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Abstract

Owing to the high power requirements of tractors, their low-power transmission gears often experience durability problems such as burning of the clutch. The operation of tractors under high load conditions also causes clutch slip, with the consequent longer operation duration exacerbating the burning of the friction plate. Solving this problem requires effective lubricant distribution. This was achieved in the present study by the development of an analysis model for predicting the lubricant flow rate. The reliability of the model was verified by comparing its predictions for various operation conditions with experimental measurements. Using the model, it was determined that effective distribution of the lubricant could be achieved without significant modification of the system, by only adjusting the gaps between the clutch piston and the housing, and between the separation plates and the case.

Keywords: Mechanical engineering

1. Introduction

Unlike automobiles, which usually operate on asphalt roads, tractors are operated in harsh environments where they are often stretched to their mechanical durability limits. The operation of tractors under high load conditions has prompted agricultural machinery manufacturers to vigorously pursue high performance,
which has necessitated the continuous upgrading of internal parts and the development of new components. The innovations are gradually introduced because of the impracticality of a simultaneous upgrade. Unfortunately, new and old components often do not exhibit the same performance level and this may result in compatibility problems.

Recent advancements in the power generation systems and other parts of tractors have enabled operation under higher load conditions. However, this has compounded the clutch slip problem, with the increased friction heat generated by the slip leading to burning of the clutch disc (Kim et al., 2011). Because the lubrication of the clutch is closely related to this burning problem, a lubricant supply system is included in the clutch of a tractor. The friction plate is also fabricated with a flow channel through which a hydraulic fluid is supplied for lubrication and cooling. The tractor clutch considered in the present study transmits power for engine speeds within 900–2,500 rpm. In this case, clutch slip occurs when the rotation speed of the output shaft is more than 3% lower than that of the input shaft; a smaller difference is considered as normal. Fig. 1 shows the components of the investigated clutch, which was partially burnt. The components were installed in a 50-hp medium-size tractor with an engine displacement of 2,505 cc and rated speed of 2,500 rpm.

Considering the importance of the lubricant flow rate, some experimental studies have been conducted worldwide (Lee and Kim, 2011; Kumar and Manonmani, 2011). A test conducted by Joo et al. (2011) revealed that the lubricant flow rate was the major determinant of the lifespan of the driving parts. In another study, the heat transfer was investigated with respect to the groove pattern of the clutch friction plate (Bae et al., 2014). However, the findings of all these studies would only be fully relevant if the lubricant flow rate could be accurately calculated in advance. To accurately calculate the lubricant flow rate, a precise flow network analysis model is required. Various studies have been conducted to develop a precise flow network analysis model (Lo, 1971; Tran et al., 1987; Mian, 1997; Cho

Fig. 1. Components of the clutch assembly (separation plate and friction plate).
and Yoon, 2005; Cho, 2006; Choi et al., 2012). This involved the use of elastohydrodynamic lubrication (EHL) analysis to obtain results that are more precise than those obtained by lubricant flow rate analysis using a one-dimensional (1D) analysis tool. Significant reductions of the lubricant flow rate calculation error were achieved. However, the present authors theorize that the relative errors between the lubricant flow test and analysis results may be due to the non-consideration of all the relevant parts in the analysis tool. An analysis model that included all the relevant parts and parameters such as the hydraulic control valves and pipe shapes was thus developed in the present study.

Generally, the analyses of the hydraulic circuits and flow networks are performed separately. This was the case in the above-mentioned previous studies, with in-depth investigations only conducted to reduce the errors relative to the experimental measurements. In the present study, however, a combined analysis of the hydraulic circuit and flow network was undertaken by securing the requisite knowledge in both fields. First, a 1D analysis model of the lubrication systems was developed. The pressure-drop characteristics of the lubrication lines were then determined by conducting a real vehicle test using different engine rotational speeds under high- and normal-temperature conditions of the hydraulic fluids, respectively. The reliability of the analysis model was verified by comparing its predictions with the test results. Based on the findings, a scheme for improving the performance of current systems is proposed in this paper. The ultimate purpose is to present a method for the efficient management of clutch fluid without significant design modifications.

2. Materials and methods

2.1. Development of the analysis model

Fig. 2 shows a diagram of the hydraulic circuit and a drawing of the forward/reverse clutch assembly, which were the target of the analysis model to be developed. Because the oil discharged from the pump is used to operate the steering and transmission systems and to engage and lubricate the clutch, it is not desirable to increase the lubricant flow rate itself. This is because such increase would decrease the flow to the other devices. In other words, a biased flow supply would disturb the balance of the entire system. This should be borne in mind when attempting to improve the lubricant flow rate. The analysis tool employed in this study was SimulationX®, developed by ITI in Germany.

2.1.1. Single-product analysis model

A hydraulic system consists of various parts and it is necessary to first perform single-product modeling when developing an analysis model of the hydraulic circuit. The development of such a model usually utilizes the dimensions indicated...
in the drawing of the system and the performance curve provided in the product catalog. If the physical characteristics of the parts, such as the pressure-drop, are not included in the catalog, the modeling is performed based on only the drawing. Because the system contains readymade products whose specifications are provided in the catalog, and a made-to-order valve whose specifications are not included in the catalog, both catalogue specifications and system fabrication drawing were used for the single-product modeling.

Fig. 3 is a simplified illustration of the modeling of the valves. Fig. 3(a) illustrates the modeling of the pressure control valve based on the catalog specifications, while Fig. 3(b) illustrates the modeling of the direction control valve based on the fabrication drawing. Both types of modeling involved function analysis, modeling, and normal operation confirmation, respectively. The third aspect was particularly important to ensure the reliability of the analysis model. Step 3 on the left side of Fig. 3 involves comparison of the catalogue and graph of the pressure generated when a current is applied to the analysis model to implement a flow rate through the valve and the same general conditions as those of the catalogue. Step 3 on the side right of Fig. 3 involves an analysis of the pressure drop characteristics after

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**Fig. 2.** Hydraulic circuit and forward/reverse clutch assembly.
implementing an accurate opening area by applying the lap condition in the valve assembly drawing to the analysis model.

2.1.2. System analysis model

The separate single-product analysis models of the hydraulic circuit valves were combined to obtain a complete system analysis model (Noh et al., 2014a). In the combined model, each of the hydraulic valves acted as a pressure drop source (Noh et al., 2014b). The valves thus hindered the smooth flow of the oil, and this had to be taken into consideration in the calculation of the lubricant flow rate. The locations of the valves were determined by considering their functions and the pressure drop that they caused, to ensure proper distribution of the limited discharge flow from the pump to all the systems.

Fig. 4 shows the complete analysis model of the lubrication system. The model reflects the shape of the flow channels and the operation mechanism of the clutch, as well as the difference between the gap heights before and after the clutch is engaged, and the changes in the flow rate with the engine rotational speed. The divided portion can be explained as follows. The circuit model depicts the oil discharge from the pump and the flow resistance of the hydraulic valves. The pump was included in the system model using the volumetric efficiency indicated in the product catalog. In contrast, the lubricant pipe model reflects the flow resistance due to the form factor of the rotating axis. The forward/reverse clutch model also reflects the changes in the flow path inside the clutch with respect to the coupling
state, with the portion labeled “Lub Q Balance” representing the distribution ratio of the lubricant. Experimental data was used to verify the reliability of the analysis model. The gap height between the piston and the housing was implemented in the analysis model using the ring gap library in SimulationX. The ring gap library can be used to reflect the piston eccentricity, which was set to 0 mm based on the assumption of an ideal assembly. The gap between the separation plate and the clutch case was implemented using the negative edge library, according to which a displacement of the piston reduces the gap.

Fig. 5 shows the variation of the opening area with the displacement of the clutch piston. When the clutch is engaged, the gap between the friction plate and the separation plate decreases, and the flow between the clutch pistons and the separation plate is blocked. The piston is moved by the projected net area to engage the clutch. The friction plate enhances the effectiveness of the coupling state by generating friction when the clutch is engaged. The separation plates are placed between the friction plates, and when the clutch is disengaged, a gap is created between the friction and separation plates, resulting in the elimination of the friction. The amount of lubricant in the analysis model is calculated using the orifice equation (Yoon and Jang, 2012). In the present study, in which various types of flow resistance existed, the same number of orifice equations used for the control volumes was employed. Although the procedure is basically very simple, the use of too many equations makes it more difficult to obtain a solution. A specialized analysis software was used to solve the equations in the present study.

Fig. 4. Lubricant flow analysis model of the forward/reverse clutch.
2.1.3. Reliability of the analysis model

To develop a scheme for improving a system through analysis, the reliability of the analysis model should be verified in advance. To verify the reliability of the present analysis model, a real vehicle test was conducted using different engine rotational speeds between 900 rpm (the engine idling speed) and 2,500 rpm (the maximum speed). The test was conducted between temperatures of 35 ± 2 and 75 ± 2 °C. The measurement range of the employed temperature sensor was −25–100 °C. The flow rate was measured using a turbine-type flowmeter, which is capable of measuring oil flows between 1.2 and 20 lpm over a temperature range of −20–90 °C. The employed pressure sensor had a measurement range of 0–60 bars for oil temperatures within −25–100 °C. All the measurements were recorded at 10 ms intervals. The test focused on the lubrication line by maintaining a static state in the steering and transmission systems, which share the discharge flow of the pump. The oil temperature was respectively set to 35 °C (normal condition) and 75 °C (a high-temperature condition). The flow into the lubrication system was controlled by the directional control valve (Lub v/v), framed by the continuous red rectangle in Fig. 2. When an electrical signal was sent to the directional control valve, the oil flowed into the lubrication system. The oil returned to the oil tank through the valve in the absence of the signal. Fig. 6 compares the simulation and test pressures.

Fig. 5. Variation of the opening area with the displacement of the clutch piston. (a) Names of the components. (b) Simulation result for the opening area.
on the clutch with respect to the engine rotational speed when the directional control valve is switched on and off, respectively. The directional control valve regulates the entry of the oil into the lubrication line. The point at which the pressure was measured is marked “sensor1” in Fig. 2. It is located before the pressure relief valve. The discharge flow from the pump passes through the pressure relief valve to reach the lubrication system. The pressure ahead of the pressure relief valve is thus also a factor that affects the lubricant flow rate. The test and simulation results were observed to be in good agreement for all the considered temperatures.

To verify the analysis model from other perspectives, the predicted lubricant flow rate and pressure in the lubrication line with respect to the engine rotational speed were compared with the test measurements, as shown in Fig. 7. The pressure and flow measurement point is marked “sensor2” in Fig. 2. It is located after the pressure relief valve. The prediction error was maximum when the directional control valve was switched off under the normal temperature condition. However, this was inconsequential because the focus of this study was the improvement of the lubricant flow rate.

Fig. 8 compares the analysis and test results for the pressure drop at the location of sensor2 with respect to the lubricant flow rate. Relatively good agreement can be observed despite the changes in the temperature and engine rotational speed. The reliability of the analysis model was thus verified.
3. Results and discussion

3.1. Improvement of the lubricant flow rate

The movement of the clutch pack comprises forward and reverse movements. When a tractor moves forward, pressure is applied on the forward clutch piston to engage it, while pressure is applied on the reverse clutch piston to engage it in reverse movement of the tractor. The clutch is designed such that more lubricant flows into the engaged side. Slip always occurs during the transmission of power to

![Fig. 7. Pressure and flow rate in the lubrication line with respect to the engine rotational speed (measurement point: sensor2). (a) Pressure and flow rate characteristics based on ON/OFF of the lubricant valve when the oil temperature is low. (b) Pressure and flow rate characteristics based on ON/OFF of the lubricant valve when the oil temperature is high.](http://dx.doi.org/10.1016/j.heliyon.2017.e00295)
the engaged side, even if it lasts for only an instant. The slip generates heat in the friction plate on the engaged side, and this may cause the clutch to burn in the absence of sufficient and timely lubricant supply. Of course, this does not mean that the supply of lubricant is unnecessary when the clutch is not engaged. If a tractor continuously repeats forward and reverse movements within a short time, residual heat would remain in the disengaged side of the clutch. The removal of the residual heat is thus another function that the cooling lubricant needs to perform. Effective distribution of the lubricant flow to both the engaged and disengaged sides of the clutch is thus an important factor that should be considered in improving the lubricant flow rate.

### 3.1.1. Adjustment of lubricant flow distribution ratio

The oil discharged from the pump flows into the lubrication line within the shaft after passing through the valve. Part of the lubricant flow is then supplied to the engaged clutch, entering through the hole in the clutch piston and the gap between the piston and housing, as shown in Fig. 9. The other part of the lubricant flow reaching the clutch is supplied to the disengaged clutch, where it only reaches to the gap height between the piston and housing, with a small portion also underlapping the piston hole.

The lubrication system is usually improved by replacing the directional control valve with another that produces a smaller pressure drop. Alternatively, fine
Abrasive polishing is used to decrease the surface roughness of the lubrication line. Both these methods reduce the pressure drop of the lubrication system, resulting in most of the discharge flow from the pump entering the lubrication system. Overall, excessive change in the flow rate into the lubrication system should be avoided. Moreover, the only factor that can be used to adjust the lubricant distribution ratio between the engaged and disengaged sides of the clutch is the gap between the piston and the housing.

Fig. 10 shows the relationships among the distribution ratio of the lubricant flow, the engine rotational speed, and the gap between the piston and the housing, currently designed to be 100 μm. In the conventional design, the distribution ratio of the lubricant flow between the engaged and disengaged sides is about 6:4 at a

![Diagram of lubricant flow distribution](image)

**Fig. 9.** Distribution of the lubricant flow in the clutch (magnified drawing).

**Fig. 10.** Lubricant flow distribution ratio between the engaged and disengaged sides of the clutch system. (a) When the oil temperature is low and the lubricant valve is ON. (b) When the oil temperature is high and the lubricant valve is ON.
high oil temperature (75 °C) and high engine rotational speed (2,500 rpm). Despite the fact that this condition is the most likely to generate heat in the clutch, 40% of the lubricant flows to the disengaged side of the clutch. However, it is possible to increase the amount of the lubricant flowing to the engaged side of the clutch by adjusting the gap between the piston and the housing on that side. Reduction of the gap induces a resistance to the oil flow to the disengaged clutch. The resistance also acts on the engaged clutch. However, because of the presence of a flow path for more lubricant to the engaged clutch piston (indicated by a hole in Fig. 9), the resistance to the flow has little effect on the engaged clutch. In other words, reduction of the gap between the piston and the housing decreases the amount of lubricant flowing into the disengaged clutch, thereby allowing more lubricant to enter the flow path to the engaged clutch. Fig. 11 shows the relationships among the lubricant flow rate, pressure drop in the lubrication system, rotational speed,

Fig. 11. Relationships among the lubricant flow rate, pressure drop in the lubrication system, engine rotational speed, and gap between the piston and housing. (a) When the oil temperature is low and the lubricant valve is ON. (b) When the oil temperature is high and the lubricant valve is ON.
and gap between the piston and housing. Fig. 11(a) shows how the pressure drop in the lubrication line varies with the adjustment of the gap between the piston and the housing. As can be seen, an increase in the flow resistance resulting from a reduction of the gap causes the pressure drop to increase. Fig. 11(b) shows the variation of the pressure generated in the lubrication line with the engine speed. Because a reduction of the gap increases the flow resistance, it also slightly increases the pressure generated in the lubrication line. Fig. 11(c) shows the total lubricant consumption with respect to the engine speed. There is no significant change (only a slight decrease) in the total lubricant consumption with reduction of the gap. The surplus flow entering the lubrication system and not utilized by the forward/reverse clutch is used to lubricate the subtransmission and other working parts. The analysis results show that the total flow into the lubrication system slightly decreases with increasing pressure drop. In other words, the surplus flow increases. It is also evident from the results that the lubrication distribution ratio between the forward and reverse clutch plates can be effectively varied by simply adjusting the gaps between the piston and the housing on both sides. A design that enhances the lubrication of other working parts can also be achieved by increasing the surplus flow.

3.1.2. Improvement of the lubricant flow distribution inside the clutch

As noted earlier, the flow of lubricant to the disengaged clutch is also necessary. Let us assume that the gear is shifted into reverse after a period of forward movement. In this case, the residual heat in the forward clutch has to be reduced during the reverse movement. In current designs, the lubricant that enters the disengaged clutch is inefficiently distributed inside the clutch. As shown in Fig. 12, most of the lubricant flows between the separation plate and the clutch case, which

![Fig. 12. Distribution of the lubricant flow inside the clutch.](image)
are not related to the cooling of the friction plate. In other words, the gap between the separation plate and the clutch case should be reduced to maximize the effect of the lubricant flow distribution, as described earlier.

**Fig. 13** shows the variation of the lubricant flow rate when the gap between the separation plate and the clutch case, and that between the piston and the housing are simultaneously reduced, as proposed above. **Fig. 13(a)** shows the variation of the lubricant flow rate in the clutch with the engine rotational speed. The proposed modification decreases the lubricant flow into the lubrication line. **Fig. 13(b)** shows

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**Fig. 13.** Improvement of the lubricant flow distribution in the clutch by adjusting the gap between the friction and separation plates (a) Lubricant flow rate with respect to the engine rotational speed (b) Variation of the lubricant flow distribution to the disengaged clutch friction plate and (c) Variation of the lubricant flow rate to the engaged clutch friction plate.
the variation of the lubricant flow distribution to the disengaged side of the clutch with the gap between the separation plate and the case. It can be seen that the lubricant flow inside the clutch is concentrated in the flow channel of the friction plate and the gap between the friction and separation plates. Fig. 13(c) indicates an increase in the flow rate through the flow channel of the friction plate of the engaged clutch with increasing engine rotational speed. It can be concluded from the results in Fig. 13 that the lubrication system can be improved by simply adjusting the gaps between the piston and the housing, and between the separation plates and the case, without redesigning the system.

4. Conclusion

An analysis model of the lubrication system of the forward/reverse clutch of a tractor was developed in this study, as well as a scheme for improving the system using the model. Following is a summary of the study and the findings:

1. The developed analysis model included all the parts that affected the oil flow and was confirmed to effectively represent the lubricant flow.
2. The results of a real vehicle test confirmed the reliability of the analysis model from various perspectives, including with respect to the variation of the engine rotational speed and oil temperature.
3. It was confirmed that the distribution ratio of the lubricant flow between the engaged and disengaged sides of the clutch could be changed by adjusting the gap between the piston and housing. This enabled the achievement of effective lubricant flow.
4. To eliminate the residual heat from the disengaged side of the clutch, a sufficient amount of lubricant was channeled to cool the disengaged friction plate by adjusting the gap between the separation plate and clutch case.

The results of this study showed that variation of the gaps between the piston and the housing, and between the separation plates and the case (minor design modifications) could be used to achieve efficient lubricant flow in the lubrication system of a tractor forward/reverse clutch. This constitutes an effective method for preventing burning of the clutch when operating a tractor under harsh conditions. Further study is, however, necessary to analyze the effectiveness of the improved lubricant flow rate with respect to heat dissipation from the clutch friction plate.

Declarations

Author contribution statement

Daekyung Noh: Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data; Wrote the paper.
Soochul Kim: Conceived and designed the experiments; Performed the experiments.

Yongjoo Kim: Conceived and designed the experiments.

Joosup Jang: Contributed reagents, materials, analysis tools or data.

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The authors declare no conflict of interest.

**Additional information**

No additional information is available for this paper.

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