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NUMERICAL ANALYSIS ON PISTON OPTIMIZATION FOR ENHANCED MASS FLOW RATE IN OIL PUMPS

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ABSTRACT

Oil pumps are used when friction occurs between components in automotive engines or compressors. This study performed piston optimization to improve the mass flow rate of oil pumps for use in linear compressors. In addition to adjusting the piston length, a taper and fluid diode was used. Under the proposed conditions, the mass flow rate improved by 19% by adjusting the piston length, 43% from using a fluid diode, and 56% from using a taper.

Keywords:oil pump, piston length, fluid diode, piston taper, optimal design, CFD.

INTRODUCTION

Oil pumps are used to lubricate engines, compressors and other systems in which friction occurs. By forming an oil film on the surface, these lubricating devices help to minimize power loss and wear between components. Moreover, they prevent high pressure gas, generated in the engine's combustion chamber and compressor's compression chamber, from slipping out through the gap between the piston and cylinder. They also act as an oil cooler, protecting the engine from high temperatures arising from explosion strokes.

Oil pumps can be classified according to operating methods. They include gear pumps, either in the form of an external gear pump or internal gear pump; rotor pumps, consisting of inner and outer rotors; vane pumps, comprised of vanes mounted to rotors; and plunger pumps, which are driven by camshafts. The oil pressure control valve on each oil pump maintains a constant pressure by opening up the relief valve when the oil pressure rapidly rises or exceeds a certain level.

While gear pumps are not as influenced by foreign substances, they tend to have lower pump efficiency and reduced durability from severe wear. Rotor pumps are not influenced by inlet/outlet pressure and oil viscosity, but cavitation during high-speed rotation causes a decrease in volumetric efficiency. Further, high-temperature and high-pressure expansion waves are generated with the bursting of cavitation bubbles, leading to more noise and erosion of metal surfaces, and eventually reduced durability [1, 2].

Over the years, many studies have been conducted on oil pumps. Frosina *et al*.[3] conducted three-dimensional CFD analysis and compared the results with experimental data for the optimal design of rotor pumps. Jing *et al*.(4) used a sliding interface and adaptive meshing technique to develop a three-dimensional transient computational model for vane pumps and gerotor pumps. Athavle*et al.*[5] developed a cavitation module for low pressure regions, enabling three-dimensional analysis according to pump rotation angle from low to high speeds. Stryczek *et al*.[6] carried out experiments to examine vortex from gear teeth during flow based on visualization of the gear pump, and Antoniak[7] employed PIV (particle image velocimetry) to examine cavitation and enhance the performance of gerotor pumps. Past research has mostly focused on reducing noise by removing gear pumps or rotor pump cavitation. However, no study exists on optimizing pistons to enhance the flow rate of oil pumps in linear compressors.

This study conducted CFD analysis to improve the mass flow rate of oil pumps in linear compressors, as shown in Figure-1. Pump performance was optimized by adjusting the piston length, and adding a taper and fluid diode within the piston.

Analysis method and conditions

The fluid flows considered in this study were two-dimensional, incompressible and unsteady flow, and the k-ε turbulence model was used for turbulent flow modeling. Pressure at the inlet and outlet was assumed to be atmospheric, and heat transfer at the walls as adiabatic. The fluid considered was water. The dynamic mesh layering technique, supported by Fluent [8], was used to describe the suction and discharge process of the piston, as shown in Figure-2. The initial piston was located at the BDC (bottom dead center) and the stroke was assumed to

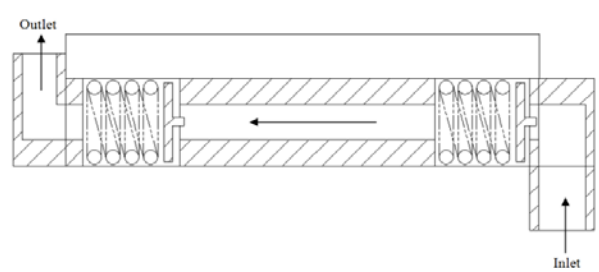

Figure-1. Schematic of an oil pump.

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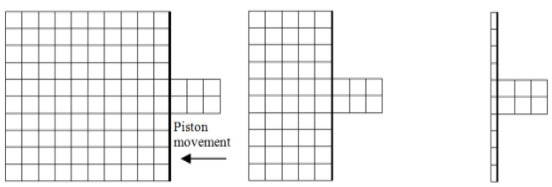
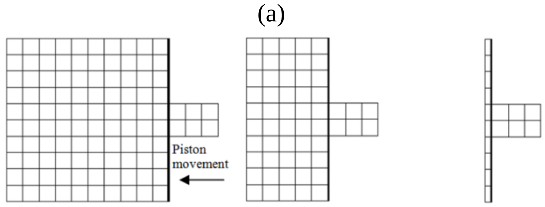
be 5 mm. In the fourth cycle, when flow stabilizes, the mean
mass flow rate was calculated using mass flow rates with piston phase angles.

Figure-3 shows the mesh system and boundary conditions of the oil pump used in the numerical analysis. Gravity acts in the -y direction, and the piston has a diameter of Ø 3.8 mm and a length of 15.7 mm. The valve opens while the piston moves from the BDC to the TDC (top dead center), and closes in the opposite direction.

Figure-4 and Table-1 present summaries of the numerical analysis models. The piston length was adjusted with reference to the baseline model, and a fluid diode was added to three different positions for analysis. To improve the mass flow rate, a taper was added within the piston. The analysis involved the baseline model with a taper angle of 5° and another model with a taper angle of 3°.The pressure difference between the inlet and outlet was defined as a subtraction of the outlet pressure from the inlet pressure. Analysis was performed at an interval of $1x10^{-5}$ seconds. The mesh consisted of 13,000 hexa and tetra cells, and the commercial program used was Fluent [8].



(a)



(b)

Figure-2. Layering mesh
(a: compression process, b: expansion process)

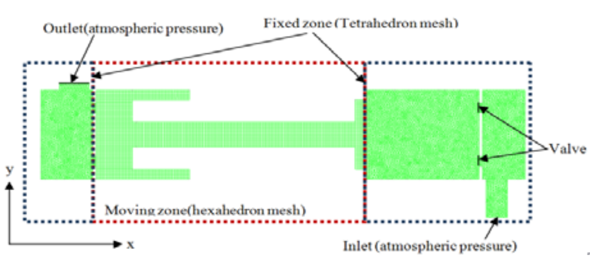
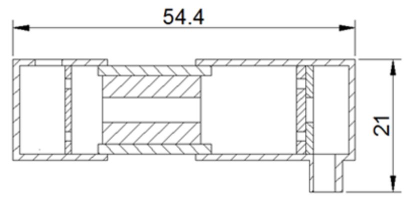
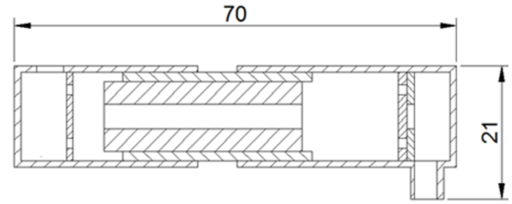


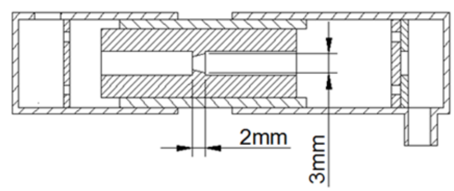
Figure-3. Mesh system and boundary conditions.



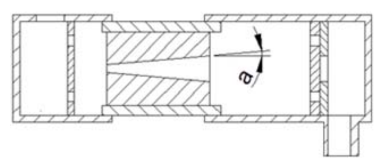
(a) Baseline model



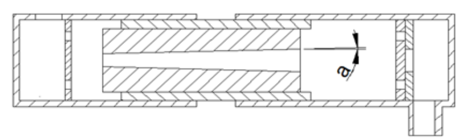
(b) Model 1



(c) Model 2,3 and 4



(b) Model 5



(e) Model 6

Figure-4. Schematic of the numerical analysis models.

Table-1. Numerical analysis models.

Models	Model description
Baseline model	Piston length 15.7 mm
Model 1	Piston length 31.4 mm
Model 2	Model 1 with front fluid diode
Model 3	Model 1 with rear fluid diode
Model 4	Model 1 with middle fluid diode
Model 5	Baseline model with taper 5°
Model 6	Model 1 with taper 3°

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RESULTS AND DISCUSSIONS

Numerical analysis of oil pumps with varying piston length

For the baseline model and model 1, Figure-5 shows the variation of inlet pressure, outlet pressure and pressure difference between inlet and outlet with piston phase angle. Here, the oil pump valve is open for a piston phase angle of 0° to 180°, and closed for angles beyond 180°. The pressure difference between the inlet and outlet is higher by an average of 55% for model 1 than the baseline model. At a piston phase angle of 0°, the pressure difference of model 1 is larger than the baseline model by 28%, but rapidly declines and eventually becomes smaller after passing the maximum flow. The initial difference is recovered before the valve closes. Model 1 show greater change in pressure difference between the inlet and outlet because its piston length is twice greater than that of the baseline model. A larger pressure difference between the inlet and outlet tends to facilitate an increase in mass flow rate.

Numerical analysis of oil pumps with fluid diodes

Figure-7 shows the variation of pressure difference between the inlet and outlet for models with fluid diodes. At a piston phase angle of 0°, when the valve first opens, model 2, model 3 and model 4 have a pressure difference larger than model 1 by 8.9%, 3.8%, and 8.7% respectively. At the point of valve opening, the highest pressure difference between the inlet and outlet is therefore observed for model 2. Model 4 shows the highest pressure difference on average, larger than that of model 1 by 60%. This implies that the additional flow resistance generated by the fluid diode leads to increased pressure difference between the inlet and outlet. Among the three models with fluid diodes, model 4 with the middle fluid diode was found to be most effective.

Figure-8 presents the variation of mass flow rate with piston phase angle for models with fluid diodes. For models with fluid diodes, maximum flow was delayed by approximately 4° - 6° compared to model 1. The most efficient model 4, with the middle fluid diode, had an

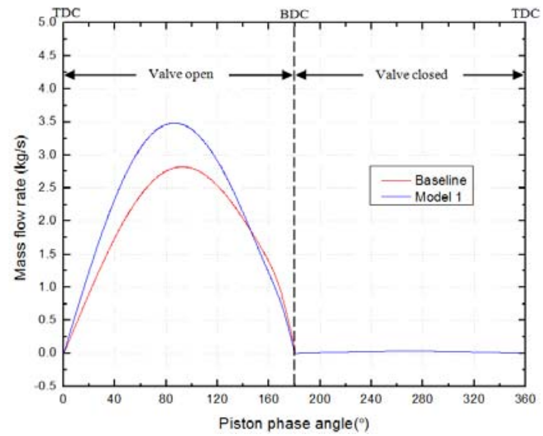

Figure-6. Variation of mass flow rate with piston phase angle for the baseline model and model 1.

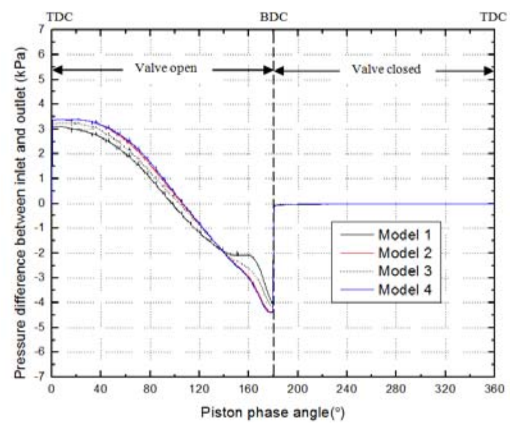

Figure-7. Variation of pressure difference between inlet and outlet with piston phase angle for model 1, 2, 3 and 4.

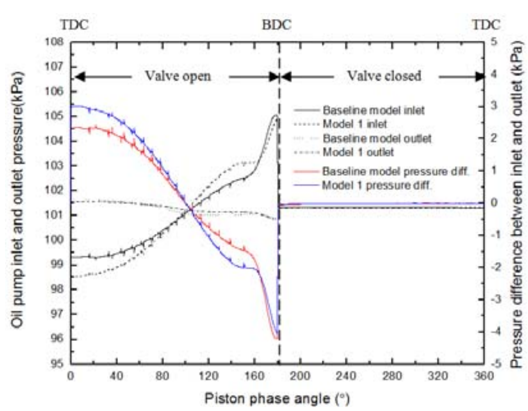

Figure-5. Variations of inlet pressure, outlet pressure and pressure difference between inlet and outlet with piston phase angle for the baseline model and model 1.

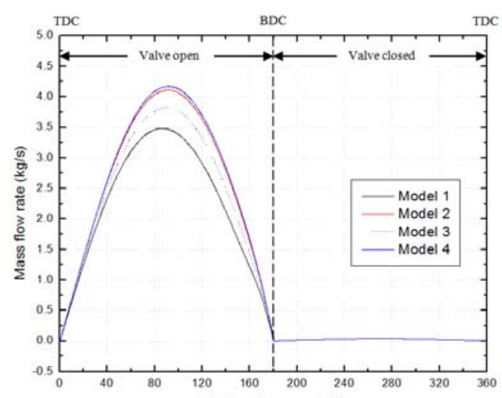

Figure-8. Variation of mass flow rate with piston phase angle for model 1,2, 3 and 4.

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increase in maximum flow of 20% compared to model 1. Model 4 also had the largest mean mass flow rate per cycle, which was 21% greater than that of Model 1. The addition of a fluid diode to the inner cylinder can thus be regarded as an effective method of improving mass flow rate.

Numerical analysis of oil pumps with tapers

Figure-9 illustrates the variation of pressure difference between the inlet and outlet with piston phase angle for models including tapers. At a piston phase angle of 0° with model 1 as a reference, model 6 has a greater pressure difference than model 5. However, as piston phase angle increases, the pressure difference between the inlet and outlet becomes smaller for model 6 compared to model 1. On the other hand, the pressure difference significantly increases for model 5 as piston phase angle increases, and the pressure difference between the inlet and outlet becomes larger on average compared to model 6.

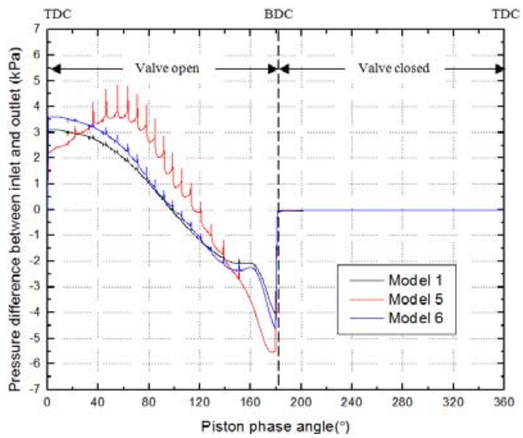

Figure-9. Variation of pressure difference between inlet and outlet with piston phase angle for model 1,5 and 6.

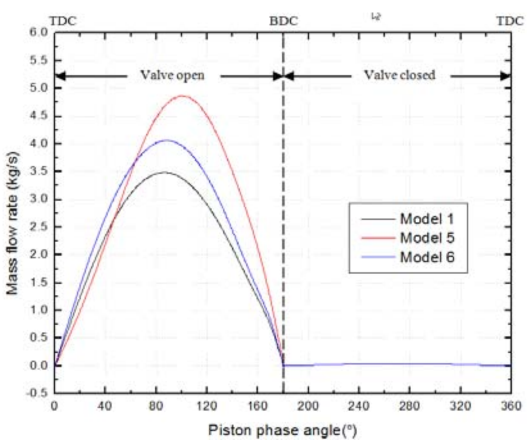

Figure-10. Variation of mass flow rate with piston phase angle for model 1, 5 and 6.

For model 5, the pressure difference between the inlet and outlet is on average 109% and 49% larger than that of model 1 and model 6, respectively. This implies that adding a taper to the inner cylinder is more effective than adjusting the piston length when it comes to improving mass flow rate. In other words, adding a taper generates greater flow resistance, which contributes to a higher mass flow rate.

Figure-10 shows the variation in mass flow rate with piston phase angle for models including tapers. Compared to model 1, the angle of maximum flow is delayed by 13° and 11° for model 5 and model 6 respectively. Model 5, which has the largest taper angle, had an increased maximum flow of 40% compared to model 1. The mean mass flow rate per cycle also increased by approximately 32% for model 5 compared to model 1. If there are spatial constraints involved in the oil pump installation, adding a taper can be more efficient than adjusting the piston length.

CONCLUSIONS

This study conducted piston optimization through CFD to improve the mass flow rate of oil pumps. The piston length was adjusted, and a fluid diode and taper were added to the piston. The following conclusions were drawn from this study.

a) The longer the piston length, the larger the pressure difference between the inlet and outlet. The mass flow rate improved by approximately 19% compared to the baseline model. As such, the length of the piston should be maximized when designing oil pumps.
b) The fluid diode is most efficient when placed in the middle of the piston, resulting in an improved mass flow rate by 43% compared to the baseline model.
c) When a taper is added to the piston, mass flow rate increases with a larger taper angle. A taper angle of 5 degrees increases the mass flow rate by 56%. If there are spatial constraints involved in oil pump installation, adding a taper may be more efficient than adjusting the piston length.

Further research is needed to examine changes in mass flow rate with varying frequency of piston motion.

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