

Parameter Designing for Heave Compensation Hydraulic System Installed in Deepwater

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Abstract

The function diagram of active heave compensation hydraulic system has been given, besides, the mathematics model for the principal hydraulic components of the compensation system has been built, and the input-output relation between components has been made clear. Aimed at compensating work capacity for the system, design and research on parameters as the bearing pressure, the initial state and the maximum flow of hydraulic cylinder, accumulator and other principal components have been made separately, and standardized design has been accomplished in accordance with relevant standards. Furthermore, calculus and verification for the capacity of the hydraulic system in different working stages have been made in order to calculate the pressure lose of the system and provide objective data for the hardware system design of the hydraulic components of the heave compensation system.

Keywords: Heave compensation; Hydraulic system; Mathematics model

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1. Introduction

The operation of dipping deepwater requires an active compensator for the heave compensation system. Through the research on the active heave compensation system we can see that components of the hydraulic unit is the core of the whole system, the selection of its parameters and the quality of its capacity directly influence the compensation effect and stability of the whole system, therefore, researches on parameter design of the hydraulic components of the active heave compensation system need to be made.

2. Research Method

2.1 Modeling for the hydraulic components of the active compensation system

The function diagram of the hydraulic unit of the active heave compensation system is shown in Figure 1.

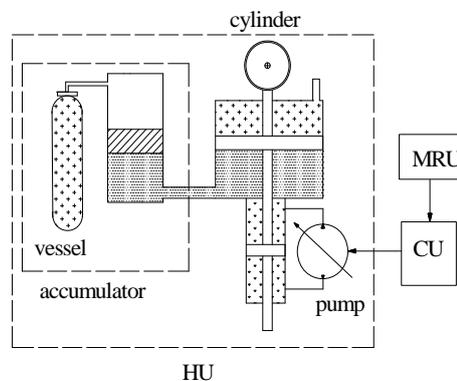


Figure 1. Function diagram of the active heave compensation system

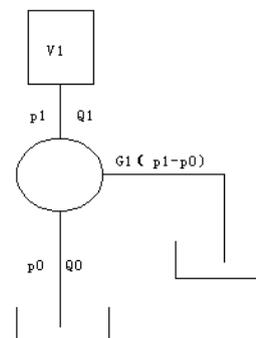


Figure 2 Schematic diagram of the model of the hydraulic pump

2.2 Hydraulic cylinder

Tables and Figures are presented center, as shown below and cited in the manuscript. Model of the hydraulic pump used in the compensation system is shown in Figure 2. Its mathematical model is

$$Q_1 = Q_0 - G_1(p_1 - p_0) - \frac{V}{K} \dot{p}_1 \quad (1)$$

In the above equation,

Q_0 、 Q_1 —the theoretical flow and the actual flow of the oil pump;

G_1 —the liquid group of the oil pump;

V —the outlet volume of the pump;

K —the elastic modulus of the volume of the oil.

2.3 Oil pipe

The input of the oil pipe is the output flow of the pump Q_1 , the discharge of the relief valve is Q_2 , the inlet of the fuel oil of the solenoid directional control valve is Q_3 , and the output is the pressure of the oil inside the oil pipe p_1 . Let the volume inside the oil pipe be V_2 , according to the flow characteristics of the oil pipe:

$$Q_1 = Q_2 + Q_3 + \frac{V_2}{K} \dot{p}_1 \quad (2)$$

We can infer that,

$$p_1 = \int \frac{K}{V_2} (Q_1 - Q_2 - Q_3) dt \quad (3)$$

2.4 Relief valve

The structure sketch of the relief valve is shown in the Figure 3.

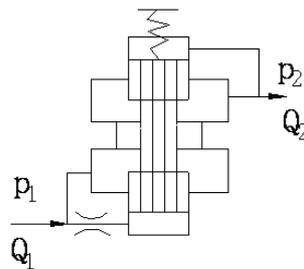


Figure 3. the structure sketch of the relief valve

Through the stress analysis of the whole relief valve we can arrive at the motion equation for the valve core of the relief valve:

$$m_r \ddot{x}_v = (p_1 - p_2) A_r - K_s (x_c + x_v) - (K_v + K_f) \dot{x}_v \quad (4)$$

In the above equation, m_r , x_v —the mass and the displacement of the valve core;

p_1 , p_2 —the oil pressure of the inlet and the outlet;

K_s , x_c —spring stiffness and the amount of preloading compression;

K_v, K_f —coefficient of transient state hydrodynamic force and of viscous damping;
 A_r —the action area of the fluid.

When it refers to large scale hydraulic system, the dynamic characteristics of the relief valve at its start are to be ignored, and its model is simplified to

$$Q_2 = G_2(p_1 - p_r) \quad (5)$$

In the above equation, G_2 —the fluid group of the relief valve;

p_r —the adjustable pressure of the relief valve.

2.5 Servo valve

Adopting four-way valve control mode, the output of the servo valve consists of the displacement of the valve core and the output flow. The output is control voltage $u(s)$, through its working characteristics we can infer that the relation between the displacement of the valve core and the voltage is as follows:

$$x(s) = \frac{k}{1 + \tau s} u(s) \quad (6)$$

In the above equation, k is the displacement gain of the valve core, τ is the equivalent time constant of displacement;

The motion equation of the valve core after it is energized is:

$$F_e = m\ddot{x} = (B_v + B_f)\dot{x} + (K_v + K_s)x + R_m \quad (7)$$

In the above equation,

F_e —the external force posed on the valve core: electromagnetic force;

m, x —the mass and the displacement of the valve core;

B_v, B_f —coefficient of transient state hydrodynamic force and of viscous damping;

K_v, K_s —coefficient of stabilized hydrodynamic force and centered spring rigidity;

R_m —Coulomb frictional resistance.

When servo valve changes its direction, the flow equation of the bypass valve can be represented as follows:

$$q = C_d A \sqrt{2(p - p') / \rho} = C \sqrt{p - p'} \quad (8)$$

In the above equation,

q —the discharge flows through the servo valve;

C_d, A —Flow coefficient and the area of the valve port;

ρ —liquid density;

p, p' —The liquid pressure of the inlet and the outlet of the solenoid valve;

C —aggregate discharge coefficient of the solenoid valve.

Since the response speed of servo valve is relatively fast, when considered about the dynamic performance of large scale system we only considered about its flow characteristics, the schematic diagram of four-way valve controlled hydraulic cylinder is shown in figure 4.

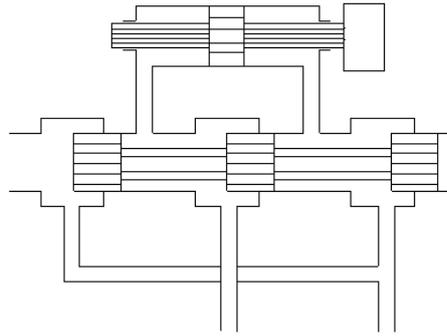


Figure 4. The schematic diagram of four-way valve controlled hydraulic cylinder

Suppose the location of x is 0 in its equilibrium, and when $x < 0$, servo valve will provide oil for the left chamber of the working cylinder, the fuel supply quantity is Q_4 , the oil of the right chamber returns, and the quantity is Q_5 , Q_6 is the fuel discharge flows from servo valve into oil tank, hence $Q_3 = -Q_4$, $Q_5 = -Q_6$, and the flow equation of the flow output is:

$$Q_4 = -Q_3 = C\sqrt{p_1 - p_2} \quad (9)$$

$$Q_5 = -Q_6 = C\sqrt{p_3 - p_4} \quad (10)$$

When $x > 0$, servo valve will provide oil for the right chamber of the working cylinder, the fuel supply quantity is Q_5 , the oil of the left chamber returns, and the quantity is Q_4 , under this situation, $Q_5 = -Q_3$, $Q_4 = -Q_6$, and the flow equation of the flow output is:

$$Q_5 = -Q_3 = C\sqrt{p_1 - p_3} \quad (11)$$

$$Q_4 = -Q_6 = C\sqrt{p_2 - p_4} \quad (12)$$

2.6 Working cylinder

For working cylinder, the input of the left chamber is Q_4 and the output is p_2 , so on the basis of the flow characteristics, the relation between the input and the output is as follows:

$$p_2 = \frac{K}{V_3} \int Q_4 dt \quad (13)$$

The input of the right chamber is Q_5 and the output is p_3 , so the relation between the input and the output is as follows:

$$p_3 = \frac{K}{V_4} \int Q_5 dt \quad (14)$$

Through the loading analysis of the piston we can get:

$$m_1 a_1 = p_2 A_1 - p_3 A_2 - F_{f1} \quad (15)$$

In the above equation, A_1 , A_2 —the area of the left chamber and the right chamber of the working cylinder;

m_1 —the mass of the piston;

F_{f1} —sum of all kinds of frictional resistance.

The flow equation is:

$$Q_4 = A_1 \dot{x}, \quad Q_5 = -A_2 \dot{x} \quad (16)$$

Through the loading analysis of power cylinder, we can get the kinetic equation:

$$m_2 a_1 = p_5 A_3 - p_6 A_4 - F_{f2} \quad (17)$$

Flow equation is:

$$Q_7 = A_3 \dot{x}, \quad Q_8 = -A_4 \dot{x} \quad (18)$$

Accumulator

Take piston type accumulator for example to establish its dynamic equation:

$$m_4 a_4 = p_6 A_6 - p_b A_6 - m_4 g - F_{f4} \quad (19)$$

In the above equation,

m_4 —the mass of the accumulator piston;

p_6 , A_6 —the fluid pressure of the inlet and the area of the piston;

p_b —the chamber pressure of the accumulator;

F_{f4} —sum of all kinds of frictional resistance.

3. Results and Analysis

In this section, it is explained the results of research and at the same time is given the comprehensive discussion. Results can be presented in figures, graphs, tables and others that make the reader understand easily [2], [5]. The discussion can be made in several sub-chapters.

Basic capabilities of heave compensation system refer to the following datum:

- Maximum compensation capability: 200 t ;
- Maximum compensation speed: 1 m / s ;
- The maximum stroke of the hydraulic cylinder piston: $\pm 3m$.

The parameter calculation method for the principal hydraulic components of the system refers to bibliography(1) and (2), through calculation and analysis we assume the diameter of the cylinder rod is $d_1 = 70mm$, its inside diameter $D = 100mm$, the maximum flow the system needed is $Q = 424L / \text{min}$, the inside diameter of the oil pipe is $d = 45mm$, the average flow oil source could provide is $489L / \text{min}$, and its control voltage could be $\pm 10V$ direct current, The volume of the oil tank is $V = 2445 \sim 3423L$.

4. Conclusion

This paper makes clear the input-output relation between hydraulic components for heave compensation system installed in deepwater, and makes design and research on parameters as the bearing pressure, the initial state and the maximum flow of hydraulic cylinder, accumulator and other principal components separately. Furthermore, it makes calculus and verification for the capacity of the hydraulic system in different working stages in order to calculate the pressure lose of the system, meet practical engineering requirement and provide objective data for the hardware system design of the hydraulic components of the heave compensation system.

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