SKF Pulp & Paper Practices

SKF Global Pulp & Paper Segment | No. 20 | April 2017

Spare a little thought for your fans?

As an SKF Application Competence Centre engineer specialising in fluid machinery, I was more than happy to agree to write the introduction for this particular issue of SKF Pulp and Paper Practices. That's because the main focus this time is fans.

My experience is that many people focus on the main process equipment and I think it's fair to say that the balance of plant used in process industries sometimes doesn't get the attention that it really deserves. This is a little surprising. After all, I remember an engineering colleague of mine, now retired, who travelled the world visiting paper mills telling us that most of the problems he came across were related to ancillary equipment.

I don't think things have changed so much since my colleague retired a few years ago, so problems with balance of plant, including fans, are likely a source of frustration and cost to many of you too. If so, I hope you will be interested in the two articles in this issue of the newsletter. The first covers fan problems caused by inadequate support and the second at possible solutions when oversized bearings have been designed in.

Regards,

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The importance of proper support for your fans

The traditional bearing arrangement for the many rather small and lightly loaded fans found in paper mills is self-aligning ball bearings of 22 or 23 series mounted on adapter sleeves in plummer block housings. Most of these fans are grease lubricated and often function well for many years, but this is not always the case. I have visited mills were the fan bearing life was very short at three to four months after mounting new bearings in new housings. Even after optimizing the mounting procedure and the lubrication, the bearing life was still unacceptable.

All the self-aligning ball bearings that I inspected had a wide path pattern on the whole circumference of the inner and outer ring raceways, cage wear - and sometimes breakage - and fretting corrosion on the outside of the outer ring (see **figures 1-4**)

Eventually, I decided to change the bearing arrangement to deep groove ball bearings of 63 series with C3 clearance in a single PDN 3 series housing. The life of the bearings increased enormously, but at the time the reason for this was not clear to me.

I did recognize that the supports on which the plummer blocks were mounted were mostly made of 8 mm metal plate and had the idea that there should be a connection between this weak support and the short service lives of the bearings. Then I visited a steel plant where they had similar problems

Fig. 1: Path pattern on an outer ring raceway

Fig. 2: Path pattern on inner ring raceways

Fig. 3: Broken cage

Fig. 4: Fretting corrosion on an outer ring

and had measured vibration on their plummer blocks and the supports. They told me that vibration analysis had shown them that their plummer block housings were moving in an axial direction, in a pendulum motion, with a certain frequency (see **fig. 5**)

When a plummer block housing moves like this, the outer ring of the bearing in it makes the same movement relative to the balls and inner ring. When this happens with a certain frequency, the balls move in both a radial and an axial direction over the raceways. This creates a kinematic disturbance in the rolling contact between ball and raceway leading to surface distress when the lubrication film collapses.

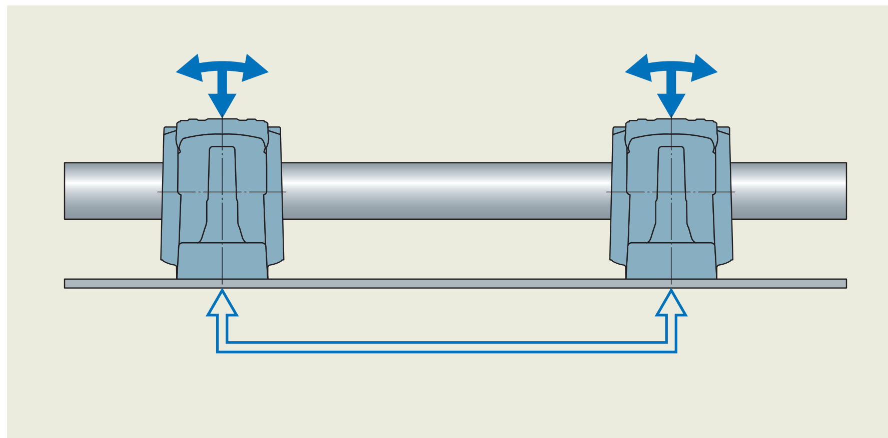

Fig. 5: The pendulum motion of inadequately supported housings

Better support with PDN two bearing housings

By mounting a single PDN two bearing housing instead of two separate plummer block housings the bearing arrangement is more rigid and the bearings will not experience the pendulum movement. The support also has a higher stiffness as the two bearing housing is connected over a larger area than would be the case with two individual housings. With such an arrangement, the outer rings of the bearings are more restrained in their movement (see **figure 6**).

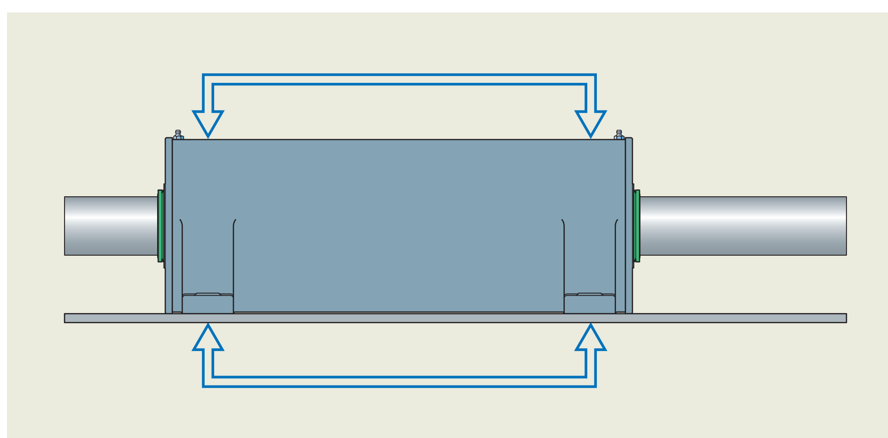

Fig. 6: Support in a PDN two bearing housing

PDN housings versus plummer block housings

At the time, I always opted for PDN 3 series housings with 63 series deep groove ball bearings because the housings were always available from stock. Whenever possible, I chose a PDN housing with the same height (H1) as the SNL/SE housings. An added advantage with the PDN housings is that the stepped shaft is 5-10 mm bigger than the straight shaft of an SNL or SE housing in most cases. This means that there is an increase in the shaft stiffness of the application.

Support rings for V-ring seals are recommended

With the relative high speed of the shaft, there is a risk that the V-ring can work loose from the cover or even from the shaft due to centrifugal forces.

The maximum circumferential speed for a felt seal is 4 m/s and for a V-ring seal it's 7 m/s (see **pages 64-7** of the *SKF bearing housings and roller bearing units* catalogue). Therefore, I always mounted a support ring against and over the V-ring.

The support ring has three functions: it keeps the V-ring in place, it works as an extra flinger by forming a gap with the cover of the housing and it protects the V-ring from getting damaged in operation (see **figure 7**). For the recommended dimensions of support rings, see the *SKF bearing housings and roller bearing units* catalogue.

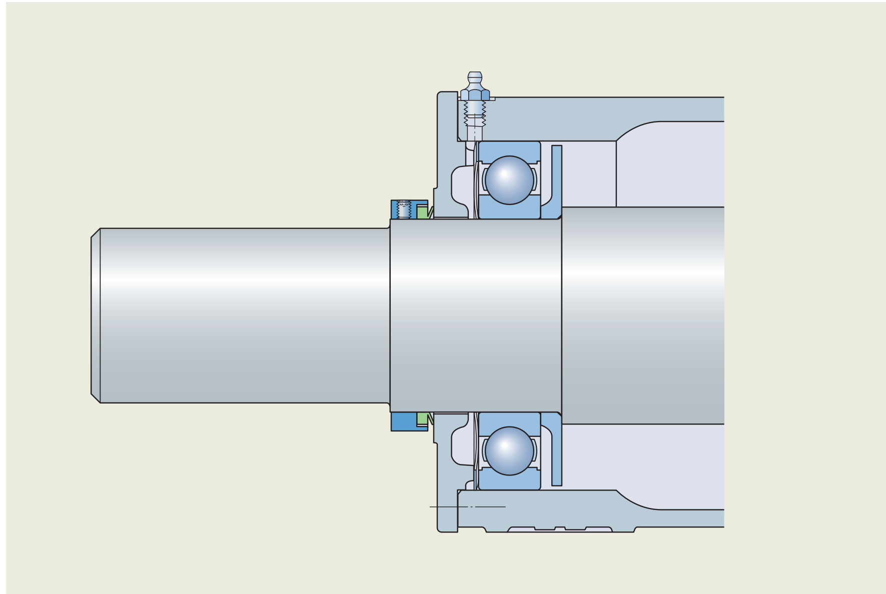

Fig. 7: A support ring for a V-ring seal

Recommended fits

SKF normally recommends a k6 shaft fit for deep groove ball bearings in these kind of applications. However, I experienced a few cases of creep between the inner ring and the shaft due to relative movement when the k6 fit was at its minimum (see **figure 8**). Consequently, I always use a m5 shaft fit.

According to the SKF *Rolling bearings* catalogue for d = 65 mm, a k6 fit results in –6 to –32 µm interference while m5 is –15 to –35 µm.

As I mount a 63 series deep groove ball bearing with C3 clearance, there is no risk of preloading the bearing during operation with the tighter fit. Furthermore, the maximum interference fit for k6 and m5 tolerances is roughly the same.

Note that C3 clearance also has a positive effect on bearing life as the axial force on the shaft caused by pressure difference in the fan is lower because the contact angle between the inner and outer ring is larger than it would be with normal clearance.

Support

The support must be sufficient. I prefer approximately 30 mm, but this depends on the size of the bearing housings.

Recommendations for the tolerance of the support are given on page 36 of the SKF *SNL plummer block housings* brochure and are reproduced below.

- Check that the roughness of the support surface is R_a 12,5 µm. The flatness (planicity) tolerance should be to IT7. Make sure that the mounting surface is clean. If the mounting surface is painted, the paint has to be removed. If shims are used, the whole surface must be covered by the shims. The mounting surface (frame) must be designed to accommodate actual load, vibrations and settings.

Example for the flatness (planicity) tolerance:

- The feet of bearing housing SNL 524-620 have a length (L) of 410 mm and a width (A1) of 120 mm.
- According to ISO the length dimensions 410 mm has a IT7-tolerance of 63 µm.
- According to ISO the length dimensions 120 mm has a IT7-tolerance of 35 µm.

These values can be checked with feeler gauges and you can visually check whether a support is flat or not by laying a ruler on it and shining a flashlight underneath it.

Fig. 8: Evidence of ring creep on a bearing inner ring

Reference speed

The reference speed of deep groove ball bearings is slightly higher than that of self-aligning ball bearings. A 6313 deep groove ball bearing, for example, has a reference speed of 10 000 r/min whereas for a 2313 K self-aligning ball bearing it's 9 000 r/min. As such, you can expect a lower running temperature from a deep groove ball bearing. In practice, we measured 10 degrees Celsius lower temperature on PDN two bearing housings compared to SNL/SE plummer block housings.

Other advantages with PDN housings are better heat removal because of their greater mass and the avoidance of kinematic disturbance due to their greater rigidity.

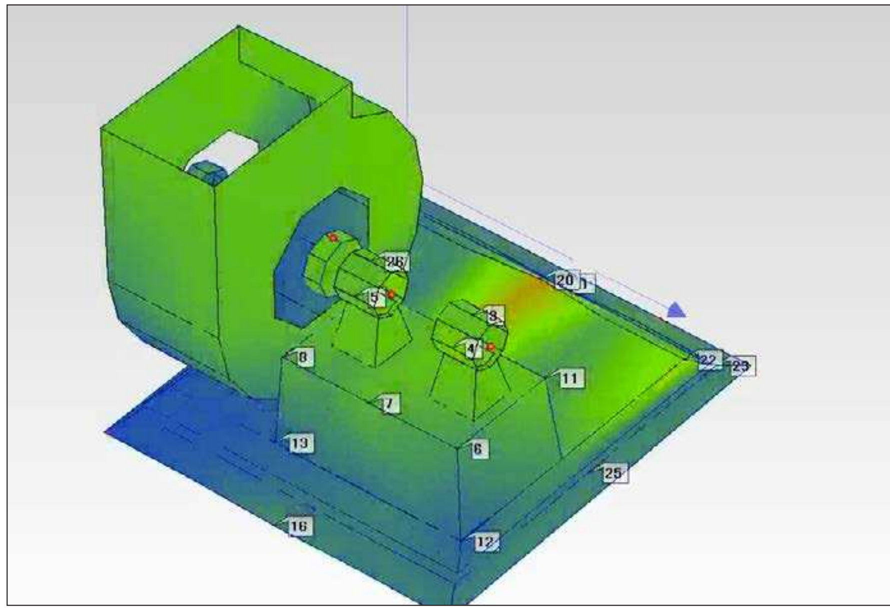

Fig. 9: ODS plot before the rebuild

Operating Deflection Shape (ODS)

SKF uses ODS to measure structures and to decide how they can be strengthened to overcome any weaknesses. By measuring and displaying the dynamic movement behaviour of a running machine, problems in the structure are visualised. When combined with knowledge of the frequency of the structure, this technique can be used to solve structural problems.

While changing a fan bearing arrangement from two SNL plummer block housings to one PDN housing at a paper mill where we have a service contract, I had the opportunity to see ODS measurements before and after the rebuild thanks to the support of a colleague in our condition monitoring department.

There was a clear difference in the movement of the bearings before and after the rebuild with the two SNH plummer block housings seen to be moving against each

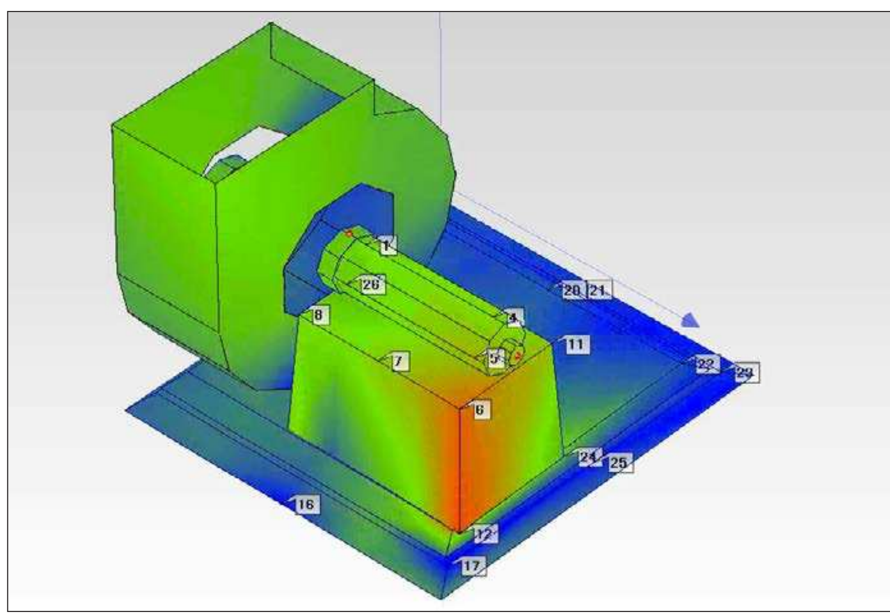

Fig. 10: ODS plot after the rebuild

other (see **figures 9 and 10**). Note that the ambient temperature was different when the two measurements were taken.

Condition Monitoring

For critical fans in paper mills, I recommend condition monitoring. Ideally, sensors should be mounted in both radial and axial positions as taking measurements in both directions gives a very good overview of what is happening to a bearing (see **figures 11** and **12**). The axial sensor should be mounted on the housing foot if space allows this.

Lubrication

Due to the relatively high speed, my standard lubricant choice is SKF LGHP 2 grease, but paper mills have supplied their own grease on occasions.

The PDN two bearing housings are well designed for relubrication with the bearings at both housing ends. The flinger rings move old grease into the middle of the housings where there is space enough for it to collect (see **figure 13**)

Fig. 11: Radial sensor placement

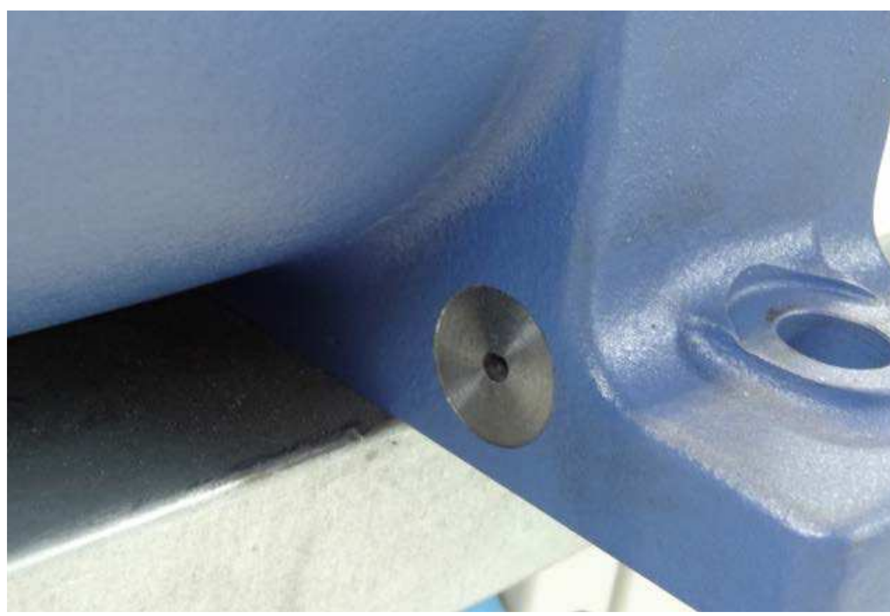

Fig. 12: Housing foot prepared for axial sensor placement

Conclusion

The PDN two bearing housing is a reliable solution for fans that experience bearing service life problems due to weak support of their existing plummer block housings. Over the years, I have mounted many of these housings on fans and seen the life increase from three to four months to six to eight years.

There is one disadvantage compared to using two plummer block housings though. This is that the fan wheel has to be dismounted from the shaft to change the PDN housing after six to eight years in operation. That said, this has never led to the paper mills that I have worked with refusing to adopt the PDN housing solution.

Bearing arrangements with two separate SNL plummer block housings are also a reliable solution particularly where there is an issue with shaft bending and/or bearing misalignment. However, they do need to have a support structure that is rigid enough in order to perform well.

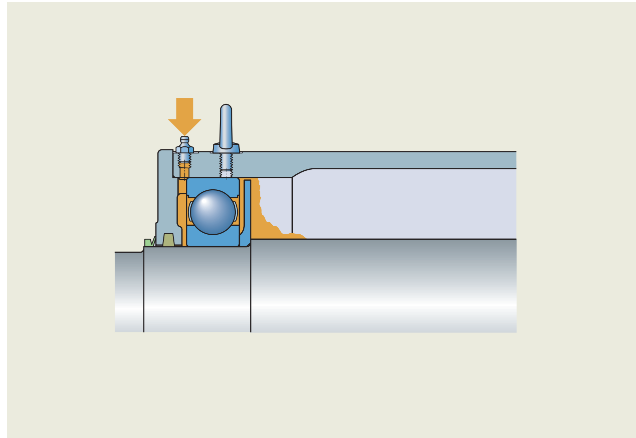

Fig. 13: Relubricating PDN housings

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Issues with oversized bearings and a quick solution

Some years ago, a customer complained about the very low service life on a spherical roller bearing mounted on a hot gas fan. They were getting just two weeks' life. The bearing was changed, but the new one didn't last any longer. They mounted another bearing, but it failed and the housing caught fire.

I arrived at the mill with very little information. Nothing had been sent to me as my visit had been planned only the day before. All I knew was that the problem was with an SKF bearing in an SKF housing on a hot gas fan. I was taken to see what was left of the failed bearing and housing, but the bearing components were so badly damaged that there were no clues about the root cause of the failure.

I did know that the bearing was running at high speed in a hot environment and that it was mounted in an SKF SOFN housing (see **figure 13**). Note that these housings are now mostly replaced by SKF SONL ones (see **figure 14**). Both SOFN and SONL housings have pick-up rings that hang loosely on a sleeve on the shaft on one side of the bearing and contact the oil bath in the lower half of the housings (see **figure 15** where the ring is shown with a red circle). When the shaft rotates, the ring follows and transports oil to a collecting groove to allow the lubricant to get to the bearing. This allows higher speeds than would be possible with standard oil bath or grease lubrication. Furthermore, the oil in the housing can be cooled though it wasn't in this case.

There was no sign of the pick-up ring with the damaged bearing and housing, so I asked if the maintenance team had forgotten to mount it. The answer I was given was such that I didn't dare ask if they had forgotten to fill the bottom of the housing with oil.

The customer then showed me a study undertaken by a consultant with the aim of increasing the reliability of the fan. As a result of this, the fan was modified by mounting larger bearings and the new calculated life wasn't in the thousands of hours, but in the thousands of years! I understood

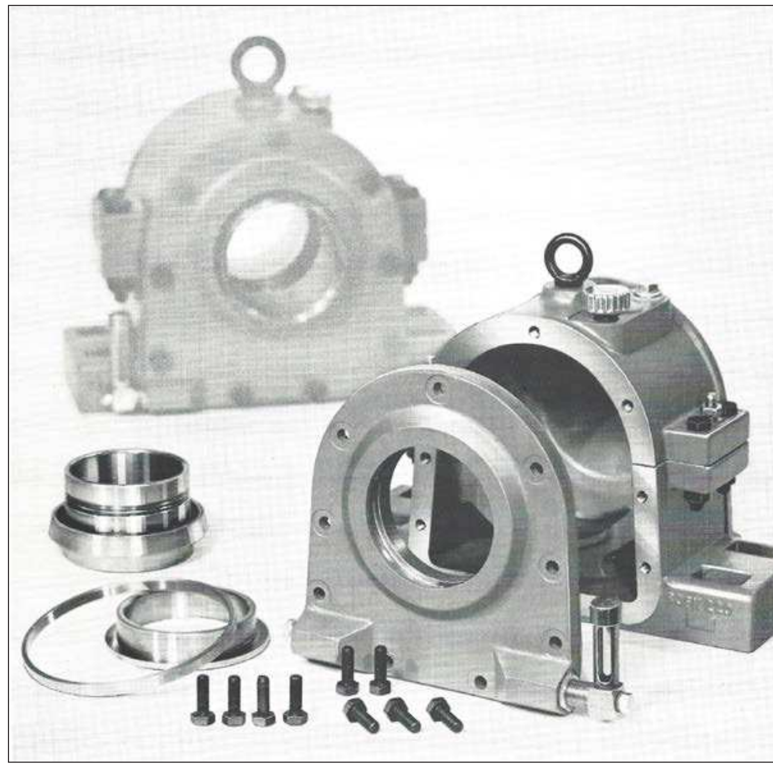

Fig. 13: SOFN housing

Fig. 14: SONL housing

straight away that the bearing was running without enough load and the failure was perhaps due to the rollers sliding rather than rolling in the bearing's loaded zone. I wasn't 100% sure because some bearings can operate well with loads much lower than recommended minimums. It's a grey zone; sometimes they work and sometimes they don't.

The customer wanted the problem fixed fast, so I suggested a very quick solution i.e. remove some rollers from the bearing. Doing this increases the load on the remaining rollers and forces them to roll in the loaded zone of the bearing. There was a long silence when I suggested removing two rollers from every three leaving only five rollers per row.

A bearing with fewer rollers was mounted under SKF supervision. The SKF service engineer involved told me that he was sure that the pick-up ring was forgotten during previous mountings and that lack of oil was the root cause of the failure. Either way, ten years' later we got an order from the same customer for the same bearing with fewer rollers.

This wasn't the first time that I had recommended removing rollers. I'd done it before for a 239/500 CA/W33 mounted on a deflection compensating roll which ended up with 14 rollers per row instead of the 38 that it originally came with. Also for a 23038 CCK/C4W33 mounted on a heated press roll which lost 50 per cent of its rollers. The most extreme example was a 23972 CC/W33 mounted on a printing machine where the bearing had been selected based on the shaft size. It started with 40 rollers per row and ended up with 8 left.

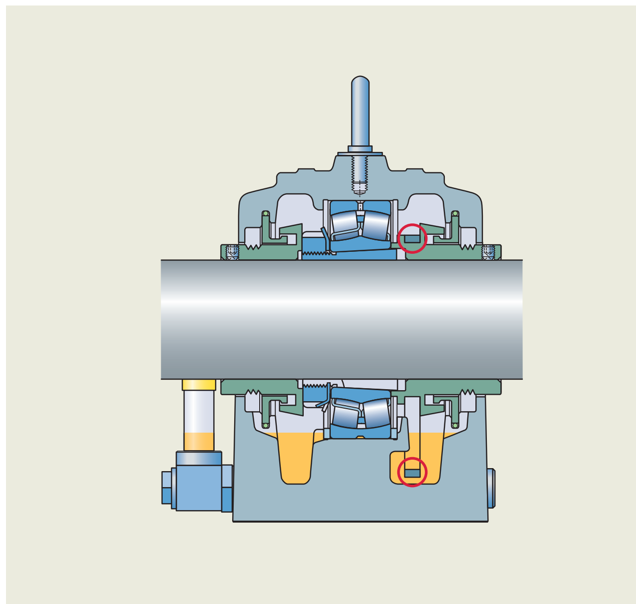

Fig. 15: Cut through of a SONL housing

Removing rollers can be a good solution when there is a risk of bearing damage caused by too low load, but there are two potential drawbacks:

1 The bearing might need to be used in another application where the full load capacity is required. This can be the case on soft calenders, for instance, where rolls can be mounted either in the upper or the lower position. When in the upper position, the NIP load reduces the load on the bearings whereas in the lower position, the NIP load increases it (see **figure 16**). As the rolls are stored with the bearings mounted on them, it's best to use bearings with full load capacity which are able to run without smearing between rollers and raceways when the loads are too low. In such cases, SKF recommends the NoWear coating on the rollers as it has been proven to eliminate smearing in soft calenders.

2 **Figure 17** shows, in a very exaggerated way, that total radial runout on a bearing also depends on the distance between two rollers and the internal radial clearance. The inner ring on the right will move up and down a greater distance than the one on the left when the bearing is rotating. **Table 1** shows the maximum addition deviation in the vertical plane for an SKF 23080 CC/W33 bearing with a radial load of 39,5 kN with different radial clearance values and numbers of rollers. Note that the inner ring is supposed to have the same wall thickness on the whole circumference, the rollers have exactly the same diameter and the standard bearing has 38 rollers per row. The ISO norm for this bearing accepts radial runout of 60 µm. Knowing the manufacturing precision for SKF bearings, a bearing with only nine rollers per row and 0,200 mm radial clearance would probably still have a total runout with the ISO range for a normal precision bearing. This means that taking away rollers doesn't create an unacceptable total runout in most bearing applications.

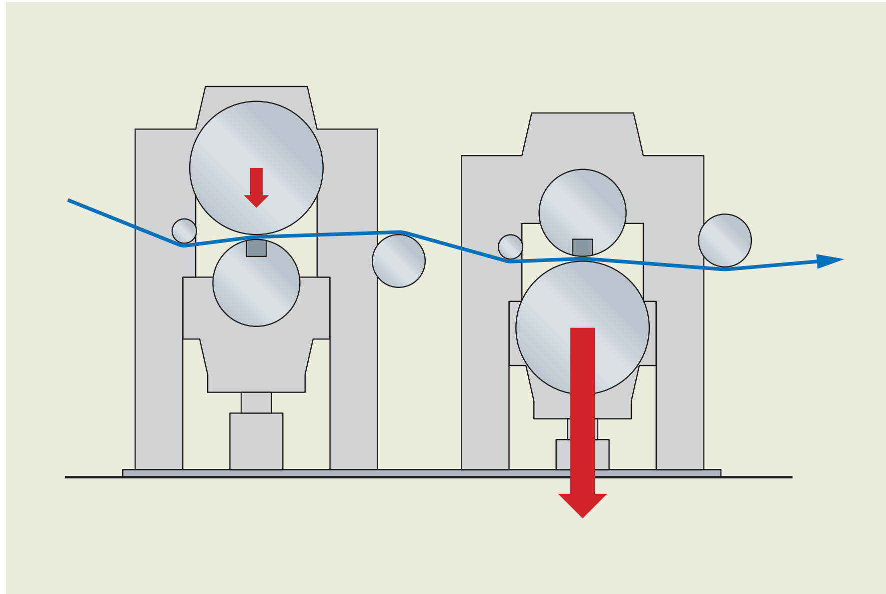

Fig. 16: Soft calender. The red arrows show load on the main bearings

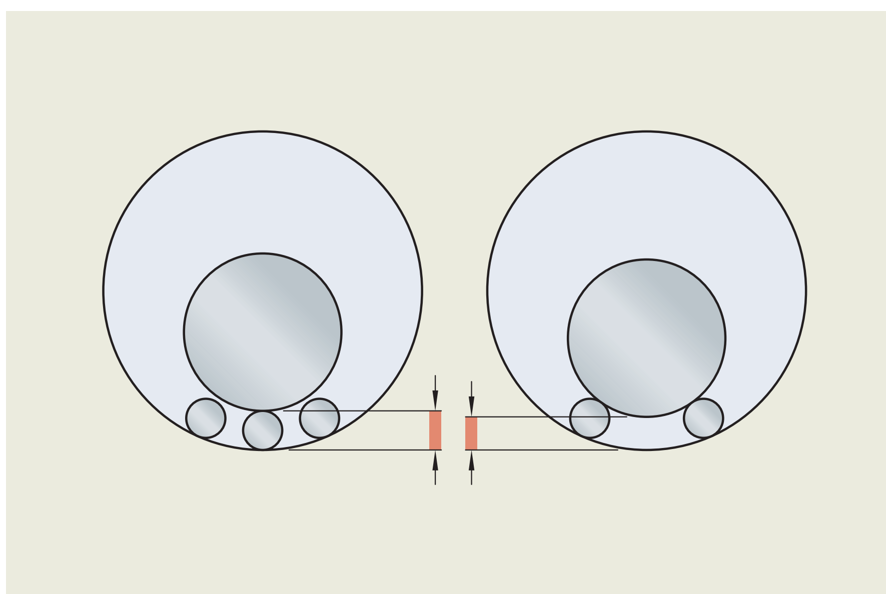

Fig. 17: The number of rollers and radial internal clearance have an influence on total radial run out

Table 1

Influence of internal radial clearance and number of rollers on the total runout in the same direction as the radial load

	Radial clearance = 0,000 mm	Radial clearance = 0,100 mm	Radial clearance = 0,200 mm	Radial clearance = 0,300 mm
9 rollers per row	0,6 µm	17,1 µm	38,5 µm	43,6 µm
10 rollers per row				43,9 µm
38 rollers per row				0.8 µm

So, how many rollers should be removed? The answer is simple if we limit ourselves to bearings that are easy to disassemble like SKF spherical rollers bearings and CARB bearings without the glass fibre reinforced PA46 cage. SKF gives a minimum recommended load based on the static basic load rating, C_0, and formulas to calculate the minimum load are given in SKF's *Rolling bearings* catalogue. The calculated minimum load is, in fact, the equivalent static minimum load and should be compared with the equivalent static bearing load. As the static basic load rating, C_0, is proportional to the number of rollers, if you remove half the rollers, C_0 is halved too.

Lost? Here's an example:

Bearing 22320 EK/C3, lubricated with grease. So, the minimum recommended load is equal to:

$P_{0m} = 0,01\ C_0$
$C_0 = 950$ kN
Then $P_{0m} = 9,5$ kN

Radial load is $F_R = 4$ kN and axial load is $F_A = 1$ kN
$P_0 = F_R + Y_0 \times F_A$
$Y_0 = 2$
$P_0 = 6$ kN
$P_0 < P_{0m}$ so there is a risk of bearing damage due to too low load.

Taking away half of the rollers leaves 8 rollers per row instead of 16. C_0 is divided by two and $P_{0m} = 9,5/2 = 4,75$ kN which is below P_0.

Recalculating the bearing basic rating life is also quite simple since the basic dynamic load rating C is proportional to $Z^{3/4}$ with Z being the number of rollers per row. With half of the rollers, the new C value will be equal to the original C value multiplied by 0,59. So for standard 22320 EK/C3, C = 847 kN. With 50% of the number of rollers, C = 847 × 0,59 = 500 kN.

If a bearing can be disassembled without damage, removing rollers is a quick way to solve the problem of low service life due to smearing under too low loads. You just have to know how to disassemble and reassemble the bearing and do it in a clean environment. However, if a bearing needs to keep its original load capacity or low radial runout is needed, SKF will recommend NoWear or black oxide coating.

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PUB 72/S9 11147/19 EN · April 2017