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ITJ

ISSN 1812-5638

INFORMATION TECHNOLOGY JOURNAL

ANSInet
Asian Network for Scientific Information
308 Lasani Town, Sargodha Road, Faisalabad - Pakistan

Information Technology Journal 12 (14): 2863-2867, 2013
ISSN 1812-5638 / DOI: 10.3923/itj.2013.2863.2867

Computer Aided Design and Simulation for the Characteristics of Energy-saving Steering Pump Based on the Speed Compensation

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Abstract: In view of practice work situation of the steering pump, this study put forward speed compensation characteristics of a new variable displacement of double-action vane pump based on the action principle and took account of the fact that the hydraulic power steering system of an automotive has serious energy loss in general. The prototype was produced by CAD technology and performance test was performed. The results show that the new pump with the variable mechanisms and rate compensation function can reduce the energy loss of the automotive hydraulic power steering system efficiently

Key words: Steering pump, simulation, variable mechanism, speed compensation, energy saving

INTRODUCTION

As China puts "energy conservation and emission reduction" forward as an achievable and sustainable development strategy and the implementation of the "fuel tax" reform plan, reduces oil energy consumption is becoming the focus of attention; especially outstanding in the field of automobile energy saving technology. Study shows that the active hydraulic steering system consumes approximately 3% of prime mover energy (Yao, 2005). While the car is driving, most of the time the engine is within the idle speed range, also over 80% of the time that the vehicles are running in straight line or in a stand by status, therefore most of the oil outputs from the pump flows through the flow control valve and the steering control valve then back to the storage tank, this process is causing a massive energy loss (Li, 2010). As the automobile steering pump has large energy loss concern, our group presents a new type of double action vane pump. This study introduces the principle and structure of new pump which can reduces the energy loss of steering system that is validated by computer simulation, reduces the steering system of vehicle fuel consumption and how to reduce automobile fuel consumption.

SPEFIC COMPENSATION PRINCIPLE

Working principle of the new pump: The new steering pump is improved on the basis of the existing quantitative double action vane pump structure (Eiichi, 2003). It is based on the structure of traditional vane pump, by changing the mechanical structure inside of the pump to achieve the goal of saving energy and reducing consumption. Compared with the traditional steering vane pump, the new type steering vane pump keeps the advantage of rotor radial force equilibrium in the existing quantitative vane pump. Moreover, it can be very convenient to automatically change the displacement that based on the different rotational speed, for steering system output pressure oil according to the change of rotating speed and to improve the traditional hydraulic power steering system of large energy loss. The working principle of the new pump is that installs up and down adjustable floating block in the rotor slot. As the change of the rotor rotational speed, floating block can slides up and down inside the slot. Therefore, the amount of space that floating block occupies in the oil chamber is related with the rotation speed of the pump and the pump is variable. When the rotor speed increases, due to the centrifugal force, floating block stretches out of the rotor slot, takes up more effective volume space between two vanes, turns the amount of suction and displacement of oil pump decreases every time. Based on the same theory, when the rotor speed decreases, floating block was retracted in the rotor slot, the effective volume of pump was increased, the pump suction and displacement of oil increases every time when it turns.

Equations for theoretical flow: When the pump is running, the floating blocks in the process of increase will take up a certain piece of space within the effective volume inner surface of between the cam ring and

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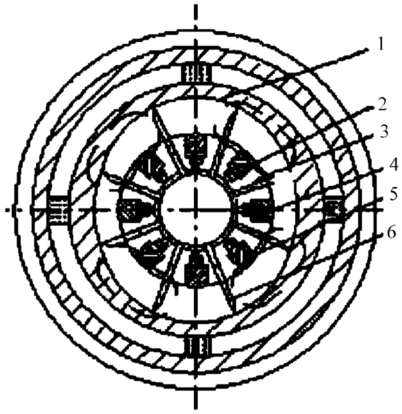

Fig. 1: Schematic diagram of new pump: (1) Stator, (2) Spring, (3) Rotor, (4) Floating block, (5) Floating tank, (6) Vane

rotor-vanes. When the floating blocks declined, it will be back soon from the starting location to the beneath of the floating slot. Therefore, the processes of the floating blocks rise or down impact on the displacement are the same. Only consider the ideal circumstances of the leakage between block bodies and floating block slot, the theoretical average flow of pump formula is possible to write (Chen *et al*., 2006):

$$q_v = \{2b[(R_2^2 - R_1^2)\pi - \frac{(R_2 - R_1)z\sigma_2}{\cos\theta}] - 8\times xb\sigma_1 - 8\times xb\sigma_1 - \frac{b'h^3(p_1 - p_2)}{12\eta l}\}n \tag{1}$$

where, x is the vane lift height, b is floating block depth and n is rotational speed of pump. Z is the number of vanes. H_1 and H_2 are the thickness of vane and floating block, respectively. θ is the vane obliquity. R_1 and R_2 are the minimum cam radius and maximum cam radius respectively. b′ is the gap depth on uprightness in velocity of flow. h is the gap height. P_1 and P_2 are the high and low pressure of gap respectively. l is the gap length and η is the dynamical mucosity degree of oil.

Then, the output power can be expressed by the following formula (Liu, 2002):

$$P_p = q_v.P \tag{2}$$

SIMULATION OF POWER IN THE SYSTEM

Based on the above analysis and the principles of pump, established the output power equation. In the light of reference (Liu, 2011) to select mathematical equations (Gao *et al*., 2003) and parameters of the hydraulic power steering system and to apply the Simulink software to establish simulation models of hydraulic power system. The dynamic simulation studies of pump output power conducted.

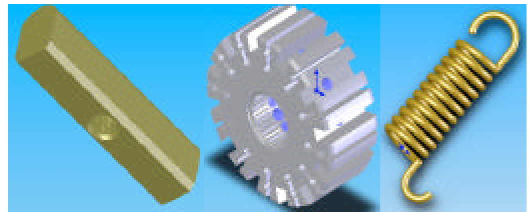

Fig. 2: Accessory configuration sketch map of the double-action pump by CAD

New pump parameters: The new pump parameters are as follows: (1) The rotational speed from 500-3200 r min^{-1}, the work pressure is 6MPa, (2) The maximum cam radius is 22.6 mm and the minimum cam radius is 19.3 mm; the rotor radius is 19 mm, the number of vanes are 10; the floating block thickness is 20 mm and vane thickness is 1.4 mm, (3) In floating block part, the width of the block is designed to be 4.5 mm, height is 6.2 mm, the thickness is 22.0 mm, the top of the floating block has 2.5 mm radius with arc top structure, the lower part of the block has 0.8 mm chamfer on both sides, the flat bottom surface has a screw hole, its diameter is 3.0 mm. The width of the slot in the rotor is designed to be 4.5 mm, the height is 6.0 mm and the thickness is 22.0 mm; the diameter of the rotor spring is 4.0 mm and the linking spring is used the common circle cylinder helical spring that used in engineering production. The section of the spring wire is round and its diameter is 0.5 mm. The pitch of the spring is 3.0 mm. According to the diameter of the spring wire, the operative numbers of spring circles are 11. Then, the accessory configuration sketch map of the double-action pump by CAD (Kumar and Sahasrabudhe, 2010) is shown in the Fig. 2.

Mathematical model: According to bond graph construction method and the pump power bond graph of the reference (Li and Quan, 2002) of this study, to draw the vehicle hydraulic power steering system bond graph shown in the Fig. 3.

In the light of the bond graph (Ahlawat *et al*., 2010), the math equations of the simulation model are established in the following:

- Redirector output force F_a:

$$F_a = P_a.\ A \tag{3}$$

where, A is the area of redirector piston.

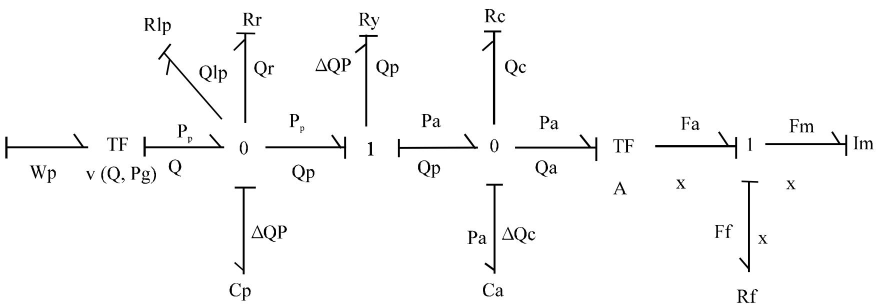

Fig. 3: Bond graph of the hydraulic steering system

Turn into redirector oil quantity Q_a:

$$Q_a = X.A \tag{4}$$

- Discharge pressure P_p:

$$P_p = \frac{1}{C_p}\int \Delta Q_p dt + P_p(0) \tag{5}$$

- Turning valve pressure P_a:

$$P_a = \frac{1}{C_a}\int \Delta Q_c dt + P_a(0) \tag{6}$$

- Redirector speed X:

$$X = \frac{1}{I_m}\int F_m dt + X(0) \tag{7}$$

- Pump leakage Q_{LP}:

$$Q_{LP} = R_{LP}.P_p \tag{8}$$

- Redirector leakage Q_c:

$$Q_c = R_c.P_a \tag{9}$$

- Valve displacement Q_p:

$$Q_p = C_d \cdot A_d \cdot \sqrt{\frac{\Delta P_v}{\rho}} \tag{10}$$

SIMULATION ANALYSIS

In order to verify the feasibility of the theory on speed compensation, according to the vane pump flow equation and set up the hydraulic power steering system model (Kim, 2010) to study the dynamic change process of output power when the new vane pump as a steering vane pump. Especially to analyze the energy saving effect of the speed compensation from the simulation curves of displacement flow for differential overflow valve in the new steering pump. Due to certain differential overflow valve, the steering pump in the system was set in a constant output flow rate for the Q_{output} = 24 l min.

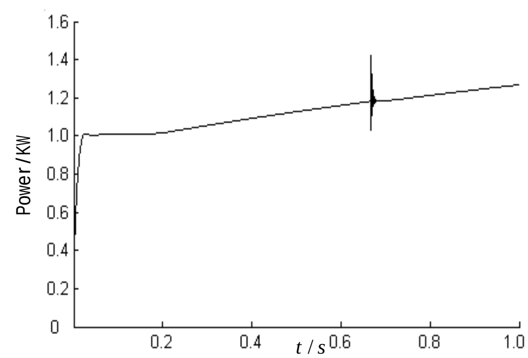

Fig. 4: Simulation curve 1 of power for the pump under the uniform accelerated speed

Uniform accelerated speed: Increased the rotational speed of pump and imposed a moment force on steering wheel to produce the step corner about 0.5 radian. The rotational speed of pump in the hydraulic power system increased from 10 r sec^{-1} to 25 r sec^{-1} in 1 second with the uniform accelerated speed by the slope of 15. The actual export power of pump system dynamic response curve as seen in the Fig. 4.

Figure 4 can be considered that the rotational speed of the system in sharp rise within 1 second and the system responded very soon. The output power changed in a smooth and stable framework. The variable machine in approximately 0.65s began to work and brought change of output oil with the pump rotational speed change. So, the system began to be instability, but quickly re-stabilized in the system. In the same conditions with other parameters, the rotational speed of pump in the hydraulic power system increased from 20-50 r sec^{-1} in 1 second with the uniform accelerated speed by the slope of 30. The dynamic response curve of the pump export power as shown in the Fig. 5.

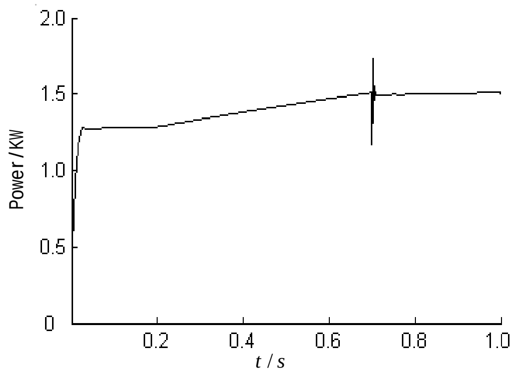

Fig. 5: Simulation curve 2 of power for the pump under the uniform accelerated speed

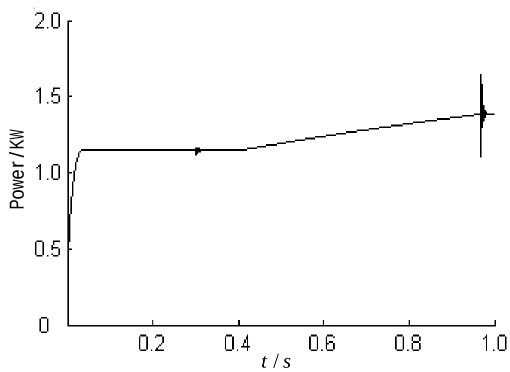

Fig. 6: Simulation curve of power for the pump under accelerated speed

From the trend changes in the simulated curves can be seen, for the same type of situations when rotational speed changes more stably, the trend changes in the system is more stable and the overshoot is a little volume. By contrast between Fig. 4 and 5, we can see that the peak output current and stable value had increased with the speed raised. The power fluctuations rose from the changed of speed and the floating bodies worked. It is consistent with the actual system.

Accelerated speed: The Fig. 6 reported that in the identical conditions of other parameters, the pump rotational speed in the hydraulic power system had increased with the accelerated speed changed by the slope of 15. As the rotational speed rose within 1 second, in the former 0.3s, the rotational speed maintained in 10r/s, from 0.3s to 1s the pump rotational speed increased from 10 to 22 r sec^{-1}, the response curve of dynamic simulation shown in Fig. 5. In the same simulation conditions, the pump speed is unchanged in the former 0.3s and then the subsequent 0.7s speed increased. Seen from the simulation curves that the emergence of the overshoot in the system appeared at 0.3s and the system speed was set to increase at 0.3s. Because the speed was changing, the system flow appeared minor fluctuations. At about 0.9s, to meet the working speed value of the variable displacement machines in the pump, therefore floating blocks began to rise and pump theoretical flow was changed again. The factors of such changes led to instability trend of the hydraulic system again, but quickly stabilized. Throughout the entire curves of the trend, with the rotational speed increased, curves of output power remained constant changing trend and a slight increase.

Fig. 7: Physical map of prototype parts

Table 1: Displacement measurement of pumps

Revolving speed (r min^{-1})	Theoretical flow (L min^{-1})	Pump Output of prototype (L min^{-1})
1000	19.4	18.6
1100	21.2	20.0
1200	22.8	21.4
1300	24.6	22.8
1400	27.2	24.4
1500	28.8	25.2
1600	30.6	26.8
1700	33.0	27.8
1800	34.8	29.0
1900	36.2	29.8
2000	38.0	30.2
2100	39.8	30.8
2200	41.4	30.4
2300	43.6	30.0
2400	45.2	30.8
2500	47.0	31.8

THE TEST

To verify the feasibility and energy saving effect under the speed compensation principle condition of new steering pump, the steering vane pump prototype was made in Dalian hydraulic parts factory of China to take the performance test-bed for performance test. The Table 1 is the displacement measurement value of prototype on the test bed and theoretical displacement value.

CONCLUSION

Simulation and experimental results have shown that the output curves of the prototype and the curves of the theory equation are almost same and the equation of vane pump's flow output is correct. Consequently to verify the feasibility of the new energy saving vane pump based on the theory of speed compensation. The new function of variable displacement vane pump can effectively reduce the excess output flow and the energy losses in the steering system according to the speed change. Therefore the pump has a certain effect on reducing the energy consumption and it has extensive application prospect.

ACKOWLEDGMENTS

The authors would like to thank for the support by the key discipline of Shanghai City Board of education(J50603).

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