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ABSTRACT

The working characteristic of scroll oil pump and the property of medium – – oil are described in this paper, the leakage of oil is analysed, and the mathematical model is set up for scroll oil pump. The effects of the discharge port on the pressure of the working pocket, the oil displacement and the required power are obtained according to the results of calculation and experiment. The optimal design criteria of the discharge port are presented.

INTRODUCTION

The oil pump is universal machine which is widely used in every area, such as in petrochemical industry, transportation field and so on. The energy consumption of the oil pump accounts a large proportion in overall value and the ratio is increasing year by year. Under the energy shortage situation, it is a main research territory to enhance the pump efficiency. One of the industry pollutions is noise, which has a serious influence on the daily life and work environment, so the demand for low noise pump becomes more and more strong as time went on. The pump used in the hydraulic pressure system should have the good stability, the fluctuations of the flow and pressure are both little. The reliability target is of great importance for the pump used in chemical process. In order to satisfy the needs of efficiency, noise, fluctuation, reliability and others, a new type pump—scroll oil pump is developed. Since the early days of 80's, the scroll compressor used for air–conditioner has been put into commercial production, it has a lot of special features, such as high efficiency, low noise and vibration, smooth operation, high reliability. These qualities have shown the inherent advantage of scroll machinery, and the scope of its application is extended continuously, it is not only used as refrigerating compressor, but also used as air compressor and vacuum pump. Scroll oil pump is a new field of application of this technique. The basic research is carried out, the prototype which scroll profile is a series of semi–circles has been developed in our laboratory, the operating datahave shown that the prototype performance is suprior to the reciprocating or centrifugal pump's in some respects.

The structure of scroll oil pump is similar to the other scroll machinery's. The principle of scroll oil pump operation shown in Fig. 1, the suction and discharge processes are nearly continuous, so the fluctuations of the flow and pressure are little and the variation of the driving moment is also little, the pump operates smoothly, the noise and vibration are low accordingly. Scroll pump is a kind of the positive displacement pump, it has the property of the compulsive discharge, so it is not necessary to install the suction or discharge valve, so the flow loss is less. Because the eccentric value of crankshaft is small, so the velocity of the orbiting scroll is low, and its friction loss is little, so the power consumption is less, the pump efficiency is high. The structure of scroll pump is simple, it has a few parts and no damageable one, therefore, the weight and size of the pump are small, the service life is long and the reliability is high.

Because the compressibility of the wroking medium——oil is very little, the scroll profile of the pump is different from the compressor's. The pump has only a pair of the working pockets, when two scroll mesh each other and form the sealed pockets, the number of the scroll profile rotation is 1.5 theoretically. The area of the discharge port is small at the initial discharge stage, because the viscosity factor of the oil is great, the effective flow area of the port becomes smaller, but the variation of the pocket volume is fast, if the oil is not exhausted promptly at the discharge process, the overhigh pressure is produced in the working pocket, this leads to the power increase and the efficiency decrease. The scroll wrap would be damaged in serious case. So the port design is of great importance for the scroll oil pump.

WORKING PROCESS MODEL

The main difference between the scroll pump and the compressor is that the pump has no obvious compression process. As shown in Fig. 1, the oil is inhaled into the crescent shaped pockets, when the crankshaft rotates a circle, the inside suction ports close, the oil is sealed in the pockets, then the inside discharge ports begin to open, the pockets connect with the high-pressure side, at last the oil is extruded from the pockets. The ports of the pump are distributed as shown in Fig. 2, the outside suction and discharge ports are both connect with the pipes, the axial inside suction and discharge ports are both located at the root of the scroll wraps, the tangential inside suction ports are located at the outside end of the scroll wrap, the tangential inside discharge ports are at the inside end.

The medium——oil density varies with the temperature, and so does its viscosity.

density equation $\rho=\rho_{20}+\alpha(t-20)$

viscosity equation $\lg\lg(\nu+0.6)=A-B\lg(t+273)$

where ρ_{20} is the density at temperature 20℃, α is the coefficient of temperature, t is the oil temperature, ν is the oil kinematic viscosity, A and B are the oil constants. The oil compressibility is very little, but when the oil contains air, the oil elastic modulus decreases greatly, accorting to the reference [1], the voluminal elastic modulus of the oil containing air is as follows:

$$k = k_f \frac{V_f / V_g + P_a / P}{V_f / V_g + k_f \cdot \frac{P_a}{P^2}}$$

where k_f is the oil elastic modulus, V_f and V_g is the volumes of oil and air respectively, P_o is the atmospheric pressure, P is the oil pressure.

Because there are too many factors influencing the pump working process, in order to simplify the calculation, the following assumptions are made.

(1) The medium is treated as the compressible viscous liquid.

(2) The medium state in working pocket is homogeneous, the flows through the ports and the clearance are steady.

(3) The heat exchange is not taken into account, the gravitational and kinematic energies of the medium are neglected.

In light of the laws of Conservation of Energy and Mass, the following fundamental equations are obtained

$$\frac{dp}{d\theta} = \frac{-k}{V(\theta)}\left[\frac{dV(\theta)}{d\theta} - \frac{1}{\rho}\ \frac{dm}{d\theta}\right]$$

$$\frac{dm}{d\theta} = \frac{dm_{in}}{d\theta} - \frac{dm_{out}}{d\theta}$$

$$P(0) = P_a$$

$$m(0) = 0$$

where $V(\theta)$ is the volume of the working pockets, it is given in reference [2], m is the medium mass, m_{in} and m_{out} are the masses flowing into the pockets and out of the pockets respectively. By use of this analysis model, the simulative calculation is carried out, the medium state parameters are obtained, the indicated power and the displacement of the pump are presented. Because the high–pressure area of the pump is large, the axial force is great, so the double acting constructure is adopted in the prototype, the axial force acting on the orbiting scroll is balanced by itself, the four–bar linkage mechanism is adopted to prevent the orbiting scroll from rotation, so only the bearings cause the friction loss in the pump.

FLOW THROUGH THE PORTS

In order to decrease the overhigh pressure, the axial inside discharge port is offered, but the axial port is restricted by the strength of the scroll, it is not prefect only to add the axial port. It is adopted in our research to cut short the scroll wrap located at the centre, so as to increase the flow area of the tangential inside discharge ports. If the wrap is cut too short, the pump will not work properly.

Flow through the Tangential Inside Discharge Port

As shown in Fig. 3, the tangential inside discharge port is long and narrow at the initial discharge stage. Because the oil viscosity is great, the flow through this port is treated as the varibale height, one–dimensional, pressure–difference flow, according to Navier—Stokes equation and the theory in reference [3], the flow is given as fallaws:

$$Q = -\frac{H\ h^3(\theta)}{12\ \mu R_1}\ \frac{dp}{d\varphi}$$

where H is the height of the scorll wrap, R_1 is the profile parameter, μ represents the dynamis viscosity. integrated as

$$Q = \frac{e^3\ H\triangle P}{3\mu R_1\left[f_1\left(\frac{\theta_1}{2}\right) - f_1\left(\frac{\theta}{2}\right)\right]}$$

where $f_1\ (\varphi) = \frac{1}{5}ctg\varphi \cdot csc^4\varphi + \frac{4}{15}ctg\varphi \cdot csc^2\varphi + \frac{8}{15}ctg\varphi$, θ_1 is obtained from the equation: $h(\theta_1\ ) = 3h_0$, The mass flow rate $\dot{m}_1\ = \rho Q$

Flow through the Axial Inside Discharge Port

The axial inside discharge port is open after the pocket is sealed, its flow condition is the same as that of slide valve, so the mass flow rate through this port is

$$\dot{m}_2(\theta) = Cd \cdot S(\theta)\sqrt{2\triangle P \cdot \rho}$$

where the coefficient Cd is determined by the port area $s(\theta)$ and others.

LEAKAGE MODEL

Leakage is the main element that influences the pump efficiency. Radial and tangential leakages are two basic leakages.

Radial Leakage

Because of the meshing clearance δ_a between the scroll wrap and end plate, there exists the radial leakage path. This is pressure–difference flow, as the viscosity of oil is great, it is calculated as the parallel layer flow, the leakage mass rate is given by

$$\dot{m}_3 = l(\theta)\left(\frac{\delta_a^3\triangle P}{12\nu t_a} + \frac{\delta_a U}{2}\rho\right)$$

where $l(\theta)$ is the leakage length, t_a is the wrap thickness, U is the orbiting scroll velocity.

Tangential Leakage

There exists the tangential leakage because of the meshing clearance δ_m between the wraps, as shown in Fig. 4, the leakage path is simplified as the parabolic cylinder with radius R_1 and the parabolic cylinder with radius R_2 , when their centre distance is $(r-\delta_m\ )$, they appears clearance, according to the reference [4], the mass flow rate is

$$\dot{m}_4 = \frac{H\delta_m^3 \triangle P}{10.02\mu\sqrt{\delta_m \frac{R_1 R_2}{R_2 - R_1}}}$$

where R_1 and R_2 are the wrap parameters.

SIMULATIVE CALCULATION

The property parameters of the medium are computed by solving the differential equations of scroll pump working process, then the performance parameter of the machine is obtained, the diagram of computer program is shown in Fig. 5, the geometrial parameters of the scroll wrap、 the operating condition、 the medium property and the like are the input data, At first, the suction and discharge processes are both regarded as the constant pressure ones, the property parameters in working pocket are calculated and is regarded as the initial data of next cycle, the differential equations are solved using Rung—Kutta method, then the parameters of the medium property and the pump performance are obtained, the flow rate and power are regarded as the convergent conditions.

EXPERIMENT

The prototype is used to set up the experimental installation. The experiments of the orbiting speed、 discharge pressure are all carried out on the pump so as to compare with the calculation results. The working medium of the experiment is 30[#] mechanical oil, the elemental design parameters of the prototype are as follows:

Discharge pressure:	0.6MP$_a$
orbiting speed:	575 rpm
flow rate:	20M^3 / h
orbiting radius:	10mm
wrap height of each side:	25mm

In order to study the influence of the discharge port on pump performance, the experiments on different working conditions are carried out using the method of shortening the scroll wrap to change the discharge prot.

RESULT ANALYSIS

The simulative calculation results are coincident with the experiment's, as shown in Fig. 6 ~ Fig. 8, this proves the correctnesses of the theoretical model and the computer program. Fig. 6 shows the change of the flow rate and power with the discharge pressures on the conditions of constant orbiting speed (700 rpm) and none shortened wrap ($\nu = 0$). It is obvious that the flow rate reduces when the discharge pressure rises, but the change is small, this shows that the flow rate is determined mainly by orbiting speed, the changing tendency of the power is proximately linear. Fig. 7 shows the change of flow rate and power with orbiting speed on the conditions of constant discharge pressure (0.6MP$_a$) and none shortened wrap ($\nu = 0$). Within this range, flow rate and power increase in directly proportion to the speed. Fig. 8 shows the change of flow rate and power with the shortened angle ν on the condition of constant orbiting speed (700 rpm) and discharge pressure (0.6MP$_a$). Within the range of $\nu = [0, 40°]$, the flow rate reduces , the power reduces greatly, when ν equals 45°, and the power increases, the main reason is that within $\nu = [0, 40°]$, the overhigh pressure in working pocket reduces with ν increase, as shown in Fig. 9 which is the practical experimental pressure curves in working pocket, its extra loss reduces, when $\nu = 45°$, because of the return flow of high pressure fluid, the pressure pulsation increases, accordingly the power increases. Fig. 10 shows the pressure of working pocket at different orbiting speed on the condition of the constant discharge pressure (0.6MP$_a$) and none shortened wrap ($\nu = 0$), the maximum pressure peak increases when the speed increases. Fig. 11 shows the pressure of working pocket at different discharge pressure when orbiting speed equals 700 rpm and ν is 40°, it is shown that the maximum pressure peak changes slightly, but the extra loss caused by overhigh pressure decreases when the discharge pressure rises.

CONCLUSION

The simulative calculation and experiment are carried out in this paper, and the influences of orbiting speed and super compression loss on pump efficiency are also analysed. The extra loss and pressure pulsation are reduced by changing the length of the scroll wrap rationally.

The orbiting speed determines the position of the tangential inside discharge port and ν mcreases with orbiting speed rising. When dibiting speed is 700 rpm and the discharge pressure is 0.6MP$_a$, the optimum ν is 40° for the prototype. When the discharge pressure is 0.6MP$_a$, the optimum positions of the tangential inside discharge port at different orbitng speed are obtained by calculation, as shown in Fig. 12.

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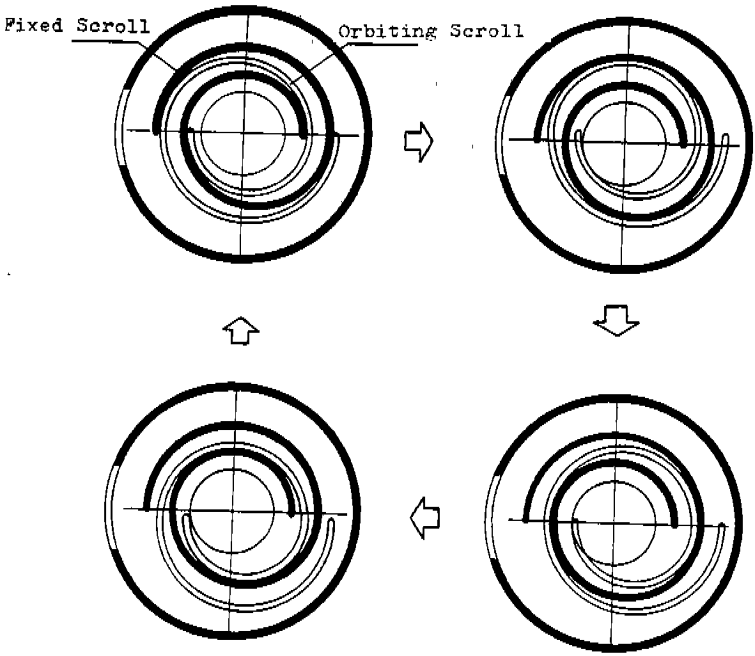

Fig.1 Operation Principle

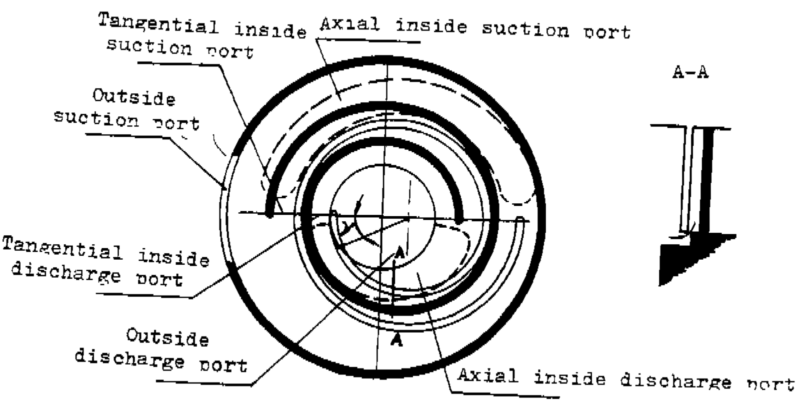

Fig.2 The ports

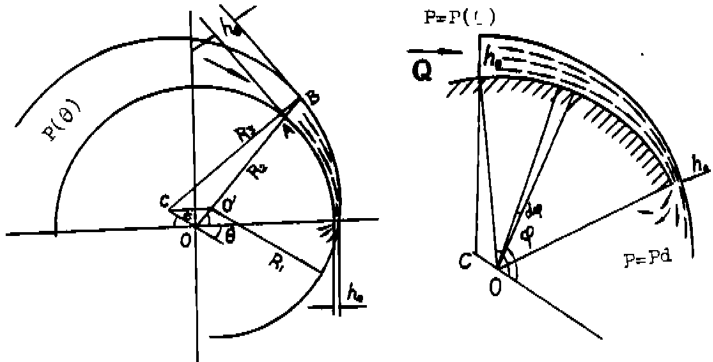

Fig.3 Leakage model

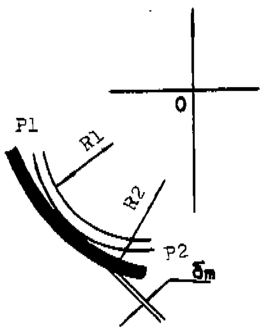

Fig.4 Leakage model

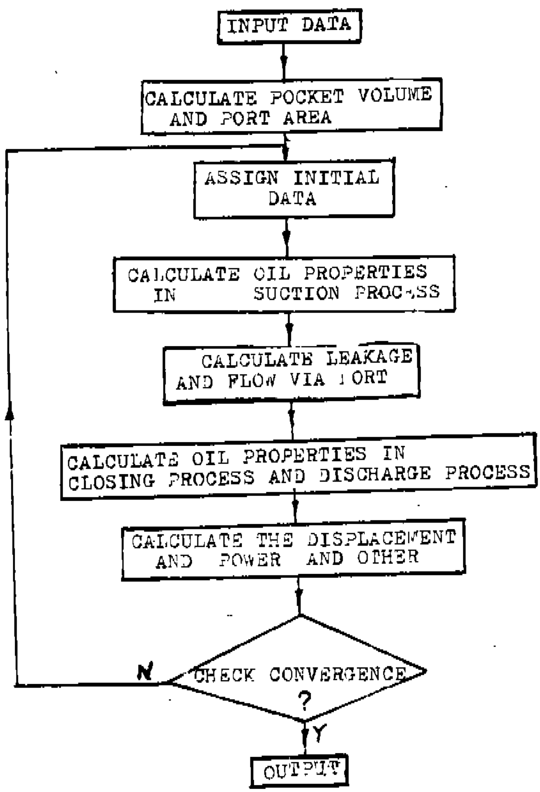

Fig.5 Computer program

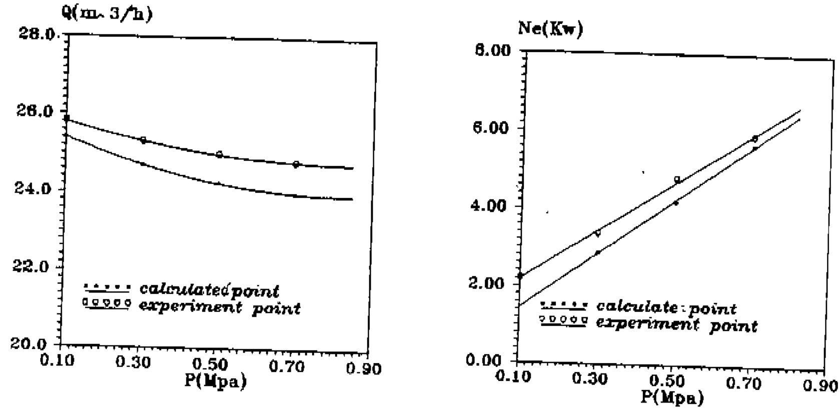

Fig.6 Variations with discharge pressure

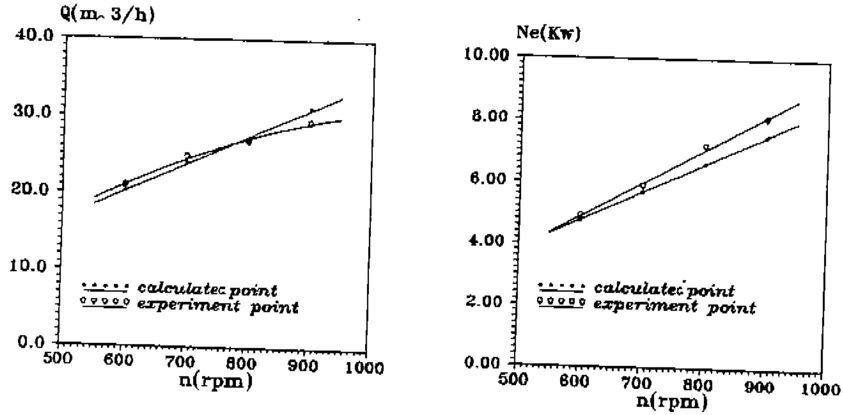

Fig.7 Variations with orbiting speed

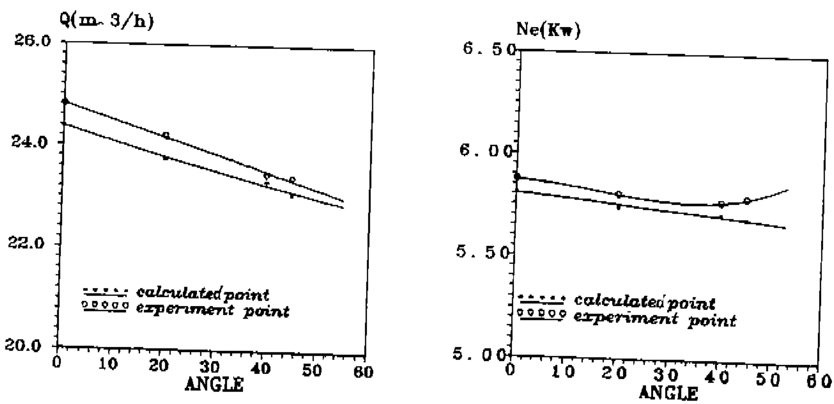

Fig.8 Variation with port angle

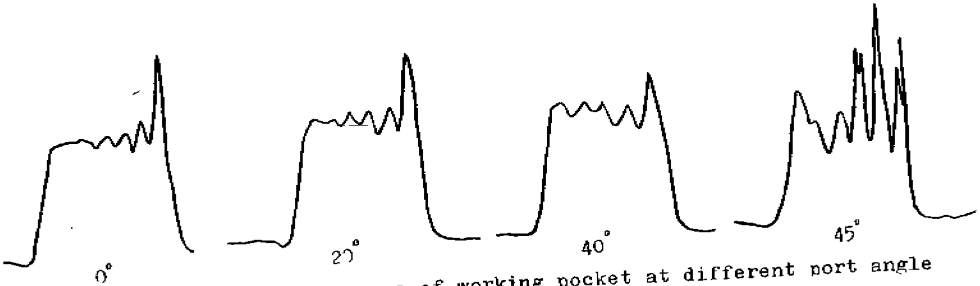

Fig.9 Pressure curves of working pocket at different port angle

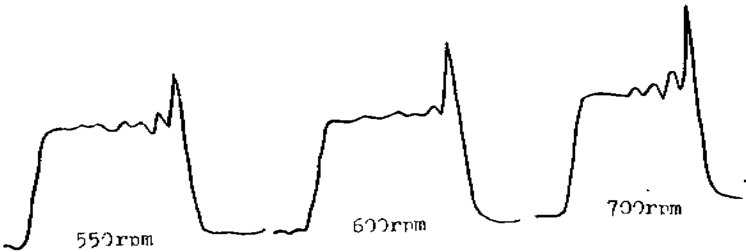

Fig.10 Pressure curves of working pocket at different orbiting speed

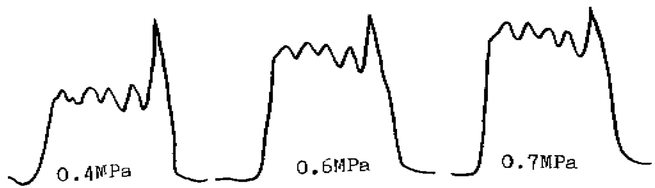

Fig.11 Pressure curves of working pocket at different discharge pressure

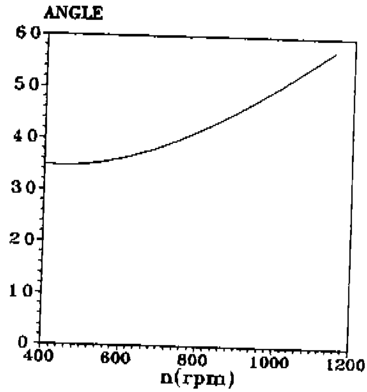

Fig.12 The optimum port angle