安徽工程大学
Anhui Polytechnic University
机械工程学院
SCHOOL OF MECHANICAL ENGINEERING

发动机接触副磨损特性及高效润滑

Wear Characteristics and High Efficiency Lubrication of Engine Contact Pair

第十一届内燃机可靠性技术国际研讨会
The 11th International Conference of ICE Reliability Technology

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发动机接触副形式多样
Various types of engine contact pairs

- 主轴承 (main bearing)
- 连杆轴承 (Connecting rod bearing)

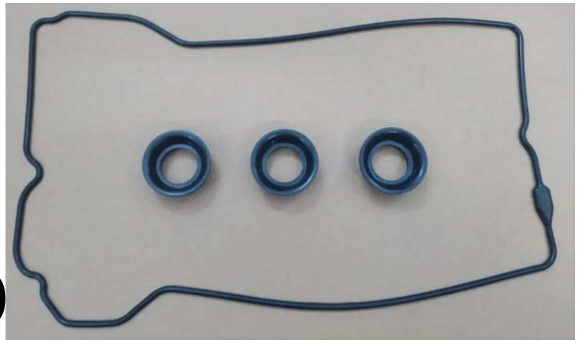
- 活塞环(Piston ring)
- 活塞(Piston)
- 缸套(Cylinder liner)

- ◆ 系统组成部件较多 (Many components)
- ◆ 接触形式复杂多样 (Complicated types of contact)

- 气阀座 (Air valve seat)
- 气门导管 (Valve guide pipe)

- 凸轮挺杆 (Cam tappet)
- 凸轮轴承 (Cam bearing)

- 密封环 (Seal ring)



发动机接触副磨损机理复杂

The wear mechanism of engine contact pair is complex

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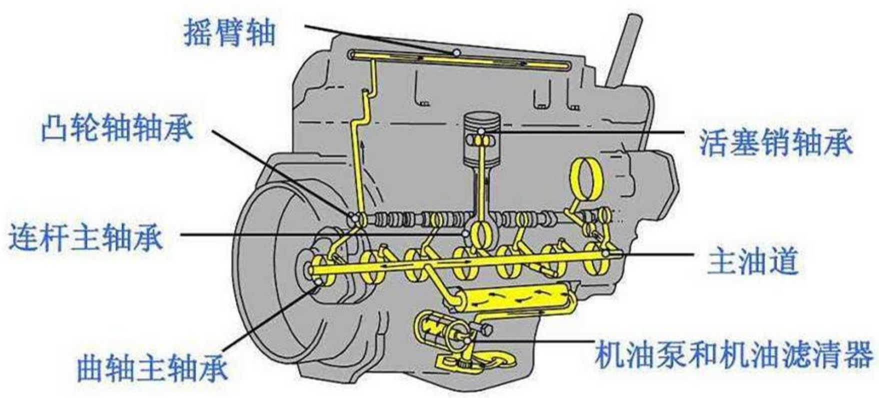
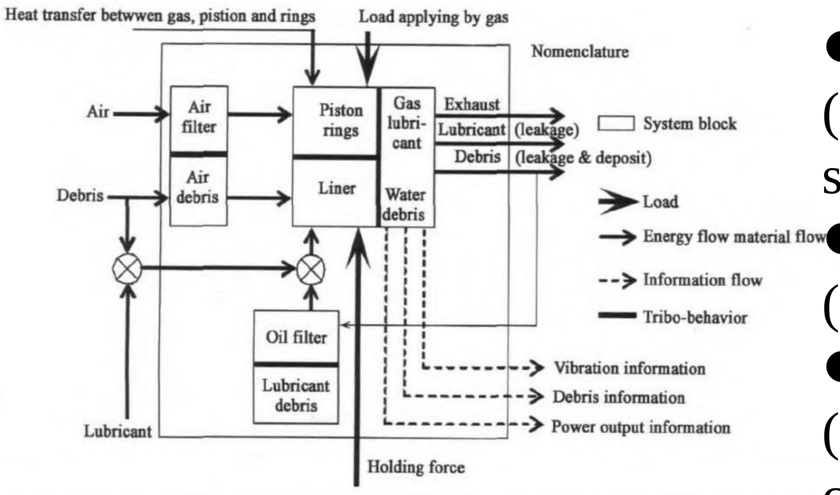




- 相对运动速度周期性变化
(Periodic change of relative motion speed)
- 载荷周期性变化
(Periodic change of load)
- 接触区域温度周期性变化
(Periodic change of temperature in contact area)

凸轮轴磨损
(Wear of camshaft)

轴承磨损
(Wear of bearing)

密封磨损
(Wear of seal)

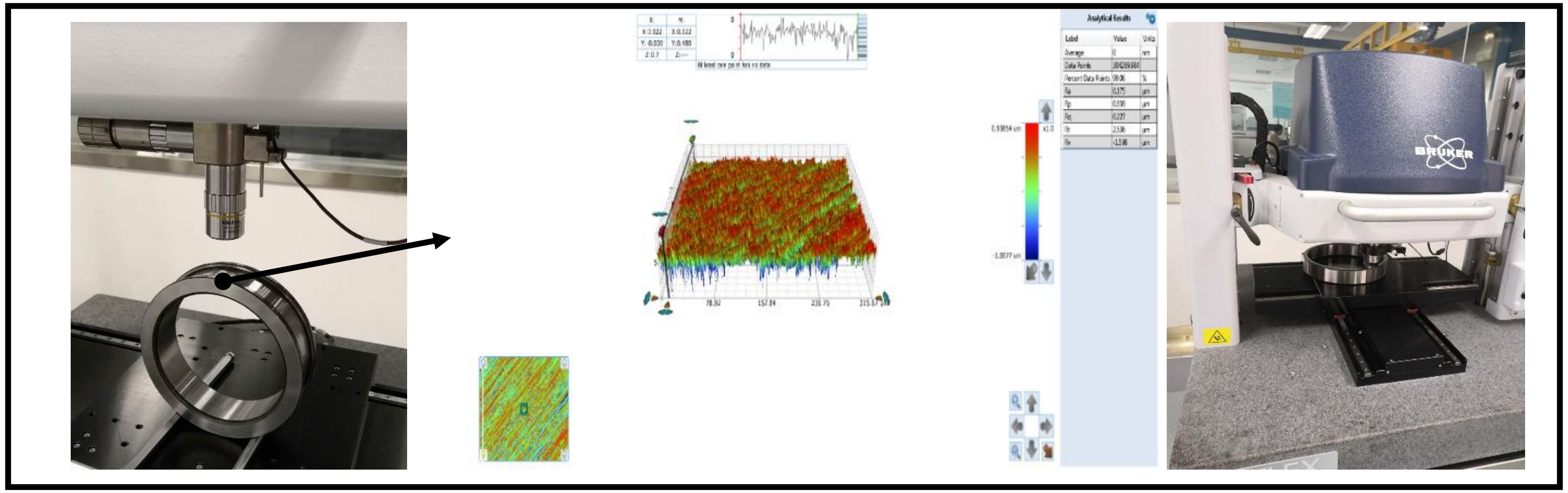
磨损机理复杂，影响发动机使用寿命、可靠性和机械效率。
(The wear mechanism is compliacted, which affects the service life, reliability and mechanical efficiency of the engine.)

发动机接触副摩擦学设计须考虑高效润滑
(High efficiency lubrication must be considered in the tribological design of engine contact pairs)

汇报提纲

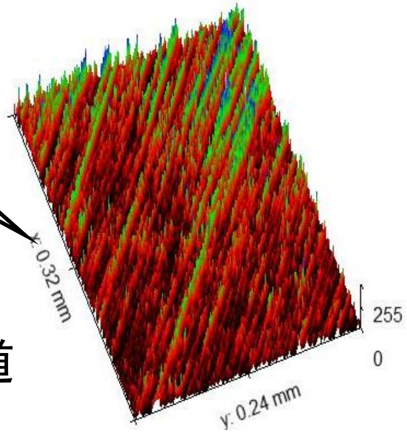
表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

◆表面纹理分布实测(Measured surface texture distribution)



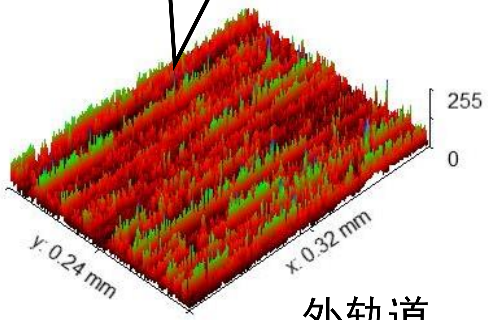
Gwyddion显示处理

与滚动体运动方向成45°



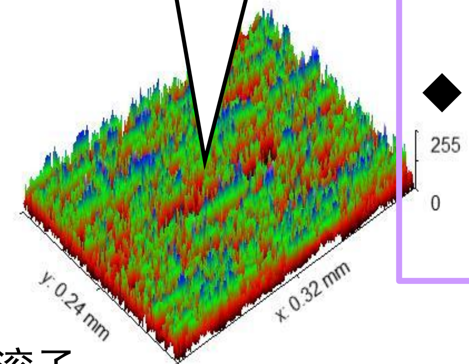
内轨道

与滚动体运动方向平行



外轨道

与滚动体运动方向平行



滚子

表面形貌的特点
(Characteristics of surface topography)

- ◆ 表面形貌纹理走向(Trend of surface topography and texture)
- ◆ 表面粗糙度比配(Match of surface roughness)

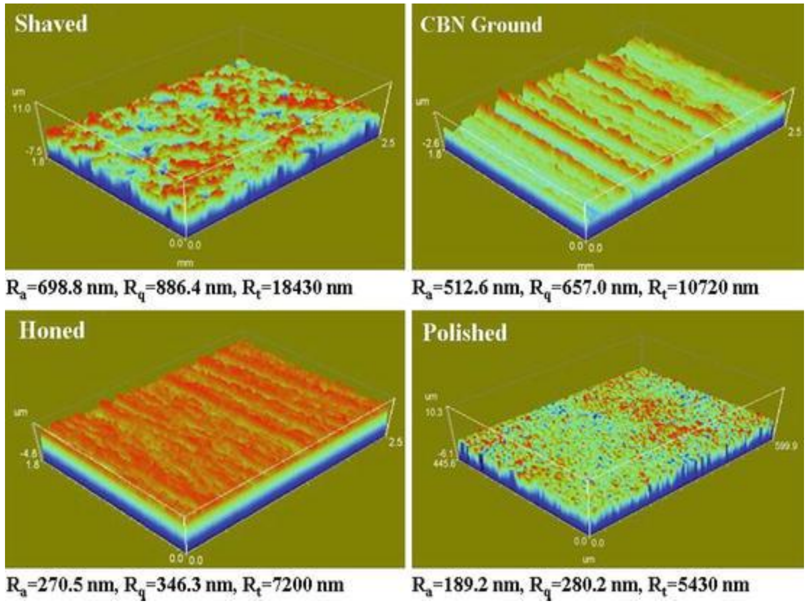
存在的问题
(Existing problems)

- ◆ 表面变化梯度太大不容易进行求解(The gradient of surface change is too large to solve)
- ◆ 实测的表面很难按照规律进行改变(The measured surface is difficult to change according to the law)

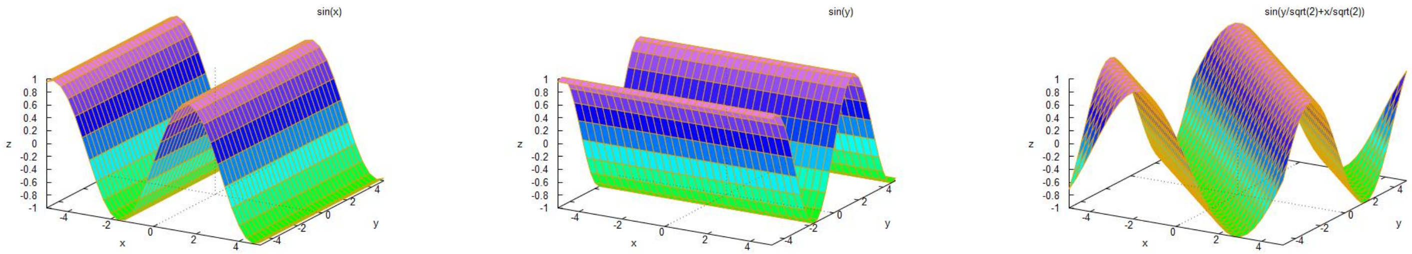
接触副表面形貌表征
(Surface topography characterization of contact pair)

表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

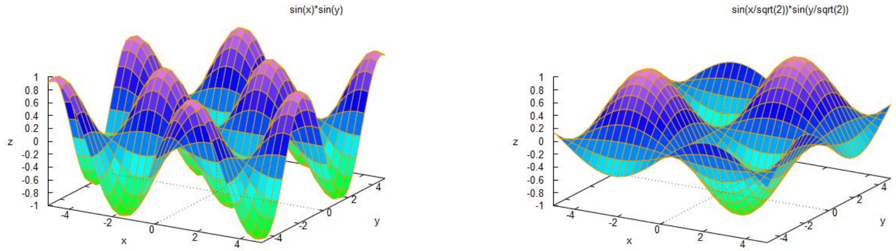
◆表面形貌简单表征(Simple characterization of surface topography)



不同加工方式的表面形貌



余弦表面

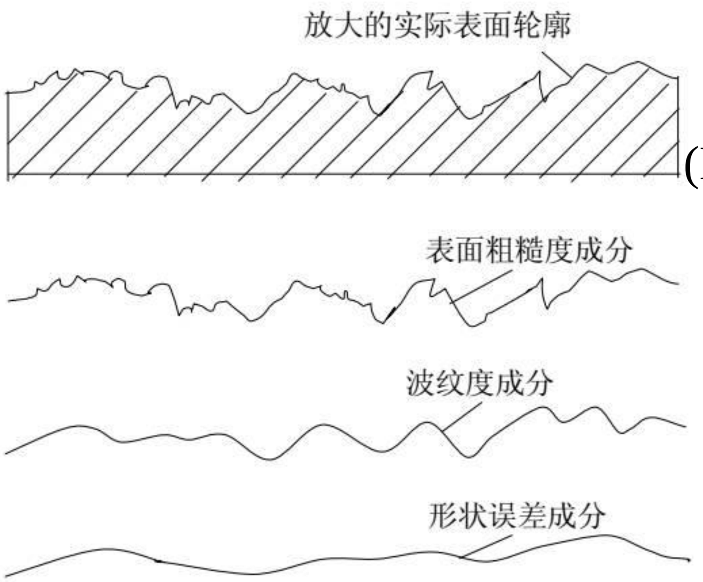


双余弦表面

便于观察，计算简单，与实际稍有差距(Easy to observe, simple to calculate, slightly different from the actual situation)

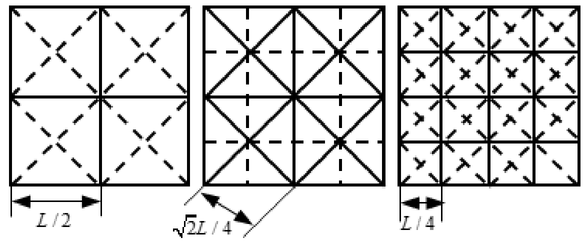
表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

◆表面形貌多尺度表征 (Multi-scale characterization of surface morphology)



多尺度表面形貌 (Multi-scale surface topography)

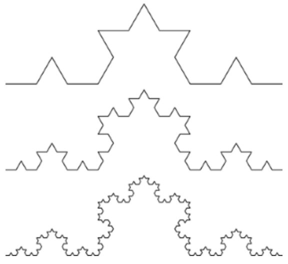
随机中点位移法



缺少粗糙度轮廓形貌参数(Lack of roughness profile parameters)

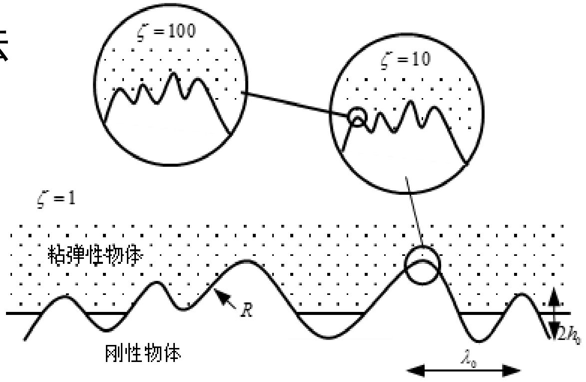
Weierstrass-Mandelbrot公式法

$$Z(\vec{x})=\sqrt{\frac{\log(\gamma)}{M}}\sum_{m=1}^{M}A_m\times\left\{\sum_{k=-\infty}^{\infty}\gamma^{(1-k)H}\left[\cos(\phi_{m,k})-\cos\left(\frac{2\pi}{\lambda}\gamma^{(1-k)}(\hat{n}_m\cdot\vec{x})+\phi_{m,k}\right)\right]\right\}$$



不能很好的描述真实的表面(Can't describe the real surface well)

功率普密度法



更符合真实表面 (More in line with the real surface)

功率谱密度法更好描述表面形貌(PSD method can better describe surface topography)

表面形貌对接触副磨损的影响

Effect of surface topography on wear of contact pair

◆功率谱密度法描述表面形貌 (Describing surface topography with PSD method)

表面粗糙峰的高度的均方根 $\sigma = \langle h^2 \rangle^{1/2}$

$$\langle h^2 \rangle = \int d^2 q C(q) = 2\pi \int_0^\infty dq q C(q)$$

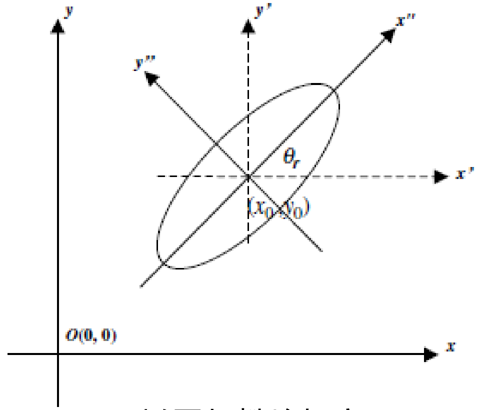
$$C(\vec{q}) = \frac{1}{(2\pi)^2} \int d^2 x \langle h(x) h(0) \rangle e^{-i\vec{q}\cdot\vec{x}}$$

$$\langle h(\vec{x}) h(0) \rangle = \int d^2 q C(q) e^{i\vec{q}\cdot\vec{x}}$$

$$z = h(\vec{x})$$

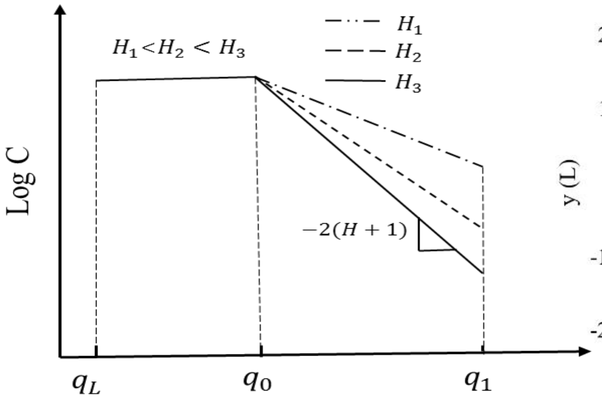
$$C(q) \propto \left(\frac{q_0}{q}\right)^{2(H+1)}$$

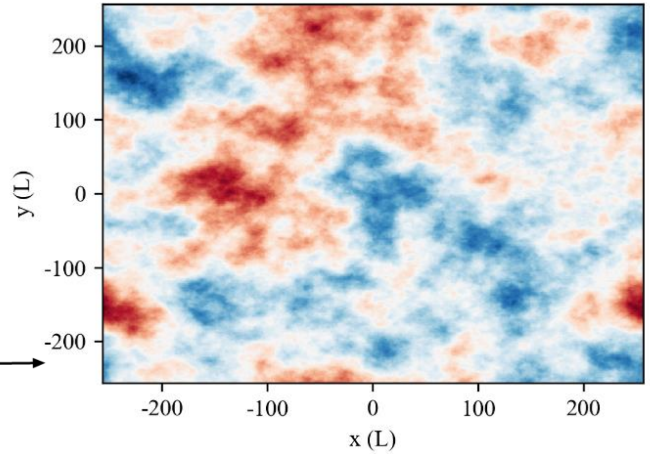
分形维度 $D_f = 3 - H$



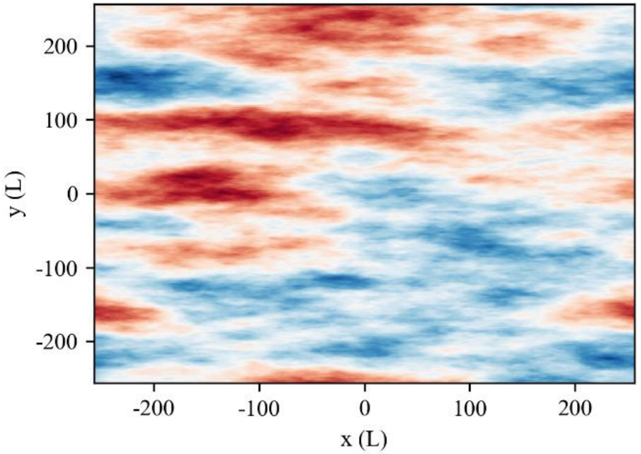
椭圆粗糙峰极座标

$$C(q) = \begin{cases} C_0\left(\frac{\cos^2(\theta)}{a^2} + \frac{\sin^2(\theta)}{b^2}\right)^{-(H+1)}, & \text{if } q_L \le q \le q_0; \\ C_0\left(\frac{q}{q_0}\right)^{-2(H+1)}\left(\frac{\cos^2(\theta)}{a^2} + \frac{\sin^2(\theta)}{b^2}\right)^{-(H+1)}, & \text{if } q_0 \le q \le q_1; \end{cases}$$

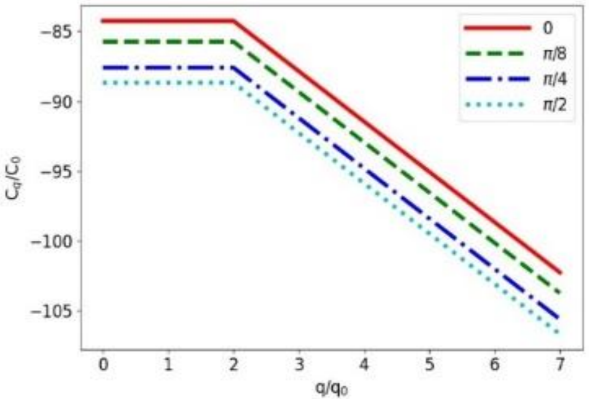




标准(Standard)



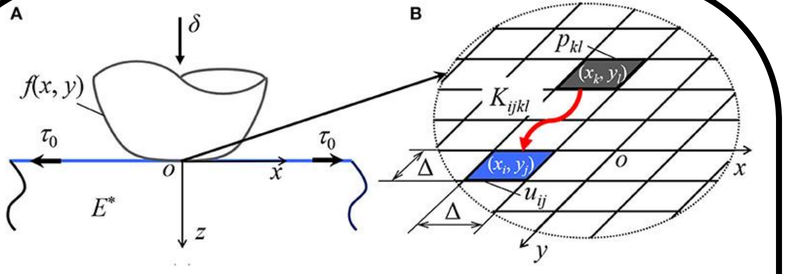
各向异性 (anisotropy)



表面形貌关联参数(Surface topography correlation parameters)

表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

◆完全边界元方法接触分析(Contact analysis with complete boundary element method)

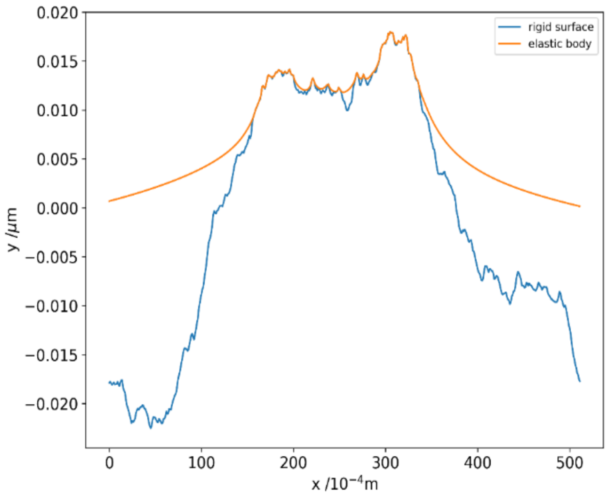


$$u^{ij}=\sum_{i'}\sum_{j'}\begin{pmatrix}K_{xx} & K_{xy} & K_{xz}\\ K_{yx} & K_{yy} & K_{yz}\\ K_{zx} & K_{zy} & K_{zz}\end{pmatrix}^{iji'j'}\sigma^{i'j'}$$

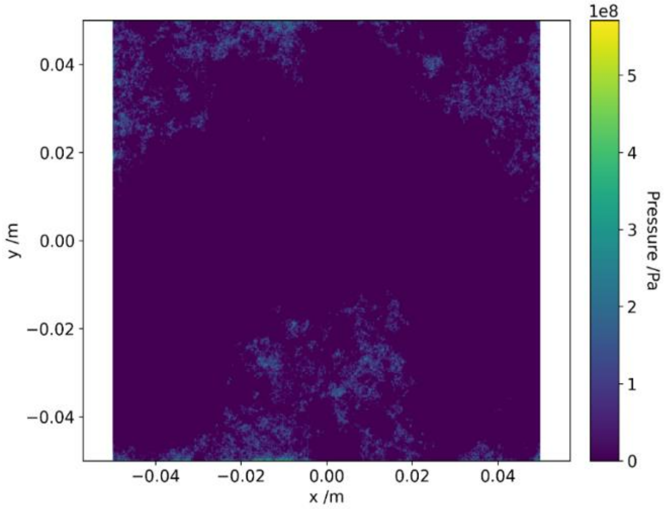
$$\begin{bmatrix}\bar{u}_n\\ \bar{u}_a\\ \bar{u}_s\end{bmatrix}=\begin{bmatrix}\bar{\bar{A}}_{oo} & \bar{\bar{A}}_{ao} & \bar{\bar{A}}_{so}\\ \bar{\bar{A}}_{oa} & \bar{\bar{A}}_{aa} & \bar{\bar{A}}_{sa}\\ \bar{\bar{A}}_{os} & \bar{\bar{A}}_{as} & \bar{\bar{A}}_{ss}\end{bmatrix}\begin{bmatrix}\bar{0}\\ \bar{\sigma}_a\\ \bar{\sigma}_s\end{bmatrix}$$

$$\bar{\sigma}_s=\mu\bar{p}$$

$$u_{sliponly}=\begin{bmatrix}\bar{\bar{A}}_{so}\\ \bar{\bar{A}}_{sa}\\ \bar{\bar{A}}_{ss}\end{bmatrix}\mu\bar{p}=FC_{ab}(\sigma_s)$$

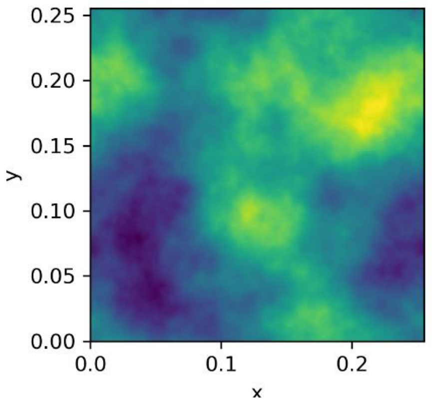


粗糙接触表面的截面

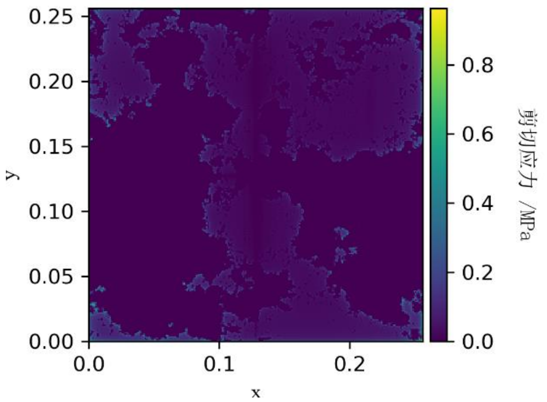


粗糙表面接触应力分布

摩擦引起的剪切应力分析



粗糙表面

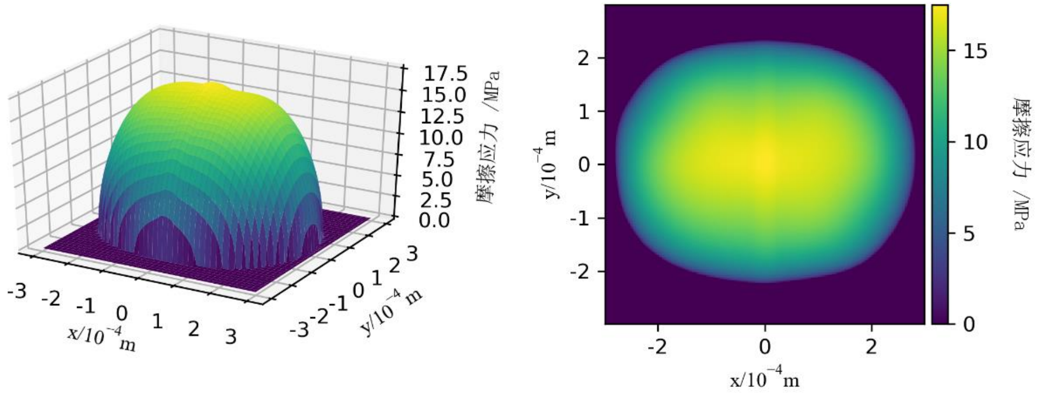


微动剪切应力

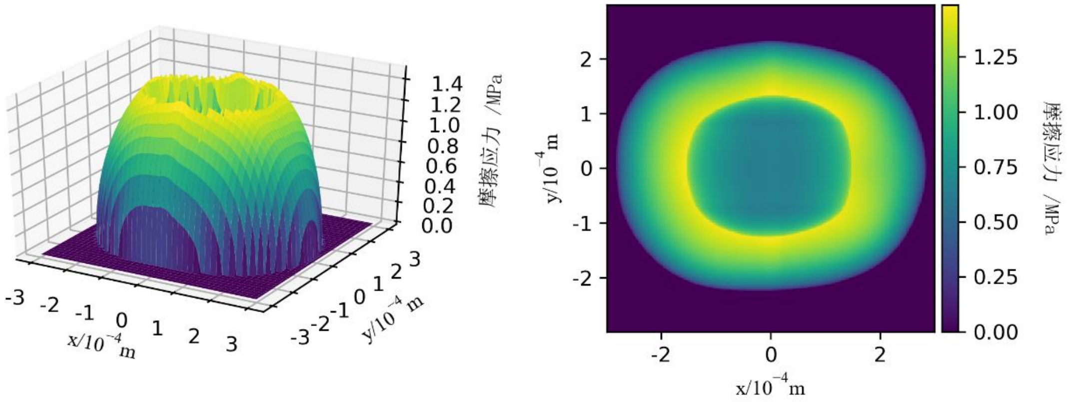
相对符合实际
(Relatively practical)

表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

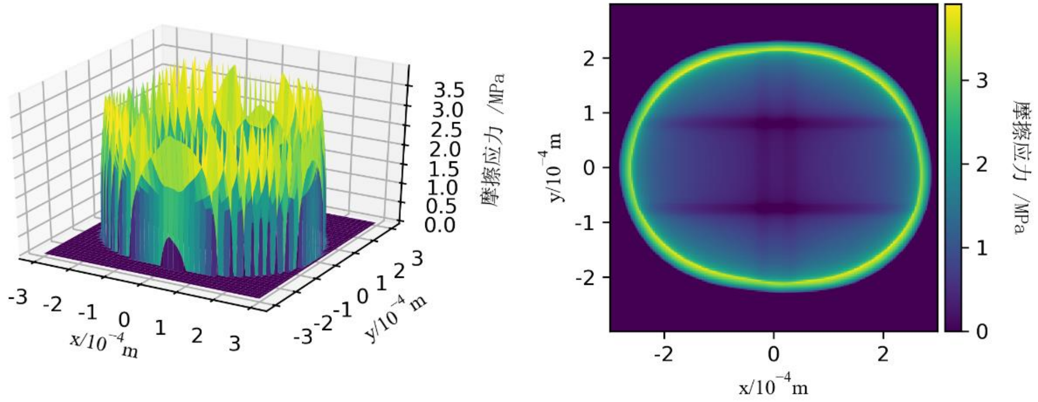
◆摩擦系数对表面剪切应力影响(Effect of friction coefficient on surface shear stress)



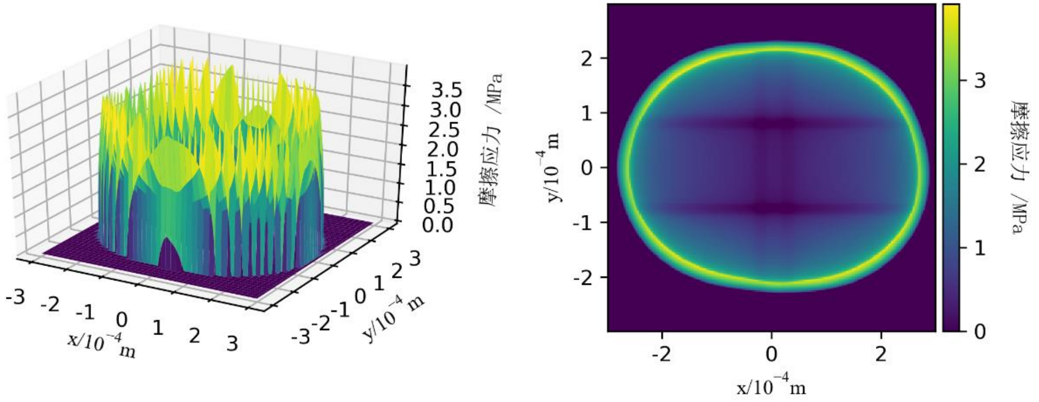
$\mu = 0.1$



$\mu = 0.01$



$\mu = 0.05$

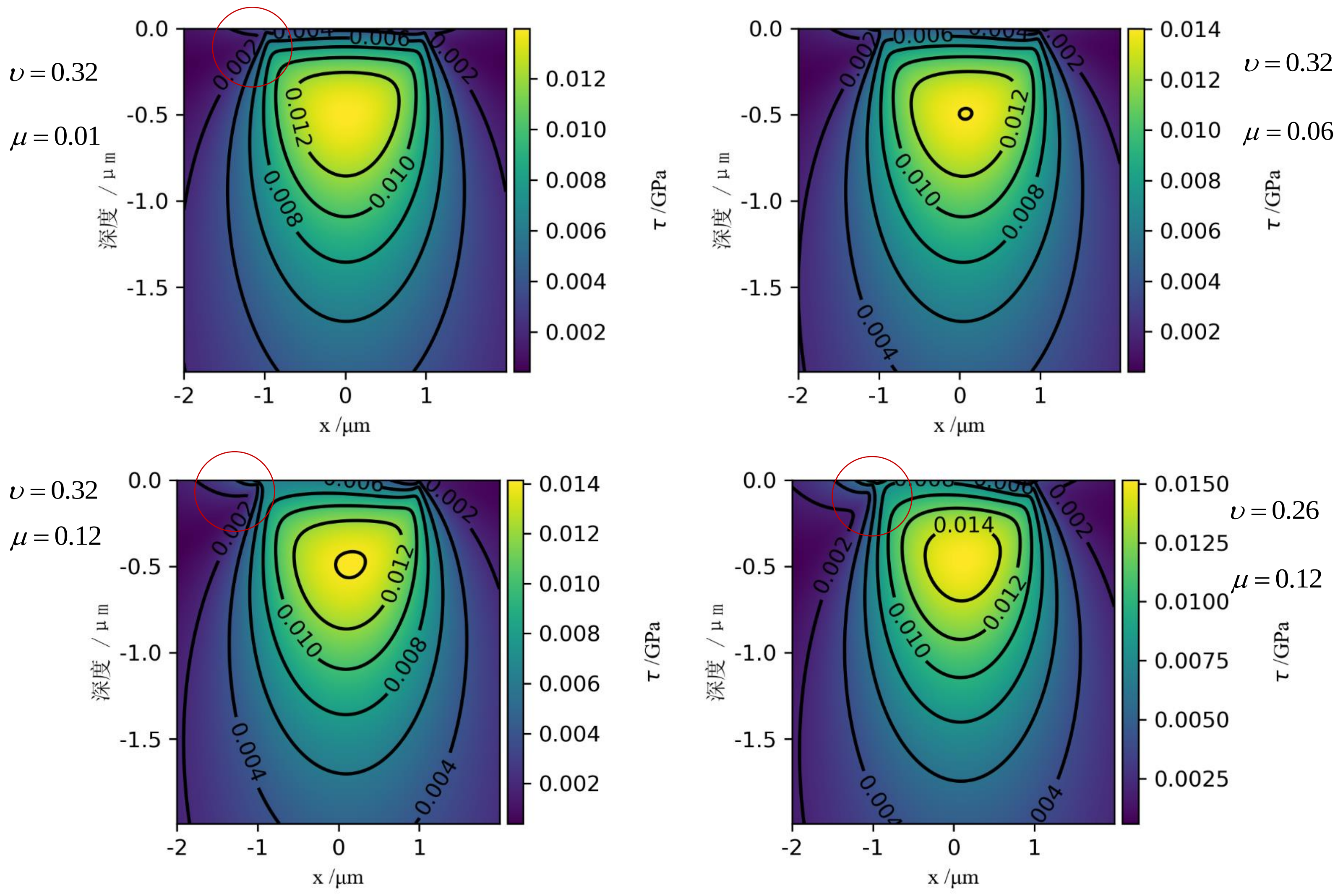


$\mu = 0.1$

纯滑动容易造成表面破坏，微动的剪切圆会随着摩擦系数的增加而变大(Pure sliding is easy to cause surface damage, and the fretting shear circle will increase with the increase of friction coefficient)

表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

◆摩擦系数对次表面剪切应力的影响(Effect of friction coefficient on subsurface shear stress)



摩擦系数增加，次表面剪切应力最大值靠近表面(The friction coefficient increases, and the maximum value of sub-surface shear stress is close to the surface)

表面形貌与摩擦系数之间的内在关系(Internal relationship between surface topography and friction coefficient)

现有弹性接触摩擦分析未考虑表面形貌对摩擦系数影响的模型，引入扩展Persson模型。(The existing elastic contact friction analysis does not consider the model of the influence of surface morphology on friction coefficient, and the extended Persson model is introduced.)

表面形貌对接触副磨损的影响

Effect of surface topography on wear of contact pair

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◆扩展Persson接触模型(Extended Persson contact model)

扩散方程 $\frac{\partial P_d}{\partial \zeta} = f(\zeta)\frac{\partial^2 P_d}{\partial \sigma^2}$

将Persson理论可以应用到不完全接触

$$\int_{q_0}^{q} dqq^3C(q)W(q) = \int_{q_0}^{q} dqq^3C(q)P(q)S(q)$$

修正系数

$$S(q) = W(q)/P(q)$$

$$W(q) = P(q)[\beta + (1-\beta)P^2(q)] \qquad \beta \approx 0.5$$

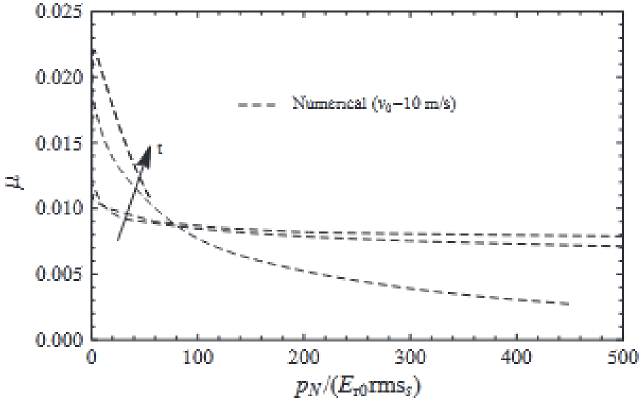
通过平均化粗糙表面斜率进行修正

$$\langle(\nabla h)^2\rangle = 2\pi\int_{q_0}^{q_1} dqq^3C(q)sr(q,v)$$

$$sr(q,v) = \frac{1}{2\pi}\int_0^{2\pi} d\phi\left|\frac{E(qv\cos\phi)}{E_0}\right|$$

$$G(q) = \left(\frac{E^*}{\sigma_0}\right)^2\frac{\pi}{4}\int_{q_0}^{q} dqq^3C(q)S(q)sr(q,v)$$

$$\begin{cases} P(q) \approx [\pi G(q)]^{-\frac{1}{2}} & \text{轻载(Light load)} \\ P(q) \approx (1+[\pi G(q)]^{3/2})^{-1/3} & \text{重载(Heavy load)} \end{cases}$$



表面硬化现象对摩擦系数的影响(Effect of surface hardening on friction coefficient)

在部分接触的基础上加入了**粗糙度变形硬化**的修正(The modification of **roughness deformation hardening** is added on the basis of partial contact)

表面形貌对接触副磨损的影响

Effect of surface topography on wear of contact pair

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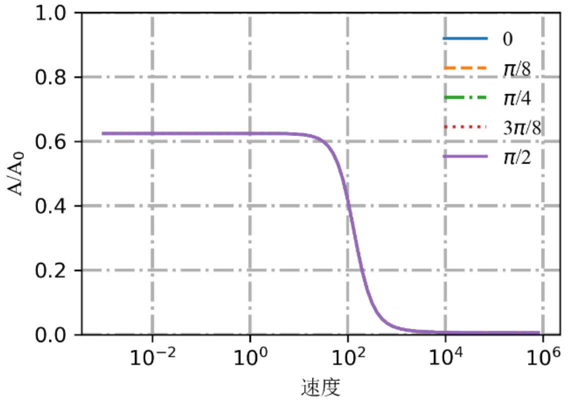
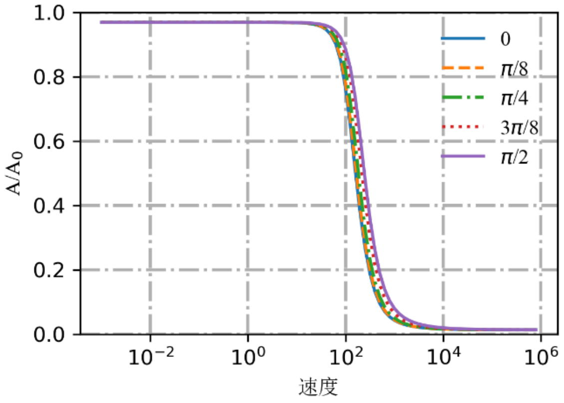
◆表面形貌对接触副接触面积的影响(Influence of surface morphology on contact area of contact pair)

运动方向和粗糙度走向的夹角，0度为垂直

$E_\infty / E_0 = 10$

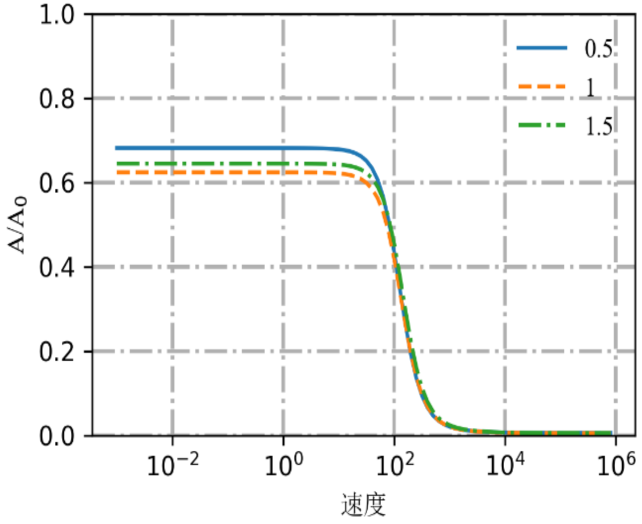
$\lambda = 1$

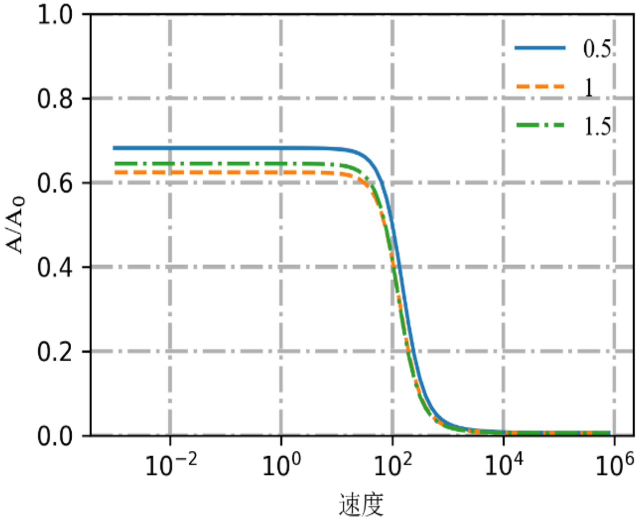




$E_\infty / E_0 = 5$

$\lambda = 0.4$



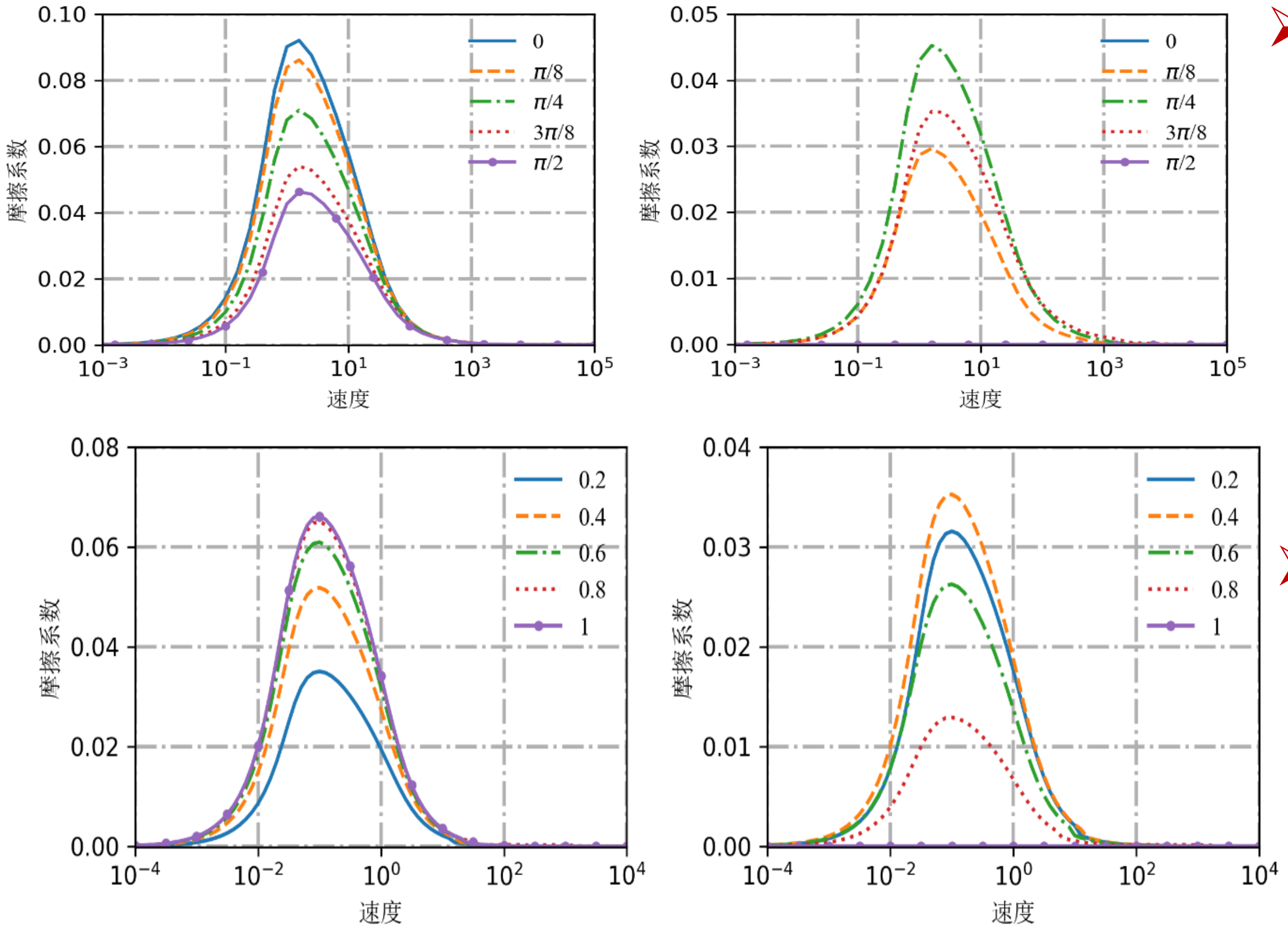


- 初始接触面积同弹性模量和椭圆比有关(The initial contact area is related to the elastic modulus and ellipse ratio)
- 粗糙表面的硬化作用使得初始接触面积减少(The hardening effect of rough surface reduces the initial contact area)
- 表面硬化后会造成接触面积减小(The contact area will be reduced after surface hardening)
- 在某些情况下表面形貌分布一样(n some cases, the surface topography distribution is the same)

表面形貌对接触副磨损的影响

Effect of surface topography on wear of contact pair

◆表面形貌对摩擦系数的影响(Effect of surface morphology on friction coefficient)



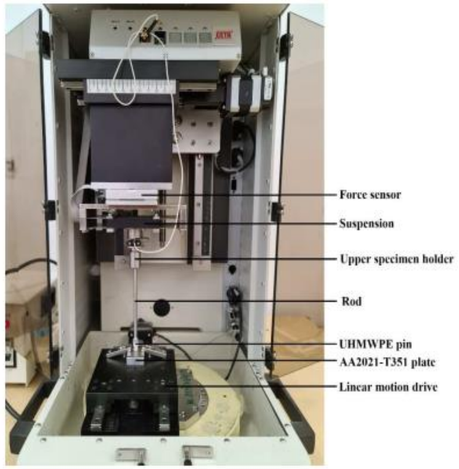
- ➢ 沿着运动方向最小，**45º**时垂直运行方向上摩擦系数最大(The friction coefficient is the smallest along the direction of motion, and the maximum in the vertical direction of motion at **45º**)

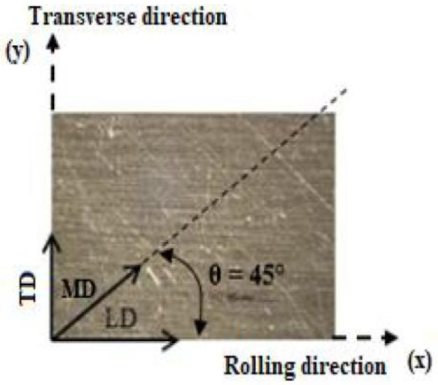
- ➢ 椭圆比大，摩擦系数大(Large elliptical ratio and large friction coefficient)

表面形貌对接触副磨损的影响
Effect of surface topography on wear of contact pair

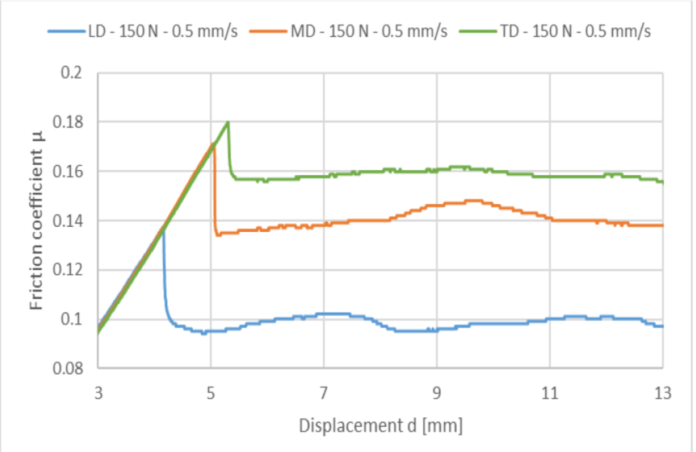
◆实验验证(Experimental verification)

通过使用AAA2021-T351铝粗糙表面与一个UHMWPE球体接触，球体在粗糙表面上滚滑运动。





在各向异性表面滚滑的方向上LD（0°）、MD（45°）、TD（90°），将TD定位为纵向方向。



表面形貌对摩擦系数的影响[1]

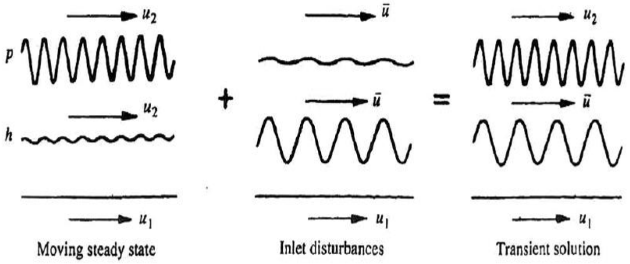
通过文献中实验结果可发现，垂直表面形貌走向摩擦系数最大，同仿真结论一致
(Through the experimental results in the literature, it can be found that the friction coefficient of the vertical surface topography is the largest, which is consistent with the simulation conclusion)

汇报提纲

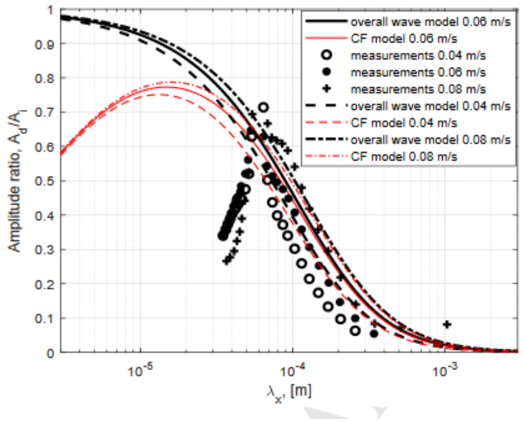
表面形貌对接触副润滑的影响

Effect of surface topography on lubrication of contact pair

◆润滑影响粗糙峰变形的拟合公式计算分析(Calculation and analysis of the fitting formula of the effect of lubrication on the deformation of rough peak)



润滑过程中的综合作用



实验验证综合作用的存在

Venner和Lubrecht的实验拟合公式

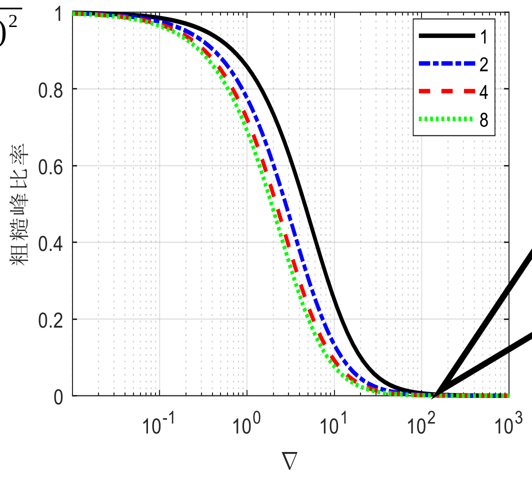
$$\frac{A_d}{A_i}=\frac{1}{1+0.15f(r)\nabla+0.015(f(r)\nabla)^2}$$

$$f(r)=\begin{cases}e^{1-1/r} & \text{if r>1}\\ 1 & \text{otherwise}\end{cases}$$

$$r=\lambda_x/\lambda_y$$

$$\nabla=\sqrt{\frac{2\pi^3}{3}}(\lambda/a)\bar{\alpha}^{3/2}L^{-2}$$

$$\lambda=\min(\lambda_x,\lambda_y)$$



优点：可以分析各向异性粗糙峰变形

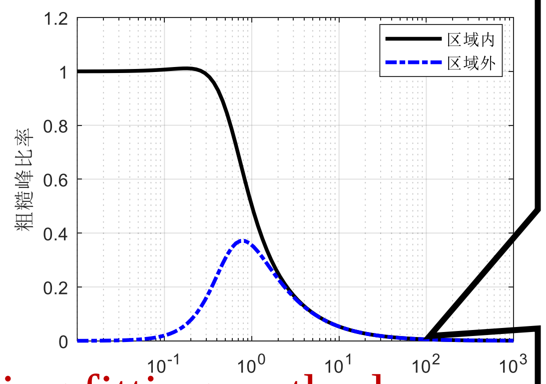
缺点：粗糙峰不能被压平

Hooke的拟合公式

$$\frac{A_d}{A_i}=\frac{1-iCQ}{1-iQ-iCQ}$$

$$Q=\frac{12}{\pi^2}\frac{\eta_x\Delta u\lambda^2}{E^*h^3}$$

$$C=\frac{\pi E^*h}{2B\lambda}$$



优点：可以考虑到非牛顿流体在滚滑状态下的粗糙峰变形

缺点：粗糙峰不能被压平

- ➢ 现有拟合方法不能综合考虑真实表面形貌、润滑油粘度。(The existing fitting methods cannot comprehensively consider the real surface morphology and lubricant viscosity.)
- ➢ 润滑油粘度的影响会带来不同的结果。(The effect of lubricant viscosity will bring different results.)

◆润滑影响粗糙峰变形的摄动计算(Perturbation calculation of the effect of lubrication on the deformation of rough peaks)

$$\delta p = p_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$
$$\delta r = r_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$
$$\delta w = w_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$
$$h_a = r_a + w_a$$
$$\delta h = h_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$
$$\delta\rho = \rho_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$
$$\delta\eta = \eta_a \exp(i\omega_x x)\exp(-i\omega_x u_x t)\exp(i\omega_y y)$$

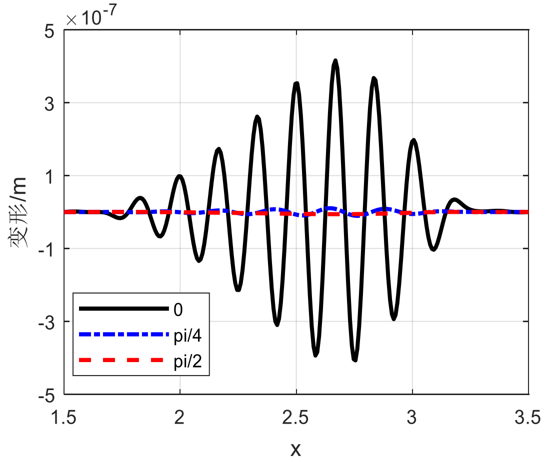
$$z = Z\left(\frac{3\delta h}{h} + \frac{\delta\rho}{\rho} - \frac{\delta\eta}{\eta}\right)$$
$$Z = \frac{\rho h^3}{12\eta}$$
$$s = \delta\rho h + \rho\delta h$$

带入Reynolds方程

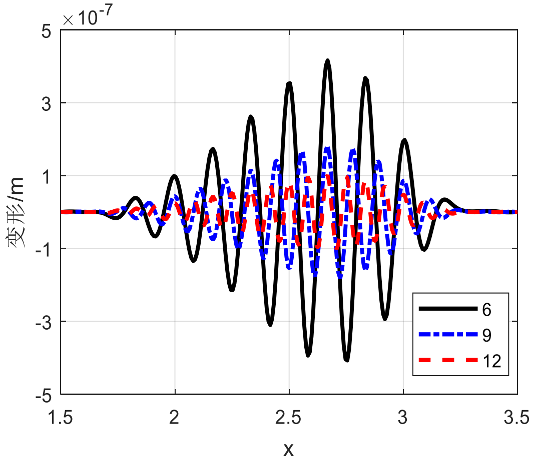
复数形式的Reynolds方程

$$\frac{\partial}{\partial x}\left(Z\frac{\partial p}{\partial x} + z\frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial y}\left(Z\frac{\partial p}{\partial y} + z\frac{\partial P}{\partial y}\right) = u\frac{\partial s}{\partial x} + \frac{\partial s}{\partial t}$$

摄动法得到表面粗糙峰的变形带来的油膜压力以及油膜厚度的波动情况



不同方向角下余弦表面-粗糙峰变形



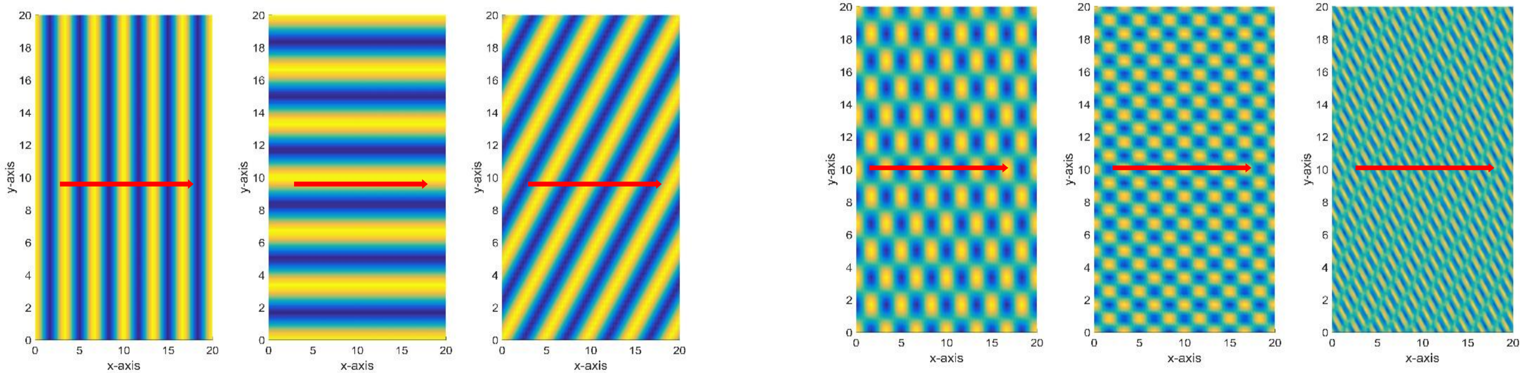
不同跨度比下余弦表面-粗糙峰变形

- ➢ 粗糙峰的变形只与流体流过的粗糙峰的**波长**有关(The deformation of the rough peak is only related to the wavelength of the rough peak that the fluid flows through)
- ➢ **高频**粗糙峰基本没有粗糙峰的变形(High-frequency rough peak basically has no deformation of rough peak)

表面形貌对接触副润滑的影响
Effect of surface topography on lubrication of contact pair

◆多尺度统计学润滑模型(Multi-scale statistical lubrication model)

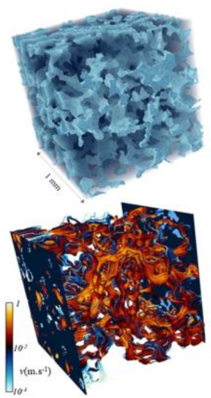
箭头代表流体流动方向

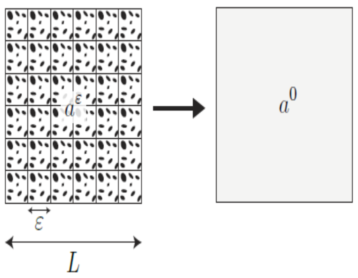


使用了**多尺度**方法。这样可以将微观局部的特性带入到宏观的解中得到考虑到耦合作用的理想化结果。

Reynolds方程homogenization化求解

Homogenization系数计算思想





$$h_\varepsilon(x) = h(x, x/\varepsilon) \quad \varepsilon > 0$$

$$\nabla \cdot (h_\varepsilon^3 \nabla p_\varepsilon) = \varsigma \frac{\partial h_\varepsilon}{\partial x_1} \quad \text{on } \Omega \quad \varsigma = 6\eta u_1$$

$$p_\varepsilon = p_0 + \varepsilon p_1 + \varepsilon^2 p_2 + \cdots,$$

$$\nabla \cdot (A(x) \nabla p_0) - \nabla \cdot B(x) = 0$$

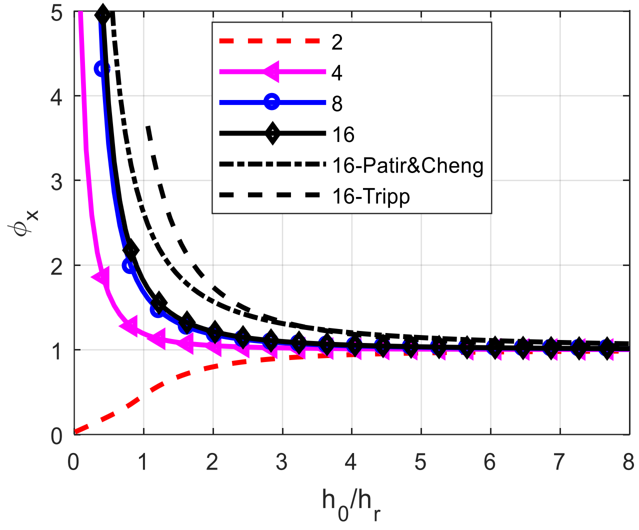
$$A(x) = \begin{pmatrix} a_{1i} \\ a_{2i} \end{pmatrix} = \int_Y h^3 (ei + \nabla_y \chi_i) dy$$

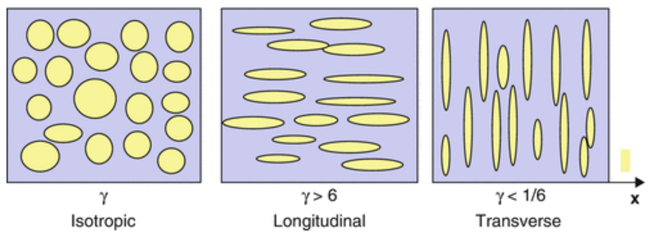
$$B(x) = \begin{pmatrix} b_{11} \\ b_{21} \end{pmatrix} = \int_Y \lambda h e_1 - h^3 \nabla_y \chi_3 dy$$

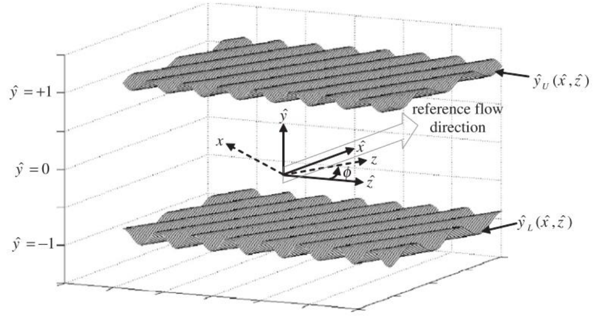
使用HMM逆解的方法求得homogenization系数，可得到压力和剪切流量因子(The homogenization coefficient is obtained by using the inverse solution of HMM, and the pressure and shear flow factors can be obtained)

表面形貌对接触副润滑的影响
Effect of surface topography on lubrication of contact pair

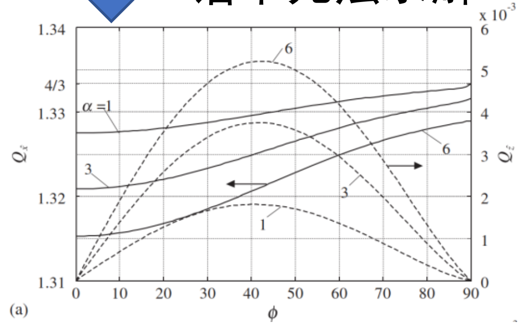
◆多尺度统计学润滑模型对比验证(Comparison and verification of multi-scale statistical lubrication models)

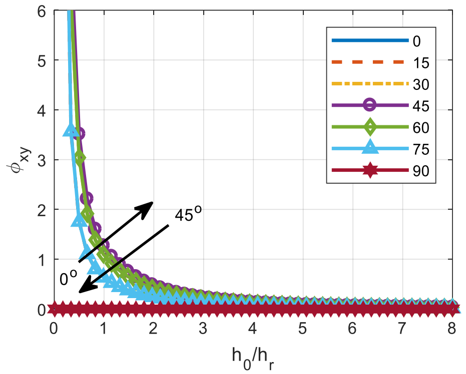






谱单元法求解NS方程





Re =1000, S =0.03沟道的高度， α 为沟道的个数

ϕ ≈42∘时，跨度流的最大压力梯度出现，同时最大的跨度流量出现[3]。

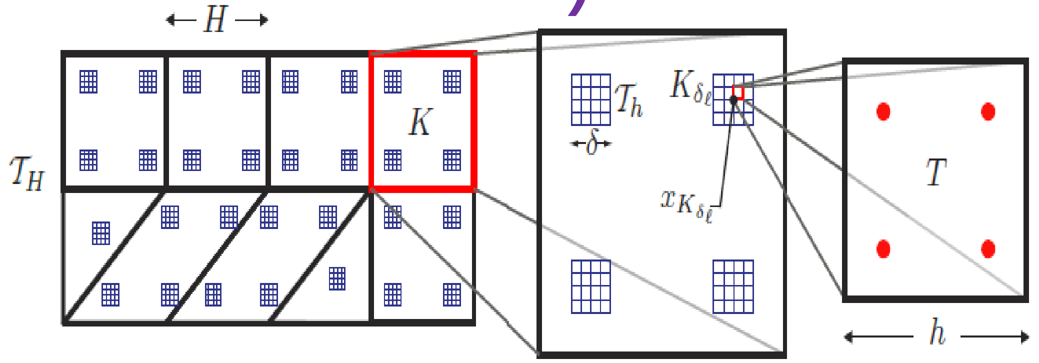
此方法得到的压力流量因子跟其他方法得到的压力流量因子基本保持一致，可以考虑到跨度流的影响(The pressure flow factor obtained by this method is basically consistent with the pressure flow factor obtained by other methods, and the influence of span flow can be considered)

表面形貌对接触副润滑的影响

Effect of surface topography on lubrication of contact pair

安徽工程大学 Anhui Polytechnic University　机械工程学院 SCHOOL OF MECHANICAL ENGINEERING

◆FE-HMM方法(FE-HMM method)



(a) macro discretization with sampling domains (micro-problems)　(b) micro-problems　(c) example of quadrature for a micro-problem

宏观有限元区域

$$V_D^p(\Omega,\mathfrak{I}_H)=\{v^H\in H_D^1(\Omega);u^H|_K\in\mathfrak{R}^p(K),\forall K\in\mathfrak{I}_H\}$$

$$\int_{\hat{K}}\hat{q}(\hat{x})d\hat{x}=\sum_{l=1}^{L}\hat{\omega}_l\hat{q}(\hat{x}_l)\qquad\forall\hat{q}(\hat{x})\in\mathfrak{R}^\sigma(\hat{K})$$

离散的宏观有限元空间，更改成双线性形式

$$B(v^H,\omega^H)=\sum_{K\in\mathfrak{I}_H}\sum_{l=1}^{L}\frac{\omega_{K_l}}{|K_{\delta_l}|}\int_{K_{\delta_l}}a^\varepsilon(x)\nabla v_{K_{\delta_l}}^h\cdot\nabla\omega_{K_{\delta_l}}^h\,dx$$

$$A_K:=B(\varphi_i^H,\varphi_j^H)_{i,j=1}^{\mu_K}$$
$$=\left(\sum_{\ell=1}^{\mathcal{L}}\frac{\omega_{K_\ell}}{|K_{\delta_\ell}|}\int_{K_{\delta_\ell}}a^\varepsilon(x)\nabla\varphi_{K_\ell,i}^h\cdot\nabla\varphi_{K_\ell,j}^h\,dx\right)_{i,j=1}^{\mu_K}$$
$$=\sum_{\ell=1}^{\mathcal{L}}\frac{\omega_{K_\ell}}{|K_{\delta_\ell}|}(\mathcal{A}_{K_\ell}^T A_{mic,K_\ell}\mathcal{A}_{K_\ell}^T)$$

通过微观函数 $v_{K_l}^h$ 计算宏观刚度矩阵，求解 $v_{K_l}^h$

$$\int_{K_{\delta_l}}a^\varepsilon(x)\nabla v_{K_l}^h\cdot\nabla z^h dx=0\qquad\forall z^h\in S^q(K_{\delta_l},\mathfrak{I}_h)$$

$$v_{lin,K_l}^H=v^H(x_{K_l})+(x-x_{K_l})\cdot\nabla v^H(x_{K_l})$$

$$S^q(K_{\delta_l},\mathfrak{I}_h)=\{z^h\in W(K_{\delta_l});z^h|_T\in\mathfrak{R}^q(T),T\in\mathfrak{I}_h\}$$

$W(K_{\delta_l})$ ——确定耦合条件或是计算微观函数的边界条件

周期耦合　$W(K_{\delta_l})=W_{per}^1(K_{\delta_l})$

Dirichlet耦合 $W(K_{\delta_l})=H_0^1(K_{\delta_l})$

变分得到

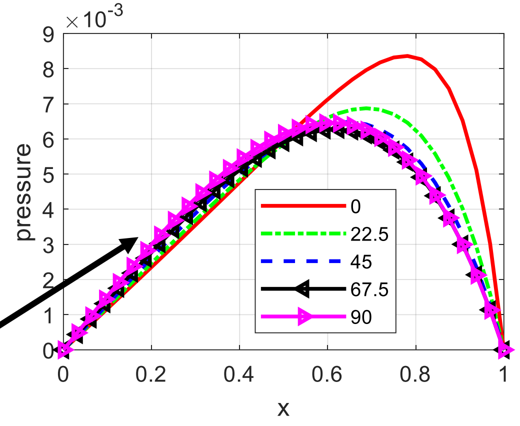
$$B(u^H,v^H)=\int_\Omega fv^H dx+\int_{\partial\Omega_N}g_N v^H dx-B(g_D,v^H)\quad\forall v^H\in V^p(\Omega,\mathfrak{I}_H)$$

表面形貌对接触副润滑的影响

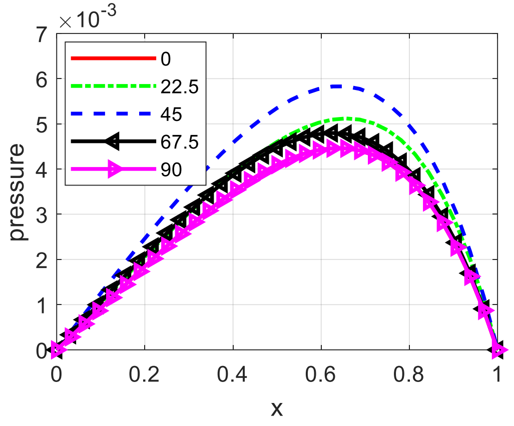
Effect of surface topography on lubrication of contact pair

◆表面形貌对油膜压力的影响(Effect of surface topography on film pressure)

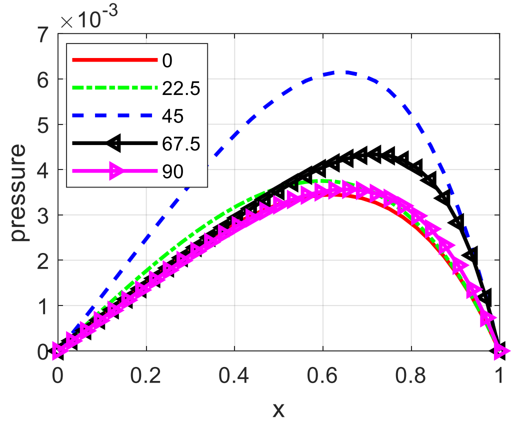
光滑表面差分法求解得到的油膜压力曲线同FE-HMM法解决结果在数值上基本一致。



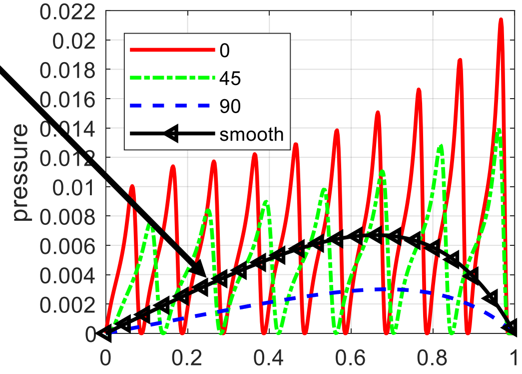
余弦表面无量纲压力分布



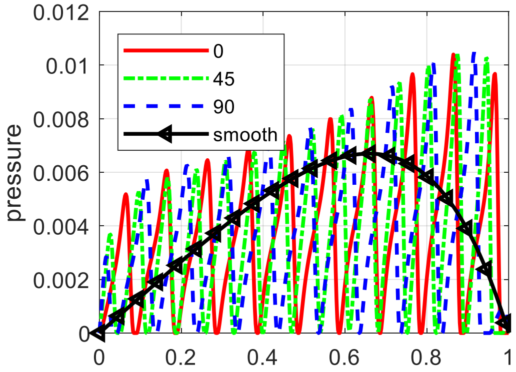
双余弦表面的压力分布



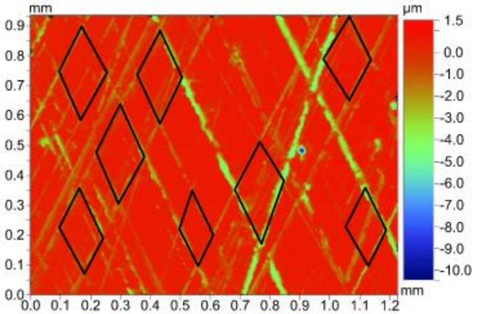
双余弦表面的压力分布



差分法计算的余弦压力分布



差分法计算的双余弦压力分布



缸套表面

- FE-HMM可以不用计算流量因子，直接计算油膜压力(FE-HMM can directly calculate the oil film pressure without calculating the flow factor)
- FE-HMM的解考虑到了表面微观结构的影响(The effect of surface microstructure is considered in the solution of FE-HMM)

汇报提纲

1. 表面形貌对接触副磨损的影响
2. 表面形貌对接触副润滑的影响
3. 缸套-活塞环润滑分析
4. 原位调控润滑方法
5. 密封泄漏机理研究

缸套-活塞环润滑分析
Lubrication Analysis of Cylinder Liner-piston Ring

◆基于JFO边界条件和粗糙度取向性的润滑分析模型(Lubrication Analysis Model Based on JFO Boundary Condition and Roughness Orientation)

基于JFO边界条件的瞬态雷诺方程

$$\sum_{i=1}^{2}\frac{\partial}{\partial x_i}\left(A_{ii}^*\frac{\partial p_0}{\partial x_i}\right)=\frac{\partial B_1^*\theta_0}{\partial x_1}+T^*$$

$$p_0\geq 0$$
$$0\leq\theta_0\leq 1$$
$$p_0(1-\theta_0)=0$$

$$A_{11}^*=\frac{1}{12\mu\overline{h^{-3}}^{Y}}$$

$$A_{22}^*=\frac{\overline{h^3}^{Y}}{12\mu}$$

$$B_1^*=\frac{U}{2}\frac{\overline{h^{-2}}^{Y}}{\overline{h^{-3}}^{Y}}$$

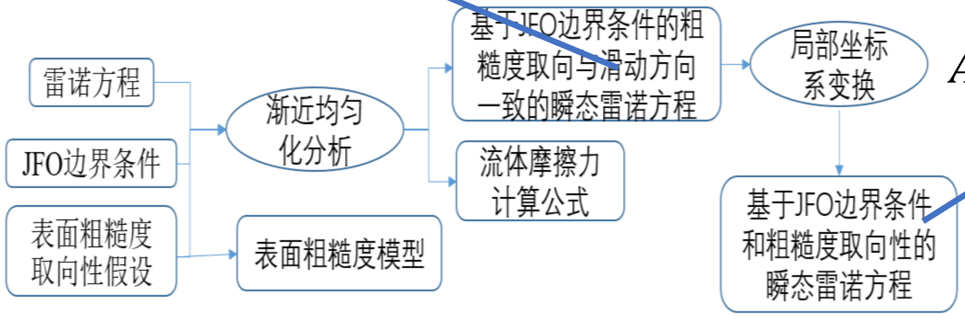
$$T^*=\frac{\partial}{\partial t}\big(\theta_0 h(x)\big)$$

计算域边界条件

$$B_1^*\theta_0-A_{11}^*\frac{\partial p_0}{\partial x_1}-A_{12}^*\frac{\partial p_0}{\partial x_2}=\mathrm{Q}$$

流体摩擦力

$$f_L=\iint_X \frac{h}{2}\frac{\partial p_0}{\partial x_1}+\mu u\overline{h^{-1}}^{Y}dx_1dx_2$$



$$\sum_{i,j}\frac{\partial}{\partial x_i}\left(A_{ij}^*\frac{\partial p_0}{\partial x_j}\right)=\frac{\partial\theta_0 B_1^*}{\partial x_1}+\frac{\partial\theta_0 B_2^*}{\partial x_2}+T^*$$

$$p_0\geq 0$$
$$0\leq\theta_0\leq 1$$
$$p_0(1-\theta_0)=0$$
$$(i,j=1,2,i\neq j)$$

$$T^*=\frac{\partial}{\partial t}\big(\theta_0 h(x)\big)$$

$$A_{11}^*=a_{11}^*cos^2\gamma+a_{22}^*sin^2\gamma$$
$$A_{22}^*=a_{11}^*sin^2\gamma+a_{22}^*cos^2\gamma$$
$$A_{12}^*=A_{21}^*=(a_{11}^*-a_{22}^*)sin\gamma cos\gamma$$
$$B_1^*=b_1^*cos^2\gamma+b_2^*sin^2\gamma$$
$$B_2^*=(b_1^*-b_2^*)sin\gamma cos\gamma$$

$$a_{11}^*=\frac{1}{12\mu\overline{h^{-3}}^{Y}}$$

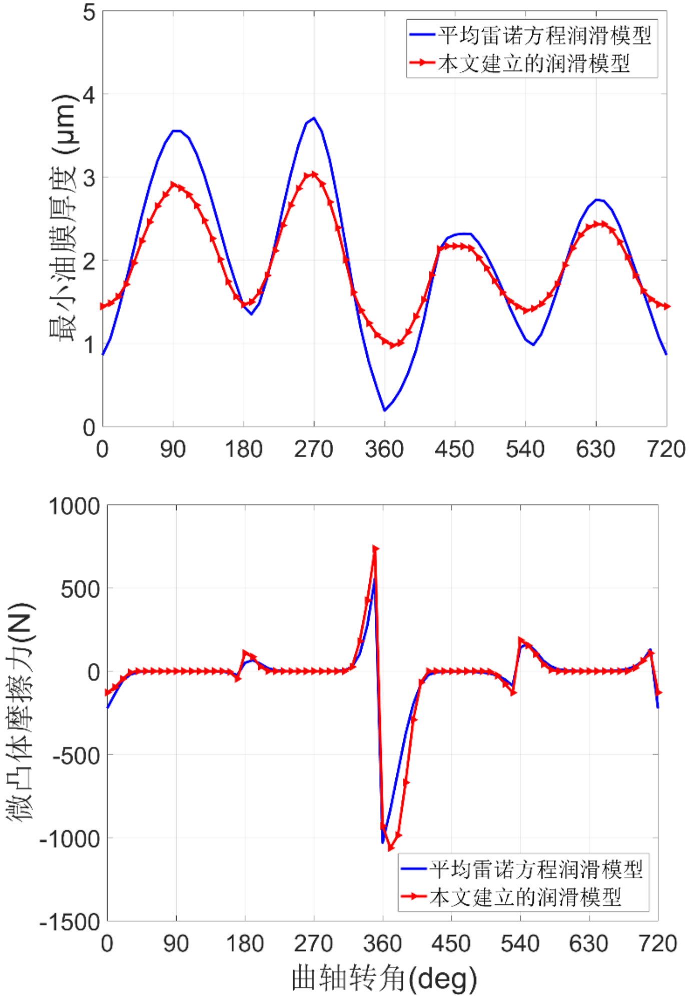
$$a_{22}^*=\frac{\overline{h^3}^{Y}}{12\mu}$$

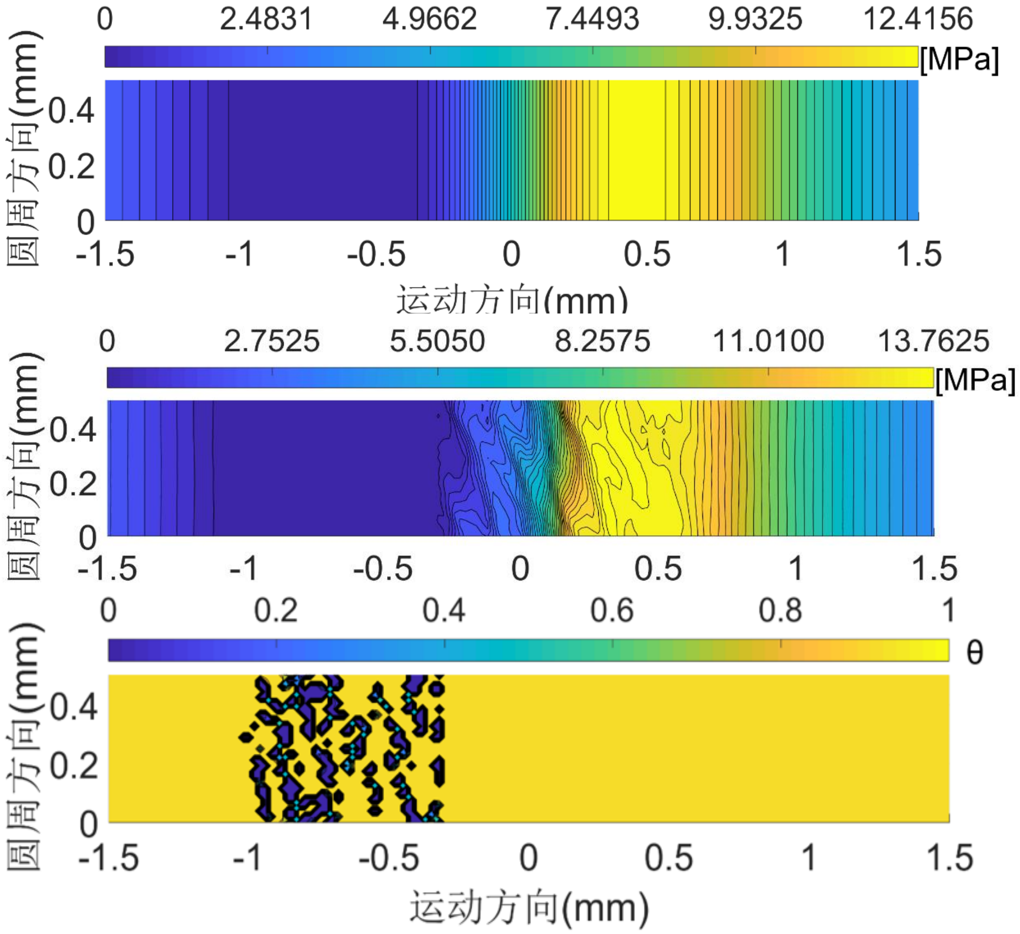
$$b_1^*=\frac{U}{2}\frac{\overline{h^{-2}}^{Y}}{\overline{h^{-3}}^{Y}}$$

$$b_2^*=\frac{U}{2}\overline{h}^{Y}$$

缸套-活塞环润滑分析
Lubrication Analysis of Cylinder Liner-piston Ring

◆新润滑模型与平均雷诺方程润滑模型的比较(Comparison between new lubrication model and average Reynolds equation lubrication model)





曲轴转角450°时本文建立的润滑模型得到的润滑油膜压力和润滑油填充率分布

两种方法结果整体趋势一致，但本方法考虑了空穴的影响。

(The overall trend of the results of the two methods is consistent, but this method considers the effect of holes.)

汇报提纲

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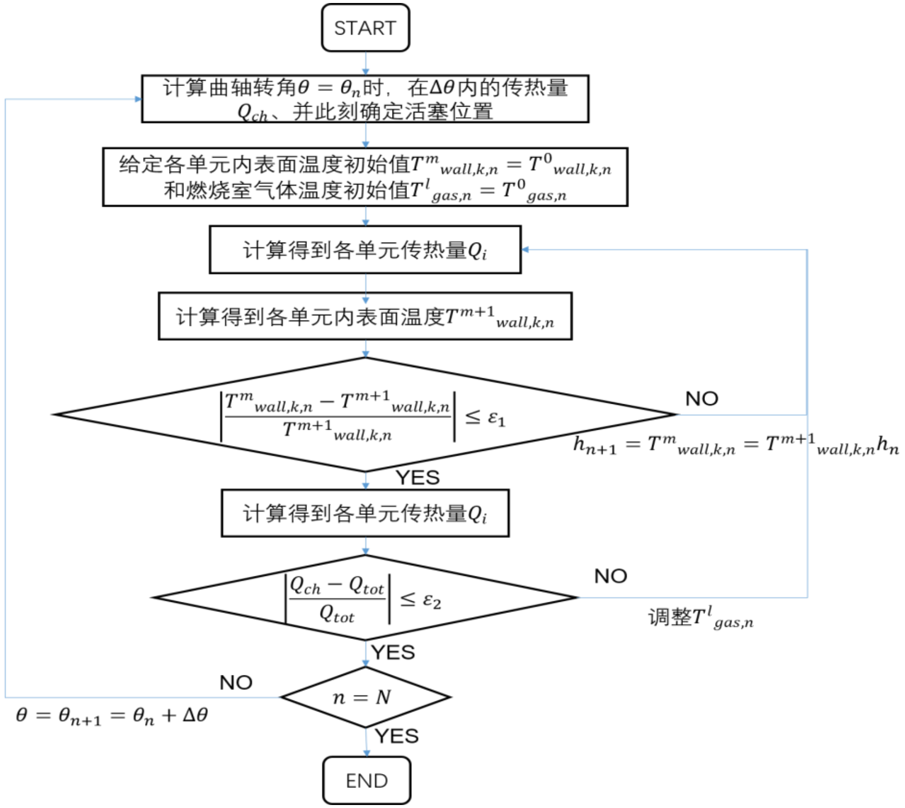
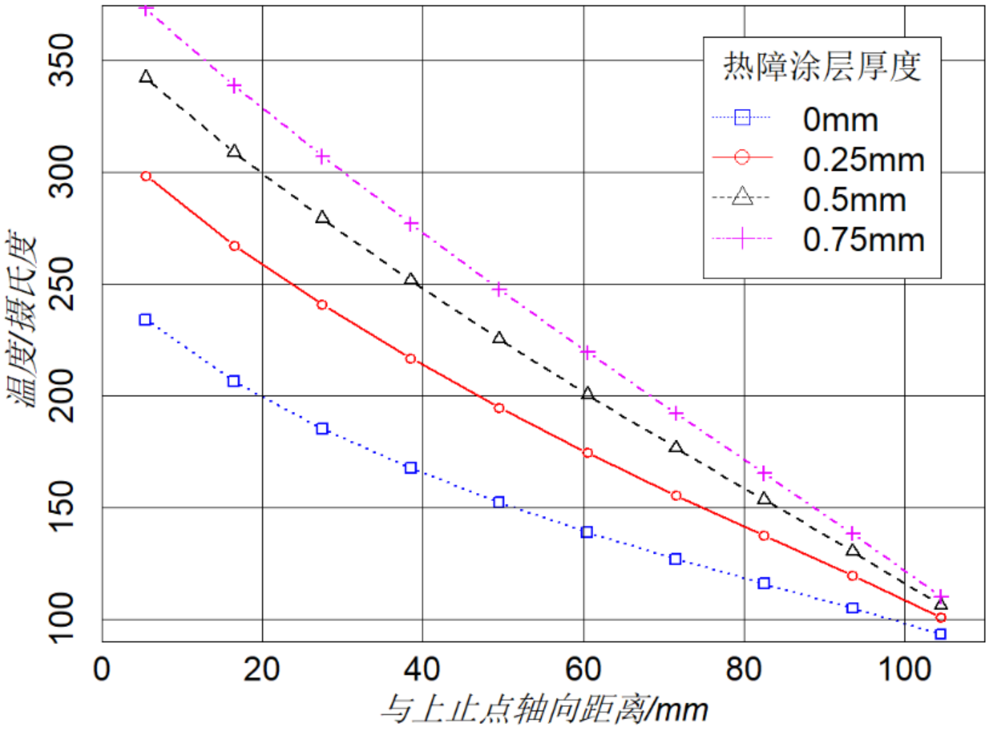
原位调控润滑方法
In-situ control lubrication method

◆缸套外侧热障涂层原位调控(In-situ control of thermal barrier coating on the outside of cylinder liner)

缸套外侧加隔热涂层，控制缸套内部温度，调节缸套润滑油粘度，减少摩擦损失。揭示涂覆位置、大小等参数对气缸套内表面温度的影响，将气缸套上止点和下止点之间的区域沿轴向平均分为N个单元。(The outer side of the cylinder liner is coated with thermal insulation coating to control the internal temperature of the cylinder liner, adjust the viscosity of the cylinder liner lubricating oil and reduce the friction loss. The influence of coating location, size and other parameters on the temperature of the inner surface of the cylinder liner is revealed. The area between the top dead center and the bottom dead center of the cylinder liner is divided into N units along the axial direction.)



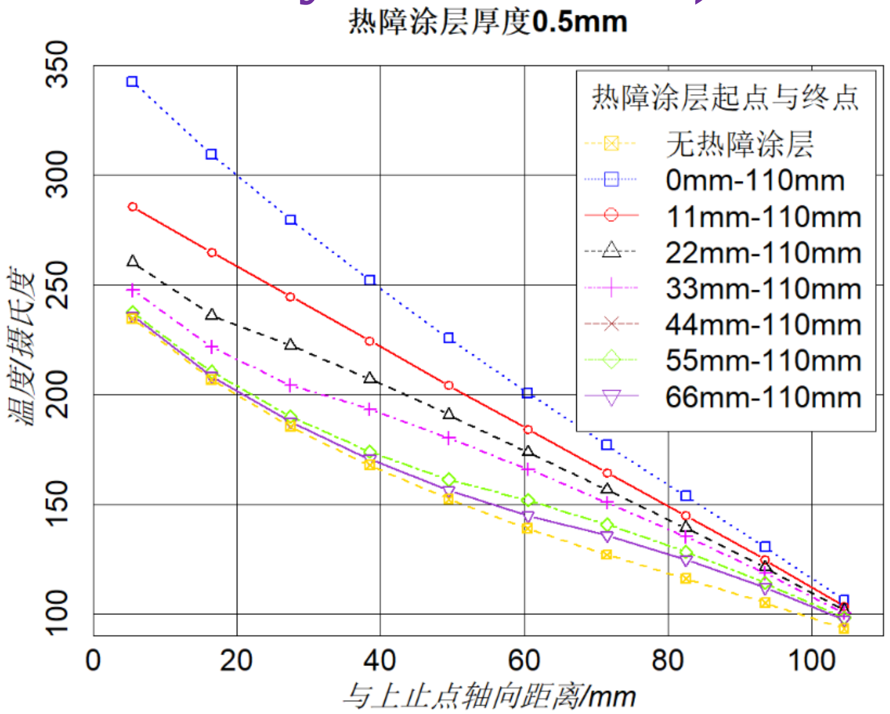
气缸套整体热障涂层

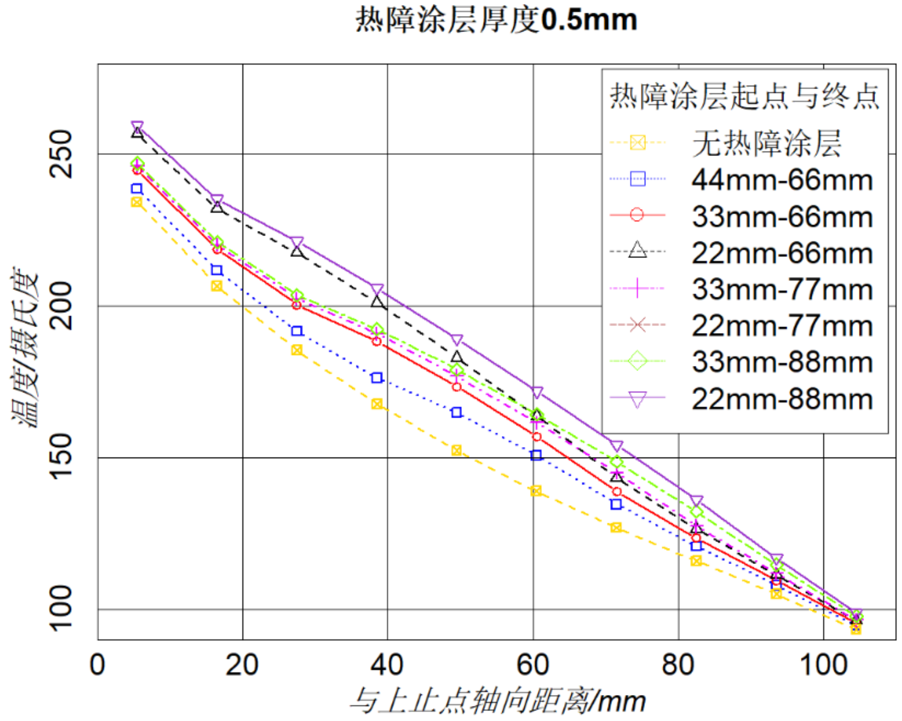


原位调控润滑方法

In-situ control lubrication method

◆缸套外侧热障涂层原位调控(In-situ control of thermal barrier coating on the outside of cylinder liner)





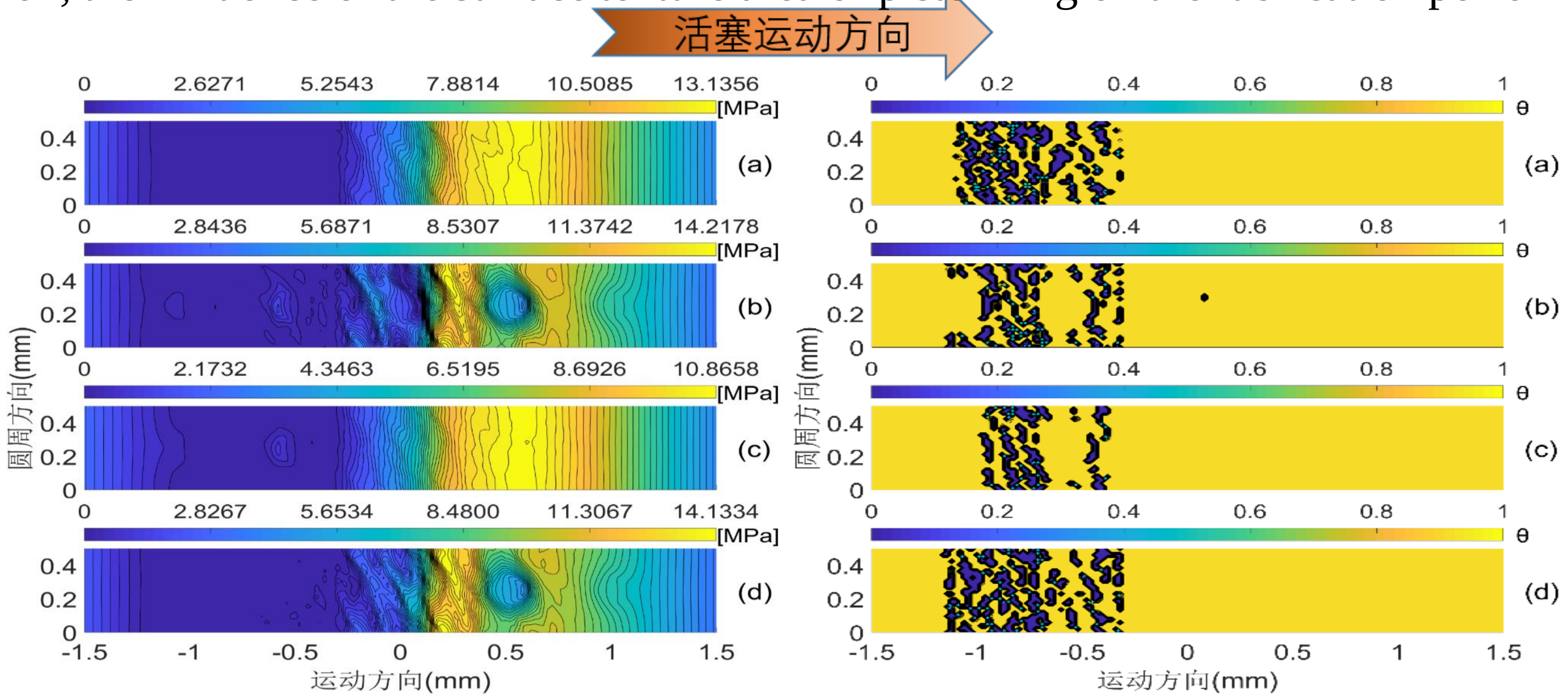
当热障涂层涂敷的起始点距离上止点为33mm时，行程中部区域气缸套内表面温度明显上升，而上止点附近气缸套内表面温度较小(When the starting point of the thermal barrier coating is 33mm from the top dead center, the temperature of the inner surface of the cylinder liner in the middle of the stroke increases significantly, while the temperature of the inner surface of the cylinder liner near the top dead center is smaller)

热障涂层涂敷起点和终点与上止点轴向距离为44mm和66mm、33mm和88mm的两个方案较好(The two schemes with the axial distance between the starting point and end point of thermal barrier coating and the top dead center of 44mm and 66mm, 33mm and 88mm are betterThe two schemes with the axial distance between the starting point and end point of thermal barrier coating and the top dead center of 44mm and 66mm, 33mm and 88mm are better)

原位调控润滑方法
In-situ control lubrication method

◆表面织构原位调控(In-situ control of surface texture)

运用基于JFO边界条件考虑粗糙度的活塞环-气缸套瞬态润滑模型，研究活塞环表面织构区域对润滑性能的影响。(Using the transient lubrication model of piston ring-cylinder liner considering roughness based on JFO boundary condition, the influence of the surface texture area of piston ring on the lubrication performance is studied.)

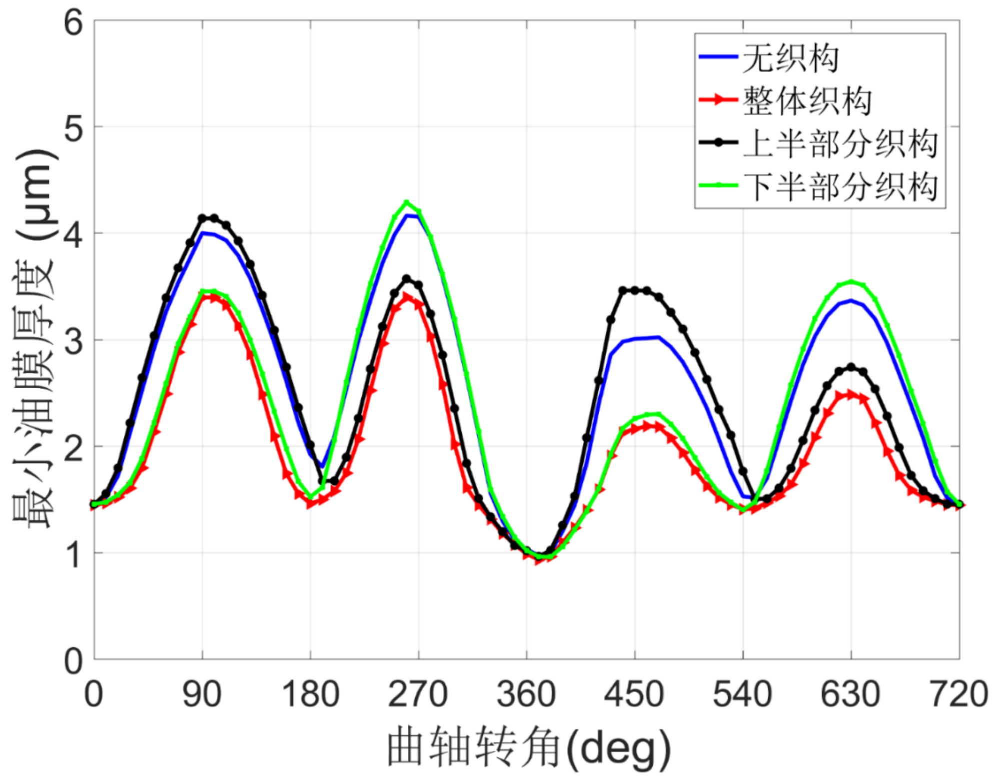


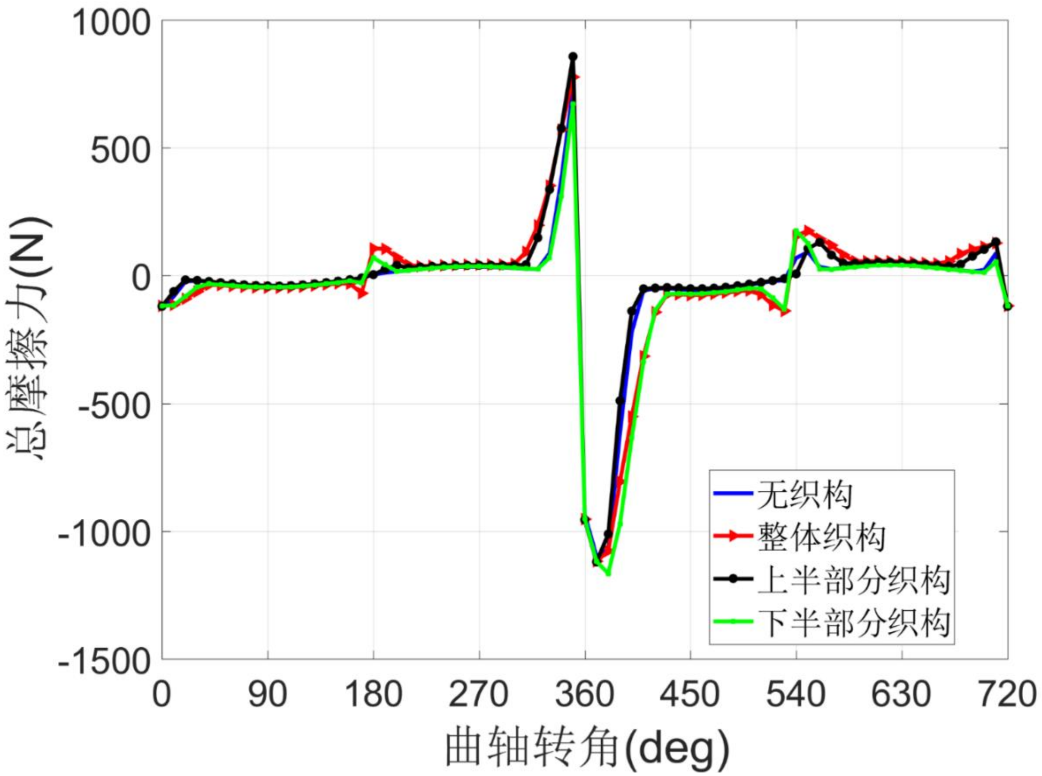
不同织构区域下油膜压力与润滑油填充率分布：

(a)无织构，(b)整体织构，(c)上半部分织构，(d)下半部分织构

原位调控润滑方法
In-situ control lubrication method

◆表面织构原位调控(In-situ control of surface texture)





- 局部织构方案优于整体织构方案(The local texture scheme is superior to the overall texture scheme)
- 局部织构对润滑性能的影响与接触副相对运动方向和载荷密切相关(The influence of local texture on lubrication performance is closely related to the relative motion direction and load of the contact pair)

汇报提纲

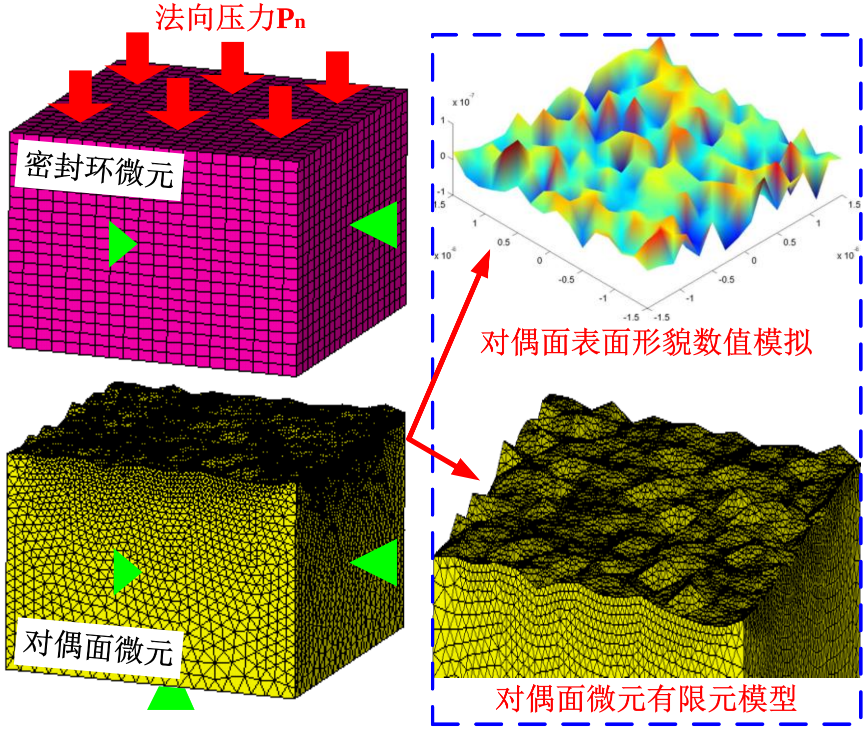
密封泄漏机理研究

Study on seal leakage mechanism

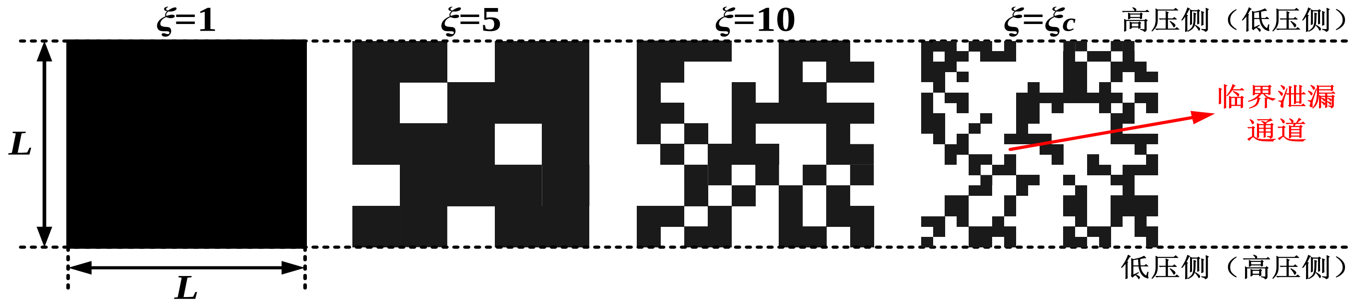
安徽工程大学 Anhui Polytechnic University　机械工程学院 SCHOOL OF MECHANICAL ENGINEERING

◆静密封泄漏机理研究(Study on leakage mechanism of static seal)

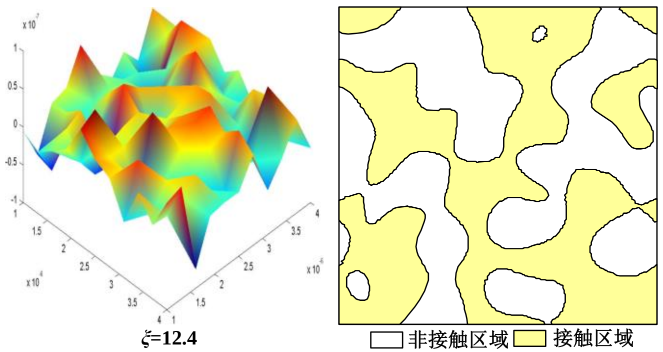
建立含表面形貌的弹性体接触模型，通过不断修改尺度系数模拟刚刚形成泄漏通道时的临界放大倍数，基于Persson泄漏理论建立泄漏计算模型。(Establish an elastic contact model with surface topography, simulate the critical magnification when the leakage channel is just formed by continuously modifying the scale coefficient, and establish a leakage calculation model based on Persson leakage theory.)



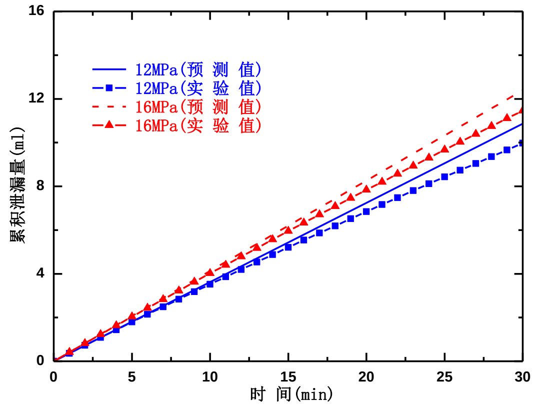
考虑表面形貌的接触模型



Persson泄漏理论



真实接触面积

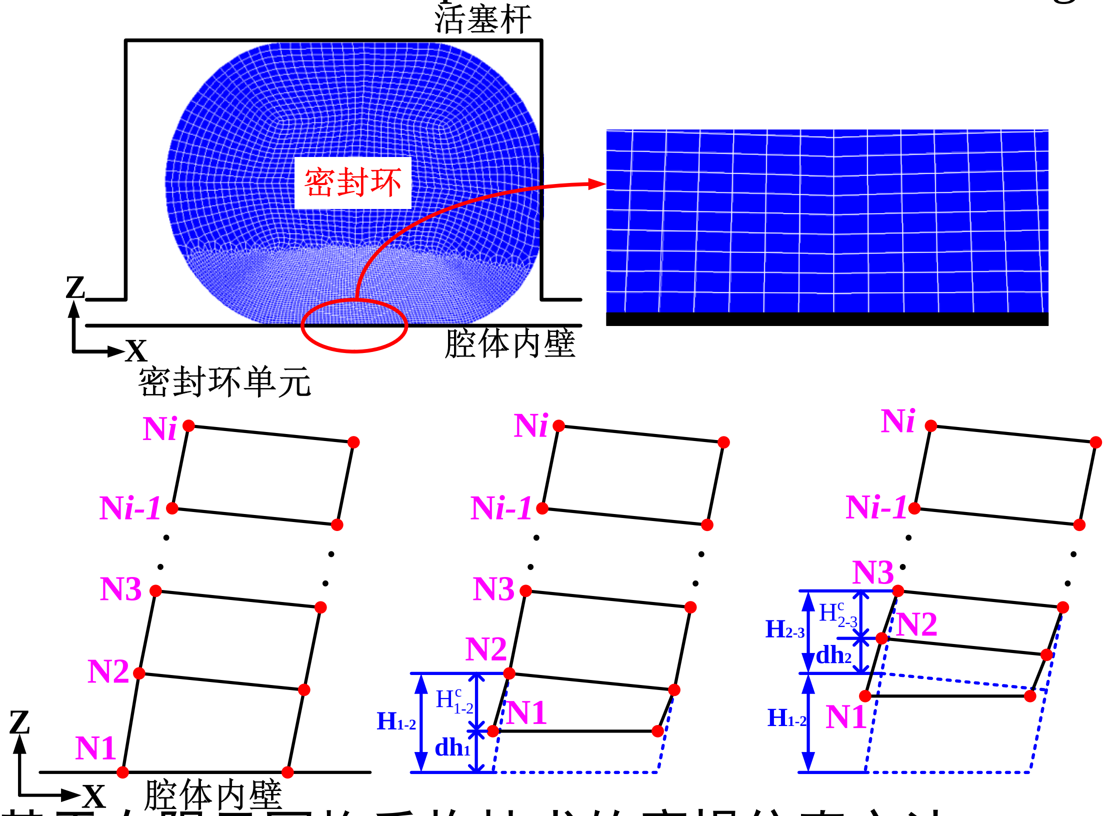


泄漏量计算

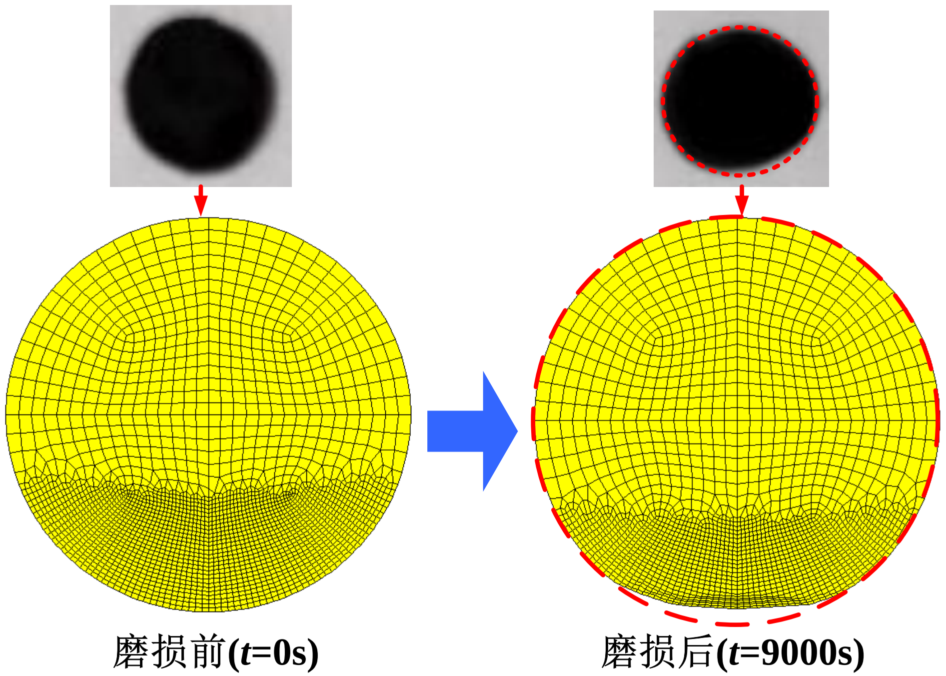
密封泄漏机理研究
Study on seal leakage mechanism

◆动密封磨损仿真研究(Simulation study on dynamic seal wear)

基于有限元网格重构技术，建立动密封元件磨损仿真策略，通过移动表面及内部节点位置更新动密封轮廓，用以模拟密封磨损后轮廓变化。(Based on the finite element mesh reconstruction technology, the wear simulation strategy of dynamic seal components is established, and the dynamic seal profile is updated by moving the surface and internal node position to simulate the change of the seal profile after wear.)



基于有限元网格重构技术的磨损仿真方法



密封圈磨损前后截面对比

谢谢！

Thank you!