SELECTION STRATEGY FOR MECHANICAL SHAFT SEALS IN PETROCHEMICAL PLANTS

by

Heinz P. Bloch

Senior Engineering Associate, Exxon Chemical Company

Baytown, Texas

Heinz P. Bloch is a Senior Engineering Associate with Exxon Chemical Company's Central Engineering Division. Located in Baytown, Texas, he assists several plants in resolving machinery-related issues.

Mr. Bloch has held machinery-oriented staff and line positions with Exxon affiliates since 1965. His assignments have included tours of duty at plants outside the U.S. in Italy, Spain, England, The Netherlands and Japan. Troubleshooting and reliability improvement missions took him to affiliated companies and machinery manufacturing plants in 12 additional countries.

The extent and depth of Mr. Bloch's previous and continuing involvement in all aspects of machinery reliability improvement is indicated in over thirty-five technical papers and two books, entitled, Practical Machinery Management for Process Plants *and* Machinery Failure Analysis and Troubleshooting *(co-authored). They have been used for lectures and consulting assignments in the U.S. and overseas. Mr. Bloch's four U.S. patents relate to high speed machinery improvements.*

Mr. Bloch graduated from the New Jersey Institute of Technology with B.S. and M.S. degrees in Mechanical Engineering. He was elected to three national honor societies, is a member of ASLE and ASME and maintains registration as a professional engineer in the State of New Jersey.

ABSTRACT

Petrochemical plants are confronted with an increasing number of available mechanical seal designs. The user must choose from among inexpensive standard seals and exotic, highly engineered seal models. We have seen seal costs range from less than \$200 to over \$6000 for pumps which didn't seem all that different to the superficial observer. Each of these has its place, although it may prove difficult to determine which seal should be used for maximum long-term effectiveness in a given application or service.

A selection strategy outline, which makes maximum use of the capabilities of knowledgeable seal manufacturers is presented. The strategy avoids unnecessary proliferation of different designs and leads to the selection of high quality seals for the largest possible number of pumps in a given installation. While primarily developed for grassroots plants, this approach can also be used to systematically upgrade mechanical seals in existing installations.

INTRODUCTION

Pump mechanical seal failure incidents outnumber any other rotating mechanical component failure event in typical petrochemical plants. Very often, 65 to 70 percent of pump maintenance expenditures must be allocated for mechanical seal repair and replacement [1]. Seal failures can be attributed to material selection errors, mechanical damage, incorrect flush, pump operating deficiencies, manufacturing defects, installation errors and so on. Failure analysis may reveal corrosion, erosion cavitation, thermal distortion and, of course, many additional modes of consequential damage.

Much has been written about these and related topics. Excellent information manuals and troubleshooting instructions are available from seal manufacturers with world-wide representation and manufacturing plants in such exotic places as Temecula and Tarzana. Their literature generally stresses failure cause identification, allowable temperatures and pressures, operating speeds, and material selection considerations. There is also a trend towards greater attention to repair costs, environmental concerns and ease of maintenance.

The user, however, continues to be confronted with an increasing number of available seal types. Each vendor understandably praises his product, lauding the merits of design features which can range from valid and desirable to hair-splittingly academic. It is left to the user to separate fact from wishful thinking, to distinguish seals suitable for a wide range of applications from those which will tolerate only limited deviations from optimum operations, and to tell low maintenance-intensive configurations. To make an intelligent choice, the user needs a seal selection strategy.

He needs guidance, a "road map" to seal selection, so to speak. This "road map" should direct him to seals which embody more than initial cost advantages. The emphasis should clearly be towards standardization without sacrificing reliability. Seals should be simple to install and maintain, they should not be vulnerable to minor deviations in pump operation or properties of pumpage. Indeed, a given seal should suit the needs of many pumping services so as to reduce spare parts inventory, streamline training requirements and leave little room for maintenance errors.

These goals can be met with a well-defined selection strategy which starts with the development of a suitable bid request package and leads to a bid evaluation which stresses technical features instead of lowest possible bid price. Such a strategy may well result in the added bonus of greatly increased seal reliability and considerably lower maintenance expenditures for most petrochemical plants. Many major manufacturers of mechanical seals have indicated their support and willingness to cooperate in the implementation of our selection strategy. Specifically, this strategy requires the development of bid request information which asks the seal vendor to assign severity categories to the various pumping services disclosed by the user. The user then screens the seal vendors' proposals to identify optimum seal configurations. Principal features of optimum seals are highlighted in this paper.

DEVELOPMENT OF BID REQUEST PACKAGE

The development of bid request information is a key element in a selection strategy leading to the procurement of mechanical seals which comply with the user needs outlined above. Bid requests must be forwarded to several experienced seal manufacturers and must make it clear that the manufacturer should submit cost proposals for only those services where his seal selection is backed by solid experience. His inability to furnish seals for some services should in no way disqualify him from submitting bids for those services where he is competent to provide a good product.

To begin with, the user should assemble API data sheets for the various pumps which need mechanical seals. These data sheets must be forwarded to the various seal vendors and the vendors asked to recommend classification categories for each pump, as shown in Table 1. Classification categories range from 1 to 10 in increasing order of severity. Category 1 would be reserved for pumping services which are easiest to seal, or where potential seal distress would not be of serious concern. Category 10 would comprise services or conditions where seal failures could result in extreme hazards and consequential damage. Category 10 seals would thus merit special designs and expert attention through all stages of design, fabrication, quality control, installation and operation. Verification of flawless operation would be assisted by state-of-art instrumentation and auxiliaries.

Classification categories can be assigned only after the user or purchaser provides the seal manufacturer with a comprehensive list of pumps which have been tentatively selected. Also, this pump tabulation must disclose all of the fluid properties and operating parameters known to the user. With disclosure thus going beyond the typical contents of API data sheets, the seal manufacturer can be requested to list the "operating windows" within which the proposed seal will function reliably in the pump tentatively selected. It is to be understood that the "operating window" refers to the *actual* seal with the chosen balance ratio, flush plan, stuffing box layout, etc. Notice, also, that we have opted to designate the pumps as "tentatively selected" because we should anticipate the possibility that even a capable seal vendor may be unable to offer his optimum seal for a given pump. Should this be the case, it might be appropriate to consider selecting a more suitable pump model for the intended service. Making the seal the weakest link in pump selection should not be considered an acceptable alternative.

Next, the seal vendor should propose a seal Type "A" from his manufacturing program which would satisfy the requirements of all pumps with service classifications 1, 2 and 3, a seal Type "B" which would satisfy the purchaser's guideline requirements for all pumps with classifications 4, 5 and 6, and finally a seal Type "C" for classifications 7, 8 and 9. Bid invitations for service classification 10 are thought to merit special handling and should go to companies whose special expertise and competence have been verified by in-depth experience checks.

Seal vendors proposing seals may elect to narrow their offer to only 2 seal types, say Type "A" for categories 1, 2 and 3, and Type "C" for categories 4 through 9; or Type "B" for categories 1 through 6, and Type "C" for categories 7, 8 and 9.

As mentioned earlier, it is recommended to mail the bid request information to three or four capable mechanical seal manufacturers. Their response should be critically analyzed to flush out significant deviations among the various proposals. These deviations may range from materials of construction to different API flush plans, and from differences in basic configuration to differences in application philosophy of stationary vs. rotating seal members. Reconciling the deviations or differences will assist the engineer responsible for final selection in determining whose seal offer has the best chance of meeting the user criteria highlighted earlier and further amplified later.

To assist the purchaser in the development of bid request information and later in the evaluation of bids received, we have developed the selection matrixes shown in Table 2. The upper matrix is concerned with general seal selection considerations. This matrix is easy to use and interpret. Under the heading "Basic Type Seal," the matrix explains that low and moderate temperatures, pressures, speed, etc., can be accommodated by the majority of basic configurations. From the second heading we would infer that Type "A" seals are probably identical to "basic" seals. They should not be applied in high pressure of high speed services, but could very well serve in high temperature environments as long as heat resistant secondary seals or effective auxiliary cooling means were applied. Type "B" seals would go one step further by featuring high speed capability, as long as the design is of the stationary type (see Figure 6, later). Finally, Type "C" seals would add high pressure capability as long as two-ply bellows and/or stress resistant face mountings were incorporated in the design.

The lower matrix points to specific seal design or material considerations and ranks their relative significance in applications which have to cope with high abrasives content, poor lubricity, extreme temperature, etc. For instance, a bellows seal could represent a significant improvement in applications where seal failure due to clogging of springs might lead to costly losses. The same seal should be considered excellent for high temperatures. Or, we could examine double seals and see that this design has no influence on high speed capability or ease of installation. However, it might significantly improve pump reliability under high temperature conditions and should be rated excellent for sealing against pumpage with high abrasives content, poor lubricity, low vapor pressure, low viscosity, hazardous or toxic properties, and high risk resulting from seal failure. The next column describes tandem seals which embody the same

Table 1. Seal Selection Strategy.

Seal type	A	B	C	D
General characteristics of pump or pumpage	Non-toxic, low pressure, moderate temperature, no limitations on pump size, but primarily AVS and ANSI pumps.	Moderate pressure, abrasive inclusions, low and moderately high temperatures. Primarily API pumps.	Severe service, high pressure and/or extreme temperature, high loss potential. Exclusively API pumps.	Extreme hazard and consequential damage risk. Special construction pumps.
Service example and classification	Filter backwash pump 5 hp, 20 gpm. 2 to 38 psig, 90°F, 1,760 rpm Classification 1	Caustic transfer pump 15 hp, 25 gpm, 0.3 to 105 psig, 100° F, 3,550 rpm Classification 4	Synthetic tar pump, 200 hp, 280 gpm, 13 to 280 psig, 520°F, 3,550 rpm Classification 7	Nuclear waste pump/reactor emergency pump. Last resort safety pump Classification 10
	Sour water pump, 20 hp, 310 gpm, 1 to 72 psig, 230°F, 3,550 rpm Classification 2	Kerosene feed pump, 125 hp, 310 gpm, 0.1 to 250 psig, 90° F 3,550 rpm, Classification 5	Intermediate synfuel pump, 500 hp, 930 gpm, 7 to 625 psig, 720°F, 1,760 rpm Classification 8	
	Slop oil pump, 40 hp, 100 gpm, 0 to 79 psig, 150°F, 3,550 rpm Classification 3	HC solvent reflux pump, 250 hp, 1,310 gpm, 65 to 270 psig, 300°F, 3,550 rpm Classification 6	Ethylene product pump, 1,250 hp, 820 gpm, 390 to 1,745 psig, −50°F, 3,550 rpm Classification 9	

design advantages as double seals, except for ability so seal against high abrasives content and poor lubricity of pumpage. Thus, if the pumpage is without abrasives and has good lubricity, it would be appropriate to select tandem seals instead of double seals. Another advantage of tandem seals is that they will not require the buffer fluid system to operate at pressures in excess of stuffing box pressure.

The remaining columns in the lower matrix of Table 2 should be used in the same fashion by the engineer specifying or evaluating mechanical seals for pumping services in petrochemical plants.

Explanation of Symbols Used in Table 2

1. Heat resistant secondary seals and/or auxiliary cooling.
2. Moderate upgrading of metallurgy to provide corrosion resistance.
3. Maximum upgrading of metallurgy to provide corrosion resistance.
4. Abrasion resistant faces.
5. Metal bellows design.
6. Stationary design.
7. Cartridge mounted assembly recommended.
8. Two-ply bellows and/or stress resistant face mounting.
9. Stepped shaft or sleeve to achieve hydraulic balance.
10. Metal, asbestos, or graphite secondary seals.
11. External steam quench in gland.
12. Heat the sealing area before rotating shaft.

L = Low   N = No significant improvement
M = Moderate   S = Significant improvement
H = High   E = Excellent

Table 2. General Seal Design Considerations.

	Basic type seal			A type seal			B type seal			C type seal		
	L	M	H	L	M	H	L	M	H	L'	M	H
Temperature	X	X 1		X	X	X 1	X	X	X 1	X	X	X 5
Pressure	X	X 9		X	X		X	X		X	X	X 8
Speed	X	X		X	X		X	X	X 6	X	X	X 6
Corrosives	X	X	X 2	X	X	X 2	X	X 2	X 3	X	X 2	X 3
Abrasives	X	X 4		X	X 4		X	X 4		X	X 4	

Specific seal design or material considerations

Equipment or product deviations	Stationary design	Metal bellows design	Isolated springs	Hydraulic balance	Double seal	Tandem seal	Cartridge mounting	Tungsten carbide	Silicon carbide	Carbon graphite
Extreme temperature	N	E/10	S	S	S	S	N	S	E	E
Coking/flashing	S	E/11	E	N	E	S	N	E	E	N
High abrasive cont.	S	S	S	N	E	N	N	S	S	N
Poor lubricant	N	N	N	S	E	N	N	S	E	E
Corrosive	N	N	E	N	N	N	N	N	E	E
Hazardous	N	N	N	N	E	E	N	N	N	N
Low viscosity	N	N	N	E	E	E	N	S	E	E
High vapor pressure	S	N	N	E	E	E	N	S	E	E
Congealing	S/12	S/12	E	N	S	N	N	E	N	N
High loss potential	S	S	N	N	E	E	N	N	S	N
Installation ease	N	N	N	N	N/7	N/7	E	N	N	N
High speed	E	N	N	E	N	N	N	S	E	E

DESIRABLE DESIGN FEATURES IDENTIFIED

Evaluation of the various bids is made easier by recognizing desirable design features incorporated in mechanical seals. Some of these merit closer consideration.

Cartridge Construction

Cartridge seals are designed for rapid installation on and removal from pump shafts. The cartridge seal is an arrangement of seal components of a shaft sleeve and in a seal gland constituting a single unit which is usually assembled and pre-set at the factory. Both bellows and spring-type seals can be cartridge arranged if the pump stuffing box is large enough.

Cartridge seal units offer major maintenance advantages. Replacement is rapid and there is far less risk of assembly error and assembly damage than with conventional mechanical seal mounting. For these reasons, we prefer cartridge seals for typical pumps in petrochemical plants. Cartridge arrangements are routinely available for horizontally split pumps. Overhung pumps are not often supplied with cartridge seals due to space and general pump design constraints.

One word of caution: Metallic gaskets under shaft sleeves may take a permanent set and cannot be reused.

"Symmetrical Package" and "Unitized" Seal Construction

Stuffing box size and general dimensional constraints have for years impeded the application of advanced, or high performance mechanical face seals in many overhung, AVS, and ANSI-type pumps. After the needs of the petrochemical industry were again discussed at a fluid sealing forum [2], experienced seal manufacturers addressed the issue and developed "Symmetrical Package" and "Unitized" seals incorporating the principal components shown in Figure 1. These pre-assembled

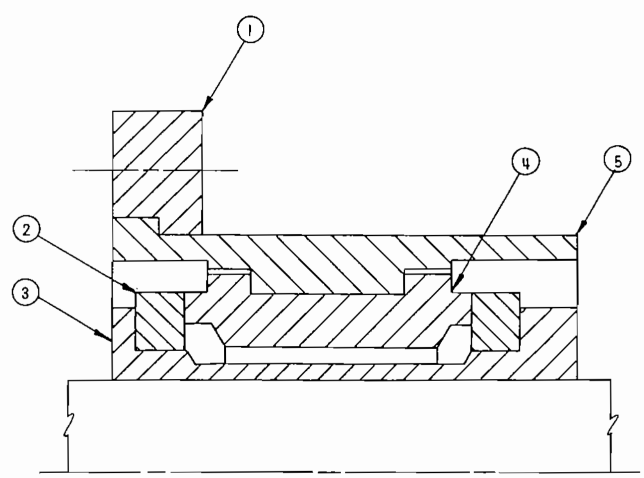

Figure 1. Symmetrical Package Seal and Major Components: (1) Mounting Flange, (2) Mating Rings, (3) Sleeve, (4) Primary Seals, (5) Housing.

package seals are now available as single or double mechanical seals. They will completely replace the stuffing box of many pumps whose hydraulic end would be adequate, but whose stuffing box dimensions are judged unsuitable for reliable long-term seal operation or for utilization of optimum seal designs.

Proper application of these self-contained seals could lead to the procurement of suitable configured hydraulic portions of centrifugal pumps from the pump manufacturer, and procurement of optimized package seals from the seal manufacturer. Symmetrical Package or Unitized Seals would be mounted to the pump casing by the seal manufacturer; as a minimum, the seal manufacturer's mounting instructions would have to govern at final assembly.

In any event, Symmetrical Package and Unitized seals open up untold possibilities for reliability improvement and cost reduction on a wide spectrum of centrifugal pumps and pumping services.

Hard Face Material and Preferred Rotating Member

Heat generated at the seal faces must be rapidly conducted away if fluid vaporization and resulting problems are to be avoided. Depending on service conditions and pump design, either the rotating or stationary seal ring must be counted on to dissipate as much frictional heat as possible. Considerably greater hardness, shock resistance and better wear properties make silicon carbide the preferred seal material in many of the more severe applications.

In applications where carbon is used as one of the seal faces, most of the heat will be dissipated through the higher conductivity silicon carbide or tungsten carbide mating ring. Heat dissipation will be enhanced if this component is being rotated instead of the carbon. It would thus be advantageous to rotate the high conductivity component whenever effective heat dissipation is of critical importance.

Placement of O-Rings

An advantageous seal design recognizes that O-ring life can be reduced by close proximity to the heat source, by swelling due to chemical attack, and by operation in a dynamic mode, especially in the presence of erosive materials. Going to PTFE chevrons or wedges may allow operation at higher temperatures and reduced risk of chemical attack, but will very often lead to fretting of metal surfaces in contact with it. Bellows seals eliminate many of these problems by using static secondary sealing. Seals with spring loaded running faces are forced to use dynamic means of secondary sealing which could, in some instances, be more prone to failure.

Mechanical Design Considerations

Important differences can exist in the mechanical designs of competing vendors. For instance, execution "E" of Figure 2 shows the method of clamping rotating hard face (1) against shaft sleeve (2) of a stationary bellows seal assembly. This clamping method ensures perpendicularity between shaft centerline and rotating seal faces. However, this clamping method also invites distortion at the running faces. Execution "F" tends to avoid distortion by mounting the rotating hard face in a resilient backing ring (3) and by pulling mounting ring (4) against the collar (5).

Two different clamping methods are shown in Figure 3. Both of these methods were devised to eliminate distortion of running faces which, incidentally, is still a possibility unless careful engineering and material selection keep the shrink-fitted hard face and mating ring carrier together throughout the anticipated temperature range. However, in execution "G", the

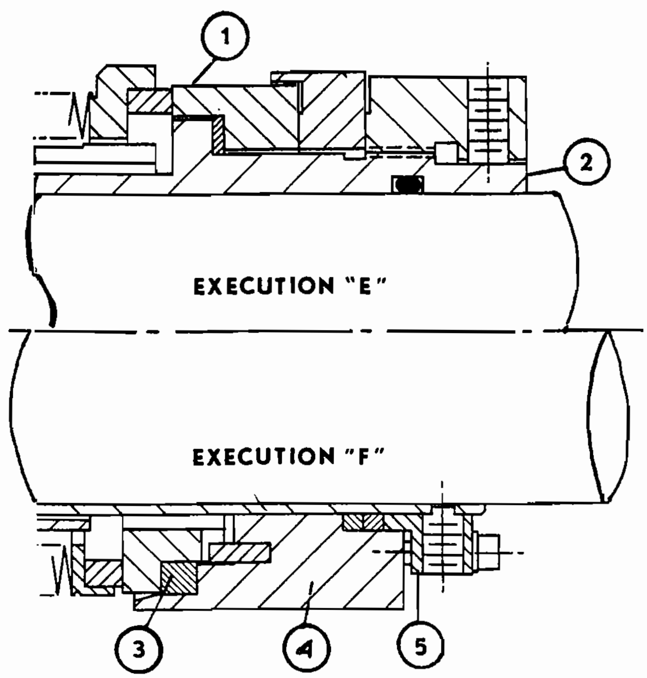

Figure 2. Competing Methods of Mounting Hard Face (1) Against Shaft Sleeve (2), Both Executions Claim a Distortion-Free Arrangement Using Resilient Backing Ring (3), Mounting Ring (4), and Collar (5).

collar is set-screwed to the shaft or shaft sleeve, whereas in execution "H" the mating ring carrier is set-screwed to the shaft or shaft sleeve. Experience shows that on some seal models' perpendicularity and seal setting accuracy are more difficult to achieve with the clamping method indicated as execution "G" in Figure 3. As shown, the seal requires very careful adjustment of cap screws inserted through the collar. This problem can be overcome if the design allows collar and carrier faces to butt-up solidly.

Another interesting difference exists in the carbon holders of the more conventional mechanical seals. The upper half of Figure 4 shows a lock ring (1) executed with a set screw (2) which engages a slot in carbon holder (3). Under certain service conditions, contact between set screw and slot may cause a wear pattern which may prevent proper seal operation.

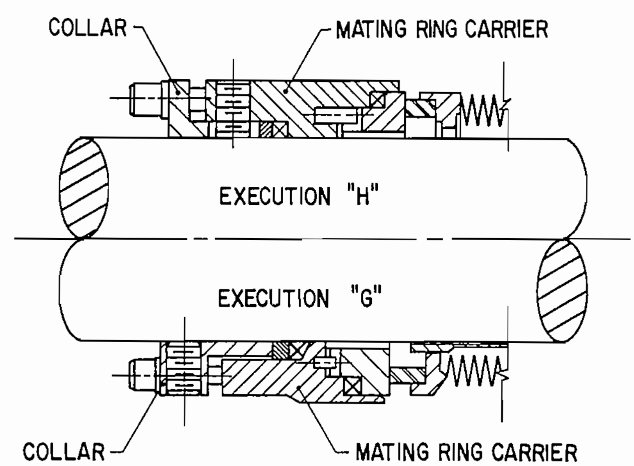

Figure 3. Competing Clamping Methods. Execution "G" Requires Careful Adjustment to Ensure Perpendicularity Unless Collar and Carrier Faces Are Designed to Butt-up Solidly.

Moreover, tightening of the set screw can distort the relatively thin lock ring and cause contact or interference between lock ring OD and carbon holder ID. The construction features shown in the lower half of Figure 4 would tend to eliminate both of these potential problems by providing an axially oriented drive pin (4) and a considerably heavier lock ring (1). Both designs shown in Figure 4 deserve credit for presenting relatively smooth, low turbulence surfaces to their respective fluid environment. This is largely accomplished by locating the springs on the atmospheric side of the seal.

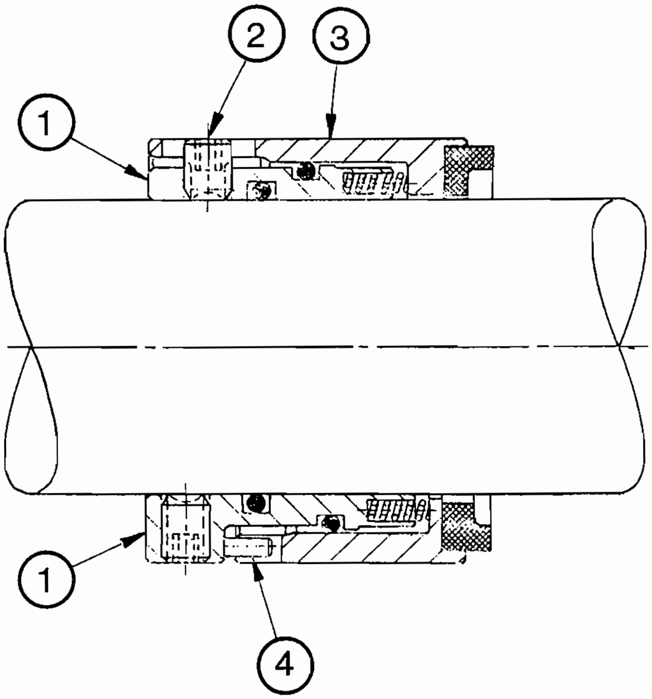

Figure 4. Lock Rings Differ in Strength; Top Half Shows Lock Ring (1), Executed with Set Screw (2), Engaging Carbon Holder (3). Bottom Half Reduces Potential Fretting by Using Heavier Lock Ring (1) and Axially Oriented Drive Pin (4).

SEAL TYPE "A"

Although the user would be wise to look for desirable design features in competing offers, the seal selection strategy would allow a Type "A" seal to be furnished by many seal manufacturers. Experience shows that the majority of pumping services with classification 1, 2 or 3 would employ AVS or ANSI-type centrifugal pumps. It will often not be possible to fit cartridge type seals in these pumps. Balanced seals should not normally be required for Type "A" seals. However, if a balanced seal is ultimately chosen, the balance should be achieved internal to the seal instead of using stepped shafts of sleeves. All Figures, 1 through 8, show seals which can incorporate this design advantage.

Simplicity of maintenance may be of somewhat greater importance for the Type "A" seal than compliance with every one of the desirable features cited earlier. Two of several seals we would expect to be offered as Type "A" seals are illustrated in Figure 5. Notice that the stationary hard faces are reversible and can probably be reconditioned several times. The top half of Figure 5 depicts a "T-clamped" stationary seat ring using gaskets which are difficult to keep evenly compressed at assembly. Also, the application of excessive clamping pressure may cause distortion at the seal faces.

The bottom half of Figure 5 shows a recently introduced low-cost bellows seal with an O-ring equipped, reversible stationary seal ring. Both seat and O-ring are available in several materials. However, neither of the two seal executions shown here should be expected to serve as an effective throttle bushing in case of catastrophic seal failure. Throttle bushings should be made from a non-sparking material, have only a small shaft clearance and be fitted into the gland plate. This requirement is given in the 6th Edition of API Standard 610, Para. 2.7.1.12, but not always observed by the vendor.

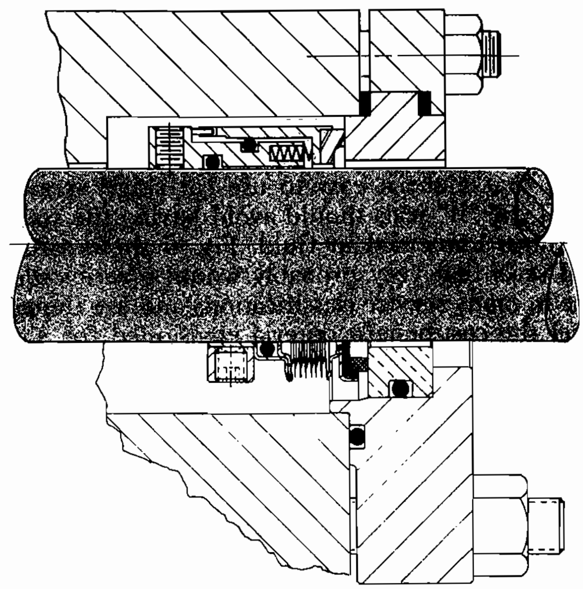

Figure 5. Typical "A" Seals: Multi-Spring Rotating Element and "T-Clamped" Stationary Seat Ring (Top) vs. Inexpensive Bellows Seal with O-Ring Seat (Bottom).

In a specific procurement situation, we would probably ask the manufacturer who proposed a Type "A" seal with the "T-clamped" stationary to re-bid on the basis of using the stationary seat execution shown in conjunction with the bellows seal at the lower half of Figure 5. When selecting a low-cost, Type "A" seal for corrosive pumpage, we would want to verify the suitability of the bellows material for satisfactory long-term operation.

SEAL TYPE "B"

Type "B", seals must be selected for satisfactory operation in pumping service classifications 4, 5 and 6. The majority of the pumps meeting these classifications would probably be API-type and our preference would be to obtain cartridge seals, even though overhung type process pumps are not routinely furnished with design features which make it easy to use cartridge arrangements. Type "B" seals should avoid having the spring (or springs) immersed in the fluid. Two of several seals which would very probably give excellent service in many service classifications and are thought to meet the cost/benefit criteria envisioned for Type "B" service severities are shown in Figure 6. These seals use one or more stationary springs and incorporate a number of desirable features. A cartridge arrangement for ease of installation; the single non-rotating spring shown with the design in the top half of Figure 6 is arranged to operate away from the product, bronze spring retainer (1) serves as a throttle bushing, and the relatively clean profile inside the stuffing box reduces seal drag. (Note that rotating

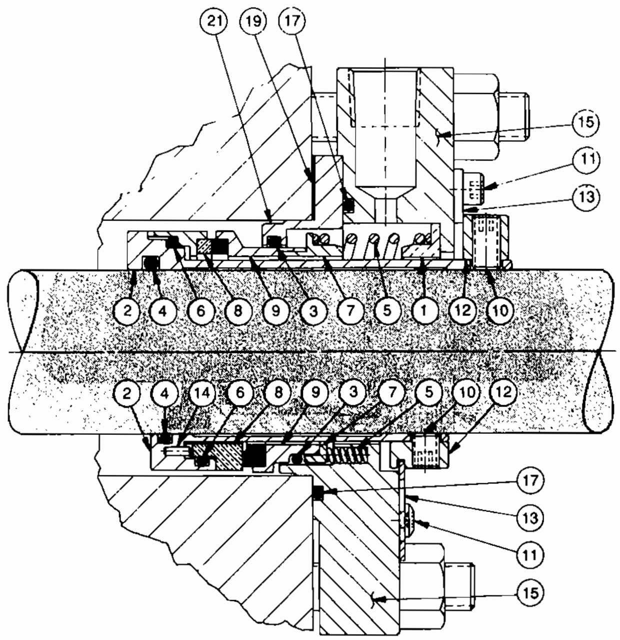

Figure 6. Typical Type "B" Seals: Single Spring Stationary Seal (Bottom). Note That Rotating Parts Carry Even, Stationary Parts Carry Odd Numbered Designations.

components are identified with even, and stationary components with odd numbers).

Except for utilizing several springs, the seal design shown in the bottom half of Figure 6 is quite similar to the design shown in the top half. Both seals incorporate the desirable features of many similar stationary seal designs:

Self-squaring faces. This feature may result in appreciably better seal life for pumps with excessive shaft deflection or pumps operating with nominal shaft deflection at high speeds.

Non-flexing springs. Spring life extension and long-term, uniform pressure can be expected.

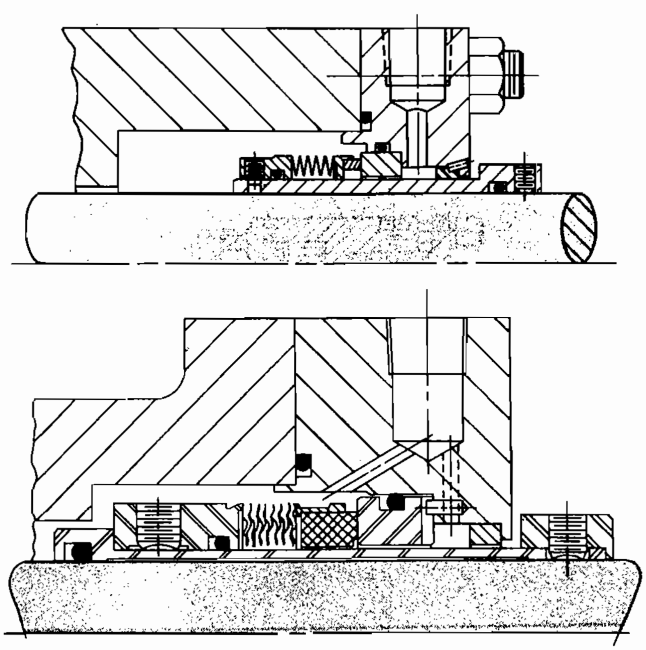

Figure 7. Typical Type "B" Seals: Cartridge-Mounted Rotating Bellows Seals. These Seals Are Also Available in Stationary Execution.

Pre-assembled cartridge construction. These seals can be shipped with the gland plate in place. No field measurements or settings will be required.

However, a closer look will show functional differences in the arrangement of O-ring (3). It could be argued that progressive wear of the seal faces shown in the upper half of Figure 6 will cause the O-ring to make sliding contact with a clean portion of part (9), whereas advancing the stationary seal face in the lower half of Figure 6 will cause O-ring (3) to slide over a wetted and potentially contaminated portion of gland plate (15).

Functionally similar features can also be found in an intermediate range of bellows seals as shown in Figure 7. Of course, these should also be considered for Type "B" services. Rotating bellows seals tend to be self-cleaning by virtue of centrifugal action. As mentioned earlier, they do not incorporate sliding (dynamic) elastomers. Instead, they use static secondary sealing means. Stationary *bellows* seals will exhibit the same advantages as given for stationary *spring* seals, above. Generally speaking, stationary-type seals should be preferred in applications encountering face peripheral velocities in excess of 4500 feet (1372 meter) per minute, serious shaft deflection, or coking after the pumpage has crossed the seal faces.

When applying bellows seals in light hydrocarbons, we should look for design features which prevent torsional windup of the bellows in case the seal faces undergo slip-stick motion relative to each other. If the design does not alleviate these concerns, the user may favor spring-type mechanical seals for light hydrocarbon services.

SEAL TYPE "C"

Severe duty, high pressure, extreme temperature and high loss potential are the predominant factors making up service categories 7, 8 and 9. The slate of experienced vendors and acceptable configurations narrows considerably for Type "C" seals, which would deserve to be called High Performance Seals. Every *pump* in these service categories should comply with the most stringent requirements of applicable API specifications. Similarly, Type "C" seals would originate from manufacturers with outstanding reputations only.

One of perhaps three possible Type "C" seals is illustrated in Figure 8. This high-temperature seal uses stationary bellows, eliminates O-rings, and employs high temperature graphite foil gasketing. Only proven materials of construction and highly sophisticated fabrication and quality control techniques should be used to produce these seals. Qualified vendors have successful experience with 2-ply bellows construction to handle higher pressures and will be able to produce these seals free from harmonic resonance which, in earlier seals of this type, has caused distress.

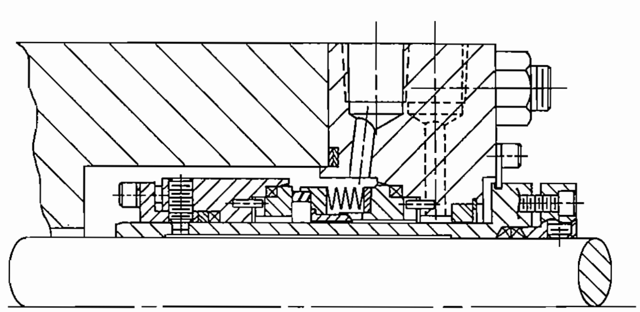

Figure 8. Typical Type "C" Seal: Cartridge-Mounted Stationary Bellows Seal with Floating-Ring Breakdown Bushing.

High performance seals usually incorporate a cushion-type stationary mating ring to avoid distortion transmitted through the gland by axial forces and to allow optimum flatness control at assembly.

BID COMPARISON

After the bids are received, they have to be tabulated and compared. The format shown in Table 3 facilitates this comparison. The reviewing engineer should look for significant differences among the competing bids and should determine which offer incorporates a majority of desirable design features. Special features beyond those specified by the purchaser may have been proposed by some bidders and would deserve extra credit for reducing the risk of catastrophic failure incidents. Each special feature must be given a separate assessment of value. Alternatively, the purchaser may decide that bidder X's offer is less expensive than, but nevertheless technically equal to the offer made by bidder Y. He may now wish to upgrade his selection by asking X to furnish the seals with optional, although not previously specified features. Floating bushings in the gland ring are only one of many available options which may prove attractive to an experienced purchaser. Auxiliary packing is perhaps another low-cost option which may be offered by some vendors, although this author agrees with the favorite saying of a wise old machinery engineer in California: "Old packings never die, they just drip away."

CONCLUSION

Systematic efforts to upgrade seal selection are possible through increased involvement of experienced application engineers working for capable seal manufacturers. These efforts are assisted by a seal selection strategy which identifies those design features which promise to lead to reduced failure risk, greater ease of maintenance, and better understanding by operators and mechanical work forces. It is believed that 98 percent of the pumps used in a typical petrochemical plant can be assigned service severity categories from 1 (least severe) to 9 (most severe). Selection of only 3 seal types to cover all 9 categories will accomplish the goals we have set for ourselves. There is little doubt that the marginally higher purchase cost for better quality seals will probably be recovered during the first year of pump operation.

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Table 3. Bid Comparison for Mechanical Seals.

Service Data: Kerosene feed pump		**Item:** P-28A/B	
Driver rating:	125 hp (92 kw)	Viscosity at P.T.:	1.9 cSt
Diff. head:	704 ft (214 m)	Vapor press at P.T.:	1.0 psia(.069 bar)
Suct. pressure:	14.8 psia (1.02 bar)	Specific gravity:	0.82
Disch. pressure:	265 psia (18.3 bar)	Temperature:	90°F (32°C)
		Speed:	3,560 rpm

	Bidder		
	Apex seal	**Custom seal**	**Superb mfg. co.**
Service classification assigned by bidder	5	6	5
Seal type assigned by bidder	B	C	B
API seal code	BTTFL	BDTFL	BTTFN
Seal model	Apex 1010	CS-500	Silicarb-X
Seal design option	Standard	Cartridge	Stationary type
API flush plan	13	13	13
API auxiliary seal plan	52	54	52
Stuffing box pressure (est)	39 psig	35 psig	40 psig
Operating window (allowable deviations)			
– Temperature	± 130°F (73°C)	± 125°F (69°C)	+ 110/ + 160°F
– Discharge pressure	– 110, + 220 psi	– 50, + 200 psi	– 100, + 200 psi
– Viscosity	– 1, + 500 cSt	– 1.2, + 200 cSt	– 1, + 400 cSt
Seal balance ratio	79%	84%	82%
Seal drawing number	578-23122	8563CDA	C-802BF-20
Special features and/or remarks	Universal gland	Floating throttle bushing	Grafoil backup
Cost, incl. gland and sleeve	$899	$995	$1,070
Cost, seal only	$472	$510	$ 532
Deviations from specified requirements			

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