EXPERIMENTAL AND NUMERICAL INVESTIGATION OF ACTIVE HEAT EXCHANGE FOR FLUID POWER SYSTEMS

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ABSTRACT

A new heat-exchanging device for fluid power systems are proposed and developed. Numerical analysis of flow in the heat-exchanging system is carried out to investigate the influence of mixing and separating air bubbles to working fluids. Performance of the system is experimentally evaluated by measurement of fluid temperature rise in a reservoir of test hydraulic circuits.

1 INTRODUCTION

Recent trend in industrial manufacturing is for fluid power systems, as well as fluid power components, to be designed in a more compact fashion. The benefits of smaller fluid power systems are obvious --- economy of materials, less energy consumption, less square-footage required. But one often overlooked drawback is the increased probability of air entrainment for fluid power systems that incorporate smaller reservoirs.

Bubbles entrained in the oil can create many problems in fluid power systems, such as acceleration of oil degradation by oxidation, decreasing lubricity caused by air emulsion, reduction of thermal conductivity, cavitation erosion, higher noise emission, increase in compressibility and decrease in dynamic characteristics, and decrease in pump output efficiency. When bubbles in oil are compressed adiabatically at high pressure, the temperature of the bubbles rises sharply, and the surrounding fluid temperature also rises [3][4][7]. Thus, it is important to eliminate the bubbles from the oil to preserve oil quality, system performance, and to avoid possible damage of the components.

Recently, one of the authors [5] has developed a new device using swirl flow for bubble elimination capable of eliminating bubbles and of decreasing dissolved gases. This device is called “Bubble Eliminator”. Using the bubble eliminator will enable the fluid power system to perform better. In our previous study, we have reported that the bubble eliminator is useful for preventing oil temperature rise caused by the bubbles under low, moderate and high system pressure conditions [2]. Numerical analysis and flow visualization have been performed for clarifying swirl flow characteristics and pressure distributions [1]. It has been experimentally and numerically confirmed that the bubble eliminator has been useful for removing tiny cavitation bubbles from the hydraulic working fluid for fluid power systems [6].

During last few years a new device for mixing gases and liquids is also developed by one of the author’s company, Ihara Science Co. We also call this device a static mixer. The static mixer can be effectively mixing up gasses and liquids by flow potential without supply power. The bubble eliminator and the static mixer are introduced and encouraged to develop a new environmental friendly device for fluid power systems.

An environmental protection movement has gained momentum under the global issue. In fluid power systems, loss of power turns to thermal energy to heat working fluid and system components. For most fluid power systems, heat exchanger is essential to reduce fluid temperature. In this paper, a new heat-exchanging device for fluid power systems are proposed and developed by using a principle of the static mixer and the bubble eliminator. We call this system an active heat exchanger. Flow pattern in the static mixer and the bubble eliminator has great influence on thermal efficiency of the active heat exchanger. Numerical analysis of the flow in the static mixer and the bubble eliminator is carried out to investigate the influence of mixing and separating the air to the oil. Performance of the system is experimentally evaluated by measurement of fluid temperature rise in a reservoir of test hydraulic circuits.

2 HEAT EXCHANGING SYSTEM

Figure 1 shows the fundamental structure of the active heat-exchanging system. The heat-exchanging system consists of an injection, mixing and elimination parts. Firstly, surrounding air is directly and occasionally infused in oil line at the injection parts. Secondary, the air is efficiently mixed with the oil by the static mixer at the mixing part. Thermal energy of the oil is transferred to the adjoining air through the surface of the air bubbles in the oil. The oil is cooling down at the mixing part and the temperature rise of the oil is restrained. Finally, the air bubbles with thermal energy are separated from the oil by the bubble eliminator at the elimination

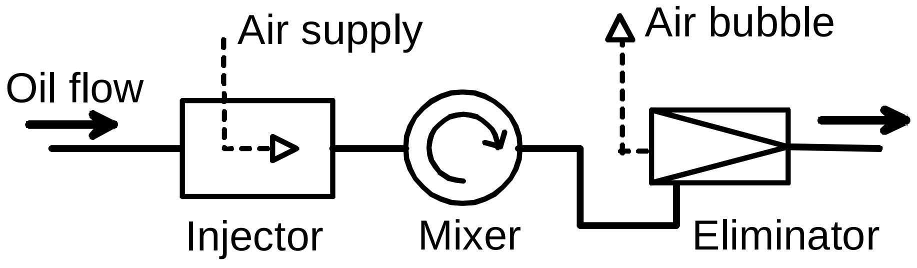
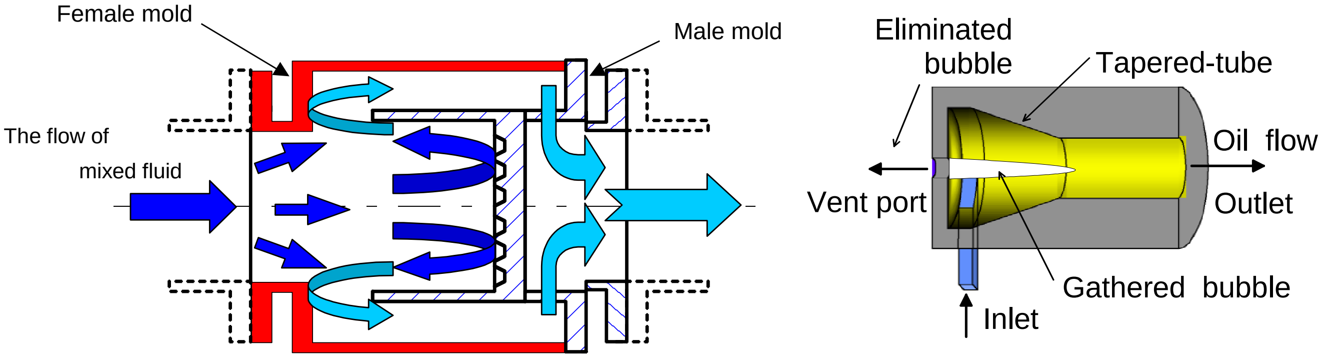


Fig.1- Heat exchange system



(a) Static mixer

(b) Bubble eliminator

Fig.2- Principle of static mixer and bubble eliminator

part. The active heat exchanger is able to prevent the temperature of the working fluid from rising with flow and pressure.

Figure 2 (a) illustrates the principle of the static mixer. The static mixer consists of a cuplike part with dimples and a pair of male and female molding joints. The oil with bubbles flows through the inlet and collide with the cuplike part. Since the oil flows reversely and windingly, the oil is mixing with small air bubbles. By mixing process, thermal energy with the oil is promoted to transfer to the air through the surface of the bubbles.

Figure 2 (b) illustrates the construction of the bubble eliminator. The bubble eliminator consists of a tapered-tube that is designed such that a chamber of circular cross-section becomes smaller, and a connected a cylindrical straight tube chamber. Oil with bubbles flows tangentially into the tapered tube from an inlet port and forms a swirl flow that circulates fluid through the flow passage. The swirl flow accelerates downstream, and the fluid pressure along the central axis decreases downstream. From the end of the tapered tube, the swirl flow decelerates downstream and the pressure increases towards the outlet port. There are certain position-dependent centrifugal forces created in all parts of the swirl flow, and the bubbles tend to move toward the central axis of the tube due to the difference in centrifugal force between the oil and the bubbles. Bubbles are trapped in the vicinity of the central axis and collected near the area where the pressure is lowest. When back pressure is applied by a check valve or an orifice located at the downstream side of the bubble eliminator, the collected bubbles are ejected through a vent port. The air bubbles with thermal energy are separated and eliminated from the oil by the bubble eliminator. In our previous paper [3], it has been experimentally verified that the bubble eliminator is effective in reducing the oil temperature rise.

3 NUMERICAL ANALYSIS

Flow pattern in the static mixer and the bubble eliminator has great influence on thermal efficiency of the active heat exchanger. We perform a three-dimensional flow analysis for the static mixer and the bubble eliminator to investigate the influence of mixing and separating the air to the oil, using the commercially available numerical calculation software; STAR-LT to investigate the influence of mixing and separating the air to the oil.

The basic equations for the numerical analysis consist of Navier-Stokes equation, the equation of continuity, the equations of motion and the energy equations. The numerical simulation has been performed for conditions of working fluids having kinetic viscosity of 32 mm^2/s, a density of 862 kg/m^3 and flow rate condition of 20 L/min. Numerical analysis of the flow in the static mixer and the bubble eliminator is carried out for single-phase and multi-phase flows.

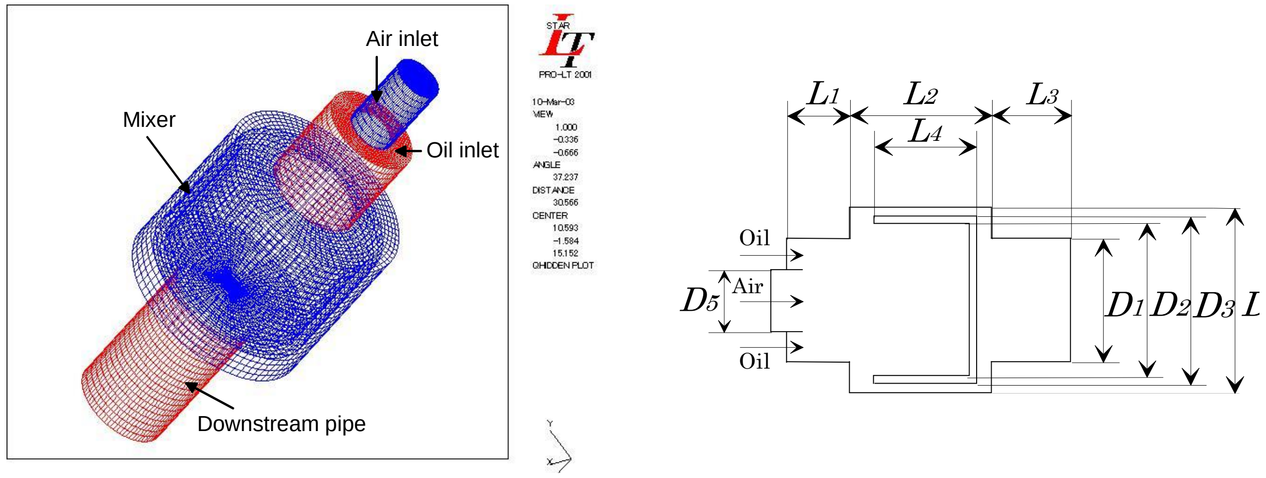


Fig.3- Mesh definition and dimensional drawing of static mixer

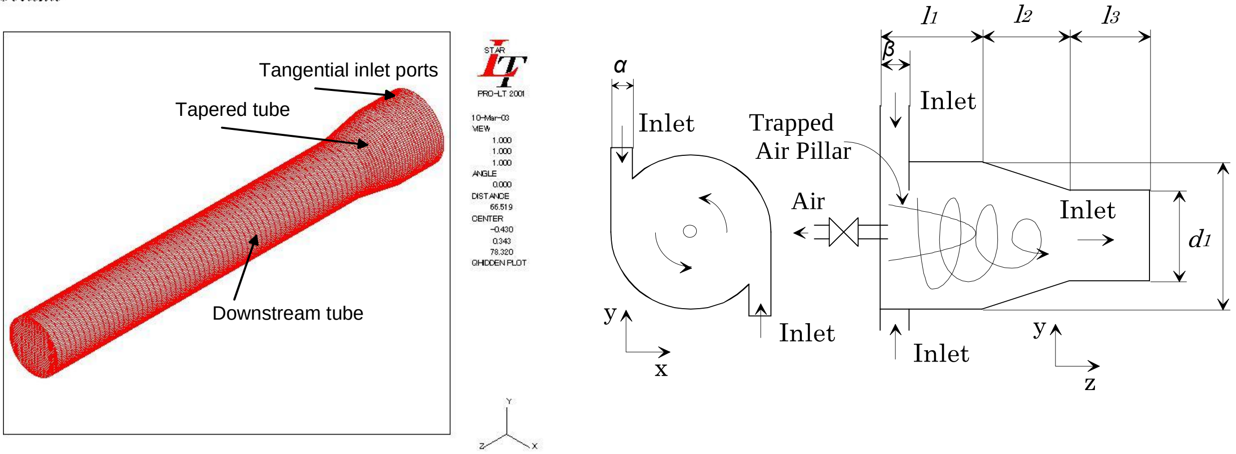


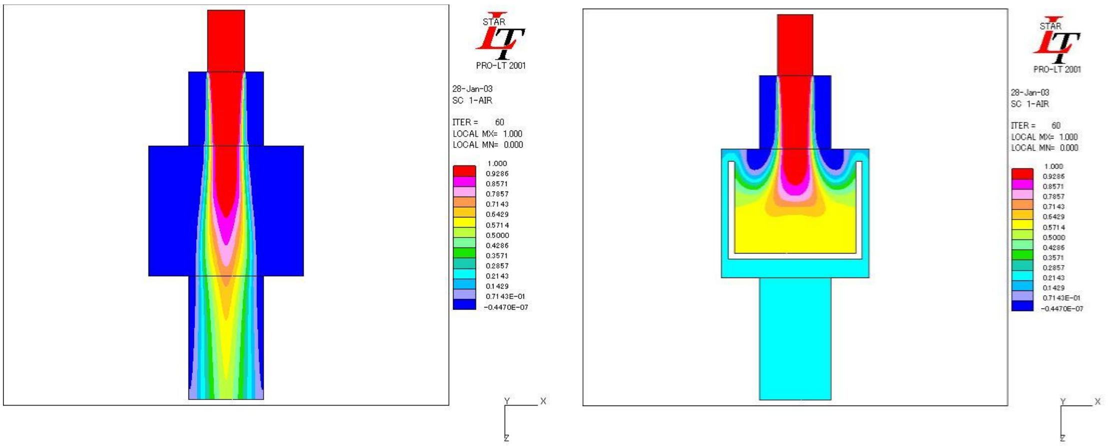
Fig.4- Mesh definition and dimensional drawing of bubble eliminator

Table 1- Dimensions of inlet ports

	α [mm]	β [mm]	α/β	s [mm^2]
No.1	1.4	15.0	0.09	21
No.2	3.0	7.0	0.43	21
No.3	4.2	5.0	0.84	21
No.4	7.0	3.0	2.33	21
No.5	10.0	2.1	4.76	21

Figure 3 shows a mesh definition and dimensional drawing of the static mixer for numerical simulation, respectively. The static mixer is divided into four definition blocks for the numerical analysis; (1) air inlet, (2) oil inlet, (3) mixer and (4) downstream pipe. The number of total cell for the configuration has 160000 cells. The geometric dimensions of the static mixer are determined as follows; D_1 = 12 mm, D_2 = 20 mm, D_3 = 22 mm, D_4 = 25 mm, D_5 = 6 mm, L_1 = 12 mm, L_2 = 21 mm, L_3 = 20 mm, L_4 = 16 mm, respectively.

Figure 4 shows a mesh definition and dimensional drawing of the bubble eliminator for numerical simulation. The overall apparatus for the bubble eliminator is divided by three blocks; (1) tangential inlet ports, (2) tapered-tube, and (3) downstream tube. The number of total cells for the configuration has 180000 cells. Dimensional conditions of the tangential inlet ports are tabulated in Table 1. The aspect ratio (α/β) is determined to keep a inlet port area s at a constant value. The other geometric dimensions of the bubble eliminator are determined as follows; d_1 = 25 mm, d_2 = 20 mm, l_1 = 15 mm, l_2 = 15 mm, l_3 = 130 mm, respectively.



(a) without cuplike part (b) with cuplike part

Fig.5- Simulation results of air content in static mixer

Simulation results for two conditions with and without the cylindrical cuplike part comparing to the distribution of volume percentage of air content in the oil mixture inside flow are shown in Fig.5. Air is infused into the centred portion of the device from the pipe located at the upper stream. In case of the device without cuplike part, air of low density and viscosity against oil is collected in the centred portion of the device and percentage of air content becomes higher compared with other wall side portion. On the contrary, when the cuplike part is equipped in the static mixer, uniform distribution of the air and oil mixture can be obtained at the downstream side of the static mixer. This is because by means of the cylindrical cuplike part agitation and mixing of air into oil occur. The simulation results discussed so far show clearly that function of the device to disperse air-bubble into oil.

We investigate influence of the geometric dimension of the inlet port for function of the bubble eliminator. Typical results of the calculated pressure distribution across a longitudinal section along z-axis for the five kinds of geometric dimensions for the inlet port as previously shown in Table 1 are plotted in Fig. 6. All the pressure data is plotted from a reference point at the one side end of the inlet port. The pressure at the center of the swirling flow continuously decreases as the working fluid flows to the downstream side. There is a minimum pressure point at a close end of the tapered tube. Subsequently, the pressure makes a recovery at the center of the adjoining straight downstream tube, because the velocity of the swirl flow is decelerated by the viscosity of the working fluid. By comparing the data for the difference of the aspect ratio, it is obvious when the pressure gradient becomes steeper from the one side of the inlet port (z=0) to the tapered starting plane (z=15), lower pressure is produced inside the device. As a result of the simulation, we should mention that the lower the minimum pressure value becomes, the effectiveness of the device for bubble elimination increases. The centrifugal effect and the radial pressure gradient become larger as oil flows through the tapered portion of the device, so that the light particles such as bubbles in the oil are collected and trapped near the center of the device. It is clearly that as the aspect ratio increases the pressure gradient becomes steeper in Fig.6. It is also to be note, however, much larger aspect ratio of the inlet dimension results in reduction of pressure gradient along the axis. Collision interference may occur between the tangential inflow and inner swirling flow in the inlet portion.

Figure 7 shows difference of the revolution speed at one side of the inlet port (z=0) and the starting plane of the tapered portion (z=15) in comparison with different geometric dimensions of the tangential inlet ports. The oil circulates through the peripheral inlet tube and tangentially flows into the tapered-tube chamber. There are certain position-dependent centrifugal forces created in all parts of the swirl flow. The swirl flow velocity has large influence on the effectiveness of bubble removal by the bubble eliminator. In case of the inlet port No.1, there is no difference of revolution speed between two cross sectional plane. On the other hand, in case of the inlet port No.2 revolution speed at the starting plane of the tapered portion is higher than the

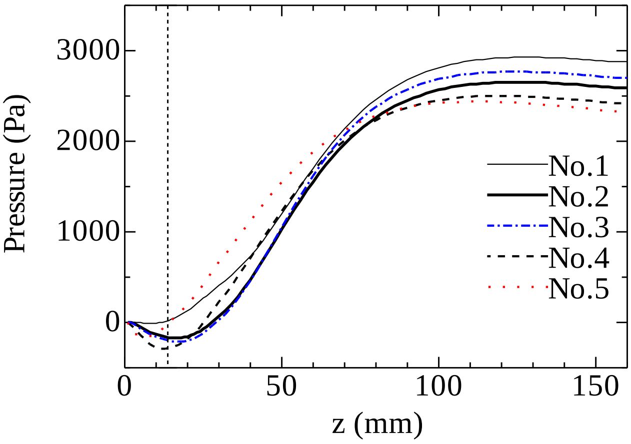
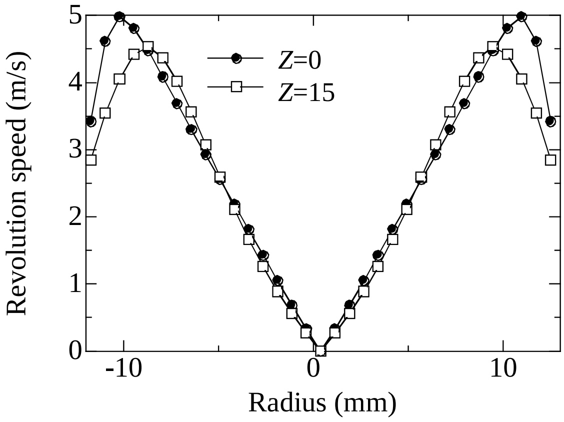
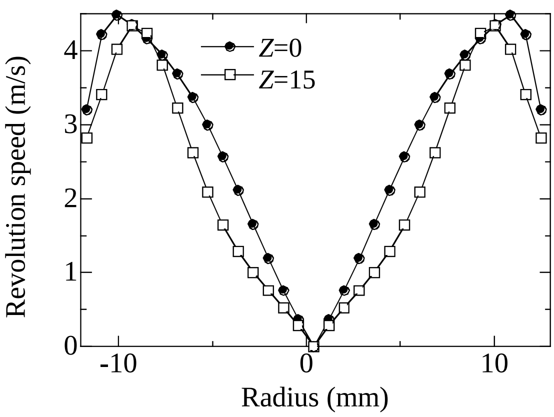


Fig.6 - Pressure distribution for the five geometric dimensions of the inlet port



(a) Inlet port No.1



(b) Inlet port No.2

Fig.7- Revolution speed of swirling flow for aspect ratio of inlet ports

revolution speed at the inlet portion. Above result has shown that difference of the revolution speed at the inlet portion and the starting plane of the tapered portion creates reverse pressure gradient. As a result, in order to collect and remove bubbles effectively, the geometric shape of the inlet port should be designed to make revolution speed high and accelerate swirl flow.

4 EXPERIMENTAL INVESTIGATION FOR OIL TEMPERATURE RISE

In order to evaluate the performance of the heat exchanging system fluid temperature rise in a reservoir of a test hydraulic circuit is experimentally measured during continuous running. Figure 8 shows an experimental hydraulic circuit, in which there are the active heat exchanger in a main line and normal pipes in a bypath line in parallel. The observation windows are installed at the downstream side of the mixer and the bubble eliminator, respectively, to observe the air bubbles in the oil. In our experiments a system pressure is set at 4.0 MPa and average flow rates delivered by the pump are kept at 22.5 L/min. The oil temperature in the reservoir is measured every 5

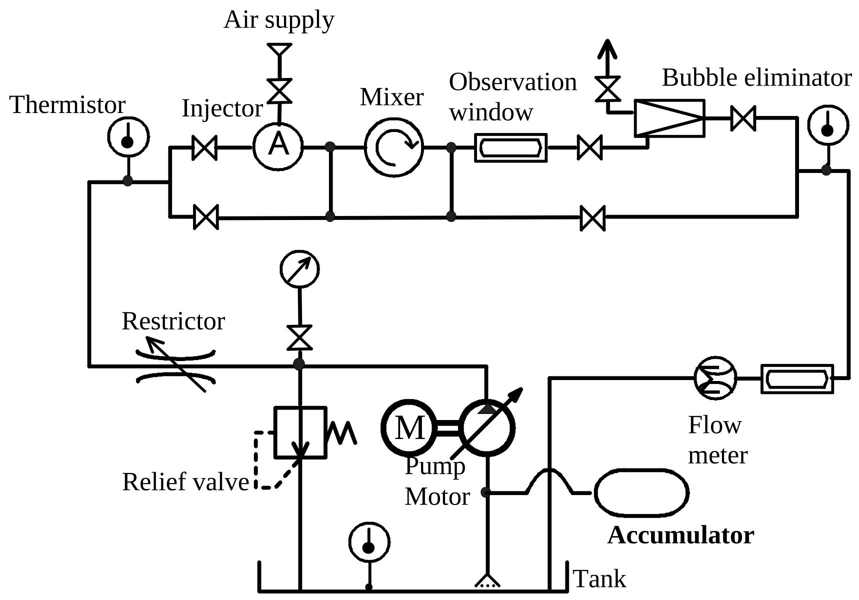


Fig.8- Test circuit for heat exchanger

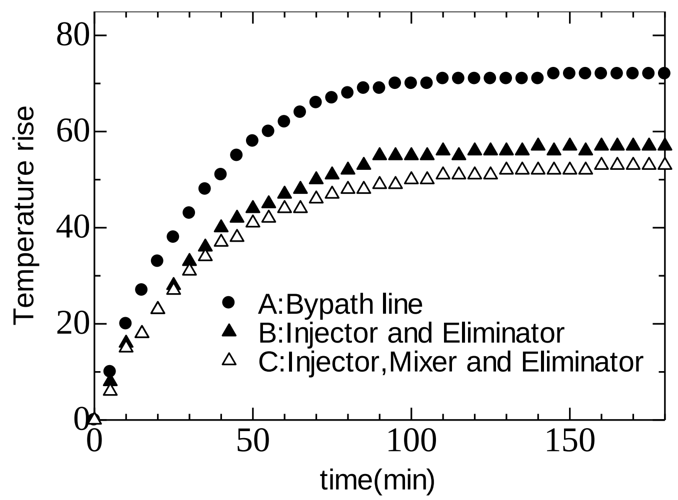


Fig.9- Temperature rise in reservoir for three test circuit conditions

minutes during 180 minutes. The reservoir has a capacity of 20 litter. Initial condition of the oil temperature is kept at 293 K for measurements.

Experiment results of the temperature rise in the reservoir are plotted in Fig. 9. In case A, the oil flows through the bypath line and oil temperature is measured. In case B, the oil flows through the injector and the bubble eliminator in the main line and make a detour around the static mixer. In case C, the oil flows to the injector, the static mixer and the bubble eliminator in the main line. In cases B and C, forced mixing air is infused at the injector every 5 minutes, and the oil temperature is also measured in the reservoir. In conditions B and C, the equilibrium temperature decreases more dramatically than condition A in order to operate the heat exchanging system. Further the equilibrium temperature in condition C slightly decreases from condition B. From these experimental results for measurement of oil temperature rise in the reservoir of the test hydraulic circuit, performance of the active heat exchanger is evaluated.

5 CONCLUSIONS

In fluid power systems loss of power turns to thermal energy to heat working fluid and system components. For most fluid power systems the heat exchanger is essential to reduce fluid temperature. In this paper a new heat-exchanging device for fluid power systems are proposed and developed by using the principle of the static mixer and the bubble eliminator. We call this system the active heat exchanger. Flow pattern in the static mixer and the bubble eliminator has numerically analyzed to investigate the influence of mixing and separating the air to the oil. In order to evaluate the performance of the heat exchanging system the oil temperature rise in the reservoir of the test hydraulic circuit is also experimentally measured during continuous running. It is numerically and experimentally verified that the proposed active heat exchanger can effectively reduce the oil temperature rise. Use of the active heat exchanger for the fluid power system may allow the hydraulic designer to reduce the system's reservoir size.

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Bubble Elimination Device in Hydraulic Systems

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Abstract

Bubbles in working fluids have much influence on the performance of hydraulic systems and cause some troubles. In this paper, the performance of the developed bubble elimination device concerning the effect of reducing the oil temperature rise is experimentally investigated. From a view point of energy balance in hydraulic systems, a simple mathematical model is proposed and validity of the model is studied.

1. Introduction

Bubbles in working fluids have much influence on the performance of hydraulic systems and cause major troubles such as bulk modulus change, cavitation and aeration inception, degradation of lubrication (Yano and Yabumoto, 1991), noise generation, oil temperature rise and deterioration of oil quality (Matsuyama and Takesue, 1996). When the bubbles in oils are compressed adiabatically at high pressure conditions, the temperature of the bubbles may rise sharply and the surrounding fluid temperature also rise (Backe and Lipphardt, 1976).

It is an important technical issue to eliminate the bubbles from the oil because of preventing a degradation of the oil and a damage of hydraulic components (Suzuki, 1994). However, it is a quite difficult problem how to separate the bubbles from the oils during the operating state of hydraulic systems. Recently, a device to eliminate bubbles by swirl flow has been developed (Suzuki and Yokota, 1994). The authors call this device "Bubble Eliminator". By using this device, the hydraulic system obtains greater performances; especially useful for reducing oil temperature rises caused by the bubbles.

In this paper, the influence of bubbles increasing the oil temperature and the effects of the bubble eliminator are studied through experiments. Furthermore, from a view point of energy balance in hydraulic systems, a mathematical model concerning the effect of reducing the oil temperature rise by use of the bubble eliminator is proposed. A thermal time constant in the mathematical model is investigated through experiments and the validity of the proposed model is studied.

2. Nomenclatures

A:	heat convection surface area [m^2]
C:	specific heat of oil [J kg^{-1} K^{-1}]
h:	average heat transfer coefficient [W m^{-2} K^{-1}]
k:	constant ($=dQ_{in}/dt$) [J s^{-1}]
m:	mass of oil [kg]
Q_{in} :	generated hydraulic energy [J]
Q_{out}:	radiative heat energy [J]
t:	time [s]
T:	oil temperature [K]
T_e:	ambient temperature [K]
T_k:	rising temperature of oil ($= k / hA$) [K]
T_∞:	final temperature of oil [K]
U:	internal energy of oil [J]
τ :	thermal time constant [s]

3. Bubble eliminator

Figure 1 illustrates the principle of the bubble eliminator (Suzuki and Yokota, 1994). The tapered-tube type device is designed such that a chamber of cross-sectional round shape becomes gradually smaller and connected to a cylindrical shaped chamber.

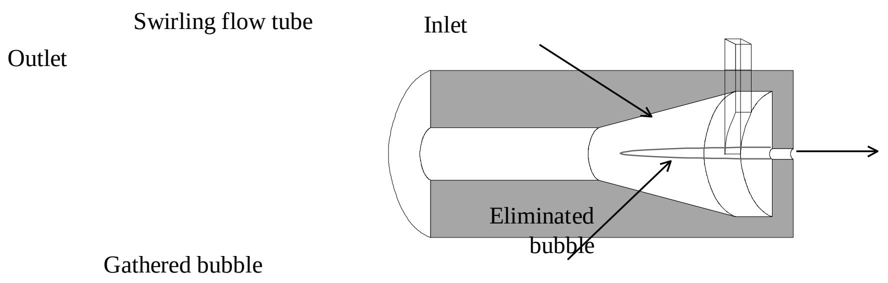

Fig.1 Bubble eliminator

Working fluids with bubbles flow tangentially into the tapered-tube from a inlet port and forms a swirl flow that circulates fluid through the flow passage. The swirl flow accelerates towards the downstream. Bubbles are trapped in the vicinity of the central axis because of a difference in the specific gravity of the oil and the bubble, and collected near the range of a vent port where the pressure is lowest. When some back pressure is applied by a check valve or an orifice located at the downstream side of the bubble eliminator, the bubbles are ejected oneself through the vent port. The dissolved gas in the fluid is also eliminated through the bubbles extracted at the pump suction side under the negative pressure. In the previous study (Suzuki and Yokota, 1994), it is experimentally confirmed that the bubble eliminator has been able to eliminate the entrained bubbles and dissolved gases from the working fluid efficiently.

4. Experimental investigation
4.1 Experimental setup

An experimental circuit of the hydraulic system is illustrated in Fig.2. The oil in a reservoir (the capacity of 20 *l*) pressurized by a variable displacement-type piston pump flows through a restrictor and returns to the reservoir. A relief valve is set for a safety pressure in the pump delivery line. The downstream line of the restrictor is divided into two lines. One goes through the stop valve [3], the bubble eliminator, the stop valve [5] and the flow meter to the reservoir. Another goes through the bypass line in which the stop valve [2] is incorporated, and the flow meter to the reservoir. The needle valve [1] at the pump suction side is used to introduce external air into the hydraulic system. A thermistor type thermometer is installed in the reservoir.

4.2 Test conditions

In order to investigate the effectiveness of the developed bubble eliminator experimentally, the oil temperature changes during several hours are measured under some conditions tabulated in Table 1. Different parameters such as the bubble eliminator "With" or "Without" and the air supply "On" or "Off" are set for the given pump delivery conditions.

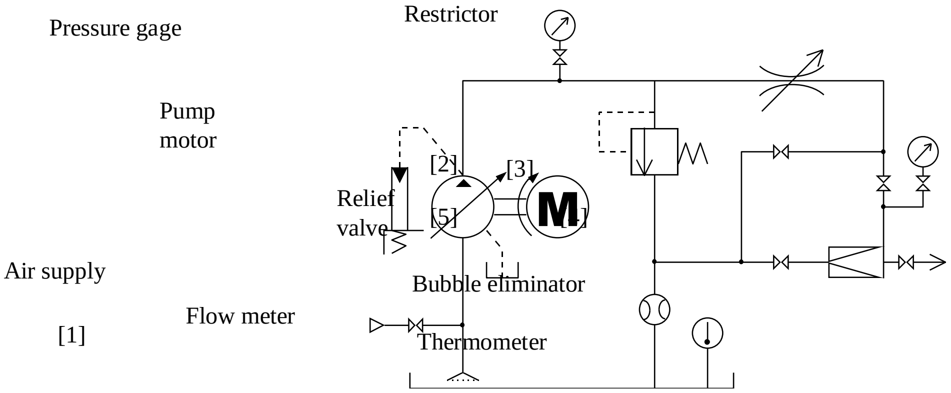

Fig.2 Experimental hydraulic circuit

Table 1 Test conditions for experiments

Condition	Bubble eliminator	Air supply
A	Without	Off
B	With	Off
C	Without	On
D	With	On

In the case of the bubble eliminator "With", the operation of the bubble eliminator is carried out by the following manner; open the stop valves [3][4][5], close the stop valve [2] on the bypass line, squeeze the valve [5] to provide the bubble release pressure at 20 kPa and then discharge the bubbles through the vent valve [4]. In the case of the bubble eliminator "Without", the stop valve [2] is opened and the valves [3][4][5] are closed.

In the case of the air supply "On", the pump drives for few minutes under no load condition and the needle valve [1] is slightly opened for admission of an external air. Few minutes later, air bubbling is stopped and system pressure is adjusted at the experimental condition.

4.3 Experimental results for reduction of oil temperature rise

Figure 3 shows the oil temperature rise for the test conditions A-D as shown in Table 1. The pump delivery pressure and flow rate are adjusted at constant values of 6.0 MPa and 20 *l*/min, respectively. In the conditions C and D, the external air is supplied in the hydraulic system for first 10 minutes after the pump operates. The time interval of temperature measurement is every 30 seconds and the experiments are carried out during 1 hour under continuous running. The initial temperature of the oil is kept within a range of 10±4℃ which is almost the same as atmospheric temperature. The temperature data are plotted as the values relative to the initial temperature.

After 1 hour of continuous running, oil temperature increases significantly high. The highest temperature rise is measured in the condition C in which the bubble eliminator is not being used and air is infused for 10 minutes. It can be explained that the bubbles in the oil causes the oil temperature rise. The air bubbles are considered as some kind of a heat insulator, so the existence of the air bubbles causes reduction of thermal conductivity of the oil. The oil temperature rise in the condition A becomes lower than in the condition C, however, it remains higher than the conditions B and D. In the conditions A, only a cavitation air has much influence on the temperature rise of the oil.

The oil temperature rise with the bubble eliminator is reduced as shown in the conditions B and D. No significant difference is clarified between the conditions B and D. These results can be explained that the bubble eliminator removes both the infused

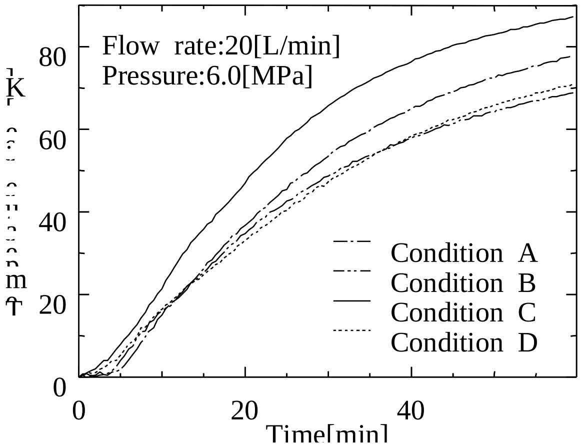

Fig.3 Experimental results of oil temperature rise

air and the cavitated air which is forced out from the oil. It is experimentally confirmed that the bubble eliminator is effective in reducing the oil temperature rise.

5. Energy balance of hydraulic system

5.1 Mathematical model

Considering the first principle of thermodynamics to the oil in hydraulic systems as a lumped-heat-capacity system shown in Fig.4, an internal energy balance of the oil is given as follows;

$$\frac{dU}{dt} = \frac{dQ_{in}}{dt} - \frac{dQ_{out}}{dt} \qquad (1)$$

where U is the internal energy of the oil, Q_{in} is the hydraulic energy generated by the pressurized pump and Q_{out} is the radiated energy from the oil to an ambiance as the heat energy.

It is assumed that the oil is non-compressible and no work is carried out by the hydraulic pressure change instead of heat transfer. As a result of the constant pressure and flow rate conditions at the pump, the hydraulic energy differentiated with respect to time is kept at a constant value and defined as follows;

$$\frac{dQ_{in}}{dt} = k \qquad (2)$$

Based on the Newton's law of cooling between the oil and the ambiance, the differential radiative heat energy Q_{out} with respect to time is derived as follows;

$$\frac{dQ_{out}}{dt} = hA(T - T_e) \qquad (3)$$

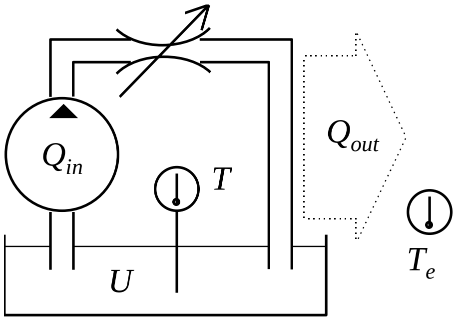

Fig.4 Energy balance of hydraulic system

where h is the averaged heat transfer coefficient and A is the surface of the hydraulic system for heat convection. Using the equations (1) to (3), the differential oil temperature change with respect to time can be derived as follow:

$$\frac{dT}{dt} = \frac{hA}{mC}\left(\frac{k}{hA} + T_e - T\right) \qquad (4)$$

By solving this differential equation (4), the following expression for the oil temperature change can be obtained.

$$T = \frac{k}{hA}\{1 - \exp(-\frac{t}{\tau})\} + T_e \qquad (5)$$

where τ is the thermal time constant and defined as follows.

$$\tau = \frac{mC}{hA} \qquad (6)$$

Consequently, equation (5) can be regarded as the first order time-lag which has the initial value of T_e and the final value of (T_e + k/hA), respectively. Let a rising temperature T_k denote as:

$$T_k = \frac{k}{hA} \qquad (7)$$

The thermal time constant defined by the equation (6) depends on the convection heat transfer coefficient which changes with time. So the thermal time constant is determined by the experimental results of the temperature changes for a heat equilibrium test. Figure 5 shows a typical example of the oil temperature rise as the result of the condition B. The oil temperature is measured until the energy balance attains thermodynamic equilibrium during 4 hours. The thermal time constant is estimated from the time which takes value 63.2 % of the rising temperature T_k.

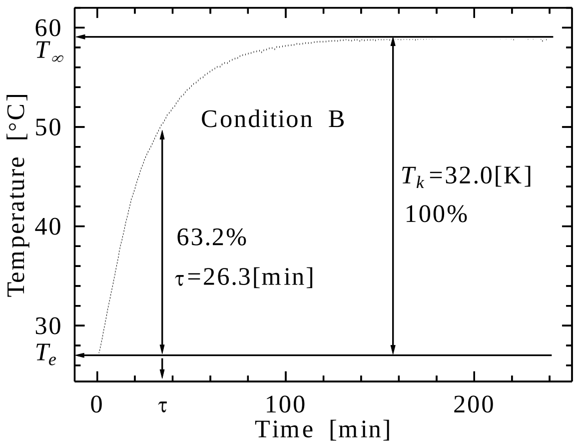

Fig.5 First order time lag

5.2 Experimental result of heat equilibrium

In the heat equilibrium tests, the pump delivery pressure and flow rate are adjusted at moderate values of 3.0 MPa and 13 *l*/min, respectively. The time interval of the temperature measurement is every 6 seconds and the tests are carried out for 4 hours under continuous running. The initial oil temperature is in the range of 26±2℃. In the conditions C and D, the external air is infused in the oil for 2 minutes before testing. The volume % of the infused bubble in the oil of reservoir is measured by Picno meter and the 2 % air in the oil is contained.

Figure 6 and 7 show the experimental results of the oil temperature rise for the heat equilibrium test. These figures also show comparative results for the mathematical model calculated by the equation (5) under the test conditions A, B and C, D respectively. The experimental results and calculations of the mathematical model indicate good similarity as a function of the operating time. Table 2 shows the thermal time constant τ and the rising oil temperature T_k estimated from the experimental results in Fig.6 and Fig.7 under the test conditions A to D. It is noticeable in the conditions B and D, the rising temperature is little reduced comparing to the conditions A and C.

We assume in the mathematical model that the thermal time constant expressed in the equation (6) is a time invariant value. However, it is our opinion that the heat transfer coefficient is varied by the air bubble content in the oil. As a result, the thermal time constant is a time variant value under different test conditions. In order to investigate the oil temperature change in more detail, the variation of the logarithm of the ratio of the oil temperature with the time is introduced. The following equation is obtained from equation (5).

$$\ln\frac{T_e + T_k - T}{T_k} = -\frac{t}{\tau} \qquad (8)$$

Figure 8 shows the variation of the logarithm of the ratio of the measured oil temperature in the left side of the equation (8) with the times. According to the equation (8), the mathematical model which has the time invariant constant is shown as a straight line with a constant gradient. In the conditions A and C without the bubble eliminator, the experimental results differ from the mathematical model significantly. On the contrary, in the conditions B and D with the bubble eliminator, the experimental results are similar to the calculation of the mathematical model. In the conditions B and D, the dissolved and infused air is eliminated from the oil by the bubble eliminator and not affected on the change of the thermal time constant. The gradient change in the condition C as shown in Fig.8 results from the averaged heat transfer coefficient changed by the infused and cavitated air. The above results lead to the conclusion that the existence of the air bubbles in the oil has influence on the heat transfer coefficient and the thermal time constant.

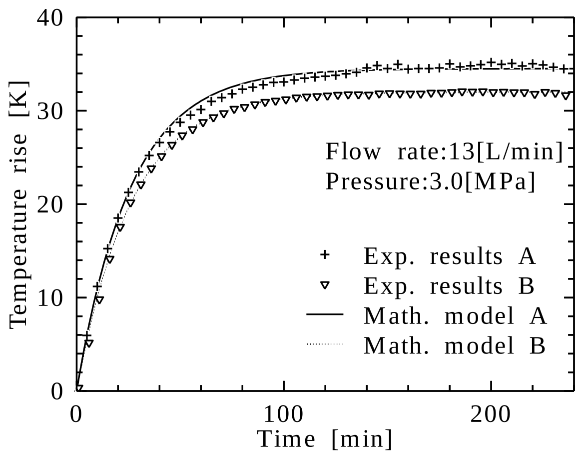

Fig.6 Comparison of experiment and mathematical model in conditions A and B

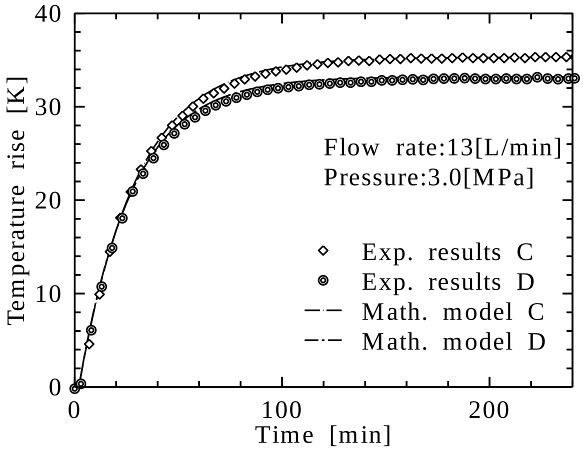

Fig.7 Comparison of experiment and mathematical model in conditions C and D

Table 2 Thermal time constant and rising temperature

Condition	τ [min]	T_k [K]
A	26.1	34.5
B	26.3	32.0
C	27.9	35.3
D	25.6	33.2

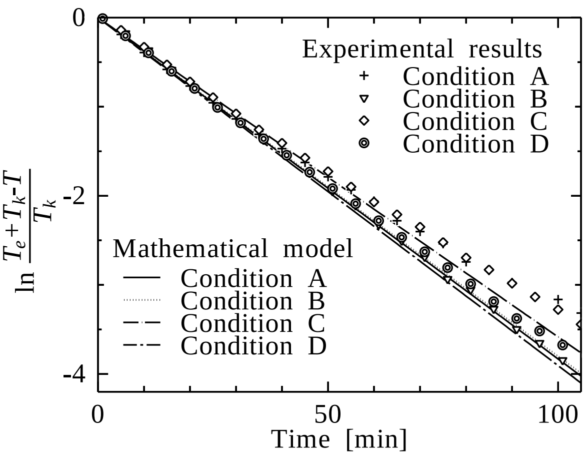

Fig.8 Variation of logarithm of ratio of oil temperature with time

6. Conclusions

In this paper, the authors have examined the influence of the air bubble and the effect of the bubble eliminator to prevent temperature rise through the experimental investigations by using the oil hydraulic system. Furthermore, from a view point of energy balance in hydraulic systems, the mathematical model concerning the effect of reducing oil temperature rise by use of the bubble eliminator is experimentally investigated. The validity of the proposed mathematical model is verified. It can be considered that many problems caused by the entrained bubbles in hydraulics and lubrication oils are solved by the bubble eliminator.

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Bubble Elimination in Hydraulic Fluids: Part II
- A New Technology for Downsizing of Reservoirs -

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ABSTRACT

Air entrainment in working fluids has greatly detrimental effects on function and lifetime of the fluid power components and systems. This may cause major problems such as cavitation and aeration, noise generation and vibration, oil temperature rise, and deterioration of oil quality. Especially, when bubbles in oil are compressed adiabatically at high pressure in a pump, the temperature of the bubble rises sharply and the surrounding fluid temperature also rises. Thus, it is important to eliminate the air bubbles from the oil to preserve oil quality, system performance, and to avoid possible damage of the components.

In mobile hydraulic systems such as commercial vehicles, hydraulic fluids are splashed and agitated in the reservoirs. To overcome air entrainment in oils, the overall dimensions should enclose a sufficient volume of oil to permit air bubbles to escape passively during the resident time of the fluid in the reservoir. However, in view point of environmental compatibility, energy saving, cost saving and safety, one trend in fluid power systems is to be designed in a more compact fashion and requiring less fluid in the reservoir.

One of the authors has developed a newly device using swirl flow for bubble elimination capable of eliminating bubbles and of decreasing dissolved gases. This device is called the Bubble Eliminator. In our previous study it has been experimentally and numerically demonstrated that using the bubble eliminator has enabled the fluid power system to perform better.

In this paper we will firstly focus on the technical issue for the air bubbles and oil temperature rise in the reservoir of a test hydraulic circuit. In our experiments,

surrounding air is intentionally introduced from the oil surface by vibration of the reservoir in the hydraulic circuit. Experimental setup for vibration of the reservoir is

shown in Fig.1. A hydraulic servo cylinder forcedly vibrates the overall equipment in order to infuse air to the oil of the reservoir from surrounding. In the hydraulic circuit shown in Fig.2, the bubble eliminator in a main line is located in parallel with a normal pipe of a bypath line. In our experimental hydraulic circuit the average flow rates delivered by the pump are kept at 12 L/min and the reservoir has a capacity of 3 Litter. It is assumed to be 1/20 model of the construction machinery with the pump delivery flow rate of 240 L/min and reservoir capacity of 60 Litter. The bubbles in the oil can be visually observed at the reservoir and observation windows. The oil temperature in the reservoir is measured every 5 minutes during 150 minutes. Experimental results of the temperature rise in the reservoir are plotted in Fig.3. In case A and B, the oil flows through the bypath line. In case C, the oil flows through the bubble eliminator in the main line. In case B and C, the experimental equipment is forcedly vibrated with the frequency of 1.2Hz and the amplitude of ±20mm. In case B and C, the equilibrium temperature decreases more dramatically than the case of A. This can be considered that the lower temperature rise is based on the cooling effect by the infused surrounding air. Further the equilibrium temperature in case C slightly decreases from the case of B. From these experimental results for measurement of oil temperature rise in the reservoir of the test hydraulic circuit, the performance of bubble eliminator is evaluated.
The swirl flow pattern and the pressure distribution in the tapered-tube chamber of the bubble eliminator greatly influence the effectiveness of bubble removal. Geometry of the tapered-tube chamber is important factor in design of the bubble eliminator. In this paper numerical analysis and flow visualization are also performed for clarifying swirling flow characteristics and pressure distributions for optimal design. The swirl flow pattern in the bubble eliminator is calculated by a three-dimensional numerical analysis for a two-phase flow. The typical example of the pressure distributions along the central axis of the bubble eliminator under the tapered-tube of 30 mm, 50 mm and 70 mm in length are plotted in Fig.4. There are minimum pressure points at a close end of the tapered-tube chamber. Air particles are trapped in the vicinity of the central axis and collected at the minimum pressure point. As a result of numerical simulation, we can design a more compact type of the bubble eliminator, which has an optimal geometry of the tapered-tube chamber.
Use of the bubble eliminator may allow the hydraulic designer to reduce the system’s reservoir size, as well as gain the following benefits:

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• a reservoir with lighter weight, smaller space, lower cost
• slow fluid degradation, which extends fluid’s useable life
• prevent pump cavitation and noise
• less fluid in reservoir, which reduces cost and increases safety
• shorter heating time in cold weather
• decrease in fluid compressibility
• easier contamination control, and
• simpler configuration of reservoir, with no baffle plate needed.

Fig.1 Overall experimental setup
Pump motor
Pressurgage
Thermocouple Observation
window
PC
Reservoir

Bubble eliminator
vent
bypass
Fig.2 Experimental hydraulic circuit
IFPE2005 in Las Vegas 4/4
0 50 100 150 0
10
20
30
B:Bypath line with vibration
C:Eliminator with vibration
Time 【min】

Temperature rise 【
K
】
A:Bypath line
Fig.3 Temperature rise in reservoir for three test conditions
0 100 200
0
0.002
0.004
0.006
Z axisis [mm]

Pressure
[MPa]
30mm
50mm
70mm
Fig.4 Pressure distribution along central axis of Bubble Eliminator