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IMPROVEMENT OF PUMP'S TORQUE EFFICIENCY DUE TO NEW COMMUTATION UNIT DESIGN

1. New Pump with Hydrostatically Discharged Oil Commutation Unit

Research results [1] show, that a typical port plate commutation unit used in axial pumps is responsible for more than 50% of all energetic losses in such a pump. The main sources of losses are oil leakage through the clearance separating cylinder barrel from port plate and friction between those elements. To reduce the leakage, especially in modern high-pressure pumps, clearance height must be kept low. A thin oil film separates two quite large and fast moving objects and additionally those objects have to be hydrostatically unbalanced and pressed together. The result is a major friction force between them severely reducing pump's torque efficiency.

This efficiency is also affected by the pressure drop in pump's channels. The main reason for that pressure drop is not the oil flow through curved ducts, but the rapid deflection of oil stream entering or leaving the fast rotating cylinder barrel [2]. Those two combined phenomena reduce the torque efficiency of all modern axial pumps. According to catalogue data this efficiency does not exceed 95% (93% for smaller units).

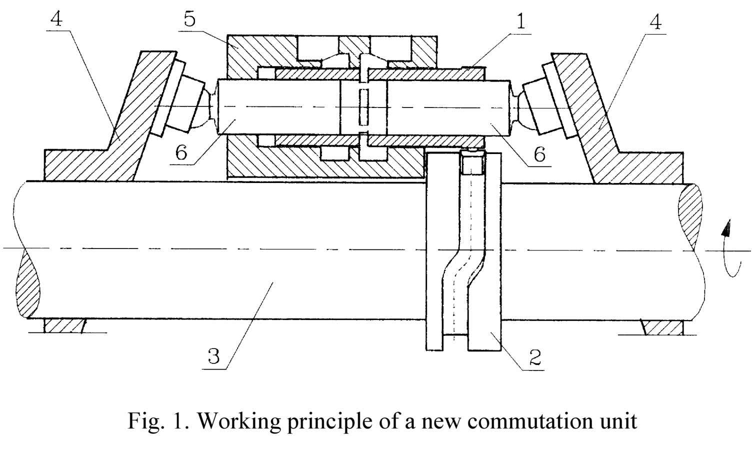

Fig. 1. Working principle of a new commutation unit

The new pump's design eliminates both sources of the energetic loss by fixing the cylinder barrel and replacing the port plate by a new commutation unit (shown in Fig 1). To make the pump with fixed cylinder barrel 5 operational its swash-plate 4 must rotate. The symmetric design with two swashplates and sets of pistons allows to discharge bearings from hydrostatic forces. The new commutation unit consists of sleeves 1 driven reciprocally by the axial cam 2 attached to the pump's shaft 3. The hole cut in the middle of each sleeve connects the working chamber alternately to low- or high-pressure channels. The working chamber is formed by the sleeve's internal surface and piston's 6 heads. This design allows to fix the height of clearance between working parts and to discharge all sleeves from hydrostatic forces.

2. Test Results

Some various prototypes of such a pump were built and tested in laboratory, their size ranged from 17 up to 68 cm^3/rev. In order to make comparisions two standard axial pumps: swashplate and bentaxis were also tested under the same conditions and using the same test equipment. Both pumps were brand new and made by world's leading manufacturers. The following diagram shows their torque efficiency values at 1500 rev/min and oil viscosity 55 cSt. Displacements of bentaxis pump A, swashplate pump B and prototype C are 28, 23 and 23 cm^3/rev respectively.

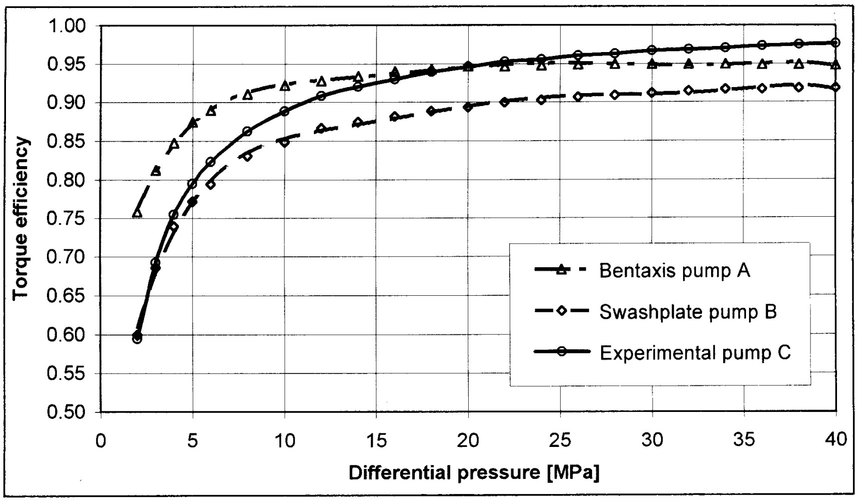

Fig. 2. Comparision of torque efficienciency values at 1500 rev/min and 55 cSt.

Results obtained for pump A comply with previous expectations, while pump B efficiency is considerably lower. Efficiency top values at 40 MPa for those pumps are 95 and 91,5%. Pump's C efficiency under the same conditions exceeds 97,5%. It means that combined torque losses in the new pump compared to pumps A and B are reduced by 55 and 75% respectively. When differential pressure was lowered to 2 MPa the efficiency of both pumps B and C dropped to 55÷60%, leaving pump A as the best performer. This is caused by the different shape of pump's C characteristic, suggesting its usefulness rather as a high-pressure unit. What's more, both pumps A and B reached their top efficiency values at 28÷30 MPa, and further pressure growth caused no change. On the contrary, pump's C efficiency grew steadily reaching 98,5% at 50 MPa.

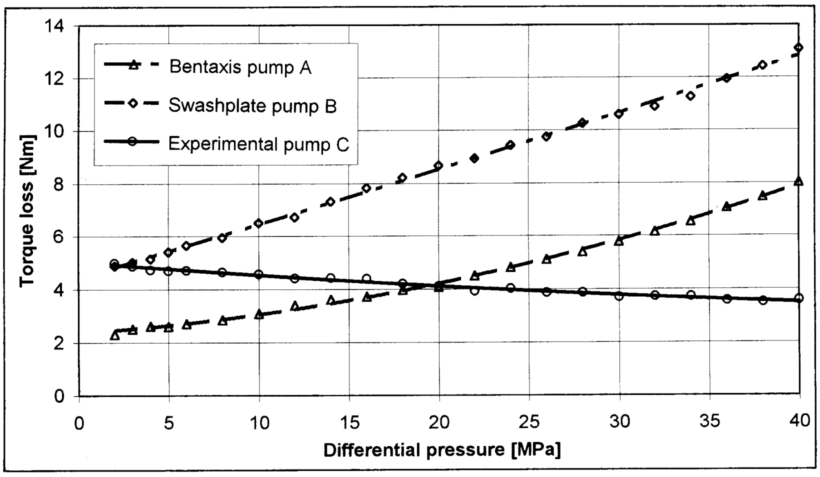

Fig. 3. Torque losses

Fig. 3 shows measured torque loss values in pressure domain. To make the comparision more precise pump's A displacement was scaled down to 23 cm^3/rev to match other pump's size.

Four main sources of torque loss in axial pumps are: friction between cylinder barrel and port plate (pumps A and B), friction between hydrostatic slippers and swashplate (pumps B and C), friction between cylinder barrel and pistons and pressure drop in pump's channels. Other sources like bearings or shaft sealing are negligible. The results shown above point at port plate and cylinder barrel as a main loss source. The hydrostatic unbalance of that unit is responsible for the quick growth of friction in pressure domain. The differences between pumps A and B can be explained by the absence of slippers sliding down the swashplate in a bentaxis pump A and by the possible difference in "the port plate unbalance ratio" between them. The pump C shows a completely different picture. The initial high value of torque loss is caused by a high number of piston-slipper units (14 versus 9 in the pumps A and B). Pressure growth however causes the slight reduction of torque loss instead of the rise. According to our previous research [3] such a behaviour of piston-slipper unit is possible and depends on the hydrostatic support's shape and dimensions. Reshaping the slipper face can cause the reduction of initial torque loss, but the friction force between the new slipper and swashplate will grow in pressure domain.

Pump's efficiency depends also on it's size. Another experimental pump C, displacing 68 cm^3/rev was tested under the same working conditions. The top value of torque efficiency for that pump at 32 MPa exceeded 99%. Since other tests and improvements of both sizes are planned, further efficiency growth is possible.

3. Conclusions

The new concept of hydrostatically discharged commutation unit and fixed cylinder barrel proved its usefulness with results exceptionally good when working pressure exceeded 20 MPa. Piston-slipper units however need some improvements to reduce the friction between slippers and swashplates under low working pressure. The reduction of the friction force may also lead to higher pump's durability.

Another advantage of full hydrostatic discharge of commutating unit parts is an ability to control variable pump's displacement directly by means of a low energy device like a proportional electromagnet or a stepping motor. Heavy and expensive hydraulic servomechanisms are no longer necessary.

Literature & References

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