QUANTIFYING THE EFFECTS OF IDLER BEARING MISALIGNMENT ON BEARING LIFE

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INTRODUCTION

It is well known that excessive misalignment of roller bearings, such as those installed in conveyor idlers, leads to a reduction in the bearing life. The excessive misalignment in the conveyor idler bearing context is a result of the cumulative effect of acceptable manufacturing tolerances and assembly, together with the load that the idler must carry. However, the reduction in life and the determination of estimations in order to quantify the reduction is currently unknown, or at least withheld from general application.

According to Mike Stewart-Lord[1], 'Idler roll bearings should be able to accommodate a misalignment of up to 0.004 radians. This takes into account shaft deflection under load and manufacturing inaccuracies which cannot be avoided. The figure of 0.004 radians is reasonable because misalignment above this value can have an adverse effect on many seal designs. Spherical roller bearings are unacceptable for idler roll applications because of the large cost disadvantage. Self-aligning ball bearings are not suitable since the axial load, especially in the wing rolls, is too high for satisfactory operation.'

It is noted that 0.004 radians is very nearly 0°-14' and is the recommended maximum deflection for 420205 bearings, which were basically the most common idler roll bearing in South Africa at the time.

So it appears that very little has changed with respect to the effects of excessive deflection on the life of the idler bearings.

Excessive misalignment can be found in the actual manufacture of the idler rolls, where the bearing housing and end disc pressing is not properly inserted into the roll tubing. It is for this reason that a misalignment tolerance is proposed, and most of the major users and mining houses in South Africa specify a manufacturing tolerance. A conservative allowance would be 0°-6' (0.001745 radians). It is very difficult to actually measure this misalignment, so it is essentially a theoretical reduction in the bearing misalignment allowance as specified by the bearing manufacturer.

Of course, the major source of misalignment should be the actual loading of the idler roll as a result of the material being carried. However, there have been instances where even return idlers (which are intended to carry the belt mass only) have shown a serious reduction in operating life. This can only be ascribed to poor manufacture and a lack of maintenance of the assembly machines in the manufacturing process, which may require third party quality checks.

THE LOAD ON THE ROLL

The load that is applied to the most heavily loaded roll in an idler set may be determined from the equation

$$W_a = \left\{ g \left[\frac{B + (n \cdot Z \cdot f_1 \cdot f_2)}{n} \right] \times 10^{-3} \right\} \text{kN/m} \qquad 1$$

Where

W_a	=	Actual load carried by the most heavily loaded roll. kN/m
g	=	Gravitation constant 9.81 m/s^2
B	=	Mass of the belting. kg/m
n	=	Number of rolls in the idler set
Z	=	Material load. kg/m
f_1	=	Dynamic load factor
f_2	=	Burden factor

The material linear loading is found by

$$Z = \frac{C_{dc}}{3{,}6 \cdot S} \text{ kg/m} \qquad 2$$

Where

C_{dc} = Design capacity (t/h)
S = Belt speed (m/s)

The dynamic load factor is found from

$$f_1 = (C_x \cdot S^2) + 1 \qquad 3$$

The value of C_x may be determined from the following table.

Lump size range	Idler form	
	Fixed	Link Suspended
-5 +0	0	0
-25 +5	0.005	0
-100 +0	0.009	0.005
-100 +50	0.014	0.009
+100	0.050	0.020

Table 1. Lump size factor (C_x)

In the majority of cases, it is perhaps wise to use only the factor for +100 mm material, since idlers carried in the plant stores will have to be available throughout the plant, based on the belt width and not the material lump size.

The burden factor f_2 is found from Table 2. The values refer to a belt loaded to 100%. It must be appreciated that the burden factor will increase with decreasing belt loading. Typically, as the belt loading approaches about 10%, the burden factor will be approaching 1.

Idler type	**f_2**
3-roll	0.66
5-roll	0.47
2-roll Vee	0.60
Flat	1.10
Picking	1.00

Table 2. Burden factor at 100% loading

These equations and tables above are extracted from the CMA Diploma notes, Chapter 5.

An estimate of the burden factor related to the belt loading may be made from the following empirical equations.

For 3-roll idlers, $$f_2 = \left[\frac{((0{,}003 \cdot \theta - 0{,}48) \cdot p) + 100}{100}\right] \qquad 4$$

For 5-roll idlers, $$f_2 = \left[\frac{(0{,}8 \cdot \theta + 142) \cdot p^{-((0{,}0016 \cdot \theta) + 0{,}26)}}{100}\right] \qquad 5$$

Where

θ = Wing roll angle (degrees)
p = Percentage loading

It is noted that the value of f_2 for flat and 2-roll Vee form idlers is as shown in Table 2 and is applicable for all percentage loadings.

It can be shown that the allowable load for deflection is determined by

$$W_d = 0{,}92\left(\frac{(\delta - 0{,}001745) \cdot d^4}{(L - 90)}\right) \text{ kN} \qquad 6$$

Where

δ = The allowable deflection limit for the type of bearing (radians)
L = The gauge length of the roll in question (See SANS 1313/1)

The deflection-based idler pitch is then given by

$$s_{i(deflection)} = \frac{W_d}{W_a} \text{ (m)} \qquad 7$$

Since the deflection-based idler pitch is determined by Equation 7 shown above, it follows that the actual deflecting load

$$W_d = (s_i \cdot W_a) \text{ kN} \qquad 8$$

and this load is applied to the roll assembly.

Manipulating Equation 6, the actual slope of deflection may be determined as

$$\delta_{actual} = \frac{W_d \cdot (L - 90))}{0{,}92 \cdot d^4} + 0{,}001745 \text{ radians} \qquad 9$$

Where d refers to the idler roll shaft diameter (given as the idler series).

The maximum allowable slope of deflection is given for each type of bearing. For example, for seize resistant cage bearings, $\delta_{allowable}$ = 0.00407 radians, while for deep groove ball bearings with C4 clearance, $\delta_{allowable}$ = 0.00436 radians. It is noted that these values are reasonably conservative and are applied before any reduction for manufacturing tolerances.

For the analysis, the active slope of deflection

$$\delta_{active} = \delta_{actual} - \delta_{allowable} \text{ radians} \qquad 10$$

Applying this to Equation 6, the component of the load which causes the excessive deflection may be determined as

$$W_{d\,active} = 0{,}92\left(\frac{\delta_{active} \cdot d^4}{(L - 90)}\right) \text{ kN} \qquad 11$$

The excess deflection results in an axial force, when the balls are forced into the inner and outer races of the bearing.

The active radius of the contact between the race and balls may be estimated from

$$r_{active} = (0{,}45 \times D_b) - 0{,}5 \text{ mm} \qquad 12$$

rounded up to the nearest whole number. In this case, the parameter D_b refers to the bearing outer race diameter.

Some values of the diameter of the bearing (D_b), the bearing width (w_b), the dynamic load rating (C) and a recommended deflection (δ) may be summarised as shown in Table 3. Note that the maximum allowable deflection is based on C4 clearance deep groove bearings, set at 0°-15′. The maximum allowable deflection through the 420 series bearings is set at 0°-14′.

Type	Series	D_b	w_b	C (kN)	δ (rad)
420204*	20	47	12	13.5	0.00407
6204		47	14	15.6	0.00436
420205	25	52	12	15.3	0.00407
6205		52	15	17.8	0.00436
6305		62	17	26.0	
6206	30	62	16	23.4	0.00436
6306		72	19	32.5	
6207	35	72	17	31.2	0.00436
6307		80	21	35.1	
6208	40	80	18	35.8	0.00436
6308		90	23	42.3	

Table 3. Values of D_b, w_b, C and δ

*Note that the 420204 bearing is not generally available.

With reference to Figure 1 below, the lever arm a_b, is estimated at $a_b = (2 \cdot d)+5$ mm, where d refers to the idler series or roll shaft diameter. The value of a_b may be accepted as 45 mm, being a weighted average value, as applied in Equation 4.

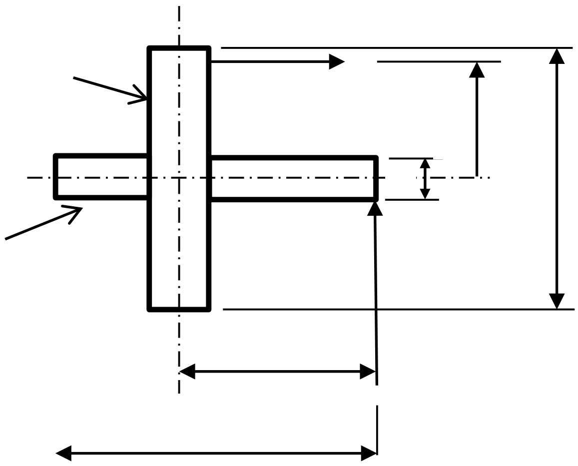


Figure 1. Idler bearing and shaft

The axial force

$$W_{axial} = \frac{W_{d\,active} \cdot a_b}{2 \cdot r_{active}} \text{ kN} \qquad 13$$

The equivalent dynamic bearing load may be determined as

$$P_{equiv} = X \cdot F_r + Y \cdot F_a \qquad 14$$

where the subscripts r and a refer to the radial and axial loads respectively.

It is clear that $F_r = 0.5 \cdot s_1 \cdot W_a$, where W_a is determined in Equation 1.

The axial load is accepted as $F_a = W_{axial}$ as determined in Equation 13.

The values for X and Y may be accepted as X = 0.44 and Y = 1.5 respectively, for the worst case condition. The values of X and Y are taken from Table 4 in the SKF General Catalogue 5000E.

Using the life equation from the CMA Diploma notes Chapter 5

$$W_L = \left[\frac{\pi \cdot D_i \cdot C^{\rho}}{3{,}6 \cdot H \cdot S}\right]^{\frac{1}{\rho}} \text{ kN for each bearing} \qquad 15$$

Where

D_i = Idler roll diameter (mm)
H = Life in hours
S = Belt speed m/s
C = Bearing load rating kN
P = 3 for ball bearings

Substituting $W_L = P_{equiv}$ (from Equation 15 above) and manipulating the equation, the bearing life may be estimated as

$$H = \frac{\pi \cdot D_i \cdot C^{\rho}}{3{,}6 \cdot S \cdot (P_{equiv})^{\rho}} \text{ hours} \qquad 16$$

Example

An overland conveyor 1 200 mm wide, running in 3-roll 35° ø127 series 25 carrying idlers fitted with deep groove ball bearings (6205), at a belt speed of 3.8 m/s handles 2 000 t/h and the belt mass is given as 28.1 kg/m. The idlers are pitched at 2.25 m. The lump size is 150 mm, which results in f_1 = 1.722 and f_2 = 0.66. Z = 146.199 kg/m.

C = 17.8 kN as obtained from catalogues; ρ = 3. Standard endurance life H = 40 000 hours. The roll gauge length is 460 mm.

Thus, from Equation 1

$$W_a = \left\{9{,}81 \times \left[\frac{28{,}1 + (3 \times 146{,}199 \times 1{,}722 \times 0{,}66)}{3}\right] \times 10^{-3}\right\} = 1{,}722 \text{ kN/m}$$

For $W_L = 2 \times \left[\frac{\pi \times 127 \times 17{,}8^3}{3{,}6 \times 40000 \times 3{,}8}\right]^{\frac{1}{3}} = 4{,}112$ kN, the ideal pitch for endurance would have to be less than $s_i = \left(\frac{W_L}{W_a}\right) = \frac{4{,}112}{1{,}722} = 2{,}388$ m

and the selected idler pitch of 2.25 m is ideal from the endurance aspect.

Equation 6 gives W_d = 2.25×1.722 = 3.875 kN and, from Equation 7

$$\delta_{actual} = \frac{3{,}875 \times (460 - 90))}{0{,}92 \times 25^4} + 0{,}001745 = 0{,}005734$$ radians (0°-19.712')

which is beyond the limit of 0.00436 radians for the specified bearings. Using Equation 8, the active slope of deflection is determined as

$\delta_{active} = 0{,}005734 - 0{,}00436 = 0{,}001374$ radians. Applying this to Equation 9, the deflection component $W_{d1} = 0{,}92\left(\frac{0{,}001374 \times 25^4}{(460 - 90)}\right) = 1{,}335\,\text{kN}$

From Equation 8, the active radius is estimated as $r_{active} = (0{,}45 \times 52) - 0{,}5 = 23\,\text{mm}$.

For the value of a_b = 45 mm, the axial force $W_{axial} = \frac{1{,}335 \times 45}{2 \times 23} = 1{,}305\,\text{kN}$ per side.
Applying Equation 11, with X = 0.44 and Y = 1.5 for bearings with C4 clearance for
F_r = 0,5(2,25×1,722) = 1,937 kN

$$P_{equiv} = (0{,}44 \times 1{,}937) + (1{,}5 \times 1{,}305) = 2{,}81\,\text{kN}$$

Substituting into Equation 13

$H = \frac{\pi \times 127 \times 17{,}8^3}{3{,}6 \times 3{,}8 \times (2{,}81)^3} = 7413$ hours (nearly) which is clearly unacceptable, when considering a required design life of 40 000 hours.

Increased bearing specification

Applying a 6305 bearing, C = 26.0 kN, D_b = 62 mm and δ = 0.00436 radians.

The value of W_L = 4.680 kN and the endurance based pitch would become 2.717 m which is acceptable.

The actual deflection remains as before, at δ_{actual} = 0.005734 radians.

The active slope of deflection then will be

$\delta_{active} = 0{,}005734 - 0{,}00436 = 0{,}001374$ radians and the deflection component will also remain at 1.335 kN as before.

With the larger bearing, the active radius is estimated as

$r_{active} = (0{,}45 \times 62) - 0{,}5 = 27{,}4$ mm and a value of 27 mm is used.

The axial force $W_{axial} = \frac{1{,}335 \times 45}{2 \times 27} = 1{,}113$ kN per side.

This means that $P_{equiv} = (0{,}44 \times 1{,}937) + (1{,}5 \times 1{,}113) = 2{,}522$ kN

Since the value of C for the 6305 bearing is increased to a value of 26.0 kN, the life is then found to be

$$H = \frac{\pi \times 127 \times 26^3}{3,6 \times 3,8 \times (2,522)^3} = 31956 \text{ hours, which is marginal.}$$

This solution has its own associated dangers, though. By far the majority of series 25 idler rolls in South Africa are made with either 420205 seize resistant cage bearings or 6205 TN9 C4 deep groove ball bearings. It is interesting to note that the clearance for the 420205 bearing is somewhere between C3 and C4 and could easily be described as C3½. It is also interesting to note that the 62 and 63 series bearings have a higher dynamic load rating than the conventional 420205 bearing.

Therefore, to specify a particular batch of idler rolls to suit a specific application (as in the example above) could easily lead to the incorrect specification of bearings being used for spares, with the premature failure of the rolls as a consequence. Idler rolls are sealed units and are not normally able to be disassembled without destroying them. The specification of the bearings inside the idler is therefore not easily seen and most certainly not visible without destroying the roll altogether. To try to identify with paint colours or stripes is also not very useful, because colours and stripes and so on can be easily painted over.

For this reason, to simply specify a heavier duty bearing in a consumable like an idler roll could lead to more maintenance problems during plant operation than they were intended to solve during the design phase. The result could therefore be much higher cost implications than any apparent savings that could have been made in the design stage.

OTHER SOLUTIONS

One of the most obvious solutions when faced with the excessive deflection of the idler bearings would be one of the following:

Option 1 The idler pitch could be reduced so that the linear load is reduced accordingly

Option 2 The idlers could be re-specified, with larger shafts

Option 3 The idlers could be designed with stepped shafts.

Discussing these, *Option 1* would immediately create a capex increase and may be unacceptable from the project costing aspect. Not only would a greater number of idler sets be required over the distance of the conveyor, but the conveyor structures (gantries and stringer modules) would have to be designed accordingly. In addition, the extra idler rolls would contribute to the annual mortality and therefore add to the system opex as well. Additional idlers as a result of a reduced pitch would also result in additional system tensions and power.

Option 2 is probably the simplest approach. However, the remarks with regard to additional capex would definitely apply. In addition, the larger rolls and shafts may impact on the standardisation of spares on the plant and could therefore become a bit of a stinging nettle.

It is also not good practice to over-size the shaft with respect to the idler shell diameter. A useful relationship is to consider the ideal shaft diameter to be about 20% of the shell diameter. On this basis, the ideal shell and shaft relationships may be tabulated as follows:

Roll dia. mm	76	89	102	127	152	165	178	219
Shaft dia. mm	15	18	20	25	30	35	35	45

Table 4. Ideal shaft-to-shell relationship for idlers

If the shaft diameter (bearing series) is oversized, the rolling resistance of the roll increases and a greater load is required in order to rotate the idler roll satisfactorily. Thus, the ø89 series 25 rolls would require a considerable load in order to rotate freely under load.

Option 3 is an one that has been mooted on several occasions in the past. There are some serious objections to stepped shafts, though, despite the attractive deflection benefits. Once again, idler rolls are sealed units and are not generally able to be disassembled without destroying them. The operating personnel would therefore be unable to easily see that the idler rolls have stepped shafts. In the case of an idler mortality, the staff would simply measure the ends of the shafts and order corresponding idler rolls, in good faith. Of course, these rolls would very rapidly fail and the idler supplier will be rather unfairly blamed. To obviate that, a well-disciplined management system would have to be introduced and spares holdings would have to be carefully determined.

Stepped Shafts

The basic deflection at the bearings of a stepped shaft can be expressed as follows:

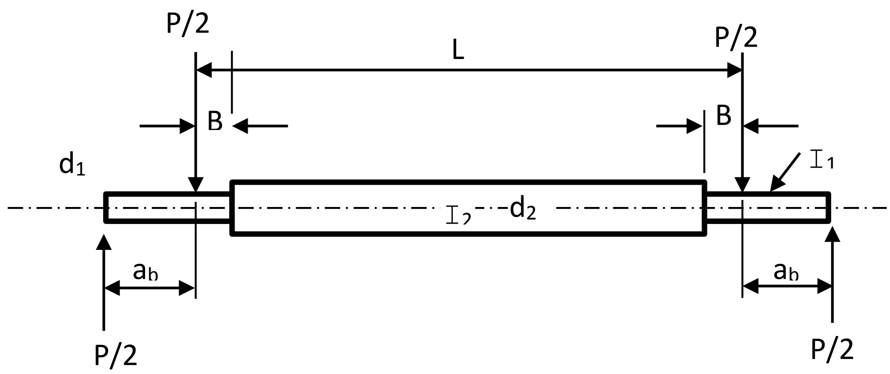


Figure 2. Stepped idler shaft

The parameter L is determined by subtracting twice the lever arm a_b from the gauge length.

$$\delta_{bearing}=\frac{64\cdot P\cdot a_b}{2\cdot\pi\cdot E}\left(\frac{B}{d_1^4}+\frac{\left(\frac{L}{2}-B\right)}{d_2^4}\right)\text{ radians}$$

In this case, P refers to the load on the roll and $P=(W_a\cdot s_i)$ kN from Equation 6.

Example

Assuming a 25/30 shaft, with the centre portion ø30 mm and using the width of the 6205 bearing 15 mm, B = 7.5 mm, with a_b = 45 mm

Using the earlier example, P = 3875 N,

$$\delta_{bearing}=\frac{64\times 3875\times 45}{2\times\pi\times 210000}\left(\frac{7{,}5}{25^4}+\frac{\left(\frac{370}{2}-7{,}5\right)}{30^4}\right)=0{,}00202\text{ radians (nearly 0°-7') and the}$$

effect of the stepped shaft is quite dramatic.

IDENTIFYING STEPPED SHAFTS

A possible standard method for identifying the presence (or otherwise) of a stepped shaft in an idler roll is presented thus:

Perhaps consideration of an additional machining on the idler roll shaft end should be considered. One idea could be to counter-bore the shaft end for a distance of (say) 2 mm, to an inside diameter specified as the difference between the series (d) and the step (D), as per the sketch, Figure 3 below.

That is $d_c=(2\cdot d)-D$ mm

Since the counter-bore will be under the closed end, there should not be a problem regarding the strength of the shaft in that area.

For the open end shafts, the counter bore will also be away from the actual bearing surface (at least, it should be) and the closest approach (for both open end and closed end) should be a minimum of 1.5 mm. This is again not too significant, because the closest approach is against the broached flat and that is not a shaft bearing surface.

However, it might become unworkable if the step is very large (say 40/25) and apart from such large steps being pointless, it may prompt a review of the idler specifications for that particular project.

This might be advantageous, because it may lead to limits on the practical stepped shaft. Nevertheless, in such a case, the counter-bore would reduce drastically.

Using the example of a 40/25 stepped shaft, the counter-bore would be reduced to $d_{counter}=(2\times 25)-40=10$ mm. The small counter-bore may then easily be overlooked by untrained personnel.

The basis of the stepped shaft for idlers is that the larger diameter should not be greater than the upper diameter of the inner ring of the bearing. This would be required in order to allow the maximum shaft diameter without interfering with the cage or rolling elements of the bearing.

Shaft dia. (Idler series)	Bearing series			
	62		63	
	Inner ring dia.	Max shaft	Inner ring dia.	Max shaft
12	18.50	17.0	19.5	19.0
15	217	21.0	23.7	23.0
20	28.8	27.0	30.4	29.0
25	34.4	33.0	36.6	36,.0
30	40.4	38.0	44.6	42.0
35	46.9	44.0	49.6	49.0
40	52.6	50.0	56.1	55.0
45	57.6	56.0	62.2	62.0
50	62.5	62.0	68.8	68.0

Table 5. Bearing inner ring diameter

The table indicates a range of shafts not necessarily in accordance with SANS 1313/1[5], but which would accommodate most idler requirements.

The maximum shaft step, based on avoiding interference with the bearing inner ring could therefore be written as

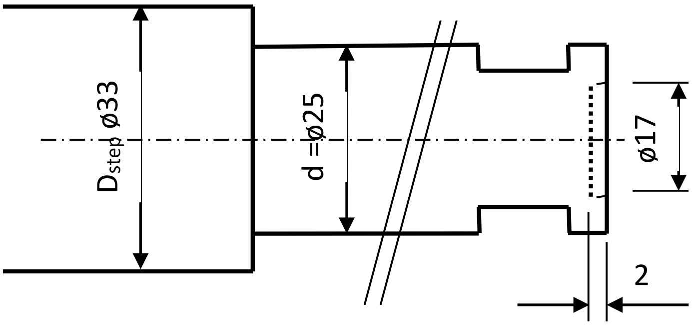
For 62 series bearings: $D = (1{,}18 \cdot d)+3$
For 63 series bearings: $D = (1{,}3 \cdot d)+3$

Of course, the tolerance of the counter-bore needs to be very loose and ISO H11 would be adequate. With this feature, when a roll needs to be replaced, the buyer will (should?) know that roll shafts with a counter-bore have stepped shafts and he/she can order accordingly, without the danger of accelerated mortality (of the rolls, of course) and the idler supplier unfairly getting a bad name.

Therefore, knowing the counter-bore diameter and the basic shaft series (bearing diameter), the stepped section can easily be found from $D = (2 \cdot d)-d_c$.

By creating a simple table, the buyers and operating staff can easily identify the shaft configuration for their particular plant and equipment.

Example, using a 25/33 stepped shaft



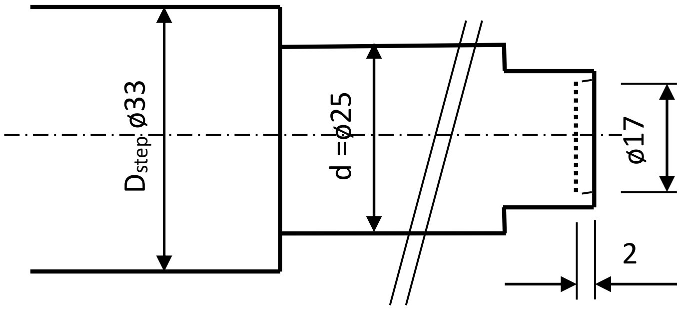
3A CLOSED END SHAFTS

Figure 3A and 3B. Stepped shaft identifying counter-bore

To identify the step, D = (2×25)-17 = 33 mm as expected, and the idler series would be specified as 25/33 in this case.

This basic exercise has highlighted that SANS 1313/1 does not seem to make allowance for open end shafts, other than to mention them (almost in passing) in Paragraph 3.15, Figure 7.

Even then, the dimension *N* (SANS 1313/1 - Table 2, columns 6 and 7) is the same for both open end and closed end shafts. The author personally would prefer to use closed end shafts (coming from the Anglo American stable), but open end rolls are



3B OPEN END SHAFTS

sometimes easier to fit for in-line idlers, which should be specified at transitions and convex curves.

An alternative identification method may be to hard-stamp the shaft ends. However, there is already provision for dating the assembly of the roll by stamping the shaft ends and this area could very easily become cluttered. In any event, many operations are quite liberal with paint and any markings on the shaft ends could easily become illegible.

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