A Simplified working procedure in designing the bearing bush for a 6” water-cooled submersible motor

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Abstract*: This paper presents a simple, working and verified design procedure for the bearing bush for a 6" water-cooled submersible motor. The paper shows the relation between eccentricity ratios, the limit for stable operation of a bearing bush system. Once the fluid film pressure is found out then load carrying capacity will have to be calculated. The power rating, speed, and weight of rotor must be known in order to estimate the torque and finally, we will get calculated diameter of the journal. From the electrical consideration, the paper shows how the useful torque and useful force can be referred in order to establish journal diameter. This simplified procedure presented in this paper will help the manufacturers in designing correct size of bearing bush for a specific motor. The resulting dimensions of bearing bush can be considered for the to generate a 3D cad model in order to conduct the FSI (Fluid Structure interaction) analysis.*

Keyword: *6" submersible motor, Bearing Bush, Fluid film pressure, useful torque, FSI.*

I. INTRODUCTION

A 6” water-cooled submersible motor uses a hydrodynamic journal bearing. The rotating journal or shaft is made from stainless steel whereas the stationary bearing bush is made from lead bronze [1] and [2]. Fig 1, 1.2 & 1.3 shows such bearing bush.

Fig 1. A Typical Submersible Bearing Bush

Fig 2. The bearing system for an 6" water cooled Submersible motor *(Courtesy: VIRA PUMPS, Kolhapur INDIA)*

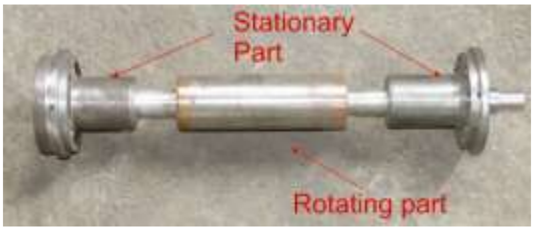

Fig 3. Rotating and stationary parts for an 6" water cooled Submersible motor *(Courtesy: VIRA PUMPS, Kolhapur INDIA)*

The surface finish of the rotating shaft must be between 0.7 to 0.72 microns. This can only be achieving by surface finish operations like grinding or burnishing [3] and [4]. The industrial practices are to just benchmark the design of some standard motors [5] and [6] One another approach in deciding the dimensions of the journal bearing is from the manufacturing catalog and selecting standard sizes. There is no comprehensive approach or method available with the manufactures to design such bearing system. It had observed that 90 % of the Submersible motor fails to owe to the bearing bush. It is a complex Tribological phenomenon and various factors affecting the design have to be addressed. H. Hirani [7] has highlighted some numerical treatment which enables to find out the fluid film pressure and load carrying capacity of the journal bearing. There is also an electrical engineering approach where the weight of rotor and useful torque is calculated and finally, dimensions of the bearing are determined [8] and [9]. A fine compromise is that which uses both the combined approaches to establishing the design. Journal bearings are analyzed by using fluid structure interaction (FSI) to find out deformation of bearing [10].

II ECCENTRICITY RATIO

At low eccentricity, the curve remains quite constant, whereas at higher eccentricity curve changes and critical speed increases. If we can operate at high relative eccentricity then we can definitely operate at high speed.

For the Journal Bearing as speed increases the relative eccentricity decreases and eventually it reaches the same speed as the critical speed and enters the unstable region.

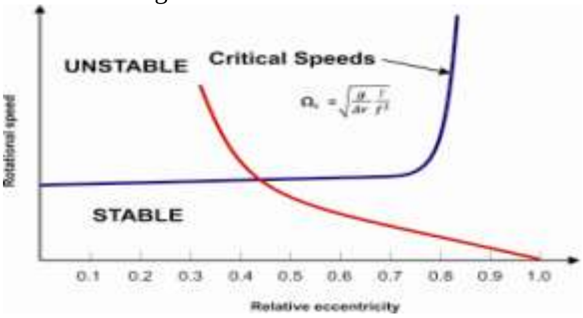

Fig 4. Relative Eccentricity v/s Rotational speed Curve

Disadvantage:

If we can operate at higher eccentricity it means that film thickness will be very small, resulting wear and vibration.

III LIMIT FOR STABLE OPERATION

When the stability limit reaches, the journal will exhibit "self-induced vibration". The vibration frequency is often near half the rotational speed and the phenomena is called "half frequency whirl"

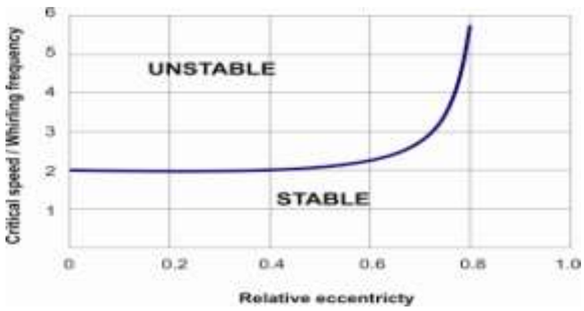

Fig 5. Relative Eccentricity v/s Critical speed Curve

Geometric configuration of journal bearing:

Film thickness h can be found out by analyzing geometric configuration

h_{max} = maximum film thickness
h_{min} = Minimum film thickness
o_j ,o_B = center of journal bearing
R_1 = radius of bearing
R_2 = radius of journal

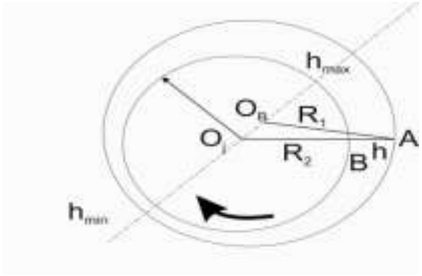

Fig 6.

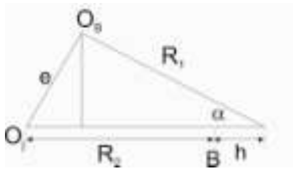

Fig 7.

e = eccentricity
o_{jB}= journal radius
o_{BA} = bearing radius
Consider equilibrium condition
h = e$\cos\theta$ + $R_1\cos\alpha$ – R_2
$\frac{e}{\sin\alpha}=\frac{R1}{\sin\theta}$α is very small
h = $e\cos\theta$ + R_1 – R_2
h = $e\cos\theta$ + C
h = C (1+ $\cos\theta$)

Deriving fluid film pressure:

$$h^3 \frac{\partial}{\partial z}\left(\frac{1}{\square}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x}$$

h = constant and $\vartheta$is not dependent on z

$$\frac{\partial^2 p}{\partial z^2}=\frac{6U\square}{h^3}\frac{\partial h}{\partial x}$$

Take integration

$$\frac{\partial p}{\partial z}=\frac{6\,U\,\vartheta}{h^3}\frac{\partial h}{\partial x}z$$

Again integration

$$P=\frac{3U\vartheta}{h^3}\frac{\partial h}{\partial x}\left(z^2-\frac{L^2}{4}\right)$$

Put value of h

$$P=\frac{3U\vartheta}{C^2(1+\varepsilon\cos\theta)^3}\left(\frac{-\epsilon\sin\theta}{R}\right)\left(z^2-\frac{L^2}{4}\right)$$

Using P=0 at Z=$\frac{+L}{-2}$

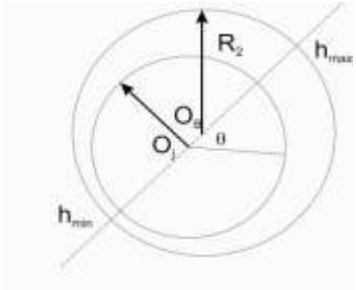

Fig.8

Load carrying capacity:

$$W\cos\emptyset = -\int_0^{\pi}\int_{\frac{-l}{2}}^{\frac{l}{2}} PR\cos\theta \, d\theta dz$$

$$W\sin\emptyset = \int_0^{\pi}\int_{\frac{-l}{2}}^{\frac{l}{2}} PR\sin\theta \, d\theta dz$$

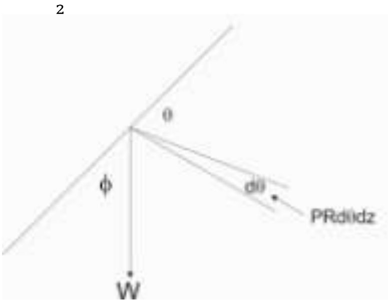

Fig.9

We need to integration to find W

$$W\cos\emptyset = -\frac{U\vartheta L^3}{2c^2}\frac{2\varepsilon^2}{(1-\varepsilon^2)^2}$$

$$W\sin\emptyset = \frac{U\vartheta L^3\pi}{4c^2}\frac{\varepsilon}{(1-\varepsilon^2)^{3/2}}$$

Taking square and adding

$$W = \frac{U\vartheta L^3\pi}{4c^2}\frac{\varepsilon}{(1-\varepsilon^2)^2}\{(\frac{16}{\pi^2}-1)\varepsilon^2+1\}^{1/2}$$

Attitude angle

$$\tan\emptyset = \frac{\pi}{4}\frac{\sqrt{1-\varepsilon^2}}{\varepsilon}$$

Calculation

We know

$$\varepsilon = \frac{e}{c}$$

Therefore consider ε is 0.6 and radial clearance is 0.05 put this value in above equation than we get eccentricity e equal to 0.03

Attitude angle
From the equation

$$\tan\emptyset = \frac{\pi}{4}\frac{\sqrt{1-\varepsilon^2}}{\varepsilon}$$

$$\emptyset = 46.32 \text{ deg}$$

Find x and y co-ordinate

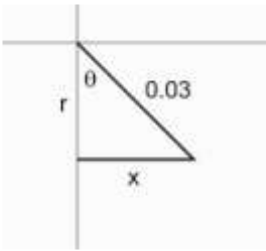

Fig.10.

$\sin\theta = \frac{x}{0.03}$;   X= 0.0217 mm

$\cos\theta = \frac{r}{0.03}$;   r = 0.0207 mm

IV DESIGN OF 6" SUBMERSIBLE MOTOR BEARING BUSH DIAMETER

In the case of the submersible motor rotor which consists of Stainless Steel shaft. On the shaft consists electrical laminations with copper ring mounted and brazed. Here, the bending is very much smaller than tensional load.

Power transmitted by shaft and torque in the shaft are related as:

$P = M_t \times \omega$ P= Power in watt
M_t = torque in N-m
w = angular velocity rad/sec

$$\omega = \frac{2\pi N}{60}$$

$$P = \frac{2\pi N}{60} \times M_t$$

$$M_t = \frac{30P}{\pi N} \text{ N-m} \quad \dots (1)$$

The shear stress and torque are related as

$$\tau = \frac{16\times10^3}{\pi d^3} \times M_t \quad \dots (2)$$

where M_t is equal

$$M_t = \frac{\pi}{16} \times \frac{\tau d^3}{10^3}$$

Than equation (1) and (2)

$$d^3 = \frac{16\times30}{\pi^2 N} \times \tau \times 10^3$$

Therefore,

$$d = 36.5 \times (\frac{P}{\tau N})^{1/3} \text{ mm}$$

The value of allowable shear stress depends upon service condition. for axially loaded shaft i.e. (rotor, bevel gear, spiral drive) the value is taken as 8 - 10 Mpa.

For Calculating bearing pressure:
Electrical consideration of rotor:
a) Useful torque (T) = $\frac{P}{2\pi N}$
Speed (N) taken in rps

b) Useful force (F) = $\frac{T}{D/2}$

Where D is diameter of stator bore size

w = useful force + w_1

w_1 rotor weight divided by two in N

Bearing Pressure (P) = $\frac{W}{LxD}$ N/mm2

Case A:

Design of submersible Bearing bush diameter of Power 15 hp and 30 hp, speed 2800rpm considering L/D = 1.25

We know

$$d = 36.5 \text{ x } (\frac{P}{\tau N})^{1/3} \text{ mm}$$

$$= 36.5 \text{ x } (\frac{15x0.745x1000}{10x2800})^{1/3}$$

d= 26.87 mm

D = d + 0.1 mm

Allowable Diametrical clearance is 0.1 mm

Useful Torque = $\frac{15x0.745}{2\pi x46.66}$

= 0.03811

Useful force = 1058.66 N

Weight of rotor is 15 kg

W = 1058.66 + 73.575

W = 1132.235 N

$P_b = \frac{1132.235}{33.71x26.97}$

P_b = 1.24 N/mm^2

Case B:

For 30 hp;

$$d = 36.5 \text{ x } (\frac{P}{\tau N})^{1/3} \text{ mm}$$

$$= 36.5 \text{ x } (\frac{30x0.745x1000}{10x2800})^{1/3}$$

d= 33.85 mm

D = 33.95 mm

W= 2117.32+117.72

W = 2235.32 N

Bearing pressure

P_b = 1.55 N/mm^2

V CONCLUSION

It is thus seen that how this simplified procedure can enable the designer as well as the manufacturer to rapidly develop the correct size of the bearing bush for a 6" water-cooled submersible motor. Using these dimensions a 3D Model can be developed which can be furthers used for the CFD Analysis using the FSI Technique.

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