Effect of design parameters on vibrational behaviour of a counter balance valve equipped hydraulic crane

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ABSTRACT

Systematic investigation of the effect of the most significant designing parameters on the vibrational behaviour of a Hiab 031 crane is carried out. A counter balance valve is used in the fluid power circuit to prevent loads running ahead of pump. The study is carried out by means of computer simulation using a mathematical model for the complete hydromechanical system. The model is subjected to experimental verification.

KEYWORDS

modelling, simulation, CAE, coupled system, counter balance valve

NOMENCLATURE

$A(x_v)$ = flow cross section area of valve orifice
 $a_1, \dots, a_5$ = empirical valve friction parameters
 $C_1 \dots C_5$ = valve parameters
 C_d = discharge coefficient
 K = variable describing opening of valve orifice
 p_{ref} = set pressure (scaled value of preload setting of valve spring)
 p_1 = load pressure
 p_2 = return pressure
 p_3 = inlet pressure
 x_1 = length of cylinder
 x_v = valve spool displacement
 ρ = oil density

INTRODUCTION

Hazardous vibrations are a major common problem when using a counter balance valve for lowering loads. Several investigations and product development work have been carried out to obtain a stable valve construction. Unfortunately, the dynamic behaviour of a pressure valve such as a counter balance valve depends quite strongly on the structure of the hydromechanical system that it is connected with. Therefore, several manufacturers have large valve series for various applications. The knowledge in selecting right valve type for a machine is often based on practical experience rather than engineering science.

It is quite complicated and expensive to carry out experimental studies for solving this kind of engineering problems. Usually, it is possible only to study the effect of the minor variations in the valve construction on the behaviour of a particular machine application because it is very expensive and time-consuming to vary dimensions of the hydromechanical system of the

machine. The only relatively cheap method to study the effects of the system parameters on the valve behaviour is computer simulation. Also utilizing computer simulation is relatively complicated because a mathematical model for the complete hydromechanical system is required. Suitable software for simulating such systems is normally not available for machine designers.

In Tampere University of Technology effective methods and software have been developed since 1984 for simulating complete hydromechanical systems. In this study these tools are used in investigating the effects of some important parameters on the vibrational behaviour of a Hiab 031 crane. Such parameters as the flexibility of fluid power circuit, the actuator friction, valve friction and valve damping are considered.

Before varying the parameters the basic model of the system is subjected to experimental verification. Semi-empirical method is used in modelling the counter balance valve as well as the other components in the fluid power circuit.

TEST SYSTEM DESCRIPTION

FIGURE 1 shows part of the Hiab 031 crane. The mechanics of the complete crane consists of five arms, five joints, two internal slides, two external slides, four cylinders, eight fluid power components and nine compressible oil volumes. The input signal to the system is a fixed negative voltage ramp to the proportional valve in order to achieve a load lowering action by means of the lifting cylinder. The return port of the lifting cylinder is equipped by a commercial commonly used counter balance valve as shown in the scheme. The crane is loaded by mass which is connected by a rope with the end of the crane boom.

SIMULATION MODEL

Crane model

The simulation model describes all significant factors that affect on the dynamic behaviour of the crane such as mechanical and hydraulic flexibility, friction, backflash in joints, dynamic behaviour of all hydraulic components etc. This is significant because the crane executes a large amplitude lifting work followed by a small amplitude vibration. The methods employed in postulating the model are introduced in previous studies [1],[2],[3],[4]. Several experiments and simulations are carried out in order to obtain good agreement between theoretical and experimental results.

Counter balance valve model

Semi-empirical method is employed in modelling the

counter balance valve [2],[3]. The method can be described roughly as follows: A conventional type of model for a fluid power valve can be referred to as analytical. It uses physical equations which describe the behaviour of the internal elements and their interconnections within the valve. In the semi-empirical method the analytical model is brought into a compact form in which the parameter values can be identified using measured characteristic curves. Postulation and experimental verification of the valve model used in this study is expressed in [5].

Figure 1. Scheme of Hiab 031 crane and lifting cylinder circuit

FIGURE 2 illustrates the schematic construction of the valve. It is a commonly used overcenter type of a counter balance valve. The valve spool is balanced by three pressures and a spring. The pressure forces generated by load pressure p_1 and inlet pressure p_3 tend to open the valve orifice. The pressure force generated by the return pressure p_2 as well as the spring force tend to close the valve orifice.

The spring constant of a counter balance valve is normally relatively high. Because of this the flow force has not as important meaning as it has in other types of pressure valves at least if the size of the valve is small. Normally, the spool mass is low and the damping orifices small providing overdamped dynamic characteristic.

One important difference between counter balance valves and other types of pressure valves is the high seal friction which causes hysteresis. It is in some valve

types so high that it has a major effect on the valve behaviour. Sometimes this makes the model postulation complicated.

The semi-empirical model [5] is expressed as follows

$$\left. \begin{aligned} \xi C_4 \dot{K}^2 + C_3 \dot{K} + K (C_1 + C_2 (p_1 - p_2)) \\ = p_1 - p_{ref} - C_5 p_2 + C_6 p_3 - p_{fric} \\ Q = K \sqrt{p_1 - p_2} \end{aligned} \right\} \quad (1)$$

where

$$\xi = \begin{cases} 1, & \text{when } \dot{K} \geq 0 \\ -1, & \text{when } \dot{K} < 0 \end{cases}$$

and

$$K \geq 0$$

Figure 2. Schematic representation of a counter balance valve

Each parameter $C_1, \dots, C_6$ is a function of parameters that appear in original analytical model i.e. internal dimensions, spring constants, physical coefficients etc. Variable K describes opening of the valve orifice. The physical description of K is

$$K = C_d A(x_v) \sqrt{\frac{2}{\rho}} \quad (2)$$

The friction force between the spool and housing is scaled as the pressure p_{fric} which is determined empirically from the steady-state characteristic curves of the valve. Normally, in counter balance valves the seal friction can be described as follows

$$p_{fric} = f(p_1, p_2, p_3, K, \dot{K}) \quad (3)$$

In case of the valve type used in this study sufficiently accurate model was obtained using a simple empirical function depending only on K and $\dot{K}$. The rate of pressure seemed not to have significant effect on the seal friction. Thus, the friction pressure is expressed as

$$p_{fric} = \psi (a_1 + a_2 \sqrt{K} + a_3 K + a_4 K^2 + a_5 K^3) \quad (4)$$

where

$$\psi = \begin{cases} 1 & \text{when } \dot{K} \geq 0 \\ 0 & \text{when } \dot{K} < 0 \end{cases}$$

and

$$p_{fric} \geq 0$$

EXPERIMENTAL VERIFICATION OF THE SIMULATION MODEL

The validated crane model in which the counter balance valve is not included is expressed in [1]. In [5] the valve model is studied in particular test circuit in which it is possible to obtain the steady-state and dynamic characteristic curves of the valve. In order to ensure the accuracy of the combined test system model simulations and experiments are carried out again in this study. FIGURE 3 compares the simulated and measured results when a negative voltage ramp is executed as an input to the proportional-four way directional valve. It can be seen that a relatively good agreement between theory and experiments is achieved. The difference between the results when $t=0$ is caused by a high rate of cylinder friction which allows several different initial states. The crane load is 280 kg. Generally, the conclusion is to be drawn from the results that the dynamic behaviour of the crane is relatively stable when lowering load.

EFFECT OF PARAMETER VARIATIONS ON CRANE BEHAVIOUR

In this section effects of the most important factors on the dynamic behaviour of the crane when lowering load

are studied. As already concluded in the previous section there are no particular problems in the test system. Because of this in this study the system is disturbed by parameter variations in order to find potential factors which may cause hazardous vibrations in the system.

Figure 3. Comparison of measured and simulated dynamic responses of a Hiab 031 crane.

Effect of valve friction

The valve size is relatively small (maximum flow about 60 l/min). Normally, in case of small sized valves the seal friction between the spool and housing is less important than in case of the valves of larger size because of shorter clearances. This is why it seems not to have significant effect in the nominal responses (see FIGURE 3). When the friction function, eq.(4),

was multiplied by 5 i.e. $p_{fric} = 5 \times p_{fric}$ high frequency vibration appeared in the simulated responses. FIGURE 4 shows the simulated pressures p_1 and p_2 when $p_{fric} = 10 \times p_{fric}$. K describes the valve dynamics (see eq.(2)). The results show that a high rate of valve friction makes the valve unstable. Although the valve spool oscillates with relatively high amplitude it has not remarkable effect on the behaviour of the load pressure p_1 because of much higher frequency than the natural frequency of the hydromechanical system. This kind of valve oscillation would probably cause only noise and, in the long term, valve and seal damages.

Figure 4. Effect of high valve friction

Effect of hydraulic flexibility

The load port of the valve is connected directly with return port of the lifting cylinder. The directional valve is connected with the upper cylinder port and the return port of the counter balance valve by two high pressure hoses about 2 m of length. The pilot port of the counter balance valve is connected with upper cylinder port by means of one meter long steel pipe (see FIGURE 1). The effect of flexibility was studied by decreasing the bulk modulus of the hoses to one third from its nominal value ($B_h = 600$ MPa). This has similar effect to as making the hoses three times longer. This causes delay in the valve behaviour and has minor effect on the natural frequency of the hydromechanical system of the crane boom. The most important result was the effect of both high valve friction and low bulk modulus. It can be concluded from the results shown in FIGURE 5 ($B_h = 200$ MPa and $p_{fric} = 10 \times p_{fric}$) that the low bulk modulus gained the instability of the valve and at the same time lowered the natural frequency of the valve. This is hazardous because now the frequency of valve vibration is low enough to disturb the hydromechanical system of the crane boom. This can be seen in the response of the load pressure p_1 (see FIGURE 5). The high frequency vibration can in case like this be seen in the motion of the crane boom. In the long term damages in the valve seat, actuator

sealing and boom structure are possible.

Figure 5. Effect of both high valve friction and low bulk modulus of hoses

Effect of damping of the valve

Normally, damping of a spool or poppet valve is achieved by small damping orifices which limit the flow velocity out and into the damping chambers that are bored in one or both ends of the valve poppet or spool (see FIGURE 2). As already mentioned the spring constant of a typical counter balance valve spring is relatively high. This provides overdamped dynamic characteristic in a sufficiently stiff circuit if the seal friction is relatively low. If there is a high seal friction in the valve sometimes the manufacturers try to improve the valve stability by increasing the damping. This sometimes results even worse behaviour because the natural frequency of the valve is decreased. Furthermore, if the diameter of a damping orifice becomes less than 1 mm contamination problems can become apparent. FIGURE 6 illustrates simulation results of the system where the valve damping is three times higher than it was in the original system. This is achieved by multiplying the parameter C_4 in the valve model, eq.(1) by three. The valve friction is $p_{fric} = 10 \times p_{fric}$ and bulk modulus of the hoses $B_h = 200$ MPa.

The results in FIGURE 6 show that although the amplitude of the valve vibration was decreased by increasing the valve damping the lower natural frequency disturbs the hydromechanical system of the crane boom dramatically. This causes poor dynamic behaviour of the complete crane and in the long term structural, component and seal damages.

Effect of actuator friction

Poor dynamic behaviour caused by stick-slip friction is a major common problem in mobile crane boom systems. Although, sometimes relatively rough sealing is used the actuator friction does not cause problems. The main sources of stick-slip friction vibration are mechanical slides in boom telescopes.

Figure 6. Effect of high valve damping with high valve friction and low circuit stiffness

However, this kind of systems are not considered in this study. In the following a simple test is carried out by increasing the steady-state friction of the lifting cylinder. FIGURE 7 shows the simulation result when the steady-state friction of the lifting cylinder is three times as it was in the original system. Other parts of the model are equal to the original model (see FIGURE 3). It can be concluded from the results that the increased stick-slip friction of the seals become a source of high frequency vibration. One interesting result is that the vibration frequency is low enough to disturb the valve. Generally, it can be said that this kind of friction can cause noise, seal damages and valve damages. To prevent the valve problem in FIGURE 8 the damping of the valve is increased three times higher from the original. From the results it can be seen that the actuator friction does not disturb the valve as much as in FIGURE 7. However the high valve damping did not have remarkable influence on the stick-slip friction vibration itself. Only a slight reduction in the amplitude of p_1 is achieved.

Figure 7. Effect of high actuator friction

CONCLUSIONS

In the present paper effects of the most important parameters on vibrational behaviour of a counter balance valve equipped Hiab 031 crane are investigated. The study is carried out by employing a simulation model describing the complete hydromechanical system of the crane. The model is postulated by combining the crane model and a counter balance valve model that were postulated in previous studies [1],[5]. The combined model is subjected to experimental verification. The model is proved to be an effective tool for investigating effects of parameters such as valve friction, valve damping, circuit stiffness and actuator friction on the crane behaviour.

The most important source of vibration is the high seal friction of the counter balance valve. Low circuit stiffness deteriorates the instability of the valve and decreases the oscillation frequency of the valve which increases the crane boom vibration. Furthermore, the increased damping of the valve gains the boom vibration by lowering the oscillation frequency of the valve.

Figure 8. Effect of high actuator friction and high valve damping

The high seal friction in the lifting cylinder generates a high frequency vibration in the system. This proved to be more hazardous when the valve damping is low because when the valve is sufficiently fast it starts to respond to the high frequency vibration and thus gain the amplitude of the vibration.

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