第十二章 Sleeve bearings 滑动轴承

§12－1 General considerations 概述
§12－2 Materials of sleeve bearings (materials of bearing shells and inserts) 滑动轴承材料（轴瓦及轴承衬材料）
§12－3 Simplified calculations for bearings operating with non-fluid friction 非液体摩擦轴承设计计算
§12－4 Load-carrying mechanics of the hydrodynamic lubrication bearings 液体动压润滑轴承承载机理

§12－1 general considerations 概述

一、 Types of sleeve bearings 分类

I With respect to friction condition: 按摩擦状态分

1） Non-fluid friction bearings 非液体摩擦轴承

2） Fluid friction bearings 液体摩擦轴承

II With respect to load-carrying direction 按承载方向分：

1） Thrust bearings 推力轴承：F_A

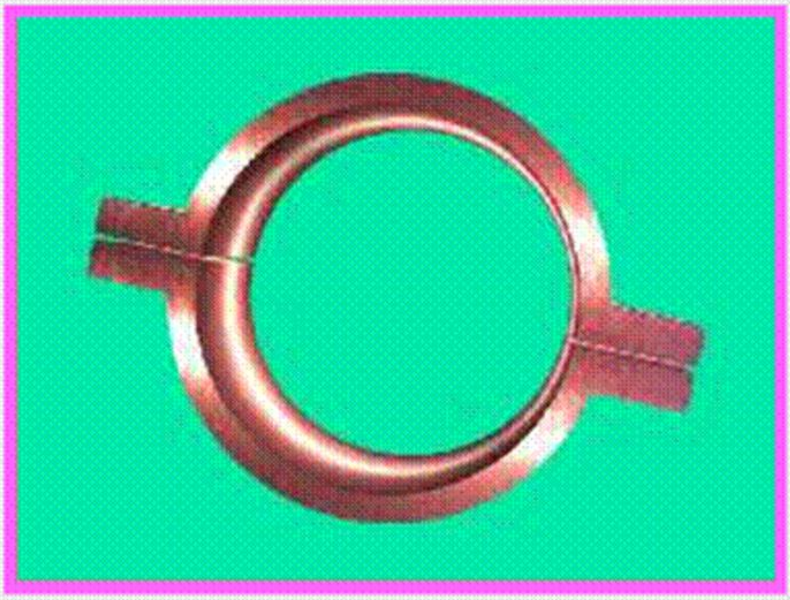
2） Radial bearings 向心轴承：F_r

Radial bearings 向心轴承：F_r

- Solid bearings :used for low speed and light load shaft (the clearance in the bearing can not be adjusted and it is difficult to assembly） 整体式：低速轻载（间隙不可调，安装困难）
- Split bearings: common application 剖分式：一般应用

Solid shell

整体轴瓦



Split shells

剖分式轴瓦

二、 Characteristics and applications

特点和应用

1、Advantages

① Operating steady , reliably and noiselessly

工作稳定、可靠、无噪音

② Under fluid friction, friction and wear are small and oil film has a marked damping effect

液体摩擦时，摩擦、磨损小，油膜吸振

2、 Shortcomings

High starting torque 起动阻力大

3、 Applications 应用

①Shafts running at exceptionally high speeds or requiring high-precision guidance or carrying heavy load （fluid friction bearings）

高速、重载、精密（液体摩擦滑动轴承）

②Shafts running at low speed noncritical mechanisms （ non-fluid friction bearings ）

低速、轻载、精度要求低时（非液体摩擦滑动轴承）

③Special occasion: for example, split shell is required or operating in aggressive media.

特殊场合:如必须剖分或者宇航、探海等

三、 Contents of the chapter

本章内容

① Selection of the bearing materials 轴承材料选择

② Design of non-fluid friction bearings 非液体摩擦滑动轴承设计计算

③ Load-carrying mechanics of the hydrodynamic lubrication bearings 液体摩擦轴承承载原理

§12－2 Materials of sleeve bearings

（ materials of bearing shells and inserts ） 滑动轴承材料（轴瓦及轴承衬材料）

Bearing shell:

Element contacting directly with journal

轴瓦：

轴承直接与轴颈相接触的零件

Bearing insert: thin layer which is adhered on bearing shell by casting or pressuring in.

轴承衬：有的轴瓦是在轴瓦整体上用浇铸或压合的方法粘附上一层另一种材料，这一薄层材料称为轴承衬。

一、 Requirements for bearing materials:

① Small friction coefficient, good heat conductivity and low linear expansion coefficient

(efficiency high, stable dimension and small temperature increase operating)

摩擦系数μ小，导热性好，热膨胀系数小（效率高、尺寸稳定、温升小）

② High wear resistance, good corrosion resistance and anti-seizing

耐磨、耐蚀和抗胶合性好

③Enough mechanical strength

(anti-impact, anti-press and anti-fatigue) (prevent from press and fatigue failures)

足够机械强度（冲击、抗压、疲劳）

（避免压溃和疲劳破坏）

④ Capacity for being run in readily and low modules of elasticity 一定顺应性和嵌藏性

顺应性：适应轴偏斜的能力→ 减小偏载

嵌藏性：容纳异物的能力→

避免轴颈伤和磨损

（弹性模量小→塑性增加→顺应性和嵌藏性好）

⑤ Good processing properties, such as castability, manufacturability, etc

良好的工艺性（容易铸造和加工）

二、Major bearing materials 常用材料

1、轴承合金（白合金、巴氏合金） babbitts

based on tin 锡锑轴承合金：ZChSnSb11－6

based on lead 铅锑轴承合金：ZChPbSb16－16－2

Advantages： Small friction coefficient and high anti-seizing Used for high speed and heavy load

摩擦系数小，抗胶合性好→用于高速、重载

Shortcomings： High cost and low strength Used for bearing inserts

价格高、强度低→用作轴承衬

2、 Bronzes 轴承青铜

Tin bronzes 锡磷青铜：ZCuSn10P1

Leaded bronzes 锡锌铅青铜：ZCuSn5Pb5Zn5

Aluminum-iron bronzes 铝铁青铜：ZCuAl9Fe4Ni4Mn2

Advantages: High strength, high wear resistance and good thermal conductivity. Used for bearing shell and inserts

强度高、耐磨性好、导热性好，即可用于轴瓦，又可以作轴承衬

3、 Brass: used for low speed and medium load

黄铜：低速中载

4、Cast iron: used for low speed and light load

铸铁：低速轻载

5、Bearings filled with oil 含油轴承:

Sintered metal powders 粉末冶金材料
- Iron-graphite Fe——石墨
- Bronze-graphite Cu——石墨

Porous structure、 Filled with oil Self-lubrication
多孔、含油→自润滑

Bad roughness 韧性差

Used for steady load and places which is difficult or impossible to lubricate 平稳无冲击、难于用油润滑处

6、 Plastics 轴承塑料

- Advantages: small friction coefficient, self lubrication and low modules of elasticity
  摩擦系数小，自润滑性和嵌藏性好
- Shortcomings: bad heat-resistance and low thermal conductivity. When they absorb moisture, sizes expansion takes place. Otherwise, under common load, dramatic deformation will take place in plastics bearings.
  耐热性、导热性差，吸水膨胀，易变形

常用作轴承衬

三、Structures of bearing shell 轴瓦结构

1、 Bearing shells 轴瓦

Single metal shell 单金属轴瓦

Double metals shell 双金属轴瓦

Three metals shell 三金属轴瓦

Bearing inserts are used 加轴承衬

2、Oil hole, oil groove and oil chamber 油孔、油沟、油室

- ★ Oil hole 油孔→ For supplying oil 供油
- ★ Oil groove 油沟
  - Conveying oil 输送油
  - Distributing oil 分布油
- ★ Oil chamber 油室
  - Equispacing oil 均布油
  - Reserving oil 贮油
  - Steadily supplying oil 稳定供油

※Oil grooves are cut in zones where there are not force or there are smaller force in order to supply oil easily and prevent from reduce load-carrying ability at same time.

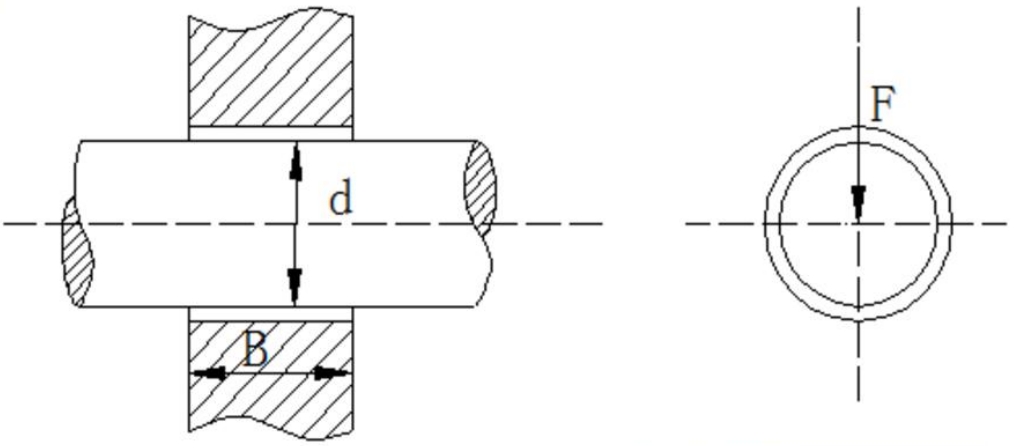
油沟开于不受力或者受力较小区，以利供油，同时避免降低承载能力

§12—3 Simplified calculations for bearings operating with non-fluid friction 非液体摩擦轴承设计计算

一、Types of failures and design criteria 失效形式和设计准则

Wear → boundary film do not lose 磨损→边界膜不破裂

This has many affecting factors and analyses are considerably complicated → simplified calculation 因素多,复杂→条件性计算



(1) Radial bearings 径向轴承:

① Average pressure is limited (prevent from oil to be force out) 限制平均压强（防油挤出→过度磨损）

$$p=\frac{F}{dB}\le[p] \quad \text{MPa}$$ （低速或间歇转动轴承，只校核p≤[p]）

Only check p<[p] for low speed or pause rotating bearings

② pv is limited: (prevent from excessive temperature increasing → oil film lose → excessive wear)

限制pv值 （防温升△t过大→油膜破裂→过度磨损）

$$pv=\frac{F}{dB}\frac{T_1 nd}{60\times1000}=\frac{Fn}{2000B}\le[pv] \quad \text{MPa}\bullet\text{m/s}$$

③v≤[v] m/s （ **Prevent from excessive speed → quick wear**）

（p较小时）（防v过大→加速磨损）

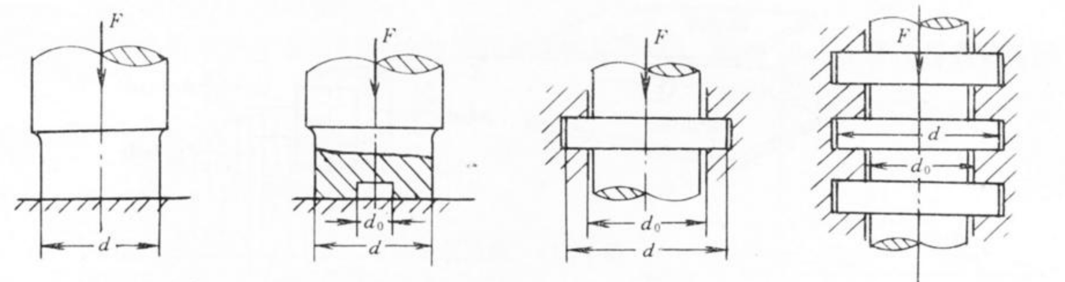
$$v = \frac{\pi dn}{60 \times 1000} \le [v]$$

★[p]、[pv]、[v]——P161表9－1

(2) Thrust bearings 推力轴承

$$\begin{cases} p \le [p] & \text{MPa} \\ pv \le [pv] & \text{MPa}\bullet\text{m/s} \end{cases} \quad \begin{cases} p = \dfrac{F}{\dfrac{\pi}{4}\left(d^2 - d_0^2\right)Z} \le [p] & \text{MP} \\ pv = \dfrac{Fn}{3000\left(d - d_0\right)Z} \le [pv] & \text{MPa}\bullet\text{m/s} \end{cases}$$

Z：油环数



a）实心端面推力轴颈　b）空心端面推力轴颈　c）环状轴颈　d）多环轴颈

★（液体轴承起动、停车：混和摩擦状态）

§12—4 Load-carrying mechanics of the hydrodynamic lubrication bearings 液体动压润滑轴承承载机理

一、Basic equation of hydrodynamic lubrication-Reynolds equation （1886年）

液体动压润滑基本方程—— 雷诺方程

$$\frac{\partial p}{\partial x} = 6\eta V \frac{h - h_0}{h^3}$$

1、 Assumptions 假设

① Compared with forces of viscous shear, inertia and gravity of the fluid are very small and are neglected

与粘滞阻力相比， 流体惯性力、重力很小，忽略

② Flow of lubricant belong to laminar flow

润滑油为层流流动

Forces of viscous shear developed in the layers can be determined by Newton’s law of viscous flow.

τ 服从牛顿剪切定律： $\tau = -\eta \dfrac{\partial v}{\partial y}$

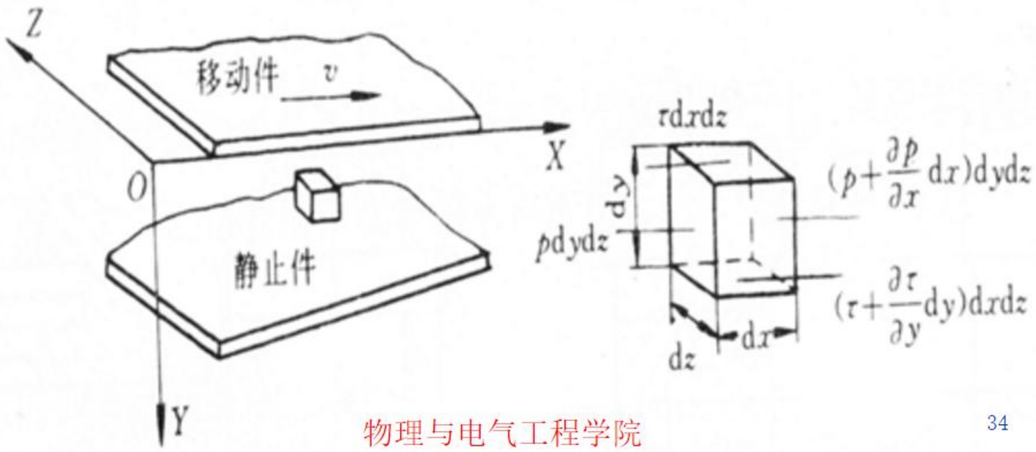
③ Sliding do not exist between lubricant and planes 润滑剂与板面无滑动

④ The dimensions in the direction perpendicular to the drawing are assumed to be infinitely large 轴承无限宽（Z向无流动）

⑤Viscosity and density of the oil do not depend on pressure

流体粘度、密度与压力无关

2、Introduction 推导



By means of:

① Equilibrium equation of the oil 流体平衡方程

② Newton’s law of viscous flow 牛顿内摩擦定律

③Principle of continuity offlow 流体流动连续性方程

$$\tau = -\eta \frac{\partial v}{\partial y}$$

$$\frac{\partial q_x}{\partial x} = 0$$

➡ Reynolds equation 雷诺方程

①Equilibrium equation of the oil

流体平衡方程

$$\sum F_x = 0$$

$$p\,dy\,dz + \tau\,dx\,dz - \left(p + \frac{\partial p}{\partial x}dx\right)dy\,dz - \left(\tau + \frac{\partial \tau}{\partial y}dy\right)dx\,dz = 0$$

$$\frac{\partial p}{\partial x} = -\frac{\partial \tau}{\partial y} \text{..................①} \quad （假设1）$$

② Newton’s law of viscous flow

牛顿内摩擦定律（假设2）

$$\tau = -\eta \frac{\partial v}{\partial y} \Rightarrow \frac{\partial p}{\partial x} = -\frac{\partial^2 v}{\partial y^2} \quad \cdots\cdots\cdots\cdots ②$$

③By integrating twice and substituting the boundary conditions 积分

$$\frac{\partial^2 v}{\partial y^2} = \frac{1}{\eta}\frac{\partial p}{\partial x}$$

$$v = \frac{1}{2\eta}\frac{\partial p}{\partial x} y^2 + C_1 y + C_2$$

由假设3: $\begin{cases} y = 0、v = V \\ y = h、v = 0 \end{cases}$

$$C_1 = -\frac{1}{2\eta}\frac{\partial p}{\partial x} - \frac{V}{h}$$

$$C_2 = V$$

$$v = \frac{V(h-y)}{h} - \frac{y(h-y)}{2\eta} \cdot \frac{\partial p}{\partial x}$$

④

The volume of oil flowing per second through a clearance of height h and unit width is (side leakage do not exist)

单位时间内流经任一剖面单位宽度面积上的流量（假设4无侧漏）：

$$q_x=\int_0^h v\,dy=\int_0^h\left[\frac{V(h-y)}{h}-\frac{y(h-y)}{2\eta}\bullet\frac{\partial p}{\partial x}\right]dy=\frac{Vh}{2}-\frac{h^3}{12\eta}\bullet\frac{\partial p}{\partial x}$$

⑤By means of principle of continuity of flow

用不可压缩流体连续性方程（假设5）

$$\frac{\partial q_x}{\partial x}=0\rightarrow q_x=\text{const}$$

$$p=p_{max}:\frac{\partial p}{\partial x}=0\Rightarrow q_0=\frac{Vh_0}{2}\ (h_0\text{—}p_{max}\text{处的}h)$$

$$\frac{\partial p}{\partial x}=6\eta V\frac{h-h_0}{h^3}\qquad\text{（雷诺方程）}$$

三、Load-carrying mechanics of the oil wedge 油楔承载机理

建立高压力$p > p_0 \rightarrow \frac{\partial p}{\partial x} \neq 0$

From Reynolds equation, we know

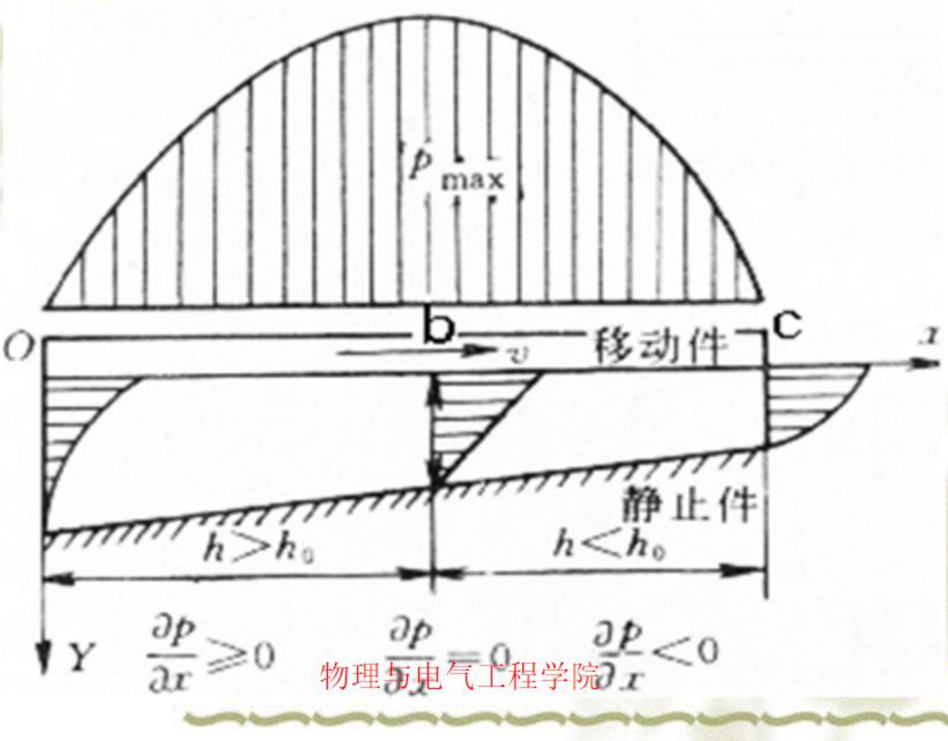
that the conditions of forming hydrodynamic lubricatio n are:

由$\frac{\partial p}{\partial x} = 6\eta V \frac{h - h_0}{h^3}$可知形成流体动压润滑的条件:

① Relative motion exists between two planes.
两板有一定相对运动速度（$V \neq 0$）

②Oil has certain viscosity润滑油有一定粘度（$\eta \neq 0$）

③ Convergence wedge is formed between relative motion surfaces.
相对运动表面成楔形，而且是收敛楔形。（$h \neq h_0$）



Convergence wedge

(1)Convergence wedge 收敛楔形

$$\begin{cases} b点: \frac{\partial p}{\partial x}=0(h=h_0) 、p_{max} \\ ob: \frac{\partial p}{\partial x}>0(h>h_0) 、p随x增大而增大 \\ bc: \frac{\partial p}{\partial x}<0(h<h_0) 、p随x增大而减小 \end{cases} \Rightarrow 产生动压力、承受外载$$

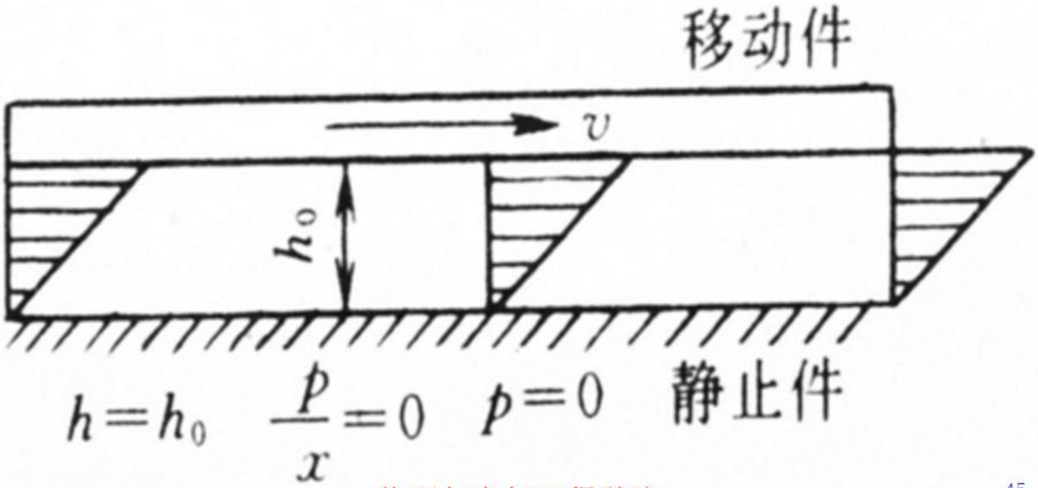
p increase along with x

Along with x increases, p drops

Hydrodynamic pressure is developed and it can support load

(2)平行油楔

$$\frac{\partial p}{\partial x}=0(h \equiv h_0) \rightarrow p \equiv p_0(大气压)$$



(3)Diffusion wedge： 扩散楔形

Negative pressure is formed and two relative surfaces pull each other

产生负压→两面相吸