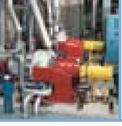

Understanding and Avoiding Lube Problems in Pump Bearing Housings

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Maintenance and engineering personnel experienced with process machinery are aware of rolling element bearings and their respective lubrication requirements. At first glance, then, it would seem a bit redundant to write an article that adds to their knowledge base or purports to have uncovered anything new . . .

Nevertheless, a letter from an experienced engineer in the western U.S. serves as a fitting reminder of occasional level discrepancies in pump bearing housings:

> *"Recently we have installed magnetic face seals and balanced constant level lubricators on approximately 25 to 30 pumps in a refinery. The result is 5 pumps with dark oil and hot bearings due to level problems associated with sealed housings and constant level lubricators with external reservoirs.*
>
> *We now believe that in installations where magnetic face seals are used and a lubricator is required, it is best to vent the housing using a filtered desiccant breather of some kind, such that oil level problems are essentially eliminated. Up to now we have been against this, as it adds an additional maintenance item; however, it seems necessary.*
>
> *Of course an alternative is to omit the constant level lubricator and install a bulls-eye sight glass."*

For lubrication problems to be rectified and avoided, especially oil level issues, the underlying physical laws must be understood. It also helps to review the pros and cons of various available lube oil application and protection options, without delving into undue amounts of mathematics and hydraulics.

Best Available Lube Application Methods and Narrow Viscosity Options

Although most bearings in modern process pumps are provided with a static oil sump, it is acknowledged that either an oil spray or pure oil mist application are the undisputed leaders for long-term highly reliable lubrication of rolling element bearings.

It is reasonable to speculate that, within perhaps a decade, oil spray and pure oil mist technologies will have been transferred into cost-effective "per-point" applications. Until then, however, reliability-focused plants must select the right bearing housing configurations, oil application methods, and bearing protection strategies. A responsible technical person must know, among other things –

1. When, where, and why oil levels should reach the center of a bearing element at the 6-o'clock position of the bearing or bearings, or
2. When, where and why it would be more appropriate to have the oil level remain below the periphery of the bearing
3. Why staying within lube viscosity limits is, sometimes, critically important

The term *technical person* infers that much of what is documented here should be shared between three job functions: engineers and technicians responsible for pump procurement and troubleshooting, the facility's maintenance technical workforce and, last but by no means least, the plant's operating technicians.

Viscosity Issues and Oil Ring Operation

Not too long ago, a petrochemical plant used ISO viscosity grade 220 in an oil mist system supplying an oil purge to a set of pump bearings. While thicker oils are sometimes advantageous, their use is an open invitation to failure of bearings that depend on viscosity-sensitive oil rings for lifting oil from a static sump into the bearings (Ref. 1, p. 236).

Oil rings (Figure 1) will generally malfunction if the viscosity differs much from the manufacturer's recommendations. More will be said about this later. The general catalogs of all bearing manufacturers explain that specific types of rolling element bearings require lubricant viscosities suitable for their speed, size and ambient temperature environment.

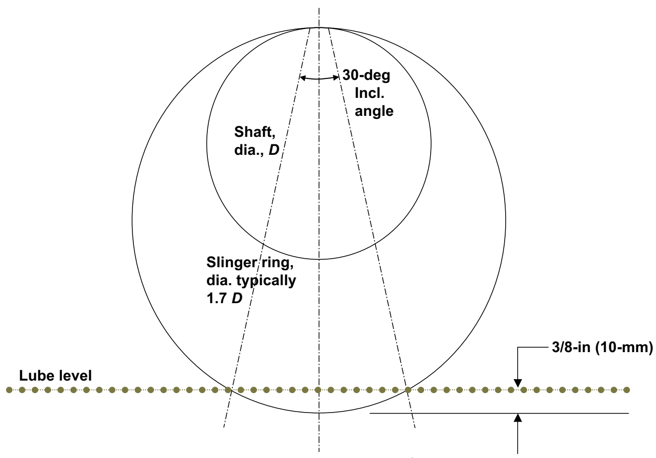


Figure 1. Typical oil ring ("slinger ring") dimensions of interest.
Source: Hydrocarbon Processing, August 2002, p. 15.

It should be noted that ISO Grade 32 *mineral* oils are considered too thin and rarely qualify for long-term, risk-free use in pumps equipped with rolling element bearings in typical North American ambient conditions. But simply switching to ISO Grade 68 *mineral oils* will be risky for bearings that are fed by oil rings (Ref. 1, p. 144), whereas appropriately formulated ISO VG 32 *synthetics* are quite acceptable here (Ref. 1, p. 188).

In any event, the user is assumed to have (a) opted for liquid oil as the pump lubricant, (b) selected the correct lube viscosity and (c) the desire to understand why oil level issues often arise in process pump bearing housings.

How and Why Oil Level Requirements Differ

It is common practice to make a clear distinction between two different level settings (oil immersion depth) for rolling element.

In one setting, the oil level reaches to the center of a ball or roller element – but never higher (!) – located at the 6-o'clock position of an assembly (Figure 2).

If this oil level is decreased to a point much below the center of the lowermost rolling element, the risk of oil overheating increases drastically. Many oil formulations typically used in pumps turn black when overheated. This black oil then shows up in the transparent bowl of constant level lubricators and does so because thermal convection currents cause oil to circulate, and because temperature-dependent cyclic compression and expansion occur in the air space at the top of constant level lubricator bowls.

In situations where the oil reaches the center of the lowermost bearing element, there will be an increase in oil temperature because the rolling elements encounter friction as they move through the lubricant. This temperature increase tends to be excessive at higher DN-values (DN is the shaft diameter (inches) multiplied by shaft rpm). An experience-based "rule-of-thumb" DN-threshold of 6000 is generally considered safe and reasonable, although somewhat higher DN-values (perhaps 8000) have been reached in some pumps.

So, while applying cooling water to effect lowering the oil temperature seems plausible, such cooling is not the preferred course of action for rolling element bearings in pump bearing housings.

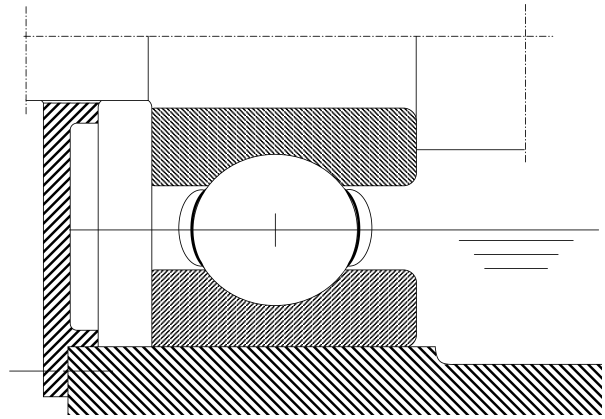


Figure 2. Oil-bath lubrication.
Source: Hydrocarbon Processing, August 2002, p. 15.

For reasons of energy and resource conservation, and also since cooling invites moisture condensation, neither water-cooled bearing housings nor cooling coils immersed in the oil are recommended for rolling element bearings in pumps.

The deletion of cooling water is feasible for all rolling element bearings in process pumps. It has been shown that acceptable operating oil viscosities and bearing temperatures can be reached in such pumps independent of DN-values (Ref. 2, pp. 434-440). Nevertheless, and to keep oil temperature and viscosity under control, a lowered lube level is needed.

As mentioned earlier, a rule of thumb allows lube levels to the center of the lowermost bearings up to a DN-value of 6000. However, once the oil level is *lowered* so as to accommodate DN-values above 6000, the lubricant needs to be "lifted" to the bearings. That's perhaps where the traditional slinger rings (Figure 1) come to mind, although slinger rings are no longer preferred here by reliability-focused users.

Why to Avoid Slinger Rings

Slinger rings tend to become unstable at DN-values somewhere between 6000 and 8000. As a rule, they must be concentric within 0.002-in (0.05-mm) and have acceptably low RMS bore surface finish. While their immersion and viscosity-sensitivities are generally acknowledged, both pump manufacturers and users often overlook the slinger rings' tendency to "run downhill" unless the shaft is installed with a degree of horizontality that would make it unrealistically close to perfection.

As slinger rings move downhill, they will inevitably contact either the edge of a shaft groove or the inside of the bearing housing. Such contact will not only tend to slow down oil rings, but will very often cause the oil rings to undergo abrasive wear. In that case, slivers of slinger ring material contaminate the lubricant (generally making the oil appear gray) and cause premature bearing failures. This explains why flinger discs should be used whenever possible although they too will have to be used within manufacturer-approved peripheral speeds.

Flexible flinger discs can be trimmed to the required diameter and, by virtue of the failure risks associated with slinger rings, flinger discs are highly cost-effective. Although primarily developed for the aftermarket, they also deserve to be specified for OEM applications (Ref. 3). However, caution is needed to ascertain that they will fit in existing bearing housings, and the fact that the elastomer disc will expand at high peripheral speeds must be considered.

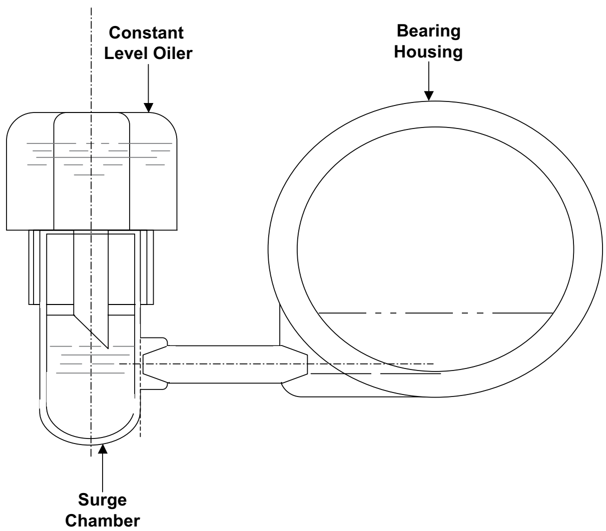


Figure 3. Elementary, unbalanced constant level lubricator.

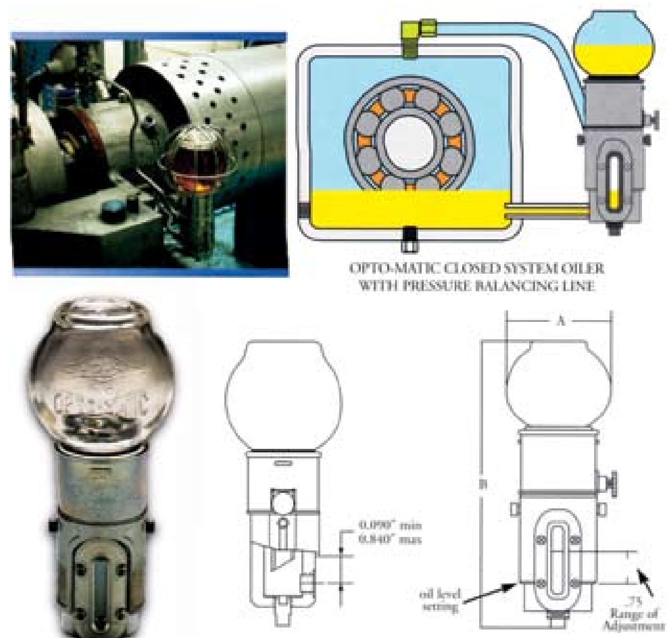


Figure 4. Pressure-balanced lubricators. *Source: Trico Mfg. Corporation.*

Oil Level Issues with Pressure-Unbalanced ("Open System") Lubricators

The "unbalanced" constant level lubricator shown in Figure 3 (Ref. 1, p. 233) has been typical of semi-automatic devices used for replenishing oil lost due to leakage or evaporation.

Here, decreasing lube oil level in the bearing housing would also result in lowering the oil level in the surge chamber, i.e. the metal base under the transparent bowl in Figure 3. As the level in the surge chamber goes down and the oil no longer surrounds all of the slanted tube, ambient air bubbles through to the top of the transparent bowl and oil feeds down from this bowl until the rising oil level again fully surrounds (and in effect "seals") the slanted tube. In open system lubricators, the oil level in the surge chamber is always in contact with the ambient air, as will the oil level in the bearing housing if this housing has a large vent to atmosphere.

However, this description of events is accurate only if the bearing housing interior is at the same pressure as the ambient air in contact with the oil level at the slanted tube. If there is *not* a large housing vent, and if the bearing housing is sealed with an effective protector seal ("bearing isolator"), a housing pressure slightly in excess of ambient will often result.

According to Bernoulli's Law of physics, such elevated pressures will cause the oil level in the bearing housing to drop and the oil level in the surge chamber of non-pressurized constant level lubricators will rise. In addition, slight compression of the air trapped in the transparent bowl will occur, which further explains the not-so-consistent level repeatability of many pressure-unbalanced (open system) lubricators.

Pressure-Balanced ("Closed System") Lubricators

As the term implies, *pressure-balanced* constant level lubricators (Figure 4, also Ref. 1, p. 234) incorporate a balance connection ensuring equal housing and surge chamber pressures.

The most important difference between the pressure-unbalanced ("open system") lubricators of Figure 3 and the pressure-balanced ("closed system") lubricators of Figure 4 is in the air supply. The air required to cause feeding when the oil level drops in the former (Figure 3) comes from the atmosphere, whereas in the latter (Figure 4) it comes from the pressure balancing line connected to the bearing housing.

The pressure balancing line must be suitably sized and configured. Nevertheless, it is not realistic to expect that absolutely constant oil levels can be maintained. Oil surface tension, configuration of slanted tube (Fig. 3), oil temperature variations, and compressibility of air in the transparent bowl are among the factors that influence oil levels. Technical information available from vendors (Ref. 3) explains that, for oil feeding to be initiated, the pressure difference between the air trapped in the top of the transparent bowl and the air in the bearing housing will have to be at least ½-psi. This feeding is cyclic and may eventually stabilize, but the overall effect may be a higher-than-intended oil level in the bearing housing.

Thus, closed system lubricators do not allow ambient air to contact the slanted feed tube and there is internal pressure balance. Compared with open systems that always have the surge chamber flooded with ambient air, closed systems are far less likely to have oil levels at the bearing drop precipitously. Moreover, a pressure-balanced (closed) lubricator system is needed to prevent water and airborne contaminant entry. Used in conjunction with appropriate housing seals, the service life of the lubricant can be greatly extended and bearing failure risk minimized with closed systems.

It must be realized, however, that many constant-level lubricator models with either the "open" or "closed" configuration use caulking at the surfaces joining the transparent bowl to the sleeve surrounding the surge chamber. Cracks can develop in this caulking after long-term service or after numerous temperature cycles. When that happens, rainwater can enter and contaminate the lubricant. That is just one more of the reasons why full sealing of bearing housings and alternative methods of maintaining oil levels are well worth considering.

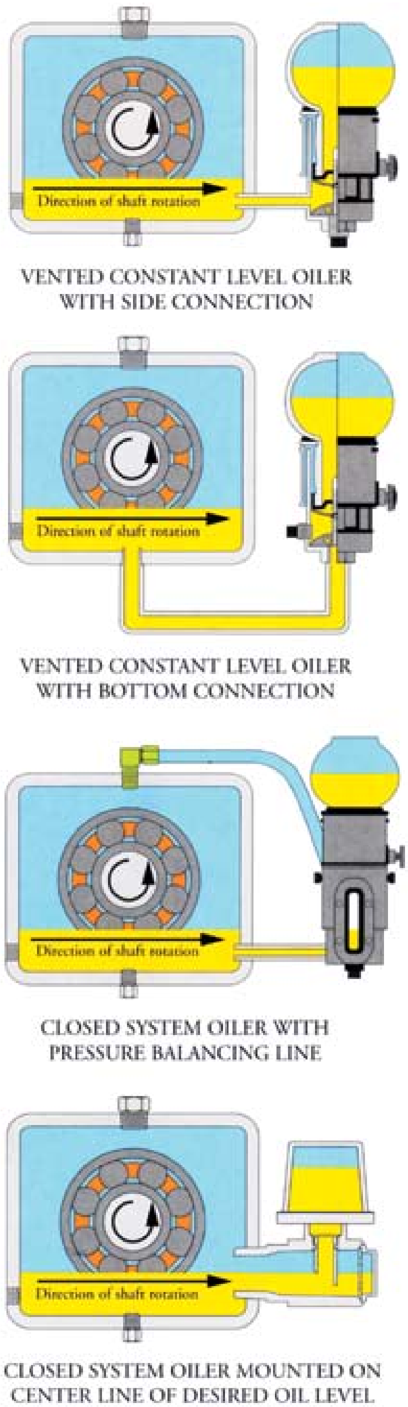


Figure 5. Vented and closed system oilers. Note mounting directions and shaft rotation.

Preferred Mounting Locations and Oil Level Upsets

Lube oil levels are also influenced by constant level lubricator mounting locations relative to shaft direction of rotation.

If these lubricators are mounted with bottom center connections, they serve as the collection point for oil sludge and wear debris. Bottom center connections would be acceptable in installations and at sites that practice frequent and timely oil replacement. But it would also presuppose that the piping is sturdy and resists inadvertent deflection if something (or someone) were to bump into it.

Sites that simply will not replace oil often and wish to use constant level lubricators would do better to mount constant level lubricators at the side of a bearing housing. However, the unidirectional mounting preference of constant level lubricators (both closed and open systems) should be observed in that instance (Figure 5, also Ref. 1, p. 233).

A side-mounted constant level lubricator belongs on the side where the shaft rotation arrow points upward. In other words, if the shaft rotation is clockwise, the up-arrow is on the left and the constant level lubricator should be mounted on the left.

With counterclockwise direction of shaft rotation the up-arrow would be on the right, and the lubricator should be mounted on the right side of the bearing housing. Rotation in the “wrong” direction, or mounting on the “wrong” side, will require slightly larger level fluctuations before feeding is initiated from the transparent bowl. In other words, the difference between the lowest level at which feeding commences and the level at which feeding stops is a function of mounting location and direction of shaft rotation.

Bearing Protector Seals Can Influence Oil Levels

As recognized by the stipulations of API-610, two types of bearing protector seals qualify for reliability-focused pump applications. They are either non-contacting labyrinth seals or contacting-face seals.

Non-contacting seals offer the advantage of extremely low friction and constitute a bearing housing vent, essentially a tiny relief port that allows small amounts of air to move across the assembly.

In contrast, there is no venting with face-contacting bearing housing seals and they work flawlessly as long as a thin film of oil separates the faces. Like all face-type seals, they cannot operate with zero friction. On the other hand, they have the advantage of simply not allowing any contaminants to enter the bearing housing. But no face-contacting seal or bearing will be reliable in situations that disregard the well-publicized cleanliness requirements of a workplace handling seals and bearings, or where metal chips are left in the bearing housing after a machining operation (Ref. 4, pp. 376-377).

Therefore, careful workmanship and good procedures go hand-in-hand with modern bearing protectors. Modern bearing protectors, and this diligent care approach, should be much preferred over relaxed procedures and inferior bearing housing protection. Moreover, magnetic seals with magnets not encapsulated are particularly vulnerable in less-than-clean situations.

Except for oil mist applications, a bypass opening beneath a bearing (Ref. 1, p. 144) is normally designed into the bottom of the housing bore surrounding a bearing outer ring, as shown in Figure 6.

Depending on housing-internal configuration (e.g. bearing design and face contact geometries), small differences may exist in the pressure acting on the lube surface between bearing and face seal, as opposed to the pressure acting on the oil in the remainder of the bearing housing. This would explain the occasional reports of level changes after retrofitting contacting face style bearing housing protector seals.

Desiccant Breathers and Expansion Chambers

Desiccant-breather combinations may address the *symptom* of a housing-internal pressure-related shortcoming and might represent a fix, in some instances. Obviously, desiccant-breathers are a maintenance item that will become part of the budgetary equation.

A somewhat similar point could

be made for expansion chambers (Figure 7) which will reduce housing pressures in fully sealed bearing housings.

They serve no technically advantageous purpose in vented applications. In fully sealed housings, the pressure increase is a function of temperature differences with ambient environment. The degree to which bearing housing pressures are reduced by expansion chambers is a function of the volumetric ratio of the chamber and the bearing housing volume.

If a pump has a relatively large bearing housing volume and/or low temperature rise, an expansion chamber will serve little purpose. As anticipated by fundamental gas laws and ample field experience, contacting-face magnetic bearing housing seals with proper face lubrication will not fail when they encounter the temperature-induced pressure rise associated with closed, non-vented bearing housings. This reinforces our findings that expansion chambers will rarely be needed and their use, in most applications, is rather discretionary.

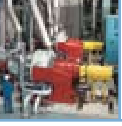
Ranking the Different Options

With so many different variables influencing oil levels in pump bearing housings, no statistics are available that allow definitive cataloging or linking of factors.

There are truly hundreds of combinations of oil application, bearing bypass options, and venting arrangements. There are also some rather unreliable wet sump oil mist applications, or dry sump oil mist introduced in a non-optimized manner (i.e. arrangements that do not conform to 10th Edition of API-610), pressure drops through desiccant containers, mismatched bearing housing and expansion chamber volumes, and so forth.

Nevertheless, even the highly subjective rankings of just a few of the more popular configurations may prove valuable if the rankings are based on several decades of experience. So, remembering that we're referring to rolling element bearings in process pumps and giving a score of 100 to the best observed, and a score of 10 that might just barely pass, reference the chart on page 32.

Finally, the observations made by the engineer-distributor that are mentioned at the outset are not unusual. He observed that, in his client's refinery and upon being fitted with face-type housing protector seals, several pumps experienced oil level fluctuations. He reacted properly to the available data and implemented measures to stabilize the oil levels. A desiccant breather relieves the pressure in pump bearing housings with unknown interior geometry, or



Configuration	Score
Jet Oil (spray of oil aimed at rolling elements) and face-type magnetic end seal	95
Jet Oil (spray of oil aimed at rolling elements) and best available rotating labyrinth seal	90
Jet Oil (spray of oil aimed at rolling elements) and no sealing	75
Dry Sump Oil Mist and face-type magnetic end seal	93
Dry Sump Oil Mist and best available rotating labyrinth seal (LabTecta RoLa)	89
Dry Sump Oil Mist and no sealing	71
Sight glass, flinger disc, face-type magnetic seal	88
Sight glass, flinger disc, best-available RoLa seal	83
Sight glass, flinger disc, no seal	58
Closed System constant level lubricator and face-type magnetic end seal, ***with*** an additional balance line ascertaining equal pressures on both sides of bearing(s). With flinger disc, DN-value not exceeding 8,000	84
Same, but with oil ring(s) and DN >8,000	60
Closed System constant level lubricator and face-type magnetic end seal, additional balance line between bearing and face-type seal not needed because flinger disc is less DN or level-sensitive	84
Closed System constant level lubricator and face-type magnetic end seal, ***without*** additional balance line ascertaining equal pressures on both sides of bearing(s), but with oil ring(s) and DN>8,000	50
Closed System constant level lubricator and best-available RoLa end seal, ***with*** additional balance line ascertaining equal pressures on both sides of bearing(s). With flinger disc	84
Same, but with oil ring(s) and DN >8,000	60
Closed System constant level lubricator and best-available RoLa end seal, additional balance line not needed because flinger disc is less DN or level-sensitive	84
Closed System constant level lubricator and best-available RoLa end seal, ***without*** an additional balance line ascertaining equal pressures on both sides of bearing(s), but with oil ring(s) and DN>8,000	50
Open System constant level lubricator and face-type magnetic end seal, ***with*** additional balance line ascertaining equal pressures on both sides of bearing(s). Flinger disc	74
Same, but with oil ring(s) and DN >8,000	50
Open System constant level lubricator and face-type magnetic end seal, additional balance line not needed because flinger disc is less DN or level-sensitive	74
Open System constant level lubricator and face-type magnetic end seal, ***without*** additional balance line ascertaining equal pressures on both sides of bearing(s), but with oil ring(s) and DN>8,000	40
Closed System constant level lubricator and no end seal, ***with*** additional balance line ascertaining equal pressures on both sides of bearing(s). Flinger disc	44
Same, but with oil ring(s) and DN >8,000	30
Closed System constant level lubricator and no end seal, additional balance line not needed because flinger Disc is less DN or level-sensitive	64
Closed System constant level lubricator and no end seal, ***without*** additional balance line ascertaining equal pressures on both sides of bearing(s), but with oil ring(s) and DN>8,000	28
Open System constant level lubricator and no end seal, ***with*** additional balance line not needed because flinger disc is less DN or level-sensitive	64
Same, but with oil ring(s) and DN >8,000	27
Open System constant level lubricator and no end seal, with flinger disc (additional balance line not needed because flinger disc is less DN or level-sensitive)	24
Open System constant level lubricator and no end seal, ***without*** additional balance line ascertaining equal pressures on both sides of bearing(s), but with oil ring(s) and DN>8,000	18
Grease Lubrication	21

unknown windage (fan effects) caused by the inclined cages of angular contact bearings, etc.

Most important is the engineer-distributor's conclusion that installing a bulls-eye sight glass and omitting the constant level lubricator altogether would be an attractive alternative. That, in combination with a properly dimensioned flinger disc and suitable face-type magnetic bearing protector seal, and perhaps even a magnetic drain plug (approximately $3) to hold tramp metal chips, would be the smartest solution by far.

Until cost-effective and reliable "per point" applications become available, face seals and large sight glasses are the right solution for plants tired of fighting symptoms. It's certainly time to tackle the real sources of oil level problems in pump bearing housings.

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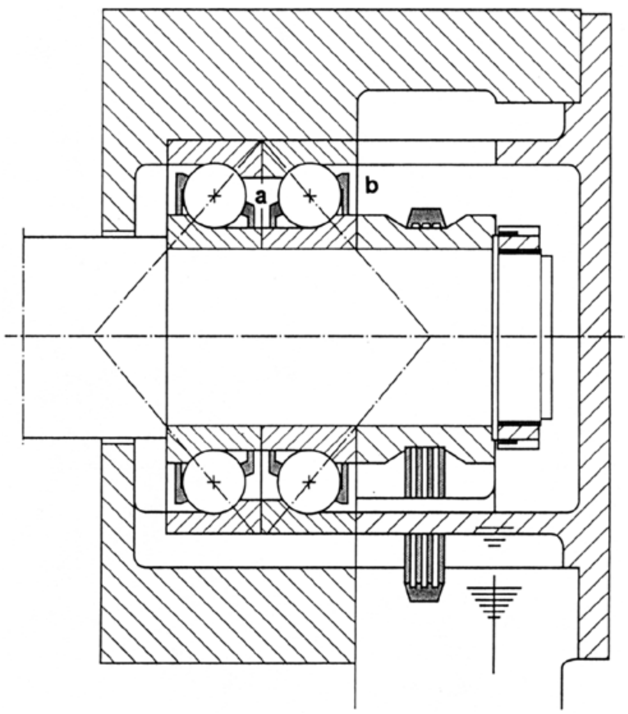

Figure 6. Grooved oil rings. Note bypass opening at the 6-o'clock position of the bearing. *Source: SKF USA Inc.*

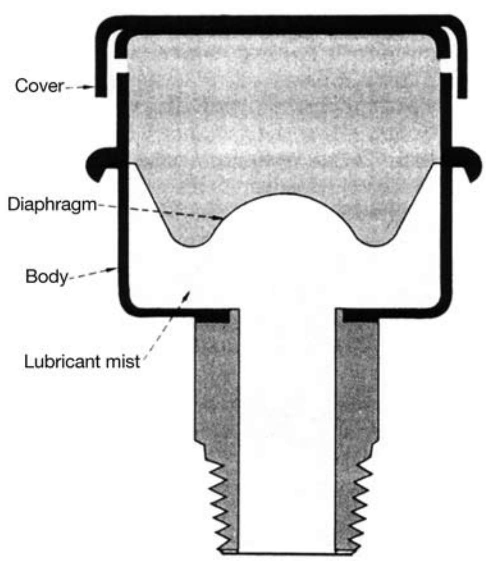

Figure 7. Diaphragm-type expansion chamber.

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