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Title page

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Effect of Texture on Energy Consumption for High Frequency Hydraulic Impact Piston Pair

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Abstract: Limited by the influence of the traditional clearance seal structure on the leakage and friction loss of the piston pair, the energy utilization ratio of the hydraulic impactor is difficult to improve. As such, a novel clearance seal structure with cylindrical texture for the impact piston is proposed to solve it. Considering the leakage and friction loss of impact piston pair, an energy consumption evaluation index is put forward. Based on the average Reynolds equation, an energy consumption analysis model for a textured high-frequency hydraulic impact piston pair is established, and the influence of piston texture parameters for the YG45 hydraulic impactor on energy consumption under rated working conditions is studied. The results show that energy consumption of an impact piston pair accounts for 29.77% of total energy loss. The variation of area ratio textured makes the ratio of energy consumption to the piston pair decline 1.32%~10.98%, where the optimum area rate textured is 0.64~0.7. The variation of depth ratio textured leads to the reduction of the ratio of energy consumption to piston pair by 3.21% ~ 5.68%, where the optimum depth ratio textured is 1~1.1. The texture structure of the piston pair significantly reduces the energy consumption of the impact piston pair, and it is an effective approach to avoid the design dilemma for the conventional impact piston pair.

Key Words: hydraulic impactor; impact piston pair; energy consumption; surface texture; cylindrical micro-texture

1. Introduction

The energy utilization ratio of the hydraulic impactor¹ is the ratio of piston impact energy to hydraulic energy. It reflects the conversion efficiency from hydraulic energy to

mechanical energy of the hydraulic impactor, and directly affects the drilling efficiency of the hydraulic impactor [1-2]. In the process of high frequency reciprocating, the friction loss and leakage loss of the piston pair are important parts in total energy loss during energy transformation of the hydraulic impactor. The friction loss of the piston pair is caused by clearance lubrication between the impact piston and cylinder block, as well as the leakage loss caused by clearance seal. In order to improve the energy utilization rate of hydraulic impactor, scholars at home and abroad have conducted a significant amount of research on the energy consumption calculation of impact piston pairs and the optimization of piston structural parameters. Through an analysis of a rock drill impact module, Oh developed the impact performance analysis tool and analyzed the variation law of energy utilization rate for hydraulic impactor relative to rock stiffness [3]. Taking into considering the leakage and friction of impact piston pair, Seo analyzed the performance of the hydraulic impactor through simulation, then put forward the design method of structural parameters of the impact piston and determined the range of piston parameters [4]. Flegner tested the friction and wear of piston during the drilling process of hydraulic impactor under different drilling pressures and rotational speeds. In view of avoiding premature wear, the design method of piston structure parameters for enhancing the lubrication effect was put forward and the range of piston parameters was determined [5]. Wang also established a kinematics mathematical model of the impact piston for the hydraulic impactor based on Reynolds equation. The bearing capacity, cavity pressure, friction force and friction energy consumption of the piston pair were studied [6]. However, it is still limited to the conventional clearance lubrication and sealing mechanism in the design optimization of the impact piston pairs mentioned above, making it difficult to realize the design of declining energy consumption for the impact piston hard to realize. Increasing clearance reduces friction loss with the increase of leakage loss, and decreasing clearance decreases leakage loss with the increase of friction loss. This brings about the failure of energy utilization improvement for the hydraulic impactor, causing energy utilization rate of the hydraulic impactor to

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be much lower than that of other hydraulic actuators.

Surface texture is a micro geometric structure fabricated by special manufacturing technology on the surface with certain size, shape, and arrangement, such as micro pits and grooves. The additional hydrodynamic lubrication effect of texture can be used to improve the surface bearing capacity and reduce the surface friction [7-8]. It has been widely recognized as an effective method in reducing friction, declining wear and increasing surface bearing capacity [9-11]. Pit texture was applied to the surface of the cylinder liner of an internal combustion engine to improve the lubricity of the piston pair, and the friction between the piston ring and cylinder liner was greatly reduced accordingly [12]. The groove texture was applied to the bearing surface to reduce the friction on the bearing surface with a better stability working parameter [13]. It has also been demonstrated by a number of scholars that different morphologies and parameters of texture have a significant influence on surface friction loss. Zhang studied the friction force of surface texture parallel sliders with different morphologies, and found that the hydrodynamic lubrication performance of circular textures is better with a lower friction coefficient [14]. Gropper analyzed the performance of bearing with variation of texture area and textured depth on surface determining that the optimum texture depth should be slightly less than the minimum film thickness of unreformed bearing, and that there existed an optimum texture depth and density with low friction existed [15]. Wang conducted research on texture density, and discovered that the friction force reduced favorably with a texture density of 5%~13%.

In order to improve energy efficiency of the hydraulic impactor, the impact piston pair structure with cylindrical texture is proposed. By analyzing the physical and mechanical mechanisms of energy consumption of the impact piston pair, energy consumption analysis model of the impact piston pair is established, then solved numerically combined with the Reynolds equation. The effect of texture parameters on the energy consumption of the impact piston is obtained, which provides a theoretical basis to design a high frequency hydraulic impactor with surface texture on the impact piston.

2. Physical model of the textured impact piston pair with high frequency

2.1 Geometric model of the impact piston pair with high frequency

The impact piston pair with high frequency is widely used in various kinds of impact machinery. The piston pair used in the impact mechanism of the YG45 hydraulic rock drill falls into this category, which is shown in Fig. 1. It

presents characteristics of periodic motion with high frequency and varied velocity. The impact mechanism of the drill is composed of the impact piston, cylinder block, cylinder sleeve, front support seat, rear support seat, front chamber, and rear chamber. The front chamber is usually filled with high-pressure oil, and the rear chamber is alternately filled with both the high-pressure oil and the return oil. The impact piston moves under the pressure of the front and rear chamber.

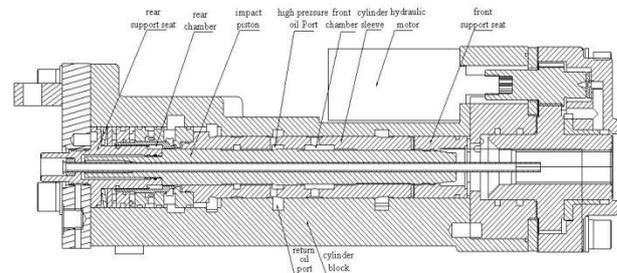


Fig. 1 Impact mechanism of the YG45 hydraulic rock drill

In the impact piston pair, clearance seal is used between the impact piston and cylinder sleeve, and the clearance value is generally set to 55-65 μm . The piston support seat is arranged in the front and rear of the cylinder block to support and guide the impact piston. In the working process of hydraulic impactor, high-pressure oil flows from the inlet into the cylinder block, as well as the front and rear chamber. Due to the difference of the bearing area of the front and rear chamber, the impact piston moves towards the right with great acceleration under the action of the pressure difference. When the load changes on both ends of the sleeve valve, the oil path will be switched. The front chamber is filled with high-pressure oil, and the rear chamber is filled with return oil. High-pressure oil within the front chamber pushes the impact piston to move leftward with acceleration.

2.2 Surface texture model of the impact piston

Taking the impact piston pair in a high frequency hydraulic impactor as the research object, the impact piston and cylinder block are fitted with clearance, the cylindrical micro-texture is machined onto the shoulder surface of the piston seal table, and the seal length is 40 mm. The shoulder surface is completely textured along the axial direction. According to the confinement of the unit area of microtexture, a total of 1647 microtexture units can be machined. The distribution of circular microtexture is presented in Fig. 2.

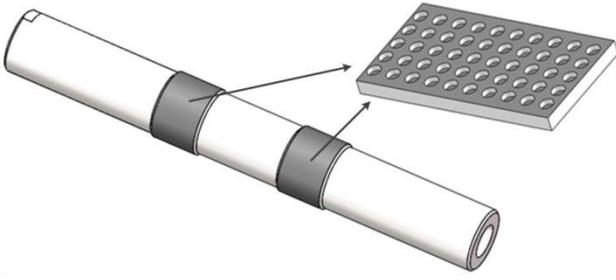


Fig. 2 Cylindrical texture geometric model of the impact piston surface

The impact piston diameter in the diagram is 30 mm, and total length of the seal is 40 mm. The radial dimension of the oil film thickness and texture is significantly smaller than the diameter of the piston. The influence of the curvature radius of oil film can be ignored, and the surface of the piston can be equivalently expanded into a plane equivalently [16]. The distribution of oil film thickness and oil film pressure varies regularly in each micro-texture unit region, and the pit texture is periodically distributed periodically. The oil film thickness and oil film pressure also change periodically in each texture unit region, so a single microtexture control unit is studied. The pressure and friction of the entire sealing area are then approximately according to the characteristics of its periodic arrangement [17]. The geometric model of the microtexture control unit is presented in Fig.3.

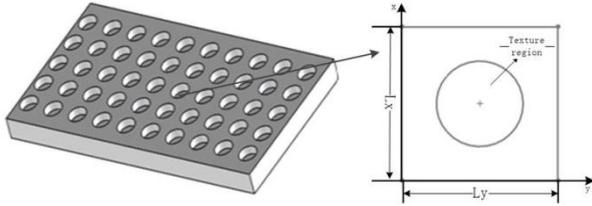


Fig. 3 Geometric model of the texture control unit

The coordinate system is established on the piston surface, in which the x -axis is along the axis of the piston, and the y -axis is set in radial direction of the piston. L_x is the length of the control unit along the x -axis, and L_y is the length of the control unit along the y -axis. A is the area of control unit. The area rate textured is S_p , and it can be expressed as:

$$S_p = \frac{A}{L_x L_y} \quad (1)$$

3. Analysis model of energy consumption for the textured impact piston pair with high frequency

A hypothesis is put forward to establish an analysis model of energy consumption for the textured impact piston pair with high frequency. The clearance of the piston and the cylinder block is assumed to fill with oil, whereas the hydraulic oil is assumed not to slip on the piston surface. In other words, the velocity of the oil

attached to the surface of the piston is the same as the velocity of the piston. The curvature radius of the piston surface is rather large compared with the thickness of the clearance oil film, and so the influence of the oil film curvature is ignored. Hydraulic oil is assumed to have the characteristics of Newtonian fluid. The viscosity value of hydraulic oil is regarded as a constant value along the thickness of the clearance oil film. The cylinder block is considered as a rigid body, whereas the impact piston is considered as an elastic body. The impact piston pair of hydraulic impactor works vertically and is used with a guiding device, so the influence of gravity and radial force to the eccentricity of the impact piston is very small. Additionally, both the piston and inner hole of the cylinder body are manufactured with high machining roundness. With these conditions, the eccentricity of the working impact piston is ignored. The change of hydraulic oil temperature during piston movement is also disregarded.

3.1 Energy consumption evaluation index of the impact piston pair

In the process of high frequency reciprocating, the friction loss and leakage loss of the piston pair are important parts of the total energy loss during the energy transformation of hydraulic impactor. The friction loss of the piston pair is caused by clearance lubrication between the impact piston and cylinder block, and the clearance seal causes leakage loss. In this paper, the energy consumption for the impact piston pair is evaluated in a cycle that takes into consideration both the friction loss and leakage loss. During the movement of the impact piston, the clearance between the impact piston and cylinder block is filled with hydraulic oil to avoid direct contact with the cylinder block. The relative motion of the impact piston and cylinder block results in dynamic oil film in the clearance. The shearing force of oil causes the friction force of dynamic pressure oil film on a solid surface. When the velocity of the impact piston increases, the friction forces increase due to the increase in shear force. In addition, the seal form between the impact piston and cylinder block is the clearance seal. Hydraulic oil leakage exists at the two ends of the impact piston pair under the action of the pressure difference between two ends of the impact piston, causing leakage loss. Considering the leakage and friction loss of the impact piston pair, the energy consumption of the impact piston pair in one cycle is calculated as follows:

$$W = W_L + W_F = \int_0^T P_{Ll} dt + \int_0^T P_{Ft} dt \quad (2)$$

where W is the energy consumption of the impact piston pair in one cycle, W_L is the leakage loss of the impact piston pair in one cycle, W_F is the friction loss of the

impact piston pair in one cycle, P_{L_t} is the leakage power of the impact piston pair at time t , P_{F_t} is the friction power of impact piston pair at time t , T is the motion cycle of the hydraulic impactor with high frequency.

3.2 Leakage loss calculation

Leakage loss power is determined by the pressure difference and leakage flow at both ends of the impact piston. The calculation of the loss energy of the piston pair leakage can be expressed by :

$$W_L = \int_0^T P_{L_t} dt = \int_0^T \Delta p_t Q_{L_t} dt \quad (3)$$

where W_L is the leakage loss of the impact piston pair in one cycle, P_{L_t} is the leakage power of the impact piston pair at time t , Δp_t is the pressure difference between two ends of the impact piston pair at time t determined by the working condition, Q_{L_t} is the leakage flow at time t , and T is the motion cycle of hydraulic impactor with high frequency.

Leakage flow varies with the change of relative velocity between the impact piston and cylinder block. The impact piston and cylinder block are completely concentric, and the seal clearance is very small. Therefore, it can be regarded as concentric ring clearance. Under the conditions of the relative motion between the piston and the cylinder, the leakage flow is the flow of the hydraulic oil in the annular clearance. It can be calculated by the flow formula of the annular clearance:

$$Q_{L_t} = \frac{\pi d h_t^3 \Delta p_t}{12 \eta l} + \frac{\pi d h_t v_t}{2} \quad (4)$$

where Q_{L_t} is the leakage flow at time t , d is the diameter of the impact piston, h_t is the leakage clearance at time t , Δp_t is the pressure difference between two ends of the impact piston pair at time t determined by the working condition, η is the dynamic viscosity of hydraulic oil, l is the runner length, and v_t is the relative velocity of the impact piston and cylinder block at time t .

The structure of the texture unit is both consistent and symmetrical. The leakage clearance at time t can be approximated by the minimum film thickness of a texture element:

$$h_t = \min h(x, y) \quad (5)$$

where h_t is the leakage clearance at time t , and $h(x, y)$ is the film thickness distribution function of texture units.

The cross section model of flow field for the control unit with cylindrical texture is shown in Fig. 4. In the figure, v is the velocity of the impact piston, h_0 is the minimum initial clearance between cylinder block and impact piston, and h_g is the height of the cylindrical texture element. Considering contact deformation under

oil film pressure, the actual oil film thickness distribution function of the texture flow field can be expressed as follows:

$$h(x, y) = \begin{cases} h_0 + h_g + \tau(x, y), & x^2 + y^2 \leq r_p^2 \\ h_0 + \tau(x, y), & x^2 + y^2 > r_p^2 \end{cases} \quad (6)$$

where h_0 is the minimum initial clearance between the cylinder block and impact piston, h_g is the height of the cylindrical texture element, $\tau(x, y)$ is the contact deformation, and r_p is the radius of bottom surface of cylindrical texture element.

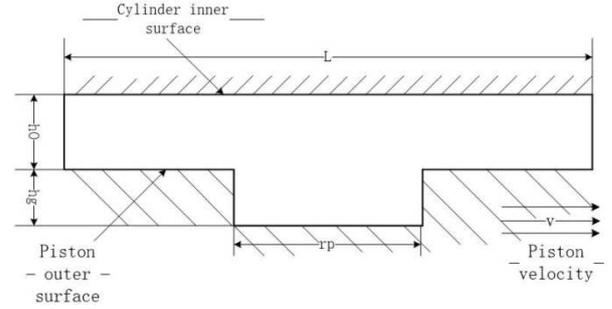


Fig. 4 Cross-section model of flow field for the control unit with cylindrical texture

$$\tau(x, y) = \frac{2}{\pi E} \int_0^{L_y} \int_0^{L_x} \frac{p(s, z)}{\sqrt{(x-s)^2 + (y-z)^2}} ds dz \quad (7)$$

where $\tau(x, y)$ is the contact deformation, E is the equivalent elastic modulus, $p(s, z)$ is the oil film pressure, L_x is the length of control unit along the x-axis, and L_y is the length of control unit along the y-axis.

3.3 Friction loss calculation

The friction loss power is determined by the friction force and the velocity of the impact piston. The equation for calculating the friction loss energy of the piston pair can be expressed by :

$$W_F = \int_0^T P_{F_t} dt = \int_0^T F_{f_t} v_t dt, \quad (8)$$

where W_F is the friction loss of the impact piston pair in one cycle, P_{F_t} is the friction power of impact piston pair at time t , v_t is the relative velocity of the impact piston and cylinder block at time t input according to experimental test data, F_{f_t} is the impact piston friction at time t , and T is the motion cycle of the hydraulic impactor with high frequency.

Assuming that the friction produced by each texture unit is the same, the friction force to be applied to the impact piston at time t can be expressed as:

$$F_{f_t} = K F_{f_t}, \quad (9)$$

where F_{f_t} is the impact piston friction applied to the impact

piston at time t , F_{fdt} is the friction produced by a single texture unit at time t , K is the total number of texture units, $K = \frac{\pi d \ell}{L_x L_y}$, d is the diameter of the impact piston, ℓ is the runner length, L_x is the length of control unit along the x-axis, and L_y is the length of control unit along the y-axis.

The shearing force of oil causes the friction force of dynamic pressure oil film on a solid surface. According to Newton's internal friction theorem, the friction force of a single texture unit can be obtained by integrating the shear stress in the contact fluid layer along the control element.

$$F_{fd} = \int_0^{L_x} \int_0^{L_y} \left(\frac{h}{2} \frac{\partial p}{\partial x} + \frac{v\eta}{h} \right) dx dy, \quad (10)$$

where F_{fd} is the friction force of single texture unit of impact piston, h is the oil film thickness, p is the oil film pressure, v is the relative velocity of the impact piston and cylinder block, η is the dynamic viscosity of hydraulic oil, L_x is the length of the control unit along the x-axis, and L_y is the length of the control unit along the y-axis.

3.4 Oil film pressure calculation for the textured impact piston pair

Based on the previous assumptions and the formation mechanism of oil film pressure, the Reynolds equation of incompressible fluid under isothermal condition is written as [18]:

$$\frac{\partial}{\partial x} \left(\rho h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\rho h^3 \frac{\partial p}{\partial y} \right) = 6\eta \frac{\partial}{\partial x} (v\rho h), \quad (11)$$

where ρ is the density of hydraulic oil, h is the oil film thickness, p is the oil film pressure, η is the dynamic viscosity of hydraulic oil, and v is the relative velocity of the impact piston and cylinder block.

The maximum working pressure is only 14 MPa in the working condition of the high frequency hydraulic impact piston pair used in the hydraulic impactor. The effect of pressure on the density and viscosity of the hydraulic oil is very small under this range. Therefore, the change of oil density and viscosity is ignored and Eq. (11) can be simplified as:

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(h^3 \frac{\partial p}{\partial y} \right) = \Lambda \frac{\partial h}{\partial x}, \quad (12)$$

where $\Lambda = 6v\eta_0$ is the simplified coefficient, η_0 is the initial dynamic viscosity of hydraulic oil, and v is the relative velocity of the impact piston and cylinder block.

There is a certain height gradient on the textured surface, which contains the convergence wedge and divergence wedge. Assuming that the oil film pressure is periodically distributed, the Reynolds cavitations boundary condition in fluid lubrication can be written as:

$$p(x, y) \geq 0, \text{ and when } p(x, y) = 0, \frac{dp}{dx} = 0 \quad (13)$$

Assuming that the pressure on both sides of the impact piston changes linearly, the single texture unit has a certain pressure drop, and the oil film boundary conditions can be obtained as follows:

$$\Delta p = \frac{P_i - P_0}{M - 1}, p(0, y) = p_0 + \Delta p, p(L_x, y) = p_0 \quad (14)$$

where, Δp is the pressure difference between two ends of the impact piston, p_i is the supply oil pressure, p_0 is the return oil pressure, and M is the number of nodes of the texture element along the x-axis.

In addition, the oil film pressure distribution should satisfy the load balance equation, as shown in Eq. (15):

$$F_L - \int_0^{L_x} \int_0^{L_y} p(x, y) dx dy = 0, \quad (15)$$

where F_L is the equivalent load, according to the boundary conditions $F_L = \frac{P_i + P_0}{2(M-1)} L_x L_y$, p_i is the supply oil pressure, p_0 is the return oil pressure, M is the number of nodes of the texture element along the x-axis, L_x is the length of control unit along the x-axis, and L_y is the length of the control unit along the y-axis.

The oil film pressure is solved with boundary conditions according to Eq. (12). If the load balance equation is satisfied, the pressure distribution value of the texture unit can be obtained.

4. Numerical solution of energy consumption model for textured impact piston pair with high frequency

After determining the size of the texture unit, the length of the texture unit along the x-axis is divided into M nodes, and the length of the texture unit along the y-axis is divided into N nodes. According to Eq. (2), Eq. (3), and Eq. (8), the calculated step size is taken as Δt . The energy consumption in a cycle can be solved numerically by the following equation:

$$W = \sum_{i=1}^{i=I+1} \Delta p_i Q_i \Delta t + \sum_{i=1}^{i=I+1} F_{fi} v_i \Delta t, \quad (16)$$

where W is the energy consumption of the impact piston pair in one cycle, I is the number of periods in a cycle $I = T/\Delta t$, T is the motion cycle of the hydraulic impactor with high frequency, Δt is the calculated step size, Δp_i is the pressure difference between two ends of the impact piston at time $i \times \Delta t$, Q_i is the leakage flow of hydraulic oil at time $i \times \Delta t$, F_{fi} is the impact piston friction applied to the impact piston at time $i \times \Delta t$, and v_i is the velocity of impact piston at time $i \times \Delta t$.

Combining Eq. (9) and Eq. (10), the friction force can be solved by using the composite trapezoid formula,

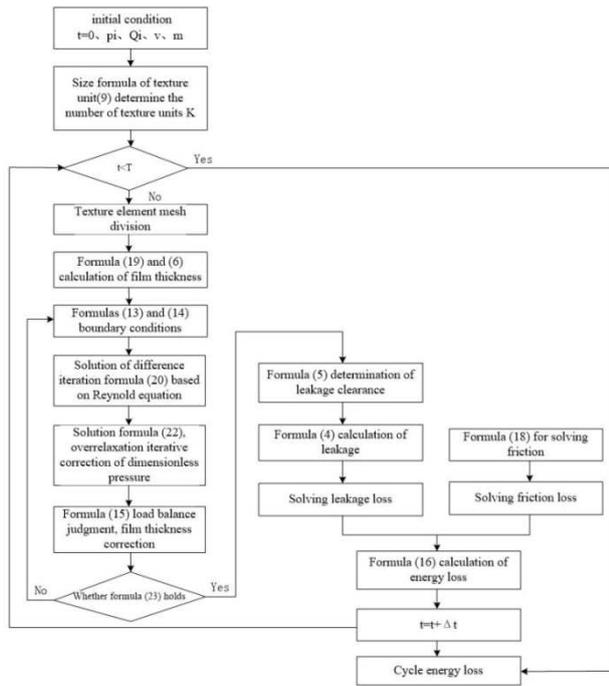


Fig. 5 Flow chart for solving energy consumption loss

5. Energy consumption analysis of the textured impact piston pair with high frequency

5.1 Analysis parameter determination of an impact piston pair with high frequency

The impact piston pair of the YG45 hydraulic impactor is taken as the analysis object. The working pressure and flow rate of the hydraulic impactor under rated working condition are constantly controlled by the system. The maximum impact velocity is obtained by strain method measurement using the test platform built by the excavator bucket arm and hydraulic oil source of excavator. The experiment is shown in **Fig. 6**. In the experiment, the supply oil pressure is set to 14 MPa, and the supply oil flow is set to 60 L/min. The test period of multi-channel high-speed waveform recorder is set to 50 ms, and the data acquisition frequency is set to 1 MHz. The amplitude-frequency characteristic curve of the stress measured after the hydraulic impactor working stably is shown in **Fig. 7**. The maximum impact velocity of piston is 12.9 m/s, which was calculated by the stress wave theory according to the test stress in **Fig. 7**. The impact frequency is 74.8 Hz, which was acquired from the amplitude-frequency characteristic curve of the stress. In other words, the motion cycle of hydraulic impactor with high frequency is 0.013 s, with the specific parameters of the test system shown in Table 1. According to the hydraulic-mechanical coupling model of the hydraulic impactor under rated working condition [1], the velocity curve of the impact piston with high frequency under rated working condition is shown in **Fig. 8**. **Fig. 8** shows that

the piston stroke and return motion need a certain amount of time to reach the steady state, but they still show a periodic change. The reason for this is that the front and rear chamber pressures have not been established in the initial operation. Therefore, the velocity of the impact piston can be taken as the velocity value of the impact piston in any period after stable operation. The pressure and flow parameters of the energy consumption analysis model can be set as the same values of the adjusted system. Specific parameters are as follows: the maximum motion velocity of the piston is 12.9 m/s, the oil supply pressure is 14 MPa, the oil return pressure is 0.4 MPa, and the motion cycle of hydraulic impactor is 0.013 s.

Table 1 Specific parameters of the test system

Supply pressure	14 MPa
Ambient pressure	0.101 MPa
System flow	60 L/min
Test period of waveform recorder	50 ms
Data acquisition frequency	1 MHz
Maximum velocity of impact piston	12.9 m/s
Impacting frequency	74.8 Hz
Motion cycle of impact piston	0.013 s



Fig. 6 Testing system of the YG45 hydraulic impactor

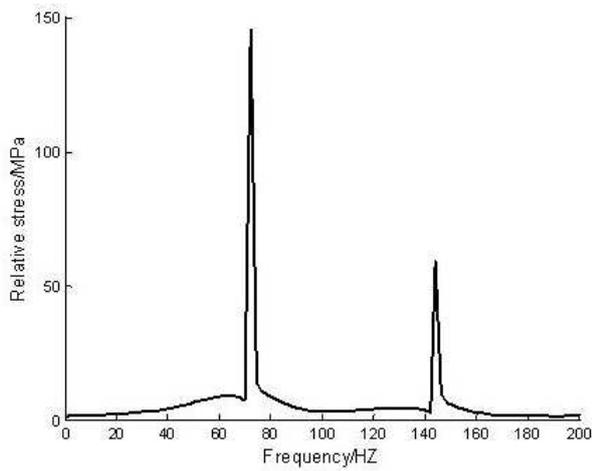


Fig. 7 Amplitude-frequency characteristic curve of stress

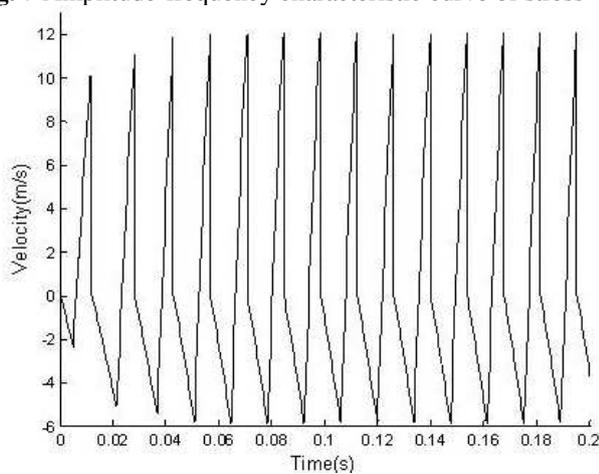


Fig. 8 Velocity curve of the impact piston under rated working condition

5.2 Energy consumption analysis of the texture impact piston pair

A cylindrical texture unit is arranged on the surface of the impact piston. The length of the control unit along the x-axis L_x is 1.5 mm, and the length of the control unit along the y-axis L_y is 1.5 mm. The area ratio textured is 30%, the dynamic viscosity of the hydraulic oil used is $0.06 P_a \cdot s$, the initial oil film thickness is $27.5 \mu m$, and the depth of cylindrical texture is $27.5 \mu m$. The energy consumption model of the impact piston pair is calculated using MATLAB. The oil film thickness and pressure drop of the first element near the front cavity at maximum velocity in return travel are shown in **Fig. 9** and **Fig. 10**, respectively. Through **Fig. 9** and **Fig. 10**, it can be observed that the cylindrical texture surface can produce hydrodynamic pressure and form hydrodynamic pressure lubrication film. The hydrodynamic lubrication effect of texture under full oil film lubrication will significantly improve the bearing capacity of the surface, and it will have better dynamic pressure lubrication performance than the non-textured surface. In addition, the area where the

pressure peak is located is on the right side of the opening area. This is due to the fact that the left side of the divergent wedge and the right side of the convergence wedge having lower pressure during the movement of the textured piston pair. Therefore, the pressure reaches the maximum on the right side of the opening region.

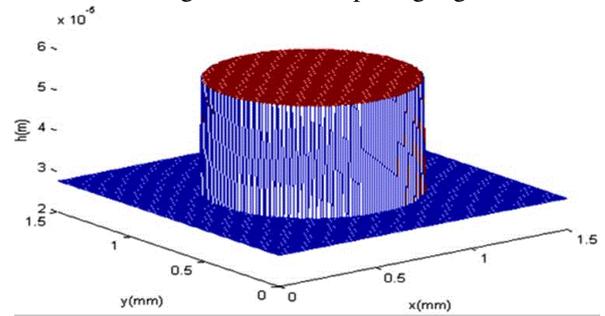


Fig. 9 Oil film thickness on textured surface

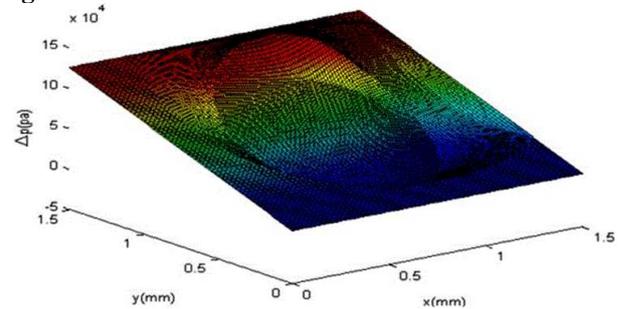


Fig. 10 Pressure drop of the first element near the front cavity at maximum backhaul velocity

The comparison of energy consumption between the non-textured impact piston pair and cylindrical textured impact piston pair using calculation results is shown in **Fig. 11**. The results show that the energy consumption of the cylindrical textured impact piston pair is 14.63 J in one cycle under the above parameters. Compared with the non-textured impact piston pair, the ratio of energy consumption to total energy loss is reduced by 5.88%. The surface friction force greatly reduces with the cylindrical texture, and the reduction of surface friction greatly improves the friction performance of the surface and significantly reduces friction loss. In addition, the leakage loss slightly increases with the cylindrical texture. However, the increased leakage loss can be ignored compared with the reduced friction loss. The cylindrical texture noticeably reduces the energy consumption of the impact piston in general. The energy consumption of the impact piston pair is 18.24 J and the ratio of energy consumption to total energy loss of the hydraulic impactor is 29.77%. Therefore, reducing the energy consumption of the impact piston pair can significantly improve the energy utilization rate of hydraulic impactor.

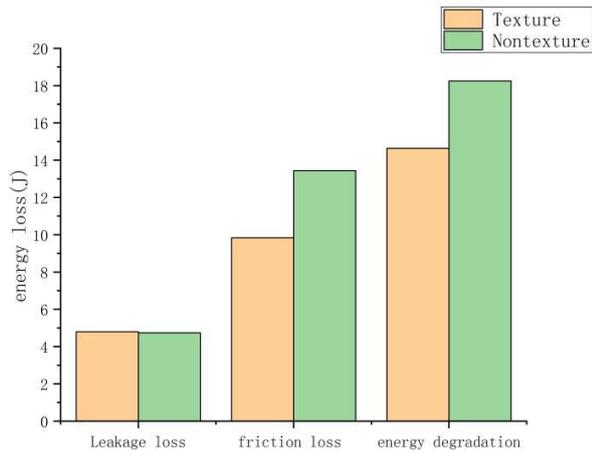


Fig. 11 Comparison of energy consumption between the non-textured impact piston pair and cylindrical textured impact piston pair

5.3 Effects of area ratio textured to energy consumption of the impact piston

The control unit is square. Limited by the shape of the control unit, the maximum area ratio textured using cylindrical texture can only reach 78.5%. The thickness of oil film is set to 27.5 μm , and the depth of cylindrical texture is set to 27.5 μm . The energy consumption and energy consumption ratio to total energy loss of the impact piston pair in one cycle with different area ratios textured are shown in Table 1. The energy consumption of the non-textured impact piston pair is 18.24 J. When the cylindrical texture is used, the energy consumption is reduced regardless of the area ratio textured, which is caused by the decline of friction loss. The energy consumption of the impact piston pair with different area ratios textured is shown in Fig. 10. As can be seen from the figure, the energy consumption of the impact piston with cylindrical texture is smaller than that without texture. The energy consumption of the impact piston first decreases and then remains unchanged with the increase of area ratio textured. The energy consumption ratio of the impact piston pair to total energy loss with different area ratios textured is shown in Fig. 11. It can be seen that the energy consumption ratio decreased by 1.32% to 10.98% with the area ratio textured ranging from 0.1 to 0.78. The decrease in energy consumption ratio reveals a trend of first accelerating and then slowing down. The energy consumption ratio reaches the minimum before the area ratio textured achieves the maximum, and the energy consumption ratio can be reduced to the lowest when the area ratio textured is in the range of 0.64 to 0.70.

Table 2 Energy consumption and energy consumption ratio of the impact piston pair in one cycle with different area ratios textured

Area ratio textured	0	0.1	0.2	0.25	0.3	0.35	0.4	0.5	0.6
Energy consumption/J	18.2	17.4	16.2	15.4	14.6	13.3	12.6	11.9	11.6
Energy consumption ratio/%	100	95.6	89.6	84.6	80.2	73.6	69.2	65.3	63.7
Area ratio textured	0	0	0	0	0	0	0	0	0
Energy consumption/J	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Energy consumption ratio/%	8.2	8.4	6.4	5.2	3.8	1.8	0.6	9.4	8.9
Area ratio textured	0	0	0	0	0	0	0	0	0
Energy consumption/J	6	4	2	1	1	1	1	1	1
Energy consumption ratio/%	33	22	11	5.5	5.5	5.5	5.5	5.5	5.5

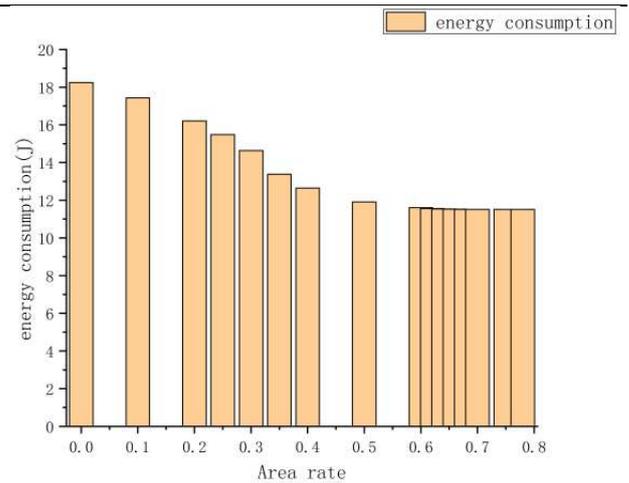


Fig. 12 Energy consumption of the impact piston pair with different area ratios textured

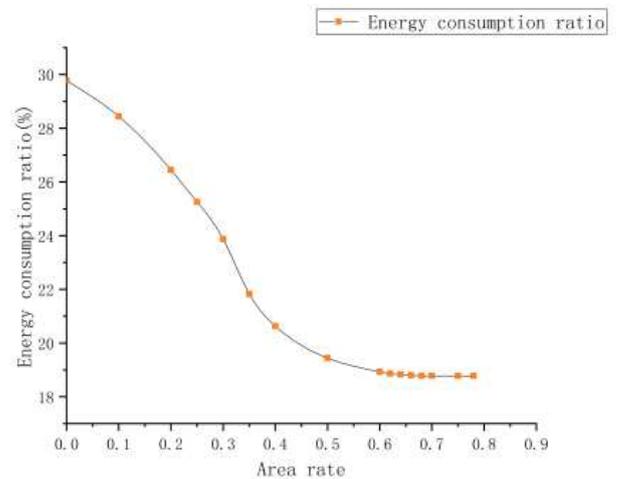


Fig. 13 Energy consumption ratio of the impact piston pair with different area ratios textured

5.4 Effects of depth ratio textured to the energy consumption of impact piston

As shown in Fig. 4, depth ratio textured is defined as the ratio of the height of the cylindrical texture element to the minimum initial clearance between the cylinder block and impact piston. The thickness of oil film is set to 27.5 μm. The energy consumption and energy consumption ratio to total energy loss of the impact piston pair in one cycle with different depth ratios textured under area ratios textured ranging from 0.3 to 0.5 are shown in Tables 2–4. When the cylindrical texture is used, the energy consumption is reduced regardless of the depth ratio textured under a certain area ratio textured. Energy consumption of the impact piston pair with different depth ratios textured under varying area ratios textured is shown in Fig. 12. As can be seen from the figure, the energy consumption of impact piston first decreases and then increases with the increase of depth ratio textured. Energy consumption ratio of the impact piston pair with different depth ratios textured under varying area ratios textured is shown in Fig. 13. It can be seen that energy consumption ratio decreased by 3.21% to 5.68% with the depth ratio textured ranging from 0.3 to 1.3, and the energy consumption ratio can be reduced to the lowest when the depth ratio textured is in the range of 1 to 1.1.

Table 3 Energy consumption and energy consumption ratio of the impact piston pair in one cycle with different depth ratios textured when the area ratio textured is 0.3

Depth ratio textured	0.3	0.5	0.7	0.9	1.1	1.3
Energy consumption /J	6.27	5.54	5.07	4.76	4.80	5.12
Energy consumption ratio/%	2	2	2	2	2	2
	6.55	5.36	4.59	4.09	4.15	4.67

Table 4 Energy consumption and energy consumption ratio of the impact piston pair in one cycle with different depth ratios textured when the area ratio textured is 0.4

Depth ratio textured	0.3	0.5	0.7	0.9	1.1	1.3
Energy consumption /J	1	1	1	1	1	1
Energy consumption ratio/%	2	2	2	1	1	2
	4.97	2.85	1.25	9.97	9.55	0.69

consumption /J	5.59	4.45	3.59	2.93	2.39	3.15
Energy consumption ratio/%	2	2	2	2	2	2
	5.44	3.58	2.18	1.10	0.22	1.46

Table 5 Energy consumption and energy consumption ratio of the impact piston pair in one cycle with different depth ratios textured when the area ratio textured is 0.5

Depth ratio textured	0.3	0.5	0.7	0.9	1.1	1.3
Energy consumption /J	5.30	4.00	3.02	2.24	1.98	2.68
Energy consumption ratio/%	2	2	2	1	1	2
	4.97	2.85	1.25	9.97	9.55	0.69

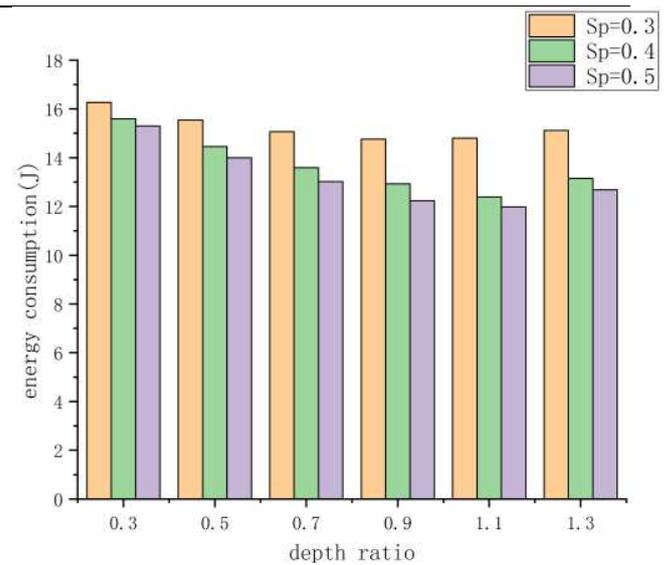


Fig. 14 Energy consumption of the impact piston pair with different depth ratios textured under varying area ratios textured

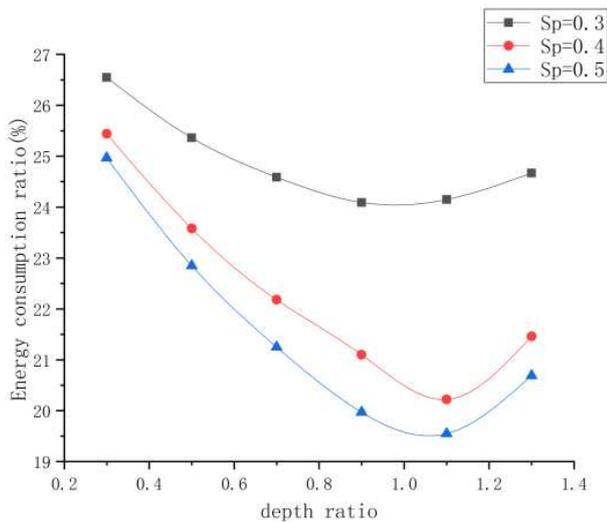


Fig. 15 Energy consumption ratio of the impact piston pair with different depth ratios textured under varying area ratios textured

The addition of a cylindrical microtexture can improve lubrication performance, reduce energy consumption, and increase the local oil film thickness. With the increase of oil film thickness, the shear rate of oil decreases, leading to the decrease of friction and the energy consumption of the piston pair. The above analysis also shows that different area ratios and depth ratios of texture have an influence on the reduction of energy consumption. Minimizing the energy consumption of the piston pair can be achieved by changing the area ratio and depth ratio of the cylindrical microtexture, which improves the energy utilization rate of the hydraulic impactor.

6 Conclusions

According to the characteristics of reciprocating periodic motion with high frequency, the energy consumption analysis model of the textured piston pair is established. The model is used to analyze the energy consumption of the impact piston pair of the YG45 hydraulic impactor. The results show that:

(1) The energy consumption of the impact piston pair accounts for 29.77% of the total energy loss of the hydraulic impactor. It is an effective way to improve the energy utilization rate of the hydraulic impactor by reducing the energy consumption of the impact piston pair.

(2) Energy consumption of the impact piston pair with cylindrical texture is reduced regardless of the area ratio textured. Energy consumption ratio decreased by 1.32% to 10.98% with the area ratio textured ranging from 0.1 to 0.78, and energy consumption ratio can be reduced to the lowest when the area ratio textured is in the range of

0.64 to 0.70.

(3) Energy consumption of the impact piston pair with cylindrical texture is also reduced regardless of the depth ratio textured, though the value is only slightly reduced. The energy consumption ratio decreased by 3.21% to 5.68% with the depth ratio textured ranging from 0.3 to 1.3, and the energy consumption ratio can be reduced to its lowest when the depth ratio textured is in the range of 1 to 1.1.

7 Declaration

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Availability of data and materials

The datasets supporting the conclusions of this article are included within the article.

Authors' contributions

The author' contributions are as follows: Yong Guo and Yanping Hu was in charge of the whole trial; Hu Mo and Song Quan wrote the manuscript; Zhewu Chen assisted with sampling and laboratory analyses.

Competing interests

The authors declare no competing financial interests.

Consent for publication

Not applicable

Ethics approval and consent to participate

Not applicable

Reference:

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Figures

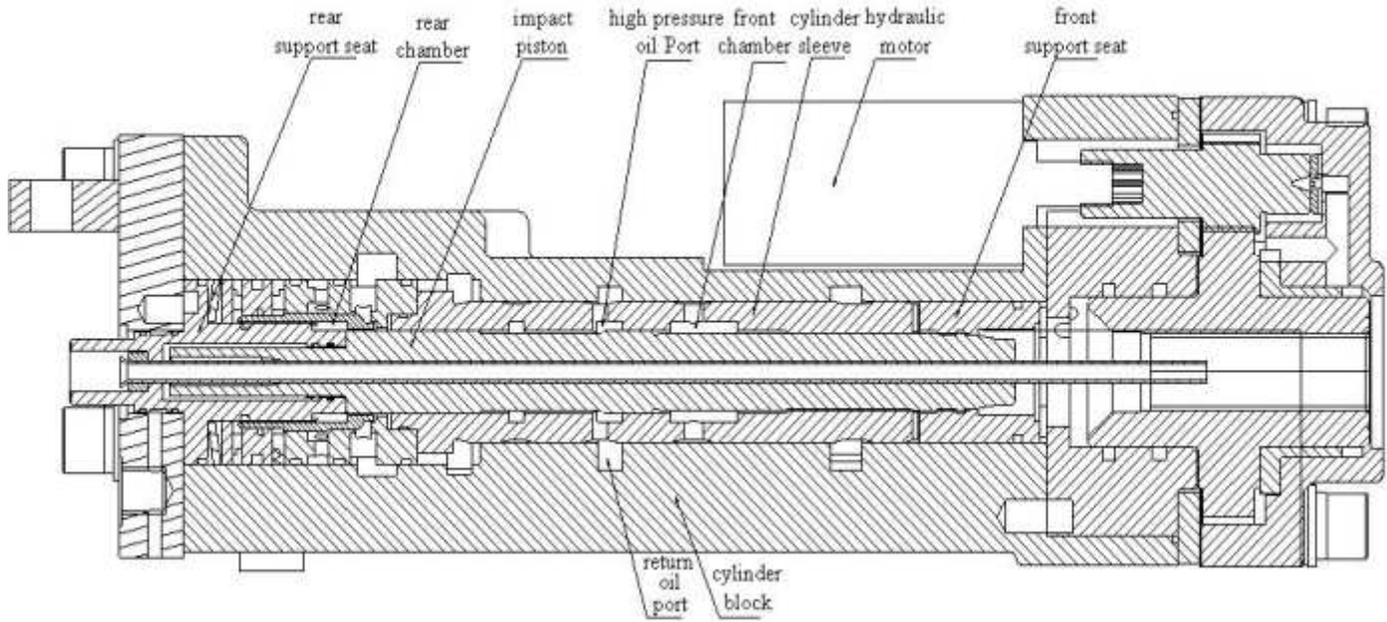


Figure 1

Impact mechanism of the YG45 hydraulic rock drill

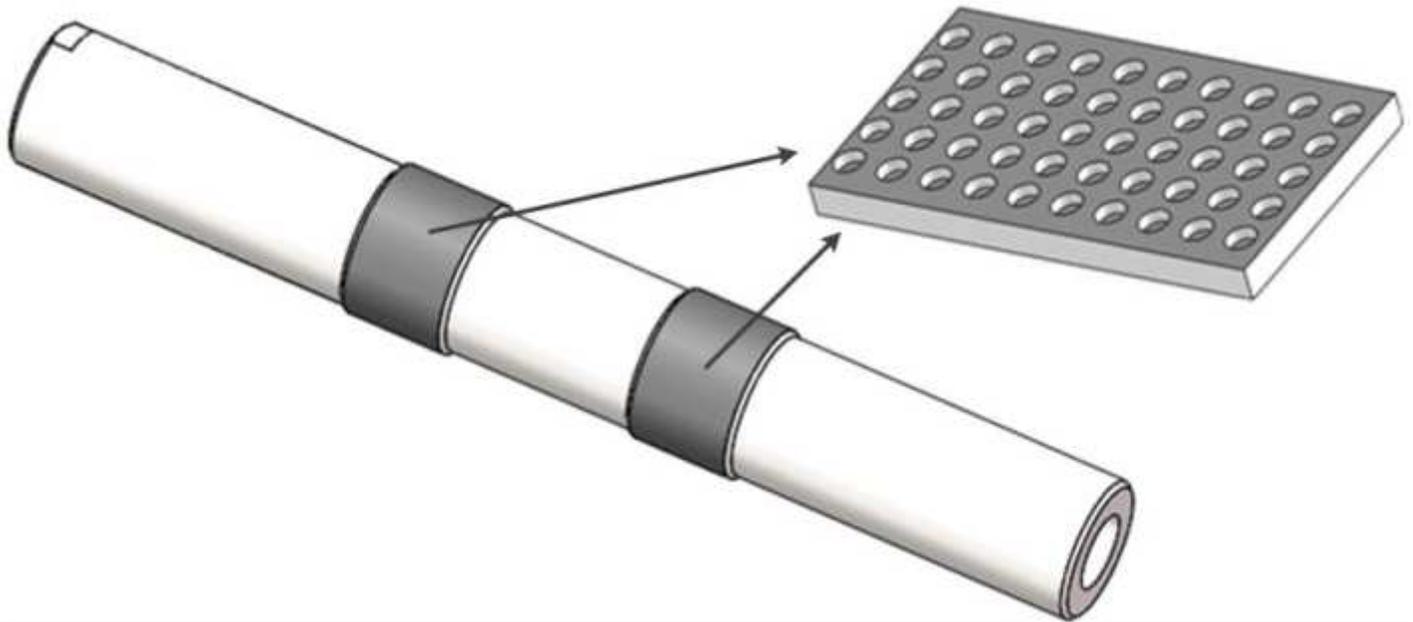


Figure 2

Cylindrical texture geometric model of the impact piston surface

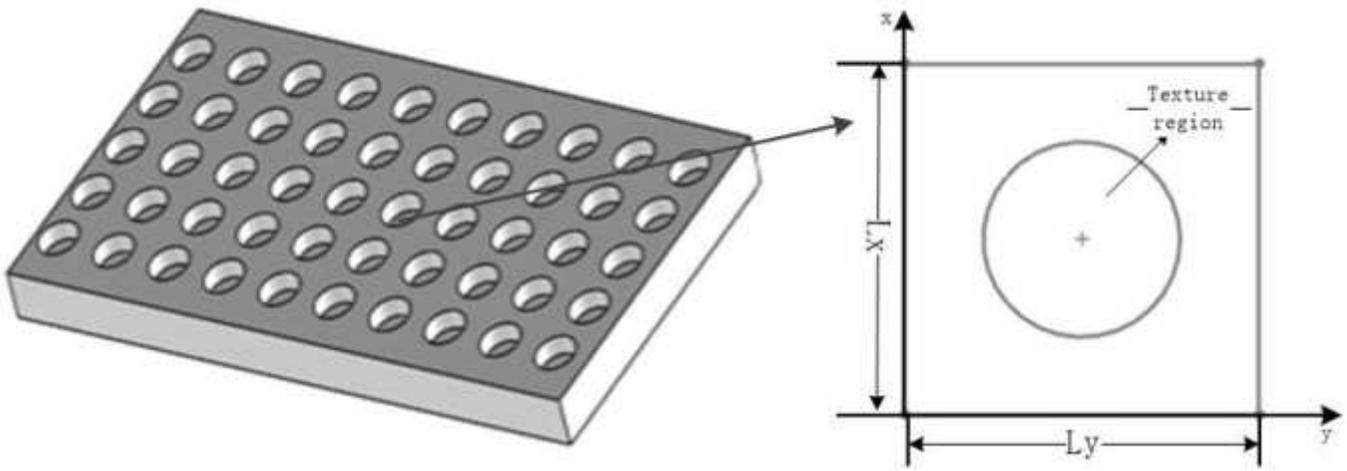


Figure 3

Geometric model of the texture control unit

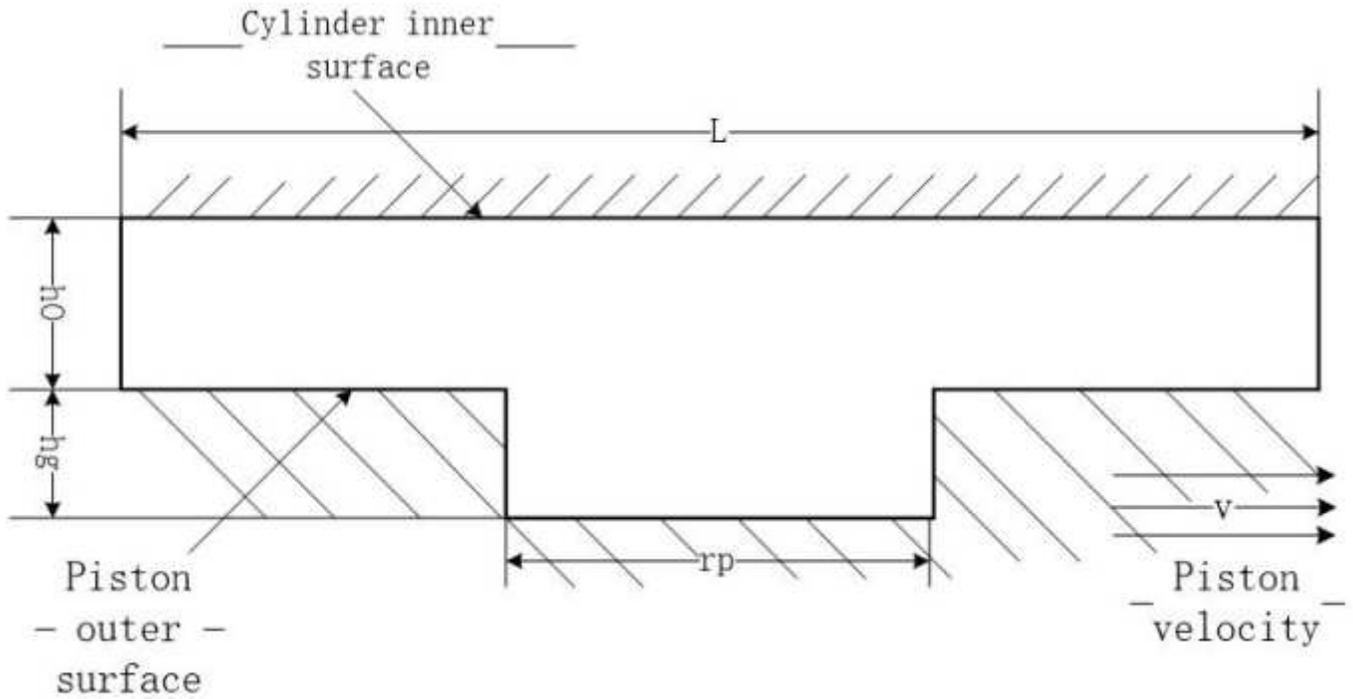


Figure 4

Cross-section model of flow field for the control unit with cylindrical texture

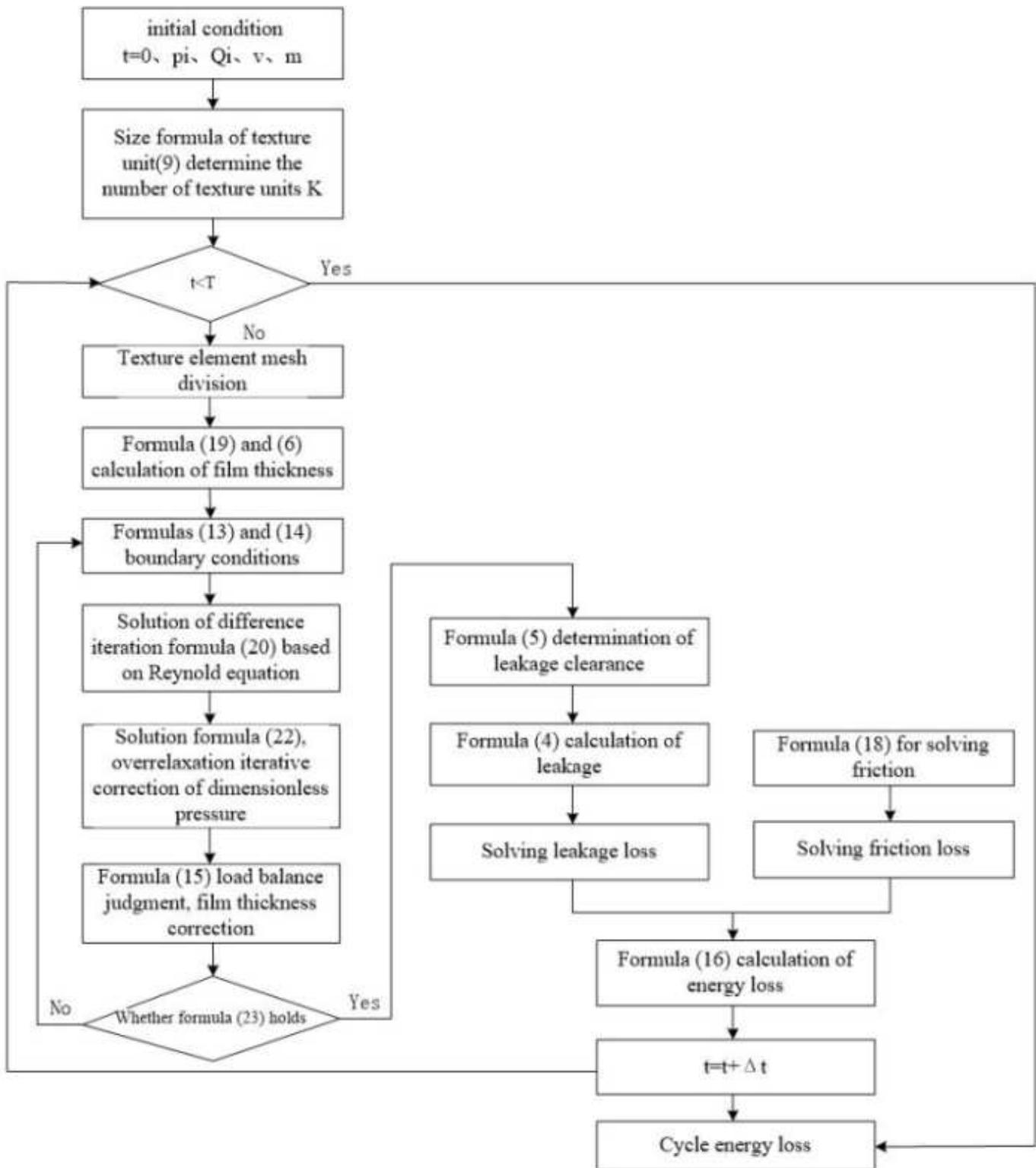


Figure 5

Flow chart for solving energy consumption loss

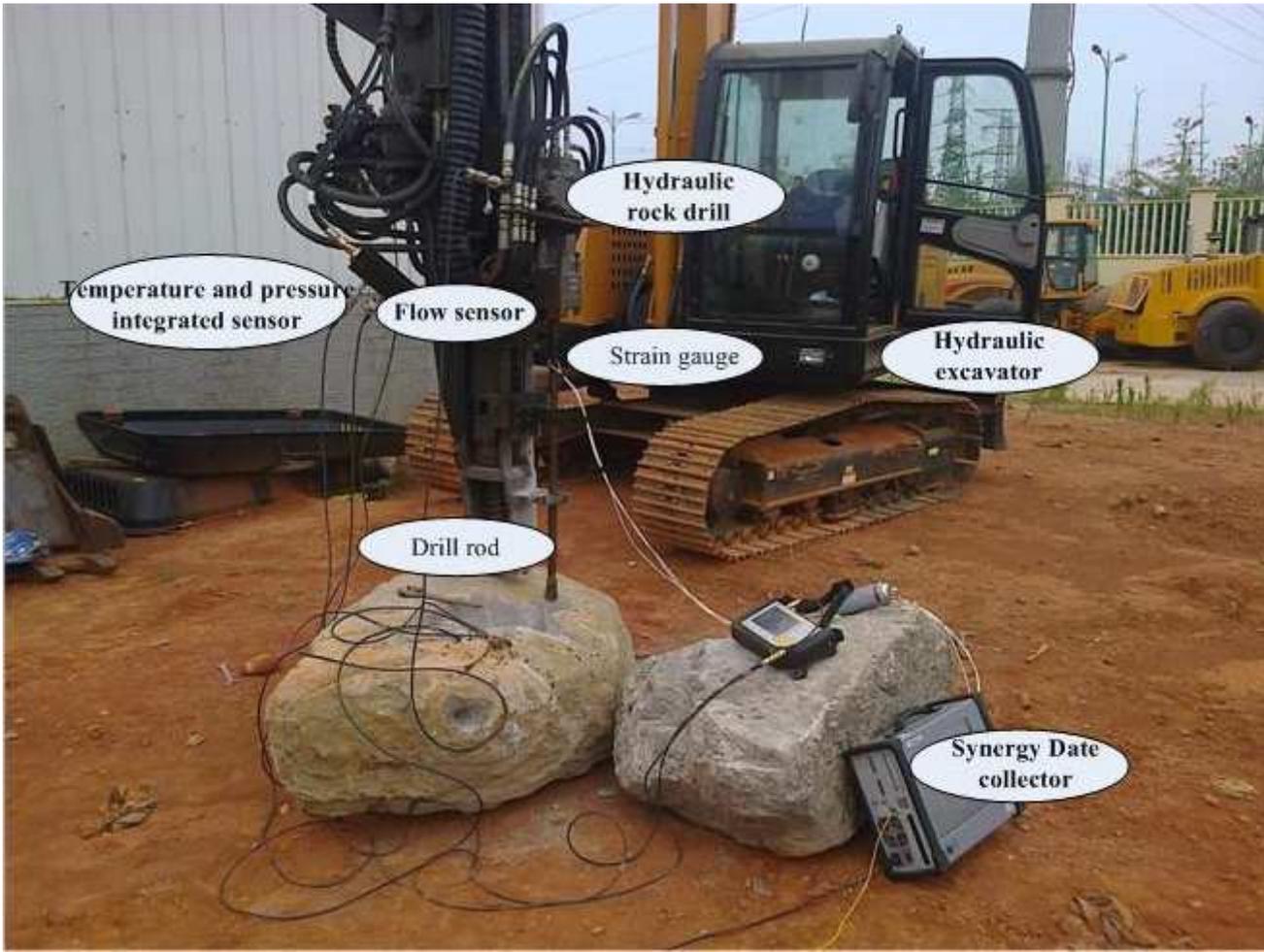


Figure 6

Testing system of the YG45 hydraulic impactor

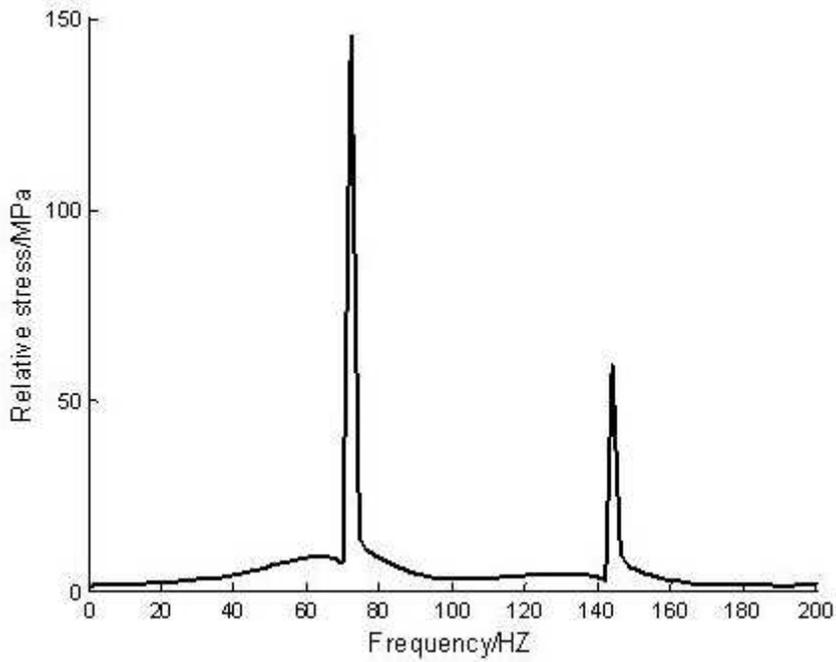


Figure 7

Amplitude-frequency characteristic curve of stress

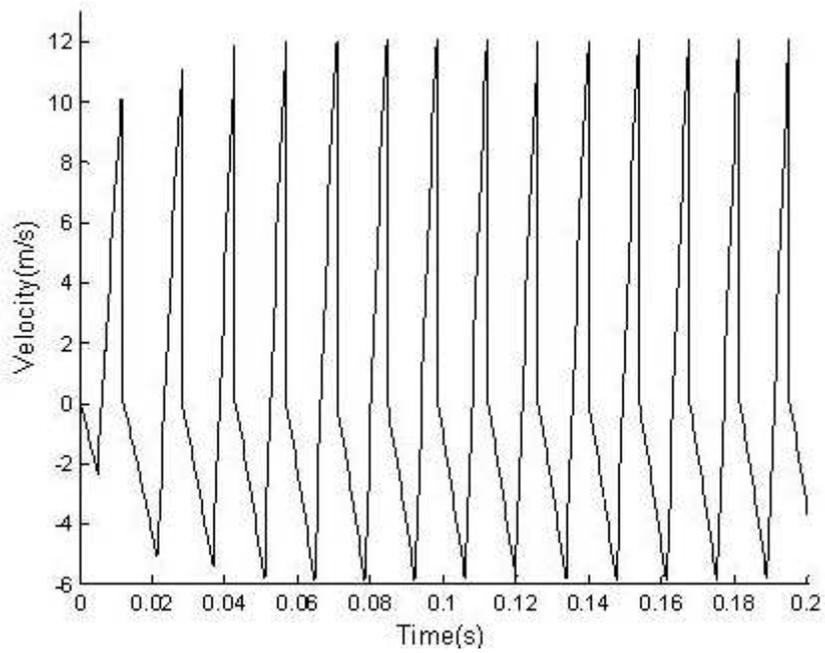


Figure 8

Velocity curve of the impact piston under rated working condition

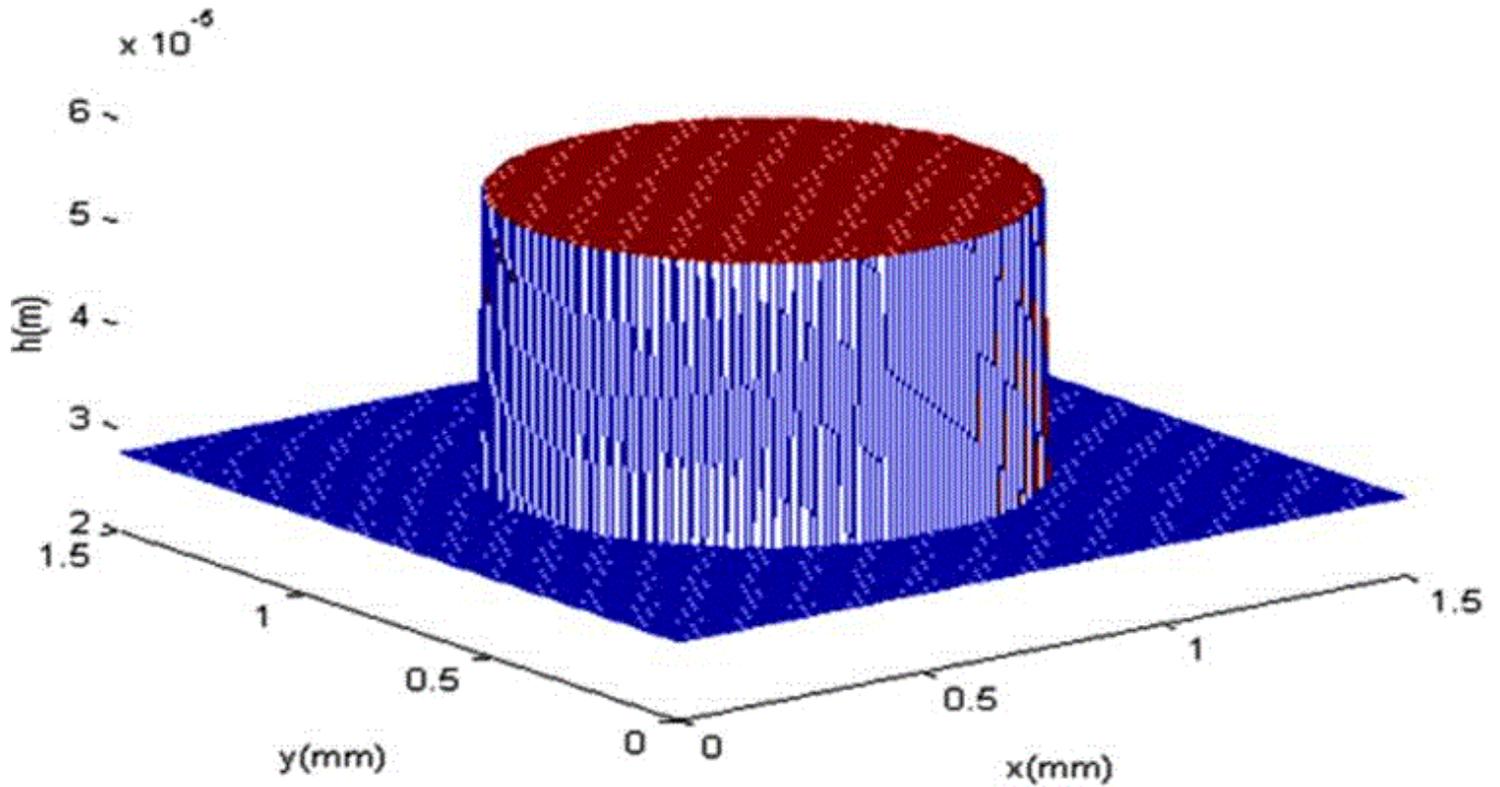


Figure 9

Oil film thickness on textured surface

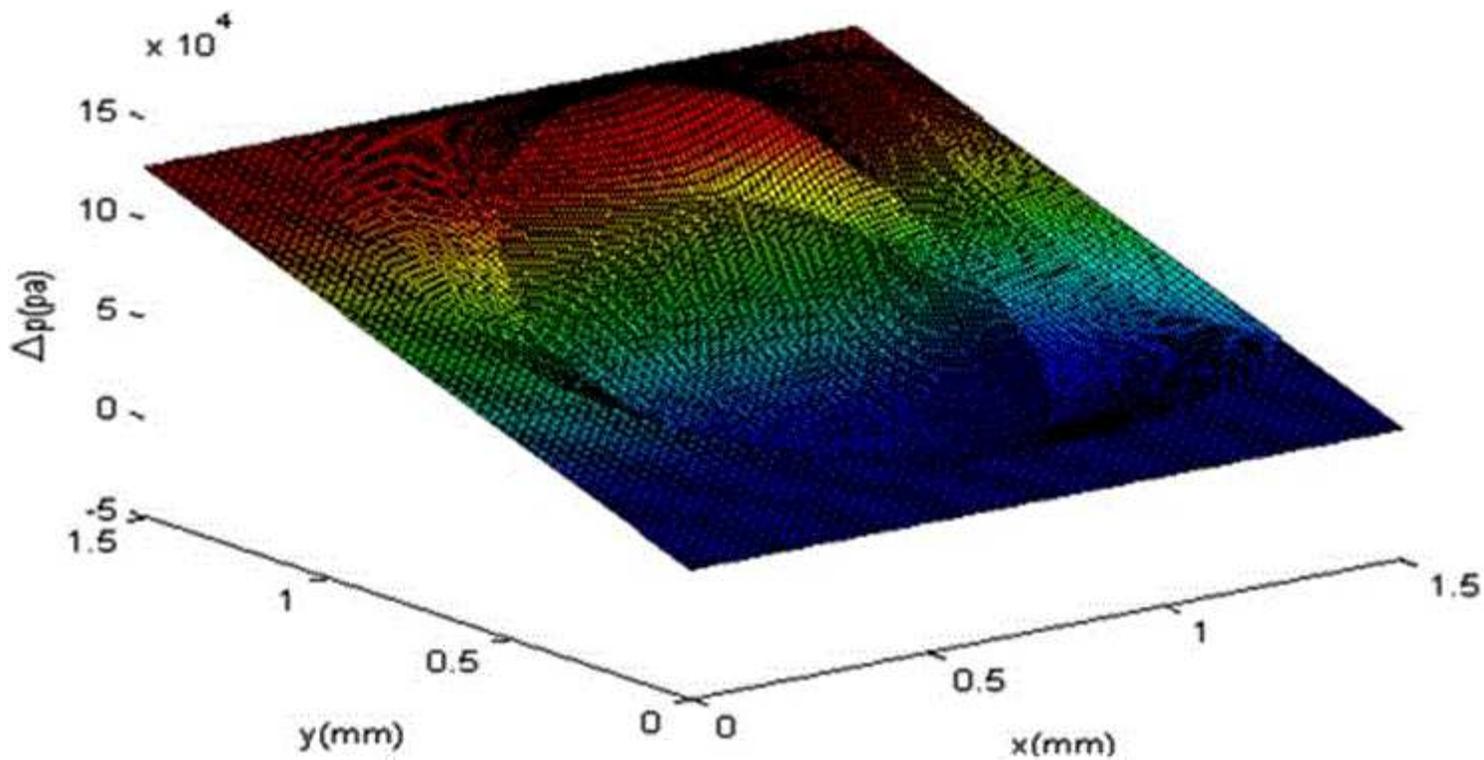


Figure 10

Pressure drop of the first element near the front cavity at maximum backhaul velocity

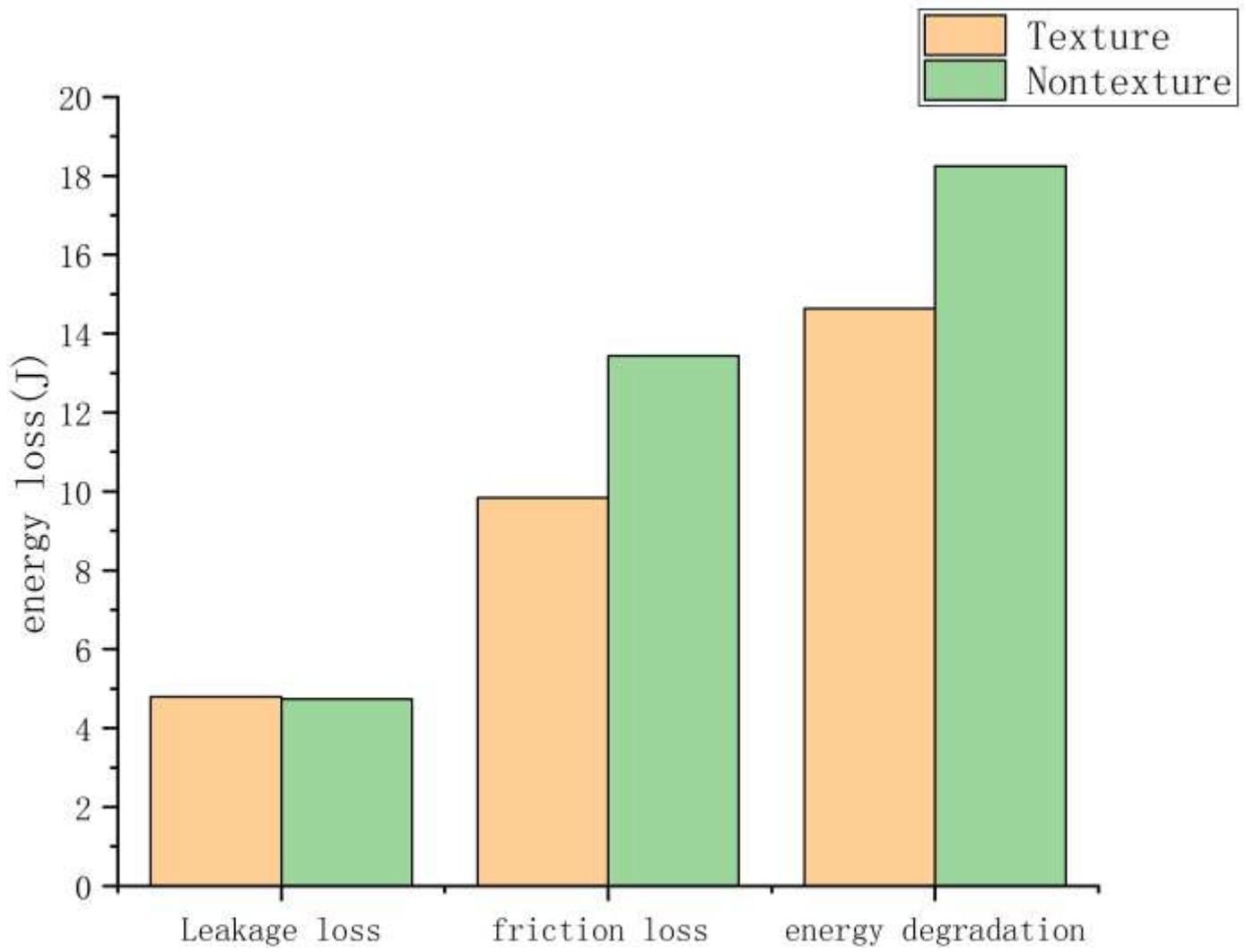


Figure 11

Comparison of energy consumption between the non-textured impact piston pair and cylindrical textured impact piston pair

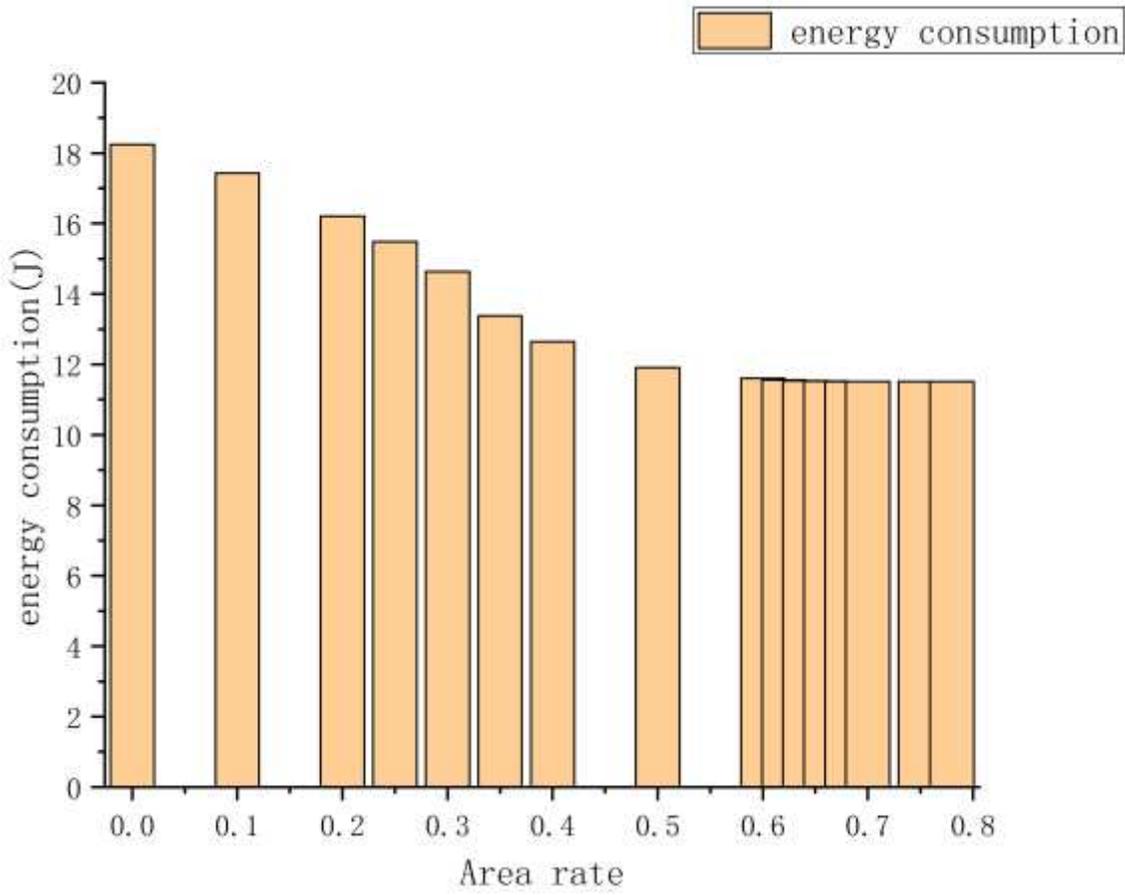


Figure 12

Energy consumption of the impact piston pair with different area ratios textured

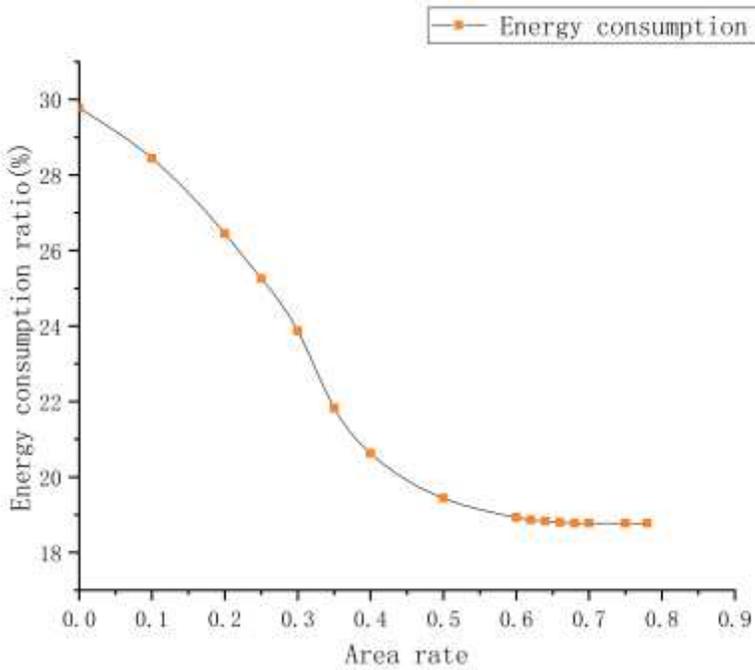


Figure 13

Energy consumption ratio of the impact piston pair with different area ratios textured

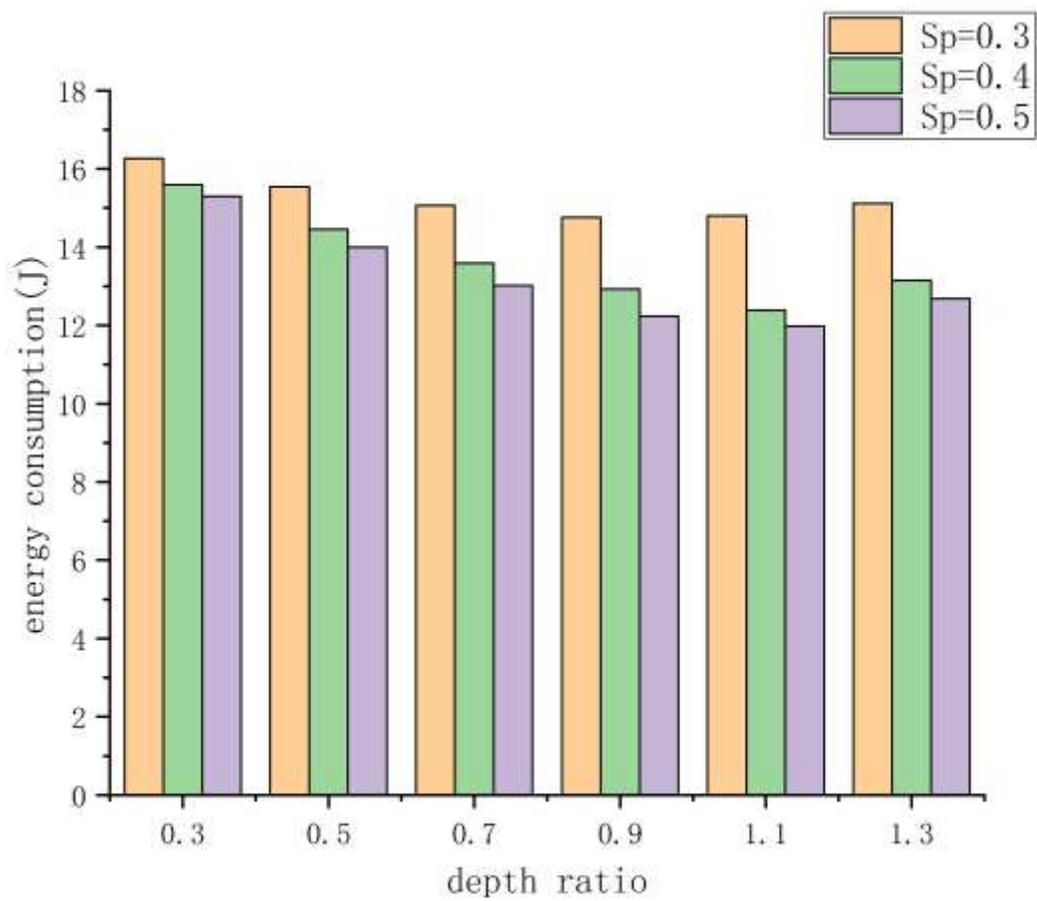


Figure 14

Energy consumption of the impact piston pair with different depth ratios textured under varying area ratios textured

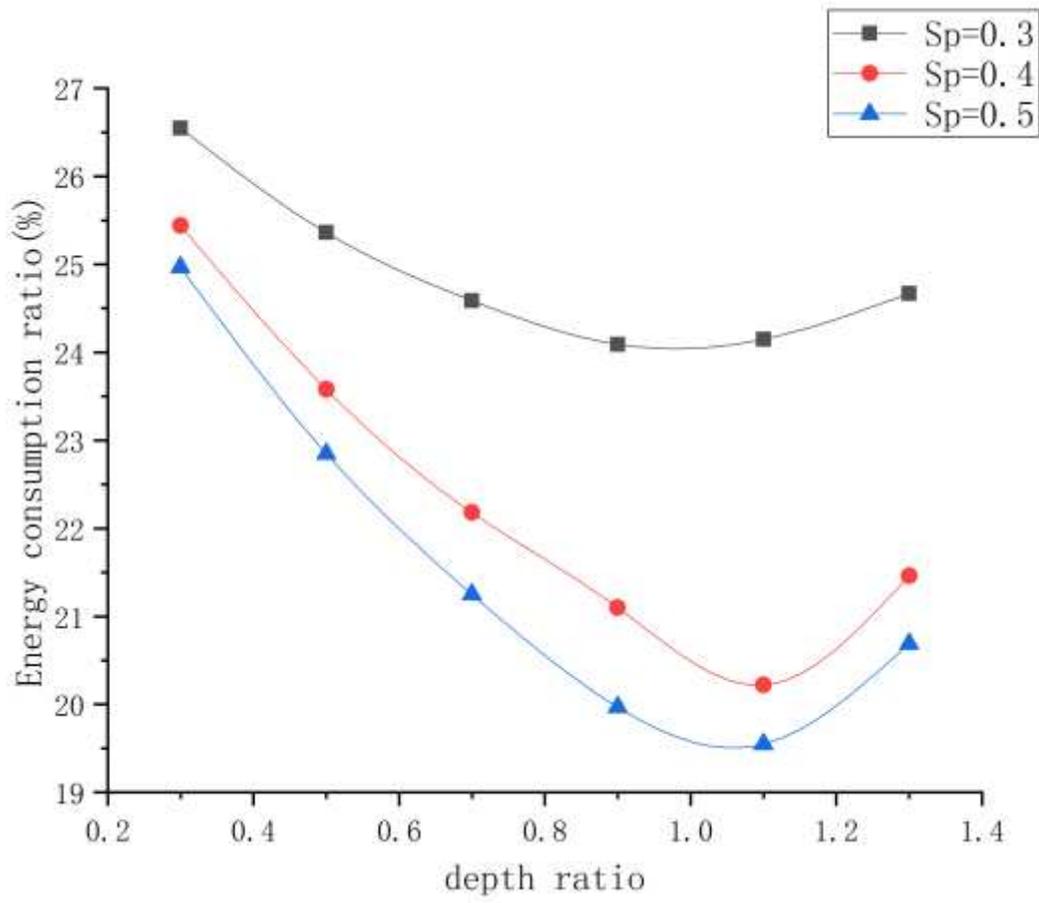


Figure 15

Energy consumption ratio of the impact piston pair with different depth ratios textured under varying area ratios textured