	-0.0001	-0.0001
2	-0.0002	-0.0002	-0.0002
3	-0.0002	-0.0001	-0.0001
4	0.0001	0	0
	Inboard Bushing Internal Diameter Change (inches)		
1 - 3	0	0.0005	0.0005
2 - 4	-0.0005	-0.0005	-0.0005
	Outboard Bushing Height Change (inches)		
1	0.0001	0.0002	-0.0002
2	-0.0003	-0.0003	-0.0002
3	0	0.0001	0
4	0	-0.0001	-0.0001
	Outboard Bushing Internal Diameter Change (inches)		
1 - 3	0.0005	0.0005	0.0005
2 - 4	-0.0005	-0.0005	-0.0005

Change to Inboard and Outboard Bushing Heights and Internal Diameters at Different Stages of the Endurance Test		
100 Hour	250 Hour	500 Hour
Inboard Bushing Height Change (mm)		
0	-0.01	-0.01
-0.02	-0.02	-0.02
-0.02	-0.01	-0.01
0.01	0	0
Inboard Bushing Internal Diameter Change (mm)		
0	0.05	0.05
-0.05	-0.05	-0.05
Outboard Bushing Height Change (mm)		
0.01	0.02	-0.02
-0.03	-0.03	-0.02
0	0.01	0
0	-0.01	-0.01
Outboard Bushing Internal Diameter Change (mm)		
0.05	0.05	0.05
-0.05	-0.05	-0.05

Figure 16 Bushing dimensions measured during the 500 hour test

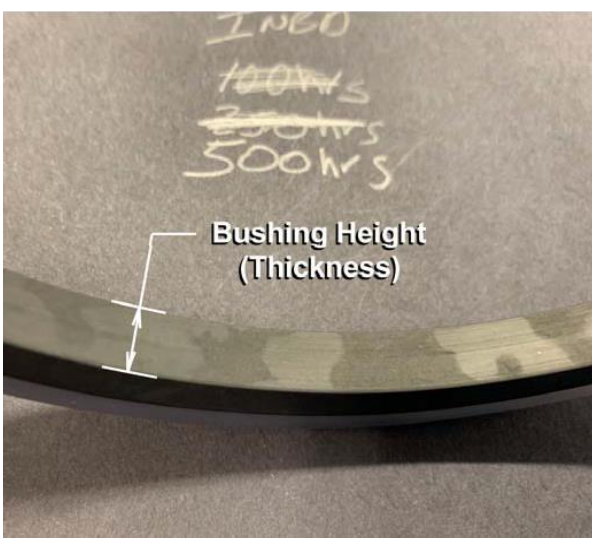

Figure 17 Photo of bushing internal diameter

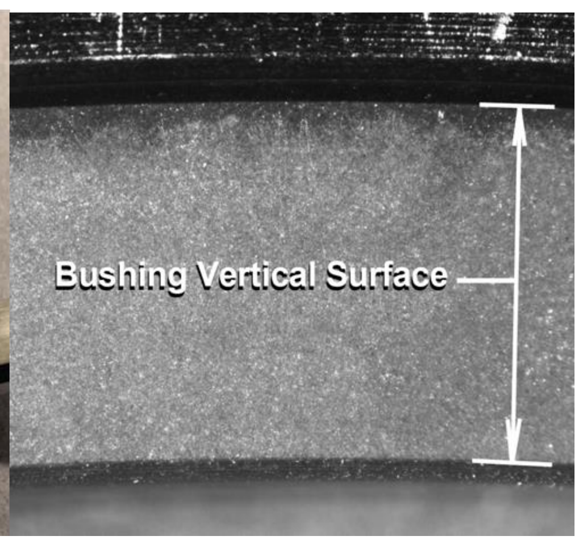

Figure 18 photo micrograph of bushing face

Requirement 8), maintaining structural integrity up to of 150% of maximum pressure during static operation (zero rpm) was tested and demonstrated successful. Pre and Post test visual and dimensional inspections reveal no damage or dimensional change to the bushing when subjected to a pressure of 2200 psig at 0 rpm. The results of the structural integrity test reveal that pressure loading which results in deformation of the primary bushing causing its collapse onto to the sleeve and axially pressing it against the bushing housing can close the leak paths resulting in near zero flow to drain during static bushing pressurization.

Requirement 9), restricting leakage past the secondary bushing in the event of primary seal failure and primary bushing failure is accomplished by design. Using the same bushing in the primary and secondary positions provides performance similarity between the positions. During catastrophic seal failure debris from a failed primary seal has been shown to migrate and cause damage to surrounding components. Depending on the material, amount, shape and size of debris, damage to downstream components can be significant to the point of jeopardizing their functional integrity especially when the damaging debris consists of hard carbide fragments from the primary seal faces. In the event that damage from the failed primary seal causes damage to the primary bushing, the secondary bushing will have full pressure operational capacity. Flow out of the pump case would be restricted to levels equal to primary bushing flows at similar pressures. Test data shows that full pressure flows may reach levels of several gallons per minute (in excess of 10 liter/min) though it is unlikely that the secondary bushing would experience full containment pressure in the vicinity of a generously sized drain.

CONCLUSION

The dual bushing design demonstrates the ability to meet the functional requirements expected when using the piping plan 66A. It can provide the elements of safety and containment when used to support a single mechanical seal. Though it should not be expected to provide an early warning to seal failure, the design successfully demonstrates the capacity to provide a warning signal in the form of a detectable pressure increase for triggering pump shut down and isolation activities when the primary seal is compromised or failing. By limiting the volume of crude escaping the pump during operational upsets, shut down and pump isolation, the design can effectively mitigate the need for clean-up operations and reduce environmental impact.

NOMENCLATURE

PV = Pressure-Velocity psi-ft/min (bar-m/sec)
API = American Petroleum Institute

FIGURES

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