Development of a Hollow Axis Swing Cylinder for the Elbow Joint of 7 Function Hydraulic Manipulator

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***Abstract*—For a 7 Function hydraulic manipulator with a 270° rotary elbow joint, the rotary actuator's axis must be hollow, so that an oil slipper ring could be integrated in it, and the 7 oil pipes for the wrist joints and jaw could be rotary connected. All the oil pipes could be hidden into the manipulator's body safely at any rotary angle and any position. Swing cylinders are the only hydraulic actuators which can realize 270° rotary angle and hollow axis at the same time. In this paper, a hollow axis swing cylinder based on double screw pairs is put forward to decrease the surface leakage to 0, so that its rated pressure could be increased to normal level (3000 psi), doubles the manipulator's full extension lift. Meanwhile, an extra axial area compensation ring is designed to decrease the unbalance axial force at high pressure, so that the friction loss, together with its control angle's uncertainty due to the axial force on the bearings could also be decreased. Either analysis or simulations and experiments all show that, the efficiency of hollow axis swing cylinder based on double screw pairs with axial force compensation structure could be as high as 81%. Its application in hydraulic manipulator's elbow joint shows that the build-in-axial oil distribution ring make the manipulator more compact, effect, and efficiency.**

Keywords—Underwater hydraulic manipulator; Manipulator elbow; Swing rotary actuator; Double-screw-pair transmission

I. INTRODUCTION

For a 7 Function hydraulic manipulator with a 270° rotary elbow joint as shown in Fig.1, the rotary actuator's axis must be hollow, so that an oil slipper ring could be integrated in it, and the 7 oil pipes for the wrist joints and jaw could be rotary connected. All the oil pipes could be hidden into the manipulator's body safely at any rotary angle and any position. Swing cylinders are the only hydraulic actuators which can realize 270° rotary angle and hollow axis at the same time. However, the surface leakage and friction of general vane-type swing cylinders are too large and too inefficient to meet the demand of most manipulators. It may cause serious issues for heavy duty hydraulic manipulators. For, the surface leakage and friction may waste too much hydraulic pressure and flow-rate while the elbow joint is swing a little bit faster. Moreover, to meet its low rated pressure, the supplied pressure of the whole manipulator had to be decreased to 2/3, and the full extension lift may decreased to its half.

Fig.1. 7 function manipulator with a 270° rotary elbow joint

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II. DESIGN OF THE HOLLOW AXIS SWING CYLINDER

A. Structure and principle of double screw pairs

The section structure of this hollow axis swing cylinder based on double screw pairs and is shown in Fig.2. It is a pure rotary component, which transfers piston's axial movement into rotary movement via 2 pairs of multiple-heads screws swing. It consists of fixed cylinder 5, shaft axis 3, piston 24, and other auxiliary components. The outer screw pairs are located between the fixed cylinder and the piston, while the inner screw pairs are located between the piston and the shaft axis. There are 2 hydraulic driving chambers at the 2 sides of the piston, which is able to drive the piston moving axially, and rotating simultaneously along the outer screw pairs inside the fixed cylinder. Since the shaft axis can only pure rotate inside the fixed cylinder instead of axial movement, it is also forced to rotate along the inner screw pairs inside the piston, so that the sum of piston's axial movement relative to the fixed cylinder, and the shaft axis' axial movement relative to the piston equals to zero. The shaft axis' axial movement are restricted by its bearings.

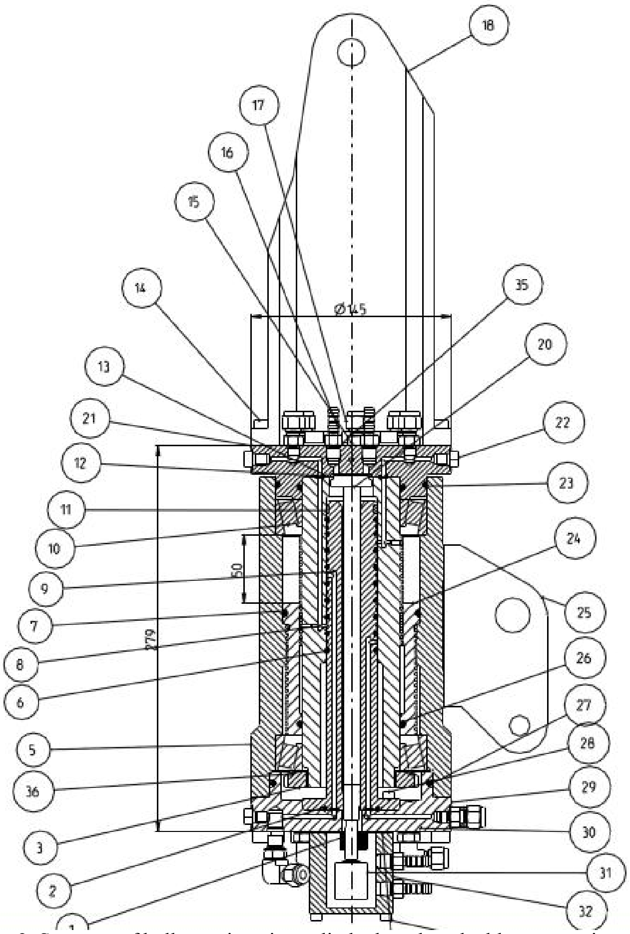

Fig.2. Structure of hollow axis swing cylinder based on double screw pairs

There are 2 kinds configuration for the 2 screw swing pairs: identical direction screw pairs and inverse direction screw pairs. The identical direction one means that the screw direction between inner and outer screw pairs are the same. It has smaller displacement ratio between the piston movement and the output swing angle, and be able to yield more torque. Its output torque could be very large while the pitches/middle-diameters of the 2 identical screw pairs are similar. The reverse direction one is the opposite. The screw direction are opposite. It has larger displacement ratio between the piston movement and the output swing angle, but the output torque is decreased. Although all the 2 kinds of swing cylinders are able to reach large rotary angle such as 270°, 360°, even much larger than 360°, the reverse configuration swing cylinder is more compact, and its torque density is still enough for the elbow joint Therefore, the reverse configuration is the most suitable actuator for the 7 function hydraulic manipulators.

B. Oil distribution and electric wire connection channels

In fig.2, the shaft axis is hollow to install a special rod (part 2) for oil distribution at the elbow joint. It is fixed on the cylinder instead of rotating together with the shaft axis. 2 sliding bearings slots (part 11), 8 O-rings slots (part 6), and 7 oil distribution slots (part 8) are circumferentially arranged on its surface. The 7 oil distribution slots are connected with the fixed inlet oil pipes through the center of the oil distribution part (part 9). The outlet of these 7 oil pipes (part 35) are connected with the other side of the 7 oil distribution slots, the shaft axis. The shaft axis is pure rotating towards this oil distribution rod. Thanks to the circumferentially arranged slot, the oil is able to be connected at any rotary angle.

Fig.3. Hollow axis swing cylinder based on double screw pairs for manipulator's elbow joint, and its parts (shaft axis, cylinder, and piston etc.)

There is another hole (part 20) at the center of the oil distribution rod. This hole allows the electric wire go through the elbow joint without any length adjustment according to its rotary angle. The assembly of this hollow axis swing cylinder' prototype, and its main parts (include its shaft axis, cylinder, piston, and oil distribution rod) are shown in Fig.3.

C. Axial force compensation structure and principle

Since the axial movement of the shaft axis is restricted by bearings, there is a fairly large axial force between the shaft axis and bearings. Besides the heavy friction losses, there is quite a little axial movement. These axial movement leads to control angle's uncertainty directly. For, the axial movement would finally result as a tangential movement according to the inner screw pair.

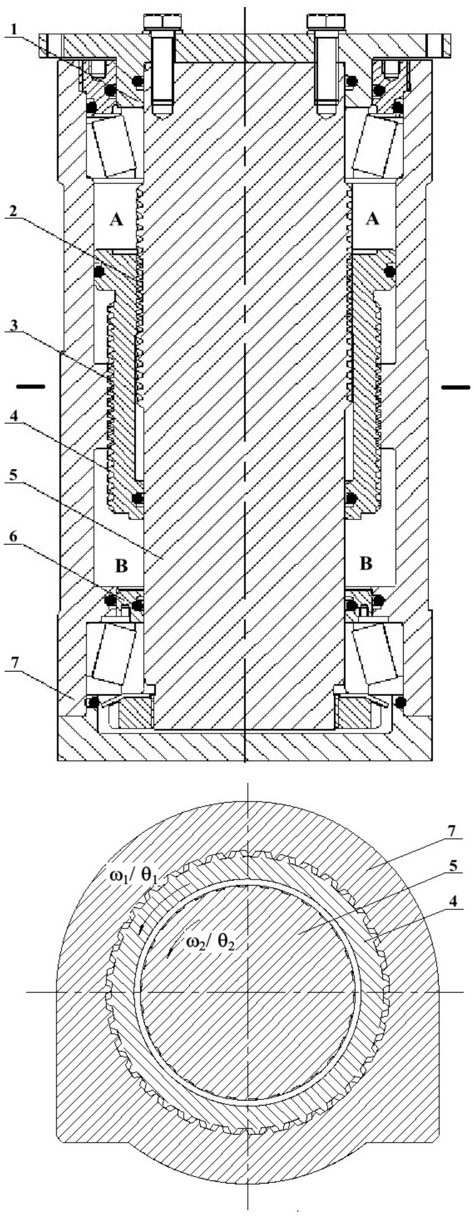

Fig.4. Structure of hollow axis swing cylinder with compensation rings

1-Compensation ring at axis, 2-Inner screw pair, 3-Outer screw pair, 4-Piston, 5-Shaft axis, 6-Compensation ring at fixed cylinder, 7-Fixed cylinder

While the rising angle is smaller than 45°, but fairly larger than its equivalent angle of friction, the axial force on the shaft axis would become fairly large, even larger than its tangential force. For, the rising angles are always limited by its structure in novel applications. The higher its rated pressure is, the lower efficiency and accuracy it have. It's essential to compensate this axial force to enhance the efficiency and angular accuracy. Although low friction material had been applied, and the pitches had been designed as large as possible, the unbalanced force of hollow axis swing cylinder based on double screw pairs is still very large without any compensation means.

For reverse direction screw pairs with compensation rings is shown in Fig.4, the direction of the shaft axis' swing speed ω_2 (relative with the piston), is the same as the direction of piston's swing speed ω_1 (relative with the cylinder). Assume the piston's axial movement speed v, and the pitches of its outer screw pairs and inner screw pairs are L_1 and L_2 separately. Then the piston's swing speed relative with the cylinder ω_1, the shaft axis' swing speed relative with the piston ω_2, and t he shaft axis' total swing speed ω are:

$$\omega_1=2\pi\cdot v/L_1 \tag{1}$$

$$\omega_2=2\pi\cdot v/L_2 \tag{2}$$

$$\omega=\omega_1+\omega_2=2\pi\cdot v\cdot(L_1+L_2)/L_1L_2 \tag{3}$$

Fig.5 shows the forces on the piston and shaft axis. α_1 and α_2 are the rising angles of the outer screw pair and inner screw pair separately. d_1 and d_2 are the screw pairs' middle diameters. A is the piston's effective area, and P_S is the pressure decrease on the swing cylinder. N_1 and N_2 are the normal press forces on the outer screw pairs and inner screw pairs, while N_{1x}, N_{2x}, N_{1y}, and N_{2y} are their horizontal and vertical vectors. f_1 and f_2 are the friction forces on the outer screw pairs and inner screw pairs, while f_{1x}, f_{2x}, f_{1y}, and f_{2y} are their horizontal and vertical vectors.

Since the swing speed is very slow, the forces are almost balanced during the whole transmission process. The hydraulic force on the piston $P_S\cdot A$ is balanced with the 2 axial force vectors from the cylinder and the shaft axis. According to the force balance equation of the piston (on vertical direction and horizontal direction respectively):

$$P_SA=N_{1y}+f_{1y}+N_{2y}+f_{2y}=N_1\cos\alpha_1+f_1\sin\alpha_1+N_2\cos\alpha_2+f_2\sin\alpha_2 \tag{4}$$

$$N_{1x}-f_{1x}=N_{2x}-f_{2x},\ \text{i.e.,}\ N_1\sin\alpha_1-f_1\cos\alpha_1=N_2\sin\alpha_2-f_2\cos\alpha_2 \tag{5}$$

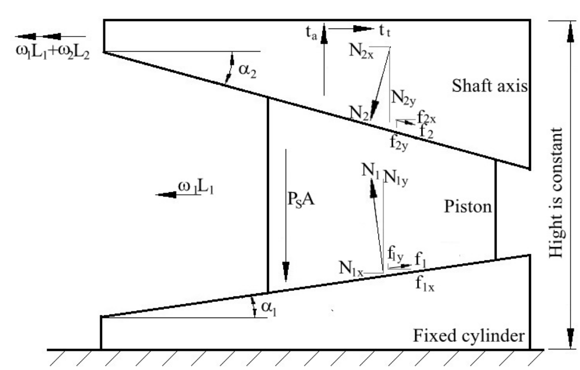

Fig.5. Interacting force between the shaft axis, piston and cylinder

Assume the friction coefficient of the two screw pair is μ: $f_1=\mu N_1$, and $f_2=\mu N_2$. And the equivalent angle of friction is γ: γ=arc tan(μ). Then,

$$P_S A\cos\gamma = N_1\cos(\alpha_1-\gamma) + N_2\cos(\alpha_2-\gamma) \tag{6}$$

$$N_1\sin(\alpha_1-\gamma)=N_2\sin(\alpha_2-\gamma) \tag{7}$$

$$N_2= P_S A \sin(\alpha_1-\gamma)\cos\gamma / \sin(\alpha_1+\alpha_2-2\gamma) \tag{8}$$

According to the force balance equation of the shaft axis (on vertical direction and horizontal direction respectively), the axial force t_a (Fig.3 shows that the direction of this axial force is as same as the direction of piston's hydraulic force) and tangential force on the shaft axis t_t are:

$$\begin{aligned} T_a &= N_{2y}+f_{2y} = P_S A \sin(\alpha_1-\gamma) \cos(\alpha_2-\gamma) / \sin(\alpha_1+\alpha_2-2\gamma) \\ &= P_S A\cdot\tan(\alpha_1-\gamma)/[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)] \end{aligned} \tag{9}$$

$$\begin{aligned} T_t &= N_{2x}-f_{2x} = P_S A \sin(\alpha_1-\gamma) \sin(\alpha_2-\gamma) / \sin(\alpha_1+\alpha_2-2\gamma) \\ &= P_S A\cdot\tan(\alpha_1-\gamma)\tan(\alpha_2-\gamma)/[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)] \end{aligned} \tag{10}$$

The output torque of shaft axis M is:

$$M = P_S A\cdot d_2\cdot\tan(\alpha_1-\gamma)\tan(\alpha_2-\gamma)/2[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)] \tag{11}$$

Equation (9) and (11) show that, if the rising angle of the inner screw pairs is much larger than the outer screw pairs, this swing cylinder is able to output large torque, accompanied with almost zero axial force. However, this rising angle is always limited by its structure in novel applications. Thus, the friction losses on the 2 screw pairs are fairly large due to the unbalanced axial force on its screw areas. While the rising angle is smaller than 45°, but fairly larger than its equivalent angle of friction, the axial force on the shaft axis would become fairly large, even larger than its tangential force. It's essential to compensate this axial force to enhance the transmission efficiency.

While the rising angles of the swing cylinder is fixed, the axial force on the shaft axis T_a is proportional with its driving pressure P_S, the coefficient is a constant. Therefore, a pair of hydraulic compensation rings are the best choice to balance this axial force. In Fig.2, part 2 and part 7 are the pair of hydraulic compensation rings, which are able to introduce driving pressure P_S into the two sides of the shaft axis. The unbalanced axial force could be fully compensated if the effective area of this compensation ring C_s is:

$$C_s=A\cdot\tan(\alpha_1-\gamma)/[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)] \tag{12}$$

III. EXPERIMENTAL RESULTS

This hollow axis swing cylinder based on double screw pairs is applied in the 270° elbow joint of an underwater hydraulic manipulator. The inverse direction screw pair type is selected to make the elbow joint and the whole manipulator is as compact as possible, while its swing angle/piston movement ratio is still enough.

This 7 function hydraulic manipulator had also been installed on a remote operated vehicle as shown in Fig.6. The black component in the red circle is this hollow axis swing cylinder based on double screw pairs. Its jaw 4 had lifted an 100kg's weight while the elbow joint can still swing freely.

TABLE I. PARAMETERS OF SCREW PAIRS FOR MANIPULATOR'S ELBOW

Name and unit of parameters	Screw	
	Outer	***Inner***
Pitch (mm)	144	120
Middle diameter (mm)	93	73
Rise angle (rad)	0.4579	0.4821
Friction angle (rad)	0.0997	0.0997

The effective area the piston A and the area of compensation ring C_s are designed to 48.1cm^2 and 23.161cm^2 respectively, and its maximum axial movement L is designed to 5cm. Therefore, the maximum swing angle is 270°, and the output torque could be as higher as 550Nm at 17Mpa's power supply pressure. The material of this ladder-shaped screw pairs are steel-albronze so that the friction coefficient μ is only 0.1. The parameters of the screw pairs is listed in table 1.

Fig.7 is the test rig for this hollow axis swing cylinder based on double screw pairs. The nature colored component in the red circle is the tested swing cylinder. A magnetic powder arrester is used to create a simulated load, and the torque sensor is used to test the torque. The shaft speed of the swing cylinder is detected by an integrated angular sensor, and be shown in an oscilloscope. The swing cylinder is driven by a hydraulic power supply and be controlled by a signal generator.

Fig.8 and fig.9 compared the output torque of hollow axis swing cylinders with compensation ring and without compensation ring. Experimental results show that, the swing cylinder with compensation ring is able to output much higher torque than those without compensation ring.

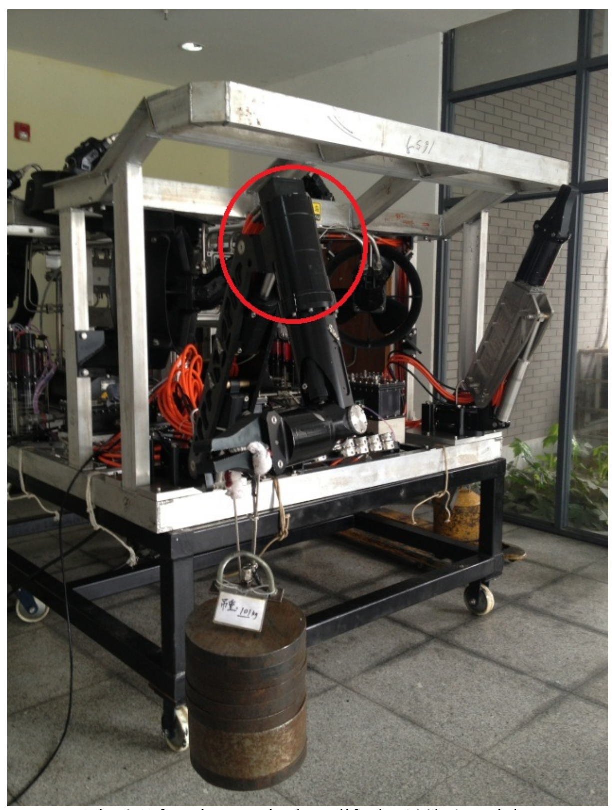

Fig.6. 7 function manipulator lifted a 100kg's weight

In Fig.9, the mechanical efficiency of swing cylinder without compensation ring begin to decrease after the input pressure is higher than 10MPa, and the output torque almost cannot increase after the input pressure is higher than 18MPa. These all due to the heavy axial force, which leads to large transmission friction and drained all the increase of input pressure away.

In Fig.8, the output torque of hollow axis swing cylinders with compensation ring matches its theoretical torque well, and have good linear relationship with its input pressure. The constant bias between the output torque and the theoretical torque may be caused by frictions at other parts, such as the sealing, bearing, etc. The total efficiency of hollow axis swing cylinder based on double screw pairs with axial force compensation structure is as higher as 81%, and the power supply's pressure can be increased to 21MPa to make the manipulator more compact, effect, and efficiency.

IV. CONCLUSION

(1) For a 7 Function hydraulic manipulator with a 270° rotary elbow joint, a hollow axis swing cylinder based on double screw pairs would be the best choice. The inverse direction screw pairs type is able to obtain larger swing angle/piston movement ratio, while the identical direction screw pairs type is able to output higher torque. The inverse one is more fit for the manipulator.

(2) For swing cylinders based on inverse direction screw pairs, the relationship between the piston's axial movement speed v and the swing speed ω are $\omega=2\pi v\cdot(L_1+L_2)/L_1L_2$, while the output torque of shaft axis is $P_S A\cdot d_2\cdot\tan(\alpha_1-\gamma)\tan(\alpha_2-\gamma)/2[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)]$.

(3) For swing cylinders based on inverse direction screw pairs, a pair of compensation rings should installed to balance the axial force. The effective area should be $A\cdot\tan(\alpha_1-\gamma)/[\tan(\alpha_1-\gamma)+\tan(\alpha_2-\gamma)]$. The output torque matches its theoretical torque well, and have good linear relationship with its input pressure after fully compensated.

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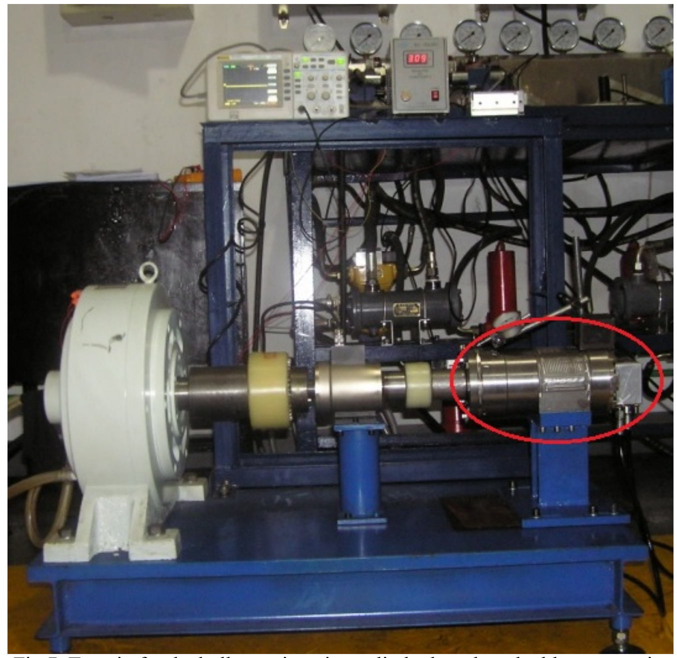

Fig.7. Test rig for the hollow axis swing cylinder based on double screw pairs

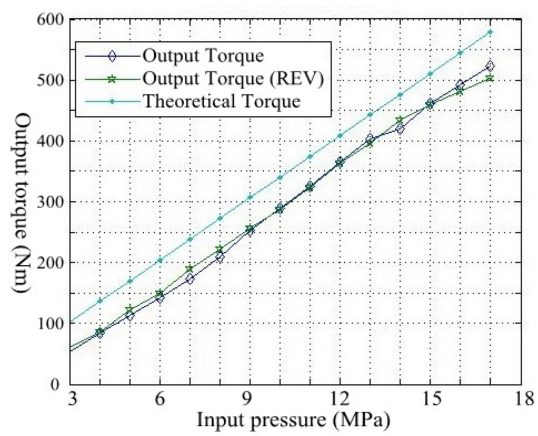

Fig.8. Output torque of hollow axis swing cylinder with compensation rings

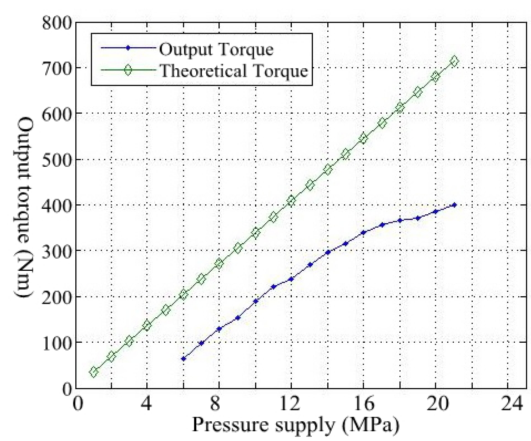

Fig.9. Output torque of hollow axis swing cylinder without compensation ring