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计入孔隙结构影响的复层含油轴承润滑特性分析*

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摘要: 基于 Darcy 定律和 Kozeny-Carman 孔隙方程,建立多孔质复层含油轴承的流体润滑模型,利用有限差分法数值模拟,分析复层结构和孔隙参数对含油轴承润滑性能的影响.得出结论如下:复层含油轴承润滑性能随轴承高度增大而变差,随孔隙率减小而变好,当总孔隙率一定时,较低的表层孔隙率有利于提高复层含油轴承润滑性能.因此设计复层含油轴承时,在保证一定孔隙含油量的前提下,应尽可能减小表层孔隙率.研究工作为复层含油轴承摩擦学性能分析与结构设计提供一定的理论基础.

关 键 词: 复层含油轴承; 流体润滑; 孔隙; 承载能力; 摩擦因数
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引 言

含油轴承又称多孔质轴承,因制造成本低,具有含油自润滑特性,广泛应用于汽车、仪表、农业工程、精密机械等领域.含油轴承在一定条件下工作时,能依靠自身孔隙中储存的油液持续向外析出,形成流体动力润滑油膜[1-2].基于这一前提,Morgan 和 Cameron 首次利用 Darcy 定律建立了含油轴承流体动力润滑模型,指出油膜承载能力和摩擦因数是含油轴承流体润滑性能的两个主要评价指标[3].此后,Darcy 定律在含油轴承润滑特性研究中被广泛采用,并通过与多种因素耦合,使数值模型更加符合实际[4].如 Gururajan 等[5]将 Christensen 随机粗糙度模型[6]应用于含油轴承润滑分析中.Chiang 等[7]和 Rao 等[8]在前者基础上分别引入偶应力流体和边界滑移效应.Shimpi 等[9]和 Kudenatti 等[10]分析了 Christensen 随机粗糙度和弹性变形对磁流挤压润滑特性的影响.王克用等[11]和李勇铜等[12]等采用 Brinkman 扩展 Darcy 定律研究了单层多孔介质孔隙通道内的流体流动和传热特性.以上研究都是以普通单层含油轴承或单层多孔介质为对象,没有考虑多孔质基体渗透率变化对润滑性能的影响.Srinivasan[13]和 Verma[14]研究发现,减小渗透率能阻止油液向多孔介质中渗漏,将提高轴承强度和油膜的承载能力,但同时也会降低孔隙含油量,使轴承自润滑性能变差.业已表明,复层结构的含油轴承能通过调整材料层间结构参数和渗透率配比,即保留一般单层含油轴承所具有的自润滑效果,同时

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也能克服孔隙结构对油膜承载性能带来的不利影响[15].Naduvinamani[16],Li[17]和 Rao 等[18]分别针对不同类型的复层含油轴承系统,研究了复层结构参数对其润滑性能的影响.含油轴承内部具有多孔结构,其渗透率不仅由外在结构直接决定,还与其内部孔隙密切相关.在多孔介质润滑分析中,常用 Kozeny-Carman 方程和 Irmay 方程来模拟孔隙结构和孔隙内部流体流动间的关系.最近,Patel 和 Deheri 对以上两种模型比较发现:在含油轴承润滑特性分析中,Kozeny-Carman 方程更加适用,这是由于含油轴承制备过程中,其内部粉末颗粒经高温烧结球化,球形颗粒间的孔隙相互连通组成含油轴承的孔隙通道,这与 Kozeny-Carman 方程的建模思想基本吻合[19].

本文以复层含油轴承为对象,基于 Darcy 定律和 Kozeny-Carman 孔隙方程,建立复层含油轴承流体润滑模型,分析复层结构和孔隙参数对含油轴承润滑性能的影响.研究内容为复层含油轴承的润滑性能分析及轴承结构和内部微观孔隙的优化设计提供一定理论参考.

1　复层含油轴承润滑模型

如图 1 所示, O_1,O_2 分别为轴和轴承中心,偏心距为 e, 内层、外层区域Ⅰ和Ⅱ的厚度为 T_1,T_2,渗透率为 k_1,k_2,轴半径为 R,轴承宽度为 b,轴承内圈和两层半径分别为 r_1,r_2 和 r_3,轴颈表面切向速度为 U_h,径向速度为 W_h.在轴承内圈建立如图所示 $Oxyz$ 直角坐标系.

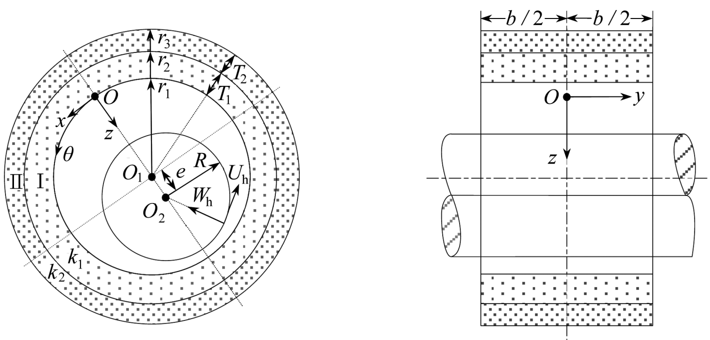

图 1　复层含油轴承几何模型

Fig. 1　The geometrical model of a multi-layer oil bearing

轴承工作过程中,两接触表面间任意点的流体膜厚为

$$h = C + e\cos\theta, \tag{1}$$

式中,半径间隙 $C = r_1 - R$.偏心距 e 和半径间隙 C 的比值为含油轴承的偏心率 ε.

1.1　修正 Reynolds 方程推导

假设润滑剂为不可压缩 Newton(牛顿)流体,多孔质基体具有均匀性和各向同性,流体流动为层流且在多孔质中满足 Darcy 流动,忽略流体惯性力作用,流体在油膜区中的流动满足 N-S 方程和流量连续性方程:

$$\frac{\partial p}{\partial x} = \eta\frac{\partial^2 U}{\partial z^2},\ \frac{\partial p}{\partial y} = \eta\frac{\partial^2 V}{\partial z^2},\ \frac{\partial p}{\partial z} = 0, \tag{2}$$

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0. \tag{3}$$

流体在多孔质基体中的流动满足 Darcy 方程和 Laplace 方程:

$$U_i = -\frac{k_i}{\eta}\frac{\partial p_i}{\partial x},\ V_i = -\frac{k_i}{\eta}\frac{\partial p_i}{\partial y},\ W_i = -\frac{k_i}{\eta}\frac{\partial p_i}{\partial z},\qquad i = 1,2, \tag{4}$$

$$\frac{\partial^2 p_i}{\partial x^2} + \frac{\partial^2 p_i}{\partial y^2} + \frac{\partial^2 p_i}{\partial z^2} = 0,\qquad i = 1,2, \tag{5}$$

其中，U_i,V_i,W_i 分别为多孔质中的周向流速、轴向流速和法向流速，内层、外层用下标 $i = 1,2$ 表示，如 U_2 表示多孔质表面的周向流速。

在复层轴承界面上满足压力和速度连续性边界条件：

$$y = -(T_1 + T_2)\text{ 时},\ \frac{\partial p_2}{\partial z} = 0,$$

$$y = -T_1\text{ 时},\ p_1 = p_2,\ k_1\frac{\partial p_1}{\partial z} = k_2\frac{\partial p_2}{\partial z},$$

$$y = 0\text{ 时},\ p = p_1,\ W_0 = W_1.$$

方程(5)两端对 z 积分，得到多孔质表面上的法向流速：

$$W_1 = -\left(T_1 + \frac{k_2}{k_1}T_2\right)\left(\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2}\right). \tag{6}$$

方程(2)两端对 z 两次积分并应用连续性方程(3)和界面边界条件，可得修正后的 Reynolds 方程：

$$\frac{\partial}{\partial x}\left[(h^3 + 12k_1T_1 + 12k_2T)\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial y}\left[(h^3 + 12k_1T_1 + 12k_2T)\frac{\partial p}{\partial y}\right] = 6\eta U_j\frac{\mathrm{d}h}{\mathrm{d}x}. \tag{7}$$

取无量纲参数：

$$\theta = \frac{x}{R}\quad (0 \leqslant \theta \leqslant 2\pi);\ Y = \frac{y}{b}\quad \left(-\frac{1}{2} \leqslant Y \leqslant \frac{1}{2}\right);$$

$$\psi = \frac{12k_1T_1 + 12k_2T_2}{C^3};\ H = \frac{h}{C};\ P = \frac{pC^2}{6\eta U_jR};\ G = \frac{C^2}{6U_j\eta R^2 b}g;\ F = \frac{fC}{\eta U_jRb}.$$

无量纲后的 Reynolds 方程为

$$\frac{\partial}{\partial\theta}\left[(H^3 + \psi)\frac{\partial P}{\partial\theta}\right] + \frac{R^2}{b^2}\frac{\partial}{\partial Y}\left[(H^3 + \psi)\frac{\partial P}{\partial Y}\right] = \frac{\mathrm{d}H}{\mathrm{d}\theta}. \tag{8}$$

无量纲 Reynolds 边界条件为

$$\theta = \frac{\pi}{2}\text{ 时},P = 0;\ \theta = \frac{3\pi}{2}\text{ 时},P = 0\text{ 且}\frac{\partial P}{\partial\theta} = 0;\ Z = \pm\frac{1}{2}\text{ 时},P = 0.$$

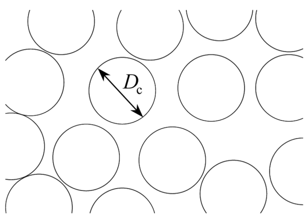

图 2 Kozeny-Carman 孔隙模型结构

Fig. 2 The pore structure of the Kozeny-Carman model

1.2 Kozeny-Carman 孔隙模型

含油轴承制备过程中，其内部粉末颗粒经高温烧结球化，球形颗粒间的孔隙相互连通组成

含油轴承的孔隙通道,Kozeny-Carman 模型正是基于这种思想,假设多孔轴承材料由平均直径为 D_c 的微小球形颗粒组成(如图 2 所示),则轴承渗透率可描述为

$$k_i = \frac{D_c^2 \varphi_i^3}{180(1-\varphi_i)^2}, \tag{9}$$

式中,φ_i 表示轴承表层和基层的孔隙率,$i=1,2$.

1.3 数值解法

将含油轴承润滑计算区域离散,首先沿轴承表面周向和轴向 $M-1$ 和 $N-1$ 等分,间距分别记为 $\Delta\theta = 2\pi/(M-1)$,$\Delta Y = 1/(N-1)$;每个节点位置用$(i,j)$二维编号表示,采用有限差分法求解润滑模型.令 $A_{i,j} = H_{i,j}^3 + \psi$,令 $\alpha = ((R/b)\cdot(\Delta\theta/\Delta Y))^2$,离散化式(8)可得油膜压力的迭代格式

$$\begin{aligned} P_{i,j} = {} & [A_{i+1/2,j}P_{i+1,j} + A_{i-1/2,j}P_{i-1,j} + \alpha[A_{i,j+1/2}P_{i,j+1} + A_{i,j-1/2}P_{i,j-1}] - \\ & \Delta\theta(H_{i+1,j} - H_{i,j})]/[(A_{i+1/2,j} + A_{i-1/2,j}) + \alpha(A_{i,j+1/2} + A_{i,j-1/2})]. \end{aligned} \tag{10}$$

迭代求出各节点压力后,代入如下方程,可得无量纲承载力:

$$G = \int_0^{2\pi}\int_{-/2}^{1/2} P\mathrm{d}Y\mathrm{d}\theta = \Delta\theta\cdot\Delta Y\cdot\sum_{i=1}^{M-1}\sum_{j=1}^{N} P_{i,j}, \tag{11}$$

离散后的无量纲摩擦力

$$F_{z=h} = \Delta\theta\cdot\Delta Y\cdot\sum_{i=1}^{M-1}\sum_{j=1}^{N}\left[3H_{i,j}\frac{P_{i+1,j}-P_{i,j}}{\Delta\theta} + \frac{1}{H_{i,j}}\right]. \tag{12}$$

通过计算摩擦力与承载力的比值可得摩擦因数 f 的大小.

2 结果分析

如前所述,复层含油轴承之所以能改善油膜承载能力和含油自润滑效果难以共存的矛盾,主要是通过调整不同层间渗透率来实现的,而轴承渗透率与轴承结构和其内部微观孔隙密切相关.因此下文将着重分析复层轴承结构及内部孔隙参数对润滑性能的影响.参考文献[20],本文选取计算参数如下:

$D_c = 1\times10^{-4} \sim 5\times10^{-4}$ m; $\varphi_i = 8\%, 10\%$; $T_i = 0.002, 0.004$ m;

$C = 5\times10^{-5}$ m; $e = 5\times10^{-6}$ m; $b = 0.024$ m.

2.1 复层结构的影响

如图 3 所示为孔隙结构一定时($D_c = 1\times10^{-4}$ m, $\varphi_2 = 10\%$, $\varphi_1 = 8\%$),复层轴承润滑性能随各层厚度及偏心率的变化.由图可见,提高轴承两层高度对润滑性能都有不利影响,即油膜的承载能力随着两层高度增大而降低,摩擦因数随着两层高度增大而增大.这主要是由于在流体润滑工况下,压装在刚性轴承座中的轴承底面及两端面具有良好密封,在油膜压力驱动下接触面间的油液向轴承多孔基体中渗漏,当轴承较薄时,多孔基体吸收渗漏量的空间较小,因此油膜渗漏量小,得以保证较高的油膜厚度和较好的润滑性能,随着轴承各层高度的提高,多孔基体中吸收渗漏量的空间增大,油膜渗漏量增大,油膜润滑性能变差.此外在图 3 还能看出,复层含油轴承润滑性能随着偏心率增大而逐渐变好,这与偏心率对普通单层含油轴承[20]和一般径向轴承(非多孔结构)的润滑性的影响规律相同[21].

2.2 孔隙结构的影响

如图 4 所示为复层轴承各层高度一定时($T_1 = T_2 = 0.002$ m),轴承润滑性能随各层孔隙率及孔隙直径的变化.由图可见:轴承润滑性随孔隙直径增大而变差,随轴承总孔隙率的减小而

变好;当轴承总孔隙率一定时,表层孔隙率较低时有利于提高轴承润滑性能.分析可知,当轴承总孔隙率较大时,接触面间的油膜更易向多孔基体中渗漏,从而使油膜润滑性能变差;当总孔隙率一定,表层孔隙率较小时,较致密表层能阻止油液向多孔基体中渗漏,使油液保持在接触面之间,提高摩擦副润滑性能.

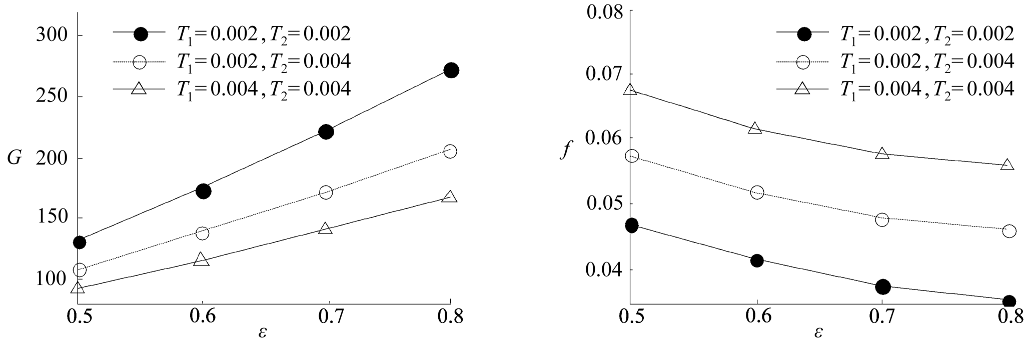

图 3 复层轴承结构对油膜无量纲承载力 G 和摩擦因数 f 的影响

Fig. 3 Effect of the multi-layer bearing structure on dimensionless bearing capacity G and friction coefficient f

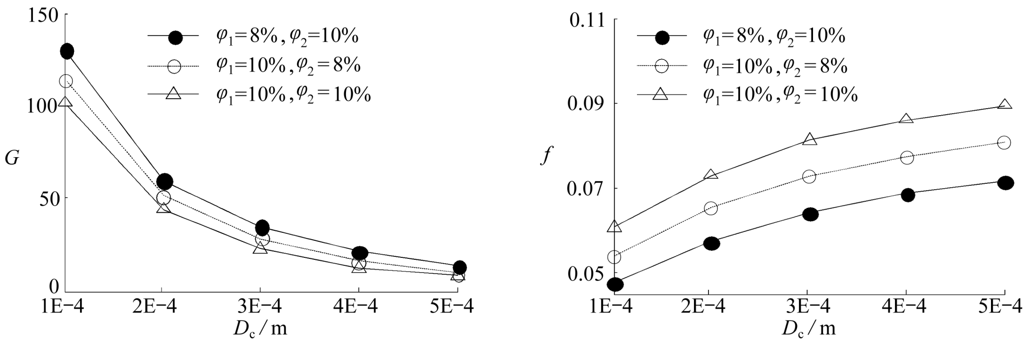

图 4 孔隙结构对油膜无量纲承载力 G 和摩擦因数 f 的影响

Fig. 4 Effects of the pore structure on dimensionless bearing capacity G and friction coefficient f

3 结　　论

本文以复层含油轴承为对象,通过在 Darcy 定律中引入 Kozeny-Carman 孔隙模型,建立复层含油轴承流体润滑模型,分析复层结构和孔隙结构参数对含油轴承流体润滑性能的影响.得出结论如下:

1) 复层含油轴承润滑性能随各层高度增大而变差,随轴承总孔隙率减小而变好.

2) 复层含油轴承总孔隙率一定时,较致密表层有利于提高轴承润滑性能.因此流体润滑工况下设计复层含油轴承时,在保证一定含油量的前提下(总孔隙率不变),应尽可能减小表层孔隙率.

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Effects of Pore Structure on Hydrodynamic Lubrication of Multi-Layer Oil Bearings

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Abstract: The hydrodynamic lubrication model for multi-layer oil bearings was established based on the Darcy's law. The pores' effects depicted with the Kozeny-Carman equation were considered in the analysis. The effects of the bearing structure and the pore structure on the lubrication properties were simulated and discussed with the finite difference method. Results show that the lubrication properties get worse with the increase of the layer heights and the porosity; a lower-permeability surface will be more beneficial to improve the lubrication performance when the total porosity is fixed. Therefore, in the design of double-layer oil bearings, the surface porosity should be reduced as far as possible as long as the oil content in pores is guaranteed. This work is referential to the analysis of the tribological properties and the structural design of multi-layer oil bearings.

Key words: multi-layer oil bearing; hydrodynamic lubrication; pore; carrying capacity; friction coefficient

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