[2B01] INFLUENCE OF CHEMICAL STRUCTURE OF SIDE CHAIN CRYSTALLINE MONOMER ON TR FLUID BEHAVIOR
*Shigeru Yao¹, Yusuke Hasebe¹, Yuri Kanazawa¹, Makoto Takeda¹, Ryoko Nakano¹, Hiroshi Sekiguchi¹ (1. Fukuoka University)
9:00 AM - 9:16 AM

[2B02] EXPERIMENTAL CHARACTERIZATION OF A MAGNETORHEOLOGICAL DAMPER WITH MULTIPLE CYLINDRICAL PASSAGES AND TOROIDAL MAGNETIC FIELD GENERATOR
Mitsuhiro Kamezaki¹, *Peizhi Zhang¹, Kenshiro Otsuki¹, Shan He¹, Gonzalo Aguirre Dominguez¹, Shigeki Sugano¹ (1. Waseda University)
9:16 AM - 9:32 AM

[2B03] SENSING FLUID PRESSURE WITH Co RICH Fe-Co SYSTEM MAGNETOSTRICTIVE ALLOY TUBE
*Takashi Mizoguchi¹, Tsutomu Takahashi¹, Toshiyuki Hashida², Yasubumi Furuya³ (1. Electronics Engineering Department, Technology and R&D Division, Nabtesco Corporation, 2. Fracture and Reliability Research Institute, Tohoku University, 3. Micro system Integration Center, Tohoku University)
9:32 AM - 9:48 AM

[2B04] DEVELOPMENT OF MANIPULATOR USING A GAS-LIQUID PHASE-CHANGE ACTUATOR
*KENYA HIGASHIJIMA¹, Tomonori KATO¹, Kazuki SAKURAGI¹, Takahiro SATO¹, Manabu ONO² (1. Fukuoka Institute of Technology, 2. Tokyo Metropolitan College of Industrial Technology)
9:48 AM - 10:04 AM

[2B05] A PUMP USING EHD FLUID
*Takahiro shimizu¹, Tetsuhiro Tsukiji¹, Keitaro Hamada¹ (1. Sophia University)
10:04 AM - 10:20 AM

[2B06] A DEVELOPMENT OF THE NEW TYPE TOURNIQUET APPLYING EHD PHENOMENON
*Yusuke Takei¹, Shota Amemiya¹, Yuki Kakinuma¹,
School of Engineering, Yokohama National University, 2. Faculty of Engineering, Yokohama National University
10:45 AM - 12:05 PM
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*Shigeru Yao¹, Yusuke Hasebe¹, Yuri Kanazawa¹, Makoto Takeda¹, Ryoko Nakano¹, Hiroshi Sekiguchi¹ (1. Fukuoka University)
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*Yusuke Takei¹, Shota Amemiyà¹, Yuki Kakinuma¹, Hiroyuki Maeda², Hideaki Iwase², Mutsuhiro Maeda³, Kazuo Kaneko², Sumitaka Terasaka⁴, Takeharu Shimoohkawa⁴, Kazuyuki Mitsui¹ (1. Tokyo Denki University, 2. Juntendo University, 3. Yamamoto・Maeda Memorial Association Maeda Hospital, 4. Sanyo Metal Industry Co.,Ltd.)
10:20 AM - 10:36 AM
INFLUENCE OF CHEMICAL STRUCTURE OF SIDE CHAIN CRYSTALLINE MONOMER ON TR FLUID BEHAVIOR

Yusuke HASEBE, Yuri KANAZAWA, Ryoko NAKANO, Hiroshi SEKIGUCHI, and Shigeru YAO

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Abstract. Recently, we reported that block copolymers constructed with a monomer having long alkane side chain and another monomer with functional ability (hydrophilic, adhesiveness, etc.) have been used to produce side chain crystalline block co-polymers (SCCBC). We have also found that the SCCBC can adsorb to polyethylene (PE) crystalline surface and can modify the PE surface to render various properties. When we added a little amount of SCCBC to a concentrated PE particle dispersion, the viscosity decreases sharply, implying that the SCCBC can work as a good dispersant for PE particle dispersion. The dispersion changed to a high-viscosity fluid or gel upon increasing the temperature, meaning that the dispersion system becomes a thermal rheological fluid. Herein we investigated the SCCBC molecular structure dependence to the TR fluid function. We found that the micelle formation function of SCCBC and the micelle collapse temperature greatly affect the TR fluid function. Furthermore, we found that this technique can be used to design many kinds of TR fluids with various transition temperatures and transition behaviors.

Keywords: Side Chain Crystalline Polymer, Block Co-polymer, Polyethylene, Dispersion, Thermal Rheological Fluid

INTRODUCTION

Polyethylene (PE) is the cheapest and highly crystalline polymer, showing good processability, moldability and solvent resistance; it is therefore used in a wide range of applications from general ones to highly functional ones like lithium ion battery separators. PE particle is also added in paint and is used as sliding and tactile property modifiers. On the other hand, PE has bad wettability to solvents and inferior adhesiveness; thus, it is known as a representative plastic that is hard to modify. Therefore, conventional surface modification was typically conducted by physical methods, such as plasma irradiation, accompanied by severing of molecular chains. However, use of these methods impairs the mechanical properties of PE, and the only directly irradiated surface is modified. Consequently, it is hard to apply to thin films, porous films, and microparticles. Hence, no good modifier and/or dispersants have been proposed for PE microparticle dispersion systems.

In contrast, we have clarified that side chain crystalline block co-polymer (SCCBC), consisting of side chain crystalline monomers with long alkane side chains and monomers with solvent affinity, adsorb to PE surface at the crystalline side chains. Therefore, the PE surface is covered with the solvent affinity block of SCCBC. So, we found that the SCCBC has an ability to modify PE surface to have solvent affinity. We also showed that application of this functionality could control and significantly decrease the viscosity of a concentrated PE microparticle dispersant system. Moreover, we found that a concentrated PE microparticle dispersant system containing the SCCBC has a function as a thermal rheological (TR) fluid, i.e. a fluid that shows low viscosity at low temperatures but becomes highly viscous or gelates at high temperatures. The adsorption and dispersant functionalities as well as the TR fluid behavior should be significantly affected by the SCCBC molecular structure. Herein, we used behenyl acrylate (BHA) as the side chain crystalline monomer in addition to the previously employed sterayl acrylate (STA) to investigate the SCCBC structure dependence on the TR fluid functionalities.

EXPERIMENTAL PROCEDURE

Side Chain Crystalline Block Co-polymer

The side chain crystalline monomer considered in this study was BHA and STA. The monomer at the solvent affinity functional sites was n-butylacrylate (nBA). FIGURE 1 shows the chemical structures of the SCCBC. Polymerization was performed as reported in a previous paper: the initiator was Blocbuilder MA (Arkema) and living radical polymerization was carried out. The molecular weights of each unit in the polymerized SCCBC are shown in Table 1. Differential scanning calorimetry (DSC; PerkinElmer Japan DSC
8500) and wide-range X-ray scattering measurements (Shimadzu XRD-6100 diffractometer) were carried out to investigate the crystallinity of the SCCBC.

**Viscoelasticity measurement samples and viscoelasticity measurement procedure**

PE microparticle dispersion systems were prepared using these SCCBC. The PE microparticles used were Ceridust®3620, and the dispersion solvent was diethyl phthalate (DEP). The mixing ratio was PE microparticle 40 wt% and DEP 60 wt%. Measurements were carried out on this original system and on systems where 0.5, 1.0, and 3.0 wt% of SCCBC was added to the PE microparticle dispersion system and then made uniform by heating in a water bath at about 70°C.

Viscoelasticity measurements were conducted using a cone-plate rheometer (Rheosol-G2000, UBM Co., Ltd.). Steady-state shear viscosity measurements were performed between 0.01 and 100 s⁻¹. On the other hand, dynamic viscosity measurements were performed at frequencies between 0.01 and 30 rad·s⁻¹, and the strain was kept within 1–2% with confirming the linearity. The temperature dependence was studied by taking measurements at different temperatures in 10 °C steps between 35 and 65 °C in systems with STA-nBA and between 35 and 85 °C in systems with BHA-nBA. Small angle X-ray scattering (SAXS) measurements were taken to investigate the structure of the SCCBCs in DEP. DEP solutions containing 0.5, 1, and 3 wt% of SCCBC were placed in capillary tubes for measuring X-ray diffraction (manufacturer: W. Muller). Measurements were conducted at beam line 11 (BL11) of the Kyushu Synchrotron Light Research Center. The beam intensity was 2.1–2.3 keV.

**RESULTS AND DISCUSSION**

The DSC fusion behavior profile and heats of fusion (FIGURE 2) show that the block copolymer are side chain crystalline polymer where the side chains crystallize. The melting point of STA-nBA is ~45 °C, whereas BHA-nBA has a higher melting point of 60 °C and a higher degree of crystallinity. Therefore, the crystallinity and melting point of SCCBC increases with increasing length of the alkane side chains.

The X-ray wide-range scattering profiles of various SCCBC and PE (FIGURE 3) show that the peaks of BHA-nBA and STA-nBA appear at almost the same position as those of the (110) plane of PE, which implies that the SCCBCs are crystalline at the alkane side chains. There are very few broad peaks in BHA-nBA compared to STA-nBA, which probably originate from an amorphous structure; therefore, the former has higher crystallinity.

FIGURE 4(a) shows the temperature dependence of shear viscosity at the shear rate of 1 s⁻¹ for the original PE microparticle dispersed systems without SCCBC and containing 0.5 wt% of SCCBCs. The original system has very little temperature dependence, and consequently has very high viscosity. In contrast, systems with STA-nBA show typical TR fluid behavior, where upon heating, the viscosity rapidly increases at around 50 °C. On the other hand, the system with BHA-nBA shows a slight local minimum in viscosity at ~50 °C, but the dispersant
The effect is significantly lower than that in the case of STA-nBA. In addition, a gradual increase in viscosity is found from 60 to 70 °C. FIGURE 4(b) shows the temperature dependence of complex viscosity at the frequency of 1 rad·s\(^{-1}\) in the same systems. Same as in FIGURE 4(a), STA-nBA shows a distinct transition around 50 °C, while BHA-nBA has a shallow local minimum at about 50 °C.

FIGURE 5 shows the temperature dependence of shear viscosity and complex viscosity for PE microparticle dispersion systems containing 1.0 wt% of SCCBCs. Results for shear viscosity (FIGURE 5(a)) show that the temperature dependence of the system with 1.0 wt% STA-nBA is similar to that with 0.5 wt% STA-nBA, and a good dispersant effect is observed and a transition temperature is 50 °C. On the other hand, the system containing 1 wt% BHA-nBA shows a similar dispersant effect as that containing STA-nBA, and the transition temperature is 60–70 °C. DSC measurement results show that the melting point is higher for BHA-nBA. Therefore, the melting temperature of the pseudo crystals formed at the PE surface should be higher in the case of BHA-nBA than in STA-nBA. In contrast, the complex viscosity results (FIGURE 5(b)) show a deep local minimum at 45 °C for BHA-nBA. A similar phenomenon was found at high SCCBC concentrations, as reported previously, indicating that BHA-nBA prefers to form micelles in the solvent compared to adsorption to PE microparticles at low temperatures.

FIGURE 6 shows the temperature dependence of shear viscosity and complex viscosity for PE microparticle dispersion systems containing 3.0 wt% of an SCCBC. The temperature dependence of shear

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**FIGURE 4** Temperature dependence of the shear viscosity and complex viscosity of PE particle dispersion with 0.5 wt% of SCCBC. (a) shear viscosity (shear rate = 0.1 s\(^{-1}\)), (b) complex viscosity (frequency = 0.1 rad·s\(^{-1}\)). Blue: STA-nBA SCCBC system, Red: BHA-nBA SCCBC system, Gray: original dispersion system.

**FIGURE 5** Temperature dependence of the shear viscosity and complex viscosity of PE particle dispersion with 1 wt% of SCCBC. (a) shear viscosity (shear rate = 0.1 s\(^{-1}\)), (b) complex viscosity (frequency = 0.1 rad·s\(^{-1}\)). Blue: STA-nBA SCCBC system, Red: BHA-nBA SCCBC system, Gray: original dispersion system.

**FIGURE 6** Temperature dependence of the viscosity and complex viscosity of PE particle dispersion. (a) shear viscosity (shear rate = 0.1 s\(^{-1}\)), (b) complex viscosity (frequency = 0.1 rad·s\(^{-1}\)). Blue: STA-nBA SCCBC system, Red: BHA-nBA SCCBC system, Gray: original dispersion system. The SCCBC content is 3.0 wt%.
viscosity (FIGURE 6(a)) indicates that upon heating, the viscosity shows a local minimum before increasing; the transition temperature is ~50 °C for STA-nBA and 70 °C for BHA-nBA, respectively. The complex viscosity also has a temperature dependency with a local minimum, as shown in FIGURE 6(b).

FIGURE 7 is a schematic of the mechanisms guiding these phenomena. Here, Td-mic is the micelle decomposition temperature of SCCBC, Tm is the melting point of the pseudo crystals of SCCBC that form on PE, ϕ is the SCCBC concentration, ϕc is the critical micelle concentration, Ead is the energy of SCCBC when adsorbing on PE, Ead is the energy of SCCBC when existing by itself in solution, and Emic is the energy of SCCBC when forming micelle in solution. We first consider the cases where the SCCBC concentration is lower than ϕc (cases 1 and 2). In case 1, where the temperature is below Tm, SCCBC is energetically stable when adsorbed on PE. Therefore, the PE particle surface gains solvent affinity, the dispersant effect comes into play, and the viscosity decreases. On the other hand, when the temperature increases above Tm, SCCBC starts to desorb from the particle surface and TR phenomenon is observed because the modification effect at the PE particle surface disappears, leading to increased viscosity. Next, we consider the case where the SCCBC concentration is higher than the critical micelle concentration. For case 3, where the temperature is lower than Td-mic, micelle formation is energetically favorable for SCCBC than adsorption on PE or dispersion of individual molecules in solution. Therefore, the dispersant effect is limited because the amount of SCCBC adsorbed on the PE particle surface is significantly less than that in case 1. In contrast, when the temperature increases beyond Td-mic, SCCBCs in micelles become unstable and they begin to adsorb on PE particles. Consequently, the viscosity significantly decreases with increasing temperature for high SCCBC concentration. In case 5, where the temperature is higher than Tm, SCCBC desorbs from PE particles and the viscosity increases, similar to case 2. If Td-mic and Tm are close to each other, viscosity increases rapidly immediately after the decreasing in viscosity; therefore, the TR phenomenon happens in a narrow temperature range. BHA-nBA has longer side chain length than STA-nBA; therefore, it can form micelles more easily. FIGURES 3–6 indicate that micelles start forming at 3 wt% for STA-nBA and at ~0.5 wt% for BHA-nBA.

FIGURE 8 shows the concentration dependence of the lowest viscosity found for the PE microparticle dispersion systems containing STA-nBA and BHA-nBA. The temperatures for each viscosity measurement are also shown. The lowest viscosity of STA-nBA is found at 35 °C at low SCCBC concentrations; however, this temperature increases to 45 °C at 3 wt% of SCCBC, as is evident from the figure. This result indicates that at 3 wt% SCCBC concentration, micelles form at low temperatures and decompose upon heating to ~45 °C and then SCCBCs adsorb on PE particle surfaces around this temperature. However, the viscosity remains almost the same, implying that the solvent affinities of PE microparticle surfaces, and therefore the amount of adsorbed SCCBCs, are also about the same. In contrast, no significant dispersant effect is observed for BHA-nBA at 0.5 wt% SCCBC concentration. At this concentration, most of the SCCBC formed micelles in the solvent, and the
amount adsorbed on PE particle surfaces is less than that for STA-nBA. At SCCBC concentration of 1 wt%, the viscosity becomes similar to that for STA-nBA; hence, dispersant effects start to appear because, in addition to micelle formation, the amount of SCCBC adsorbed on PE microparticle surfaces starts to increase. Moreover, at 3 wt% SCCBC concentration, there is a significant dispersant effect at 65 °C that is almost at par with STA-nBA. These results indicate that BHA-nBA form micelles more easily, that Td-mic of BHA-nBA is ~50 °C, and the dispersant effect of BHA-nBA becomes almost the same as that of STA-nBA if micelle decomposition proceeds sufficiently and the PE microparticle surface begins to have enough solvent affinity.

FIGURE 9 shows the SAXS profiles of DEP solutions with various SCCBC concentrations that were obtained using a synchrotron. STA-nBA (FIGURE 9(a)) shows almost no scattering at a concentration of 0.05 wt%; hence, no definite structure exists in the solution. With increasing SCCBC concentration, the scattering intensity increases and the scattering profile shifts from a weak peak to a flat profile, suggesting that some stable structure starts to form at STA-nBA concentration of ~0.5 wt%. In contrast, there is a plateau in the scattering profile even at 0.05 wt% in BHA-nBA solution, indicating that a stable structure start forming at this point. Therefore, BHA-nBA has a strong tendency to self-associate and probably form SCCBC micelles at low concentrations.

CONCLUSIONS

BHA-nBA polymerization was carried out for the first time in this study, and the obtained SCCBC has a melting point of ~60 °C.

Various concentrations of STA-nBA and BHA-nBA employed in this work were added to PE microparticle dispersion systems, and the temperature dependence of their shear and complex viscosities were evaluated. At 0.5 wt% concentration, BHA-nBA did not demonstrate a strong dispersant effect, and showed a shallow local minimum at 45 °C. Dispersant effects appeared when the concentration increased to 1 wt%, and a peculiar behavior was found where the complex viscosity shows a deep local minimum at 45 °C. At a higher concentration of 3 wt%, the temperature dependencies of steady-state shear viscosity and complex viscosity showed local minima near 45 and 65 °C for STA-nBA and BHA-nBA, respectively.

We explained the above behavior by considering the critical micelle concentration and micelle decomposition temperature of the SCCBCs. SAXS evaluation showed that BHA-nBA has greater tendency to form micelles than STA-nBA.

Transition temperature and temperature response of TR fluids could be designed by taking advantage of these properties.

REFERENCES


EXPERIMENTAL CHARACTERIZATION OF A MAGNETORHEOLOGICAL DAMPER WITH MULTIPLE CYLINDRICAL PASSAGES AND TOROIDAL MAGNETIC FIELD GENERATOR


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Abstract. In order to precisely control the stiffness, back-drivability, and output inertia of the system, several devices that employ magnetorheological (MR) fluids adjust the apparent viscosity of the MR fluids by controlling the magnetic field applied to it. A variety of devices has been developed such as valves and clutches, and however, in most of these designs, the customization parameters are restricted by the geometry of the bobbin-shaped core. As an alternative to conventional annular dampers, a new MR piston with multiple cylindrical passages and toroidal magnetic field generator was proposed. In the previous study, a prototype with a circular MR valve array integrated inside the piston was built and preliminary tests were conducted to ensure the feasibility of the system. The purpose of this study is to provide an experimental characterization of the proposed toroidal MR damper by conducting a new set of experiments to benchmark its force vs. stroke, force vs. speed, and control performance, against a conventional annular damper.

Keywords: Magnetorheological fluids, Hydraulic damper, Piston, Prototypes

INTRODUCTION

Recently, the emergence of magnetorheological (MR) devices which used magnetorheological fluids (MRFs) provided novel ways to help to create control output force in a couple of applications. These devices are able to change the viscosity of MRF obviously by adjusting the magnetic field applied to it, in order to control the stiffness, damping factor, as well as output inertia of the system. There are three modes to operate the movement of MRF, which are a valve mode, direct-shear mode, and squeeze mode [1]–[3]. These applications use the characteristics of the magnetorheological fluid to control the pressure gradient along stationary surfaces, shear stress between sliding plate, or strain force objecting the deformation of the flexible object with MRF, respectively.

Some MR devices such as MR damper [4], assistive devices [5], as well as in robotics [6], haptics [7], and other types of devices [8] are already used in the application that we familiar with. However, these devices have not been enough research done with the possibility to combine with the hydraulic linear actuator. The MR piston is needed to be created, in order to build the high output hydraulic linear actuator with backdrivability, helping to make sure the safety of hydraulic linear actuator. Based on the traditional annular dampers, a new magnetorheological piston (MR Piston) was designed, as shown in Fig. 1, which is inspired by toroidal electromagnets from the design of conventional annular dampers. In the new design of MR piston, a circular valve array integrated inside piston was built as a prototype, and then test to measure the passive performance.

This paper shows the piston performance evaluation, which contains piston force, friction force performance, step response, actuator hysteresis performance, and force control experiments. Additionally, in these experiments, the comparison between the annular piston head and toroidal piston head also shows clearly with charts and figures. By analyzing the data that already collected, the parameters, such as number and size of holes, the material of electromagnets, and coils might be modified to improve the accuracy of the model, in order to perform better in the further work.
MR PISTON PERFORMANCE EVALUATION

In order to test the damping force which was generated by the prototype, the experiments were designed by using a linear actuator. In the experiment, the linear actuator was used to drive the piston, the MR actuator assembled the accumulator, control valve, position sensor, and pressure sensor, to make it completely for the experiment. The Arduino board is used for recording data, and the power supply is used for control current. In the experiment, a conventional annular piston was also assembled and tested for comparing with the toroidal MR piston. The annular piston contains an annular gap inside the piston head, and the shaft is bobbin shape, playing the role of electromagnet core, an outer ring as a magnetic flux return path. The outer ring as a magnetic flux return path. In the both cases, the rod head of piston is combined with the rod head of linear actuator with a 2 kN load cell attached to them, while the linear actuator drives the piston to move back and forward. In the process of experiment, a data acquisition board is used to record the data including position, force, current, temperature, and pressure values at each port while the linear actuator driving the piston at a constant speed. The temperature in the piston is kept at room temperature (20–30 °C). The current in the electromagnet set manually by using an 800 W digital power supply. The load cell shows positive when the piston is pushed, and negative when pulled. In order to make sure the safety, the stroke range is set to 80 mm (from the total 110 mm), setting the start position, when the piston is close to its compressed state.

Piston Forces

In this experiment, the piston is cyclically moving at four different speed, which are 10, 20, 30, and 50 mm/s. In each round, the current is applied to toroidal design from 0.0 A to 3.0 A in 0.5 A increments, and 0.0 A to 0.5 A in 0.1 increments for the annular design. The top speed of response is able to reach 120mm/s when the current applied to the annular type is 0.3 A.

Figure 2 and Table 1 show the results, which indicate that under nominal current ranges (i.e., safe for coils), the peak force of prototype is about one-third of the conventional annular design. The reason of this is the design of annular piston head has more space than the prototype, which helps to contain a higher number of windings in the coil so that the annular design can achieve such output forces, as well as higher energy efficiency. However, from the results of Table 1, the toroidal design can also give a high output, when we run risk to add current on the coil with high current in a very short period. Based on the results, an optimization of the magnetic circuit is indicated, and gives the feasibility that the geometry of the passages could achieve comparable results to the conventional annular design.

Comparing with the annular design, the results of force generated by toroidal appear to be more stable and consistent in the whole range of current. The reason for the difference might be caused by the torque and speed controller module inside the linear actuator, which hints an easier controller for the proposed prototype. In the future work, the development of a more robust test setup is necessary for applying to a hydraulic piston, so that improve the stability of the annular type.

In the Force-Speed curves, the results of toroidal design are more independent from the piston speed, and also has lower hysteresis when compared to the results of annular design. However, this result is not expected at the...
beginning of the experiment, because the gap cross-section of the annular design is bigger, as the Table 1 shows. And the big gap cross-section normally causes to lower damping factor related forces. The reason for this result might because of the difference in the design of two kinds of pistons, the toroidal head has the empty seal grooves, while the manufacturing of the annular head has no such design. Nevertheless, the result shows that the toroidal can move each iteration steadily throughout this speed range.

Finally, from the resultant Force-Current curve, the results show the quasilinear relation between force and current perfectly, which achieve our goal. Because of the linear relation, the output force can be control easier by changing the magnetic field. In addition, the problem also is revealed in the experiment, both designs are far from the saturation limitation of the permeable materials, where the permeability of material decrease rapidly. By covering the problem of it, the toroid design of MR piston head supposed to performance better, which can be shown in the further work.

![Graphs of Force vs Current and Force vs Speed](image)

**FIGURE 2.** Top: measured piston forces vs current at different speeds, (a) Toroidal design 0.0–3.0 A, 0.5 A steps (b) Annular design 0.0–0.5 A, 0.1 A steps. Bottom: piston force vs speed at a sine wave of 120 mm/s with 1.2 Hz for (c) toroidal and (d) annular.

**TABLE 1.** MR Piston Design Performance Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Toroidal design</th>
<th>Annular design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head dry weight</td>
<td>516 gr</td>
<td>565 gr</td>
</tr>
<tr>
<td>Number of windings</td>
<td>240</td>
<td>860</td>
</tr>
<tr>
<td>Total coil resistance</td>
<td>3.3 Ω</td>
<td>10.9 Ω</td>
</tr>
<tr>
<td>Minimum force</td>
<td>57 N</td>
<td>104 N</td>
</tr>
<tr>
<td>Peak force [current]</td>
<td>607 N [5.0 A]</td>
<td>1136 N [0.5 A]</td>
</tr>
<tr>
<td>Power @ 300 N</td>
<td>24.3 W</td>
<td>0.3 W</td>
</tr>
<tr>
<td>Average response time</td>
<td>10 ms</td>
<td>18 ms</td>
</tr>
<tr>
<td>Gap active area</td>
<td>404 mm$^2$</td>
<td>992 mm$^2$</td>
</tr>
<tr>
<td>Total gap cross section</td>
<td>113 mm$^2$</td>
<td>260 mm$^2$</td>
</tr>
<tr>
<td>Core cross area</td>
<td>259 mm$^2$</td>
<td>264 mm$^2$</td>
</tr>
</tbody>
</table>

Power and force values taken at 20 mm/s, 5 A were applied to toroidal for a short period of time

**Friction Force Performance**

In the previous study, the friction force of static and dynamic was estimated when the accumulator connected. And the static friction is around 60 N, while the dynamic friction is around 40 N. However, because of the
connection of accumulator, the results also contain the uncompensated disturbance. In order to eliminate the inaccuracy, two additional tests were performed.

In the first experiment, the chamber of the piston in the state of empty, only the friction which is generated by rod against the seal, wear ring and wiper, is measured at a speed of 20 mm/s. The result shows the state of dynamic friction, which is an absolute average of 34 N. The second experiment is tested with the MRP filled, the friction force was calculated to 50 N after subtracting the zero field damper force [9]. The friction which generated by prototype is reasonable for the size of the piston, because of the rubber seals of the piston removed. Nevertheless, the friction performance is not satisfied for achieving backdrivable application. The friction needs to be reduced in the future. The further work contains improvements of low friction seals, and ball bearings instead of wear rings for the piston rod. Furthermore, by using a tapered piston with machined surfaces and alignment tolerances precisely, a no seal piston head can be achieved, in order to reduce the friction force successfully.

\textit{Step Responses}

In the previous study, the performance of step response is not so satisfied. The response time was estimated at 200 ms, which was unusually slow when compared with normal MR devices. The probable reason might be caused by the inconsistency of data recording devices. In the past experiment, the force results were recorded by a universal testing machine, while the rest of the sensor data was collected by an Arduino board. For this reason, in the new experiment, a data acquisition board is used for collecting all the measurement results simultaneously. In the improved experiment, both pistons were set at 10 mm/s using 1.0 A and 0.07 A steps for test respectively. In this process, both designs correspond to a 70 N force step from their zero-field damping force. From the results, although the annular piston has a fast initial response, because of the higher inductance caused by this design, the overshoot and ripple appear. However, the design of toroidal design prevents these disadvantages, it reaches the steady state value 5 times faster with less ripple when comparing with the annular design. The results show that toroidal design is a suitable alternative for force control application, such as contact forces in physical human-robot interaction (pHRI). The fast response might due to the low coercivity of the material of electromagnetic core which used in the prototype. In the case of annular design, the overshoot can be compensated by using a current controller, so that the device obtains the optimal response seen in current damper designs.

Figure 3 shows the pressure difference, which is calculated by the difference of the measurements from two pressure sensors connected next to the piston ports. The position of pressure sensors is crucial to the measurement. Generally, the response of pressure is slower than the response of force slightly, and the behavior is obviously independent of the effect of the accumulator, so that the pressure sensors could be a suitable force feedback device. Nevertheless, the noise from the pressure sensor is more frequently than force sensor, which shows in the graphs.

\textbf{FIGURE 3.} Left: Hysteresis measurement performed at 500mm/min, Force control experiments using a PID controller to follow a sine wave with an amplitude of 50 N and a frequency of 0.5 Hz, 1.0 Hz, 2 Hz and 5 Hz on the top, and a 100 N step on the bottom. On top, showing a more stable response in the toroidal prototype.

\textit{Actuator Hysteresis Performance}

The purpose of these experiments is to determine the performance of the toroidal magnetic circuit with respect to
carrying current. In the experiment, the piston is moved by the linear actuator with 500 mm/s (8.33 mm/s) and the current was raised from 0 A to 3 A, then decrease back to 0 A with 0.5 A step.

The hysteresis curve shows in Fig. 3, the result indicates that the actuator has very low hysteresis comparing with other similar actuator found in articles, showing its potential to achieve a good ability to be controlled. This characteristic can be attributed to the small size of cores, as well as the low coercivity of the material used in the construction. As we can observe from the curve, from 0 A to 1.0 A the relation between shows strong non-linear behavior, which is caused by the material properties, the relation between the Newtonian and yield stress induced forces as well. After the point of 1.0 A, the permeability of the material should become more stable, and the yield stress of the MRF become the dominate force at the same time, helping to perform a better linear relation. These features need to be considered in the future design of the force controller, where a compensation strategy such as piecewise linear controller [10] could be used.

**Force Control Experiments**

The experiments for force control of toroidal piston is necessary, since the prototype is purposed to be applied in the future construction work. A PID controller was programmed in LabVIEW, so that the controller is able to drive the piston by outputting control signal. The inputs and outputs of the controller are taken charged by the DAQ board NI USB-6212. The control output signal is amplified using a chopper current controller, TI LMD18245, in order to create enough driving current. In the experiment, four sine waves and a square step of 100N are used as inputs. All the input sine waves have same amplitude of 50 N. However, the frequency is applied from 0.5 Hz to 5 Hz.

Figure 3 shows the result, which indicates the ability of prototype piston accurately follow the sine input, without apparent different comparing to the annular design. Nevertheless, when the frequency applied is 2 Hz, the small ripple occurs in the annular type. Similarly, in the case of 5 Hz, both designs have a tendency to produce higher ripples, which because of the higher pressure stored in the accumulator. Because it is not implemented in real time, the case of ripple fluctuates under the condition of variations of the frequency of the controller. Although these problems occur, the proposed head achieves smooth force control until 2 Hz.

In the experiment of step response, the annular design performs better in the rising edge, which is faster than toroidal design, where 20 ms for annular design against 45 ms for toroidal design. However, the falling edge is the same 60 ms for both. Nevertheless, a reduced 8% overshoot appears in the case. The reason of this might attribute to the difference in driving current. Moreover, at the steady state, the toroidal shows to be more stable, which is necessary for force interaction application. In one word, the PID controller improved the response for both design, also eliminate the effect from accumulator successfully. However, the performance in low-frequency state still not as good as we expected, which need to be improved by a more complete strategy such as [11].

**FUTURE IMPROVEMENTS**

At earlier stages of work, the active length of gap walls in annular is considered as an important factor that might limit the performance of the device. However, during the process of design several other key performance parameters were identified. In these parameters, the effect of magnetic characteristics of electromagnet materials plays an important role in output force of the actuator, which including the non-linear permeability and saturation, the geometry of the gap and its relation to the magnetic flux density, the size of the core, and the space for the electromagnet coils. Additionally, the improvement of design still needs to be done, since the results from FEM analysis shows a small magnetic field leakage. From the experiment results and FEM analysis, some ideas might be brought on how to bring the current devices to suit the current annular devices by improving the parameters that mentioned above.

In the study, by using FEM analysis software the physical mode was improved, and helping to give a better understanding of relations between parameters of the toroidal design. However, improvement, such as building reluctance model still needs to be done further, in order to account for the non-linearity of the parameters, as well as estimate the magnetic leakage path. In addition, a more straightforward design methodology can be achieved by the study of the trade-offs between key performance parameters.

Currently, the mathematical model shows a high sensitivity to the estimation of material and geometric parameters. Therefore, experiment measurements as well as accurate and complete data is needed to identify possible causes for the model discrepancies, and further improve its accuracy. Moreover, improvement of design and construction
of the electromagnetic circuit is proved useful, which not only beneficial for the model, but also the performance of actuator. Besides of the key parameters, for fixing the problems driven in the experiments, other important mechanical design modifications are needed. The double rod design needs to be tested and compared with the accumulator connected design, for finding which design has lower friction performance. The choice for sealing also important, since by changing the sealing the friction might be reduced.

Because of the development of the MR toroidal piston still in early stage, for sake of the feasibility in a wider range of the application, additional tests needs to be evaluated. Temperature tests can give the relation between the temperature and viscosity, as well as the effect of output force. After optimization of the force device, the force control experiment could be evaluated, to analyze the force repeatability, response of the device to different kinds of control inputs and bandwidth. Above all, the test base for an active setup is necessary, which will show its potential to bring backdrivable operation to hydraulic linear actuation.

CONCLUSION

The purpose of the new MR piston head design is improving the backdrivable capabilities of hydraulic linear actuator devices. Moreover, we aimed to shrink the gap between the practical utilization of MRF and theory, as well as to increase the MR devices by changing the customization options in order to suit the particular application demands. Through the data and results shown in the evaluation, we learned that there do exists the limitation for the proposed system in current force performance, but the prototype still performs well in speed force characteristics. In addition, it also realizes a more stable output in the force application than the conventional annular piston. Nevertheless, in order to build the toroidal piston into a fully functional suitable alternative to current technologies, there still remains a lot of work to be done, such as increasing the customization options and manufacturing the parts precisely to reduce friction for enhanced backdrivability.

ACKNOWLEDGMENTS

This work was supported by Strategic Advancement of Multi-purpose Ultra-human Robot and Artificial Intelligence Technologies (SAMURAI), the New Energy and Industrial Technology Development Organization (NEDO), JSPS KAKENHI Grant No. 16K06196 and 25220005, and the Research Institute for Science and Engineering, Waseda University.

REFERENCES

SENSING FLUID PRESSURE WITH Co RICH Fe-Co SYSTEM MAGNETOSTRICTIVE ALLOY TUBE

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** Fracture and Reliability Research Institute,
Tohoku University
*** Jun-ichi Nishizawa Research Center,
Tohoku University

Abstract: This paper proposes a new pressure sensor with Co rich Fe-Co system magnetostRICTive alloy tube. If fluid oil in a magnetostRICTive tube is pressurized, the tube is strained by fluid power and its magnetic permeability is drastically changed by the magnetoelastic effect or Villari effect. In order to capture the magnetic flux density change from the change in magnetic permeability, Hall IC is placed around the outside of the magnetostRICTive tube, and a pressure sensor is placed in the pressure path as a reference sensor to monitor actual oil pressure. The result shows a good relation between the pressure and magnetic permeability of the Co rich Fe-Co magnetostRICTive tube. It is also characterized this sensing system can monitor the pressure outside the tube, which means that indirect sensing is not a necessary concern for changing the sensor sensitivity due to the combustion deposits in case of the application of diesel engine’s transducers.

Keywords: Magnetoelastic effect, Villari effect, pressure sensor, transducer, Hall IC, magnetic permeability

INTRODUCITON

Transducer sensors are as an important method of measuring pressure inside diesel engines or gas engines because it needs to manage the high combustion efficiency of those engines and their running cost. The sensing environment is required to maintain a high temperature of up to 350°C and a high pressure of up to 25MPa (Overload 40MPa) when the engines run. Therefore, transducers have limited types based on Piezo, diaphragm made of low conductivity material, and etc [1]. That transducer needs the sensor head to be inserted into a pipe and be sealed in order to measure the fluid pressure directly. If engine combustion in a pipe tube contains combustion deposits, some deposits attach to the surface of the sensor head. Once those deposits attach to the surface, it’s really hard to eliminate them unless taking out the sensor from the pipe and cleaning it carefully. That deposition would affect sensor sensitivity, and might not monitor the true pressure value accurately. And as another concern, when the engine oil fuel is combusted, the combustion pressure drastically heats the sensor head like a heat shock from compressed gas. Taking into the consideration of heating effect, the sensors are required to be of a low sensitivity variation with temperature at mounting position because engine gas temperature drastically changes up to around 350°C. A transducer sensor needs sufficient stability and robust specification.

This paper proposes a new method of indirectly measuring inside pressure with magnetostRICTive material tube. First, FIGURE1 shows the overview of the magnetostRICTive effect [2]. When a magnetostRICTive material plate is placed in the magnetic field H, the material strains by Δl. This effect depends on a material characteristic itself. MagnetostRICTion constant λ is indexed with λ = Δl / l, setting the material length to l and strain amount to Δl. The λ values of representative Galfenol [3][4] and Tarfenol [5][6] called giant magnetostRICTive materials are shown in TABLE1. When the material is loaded with the external compressive force F, the material magnetized with M1 is strained by Δl2, meanwhile magnetization changes M1 to M2, called the Villari effect [7][8]. This effect is also called the inverse magnetostRICTion effect [9][10]. In case of tensile force like F1, magnetization changes M1 to M3 as well.

A new concept is designed to measure the inside pressure from outside by using the magnetostRICTion effect in shown FIGURE2. When the magnetostRICTive material is extended by compressed pressure like tensile force F3 in shown FIGURE1, magnetic permeability would be changed. If it is possible to measure the changing magnetic permeability of magnetostRICTive material, it would monitor inside oil pressure in the tube from outside. In addition, it’s easy to redesign the tube in terms of outer size and thickness to suit the maximum pressure of the fluid that it contains.
Co rich Fe-Co magnetostriction alloy developed by Tohoku University has high Curie point in TABLE 1 [11] [12], and it is understood that Fe-Co consists of inexpensive metals, not including the rare earth metal. Co rich Fe-Co alloy is investigated to apply for a transducer sensing the pressure.

Table 1. Magnetostriiction constant and Curie point

<table>
<thead>
<tr>
<th>Material</th>
<th>$\lambda$ (ppm)</th>
<th>Curie temperature ($^\circ$C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Terfenol(Fe-Dy-Tb)</td>
<td>1000</td>
<td>350</td>
</tr>
<tr>
<td>Gallenol (FeGa)</td>
<td>200</td>
<td>700</td>
</tr>
<tr>
<td>Fe-Co71</td>
<td>150</td>
<td>960</td>
</tr>
</tbody>
</table>

Magnestrictive Co rich Fe-Co alloy

First, Magnestrictive Co rich Fe-Co alloy is examined to meet the transducer specification as a tube material.

(A) Strain amount $\lambda$.

The distortion amount of the magnetization for the Co rich Fe-Co magnetostrictive alloy was investigated. A sample plate as shown in FIGURE 3 was made of Co rich Fe-Co material, a strain gauge (Kyowa: KFG - 1 N - 120 - C 1 - 11) was mounted on to a plate 5x2mm with glue (Kyowa: CC-33A). The strain amount $\lambda$ is calculated by the equation as shown in Eq. (1), and the result of strain amount is shown in TABLE 2. The result shows almost the same result as the result of TABLE 1, and then the magnetostrictive effect of Co rich Fe-Co metal material was confirmed.

Horizon : $H_{//}$

Vertical : $H_{\perp}$

FIGURE 3. Sample dimension and direction of Magnetic field
\[ \lambda = (H_{\parallel} + H_{\perp}) \times 2/3. \]  

(1)

<table>
<thead>
<tr>
<th>Sample</th>
<th>Saturation Magnetostiction (\lambda) (ppm)</th>
<th>Coercivity (H) (Oe)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No1</td>
<td>193.3</td>
<td>169.5</td>
</tr>
<tr>
<td>No2</td>
<td>193.3</td>
<td>134.8</td>
</tr>
</tbody>
</table>

(B) Temperature stability of magnetic property
Assuming the usage of transducer sensor under the high temperature for engine combustion, the magnetic stability is examined at room temperature RT and temperature 300°C. Co rich Fe-Co alloy was cut to the size of 3.5x5 mm plate, and put into VSM (Toei Industry VSM-5) to measure M-H curve under a fixed dc magnetizing field \(H\).
As a result, there is no significant difference on M-H curve between the case of RT and that of 300°C. The magnetic character of Co rich Fe-Co alloy is stable between under RT and 300°C.

(C) Designing Fe-Co alloy tube
In order to design a robust Fe-Co tube against engine combustion pressure, Co rich Fe-Co test pieces are made for tensile test to determine the limited maximum force from upper yield point, and were tested by Tensile testing machine (SHIMAZU AG-25TA).

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TABLE 3. Upper yield point result and tensile strength result

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Upper yield point (MPa)</th>
<th>Tensile strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room Temperature (~23°C)</td>
<td>---</td>
<td>690</td>
</tr>
<tr>
<td></td>
<td>414</td>
<td>715</td>
</tr>
<tr>
<td>350°C</td>
<td>---</td>
<td>688</td>
</tr>
<tr>
<td></td>
<td>---</td>
<td>516</td>
</tr>
<tr>
<td></td>
<td>---</td>
<td>522</td>
</tr>
<tr>
<td></td>
<td>---</td>
<td>530</td>
</tr>
</tbody>
</table>

As the results, it is confirmed that there is comparable mechanical strength to that of carbon steel.

In order to determine the actual tube sizes, outer radius and inner radius to withstand demanded pressure and temperature are calculated, showing a sectional view of a tube in FIGURE6.

When the inner radius and outer radius are set to \( r_a \) and \( r_b \), the tube thickness is set to \( t \) and inside tube pressure is set to \( p \), the strain stress \( \sigma_0 \) is calculated by the following Equation (2). When the critical pressure, strain stress \( \sigma_0 \) is obtained from the upper yield point of TABLE 3. If the outer radius \( r_b \) is 3 mm or 5mm, the inner radius \( r_a \) (thickness \( t \)) is calculated as shown in TABLE4 from equilibrium equation, case of critical pressure.

FIGURE6. Profile of a cylinder tube

\[
\sigma_0 = p \left( \frac{r_b}{r_a} \right)^2 + 1 \frac{\left( \frac{r_b}{r_a} \right)^2 - 1}{p} \tag{2}
\]

TABLE 4. Inner radius and thickness of FeCo tube

<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Factor of safety</th>
<th>Inner radius ( r_a ) (mm)</th>
<th>Thickness ( t ) (mm)</th>
<th>Inner radius ( r_b ) (mm)</th>
<th>Thickness ( t ) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25 (Maximum)</td>
<td>---</td>
<td>2.8</td>
<td>0.17</td>
<td>4.7</td>
<td>0.28</td>
</tr>
<tr>
<td>40 (Overload)</td>
<td>1.0</td>
<td>2.7</td>
<td>0.26</td>
<td>4.6</td>
<td>0.43</td>
</tr>
<tr>
<td>80</td>
<td>2.0</td>
<td>2.4</td>
<td>0.50</td>
<td>4.2</td>
<td>0.84</td>
</tr>
</tbody>
</table>

If a pressure sensor has sensing range from 0 MPa to 25MP to monitor the pressure and overload is 40MPa, inner radius and thickness is calculated in shown TABLE4.

FIGURE7 shows the picture of FeCo tubes made of Co rich Fe-Co alloy ingot. The left tube is \( \phi 6 \)mm with 0.5mm thickness; right tube is \( \phi 10 \)mm with 1.0mm thickness, satisfied with safety 2.0 if the safety 1.0 is set to 40MPa at overload point from the TABLE4. This picture also shows a good material workability of Fe-Co alloy.

FIGURE7. Left: \( \phi 6 \)mm tube with 0.5mm thickness, Right: \( \phi 10 \)mm with 1.0mm thickness made of Co rich Fe-Co alloy
EXPERIMENTS

First, the FeCo tube which has 6mm diameter with 0.5mm thickness is evaluated to be robust against up to 40MPa when oil is compressed in the tube by pumping GE sensing DPI610HC under the room temperature (around 23°C). It is confirmed that there are no cracks of neither tube nor oil leakage after keeping 40MPa oil in the FeCo tube for a couple hours.

Second, in order to measure the magnetic flux leakage around Co rich FeCo tube, Hall IC (Asahi Kasei Microdeiveces EQ-431L, 65mV/mT) is placed, and SmCo magnet (W5xD5xH10m, 438mT measured at the surface) is placed at the opposite position to the FeCo tube. Fe-Co tube, Hall IC and magnet are assembled shown in FIGURE8 as a prototype pressure sensor.

As a reference sensor, pressure sensor (Nagano Keiki KM31-174-50) is set in test line to monitor actual oil pressure in a test line as shown in FIGURE8 right picture. With the hand pump, the oil in the line was pressurized or decompressed several times to control the oil pressure.

FIGURE8. Experiment setup of sensing the magnetic flux around Fe-Co tube during pumping the oil in a tube

FIGURE9 shows the foot print of output of Hall IC and pressure sensor during pumping. It is understood, as the measured oil pressure increases, the measured magnetic flux density also increases. The magnetic flux density is able to calculate from Hall IC result, so the linearity between oil pressure and the magnetic flux density detected around FeCo tube is plotted in shown FIGURE10. The result shows that the result has a good relation between oil pressure and magnetic flux density.

![Graph](image)

FIGURE9. History of Hall IC and pressure when pumping oil
SUMMARY AND CONCLUSION

A new concept transducer has been designed based on a magnetostrictive tube made of Co rich Fe-Co alloy developed by Tohoku University.

First, the material characteristic of Co rich Fe-Co is evaluated as being a magnetostrictive material and a FeCo tube for an engine transducer. The magnetic stability during temperature changes of Co rich Fe-Co material is confirmed from M-H curve result by VSM even when the temperature is 300°C. From tensile test results, the FeCo tube diameter and thickness are calculated and determined.

Second, the detected magnetic flux by Hall IC outside φ6mm FeCo tube is related to the output sensing value of the actual pressure inside the tube in comparison with the reference pressure sensor being at room temperature. This system monitors the inside oil pressure from outside. It is also recognized that magnetostrictive material Co rich Fe-Co is a very useful material and easy to control and work as magnetostrictive material to make use of Villari effect.

In the future, since there are many other items in the severe specification to apply this system for an engine transducer, it is especially necessary to confirm whether this system also has good linearity at high temperatures (300°C) between oil pressure and magnetic flux density at first. In case of usage for diesel engines, the heated tubes temperature is much higher than Hall IC operation temperature even if Hall IC doesn’t attach the heated tube. Another method of sensing or device will become necessary to replace Hall IC in the actual operation applications.

ACKNOWLEDGMENTS

We thank Professor Fumio Narita, Tohoku University, Japan for his support of the analysis of magnetic flux leakage density from the FeCo alloy pipe as well as the sensor structure of the closed magnetic circuit in this test.

REFERENCES


DEVELOPMENT OF MANIPULATOR USING A GAS–LIQUID PHASE-CHANGE ACTUATOR

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Abstract. The purpose of this study is to develop a manipulator driven by a miniaturized artificial muscle in which a tiny compressor can be installed. Pneumatic actuators, such as pneumatic artificial rubber muscles (PARMs), have been widely used in many industrial and robotic research applications because they are compact and lightweight. However, the compressors driving such actuators are relatively large, and the peripheral devices such as filters and valves also tend to be large. To solve this size problem, the authors have been researching soft actuators driven by gas–liquid phase changes (GLPCs). In this research, a manipulator using an artificial rubber muscle driven by GLPCs was fabricated and a gripping experiment was conducted.

Keywords: Pneumatic artificial rubber muscle, Gas–liquid phase change, Pressure control, Manipulator

1. INTRODUCTION

The purpose of this study is to develop a manipulator driven by a miniaturized artificial muscle in which a tiny compressor can be installed. Pneumatic actuators, such as pneumatic artificial rubber muscles (PARMs) (FIGURE 1 (a)), have been widely used in many industrial and robotic research applications. PARMs are a type of pneumatic actuator driven by air pressure used to imitate muscle contractions. Air pressure supplied to the inside of a PARM results in its expansion in the radial direction and contraction in the axial direction. This process generates a displacement, as shown in FIGURE 1 (a). PARMs are flexible and lightweight, and they can generate a pulling force several times that generated by an air cylinder of the same diameter. In addition, PARMs have beneficial properties such as explosion proofing, action holding, and applicability to robots that require soft-touch operation. The use of PARMs in manipulators and power-assist devices has also been studied [1] [2]. However, the compressors driving such actuators are relatively large, and the peripheral devices such as filters and servo valves (FIGURE 1 (b)) also tend to be large. To solve this size problem, the authors have been researching soft actuators driven by gas–liquid phase changes (GLPCs) [3] [4] [5]. By applying feedback control of the inner pressure generated by GLPCs, a relatively high-response actuator with a time constant of several tenths of seconds was realized.

(a) Pneumatic artificial rubber muscle (PARM) (b) Peripheral devices

FIGURE 1. Pneumatic artificial rubber muscle (PARM) and the peripheral devices necessary to drive the PARM.
In this paper, the feedback control method of a PARM driven by GLPCs is summarized. Then, the design and fabrication of a manipulator using an artificial rubber muscle driven by GLPCs are explained. Finally, a performance test of the manipulator is presented.

2. CONCEPT OF ACTUATOR DRIVEN BY GAS–LIQUID PHASE CHANGES

2.1 Gas–liquid phase change (GLPC)
GLPC is a phenomenon in which a substance transitions from liquid to gas or from gas to liquid. When a liquid is heated, it begins to boil and transitions to the gas phase, and this transition causes its volume to expand and the pressure in the container to increase. When the substance is removed from the heat source, it loses its energy and returns to the liquid phase because of the heat transfer from inside the container to the ambient environment. Consequently, the substance volume contracts and the pressure decreases.

2.2 Actuator driven by GLPCs
FIGURE 2 illustrates the concept of an actuator driven by GLPCs. The actuator takes advantage of the phenomenon of volume expansion during the liquid-to-gas phase change. A working fluid is added to an actuator such as a PARM, and an electric heater (e.g., constantan heater) is installed. When the heater in the actuator is powered on, the heated liquid expands and boils into gas. The actuator is driven by the pressure generated by the volume expansion due to thermal expansion of the liquid and the following GLPC. When the heat source is removed, the generated gas returns to liquid and the volume contracts. Because the GLPC actuator does not require a compressor and other peripheral pneumatic components, it is possible to miniaturize the entire apparatus for driving the actuator.

2.3 Working fluid
In this study, the fluorocarbon \( C_5F_{11}NO \) was used as the working fluid. Its characteristics are compared with those of water in TABLE 1. Because this fluorocarbon has a low boiling point of 50 °C at atmospheric pressure and a heat of vaporization of 104.65 kJ/kg, which is 1/22 that of water (2260 kJ/kg), a small thermal energy supply can induce the liquid-to-gas phase change. The coefficient of thermal expansion of the fluorocarbon at 20 °C is 0.00154 °C\(^{-1}\), which is approximately seven times that of water. Moreover, it is non-poisonous, incombustible and its properties are not changed by heating (according to its product manual issued by 3M). As such, this working fluid is very suitable for the GLPC actuator.

![FIGURE 2. Concept of actuator driven by GLPCs.](image-url)

<table>
<thead>
<tr>
<th>Working fluid (Chemical formula)</th>
<th>Boiling point (1 atm) [°C]</th>
<th>Heat of vaporization [kJ/kg]</th>
<th>Coefficient of thermal expansion [°C(^{-1})]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluorocarbon ( C_5F_{11}NO )</td>
<td>50</td>
<td>104.65</td>
<td>0.00154</td>
</tr>
<tr>
<td>Water ( H_2O )</td>
<td>100</td>
<td>2257</td>
<td>0.00021</td>
</tr>
</tbody>
</table>
3. PARM DRIVING EXPERIMENT USING GLPCs

3.1 Experimental configuration
A PARM driving experiment was conducted on the configuration shown in FIGURE 3. The PARM used in this research is FESTO MXAM-5-AA and its specifications are shown in FIGURE 3. Beneath the PARM, a fixed container with a volume of 3.93 cm³ is installed. The PARM and the container are filled with the fluorocarbon working fluid. The working fluid is heated and boiled by powering the constantan heater (Tokyo Wire Works, Ltd., diameter: 0.231 mm, resistance per length: 16.02 Ω/m, total resistance: 3.7 Ω) at the bottom of the device. In addition, a pressure sensor (SMC PSE510-R06) is installed between the PARM and the container. The control signal (voltage: 0–10 V) is generated by the digital signal processor (DSP) (MTT s-BOX). By inputting the control signal into the power source, a voltage of 0–35 V is applied to the heater. The working fluid in the container is heated by the heater to induce the GLPC. The pressure generated by the GLPC is measured by a pressure sensor and the measurement is sent back to the DSP as the feedback control, as shown in FIGURE 4.

3.2 Experimental procedure and result
In this experiment, the PI control gains were set as follows: proportional gain: 1000 V/Pa, integral gain: 3 V/(Pa·s). The reference pressure P_ref was initially set at 0.3 MPa (gauge). At 40 s, P_ref was increased to 0.35 MPa and then at 80 s, it was decreased back to 0.3 MPa. The generated pressure P was measured by the data logger. The experimental result is shown in FIGURE 5. The time constants were 0.51 s when the pressure was increased from 0.3 to 0.35 MPa, and 0.37 s when the pressure was decreased from 0.35 to 0.3 MPa. During the period of 40 s to 80 s, the pressure was almost steadily kept at 0.35 MPa. According to our former report [5], the pulling force generated by the PARM differs depending on its contraction ratio, shown in FIGURE 6. At the inner pressure of 0.35 MPa, when the contraction ratio is 0.04, the pulling force generated by the PARM is approximately 20 N.

FIGURE 3. Experimental configuration of PARM driving experiment using GLPCs.

FIGURE 4. Block diagram of PARM driving experiment using GLPCs.
4. MANIPULATOR DRIVEN BY PARM USING GLPCs

4.1 Design and fabrication of manipulator
In this section, a manipulator whose driving force is generated by a PARM using GLPCs is designed and fabricated. The fabricated manipulator is shown in FIGURE 7. Most of the components used in this manipulator are the same as those in FIGURE 3. The manipulator consists of a container (3.93 cm$^3$), a constantan heater (Tokyo Wire Works, Ltd., diameter: 0.231 mm, total resistance: 3.7 Ω), a pressure sensor (SMC PSE510-R06) and working fluid (fluorocarbon). At the top of the PARM, a manipulating hand was newly attached. According to the kinematic calculation, when the pulling force generated by the PARM is 20 N, the hand can generate a gripping force of about 5 N.

4.2 Manipulator driving experiment (Gripping)
The experimental configuration for the manipulator driving experiment is shown in FIGURE 8. As stated in Section 3.2, the PI control gains were set as follows: proportional gain: 1000 V/Pa, integral gain: 3 V/(Pa·s). The reference pressure $P_{\text{ref}}$ was initially set at 0.3 MPa (gauge). At 40 s, $P_{\text{ref}}$ was increased to 0.35 MPa and then at 80 s, it was decreased back to 0.3 MPa. The gripping force was measured by a load cell (Tokyo Sokki, TCLZ-20NA, Dynamic Strain Meter DA-18A). The generated pressure $P$, gripping force $F$, and voltage to the heater $E$ were measured by the data logger.
The experimental results are shown in FIGURE 9. The time constants were 0.18 s when the pressure was increased from 0.3 to 0.35 MPa, and 1.35 s when the pressure was decreased from 0.35 to 0.3 MPa. During the period of 40 s to 80 s, pressure $P$ was almost steadily kept at 0.35 MPa, and therefore the gripping force was kept at around 5.6 N.

(a) Front view

(b) Upper side view

FIGURE 7. Fabricated manipulator driven by a PARM using GLPCs.

FIGURE 8. Experimental configuration of manipulator driving experiment.

5. CONCLUSIONS

In this paper, the feedback control method of a PARM driven by GLPCs is summarized. Then, the design and fabrication of a manipulator using an artificial rubber muscle which is driven by GLPCs are explained. Finally, a performance test of the manipulator is presented.
FIGURE 9. Experimental results of manipulator driving experiment.

ACKNOWLEDGMENTS

This research is funded by the Strategic Research Foundation Grant-Aided Project for Private Universities from MEXT.

REFERENCES

A PUMP USING EHD FLUID

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Abstract. One of an electrohydrodynamics (EHD) phenomenon is the induced flow of an EHD fluid in the presence of an electric field. In this paper, we describe a small pump in which the flow is generated by such an EHD phenomenon. In the case of pumps based on cylindrical electrodes, which thus far have been the focus of our research on EHD pumps, the total circumference length of the edge of the holes in the electrode that generates the rotational flow is small, leading to a small overall one-directional flow velocity. To investigate the influence of the total circumference length of the edge of the holes in the electrode, the area of the holes and the electric field intensity on pressure-flow rate characteristics, we produced ten different electrode pumps in which the electrode contains multiple holes. We measured the pressure-flow rate characteristics of our pumps and compared their performances.

Keywords: Electrohydrodynamics (EHD), Pump, EHD fluids, Flow visualization

1. Background/ Objectives and Goals

The induction of an EHD fluid flow through the application of an electric field is one of an electrohydrodynamics (EHD) phenomenon. In this paper, we describe a small pump in which the flow is generated by such an EHD phenomenon. Although EHD pumps have been studied in recent years, simplifying their structure to improve their efficiency and compactness remains an important challenge. Many electrode shapes were designed in various research organizations and have been tested. For example, mesh electrode pair was designed by Sakurai et al in Ashikaga Institute of Technology [1], and Triangular-slit electrode pair was designed by Gu et al in Tokyo Institute of Technology [2]. However, it has not been lead to practical use yet and there is still room for the development in the structure of the EHD pump (e.g. electrode shape, distance between electrode and width of electrode). To this end, in this study we focused our attention on the electrode shape, which is an important element of EHD pumps. Thus far, we have been researching EHD pumps based on cylindrical electrodes developed by our previous study[3][4]. For such a geometry, the total circumference length of the edge of the holes in the electrode that generates the rotational flow is small, resulting in a small one-directional flow velocity. Therefore the multi-holes electrodes were developed to increase the circumference length [5][6]. We designed new pumps containing multiple holes in the electrode pair to produce the rotational flows. As the results, the pressure-flow rate characteristics was modified with the pumps. However the influence of the parameters on the characteristics is not clear because the area of the holes also increased with the length of the edge of the holes in the electrodes.

In the present study, our aim is to investigate the influence of the total circumference length of the edge of the holes in the electrode, the area of the holes and the electric field intensity on the pressure-flow rate characteristics. Our pumps are structurally simple, easy to fabricate, and the flow direction can be reversed by simply swapping the positive and negative electrodes. Previously, we used liquid crystal (MJ-0669) as the EHD fluid in our pumps; however, due to its high cost, we decided to employ less expensive EHD fluids such as silicone oil and two hydrofluoroethers (HFE-7100 and HFE-7300) in this study. The physical properties of these EHD fluids are summarized in TABLE 1. We compared the behavior of various EHD fluids exhibiting the EHD phenomenon and selected the best performing fluid (HFE-7100) to investigate the pressure flow rate characteristics of the newly designed pumps.
TABLE 1. Physical properties of various EHD fluids

<table>
<thead>
<tr>
<th>No.</th>
<th>Fluids</th>
<th>Operating temperature region{[°C]}</th>
<th>Kinematic viscosity ([\text{mm}^2/\text{s}]) (25°C)</th>
<th>Relative permittivity (25°C)</th>
<th>Density ([\text{g/cm}^3]) (25°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Liquid crystal</td>
<td>~45.9</td>
<td>11.0</td>
<td>3.9</td>
<td>11.2</td>
</tr>
<tr>
<td>2</td>
<td>Silicone oil</td>
<td>~160</td>
<td>10</td>
<td>2.65</td>
<td>0.94</td>
</tr>
<tr>
<td>3</td>
<td>HFE-7100</td>
<td>~61</td>
<td>0.38</td>
<td>7.52</td>
<td>1.52</td>
</tr>
<tr>
<td>4</td>
<td>HFE-7300</td>
<td>~98</td>
<td>0.7</td>
<td>6.14</td>
<td>1.66</td>
</tr>
</tbody>
</table>

2. Experimental apparatus

2.1 Flow visualization

To better understand the EHD-induced flow velocity characteristics of the new EHD fluids being employed, flow visualization was carried out by particle image velocimetry (PIV). The flow channel and a plate electrode pair used in these measurements are illustrated in FIG. 1. The former was made of transparent acrylic resin and the latter was produced by etching copper. Each electrode plate had a width of 1 mm and the gap between the two plates was 0.2 mm. An electric field was generated at the bottom of the flow channel by applying a voltage to the plate electrode pair. Specifically, the cathode and anode of a DC power supply were connected to electrodes A and B, respectively.

To enable visualization, the EHD fluid was first mixed with green fluorescent polymer microspheres (Duke Scientific Corporation Co., Ltd, particle diameter 8 µm), after which the mixture was introduced into the flow channel. The channel was placed on a hot plate and the temperature of the mixture was maintained at 30 ± 1 °C. A voltage was then applied to the electrode pair to generate an electric field, thus initiating the fluid flow. A laser sheet was used to illuminate the center of the flow channel, and the flow behavior was observed from the front of the flow channel using a high-speed video camera (Photron Co., Ltd, ultima RGB). Similarly, the laser sheet was then used to illuminate the flow channel from the side, and the camera was positioned above the flow channel. The two video recording configurations are illustrated in FIG. 2. The flow was observed both above the electrode pair and in the entire channel during experiments, which were conducted at 1.5 kV and 2.0 kV.

2.2 Multi-holes electrode pump characteristics

Ten types of multi-holes electrode pumps, termed Types I ~ X which are illustrated in Figs 3(a) ~ 3(j) respectively. The disk electrodes used in the Type I pump contain seven holes (1 mm diameter), and those used in Type II contain nine holes (1 mm diameter) to reduce the flow resistance. The distance between the electrodes of Type I and Type II were 1mm. Type III has nine holes (1mm diameter) and the distance between the electrodes was reduced to 0.4mm to increase electric field strength between the electrodes. Type IV has same electrodes of Type II and Type III. However, the distance between the electrodes was 0.2mm. Type V has 18 holes (0.5mm diameter) and distance between the electrodes was 1 mm. Type VI has 36 holes (0.5mm diameter) and distance between the electrodes was 1mm. The number and the diameter of holes of Types VI~X are shown in Figs.3(f)~3(j). The electrode material was copper and the flow channel between the electrodes was produced from transparent acrylic resin. An electric field was generated in the flow channel of the multi-hole...
electrode pair by connecting the cathode and anode of a DC power supply to electrodes A (left side) and B (right side), respectively. As illustrated in FIG.4, the one-directional flow generated by the electric field is caused by rotational flows.

A schematic depiction of the entire experimental apparatus is shown in FIG.5. In this figure, H is the total pump head and θ is the angle of the acrylic board with respect to the floor. The multi-hole electrode pumps were connected in series using silicon pipes. The temperature of the EHD fluid was maintained at 30 ± 1 °C using a hot plate. The EHD fluid flow velocity was measured by recording the change in the fluid level in the discharge pipe of the pump with time, and the flow rate Q was calculated by multiplying the measured flow velocity by the cross-sectional area of the pump. When the flow rate was zero, the pressure was at its maximum value.

FIGURE.3. Schematics of the multi-hole electrode pumps

FIGURE.4. Flow in a pump

FIGURE.5. Schematic depiction of the experimental apparatus
3. Results

3.1 Flow visualization

The results of the PIV visualization of HFE-7100 (applied voltage, 2.0 kV) carried out with the plate electrode pair are shown in FIG. 6. The cathode and anode of the DC power supply were connected to electrodes A and B, respectively. We located the rotational flow between the electrodes and noticed that it induced a one-directional flow around the entire flow channel with a relatively steady velocity. A flow velocity measurement point was a section of the center of a channel of the side opposite to the channel in which the electrode is installed. We photographed the state of one-second the flow using a high-speed camera (500FPS), and displayed velocity vector using PIV (Particle Image Velocimetry), and took the average for 500 frames of the velocity vector. Five velocity vectors from average of 500 frames were sampled. The flow velocity in the channel is calculated from the average value of five velocities. The one-directional flow velocity is 14.7mm/s for 1.5kV and 23.1 mm/s for 2kV.

TABLE 2 compares our current and previous experimental results by Miyahara et al[3], and revealed that of all the functional fluids examined, HFE-7100 exhibited the highest flow velocity for 1.5kV because the relative permittivity is high and the kinematic viscosity is low as shown in TABLE 1. We therefore chose to use HFE-7100 to investigate the characteristics of the multi-hole electrode pumps, as detailed below.

![Flow velocity map measured using PIV near the electrode(HFE-7100:2.0kV)](image1)

![Flow velocity map measured using PIV in the entire flow channel(HFE-7100:2.0kV)](image2)

**FIGURE 6.** Flow visualization with the plate electrode pair using HFE-7100 and 2.0 kV

<table>
<thead>
<tr>
<th>No.</th>
<th>EHD fluid</th>
<th>One-directional velocity[mm/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Liquid crystal(MJ-0669)</td>
<td>0.48</td>
</tr>
<tr>
<td>2</td>
<td>Silicone oil(KF-96-10CS)</td>
<td>0.13</td>
</tr>
<tr>
<td>3</td>
<td>HFE-7100</td>
<td>14.7</td>
</tr>
<tr>
<td>4</td>
<td>HFE-7300</td>
<td>1.53</td>
</tr>
</tbody>
</table>

3.2 Multi-holes electrode pump characteristics

HFE-7100 was chosen as the EHD fluid for designing the multi-hole pump because the flow velocity was found to be the largest of those studied using the plate electrode experiment described in the previous section. First, the maximum pressure and maximum flow-rate of Type I～Type X was investigated at 3.0kV. The experimental results are summarized in TABLE 3. However, in Type IV, the dielectric breakdown occurred at 3.0kV. Therefore, Type VI pumps were measured at 2.0kV.
We performed the investigation using two Type I pumps connected in series and measured the pressure-flow rate characteristics at 2.5kV and 3.0kV. We compared the pressure-flow rate characteristics to those previously measured using a cylindrical electrode pair pumps and liquid crystal as the EHD fluid by Tsukiji and Miyahara[4]. The results, shown in FIG.7, reveal that the plot of pressure-flow rate is approximately linear. Furthermore, approximate lines through the data (various dashed lines in the figure) are translated in the direction of increasing pressure and flow rate when the applied voltage is increased. Therefore, it can conclude that the pressure and flow rate of the pump increase with the applied voltage. In addition, for the same applied voltage, the pressure and flow rate in the Type I pumps using HFE-7100 were higher than those in the cylindrical electrode pair pumps using liquid crystal. It was hypothesize that this is because the total circumference length of the edge of the holes that generates the rotational flow in the Type I pumps was larger (22.0 mm vs. 12.6mm) than that of the cylindrical electrode pair pump. However, changing the EHD fluid from liquid crystal to HFE-7100 is also thought to have been a major factor.

Next, we performed the experiment using five Type I pumps connected in series and measured the pressure-flow rate characteristics at 3.0kV. A comparison of the results with those of the two-type I setup is shown in FIG.8. Both pressure-flow rate plots are approximately linear, but the maximum pressure was about 2.5 times higher (188.9 Pa vs. 75.6 Pa) for the five-pump setup. We therefore concluded that the pressure is directly proportional to the number of pumps.

The pressure-flow rate characteristics of the two pumps also were measured at an applied voltage of 3.0kV using HFE-7100 as the EHD fluid. Typical results comparing Type I and Type II pumps are shown in FIG.9.
From the results shown in FIG.9, it was noticed that when the number of holes was increased, both the pump pressure and flow rate increased. Therefore, we can conclude that the pressure and flow rate of Type II pump were larger than those of Type I pump because the total circumference length of the edge of the holes to generate rotational flows of Type II pump was also larger.

The pressure-flow rate characteristics of the two pumps also were measured at an applied voltage of 3.0 kV using HFE-7100 as the working fluid. Typical results comparing Type II and Type III pumps are shown in FIG.10. Both pumps were produced using the electrode of nine holes(diameter 1mm). Type II has 1mm of the distance between the electrodes, however, Type III has only 0.4mm of the distance between the electrodes. Thus, electric field strength of Type III pump was stronger than that of Type II pump. Therefore, it was expected that the pressure and flow rate of Type III pump was larger than those of Type II pump. From the results shown in FIG.10, as expected the pressure and flow rate of Type III pump was larger than those of Type II pump. It was noticed that when the distance of between the electrodes was decreased, both the pump pressure and flow rate increased. The electric field strength increases when the distance of between the electrodes is reduced. Therefore, it can conclude that the pressure and flow rate increased because the electric field strength increased. In addition, it was noticed that the plot of pressure-flow rate is approximately linear.

Furthermore, approximate lines through the data (various dashed lines in the figure) are translated in the direction of increasing pressure and flow rate when the distance of between the electrodes is reduced. Tendency of the pressure-flow rate characteristic of comparing between Type II (applied voltage 3.0kV) and Type III (applied voltage 3.0kV) is similar to the pressure-flow rate characteristic of comparing between Type I (applied voltage 2.5kV) and Type II (applied voltage 3.0kV). It was thought that this is because the electric field is increased.

Next, results of maximum pressure and flow-rate comparing Type II, Type III and Type IV pumps are shown in FIG.11 and FIG.12. However, in Type IV, the dielectric breakdown occurred at 3.0kV. Therefore, Type IV pumps were measured at 2.0kV. Because applied voltages were different, it was evaluated by electric field strength not the distance between the electrodes.

From the FIG.11 and FIG.12, it was found that the maximum pressure and maximum flow-rate were increased almost linearly by increasing electric field strength. It is possible that maximum pressure and maximum flow-rate is increased by increasing electric fields strength.
Next, typical results comparing Type II, Type V, Type VII and Type VIII pumps are shown in FIG.13. Maximum pressure of four pumps for no flow are almost same, however maximum flow rate are different. The relation between the total area of the holes of the electrodes and maximum flow rate is shown in FIG.14.

From comparative results of the Type II, Type V, Type VII and Type VIII, it was found that it is possible to increase the maximum flow rate by increasing total area of the holes under the condition of the same length of total circumference of the holes.

![FIGURE.13. Flow rate-pressure data](image1)

![FIGURE.14. Total area of the holes-maximum flow rate data](image2)

Next, typical results comparing Type II, Type VI, Type IX and Type X pumps are shown in FIG.15. Maximum flow rate of four pumps for zero pressure are almost same, however maximum pressure are different. The relation between the length of total circumference of the holes of electrode and maximum pressure is shown in FIG.16.

From comparative results of the Type II, Type VI, Type IX and Type X, it was found that it is possible to increase the maximum pressure by increasing length of total circumference of the holes of electrode under condition of same total area of the holes of the electrodes.

From the above results, it can be concluded the pressure and flow-rate of multi-holes electrode pair pump can be increased by increasing length of total circumference of the holes of electrode, increasing the total area of the holes of the electrodes and decreasing the distance between the electrodes.

These pumps are aimed at small-size liquid-cooled cooling system. For this purpose, it is necessary to pressure of at least a few kPa. However, these pumps have not reached its pressure. Therefore the further improvement of the pump shape is necessary.

![FIGURE.15. Flow rate-pressure data](image3)

![FIGURE.16. Length of total circumference -maximum pressure data](image4)
4. Conclusions

We performed flow visualizations with a plate electrode pair for various EHD fluids and produced multi-hole electrode EHD pumps, for which we investigated the pressure-flow rate characteristics using HFE-7100. The conclusions of this study are as follows:

1. When a voltage was applied to the two electrodes positioned at the bottom of a flow channel filled with EHD fluids (HFE-7100, etc.), rotational flows were generated between the electrodes, which induced a one-directional flow in the channel.
2. For a given applied voltage, HFE-7100 exhibited the highest flow velocity among the EHD fluids studied because the relative permittivity is high and the kinematic viscosity is low.
3. The flow velocity increased with electric field strength for HFE-7100.
4. Using HFE-7100 as the EHD fluid in the multi-hole electrode pair pump, an approximately linear relationship was found between the pressure and flow rate. Both of these quantities increased proportionally with an increase in the applied voltage.
5. The pressure and flow rate of the multi-hole electrode pair pumps using HFE-7100 were larger than those of the previously studied cylindrical electrode pair pumps using liquid crystal for the same applied voltage.
6. The maximum pressure of the five Type I pumps connected in series was about 2.5 times that of the two Type I pumps connected in series. In other words, the maximum pressure is directly proportional to the number of pumps.
7. The pressure and flow rate of Type II pump were larger than those of Type I pump because the total circumference length of the edge of the holes to generate rotational flows of Type II pump was also larger.
8. When the distance of between the electrodes was decreased, both the pump pressure and flow rate increased because the electric field strength between the electrodes increased.
9. It is possible to increase the maximum flow rate of the pump by increasing the total area of the holes of the electrodes at the constant length of total circumference of the holes.
10. It is possible to increase the maximum pressure of the pump by increasing length of total circumference of the holes of the electrodes at the constant total area of the holes of the electrodes.

Acknowledgments

The authors thank the Techno center of Sophia University for their generous support.

5. References

A development of the new type tourniquet applying EHD phenomenon

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Abstract. A tourniquet is a medical device which compresses upper limbs or lower limbs, by using a cuff in orthopedic surgery in order to reduce the bleeding. This device has the problem that the compression of a limb by a fixed pressure for a long time gives the patient a burden. In order to solve this problem, we found out that it was effective to finely adjust the pressure according to the vital signs. However, the fine adjustment of the pressure of the pneumatic tourniquet is difficult. Then, we developed a new tourniquet system, which uses the EHD pump as the driving source, to solve this problem. In this study, we created a compact EHD pump aiming to miniaturize the tourniquet system and we made a prototype tourniquet driven by the EHD pump for human.

Keywords: Tourniquet, EHD phenomenon, Functional fluid

INTRODUCTION

Fluid power has already been deployed in various fields and extends to the medical field. Among several medical applications, we focused on the tourniquet which use pneumatic pressure. A tourniquet is a hemostatic device used during orthopedic surgery of upper limbs and lower limbs. For example, as shown in FIGURE 1, it consists of a driving source with a built in compressor and a clamping unit called a cuff. With this device, it is possible to stop the bleeding by expanding the cuff attached to the upper arm or the thigh with pneumatic pressure. By using a tourniquet, the surgeon can work smoothly while maintaining a clear view for surgery. However, when compressing a patient with a cuff, the limbs are generally compressed for a long time with a strong, constant pressure. In consequence, the burden on the patient increases which has begun to be regarded as a problem in recent years [1]. Moreover, these devices have many problems like the vibrations and the noises generated by the built in compressor, which hinder the surgeon’s concentration ability. In accordance with the occurrence of vibrations and noises, it is possible to deal with it by incorporating other elements, such as the installation of a silencer or a vibration isolation equipment. However, since it leads to an increase in the size of the apparatus and some complications of the control method, it is insufficient as a solution. Further, to prevent damages to patients, we discovered that the adjustment of the pressure according to the vital signs could reduce the burden on patients [2]. Still, it is difficult to finely adjust the pressure of the tourniquet driven by a compressor. Therefore, we decided to develop an unprecedented tourniquet system using a new driving source instead of the pneumatic pressure. We decided that fundamental countermeasures are necessary, rather than alleviating problems such as vibrations and noises.

1
The purpose of this study is to develop a new tourniquet that allows the adjustment of the compression pressure during surgery, and the reduction of sequelae after surgery. We focused on the EHD phenomenon, which creates a flow of fluid with electric fields without a mechanical system, as a new driving source alternative to a pneumatic one. In particular, this time we developed a compact EHD pump as a driving source, aiming at downsizing the tourniquet system constructed in the past, and conducted an evaluation of performances. With this EHD pump, we produced a prototype of EHD tourniquet for humans.

EHD PHENOMENON AND EHD PUMP [3]

The EHD phenomenon is an abbreviation for electrohydrodynamic phenomena. It is a phenomenon in which a flow of fluid is generated by applying a high voltage to two electrodes immersed in an electrically insulating fluid as shown in FIGURE 2 (a). With this phenomenon we developed an EHD pump capable of converting a flow between the electrodes in one direction, by arranging a tilted GND electrode placed above a flat plate positive electrode as shown in FIGURE 2 (b). Furthermore, it has already been found that the discharge pressure is increased by placing successively this electrode structure, as shown in FIGURE 3.
CONSTRUCTION OF THE EHD DRIVEN TOUNIQUET SYSTEM

We previously developed an EHD pump (unit type EHD pump), shown in FIGURE 4, for the realization of a new tourniquet. This EHD pump has a unit structure in which eight pumps are connected in series. The performance of this pump has been measured (FIGURE 5). It can discharge 40 [kPa] which is the maximum pressure used for hemostasis [4]. Furthermore it has a slope of 9 [kPa / kV] with a negligible hysteresis during the decompression. Therefore, it is possible to control the discharge pressure by simply adjusting the input voltage. Using this pump as a driving source, we constructed an EHD driven tourniquet system shown in FIGURE 6. However, this system emphasizes on the functional aspect, thus the components are simply arranged. If we use this system in the operating room, it is necessary to condense as one structured system. Therefore, it is necessary to miniaturize each element, and in particular, the EHD pump which is the driving source.

![FIGURE 4. Unit type EHD pump](image)

![FIGURE 5. Measurement result of pressure](image)

![FIGURE 6. EHD driven tourniquet system](image)

NEW COMPACT EHD PUMP

In order to miniaturize and integrate the tourniquet system, we began by developing a compact EHD pump, that replaces the unit type EHD pump used in the past tourniquet system. We aimed to create a compact EHD pump with the same discharge pressure as the unit type but smaller. The developed compact EHD pump is shown in FIGURE 7. In order to evaluate the performance of this EHD pump, pressure and flow rate were measured and compared with the unit type EHD pump previously manufactured. Moreover, we compared the size of each EHD pump.

![FIGURE 7. Compact EHD pump](image)
Measurement method of discharge pressure and flow rate

The pressure was measured by connecting a pressure gauge (PS-LLLW, Fujifilm Corporation) to the manufactured compact EHD pump as shown in FIGURE 8. The flow rate was measured by connecting a flow meter (Ultrasonic micro liquid flow meter μLF-100, Sonic Corporation) to the compact EHD pump as shown in FIGURE 9. The voltage during experiments was applied from 0 [kV] until the electric discharge occurs, with an interval of 1 [kV]. We repeated the measurements 3 times.

Measurement results of discharge pressure and flow rate

The discharge pressure of the compact EHD pump increase and decrease accordingly to the applied voltage as shown in FIGURE 10. Since the discharge pressure at the applied voltage of 9 [kV] was 42 [kPa], it was possible to apply the 40 [kPa] required for the hemostasis. Further, we could observe that the increasing and decreasing voltage curves are close, so the hysteresis was small. Moreover, on the range from 5 [kV] to 9 [kV], which was assumed to be the range of use during surgery, we obtained a slope of the near-straight line of 8 [kPa / kV]. It was observed that the flow rate of the compact EHD pump increases as the applied voltage rises, as shown in FIGURE 11. The flow rate when applying a voltage corresponding to the pressure output of 40 [kPa], was compared with the unit type EHD pump. The flow rate of the compact EHD pump was 5.4 [mL / s] and the flow rate of the unit type EHD pump was 5.6 [mL / s]. It could be said that they have the same ejection performance.

Comparison of size and mass

We compared the sizes of the unit type EHD pump and the compact EHD pump as shown in FIGURE 12. The size of compact EHD pump is 90 × 35 × 18 [mm], which is about 1/22 of the unit type EHD pump, the mass is 93 [g], which is about 1/8 of the unit type EHD pump. Both size and mass were greatly reduced.
As a result of the performance evaluation, it was found that the compact EHD pump can produce the pressure required for hemostasis, and can maintain the same flow rate as the unit type EHD pump. Moreover, it was able to realize hysteresis and resolution improvement and drastic miniaturization.

**EHD TOURIQUET FOR HUMAN**

Since the EHD pump has been miniaturized, we integrated it to a new EHD tourniquet aiming at the use in clinical experiments and surgery. The developed EHD tourniquet for human is shown in FIGURE 13. This device is composed by a cuff and a box with a control board. The power supply (700-BTL 017 BK, Sanwa Supply Inc.), the microcomputer, the compact EHD pump, the reservoir, and the pressure gauge are incorporated inside a 290 × 240 × 180 [mm] box. Since the power supply is a mobile battery, it can be used even in places without outlet. The chosen cuff (single cuff, Mizuho Corporation) is a product actually used in operating rooms. There are three red and two yellow switches on the control board. Red switches are the power switch of the microcomputer, the pressure gauge and the EHD pump. By pressing the yellow switches, it is possible to directly apply 30 [kPa] or 40 [kPa], which are often used during surgery. Furthermore, the pressure can be finely adjusted by a knob and the adjustment accuracy at the present stage is about 2 [kPa / kV]. It is also possible to interrupt the compression by pressing both yellow switches at the same time. When wearing and using the EHD tourniquet, it is possible to reach the set pressure in about 40 seconds after pushing the switch, to change the set pressure by pushing the other switch, and to adjust the pressure with the knob without any problem.

**CONCLUSION**

The purpose of this study was to develop a new tourniquet driven by an EHD pump. We developed a new EHD pump and a tourniquet for human. The developed compact EHD pump has a small hysteresis and a good resolution with a pressure output capacity equivalent of the previously developed unit type EHD pump but with a size of 1/22 and a mass of 1/8. Further, the developed EHD tourniquet for human can set the two frequently
used pressure by two switches, and a fine pressure adjustment is possible with the knob. Although the adjustment accuracy at the present stage is about 2 [kPa / kV], it is possible to easily improve the accuracy. In future, we will develop a tourniquet which the pressure can be controlled by a microcomputer in order to reduce the burden on the patient.

REFERENCES

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[2D50] PROPOSAL OF MEASUREMENT METHOD OF FRICTION COEFFICIENT FOR PIPE FLOW IN AQUA DRIVE SYSTEM
*Yasuo Sakurai¹, Takahisa Nagasawa² (1. Ashikaga Institute of Technology, 2. KOYO Seiki Co., LTD.)
10:45 AM - 12:05 PM

[2D51] CAVITATION PHENOMENON IN A SPOOL VALVE MODEL FOR AQUA DRIVE SYSTEM
*Hitomi Okabe¹, Yukiko Tanaka¹, Futoshi Yoshida², Shouichiro Iio³ (1. Shinshu university faculty of engineering, 2. KYB corporation)
10:45 AM - 12:05 PM

[2D52] DEVELOPMENT OF A DIRECT TYPE WATER HYDRAULIC RELIEF VALVE FOR SMALL FLOW RATE
*Kenji Suzuki¹, Yohichi Nakao¹, Tsutomu Iguchi², Futoshi Yoshida³ (1. Kanagawa University, 2. Hirose Valve Industry Co., Ltd., 3. KYB Corporation)
10:45 AM - 12:05 PM

[2D53] CHARACTERISTICS OF WATER HYDRAULIC CYLINDER
*Hideki Yanada¹, Yuhi Ito¹, Yutaka Fujimoto¹ (1. Toyohashi University of Technology)
10:45 AM - 12:05 PM

[2D54] A PREDICTION METHOD OF WATER HAMMER
*Yuhi YOSHIDA¹, Tatsuya UCHIDA¹, Kazushi SANADA² (1. Department of Systems Integration, Graduate School of Engineering Yokohama National University, 2. Faculty of Engineering, Yokohama National University)
10:45 AM - 12:05 PM

[2D55] TEST METHODS OF WATER HYDRAULIC PUMPS
*Tatsuya UCHIDA¹, Yuhi YOSHIDA¹, Kazushi SANADA² (1. Department of Systems Integration, Graduate School of Engineering, Yokohama National University, 2. Faculty of Engineering, Yokohama National University)
10:45 AM - 12:05 PM
PROPOSAL OF MEASUREMENT METHOD OF FRICTION COEFFICIENT FOR PIPE FLOW IN AQUA DRIVE SYSTEM

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Abstract. This paper deals with measurement method of friction coefficient for pipe flow in aqua drive system. A pump is used as a power source in aqua drive system and then pressure pulsation is generated. When designing aqua drive system, pipe friction is one of important factor and friction coefficient for pipe flow is required to calculate energy loss due to pipe friction. However, in general, friction coefficient for pipe flow had been determined through experiments using a head tank. Therefore, it is necessary to investigate friction coefficient for pipe flow under pressure pulsation. In this study, the analysis method based on interval estimation is introduced to eliminate the influence of pressure pulsation from the measured differential pressure. By using the analyzed differential pressure, friction coefficient for pipe flow is estimated.

Keywords: Aqua drive system, Pipe, Friction, Measurement, Confidence interval

INTRODUCTION

In design and improvement of an aqua drive system [1], it seems to be necessary to evaluate energy loss by pipe friction. In the estimation of the energy loss caused by pipe friction, it is required to determine friction coefficient for pipe flow. When measuring friction coefficient for pipe flow, in general, a head tank is used to prevent pressure pulsation by pump and to make steady flow. However, when designing an aqua drive system, it seems to be necessary to confirm friction coefficient for pipe flow in actual operating condition, where water is delivered by a pump. In such condition, pressures at upstream and downstream of a pipeline are required to be measured by pressure transducers. Then, the signals from these pressure transducers involve the influence of pressure pulsation by the pump. Therefore, to accomplish the above mentioned measurement, it is necessary to cope with these problems.

In this study to realize the above mentioned measurement, an analysis method based on interval estimation [2] is used when calculating the average values of the signals from the pressure transducers. By introducing this method, the influence of pressure pulsation by pump seems to be able to be reduced without employing an accumulator or the equipment to remove pressure pulsation.

EXPERIMENT AND ANALYSIS

Experimental apparatus to investigate friction coefficient for pipe flow is shown in Fig.1 and Fig.2. This apparatus is composed of a centrifugal pump, test pipeline involving straight pipeline of 2.0m, and a metering valve to set flow rate. Instead of piston pump, centrifugal pump is employed to carry out experiment up to large Reynolds number, namely almost $8 \times 10^4$. Diameter of the straight pipeline is 10.22mm. Upstream pressure $P_1$ and downstream pressure $P_2$ of the test pipeline were measured by semiconductor pressure transducers. The signals from these pressure transducers were recorded by a data logger. In this study, the sampling time of the data logger was set at 10μs.

When calculating the average values of the signals of the pressure transducers, an analysis method based on interval estimation is introduced. Sample size $N$ is determined, and sample average $\bar{x}$ is calculated. The standard deviation $S$ of sample can be calculated as follows:

$$ S = \sqrt{\frac{\sum_{i=1}^{N}(x_i - \bar{x})^2}{N - 1}} $$

(1)

Consequently, confidence interval for population mean value $\mu$ can be estimated as follows:
\[
\bar{x} - t \frac{S}{\sqrt{N}} \leq \mu \leq \bar{x} + t \frac{S}{\sqrt{N}}
\]  
(2)

where \( t \) is determined from \( t \)-distribution table.

**FIGURE 1.** Experimental apparatus

**FIGURE 2.** Photo of experimental apparatus

**EXPERIMENTAL RESULTS**

Based on measurement, the roughness of the inner wall of the test pipe is less than 10μm. Therefore, the test pipe is the hydraulically smooth pipe within this experimental range. Therefore, friction coefficient for pipe flow \( \lambda_e \) in experiment is compared with friction coefficient for pipe flow \( \lambda_p \) by the equation of Prandtl-Karman-Nikuradse. Furthermore, relative error \( e \) is calculated as follows:

\[
e = \left| \frac{\lambda_p - \lambda_e}{\lambda_p} \right| \times 100
\]  
(3)

Experimental results are described in Fig.3 and Table 1. When calculating \( P_1 \) and \( P_2 \), sample size \( N \) is determined so that these values are within a sufficiently narrow range with the probability of 99 percent. As can be seen from these results, when using the experimental apparatus and an analysis method based on interval estimation, \( \lambda_e \) can be predict with the precision of 5% from \( \text{Re}=6.0\times10^4 \) to \( 8.0\times10^4 \).

**CONCLUSIONS**

In this study, the measurement of friction coefficient for pipe flow was carried out in actual operating condition by introducing an analysis method based on interval estimation without an equipment to remove pressure pulsation. Consequently, it is made clear that friction coefficient for pipe flow can be estimated from \( \text{Re}=6.0\times10^4 \) to \( 8.0\times10^4 \) with the precision of 5%.
Next step is to make clear the reason why the precision is worse when $\text{Re} < 6.0 \times 10^4$.

![Figure 3. Comparison of $\lambda_\epsilon$ and $\lambda_p$](image)

### Table 1. Experimental and calculated results

<table>
<thead>
<tr>
<th>Q(L/min)</th>
<th>$P_1$ (MPa)</th>
<th>$P_2$ (MPa)</th>
<th>$\Delta P$ (MPa)</th>
<th>$\mu$ (m/s)</th>
<th>$\lambda_\epsilon$</th>
<th>$\lambda_p$</th>
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ACKNOWLEDGMENTS

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.

REFERENCES

CAVITATION PHENOMENON IN A SPOOL VALVE MODEL FOR AQUA DRIVE SYSTEM

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Abstract. Cavitation sometimes occurs serious problems in hydraulic system. Especially in water hydraulic system the influence becomes obvious because of high saturated vapor pressure of water. This study is focused on cavitation phenomenon in a spool valve for Aqua Drive System (ADS). Visualization of cavitation jet issuing from a control gap of a spool valve was conducted, and pressure-flow rate characteristics was measured for some differential pressure and for some gap widths. As a result, the cavitation jet behavior is dependent on the differential pressure and the gap width. The cavitation jet angle becomes steep with increase the gap width. The area of cavitation jet becomes larger for higher differential pressure. The both of the gap and the pressure has no influence on the pressure-flow rate characteristics for $\Delta P \leq 5.0 \times 10^6$ Pa.

Keywords: Spool valve, Cavitation, Visualization, Aqua Drive System, Pressure-flow rate characteristics

INTRODUCTION

Aqua drive system (ADS) that uses tap water as working fluid has been much attention in recent years [1]-[3]. The ADS has some advantages and disadvantages comparing with oil hydraulic system. The most different point between the ADS and oil hydraulic system is working fluid character especially for viscosity and saturated vapor pressure from a practical point of view. Oil hydraulics is easier to prevent leakage, rust, cavitation and to reduce friction than the ADS. On the other hand, the ADS can be supplied more hygienic, more safety, more environmentally-friendly system than oil hydraulics. Their markets are perfectly different for the reasons above. Oil hydraulics is for high-pressure smokestack industries. The demand of ADS is low-pressure fields; food processing machines, medicine and semiconductor manufacturing equipment.

In these driving system, it should be controlled speed, force and flow direction for actuators by spool valves. The spool valve can be realized precisely control them by change the valve stroke which has small flow passage gaps. The water velocity through these gaps is dependent on the differential pressure. The maximum pressure of the ADS is reached at $14 \times 10^6$ Pa, in this case, the water velocity is over 100 m/s and the pressure downstream the gap bellows the saturated vapor pressure. Thus, cavitation is easily generated in the ADS because of that water saturated vapor pressure is pretty higher than that of oil. So, cavitation is serious phenomenon for the ADS, it occurs some problems; erosion, noise and vibration, performance degradation. There are many studies on cavitation in the spool valve for oil hydraulic system whereas much less knowledge exists on the spool valve in the ADS for new technology.

This study is focused on cavitation generated in a spool valve for the ADS. It is not fully understood the cavitation behavior in the valve, so the authors aim to observe the behavior and to evaluate pressure-flow rate characteristics with cavitation experimentally.

EXPERIMENTAL PROCEDURE

Figure 1 illustrates a testing water hydraulic circuit. The working fluid is tap water. Dissolved oxygen concentration is approximately 6 ppm. Water was supplied with a piston pump unit. The output water pressure was regulated by a relief valve and the flow rate was controlled by pump rotation speed. The rotation speed was set to supply required water flow rate to the test section. The water flows into a test section, and then goes back into a reservoir under the pump unit through a chamber with a vacuum pump. Water flow rate was measured upstream the test section by an ultrasonic flowmeter. The pressures upstream and downstream the test section, $P_1$, $P_2$, were measured with pressure sensors individually. The downstream pressure, $P_2$, can be set arbitrary value by adjusting the suction time of the vacuum pump. The differential pressure between the test section was set of
ΔP=0.5~5.0×10^6 Pa(G). Cavitation and flow field were captured by a high-speed camera (Photron FASTCAM SA-X2) and CW-YAG laser (KANOMAX CW532-10-3W). The camera and light was mounted front and side of the test section as shown in Fig.1. The shutter speed and flame rate was set from 1/2325 to 1/2380 and 1000fps, respectively.

Figure 2 shows a spool model used in this experiment. The model is made of transparent acrylic resin for easy optical access around a control gap in the model. The control gap has a rectangular notch. The notch size is 2 mm×5 mm along circumferential and axial direction of the spool. The spool diameter is 12 mm, and the axial gap of the control area was selected three conditions, w=0.06, 0.24, 1.05 mm. There is no radial gap between the spool and the sleeve. The model has a large space downstream the notch to capture cavitation and flow behavior easily.

RESULTS AND DISCUSSION

Cavitation images were captured for 1 sec. by the camera. Figure 3 shows visualization results of cavitation jet issuing from the control notch at left-down location in the picture. These images are time averaged to evaluate mean flow field. Dark portion is the background and brighter region is cavitation. It is noticed that the issuing angle of cavitation jet is different for opening width, w. The inclination angle of the jet centerline from the horizontal is approximately 40°, 65°, 70° at w=0.06, 0.24, 1.05 mm, respectively. On the other hand, the angle does not change with the differential pressure, ΔP. The area of bright region is increased with increase of ΔP. From these results, it is clarified that the jet behavior can be changed with the gap and the differential pressure in a spool valve.

Figure 4 shows the relationship between the opening gap ratio and inflow angle for oil hydraulic spool valve [4]. As shown in this graph, the flow angle continuously changes with the gap ratio. Three plots in the graph shows

![Figure 1. Testing circuit for water hydraulic spool model](image1)

![Figure 2. Spool model](image2)
FIGURE 3. Time averaged images for 1 sec.

FIGURE 4. Variation of flow entry angle with control gap geometry of oil hydraulic spool valve [4]

the measurement result shown in Fig.3. The distribution pattern of these plots for water spool valve is almost corresponded with the line for an oil spool valve. The Reynolds number is difference of 10 times between these results. From this fact, it is thought that the angle is only dependent on the geometry.

Figure 5 demonstrates relationship between the differential pressure and the cavitation area. The cavitation area was measured from binarization image processing. Otsu’s method was used for threshold criteria of the binarization. It is easily recognized that the cavitation area grows with the differential pressure. But increasing rate and tendency is quite different in the both results of $w=0.06, 0.24$ mm. The reason has not been revealed yet.

Figure 6 illustrates the relationship between cavitation number and cavitation area. The cavitation number, $\sigma$ is defined by equation (1).

$$\sigma = \frac{P_r - P_t}{\Delta P} \quad (1)$$
Where, $P_v$ is the saturated vapor pressure of water. The cavitation area is increased with decrease of the cavitation number. It is interesting and should be clarified why the variation tendency is changed with $w$.

**FIGURE 5.** Differential pressure vs. cavitation area

**FIGURE 6.** Cavitation number vs. cavitation area

**FIGURE 7.** Pressure–flow rate characteristics

One of the problem with cavitation is blockage of flow passage area. Figure 7 shows the pressure–flow rate characteristics with cavitation. The experimental results are plotted in the graph for three gap conditions. Dotted lines are theoretical value which calculated by equation (2). Red and blue line shows the result for $C=0.7$ and $C=0.8$, respectively.
Where, $Q$, $A$, $C$, $\rho$ is flow rate, opening area at control gap, discharge coefficient, water density, respectively. The experimental values are almost corresponded with theoretical value. Here, it is noticed that experimental result for $w=0.06$ mm is close to the line of $C=0.7$, the results for $w=0.24$ and $1.05$ mm is close to the curve of $C=0.8$. It is naturally thought that the variation shows opposite tendency if some influence by cavitation, e.g. cavitation blockage is occurred. Now the authors are trying to reveal the reason why the difference mentioned above occurs including experimental errors.

CONCLUSION

This study is focused on cavitation and pressure-flow rate characteristics of a spool model to get fundamental knowledges about the spool valve for Aqua Drive System. Observation of cavitation jet issuing from the control gap and measurement of pressure-flow rate character were conducted experimentally. As a result, the following conclusion can be drawn:

1) Inflow angle of cavitation jet is dependent on the axial opening width of spool valve.
2) Cavitation is enhanced with increase the axial opening width and differential pressure between the valve.
3) Cavitation area becomes large with decrease the number of cavitation.
4) Cavitation blockage is not occurred on the pressure–flow rate characteristics of $\Delta P \leq 5.0 \times 10^6$ Pa.

REFERENCES


ACKNOWLEDGEMENT

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.
DEVELOPMENT OF A DIRECT TYPE WATER HYDRAULIC RELIEF VALVE FOR SMALL FLOW RATE

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Abstract. In this report, a direct type water hydraulic relief valve for small flow rate was newly designed. The rated pressure and rated flow rate is 14 MPa and 2 L/min, respectively. The developed valve will be used as a pilot valve of the pilot-operated relief valve. An influence of the shape of the valve parts on the static characteristics was investigated. There is each three kinds of shape of the valve and the trim, where a vortex chamber are formed. The valve shape is similar to conventional poppet valve. However, a sliding rod is attached to the valve, where the spring force acts on. The experimental results shows that valve parts alignment exerts a great influence on hysteresis of the static characteristics of the valve.

Keywords: Water hydraulics, Relief valve, Cavitation, Hysteresis, Override pressure

INTRODUCTION

Most water hydraulic valves have the same basic structure as those used in oil hydraulics. Such water hydraulic valves are produced using different materials and with slightly different dimensions to oil hydraulic valves [1]. However, cavitation associated with the high vapor pressure of water is difficult to resolve with such mild improvement of valve design.

Reducing pressure gradually by making complex paths in a valve throttle is effective for suppressing cavitation [2]. Another way to prevent cavitation is multi-stage pressure reduction. Some relief valves developed for water hydraulics adopts two serial throttles for the main valve [3]. The aim of this study is development of water hydraulic relief valve that rated pressure of 14 MPa and rated flow rate of 40 L/min. To reduce pressure override of the relief valve, pilot-operated relief valve type is suitable.

In this report, a direct type water hydraulic relief valve for small flow rate was newly designed as the pilot valve. An influence of the shape of the valve parts on the static characteristics was investigated.

VALVE DESIGN

Figure 1 shows the cross section of the developed valve. The typical dimensions are shown in Table 1. The valve shape is similar to conventional poppet valve. However, a sliding rod is attached to the valve, where the spring force acts on.

There is each 3 kinds of shape of the valve and the trim, where a vortex chamber are formed. Figure 2 illustrates combinations of the valves and trims.

FIGURE 1. Cross section of the developed valve
### TABLE 1. Valve specifications and dimensions

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<tr>
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<tr>
<td>Rated flowrate</td>
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<td>Half cone angle of poppet valve</td>
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<tr>
<td>Spring stiffness</td>
<td>19.6 N/mm</td>
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</table>

### FIGURE 2. Configurations of valves and trims

### EXPERIMENTS

#### Experimental setup

Figure 3 illustrates the experimental circuit. An axial piston pump was used as the water pressure source. The rated pressure is 14 MPa and the rated flow rate is 27 L/min. The flow to the test relief valve was adjusted by controlling a bypass flow with a manually operated needle valve connected parallel to the test relief valve. The relief flow rate was measured by switching two kinds of turbine-type flow meters with different ranges.

![Experimental circuit diagram](image)

**FIGURE 3. Experimental circuit**

#### Static characteristics

The static characteristics was measured with 9 combinations for 3 kinds of each of the valve and the trim. Figure 4 shows an influence of the valve shape for the same trim, that override pressure slightly changed. On the other hand, an influence of the trim shape is smaller as shown in Fig. 5. When the flow rate is larger, the influence of the shape of the trim exerts on occurrence of cavitation would become larger.
**Discussion**

Figure 6 shows the influence of an alignment of the valve parts. Even though the same valve parts were installed, assembling way of the parts greatly influences hysteresis of the static characteristics.
CONCLUSION

A direct type water hydraulic relief valve for small flow rate was newly designed. An influence of the shape of the valve parts on the static characteristics was investigated. The valve parts alignment exerts a great influence on hysteresis of the static characteristics of the valve.

ACKNOWLEDGMENTS

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.

REFERENCES

CHARACTERISTICS OF WATER HYDRAULIC CYLINDER

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Abstract. A durability test of a double-ended rod type of water hydraulic cylinder is conducted until 300 km operating distance. The water hydraulic cylinder is operated at a velocity of 0.25 m/s under a supply pressure of about 10.5 MPa by an electro-oil hydraulic servo cylinder system. The friction characteristic, low speed performance and leakage characteristic of the water hydraulic cylinder are measured at different operating distances. Friction force decreases at first with increasing operating distance and becomes an almost constant value after about 100 km operation. Friction force vs. velocity curves show small or null velocity dependence of friction force and no negative resistance characteristic at low velocities. The water hydraulic cylinder can move very smoothly at velocities lower than 0.1 mm/s without stick-slip. Leakage flow rate tends to increase with increasing operating distance.

Keywords: Aqua drive system, Water hydraulic cylinder, Friction, Leakage, Low speed performance

INTRODUCTION

A water hydraulic system, which is also called aqua drive system (ADS), has many advantages such as great power density, environment- friendliness, noninflammability, and low pipe friction. Promising applications of ADS include food processing machines [1, 2] and human assist devices utilizing low pressure water like water from waterworks [3]. However, ADSs have not been widely used yet. This is partly because international standards of water hydraulic components have not been established yet. In order to standardize water hydraulic components, their characteristics need to be experimentally investigated and appropriate methods to measure the characteristics have to be proposed. Water hydraulic cylinders and seals suited to them were investigated [3, 4]. However, to the best of the authors’ knowledge, published data of friction characteristic, low speed performance, leakage characteristic, and durability of a water hydraulic cylinder are very limited. In this paper, those characteristics are experimentally investigated.

EXPERIMENTAL APPARATUS AND METHOD

Experimental Apparatus

Figure 1 shows the experimental apparatus used. A test water hydraulic cylinder was driven by an oil hydraulic servo cylinder set in alignment with the water hydraulic cylinder on a surface plate. The piston diameter, rod diameter, stroke, and rated pressure of the test cylinder are 50 mm, 28 mm, 250 mm, and 14 MPa, respectively. The piston and rod seals are made of polyethylene resin. Each polyethylene resin seal has a rectangular cross-section and is used in combination with an O ring. The throttle valve in the water hydraulic circuit was used only when the low speed performance of the test cylinder was measured. Tap water filtered by a nominal 10 μm filter was used as the working fluid.

Experimental Method

Durability of the test cylinder was examined by moving the piston right and left at a velocity of about 0.25 m/s. The relief pressure of the water hydraulic circuit was set at 14 MPa, but the pressure in a cylinder chamber connected to the pump was reduced to about 10.5 MPa during the operation due mainly to the override characteristic of the relief valve. The temperature of water was kept at 301 K (28 °C) at the test cylinder’s ports. The durability test was conducted until the total operating distance reached 300 km. Friction characteristic, low speed performance, and leakage characteristic of the test cylinder were measured under different pressures at the total operating distance of 0, 100, 200, and 300 km or more frequently for leakage characteristic. The friction force, \( F_r \), was calculated by the following equation:
\[ F_r = F_L + (p_1 - p_2)A \quad (1) \]

where \( F_L \) is the force measured by the load cell, \( p_1 \) and \( p_2 \) are the pressures in the test cylinder’s chambers, and \( A \) is the pressure receiving area of the piston.

The low speed performance was evaluated as the lowest operating speed under different operating pressures in one cylinder chamber, which were adjusted by changing the supply pressure to the servo valve. The speed of the cylinder was controlled by adjusting the opening of the meter-in throttle valve.

The leakage characteristic was also measured during the durability test by absorbing the water leaked through the rod packings into a piece of paper during the time period of 100 m operation and by measuring the weight of the paper.

![Schematic of experimental apparatus](image)

**FIGURE 1.** Schematic of experimental apparatus

**RESULTS AND DISCUSSION**

Figure 2 shows the variation of friction force with operating distance during the durability test. It is seen that the magnitude of the friction force is varied at first and tends to be saturated after about 50 km for right stroke and after about 100 km for left stroke.

![Friction force variation with operating distance at \( p_1 \) or \( p_2 = 10.5 \) MPa](image)

**FIGURE 2.** Friction force variation with operating distance at \( p_1 \) or \( p_2 = 10.5 \) MPa

Figure 3 shows the friction characteristics measured at the operating pressures of 3.5 and 10.5 MPa. The friction force tended to increase with increasing velocity at low velocities at 0 km operating distance but did not depend on
velocity at and after 100 km. This means that water film is hardly formed between the seals and cylinder inner wall or piston rod. In addition, friction force decreased with decreasing velocity even near zero velocity and no negative resistance characteristic was observed. The values such as 95 % in Fig. 3 show the mechanical efficiency at the velocity of about 0.2 or -0.2 m/s at 0 and 300 km operating distance. The efficiency of the test water hydraulic cylinder is high.

![Graph showing friction characteristics and their variation with operating distance and pressure](image)

**FIGURE 3.** Steady-state friction characteristics and their variation with operating distance and pressure

Figure 4 shows the low speed performance at $p_1=10.5$ MPa and $p_2=0$ MPa. The test cylinder could operate smoothly at very low speeds lower than 0.1 mm/s and the lowest operating speed was hardly affected by the operating distance. It is considered that such good low speed performance results from no negative resistance characteristic in the steady-state friction characteristic shown in Fig. 3. Good low speed performance was obtained under the other operating pressures.

![Graph showing low speed performance at $p_1=10.5$ MPa](image)

**FIGURE 4.** Low speed performance at $p_1=10.5$ MPa (right stroke)

The leakage of water from $p_2$ side (see Fig. 1) was nearly zero throughout the 300 km operation but the leakage from $p_1$ side increased with increasing operating distance, as shown in Fig. 5(a). The maximum quantity of leakage was 2.2 g per 100 m operation (about 400 s) and the leakage quantity was varied with operating distance. When the leakage quantity was large, the leaked water adhered to the rod surface as many droplets as shown in Fig. 5(b). The droplets were scraped up during the retracting motion of the rod by a dust seal mounted on $p_1$ side and became a few bigger water drops, which sometimes dripped from the rod.

The 10th JFPS International Symposium on Fluid Power 2017

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After the 300 km operating distance, the variations of the dimensions and hardness of all the seals were measured. It was found that the mass reduction of the rod seal was 20.2% for \( p_1 \) side and 10.9% for \( p_2 \) side, respectively and that the interference of the rod seal on \( p_1 \) side became negative. This is regarded as the principal cause of the increased leakage from \( p_1 \) side. The variation of the leakage with operating distance may be caused by the variation of the number and/or size of the wear debris entering in the gap between the rod seal and rod. The leakage can be an index for evaluating the durability performance of the seals for water hydraulic cylinder.

![Graph showing leakage characteristic with operating distance.](image)

**FIGURE 5.** Leakage characteristic

**CONCLUSION**

The test water hydraulic cylinder could operate 300 km under 10.5 MPa operating pressure. Friction characteristic, leakage characteristic, low speed performance, and their variations with operating distance were investigated. Friction force was small, hardly depended on velocity and became almost constant after about 100 km operation. Negative resistance characteristic was not observed and because of that, the water hydraulic cylinder has a good low speed performance. Leakage tended to increase with increasing operating distance and can be an index of the durability of seals.

**ACKNOWLEDGMENTS**

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.

**REFERENCES**

A PREDICTION METHOD OF WATER HAMMER

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Abstract. In this paper, the purpose is to grasp water hammer occurring in water hydraulic piping as a physical phenomenon and to control that through experiment and analysis. Specifically, as one of the criteria for choosing an appropriate inner diameter of hydraulic piping that flows a necessary flow rate, the water hammer phenomenon is considered. An apparatus was prepared to allow water from the pump to flow through the piping and to the cylinder as working fluid. At the time, water hammer occurs and pressure rise occurs. An apparatus was prepared to allow water from the pump to the cylinder through the piping and as working fluid. When pressure is applied from the cap side of the cylinder, the piston stops at the rod side. At the time, a water hammer occurs and pressure rises. This experiment was conducted by changing the flow rate in the hydraulic piping then the experimental value and the theoretical value of were compared.

Keywords: Aqua drive system, Water hydraulics, Water hammer, Surge pressure

INTRODUCTION

Working fluid of water hydraulic system is tap water without any additives. By comparison with oil hydraulics, velocity of the water in pipe is tend to be high due to lower viscosity of water. If the water flow is stopped instantaneously by sudden stoppage of cylinder or valve operation, the kinetic energy of the water is transformed to the elastic energy of the water resulting in surge pressure. It is known as water hammer phenomena and there is a danger that equipment will be damaged by large pressure occurring in a moment. It is needed to predict the surge pressure and prevent the water hammer. In this study, a prediction method of surge pressure occurring when cylinder is moved from cap side to rod side is studied by experiments. In theory, it is known that the surge pressure is proportional to the flow velocity in the pipe. The surge pressure is measured by a pressure sensor attached to the rod side of the cyliner and indirectly measured the flow velocity in the pipe by tracking the displacement of the cylinder with a laser sensor. By this experiment, the theoretical value and the experimental value were compared. The final goal of this study is to propose a draft to ISO working group on prediction and prevention of water hammer for water hydraulic systems.

NOMENCLATURES

c wave speed
$c_1$ parameter of anchoring pipe
$D$ diameter of pipe
$E$ Young’s modulus of pipe wall material
$e$ wall thickness of pipe
$K$ bulk modulus of water
$\Delta p$ surge pressure
$\Delta V$ velocity change
$\mu$ Poisson’s ratio
$\rho$ density of water

TEST CIRCUIT

A test circuit of water hammer is shown in FIGURE 1, FIGURE 2 and FIGURE 3. Working fluid is tap water. Stainless steel pipe [1] is used. A water hydraulic pump ① generates high pressure. Operating a water hydraulic servo valve ⑥, the high-pressure water flows into a cylinder ⑦ (stroke: 200mm, piston diameter: 20mm) through a pipe of 8m in length and 6mm in inner diameter. A piston ⑨ is extracted and suddenly stopped at its...
stroke end. At the instance, water flow stops instantaneously and surge pressure is generated. The surge pressure is measured by a pressure sensor ⑩ (shown in FIGURE 4). TABLE 1 shows summary of all measuring instruments. The experiment was conducted by some velocity in the pipe altering pump rotating speed and servo valve input voltage. As a preparation of experiment, opening two stop valves ⑧ and letting water flow bypass the cylinder piston, if air bubbles exist inside the pipe, they will go out from the pipe. The flushing is needed for this experiment.

FIGURE 1. A photo of test pipeline

FIGURE 2. A photo of test pipeline with cylinder
FIGURE 3. A test circuit of water hammer

TABLE 1. Measuring instruments

<table>
<thead>
<tr>
<th>Instrument</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure gauge</td>
</tr>
<tr>
<td>temperature sensor in the tank</td>
</tr>
<tr>
<td>level indicator</td>
</tr>
<tr>
<td>pressure sensor</td>
</tr>
<tr>
<td>laser displacement gauge</td>
</tr>
<tr>
<td>Servo valve driving oscillator</td>
</tr>
<tr>
<td>Dynamic distortion amplifier</td>
</tr>
<tr>
<td>Servo amplifier</td>
</tr>
<tr>
<td>Data collecting system</td>
</tr>
</tbody>
</table>

FIGURE 4. A photo of pressure sensor and laser sensor
SURGE PRESSURE

At the instance when water flow of velocity $\Delta V$ stops instantaneously, water pressure rises by the amount of $\Delta p$:

$$\Delta p = \rho c \Delta V.$$  \hspace{1cm} (1)

where $\rho$ is the density of water and $c$ is the wave speed. The wave speed of water is written as:

$$c = \frac{\sqrt{K}}{\rho}$$ \hspace{1cm} (2)

where $K$ is the bulk modulus of water (2.2GPa). When considering elasticity of pipe wall, the equation of wave speed can be written as:

$$c = \frac{\sqrt{K/\rho}}{\sqrt{1+[(K/E)(D/e)c_1]}}$$ \hspace{1cm} (3)

The parameter $c_1$ represents a condition of anchoring pipe:

(a) at its upstream only,

$$c_1 = 1 - \frac{\mu}{2}$$ \hspace{1cm} (4)

(b) throughout against axial movement,

$$c_1 = 1 - \mu^2,$$ or \hspace{1cm} (5)

(c) with expansion joints throughout.

$$c_1 = 1$$ \hspace{1cm} (6)

The parameters used to calculate the theoretical value are as in TABLE 2.

<table>
<thead>
<tr>
<th>$\rho$</th>
<th>Density of water (20°C)</th>
<th>998</th>
<th>[kg/m$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K$</td>
<td>Bulk modulus of water</td>
<td>2.23</td>
<td>[GPa]</td>
</tr>
<tr>
<td>$E$</td>
<td>Elastic coefficient of SUS304TP</td>
<td>197</td>
<td>[GPa]</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter of pipe</td>
<td>6</td>
<td>[mm]</td>
</tr>
<tr>
<td>$e$</td>
<td>Thickness of the pipe</td>
<td>2</td>
<td>[mm]</td>
</tr>
<tr>
<td>$c_1$</td>
<td>State parameter of fixed pipe</td>
<td>1</td>
<td>[-]</td>
</tr>
</tbody>
</table>

MEASUREMENT RESULTS

An example of experimental results is shown in FIGURE 5. Supply voltage $u$ of servo valve changed at 4.22 s and the piston displacement $x$ decreased (the piston extracted). When the piston stopped at its stroke end, the pressure measured by the sensor increased suddenly. The pressure rise is surge pressure $\Delta p$. Mean velocity $V_{pipe}$ is measured from the slope of the piston displacement just before the piston stopped (FIGURE 6). Therefore, the velocity change $\Delta V$ is equal to the mean velocity $V_{pipe}$. As shown in FIGURE 7, measurement results of surge pressure (triangle points) are precisely predicted by Eq. (1) (solid and dotted lines). An example measured value is shown in TABLE 3 and TABLE 4. Incidentally, the difference between theoretical value and experimental value is 0.1% at least and 6.2% at most.
FIGURE 5. An example of transient phenomena of water hammer

FIGURE 6. The method of calculating $V_{\text{pipe}}$

FIGURE 7. Surge pressure vs. mean velocity
TABLE 3. an example of experimental condition and surge pressure(MPa)

<table>
<thead>
<tr>
<th>Rotation speed of pump [rpm]</th>
<th>Driving voltage of servo valve [V]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4</td>
</tr>
<tr>
<td>400</td>
<td>4.7</td>
</tr>
<tr>
<td>500</td>
<td>5.4</td>
</tr>
<tr>
<td>600</td>
<td>6.5</td>
</tr>
<tr>
<td>700</td>
<td>7.6</td>
</tr>
<tr>
<td>800</td>
<td>8.4</td>
</tr>
<tr>
<td>900</td>
<td>8.4</td>
</tr>
<tr>
<td>1000</td>
<td>8.0</td>
</tr>
</tbody>
</table>

TABLE 4. an example of experimental condition and $V_{pipe}$(m/s)

<table>
<thead>
<tr>
<th>Rotation speed of pump [rpm]</th>
<th>Driving voltage of servo valve [V]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4</td>
</tr>
<tr>
<td>400</td>
<td>3.1</td>
</tr>
<tr>
<td>500</td>
<td>3.9</td>
</tr>
<tr>
<td>600</td>
<td>4.2</td>
</tr>
<tr>
<td>700</td>
<td>5.1</td>
</tr>
<tr>
<td>800</td>
<td>5.6</td>
</tr>
<tr>
<td>900</td>
<td>5.6</td>
</tr>
<tr>
<td>1000</td>
<td>5.3</td>
</tr>
</tbody>
</table>

CONCLUSIONS

The deviation between the theoretical value and the experimental value is at most 6.2%. Thus, prediction of water hammer is investigated by the experiments. Future work is to study the method of prevention of water hammer. Currently, we plan to reduce surge pressure by installing an accumulator.

ACKNOWLEDGMENTS

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.

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TEST METHODS OF WATER HYDRAULIC PUMPS

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Abstract. Although water hydraulic pumps exist, they are used at only a few manufacturers. Because there is no criterion of performance that can be applied to all pumps when water hydraulic pumps are introduced to the manufacturers. Therefore, in this research, test methods of water hydraulic pumps for the qualitative evaluation of pumps are proposed. The purpose of this research is to propose a draft to ISO working group on the test method of water hydraulic pumps. As a research method, prepare a tester, do experiments, summarize the experiment results and document the test method for formulating the global standard through experiments and considerations. As a result, the draft to water hydraulic pump performance test method was completed.

Keywords: Aqua drive system, Water hydraulics, Test method, Pump.

INTRODUCTION

Modern water hydraulic system is characterized by newly developed displacement pumps that generate high pressure to drive motors and/or cylinders through valves using tap water as working fluid without any additives. The modern water hydraulic system is named as “Aqua Drive System”. As an example, the aqua drive system has been applied to food processing machines. Only a few manufacturers have put water hydraulic pumps in the market. The reason why the market is not big is that there is no criterion for comparing performance of pumps with the reliability when water hydraulic pumps are introduced to manufacturers. For the qualitative evaluation of pumps, it is needed to develop an international standard on the test method of pump. The purpose of this study is to propose a draft to ISO working group on the test method of water hydraulic pump.

NOMENCLATURES

\[ D \] displacement of pump
\[ p_s \] suction pressure
\[ p_o \] output pressure
\[ Q \] flow rate
\[ T \] shaft torque
\[ \Delta Q \] leakage
\[ \Delta T \] torque loss
\[ \eta \] overall efficiency
\[ \eta_m \] mechanical efficiency
\[ \eta_v \] volumetric efficiency
\[ \omega \] rotating speed

MESUREMENT METHOD

Test Circuit

A test circuit proposed in this study is shown in Figure 1. And FIGURE 2. A water hydraulic pump to be tested \( \Box \) is installed on the test stand. The pump is driven by an electric motor \( \Box \) whose rotating speed is controlled to
be constant against output pressure of the pump. Water temperature in the tank is regulated to be 20℃ with a heat exchanger ⑪. Shaft torque and rotating speed are measured to calculate input power supplied to the pump. Output pressure of pump is regulated by a pressure relief valve ⑨. Output flow rate is measured by a flowmeter ⑩. Suction and output pressure of pump are measured. They are used to calculate output power of pump. Before measurement, it is needed to properly perform running-in of a test pump.

FIGURE 1. A test circuit of water hydraulic pump

FIGURE 2. A photo of a test stand
Measuring Instrument

- Rotating speed meter
- Torque meter
- Pressure gauge for suction
- Pressure gauge for output
- Flowmeter
- In-tank thermometer
- Tank water level gauge

Test Procedure

A preparation procedure before measurement is:
(1) Turn on the test equipment.
(2) Turn on chiller and operate at the set temperature.
(3) Increase the rotation speed to 500rpm at no-load-pressure condition and check for water leakage from flow path. For a while, run at 500rpm and step up the rotation speed to 1800rpm. At this time, increase the rotation speed while confirming the flow rate, torque, pressure with each measuring instrument.
(4) With the rotation speed kept at 1800rpm, adjust the output pressure setting with relief valve and stepwise raise the discharge pressure to 16MPa.
(5) Run at a rotational speed of 1800rpm and output pressure of 16MPa for a while.
(6) When the water temperature settles at the set temperature of the chiller and the torque decreases and it does not change, it is in a condition that test can be performed.

A typical test procedure is done with following the operation and using the test circuit as shown in FIGURE 1.
(1) set rotational speed of test pump,
(2) set output pressure of test pump by adjusting pressure relief valve ⑨, step by step,
(3) measure shaft torque and rotational speed by sensors ② and ③,
(4) measure suction pressure, output pressure, and output flow rate by sensors ⑥, ⑤, and ⑩,
(5) repeat the same procedure from (2) to (4) for other pressure values with rotation speed determined by (1) constant.
(6) if an abnormal value can be seen from the experimental results, re-measurement about only that condition is performed according to the procedure of (1) to (4).

NOTE: By closing a stop valve ⑭ and opening a stop valve ⑬, a booster pump ⑦ is used to pressurize a suction port of pump.

MEASUREMENT RESULTS

Overall efficiency $\eta$ is defined as Eq. (1):

$$\eta = \frac{(p_o - p_i)Q}{\omega T}.$$  \hspace{1cm} (1)

Efficiency $\eta_V$ is defined as Eq. (2):

$$\eta_V = \frac{Q}{\omega D}.$$  \hspace{1cm} (2)

Mechanical efficiency $\eta_M$ is defined as Eq. (3):

$$\eta_M = \frac{D(p_o - p_i)}{T}.$$  \hspace{1cm} (3)

Leakage $\Delta Q$ is defined as Eq. (4):

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\[ \Delta Q = \omega D - Q. \]  

(4)

Torque loss \( \Delta T \) is defined as Eq. (5):

\[ \Delta T = T - \frac{D(p_o - p_i)}{2\pi}. \]  

(5)

Displacement of pump that is used to calculate various performance is the volume of fluid per revolution of the pump which is determined by the following method. An example of calculation method for displacement of pump is shown in FIGURE 3.

1. Create a scatter plot with constant rotation speed, with outlet pressure on the horizontal axis and \( Q/N \) on vertical axis.
2. Linearly approximate a point group plotted by the method of least squares in a range where there is little variation with no load or low pressure state (e.g. 0 MPa to 5 MPa), and determine the intercept of the vertical axis and the linear approximation line as displacement of pump \( D \).
3. Calculate displacement of pump \( D \) for each rotation speed according to (1),(2).

An example of efficiency plot of one of commercial water hydraulic pumps is shown in FIGURE 4, FIGURE 5, FIGURE 6, FIGURE 7, FIGURE 8 and FIGURE 9. The efficiencies are plotted as a function of output pressure for four values of the rotating speed of pump. The same test procedure are adopted to other commercial pumps.

\[ \begin{align*}
\text{FIGURE 3.} & \quad \text{example of calculation method for displacement of pump} \\
\text{FIGURE 4.} & \quad \text{overall efficiency}
\end{align*} \]
FIGURE 5. Volumetric efficiency

FIGURE 6. Mechanical efficiency

FIGURE 7. Leakage
CONCLUSION

A test circuit and a test procedure are proposed for the test method of water hydraulic pump. Example of pump efficiencies are measured. A future work is to test other water hydraulic pumps and improve the test method to propose to ISO working group.

AKNOWLEDGEMENTS

This research was carried out as part of the ADS International Standardization Promotion Project (Japan Fluid Power Association) supported by Ministry of Economy, Trade and Industry.

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