Circulation Systems for Single and Multiple Seal Arrangements

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This discussion opens a three-part series covering mechanical seal piping plans that provide guidelines for various seal arrangements, fluids and control equipment to help you determine what support system requirements will maximize the performance reliability of your application.

Introduction

The American Petroleum Institute (API) created a numbering system for a variety of seal flush plans. The API flush plans are now located in API Standard 682 and the corresponding ISO standard, ISO 21049. The American National Standard Institute (ANSI) adopted a slightly different designation system.

General Information

These plans are utilized to provide the seal with the proper environment, depending upon the type of equipment used and the application the seal is exposed to. This series of articles discusses the basic flush plans, providing some general guidelines to be used along with the advantages/disadvantages of the plans, and, where appropriate, information on sizing and proper control of the system.

In this first part of the series, the basic concepts, criteria and considerations for circulation systems are established.

The various flush plans can be grouped by a variety of categories. One method for grouping is as follows:

API Plan	Description
01, 02	Internal system for single seals
11, 12, 13, 14, 22	Simple recirculation system for single seals
21, 23, 31, 41	Recirculation systems with auxiliary equipment for single seals
52, 53A-C, 54, 74	External systems for dual seals
32, 62	External injection systems
72, 75, 76	External control system for containment seals

These groups can have similarities in advantages/disadvantages, sizing of the system, and system controls.

Advantages and Disadvantages

The internal and recirculation systems have the advantage that the flush source comes from the pumpage and goes back to the pumpage, so no product contamination occurs. In addition, these flush plans, unlike an external injection, do not require any reprocessing of the product.

These same flush plans share the disadvantage that if the product pumped is not a good face lubricant, then the seal can become damaged. For some of the plans noted, circulation from the pump discharge back to pump suction or vice versa will decrease pump efficiency and increase power required for the application. The volume of flush is usually very small compared to the capacity of the pump and, therefore, the decrease in efficiency is very small.

Sizing

Generally, the flush rate must be calculated based on fluid

Table 1. Comparison of features of categories.

API/ISO	ANSI	General Description
01	7301	Internal recirculation from pump discharge.
02	7302	Dead-ended no circulation.
11	7311	By-pass from discharge to seal chamber.
12	7312	By-pass from discharge through strainer to seal chamber.
13	7313	Recirculation from seal chamber to pump suction.
14	7314	By-pass from discharge to seal and back to pump suction.
21	7321	By-pass from discharge through cooler to seal chamber.
22	7322	By-pass from discharge through strainer, orifice, cooler to seal chamber (not shown). Similar to Plan 21 with addition of a strainer.
23	7323	Recirculation from pumping ring through cooler to seal chamber.
31	7331	By-pass from discharge through cyclone separator to seal chamber.
32	7332	Injection from external source to seal chamber.
41	7341	By-pass from discharge through cyclone separator and cooler to seal chamber.
52	7352	Nonpressurized external reservoir with forced circulation.
53A	7353A	Pressurized external reservoir with forced circulation.
53B	7353B	Pressurized external bladder type reservoir with forced circulation. Has been known as Plan 53 Modified.
53C	7353C	Pressurized external piston type reservoir with forced circulation.
54	7354	Circulation of clean fluid from external system.
61	7361	Tapped connections only. Usually used for Plan 62 later. Not shown.
62	7362	Quench fluid from external source.
65	--	Single seal leakage alarm for high leakage.
71	7371	Tapped connections only. Usually used for Plan 72, 75, 76 later. Not shown.
72	7372	External buffer gas purge for secondary containment seals.
74	7374	Pressurized external barrier gas for Dual Gas Seals.
75	7375	Secondary containment seal drain for condensing leakage.
76	7376	Secondary containment seal drain for non-condensing leakage.

properties, system pressure, shaft speed, and seal size. See the “Flush Rates” section for more details.

System Control

The preferred method for controlling flow is with an orifice. The orifices should not be less than .125-in, unless the product is very clean and customer approval is obtained. Many small or low speed pumps have a low differential pressure and no orifice would be required in the piping. On the other hand, when the differential pressure is high, a single .125-in orifice would allow for more flow than desired. In such cases, multiple orifices, choke tubes or valves must be used to control flow.

Flush Rates

With few exceptions, any flush system works hand-in-hand with the hardware and seal components. If the seal is set up with a distributed or single point flush, and/or an enlarged bore seal chamber, the effectiveness of the system will be better and the seal will run cooler no matter how much or little the flush flow rate is.

Flush requirements for seals should be given in terms of a minimum and a recommended flow rate. Some seals can actually operate satisfactorily without a flush. Such applications usually involve non-volatile fluids at low pressures and low speeds. Heat transfers from the faces, through the liquid and into the metal surrounding the seal chamber. Analysis of these cases is beyond the scope of this article.

The minimum flush rate is necessary to obtain the performance rating given by the product technical bulletin; it is determined by an energy balance computation. The assumption is that heat generated by the seal faces is absorbed by the flush through ideal mixing. This raises the temperature of the flush. Typically, an increase of 15-deg F for water and low vapor pressure hydrocarbons, 30-deg F for lube oils, and 5-deg F for high vapor pressure hydrocarbons is allowed. Frequently, the minimum flush rate is relatively low, often less than 1-gpm.

Field experience indicates, and laboratory tests confirm, that seal performance generally improves when the flush rate is greater than the minimum. In particular, heat transfer usually improves and the average temperature around the seal decreases with increased flush rate; as a result, the face temperature and wear rate decreases. The recommended flush rate promotes these benefits.

The recommended flush rate should be based on experience with similar applications. Some considerations include performance goals and fluid properties as well as the design and interaction of the seal chamber, gland, flush plan and seal. In the absence of specific experience, a simple rule of thumb is: the recommended flush rate is the larger of 1-gpm per inch of seal size or the minimum flush rate.

Questions are sometimes asked about the maximum flush rate. Although increasing the flush rate beyond the recommended value may produce further improvements, by definition this effect is rapidly diminishing beyond that point. At very high flush rates and close clearances, erosion can occur. As an example, when sealing water at 250-psig using a balanced 2-in seal at 3600-rpm, the minimum flush rate might be computed as 0.4-gpm based on an allowable temperature rise of 15-deg F. The rule of thumb yields 2-gpm for a 2-in seal. Therefore, the recommended flush rate would be 2-gpm.

On the other hand, when sealing propane under the same conditions, the minimum flush rate is computed as 2.5-gpm based on an allowable temperature rise of 5-deg F. Thus, for propane the recommended flush rate would be 2.5-gpm.

Radial Flow Pumping Ring

Axial Flow Pumping Ring

Pumping Rings

Pumping rings are used in closed loop sealing systems such as Plans 23, 52, and 53A-C to produce flow through coolers and reservoirs. There are two basic pumping ring designs: radial flow and axial flow. Either design can be effective. Just as the performance of a centrifugal pump is a function of the impeller and volute, the performance of the pumping ring depends on the design of the seal chamber. In particular, the design, size and placement of the inlet and outlet ports are crucial to the performance of the pumping ring.

The first rule for pumping rings is to make the diameter of the inlet and outlet ports as large as possible. This includes the pipe tap and the final drill-through. There is no particular reason to make the outlet port smaller than the inlet port unless there is not enough space for both to be large. For radial flow pumping rings, a tangentially directed outlet port is absolutely essential. This requirement applies to all variations of radial flow pumping rings, including those with vanes, drilled holes, slots, paddle wheels, knurled surfaces, etc.

According to simple theory, there is no reason to expect any flow through a radially directed outlet port. In actual practice, a small flow rate, usually about 25 percent of the amount expected from a tangential outlet, is produced by a radial outlet providing that the outlet port is large.

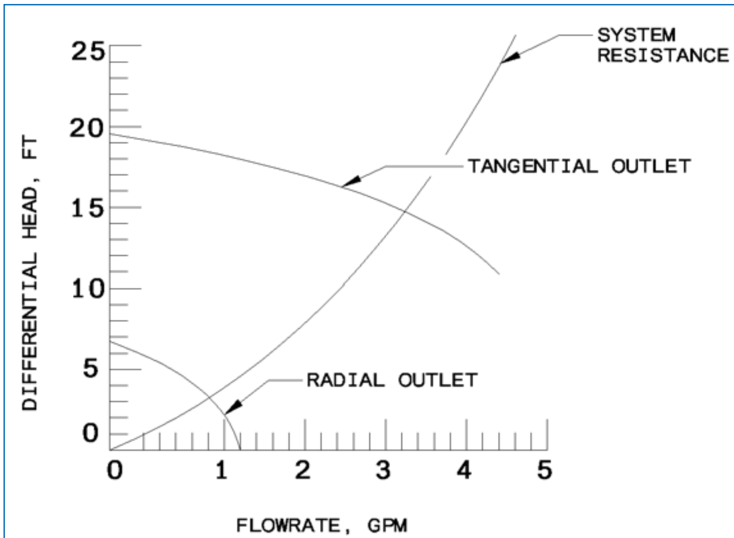
For axial flow pumping rings, the inlet and outlet ports must not be directly over the vanes of the pumping ring. There should be an inlet and outlet region at the ends of the pumping ring to assure even distribution of the liquid. Although a tangential outlet is not essential for axial flow pumping rings, significant performance improvements are realized when the outlet is tangential. In effect, an axial flow pumping ring with a tangential outlet becomes two pumping rings in series.

Unlike outlet ports, inlet ports can and should be radially directed. Just as is the case for centrifugal pumps, a tangential inlet would cause pre-rotation of the liquid, which would adversely affect performance.

In general, the pressure and flow from a pumping ring increases with diameter and shaft speed. Pressure and flow decrease with increased radial clearance.

The circulation rate in a seal system is a function of the fluid properties and system piping as well as the pumping ring. Small piping, numerous directional changes, and viscous liquids result in low flow rates. The procedure for estimating the circulation rate is to first construct a piping system (resistance) curve and then superimpose the pumping ring performance curve. The intersection of these curves defines the circulation rate.

When both the pumping ring and the system are properly designed, circulation rates of about .5-gpm to 1.5-gpm per inch of seal size are easily attainable.



Circulation Rates for Pumping Rings

Thermosyphon Systems

A *thermosyphon* is a closed loop system in which fluid flow is produced by gravity through the effects of temperature on density. This natural circulation results from the differential head that exists between the cold and hot sections of the system. The cold fluid has the greater density and displaces the hot fluid. The saying "warm air rises" is better described as "cold air sinks."

Thermosyphons can provide cooling for liquid sealing systems; however, great care must be taken because thermosyphon flow rates are small and easily stopped by bubbles from vaporization or dissolved gases. A single bubble that is about the same diameter as the piping can stop flow; this is called vapor-locking. To prevent vapor locking and maximize flow, large diameter piping, connections, and drill-throughs should be used. The cooler or reservoir should be 2-ft to 5-ft above the seal chamber. If thermosyphoning is not a concern, a cooler or reservoir height of 1-ft to 2-ft can be used as this will reduce the system resistance slightly. Liquid should flow "in the bottom and out the top" of the seal chamber. The system must be periodically, or continuously, vented. To assist in the thermosyphon effect, the return or hot piping leg should be insulated so that no cooling occurs in this line.

Because of the quirky and sensitive nature of thermosyphons, most specifications require a positive circulation using some type of pumping ring. Even so, the effects of thermosyphoning should always be considered when designing seal circulation systems. That is, the system should always be designed to promote thermosyphoning.

Quenches For High Temperature

A *quench*, as defined by API 682, is "a neutral fluid, usually water or steam, introduced on the atmospheric side of the seal to retard formation of solids that may interfere with seal movement." Nitrogen is another quench medium.

In high temperature services, a steam quench may provide several benefits:

- Retard formation of solids.
- Wash away solids that do form.
- Provide cooling during normal operation.
- Provide heating before startup.

Nitrogen quenches, based upon general observations, are not as effective as steam for quenching high temperature seals. Product decomposition ("coking") is related to temperature. Not only does coke form more quickly in hot pumps, but it also forms more quickly around seals that run hot because of heavy load or inadequate flushing.

Steam quenches can be used with either rotating seal heads or stationary designs. Quenches on rotating seals, sometimes called a "steam blanket", is not particularly effective because very little steam is circulated within the quench area. Depending upon the type of bushing used, the steam can even be directed towards the pump bearings. A steam quench used with a stationary design, such as the Type 1604 (metal bellows seal), is more effective. The steam must enter underneath the bellows assembly, between the bellows and the anti-coking baffle, and is guided around the seal to wash away the leakage from the seal faces.

Care should be taken to make the drain port as accessible as possible with as large a "drill-through" as possible to prevent the drain hole in the gland from clogging up with coke. On a design like the Type 1604, if a quench is not going to be used then the baffle should be removed or modified, as this will provide additional clearances to counteract the accumulation of solids.

Determining the Quench Rate

Four considerations determine the recommended quench rate:

1. Is a quench required to improve MTBPM?
2. Minimum rate to purge the quench volume of the gland.
3. Minimum rate based on velocity to wash away leakage.
4. Minimum rate for cooling leakage below decomposition temperature.

Table 2. Quench rates for typical high temperature pumps.

Pump Temperature ºF	Steam Requirement 250ºF Saturated Steam CFM	Steam Requirement 250ºF Saturated Steam lb/hr	Nitrogen Requirement 100ºF Supply CFM	Nitrogen Requirement 100ºF Supply lb/hr
400	0.3	1.0	0.2	1.0
500	0.8	1.5	0.4	1.0
600	1.2	2.4	0.6	1.1
700	1.7	3.4	0.8	1.6

Note: Rate is volume per inch of seal size.

__Is a quench required to improve MTBPM?__

For high temperature hydrocarbon services, the general guideline is to apply a steam quench if the pumping temperature is above 350-deg F. The relative effectiveness depends upon many variables, but quenches used on lower temperature services have a reduced effect on extending MTBPM, other things being equal.

__Minimum Rate based on Purging__

If a quench is to be applied, then the minimum quench rate can be thought of as a purge. In that case, the minimum rate is a function of the volume being purged and the leakage being diluted. For typical seal gland plates and a contingency plan for high leakage rates, dilution of leakage usually governs.

__Minimum Quench Rate Based on Washing__

Another consideration is that the quench should wash away the leakage. This is based upon the quench rate with a certain velocity thru the quench area. The velocity should be in the range of 10-fps to 15-fps through the flow area to be effective. This consideration may call for more quench than the consideration for purging.

__Minimum Quench Rate Based on Cooling__

Steam is usually readily available in plants and the flow rates are typically not regulated very closely due to the availability. This is also due, in part, to the cost versus other quench media. The relative cost of quench media is:

Water = 1 (datum) Steam = 0.005
Plant nitrogen = 0.006 Bottled nitrogen = 1.4

The cooling effect of gases such as steam and nitrogen on the face temperature of hot seals is small. The order of magnitude is less than 500-btu/hr removed from the seal faces. If the quench rate is too small, the temperature of the quench will heat up to nearly the pump temperature and allow decomposition and coking to occur. To prevent this, the average temperature in the quench volume can be estimated from an energy balance using the seal leakage rate, quench flow rate and heat soak from the surrounding metal. By constraining this average temperature to be less than some critical "coking" temperature, the quench rate can be computed.

Recommended Quench Rate

After all the above considerations, the recommended quench rate is the largest of the values. For most pump seals the recommendation can be simplified per Table 2.

__Controlling the Quench__

The recommended quench rates are low enough that the flow rate may be somewhat difficult to control with any accuracy. There is little need for precision, especially with steam. Table 3 shows the approximate flow rate through a simple orifice, for steam and nitrogen.

Water is typically used as a quench medium when the fluid being sealed has solids in solution or will crystallize upon exposure to atmosphere. The flow rate for water does not have to be very large. In some cases it can just be enough to keep a volume of fluid on the atmospheric side of the seal, while in other cases a slight flow rate of .125-gpm to .250-gpm is sufficient to prevent build up of product underneath the seal faces. This is one case where the containment device may be a lip seal.

Secondary Containment Seals

Plans 71, 72, 75, and 76 are new plans for dry running secondary containment seals used in conjunction with a liquid lubricated primary seal. The process, or inner seal, of the dual unpressurized arrangement usually has its own flush plan. For example, the flush plans for a dual unpressurized seal arrangement with a dry running secondary containment seal might be written as Plan 11/71, 11/71/75, 11/71/76, or as noted below 11/72/75 or 11/72/76. The Plan 11 for the inboard seal can be any of the plans normally associated with a single

SPECIFICATIONS vb7 REMARKS

ITEMS IN RED INDICATE A SIGNIFICANT CHANGE FROM THE VB3000 INSTRUMENT.

ALL SPECIFICATIONS ARE SUBJECT TO CHANGE WITHOUT NOTICE PENDING FINAL ENGINEERING APPROVAL **14 March 2007**

SPECIFICATIONS	vb7	REMARKS
Sensor Input	2 channels	Simultaneous sampling
Sensors	Accelerometer, velocity, displacement, current	
AC coupled range	16 V peak-peak	Allows for ± 8 V sensor output swing (± 80 g)
DC coupled ranges	0 V to 20 V, -10 V to 10 V, -20 V to 0 V	E.g. for reading prox-probe gap
Connectors	BNC	Safety feature: break-free inline connector
Analog to digital conversion	24-bit ADC	
Sensor excitation current	0 mA or 2.2 mA (configurable), 24 V maximum	2.2 mA required for ICP® - type accelerometer
Sensor detection	Warns if short circuit or not connected	
Tachometer		
Sensor	Laser sensor with reflective tape included in kit	Sensor triggers when the tape reflects its beam
Laser sensor range	10 cm to 2 m nominal	Dependent on size of reflective tape
Other sensor types supported	Contact, TTL pulse, Keyphasor®	Instrument has optically isolated input
TTL pulse rating	3.5 V (4 mA) min, 28 V (6 mA) max, off-state 0.8 V	
Power supply to sensor	6 V to 8 V, 50 mA	Battery voltage with current limit
Keyphasor® threshold	13 V ± 1 V	
Speed range	30 RPM to 300 000 RPM (0.5 Hz to 5 kHz)	
Parameter Indication		
Maximum levels	>1000 g (10 000 m/s2), >1000 in/s (25 000 mm/s), 100 in (2500 mm), >10 000 amps	Effective limit is sensor sensitivity and output voltage
Dynamic signal range	> 95 dB (typical at 400 line resolution)	Acceleration and velocity
Harmonic distortion	Less than -70 dB typical	Other distortions and noise are lower
Units	g or m/s2, in/s or mm/s, mil or mm or µm	0-peak, peak-peak or RMS. Auto-scale by 1000x when required
AdB, VdB, amps	US and SI options for both adB and vdB	
Magnitude and cursors	Overall RMS value, dual cursors, harmonics	Digital readouts on chart
Accuracy	± 1% (0.1 dB)	For DC level (%F.S.) and AC measured at 100 Hz
Frequency response	± 0.1 dB from 10 Hz to 15 kHz; ± 3 dB from 1 Hz to 40 kHz	Acceleration and velocity. From value measured at 100 Hz High freq response also applies to DC ranges
Spectrum Display		
Fmax possible ranges	25, 50, 100, 125, 150, 200, 300, 400, 500, 600, 800 Hz 1, 1.2, 1.6, 2, 2.5, 3, 4, 5, 6, 8, 10, 15, 20, 30, 40 kHz	Or equivalent CPM values Or orders-based from 1X to 999X
Fmin possible range	0 to Fmax	Instrument zeroes all spectral lines below Fmin
Resolution	400, 800, 1600, 3200, 6400 lines	3200 lines max for dual channel measurements
Frequency scale	Hz, CPM, orders	Linear scale with zooming
Amplitude scale	Acceleration, velocity, displacement or current	Linear or log scales, auto or manual scaling.
Window shapes	Hanning, rectangular	
Overlap	(0, 12.5, 25, 37.5, 50, 62.5, 75, 87.5) %	Dependent on Fmax and number of lines
Number of averages	1, 2, 4, 8, 16, 32, 64, 128	Increases sampling time proportionally
Averaging types	Linear, exponential, peak hold, synchronous	
Demodulation bandwidths	20 bandwidth options	From 125 Hz to 1250 Hz up to 16 kHz to 20 kHz
Waveform Display		
Number of samples	1024, 2048, 4096, 8192, 16 384	
Time scale	10 ms to 256 seconds	Or orders based from 1 to 999 revs
Time synchronous averages	1, 2, 4, 8, 16, 32, 64, 128	Only available when tachometer triggered
Long time waveform	Up to 10 kHz Fmax	
Logging Features		
Output formats	LCD screen, transfer to Ascent PC-based software	
Data storage	1 GB non-volatile flash memory	Virtually unlimited recording storage
Data storage structure	Folders / machines / points / locations / routes	No limits are applied, 50 character names
Max folder size	10 000 measurement locations	
Keypad entry value range	± 999 999.999 999	50 character prompt string
Balancing	Planes 1, 2	
Speed range	30 RPM to 60 000 RPM	
Measurement type	Acceleration, velocity, displacement	
Weight modes	Angle 0° to 360°, fixed position, circumference arc	E.g. weights on fan blades, linear dist around circumference
Manual data entry	Yes	Allows re-entry of previous balance jobs
Storage	Against machines in data structure	No limits are applied
Display and Communication	Graphic grayscale LCD	
Resolution	480 x 320 pixels (HVGA)	
Viewing area	4.6" x 3.1" (117 x 79) mm	
Backlight	White LED, 4 V, 100 CD/m²	
Communications with PC	USB and Ethernet	PROFLASH allows instrument software to be upgraded
Battery and Charger		
Battery type	Custom Lithium Ion pack, 7.4 V, 4500 mAh	
Operating time	10 hours	Backlight on (60 second timeout)
Charger type	Internal charging, automatic control	External power pack 12 V DC, 3 A output, included in kit
Charge rate	3 A nominal	3 hours for complete charge
Mechanical		
Size	9.9" W x 5.8" L x 2.4" H (252 x 148 x 60) mm	
Weight	2.6 lb (1.2 kg)	Including strap
Environmental		
Operating temp	14 °F to 122 °F (-10 to 50) °C	
Storage temp and humidity	-4 °F to 140 °F (-20 to 60) °C, 95% RH	
EMC	EN61326	
Ruggedness	MIL-STD-810F-IV, 4' (1.2 m) drop onto concrete	Procedure: MIL-STD-810F-IV

Table 3. Approximate flow rates through orifices. (ACFM = Actual Cubic Feet per Minute)

	Steam Flow, ACFM Orifice Size, inch				Nitrogen Flow, ACFM Orifice Size, inch			
Differential Pressure, psi	1/16	3/32	1/8	3/16	1/16	3/32	1/8	3/16
1	0.4	0.9	1.6	3.6	0.3	0.6	1.1	2.5
2	0.6	1.2	2.2	4.9	0.4	0.8	1.5	3.4
3	0.7	1.5	2.7	6.2	0.5	1.0	1.9	4.3
4	0.8	1.7	3.1	6.8	0.6	1.2	2.1	4.7
8	1.2	2.5	4.4	9.9	0.8	1.7	3.0	6.8
16	1.5	3.6	6.2	14.0	1.0	2.4	4.3	9.7

mechanical seal.

A secondary containment device is a means of containing and controlling the primary seal leakage from a mechanical seal. In contrast to a dual liquid lubricated mechanical seal, which operates in a buffer or barrier fluid, a secondary containment device operates primarily in the leakage from the process seal, although purges may be added.

There are many different types of secondary containment devices from simple bushings to mechanical seals. Leakage rates for the various secondary sealing devices can vary by several orders of magnitude. Selection of the secondary containment device and system will depend on the level of leakage to atmosphere that is considered acceptable as well as performance requirements for normal operation, upsets, and in the event of process seal failure.

By definition, the secondary containment device does not necessarily have the performance or rating of the primary seal; however, it may be able to temporarily tolerate seal cavity pressure and fluid in the event of a failure of the primary seal.

Large clearance devices like fixed bushings have the highest leakage rates; floating bushings with reduced clearance are much better. Floating segmented bushings have still lower leakage rates. Dry running mechanical seals, both contacting and non-contacting, may also be used as secondary containment devices and can approach the level of performance of a dual unpressurized liquid lubricated seal arrangement.

Purge Rates for Secondary Containment Seals

API Plan 72 is designed to have an inert gas purge through the containment seal area with the intent to reduce emission levels to the atmosphere. The purge gas mixes with leakage from the primary seal, thereby reducing the concentration of the hazardous fluid (liquid or gas). Leakage rates from the various types of containment devices will vary from high rates with bushings to low leakage rates with contacting face seals.

Leakage to atmosphere will also have a wide variation depending upon operating conditions, length of time in service and equipment conditions, as well as a myriad of other lesser considerations. When deciding on the purge rate, consideration should be given to the type of containment device, the flow rate past the orifice, the fact that excessive purge rates can dry out the sealing cavity and possibly decrease the life of contacting face seals, and that excessive containment seal cavity pressures can decrease the life of the containment sealing device, with the possible exception of non-contacting containment seals.

A simple rule of thumb is to have a flow rate on the order of ½ SCFM to the containment seal cavity. This relates to the rough flow rate for a 5-psi differential pressure across a 1/16-in orifice. This rate can be adjusted upwards or downwards depending upon the specific application.

Influence of Static and Dynamic Dual Gas Seal Leakage on Pumps

Even though leakage from dual gas seals is normally very low, the following issues related to pump design and installation may require attention, depending upon the seal duty:

1. Static gas leakage can displace the liquid in the pump and prevent start up. This is particularly relevant in the case of vertical standby pumps.
2. Dynamic pump performance can also be affected by a loss of pump efficiency, differential head and increased NPSHR (Net Positive Suction Head Requirements).

Summarized below are the background and recommendations to eliminate these potential problems.

Static Gas Leakage

Inboard static gas barrier leakage may be at a minimal rate, but in a vertical pump in a standby condition, or stationary in a stop/start batch process, barrier gas can collect in the pump casing and disable the ability of the impeller to prime on start-up.

Some exceptional horizontal installations also suffer the same circumstance when suction pipework originates from below the shaft centerline. Not all vertical pumps are vulnerable, as the sensitivity is dependent on the relative positions of the impeller and the suction inlet. Some in-line units using a

Plan 13 flush (in conjunction with a Plan 74 for the dual gas seals) have the ability to naturally vent through the suction valve, if the piping orientation permits.

It is customary to leave suction valves on standby pumps open. Static barrier gas leakage will eventually vent through this opening. On vertical pump installations, this venting is liable to occur in sudden and significant volumes when meniscus forces are broken. In these instances, there is a possibility that the volume of gas entering the main pump suction can be sufficient to affect the operation of the main pump, if it is of a low enough flow rating.

To accommodate these issues in vertical pump installations or horizontal pumps with non-venting suction lines, a provision for manual or continuous automatic venting of seal chambers must be incorporated within the total pump installation.

If for operational or hazard reduction reasons it is required to shut both the suction and the discharge valves and isolate a standby pump, it can be expected (as with any dual pressurized seal) that the pump casing stands the risk of becoming pressurized to the same pressure as the gas barrier source. Depending upon the effectiveness of the valve seats, the casing pressure could also rise to that of the pump discharge manifold, which might be in excess of the barrier gas pressure.

Field experience indicates, and laboratory tests confirm, that seal performance generally improves when the flush rate is greater than the minimum. In particular, heat transfer usually improves and the average temperature around the seal decreases with increased flush rate; as a result, the face temperature and wear rate decreases.

Even though the dual gas seal may have a reverse pressure design feature on horizontal units, it is possible that a small quantity of process fluid may contaminate the gas barrier chamber. This is not detrimental to the seal (unless the process crystallizes or hardens), but when restarting the pump there is a risk that this small volume of process fluid will be pumped through the outer seal to the atmosphere.

On horizontal installations requiring zero atmospheric emissions, which may be required to operate in a standby mode with the pump suction and discharge valves closed, it is necessary to connect the casing to a low pressure environment.

Dynamic Gas Leakage

Barrier gas leakage across the inner seal face during dynamic operation will eventually mix with the process flow. Depending on the seal size, operating conditions, pump size, pump design, and operation, this leakage can affect the seal's performance. This may be an increase in the NPSHR, a reduction in differential head, and in extreme cases a loss of prime.

At normal leakage levels this may not be an issue, but when leakage levels approach a condition when failure is deemed imminent, the effect on pump operation should be minimized. The seal size, shaft speed, barrier gas pressure, pump flow capacity, impeller design, and level of operational flow compared to the pump's design BEP (best efficiency point) are all factors that determine the effect on normal pump operation.

To prevent the likelihood of dual gas seal leakage in dynamic operation affecting the design pump performance, screening by consultants is advised on pumps operating between 40-gpm and 90-gpm, when operating at less than 50 percent of its BEP.

At high vacuum suction conditions the effect of dual gas seal leakage into the process fluid is exaggerated because the gas expands at the low pressure. This is not a normal pump operating condition, but on pump NPSHR proof testing it may occur. The normal measurement criteria of a loss of 3 percent in the head generated can be created by gas entrainment. In an NPSHR proof test with a low capacity pump design and dual gas seals, a conservative and inaccurate value may be indicated.

It is advised that if NPSHR proof tests are applied to pumps with a BEP capacity less than 40-gpm, the influence of gas seal leakage must be evaluated and if necessary use an alternate seal design.

Plan 01

Plan 01 is an integral (internal) recirculation from the pump discharge to the seal chamber, which is typically at a pressure slightly above pump suction pressure. It is similar to Plan 11 in that it uses the pressure differential between pump discharge and pump suction to develop flow, but is different in that there

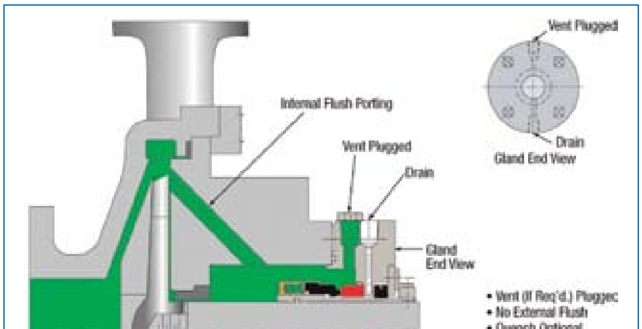


Figure 1. Seal Flush Plan 01

are no external lines (piping or tubing) on the pump. It is recommended for clean pumpage only and is typically limited to pumps with a Total Discharge Head of less than 125-ft.

Advantages
- No product contamination. The flush source is coming from the pump and going back to the pump.
- No reprocessing of product. Unlike an external flush, the product does not have to be reprocessed.
- Simplified piping. There is no external piping on the pump.
- Useful arrangement on fluids that are highly viscous at normal ambient pumping temperatures so as to minimize the risk of freezing if exposed to low temperatures in external piping plans, such as a Plan 11.

Disadvantages
- Must be used for clean pumpage only as dirty pumpage could easily clog the passageway and it would require pump disassembly to repair.
- Flush has to be repumped. The circulation from pump discharge back to pump suction will decrease pump efficiency and increase power required for the application. The volume of the flush is based upon the pump OEM's design and the pressure differential.
- There is no external way to control flow. Unlike Plan 11, which can have an externally replaceable orifice to control flow, the internal design of a Plan 01 eliminates this possibility.
- The flush is not usually directed right at the faces, but may come in over the seal head.

Sizing and Controlling
The flow rate is dependent upon the pressure differential in the pump and the design of the line running internal to the pump casing. Changing the impeller design can affect the pressure differential and thus the flush rate. The pump OEM should be contacted to ensure that the flow rate is adequate to maintain a stable condition at the seal faces.

General
This flush system can perform its function well when used properly. Changes in pump impellers, or changing seal designs that can move the seal faces away from the flush hole can cause problems that result in seal failures. This system is not recommended on vertical pumps.

Plan 02

Plan 02 is a non-circulating flush plan. In Plan 02 the process is not directed into or out of the seal chamber. Seal generated heat is removed by convection and conduction to the process fluid, pump components, and the surrounding environment. Also, some seal chamber designs promote cooling, by mixing of process fluid between the pump cavity and seal chamber. Often, this plan is used in conjunction with API Plan 62 and/or the optional use of a cooling jacket, which will provide some additional cooling. This plan should only be used for services where adequate vapor suppression can be assured, so that vaporization of the process in the seal chamber or at the seal interface does not occur. Plan 02 is often used with a self venting, open seal chamber, i.e. no throat bushing.

Advantages
- No external hardware required.
- Solids are not continually introduced to the seal environment.
- Pump efficiency is not affected, as there is no recirculation of pumped or externally supplied fluid.
- Natural venting occurs with an open chamber throat.

Disadvantages
- Success of this plan can be difficult to predict, reliance on previous experience with a specific process or pump design is often required.
- If the fluid in the seal chamber vaporizes, the life of the seal will be drastically reduced.
- Fouling of the cooling jacket, if so equipped, over time will reduce its effectiveness, resulting in higher seal chamber temperatures.
- Careful design of impeller/chamber interaction is necessary on low head pumps to prevent air ingression.

General
Low duty, chemical service pumps are often a prime candidate for Plan 02. In these services, it is also advantageous to apply Plan 02 in conjunction with a large (open bore) or taper bore seal chamber. Often, in these services, suspended solids may be included in the process stream. In these cases, devices which encourage seal chamber circulation, while excluding solids from the seal chamber, are available and offered by many OEM and after market suppliers. Applications where these devices have been applied often work well with Plan 02.

Hot, refinery and petrochemical heavy oil services can be

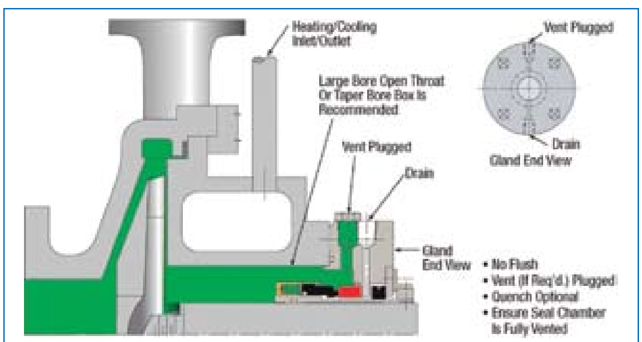


Figure 2. Seal Flush Plan 02

successfully sealed with Plan 02. Often, these services congeal or become highly viscous at ambient conditions. This can result in fouling and plugging of the recirculation plans, such as Plan 11, 13, 23 and their derivatives, unless effective temperature control schemes are employed. In these services, Plan 02 offers a relatively simple, cost effective way to obtain reasonable seal life. Only Plans 32 or 54 may be found to provide superior seal life. Use of Plan 02 in hot oil applications normally requires the use of Plan 62, using steam or nitrogen. In most cases, use of a seal chamber cooling jacket is helpful.

Successful use of Plan 02, as with other plans, is dependent on maintaining a lubricating film between the seal faces. This can be accomplished only if vapor formation in the seal chamber can be adequately suppressed. Plan 02, with no forced circulation through the seal chamber, requires thorough venting. This can be accomplished before startup (after pump inventory) or on a continuous basis by means of a self venting seal chamber design. Further, this Plan should be used with caution if the process has entrained gas or other components, which may vaporize easily. This plan is not recommended for vertical pumps.

Plan 11

Plan 11 is the most common flush plan in use today. This flush plan simply takes an appropriate amount of fluid from the discharge of the pump (or the discharge of one of the intermediate stages if applicable) and puts it into the seal chamber to provide cooling and lubrication to the seal faces.

Advantages

- No product contamination. The flush source is coming from the pump and going back to the pump.
- No reprocessing of product. Unlike an external flush, the product does not have to be reprocessed.
- Simplified piping. Piping consists of only pipe (or tubing) and an orifice, if required.
- With a properly sized orifice and throat bushing that results in a higher seal chamber pressure, the vapor pressure margin can be increased.

Disadvantages

- If the product in the pump is not a good face lubricant or is dirty, the seal can become damaged or clogged.
- Flush has to be re-pumped. Circulation from the discharge back to the pump suction will decrease pump efficiency and increase power required for the application. Usually the volume of flush is very small compared to the capacity of the pump and therefore the efficiency effect is very small.

Sizing

Generally the flush rate must be calculated based on service conditions, pump speed and seal size. The rule of thumb is for not less than 1-gpm per inch of seal size, but the flush requirement may be greater if the pressure or speed is high. For application above 3600-rpm or box pressures above 500-psig the flush rate should be calculated to avoid excessive heat at the seal.

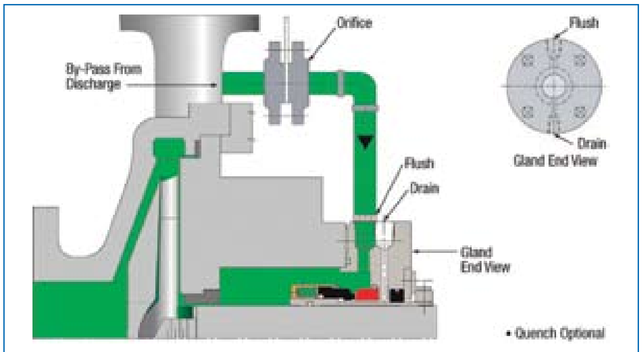


Figure 3. Seal Flush Plan 11

Controlling

The flush flow rate is usually controlled by an orifice in the flush line. Orifices should not be less than .125-in unless the product is very clean and customer approval is obtained. Many small or low speed pumps have a low differential pressure and no orifice is required in the piping.

An interesting challenge arises when the differential pressure is high and a .125-in orifice allows for more flow than is desired. This can be addressed two ways. One option is to use two or more orifices in series. The number is dependent on the differential pressure. The other way is to use a “choke tube”. This is a piece of tubing generally .250-in heavy wall. The length of the tubing is calculated using a piping pressure drop calculation such that the pressure drop across the tubing is equal to the difference between the discharge pressure and the seal chamber pressure at the flow rate desired.

General

Any flush system works hand in hand with the hardware and seal parts. If the seal is set up with a distributed or extended flush, the effectiveness of the system will be better and the seal will run cooler no matter how much or little the flush flow rate.

Next month: *Part Two of our series continues next month.*

P&S

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