Evaluation of Oil Supply System for Rotary Compressor using Two-phase flow analysis

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ABSTRACT

The oil supply system is a key factor that greatly affects the performance of a rotary compressor. Therefore, verification of the system is required when designing the compressor. However, it is difficult to perform experimental evaluation of the oil supply system due to the characteristics of the rotary compressor such as high pressure and closed structure. In this study, numerical analysis schemes for evaluating the oil supply system was established. The Volume of Fluid(VOF) method was applied to simulate the phase separation phenomenon for the operating fluid, and unsteady-state analysis was performed to verify the supply characteristics over time. In addition, the loss coefficient at each outlet was defined to reflect the effect according to the oil supply flow path shape. The reliability of the established analysis schemes was verified by comparing the results of the analysis and experiment in the unloaded state. The error range of the analysis and experiment was within 6.5% and showed very high reliability. Finally, numerical analysis of the oil supply system under actual load conditions was performed to predict the supply flow rate at the time of compressor operation. In addition, performance impact tests were performed according to the assembly characteristics, and the cause of the loss was analyzed through internal flow visualization.

1. INTRODUCTION

Lubrication has a great influence on the performance and reliability of a hermetic compressor such as a rotary compressor. As shown in Fig. 1, the rotary compressor supplies oil to the compression and bearing part of the pump through the flow path inside the shaft. If oil is not smoothly supplied into the compression part, the sealing ability at the compression part decreases, causing cooling capacity loss. In addition, if the oil supply of the bearing part is insufficient, it causes reliability problems such as damage or an increase in input power. Due to the importance of the oil supply system, related studies have been steadily conducted.

In the past, studies were mainly conducted to predict oil supply in compressors in theoretical and experimental ways (Asanuma et al., 1984, Fukuta et al., 1996, Kim et al., 2000). However, these experimental studies had limitations in that they had to be reviewed under no-load operating condition due to the characteristics of a hermetic compressor. Recently, as various multiphase fluid analysis schemes have been developed and computing power has improved rapidly, research on oil supply systems using computational fluid dynamics is being actively performed (Zhai et al., 2008, Liu et al., 2012, Yanzhen et al., 2018, Zhu et al., 2019). As research using numerical analysis was conducted, it became possible to evaluate the performance of the oil supply system in consideration of actual operating conditions. However, since the numerical analysis technique and the way of evaluation can vary depending on the compressor type or specification, it is important to apply an appropriate numerical analysis method to the compressor to be evaluated. In this study, the suitability evaluation for the oil supply system of the rotary compressor currently being developed was performed using two-phase flow analysis technique. A numerical analysis technique for two-phase flow was established, and the reliability of the established numerical analysis technique was verified through comparison with oil supply test data. Finally, the performance of the oil supply system for actual operating conditions was evaluated using numerical analysis, and the effect of the shape, deformation, installation state of the oil pick-up module on the lubrication performance was verified.

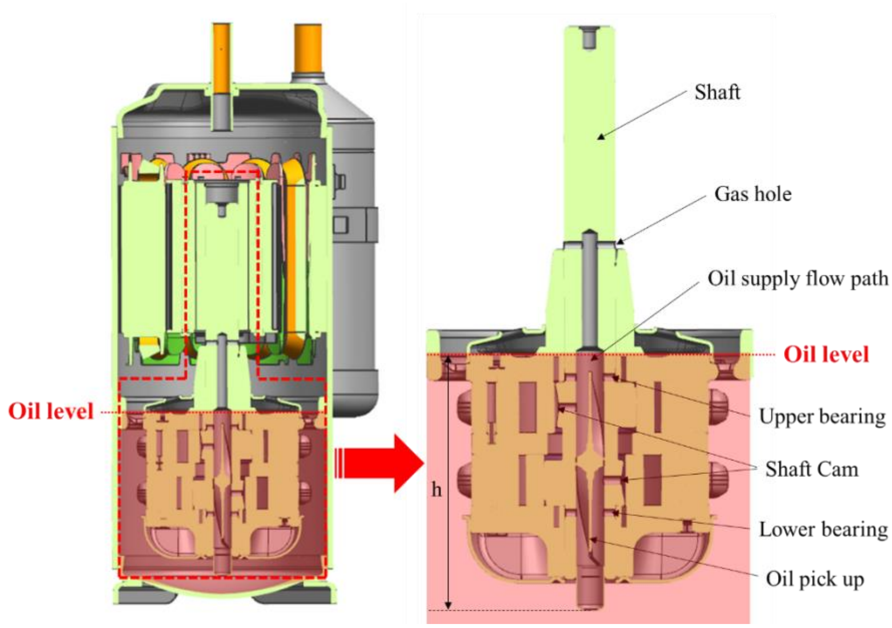

Figure 1: Schematic diagram of the oil supply system in a rotary compressor

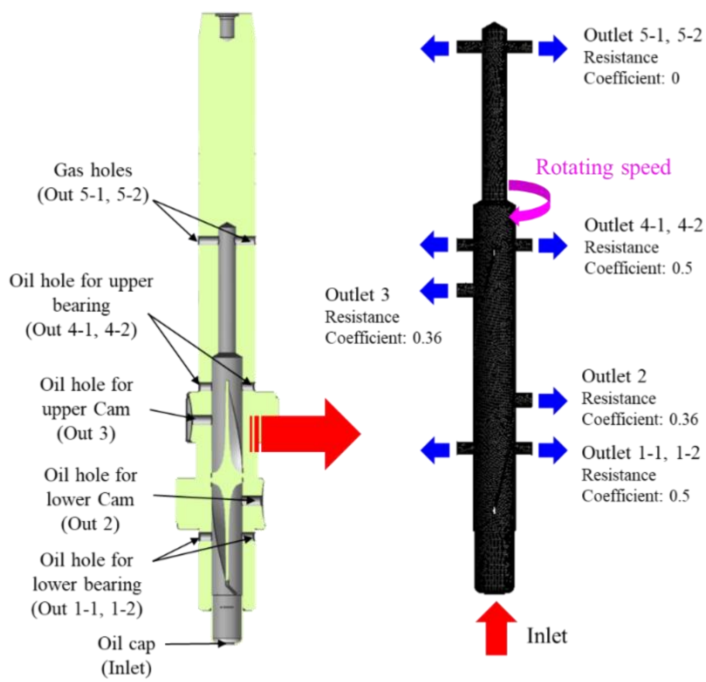

Figure 2: Computational methods to evaluate the oil supply system

2. COMPUTATIONAL METHODS

As shown in Fig. 2, a computational domain for the oil supply system inside the shaft was generated. The grid system consists of unstructured grid, and a prism mesh was applied near the wall. The total number of nodes generated is approximately 170,000. ANSYS CFX was used to perform the numerical analysis about the oil supply system. In order to evaluate the oil supply over time, Transient analysis was applied. The standard k-ε model was applied as a turbulence model for flow. A homogeneous model was used to predict two-phase flow. This model is assumed that the velocity of each phase is the same, and the governing equation for each phase is defined as follows (Soares et al., 2013).

Mass Conservation:

$$\frac{\partial(\rho_R f_R)}{\partial t} + \nabla \cdot (\rho_R f_R u) = 0 \tag{1}$$

$$\frac{\partial(\rho_O f_O)}{\partial t} + \nabla \cdot (\rho_O f_O u) = 0 \tag{2}$$

Momentum Conservation:

$$\frac{\partial(\rho u)}{\partial t} + \nabla \cdot (\rho u u) = -\nabla p - \nabla \cdot (\tau + \overline{\rho u' u'}) + \rho g \tag{3}$$

where ρ_R is density of the refrigerant, ρ_O is density of the oil, u is velocity vector.

Opening pressure condition was assigned to the inlet and outlet as a boundary condition. In the case of the rotary compressor, which is the subject of this study, oil is filled up to the height of the top flange surface(h) as shown in Fig. 1. Accordingly, the inlet pressure can be calculated by the definition of the head pressure as follows.

$$P_{inlet} = \rho_O g h \tag{4}$$

The rear shape of each oil discharge hole outlet is different. Therefore, it is necessary to take into account the pressure loss caused by the flow path to apply appropriate outlet pressure conditions. In this study, the concept of abrupt contraction was introduced (Weisbach J., 1855). The head loss coefficient(ζ) in abrupt contraction was calculated as follows.

$$H_L = \zeta \frac{{V_2}^2}{2g} = \zeta_a \frac{{V_c}^2}{2g} + \frac{(V_c - V_2)^2}{2g} \tag{5}$$

where V_2 is flow velocity at rear point, V_c is flow velocity at contraction point. The operating condition of the compressor was ASHRAE-T, and the rotating speed was 60rps. R134a refrigerant and POE oil were applied as working fluids. Since the initial operating condition was targeted when performing the numerical analysis, the refrigerant and oil were assumed to be completely separated, and chemical properties such as solubility were not considered.

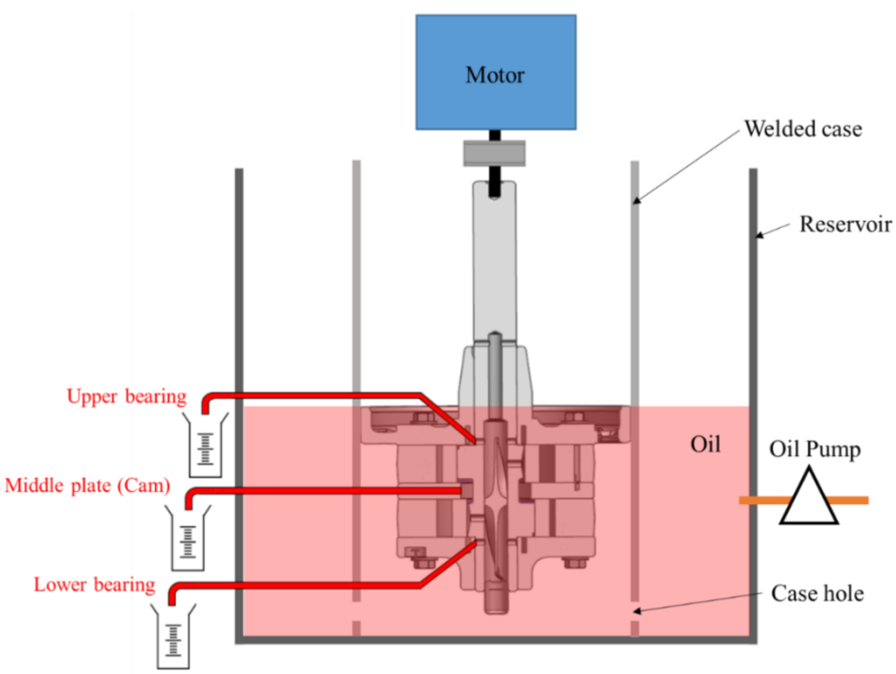

Figure 3: Schematic diagram of the experimental apparatus

Table 1: Evaluation result of the oil flow rate (base model)

<table>
<tr><th colspan="2">Base model</th><th>Lower bearing</th><th>Lower CAM</th><th>Upper CAM</th><th>Upper bearing</th><th>Gas hole</th></tr>
<tr><td rowspan="5">Oil flow rate (Dimensionless)</td><td>Experiment / 60 [rps] (1atm, 35 °C)</td><td>0.197</td><td colspan="2">0.523</td><td>0.280</td><td>No data</td></tr>
<tr><td rowspan="2">CFD / 60 [rps] (1atm, 35 °C)</td><td rowspan="2">0.205</td><td>0.183</td><td>0.368</td><td rowspan="2">0.309</td><td rowspan="2">0</td></tr>
<tr><td colspan="2">0.551</td></tr>
<tr><td rowspan="2">CFD / 60 [rps] (34.52 Kgf/㎠ A, 101 °C)</td><td rowspan="2">0.219</td><td>0.203</td><td>0.424</td><td rowspan="2">0.366</td><td rowspan="2">0.097</td></tr>
<tr><td colspan="2">0.627</td></tr>
</table>

3. EXPERIMENTAL METHODS

To verify reliability of the numerical analysis, an oil supply test was performed on the pump unit of the compressor. A schematic diagram of operating and flow measurement is shown in Fig. 3. A pump fixed to the welded case was installed in the reservoir and immersed in oil to the top flange. Holes with a diameter of 5 mm were created in the middle plate, and the boss area of top and bottom flange respectively, and each hole was connected to a pipe to measure the oil flow rate. In this study, all oil supply data were dimensionless based on the total flow rate calculated by the experiment.

The shaft key groove was connected to the motor shaft using flexible coupling to operate the pump. The oil flow rate was measured for 20 seconds, and the average value was calculated as the oil supply amount. In addition, an oil pump was installed to maintain the initial oil surface height so that oil was continuously supplied to the reservoir.

The oil supply test was performed under atmospheric pressure due to the limitations of the experimental facility configuration. When the compressor is operating, the internal pressure is very high. However, it was judged that the influence of the reference pressure would not be significant because oil is supplied due to the internal pressure difference. The change in the properties of the refrigerant and oil due to pressure and temperature can be an error factor of the result.

4. RESULTS AND DISCUSSION

4.1 Validation Results for Two-phase Flow Analysis

Table 1 shows the results of oil supply performance by experiment and numerical analysis. The amount of oil supply predicted by the numerical analysis changes over time because the unsteady state analysis is performed in this study. Accordingly, after the oil supply became stabilized, the average value for 0.1 seconds was calculated as the analysis result. In addition, numerical analysis under actual load conditions was performed to confirm the influence of oil supply performance depending on conditions such as pressure and temperature.

As seen from the results, oil supply was smooth in the CAM region, where the area of the flow path after the oil discharge hole was less reduced than in the bearing region. It means that the head loss of the oil discharge area has a great influence on the oil supply performance. Also, the amount of oil supply in the oil discharge hole in a relatively high position was higher. An axial velocity component is rapid as oil flows in near the lower oil cap. As a result, it was judged that the amount of oil supply decreases due to the relatively weak pressure acting in the radial direction in the oil discharge hole existing below. This tendency was the same in both the experimental and numerical results. The error range of numerical analysis was within about 6.5%, which means high reliability.

The oil supply tendency under actual operating condition showed a slight difference from the result under no-load condition. The total amount of oil supply under actual operating condition increased by about 13.8% compared to the result under no-load condition. In particular, the higher the position of the oil discharge hole, the greater the increase in the amount of oil supply. This difference in oil supply tendency is judged to be due to change in physical properties of oil and refrigerant because of the increase of internal pressure and temperature.

Table 2: Evaluation result of the oil flow rate (test case)

Effect review case		**Lower bearing**	**Lower CAM**	**Upper CAM**	**Upper bearing**	**Total**	**Gas hole**
Oil flow rate (Dimensionless)	Base model	0.219	0.203	0.424	0.366	1.212	0.097
	Case 1	0.168	0.142	0.397	0.325	1.032	0.139
	Case 2	0.221	0.386	0.451	0.394	1.452	0

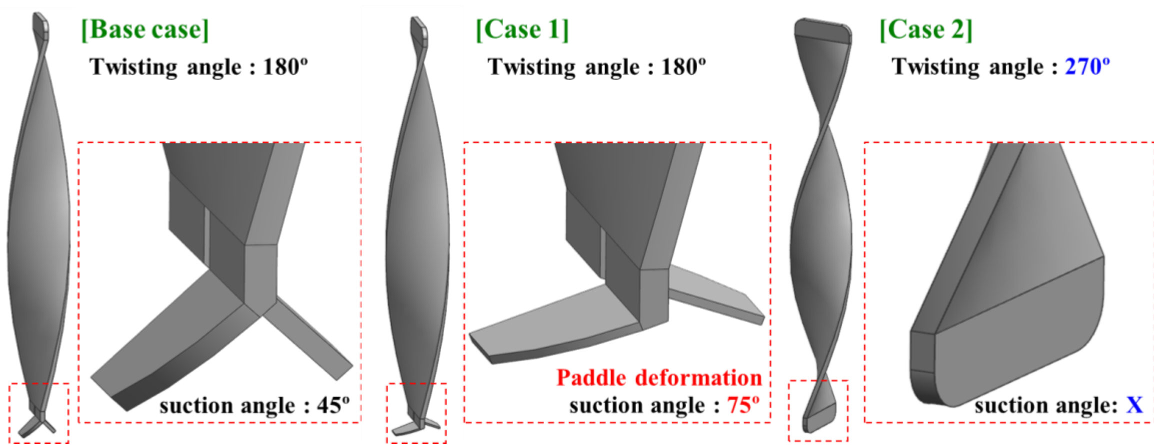

Figure 4: Review case for the oil pick-up module

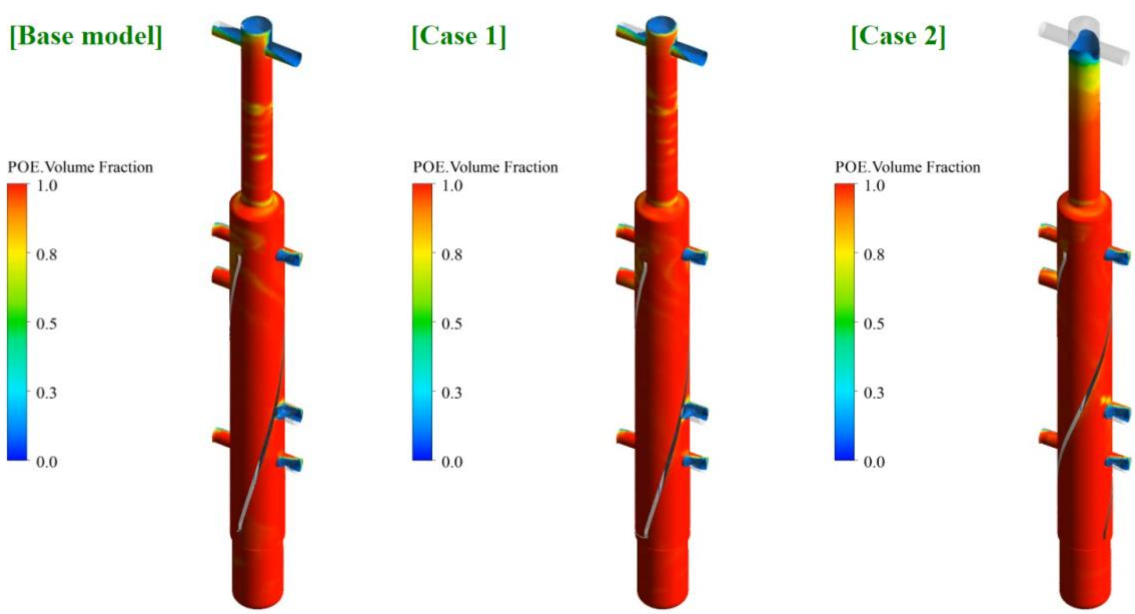

Figure 5: Oil volume fraction (1 second after operating)

4.2 Effect of Oil Pick-up Module

Oil supply inside the rotary compressor is basically made by the head of the storage oil, and an oil paddle is installed inside the shaft to increase the oil supply speed at the beginning of operation. In order to examine the effect of oil paddle on oil supply performance, two additional review cases were selected as shown in Fig. 4. Case 1 is a review model that applies defect factors that can occur during product assembly, and it is deformed on the front of the oil paddle. In addition, it was assumed that the insertion angle of the oil paddle was twisted by 5º, resulting in interference with some oil discharge holes. Case 2 is a review model to improve the defect factors that can occur, and to prevent deformation the shape of the front part of the oil paddle has been changed. Also, by increasing the twisting angle of the oil paddle, the interference with all oil discharge holes was prevented.

Table 2 is the evaluation results of the oil supply performance for each review case, and Fig. 5 is the oil volume fraction distribution of the internal flow path at the time of stabilizing oil supply after operation (1 second after operation). In case 1, the total oil supply flow rate decreased by about 14.9% compared to the base model. In particular, the decrease ranges in the oil discharge holes existing in the lower area were large. In addition, it was expected that the amount of oil discharged from the gas hole would increase, which would adversely affect the OCR. On the other hand, in case 2, the total oil supply flow rate increased by about 19.8% compared to the base model. The effect of improving the oil supply flow rate in the oil discharge holes existing in the lower area was great. In addition, as oil supply from the overall oil discharge hole was smoothly carried out, oil discharge from the gas hole was suppressed.

Internal flow field analysis was performed to determine the cause of the difference in the oil supply characteristics. Figure 6 is the comparison result of the flow velocity distribution in the suction region of the oil paddle, and Fig. 7 shows the oil volume fraction distribution in the lower area of the oil supply system. In case 1, flow velocity loss occurred as the suction flow path area decreased due to deformation of the front part of the oil paddle. It means energy loss in the suction area. In addition, the proportion of discharged oil at the lower CAM decreased sharply due to interference between the oil discharge hole and the oil paddle. In case 2, the loss of suction flow velocity decreased significantly as the front part of the oil paddle was changed to a straight shape. In addition, as the interference section was removed by changing the twisting angle of the oil peddle, the proportion of discharge oil at the lower CAM also increased.

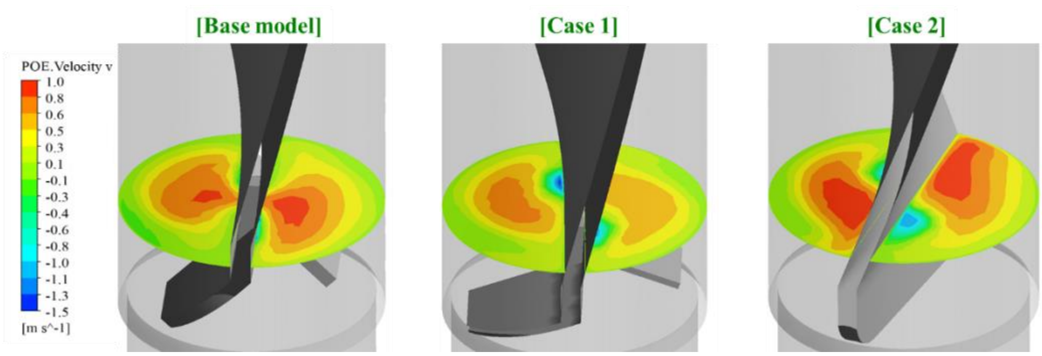

Figure 6: Flow velocity contour at suction part of the oil paddle

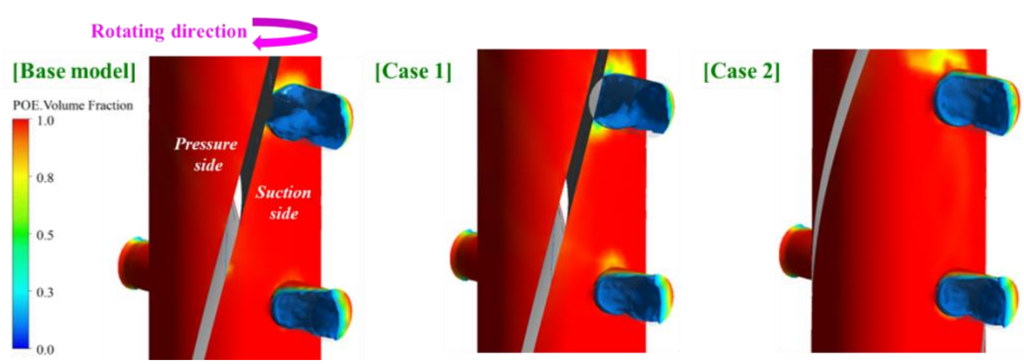

Figure 7: Oil volume fraction in lower part

5. CONCLUSIONS

In this study, a numerical analysis schemes were established to evaluate the oil supply performance of the rotary compressor, and reliability verification was performed on the established numerical methods. In addition, the performance of the review models was evaluated using numerical analysis to review the design impact of the oil supply system. The results are as follows.

- in order to perform numerical analysis of the two-phase flow, numerical analysis technique based on the homogenous model was constructed. The concept of the abrupt contrast loss was applied to the oil discharge holes so that the loss due to the change in the flow path could be simulated. The constructed numerical analysis technique verified reliability through comparison of actual experiments and numerical data. As a result, the error range of the numerical analysis was within about 6.5%, indicating a high reliability level.
- In case 1(problem case), the overall oil supply flow rate decreased due to the energy loss of the suction region. In particular, the oil supply flow rate at the point where interference between the oil discharge hole and the oil paddle occurs was greatly decreased.
- In case 2 (improve case), the energy loss decreased due to the change in the suction shape. As a result, the overall oil supply flow rate increased by about 19.8%. In addition, the twisting angle of the oil paddle was adjusted to suppress interference between the oil discharge hole and the oil paddle, thereby improving the reduction of the oil supply performance.
- In the future, the review technique for the oil supply system established through this study will be applied to various development models.

NOMENCLATURE

f	volume fraction	(–)
g	gravitational acceleration	(m/s^2)
H_L	head loss	(Pa)
P_{inlet}	inlet pressure	(Pa)
t	time	(s)
u	velocity vector	(m/s)
V_2	flow velocity at rear point	(m/s)
V_c	flow velocity at contraction point	(m/s)
ρ	density of the fluid	(kg/m^3)
τ	viscous stress	(N/m^2)
ζ	loss coefficient	(-)

Subscript

O	Oil
R	Refrigerant

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Weisbach, J. (1855). Die experimental-Hydraulik: eine Anleitung zur Ausfuhrung hydraulischer Versuche im Kleinen, nebst Beschreibung der hierzu nothigen Apparate und Entwickelung der wichtigsten Grundformeln der Hydraulik, so wie Vergleichung der durchdiese Apparate gefundenen Versuchsresultate mit der Theorie und mit den Erfahrungen im Grossen. *JG Engelhardt*.