Recent Development of Fluid Mechanics in Fluid Power Engineering*

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The recent development of fluid mechanics in the field of fluid power engineering is surveyed. Topics covered include unsteady flow in pipelines, unsteady flow measurement, cavitation in unsteady flow, flows in fluid power components, and flow-induced vibrations.

Key Words : Fluid Mechanics, Fluid Power Systems, Unsteady Flow, Pipe Flow, Flow Measurement, Cavitation, Numerical Analysis, Flow-Induced Vibration

1. Introduction

Fluid power engineering has undergone great development in the past three decades because of the high force density, high response and excellent controllability of the fluid power system.

The components used in a fluid power system fall within the category of fluid machinery, while the main function of the system is to control the power transmitted from the power source to the load. Consequently, fluid power engineering is based on both fluid mechanics and control engineering.

This paper describes the fluid mechanics developed in the field of fluid power engineering, especially in Japan. Fluid power technology is based on Pascal's principle concerning hydrostatics. The flow in hydraulic components, however, has high local velocities which cannot be treated as hydrostatics but

should be treated as hydrodynamics.

The fluid mechanics applied in fluid power engineering has the following special features :

1) The flow is laminar except near a restriction because of the high viscosity of the hydraulic fluid.

2) Fluid mechanics in unsteady flow is important because the behavior of the fluid power system is essentially dynamical.

3) Fluid mechanics plays an important role in the determination of flow forces because they affect the behavior of control valves which are the main control elements of the system.

2. Unsteady Flow in Pipelines

As stated in Section 1., most flows in the pipelines of a fluid power system are laminar and unsteady. Consequently, many papers on such flows have been published. In this section, some topics involving the laminar unsteady flow in pipelines are surveyed.

2.1 Numerical methods in laminar unsteady flow

Numerical methods in the unsteady flow of a circular pipe are classified into two types : Laplace

* Received 20th February, 1989

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transform methods and characteristics methods.

The characteristics methods^{(1)~(4)} may be more useful when the piping system to be analyzed includes several hydraulic components having nonlinear characteristics.

Zielke⁽¹⁾ succeeded in expressing the friction term in the Navier-Stokes equations of motion using a weighting function. Trikha⁽²⁾ improved Zielke's method to decrease the computing time, though the accuracy became lower than in the latter.

Kitagawa et al.⁽⁴⁾ improved the weighting function to speed up computation. The governing equations for the unsteady flow of a liquid in a straight pipe are the following: Equation of motion

$$\frac{\partial V}{\partial t} + V \frac{\partial V}{\partial x} + g \frac{\partial H}{\partial x} + gh_f = 0 \quad (1)$$

Continuity equation

$$\frac{\partial H}{\partial t} + V \frac{\partial H}{\partial x} + \frac{a^2}{g} \frac{\partial V}{\partial x} = 0, \quad (2)$$

where H is the pressure head, V is the mean velocity at a flow cross section, t is the time, x is the distance, g is the acceleration of gravity, a is the sound velocity and h_f is the head loss per unit length.

The wall friction term h_f in Eq. (1) is expressed as follows:

$$h_f(t) = \frac{8\nu}{gR^2} V(t) + \frac{4\nu}{gR^2} \sum_{i=1}^N y_i(t), \quad (3)$$

where

$$y_i(t) = \int_0^t m_i e^{-n_i(\nu/R^2)(t-u)} \frac{\partial V}{\partial t}(u) du \{ V(t+\Delta t) - V(t) \},$$

Where R is the inner radius of the pipe, ν is the kinematic viscosity and $\Delta\tau = \nu\Delta t/R^2$. At time $t+\Delta t$, h_f becomes

$$h_f(t+\Delta t) = \frac{8\nu}{gR^2} V(t+\Delta t) + \frac{4\nu}{gR^2} \sum_{i=1}^N y_i(t+\Delta t), \quad (4)$$

where

$$y_i(t+\Delta t) \approx e^{-n_i\Delta\tau} y_i(t) + m_i e^{-n_i(\Delta\tau/2)}. \quad (5)$$

The values of m_i and n_i up to $i=10$ are shown in Table 1.

Table 1 Constants m_i and n_i in Eq. (5)

i	n_i	m_i
1	26.374 4	1.0
2	72.803 3	1.167 25
3	187.424	2.200 64
4	536.626	3.928 61
5	1 570.60	6.787 88
6	4 618.13	11.676 1
7	13 601.1	20.061 2
8	40 082.5	34.454 1
9	118 153.	59.164 2
10	348 316.	101.590

The variations of pressure and velocity can be calculated by applying the characteristics method taking into account the functions $h_f(t)$ in Eq. (3). The computing time has been reduced to the same order as in Trikha's method⁽²⁾, the accuracy being kept nearly the same level as in Zielke's method⁽¹⁾.

2.2 Unsteady flow in flexible hoses

Flexible hoses used in fluid power systems have viscoelastic properties. The motion of their walls in the radial direction as the pressure inside them fluctuates is considerably larger than that of normal pipes made of metals and cannot be neglected. Cox⁽⁵⁾ investigated the effect of the viscoelastic properties of the blood vessel on the blood flow, but he did not confirm it by experiments.

Nakano and Yoshimoto⁽⁶⁾ assumed a modified Voigt model as shown in Fig. 1 for high pressure flexible hoses used in hydraulic systems, where d is the damping factor of the dash pot, and k_1 , k_2 and k_h are spring moduli. The ratio r of the phase velocity of the viscoelastic pipes to that of the elastic pipe is shown in Fig. 2, where $T_1 = d/k_2$ and

$$T_2 = \frac{d}{k_2 + \frac{k_1 k_h}{k_1 + k_h}}. \quad (6)$$

Parameters T_1 and T_2 represents the property of viscoelasticity. By carrying out experiments on frequency response, T_1 and T_2 can be estimated from Fig. 2.

Transient phenomena in viscoelastic pipes were also studied by Oguchi and Nakano⁽⁷⁾ Gally et al.⁽⁸⁾ and Hashimoto et al.⁽⁹⁾.

Fig. 1 Modified Voigt model for the combination of the flexible-hose wall and the slightly compressible fluid

Fig. 2 Ratio of the phase velocity of the viscoelastic pipe to that of the elastic pipe

2.3 Transient flow in the inlet length of a straight pipe

In fluid power systems, short circular pipes known as "chokes" are frequently used. In such a case it becomes very important to know the dynamic behavior of the laminar flow in the inlet length of that pipe.

Atabek et al. analyzed the oscillatory flow⁽¹⁰⁾ and transient flow⁽¹¹⁾ in the inlet length of a pipe, assuming that the velocity distribution of the inlet section of the pipe was uniform and given by a periodic function of time t and approximated the convective term as follows:

$$u \frac{\partial u}{\partial x} \approx w(t) \frac{\partial u}{\partial x} \quad (7)$$

where w is the mean value of the velocity at the inlet section. Nakano and Miyata⁽¹²⁾ analyzed the flow using Eq.(7), varying the uniform velocity at the inlet section of the tube from a finite value to another finite value, while measurements were carried out using a laser-doppler anemometer. The theoretical results were in good agreement with the measured ones. Nakano and Sakatani⁽¹³⁾ further investigated the consistency of Eq.(7) by solving the Navie-Stokes equations numerically using the finite difference method. The results were in good agreement with the calculated ones⁽¹²⁾.

Figure 3⁽¹³⁾ shows the variation of normalized velocity u^* with normalized time $\bar{t}$ where $u^* = \{u(\bar{x}, \bar{r}, \bar{t}) - u(\bar{x}, \bar{r}, 0)\} / (w_2 - w_1)$, u is the velocity in the axial direction, $\bar{r} = r/r_0$, r is the radial coordinate, r_0 is the inner radius of the tube, $x = 2x/(Re r_0)$, x is the axial coordinate, Re is the Reynolds number $= 2r_0 w_2 / \nu$, $\bar{t} = 2w_2 t / (r_0 Re)$, t is the time, w_2 is the final value of w , and w_1 is the initial value of w . Nakano

Fig. 3 Numerical solution of normalized velocity u^* at $Re=100$ and $w_1=0$.
 — : $x=0.105$, --- : $x=0.0346$, - - - : $x=0.0695$, ··· : $x=0.0182$, ● : Approximate solution

and Yokota also analyzed the oscillatory flow^{(14)~(16)} and showed that the time-average velocity distribution at a flow cross section agreed well with the velocity distribution of the steady flow at the same section, and that the fluctuating component became flat over any cross section as Strouhal number increased.

2.4 Unsteady flow in a convergent tube

In analysis of the laminar flow in a tapered tube with a small taper angle, the convective terms of the Navier-Stokes equations are usually neglected^{(17)~(19)}. However, this assumption is not always in good agreement with the experimental results.

Takahashi et al.⁽²⁰⁾ expanded normalized velocity u_p^* and normalized differential pressure Δp^* between upstream and downstream into Fourier series with respect to time t as follows:

$$u_p^* = \sum_{k=0}^{\infty} a_{2k} r^{*2k} + \sum_{k=0}^{\infty} b_{2k} r^{*2k} \cos \omega t + \sum_{k=0}^{\infty} c_{2k} r^{*2k} \sin \omega t \quad (8)$$

$$\Delta p^* = \Delta p_m^* + \Delta p_{1c}^* \cos \omega t + \Delta p_{1s}^* \sin \omega t \quad (9)$$

Substituting these equations into Navier-Stokes equations, they obtained the coefficients which are functions of radial coordinate r^* , assuming that the pressure is nearly constant over the flow cross section. They showed that the velocity distribution in the radial direction is flatter than that of a straight tube, especially near the throat of a convergent tube.

Ikeo and Uzawa⁽²¹⁾ measured the velocity distribution in the radial direction of a convergent tube using a laser-doppler velocimeter. The results agreed very well with the calculated ones⁽²⁰⁾.

3. Unsteady Flow Measurement

In order to improve the dynamic characteristics of hydraulic components, the accurate measurement of unsteady flow rate has long been requested in fluid power engineering. Flow meters with movable parts are not preferable because of their inertia. The most promising methods would be use of a laser doppler velocimeter (LDV), differential-pressure-type flow meter, or electromagnetic flow meter. The electromagnetic flow meter cannot be used for oil flows because of its insulation properties, so it is not taken up here.

3.1 LDV-method

a) Beam-scanning⁽²²⁾ The velocity distribution of a flow cross section is measured by scanning the beam quickly over that section. Integration of this distribution gives the instantaneous flow rate through that cross section.

b) Multi-beam⁽²³⁾ The velocity distribution of a flow cross section is measured by measuring the

velocities at several points at the same instant using several beams. These beams can be generated from a single laser generator.

c) Estimation by an "observer"⁽²⁴⁾ The velocity at the center axis of the pipe is measured. The flow rate is estimated from that velocity using an "observer" which has been set up in advance.

d) Estimation by convolution integral The velocity at the center axis of the pipe is measured. The convolution integral is carried out using this velocity multiplied by a weighting function which describes the relation between the center velocity and the flow rate. There are two types of device available: a digital type^{(25)~(27)} and an analog type⁽²⁸⁾.

It is necessary that the velocity distribution be known in advance for methods c) and d). Therefore, these two methods can currently be applied only to laminar flows. The hot film anemometer⁽²⁹⁾ is also available instead of LDV.

3.2 Differential pressure method

a) Straight tube The instantaneous pressure difference between two points of a tube is measured⁽³⁰⁾. The instantaneous flow rate is calculated from this differential pressure using a transfer function which relates the differential pressure with the flow rate. The transfer function is approximated as

$$\tilde{G}_2(0, s^*) = A_2 + \sum_{i=1}^2 \left(\frac{m_{2i}}{s^* + n_{2i}} \right) + \frac{K_{c2}s^* + K_{a2}}{s^{*2} + 2\zeta_2\omega_{n2}s^* + \omega_{n2}^2}, \quad (10)$$

where $A_2=0.99$ and $\tilde{G}_2(0, s^*)$ is the transfer function of the mean velocity to the velocity at the center of the pipe. The constants are shown in Table 2. An analog electronic circuit⁽³⁰⁾ corresponding to Eq. (1) has been prepared for use. The flow rate can be calculated in real time using this circuit.

b) Restriction The pressure difference between the points upstream and downstream of a restriction is measured. The instantaneous flow rate is estimated from this differential pressure. Several methods have been reported using different restrictions^{(20), (31)~(33)}.

The method⁽³⁴⁾ using a cylindrical choke is one of the most convenient and simplest ways to measure the unsteady flow. Figure 4 shows the optimal diameter d of the choke corresponding to the mean flow rate q_c

Table 2 Constants for transferfunction $\tilde{G}_2(0, s^*)$

n_{21}	n_{22}	m_{21}	m_{22}
148.724 340	2 439.003 3	-27.079 510	-113.846 57
ζ_2	ω_{n2}	K_{c2}	K_{a2}
0.864 440 4	52.044 138	-0.010 915	-721.404 46

with the pressure drop p_d at the restriction for steady flow as a parameter, where l is the length of the choke.

4. Cavitation in Unsteady Flow

Since hydraulic fluids used in hydraulic systems have much smaller vapor pressure and more air bubbles than water, the cavitation phenomena occurring in hydraulic fluids are supposed to be different from those in water. Furthermore, hydraulic components have many restrictions, and flows through them are unsteady. Therefore, it is difficult to apply directly the results of watercavitation to the cavitation of hydraulic fluids.

4.1 Gaseous and vaporous cavitation

Backé and Riedel⁽³⁴⁾ investigated the cavitation downstream of a cylindrical choke with a 45-degree chamfer, varying the pressure downstream sinusoidally, and reported that the degree of hysteresis increases with an increase of pressure variation rate.

Ishihara et al.⁽³⁵⁾ investigated the cavitation occurring downstream of sharp-edged orifices in detail. They created unsteady flows with constant rates of pressure variation and observed the cavitation using the optical method. Two different types of cavitation were observed: gaseous cavitation and vaporous cavitation.

a) Gaseous cavitation This is recognized as a bubble flow which is either laminar or cloudy and appears cylindrical in shape directly downstream of the orifice. This can only be observed by the optical method; i.e., it can be observed neither by visualization nor by sudden change in pressure.

b) Vaporous cavitation This is recognized as a crowd of bubbles generating at the transition region of a turbulent jet and also at a farther distance downstream of the orifice.

The inception and the behavior of cavitation are greatly affected by the initial conditions of the hydraulic fluid and oil temperature. The hysteresis, i.e., the

Fig. 4 Selection of choke diameters

difference between inception and desinence of cavitation, increases with an increase of the degree of flow-unsteadiness, i.e., the rate of pressure variation.

4.2 Oil-column separation

When the pressure in a liquid decreases below absolute zero, the liquid is subject to the tension. Normally, it cannot bear the tension, and column separation occurs. There are some cases, however, where a liquid can bear tension, and the pressure goes below zero.

Lee et al.⁽³⁶⁾ investigated the transient flow in a pipeline and showed that a negative pressure can appear. Figure 5 illustrates the pressure variations after a valve upstream of a long pipeline is suddenly shut. Lee et al. deduced that the minimum pressures must be below absolute zero in the case of Fig. 5 (a); in other words, the liquid bore tension. Oil-column separation was observed at $x=12$ m (x is the

distance from the valve upstream) in the case of Fig. 5 (b).

Lee⁽³⁷⁾ further investigated such phenomena, adding gas bubbles and solid particles. The relations between the minimum pressure $p_{\min}$ and the initial set pressure p_s are shown in Fig. 6, where V_G is the gas volume percentage. It can be seen that cavitation occurs when p_s exceeds a critical value called the inception point of cavitation. He concluded that incipient pressure of cavitation in the transient flow of oil is considerably lower than absolute zero, since gas bubbles are apt to resolve into oil because of the high level of pressure in hydraulic systems, and solid particles are apt to get wet in normal hydraulic fluids.

5. Flows in Fluid Power Components

5.1 Analysis of the flow in control valves

In order to design a control valve, it is very

(a) $p_s = 1.0$ MPa

(b) $p_s = 1.4$ MPa

Fig. 5 Records of the pressure waves

(a) Clean oil ($V_G = 8.8\%$)

(b) Teflon particles in oil ($V_G = 8.5\%$)

(c) Effects of gas

Fig. 6 Relation between the minimum pressure and p_s obtained from the experiments using hydraulic fluids

important to estimate the discharge coefficient of the valve and the forces acting on the valve body. In order to estimate them, it is necessary to know the flow pattern inside the valve.

In general, solid boundaries inside the valve have complicated shapes, and it is difficult to give the boundary condition if common coordinates such as cartesian coordinates are employed. Several kinds of streamline-coordinate methods have been developed^{(38)~(40)} to solve this problem.

A control valve has one or more restrictions, and the flow through a restriction forms a jet with a discontinuous surface. Another problem arises in such a flow because the shape of this discontinuous surface is not initially known.

To overcome this difficulty, a new method using a streamline coordinate system was developed^{(41)~(43)}. In this method, it is assumed that the fluid surrounding the jet does not affect the latter.

Figure 7 shows the streamline coordinates; the s -coordinate curve is taken along a streamline and the n -coordinate curve orthogonal to it. Since the unit of the length is indefinite in system (s, n) , coordinate system (s, n) is transformed into another coordinate system (ϕ, ψ) by the following equations:

$$\frac{\partial \phi}{\partial s} = \lambda \quad \text{and} \quad \frac{\partial \phi}{\partial n} = 0 \quad (11)$$

$$\frac{\partial \psi}{\partial n} = q \quad \text{and} \quad \frac{\partial \psi}{\partial s} = 0 \quad (12)$$

where λ is the measuring ratio with respect to s and ϕ , and q is the normalized velocity and at the same time the measuring ratio with respect to n and ψ .

Since ϕ is constant along an n -coordinate curve ($\overline{AD}$ and $\overline{BC}$ in Fig. 7),

$$\frac{\partial \theta}{\partial s} = \frac{1}{\lambda} \frac{\partial \lambda}{\partial n} \quad (13)$$

and ψ is constant along an s -coordinate curve ($\overline{AB}$ and $\overline{DC}$ in Fig. 7),

Fig. 7 Streamline coordinates

$$\frac{\partial \theta}{\partial n} = -\frac{1}{q} \frac{\partial q}{\partial s}, \quad (14)$$

where θ is the angle shown in Fig. 7.

The Navier-Stokes equations of motion in (ϕ, ψ) coordinates are expressed as

$$\lambda q \frac{\partial q}{\partial \phi} = -\lambda \frac{\partial p}{\partial \phi} + \frac{1}{Re} \left[\lambda \frac{\partial}{\partial \phi} \left(\lambda \frac{\partial q}{\partial \phi} \right) + q \frac{\partial}{\partial \psi} \left(q \frac{\partial q}{\partial \psi} \right) - \frac{q^2}{\lambda} \frac{\partial q}{\partial \psi} \frac{\partial \lambda}{\partial \psi} - \frac{q^3}{\lambda^2} \left(\frac{\partial \lambda}{\partial \psi} \right)^2 - \frac{2\lambda^2}{q} \left(\frac{\partial q}{\partial \phi} \right)^2 \right], \quad (15)$$

$$\frac{q^3}{\lambda} \frac{\partial \lambda}{\partial \psi} = -q \frac{\partial p}{\partial \psi} + \frac{1}{Re} \left[-q^2 \frac{\partial}{\partial \psi} \left(\frac{\lambda}{q} \frac{\partial q}{\partial \phi} \right) + q \lambda \frac{\partial}{\partial \phi} \left(\frac{q}{\lambda} \frac{\partial \lambda}{\partial \psi} \right) - 2\lambda \frac{\partial q}{\partial \psi} \frac{\partial q}{\partial \phi} + 2q \frac{\partial q}{\partial \phi} \frac{\partial \lambda}{\partial \psi} \right], \quad (16)$$

where p is the normalized pressure and Re is the Reynolds number.

The vorticity transport equation is

$$q \lambda \frac{\partial \Omega}{\partial \phi} = \frac{1}{Re} \left[\lambda \frac{\partial}{\partial \phi} \left(\lambda \frac{\partial \Omega}{\partial \phi} \right) + q \frac{\partial}{\partial \psi} \left(q \frac{\partial \Omega}{\partial \psi} \right) - \frac{\lambda^2}{q} \frac{\partial \Omega}{\partial \phi} \frac{\partial q}{\partial \phi} - \frac{q^2}{\lambda} \frac{\partial \Omega}{\partial \psi} \frac{\partial \lambda}{\partial \psi} \right], \quad (17)$$

where Ω is the vorticity defined by the following equation:

$$\Omega = \frac{q^2}{\lambda} \frac{\partial \lambda}{\partial \psi} - q \frac{\partial q}{\partial \phi}. \quad (18)$$

The set of Eqs.(19), (14), (17) and (18) can be solved numerically, e.g., using the finite difference method. Figure 8⁽⁴²⁾ shows an example of the solution for the flow issuing from the restriction of a spool valve.

This method can also be applied to the image processing⁽⁴⁴⁾ of streamlines obtained from flow visualization, from which flow quantities such as velocity, vorticity and pressure distributions are easily computed.

5.2 Unsteady flow between disks with a varying gap

Unsteady flow between parallel disks with a time-varying gap width and a central fluid source is fre-

Fig. 8 Streamlines in the jet from the restriction of the spool valve, $Re=100$, clearance/opening = 0.06

Fig. 9 Coordinate system

quently found in machine elements such as statical bearings used in axial piston pumps^{(45)~(48)}. Recently, a method⁽⁴⁹⁾ using “multifold series expansion⁽⁵⁰⁾” was developed to solve such problems.

Coordinates are taken as shown in Fig. 9. The velocity components u and v and pressure p , which are functions of r , y and t , are expanded with respect to r as follows:

$$u = -\frac{1}{2} \frac{\nu r}{h^2} \phi_\eta(\eta, t) + \frac{q_s}{hr} F_\eta(\eta, t) + \frac{q_s^2}{\nu r^3} G_\eta(\eta, t) + \dots \quad (19)$$

$$v = \frac{\nu}{h} \phi(\eta, t) + 0 + \frac{2q_s^2 h}{\nu r^4} G(\eta, t) + \dots \quad (20)$$

$$\frac{p}{\rho} = K(t) + \frac{\nu^2}{h^2} P_0(\eta, t) + \frac{\nu^2 r^2}{h^4} P_1(\eta, t) + \frac{\nu q_s}{h^3} \times \log\left(\frac{r}{r_s}\right) \cdot P_2(\eta, t) + \frac{q_s^2}{h^2 r^2} P_3(\eta, t) + \dots \quad (21)$$

where $\eta = y/h(t)$, ρ is the density of the fluid, ν is the kinematic viscosity of the fluid, $q_s = q/(2\pi)$, and q is the source flow rate.

Substituting Eqs.(19)-(21) into the Navier-Stokes equations of motion, we have a set of partial differential equations of $\phi(\eta, t)$, $F(\eta, t)$, $G(\eta, t)$, $\dots$, $P_1(\eta, t)$, $P_2(\eta, t)$, $\dots$. Then, the variable t is transformed into a set of new variables S_k , $k=1, 2, \dots$, which includes the k th derivative of h with respect to t . Using S_1 , S_2 , $\dots$, we expand $\phi(\eta, t)$, $F(\eta, t)$, $G(\eta, t)$, $\dots$ as follows:

$$\phi = S_1 \phi_1(\eta) + S_2^2 \phi_{11}(\eta) + \dots \left. \begin{array}{l} + S_2 \phi_2(\eta) + \dots \\ + \dots \end{array} \right\} \quad (22)$$

$$F = F_0(\eta) + S_1 f_1(\eta) + S_1^2 f_{11}(\eta) + \dots \left. \begin{array}{l} + S_2 f_2(\eta) + \dots \\ + \dots \end{array} \right\} \quad (23)$$

$$G = G_0(\eta) + S_1 g_1(\eta) + S_1^2 g_{11}(\eta) + \dots \left. \begin{array}{l} + S_2 g_2(\eta) + \dots \\ + \dots \end{array} \right\} \quad (24)$$

which are called multifold series expansion.

Substituting these series into the partial differential equations of ϕ , F , G , P_1 , P_2 , $\dots$ and setting the coefficients of S_k , $k=1, 2, \dots$, equal to zero, we have a set of ordinary differential equations of $\phi_1(\eta)$, $\phi_{11}(\eta)$, $\phi_2(\eta)$, $F_0(\eta)$, $f_1(\eta)$, $f_{11}(\eta)$, $f_2(\eta)$, $G_0(\eta)$, $g_1(\eta)$, $g_{11}(\eta)$, $g_2(\eta)$, $\dots$ with respect to η . These equations can easily be solved by the numerical method.

(a) Pressure ($a=0.38$) (b) Flow force ($a=0.43$)

Fig. 10 Range of applicability of the theory
 a : normalized amplitude of the gap-width,
 r^* : normalized radial coordinate

Figure 10 shows the range of applicability of the theory for the case of sinusoidally oscillating gap width with time, where R_q and R_w are two kinds of Reynolds numbers defined as $R_q = h_m q_s / (\nu r_e^2)$, $R_w = h_m^2 \omega / \nu$, and h_m is the mean gap width.

5.3 Flow visualization inside fluid power components

The flow visualization technique has undergone great development in this decade, and, in the field of fluid power engineering, this technique has come to be used mainly for the study of flows inside valves.

Ohshima et al. observed cavitation⁽⁵¹⁾ occurring in the half-cut model of a poppet valve with a transparent cover and investigated the characteristics of the valve under cavitation. They⁽⁵²⁾ also investigated the flow force acting on a poppet valve using a similar model.

Nakano et al.⁽⁵³⁾ measured the axial force acting on the spool of a valve, using the half-split model of a spool valve with transparent cover and observed the flow pattern. Based on this flow pattern, they concluded that the measured force is consistent.

6. Flow-Induced Vibration in Fluid Power Systems

The pressure level in a hydraulic system is generally so high that the pressure drop across a control valve varies greatly with a small change in the opening of the variable orifice of the valve; in other words, the gain of pressure drop to valve-opening is very large. The change in pressure drop affects flow forces acting on the valve body, and these forces in turn vary the opening. Consequently, a control valve constructs itself a closed feedback system with a high gain element; this means that it may fall into a self-oscillation.

Many studies have been reported on such phenomena. They are classified into three groups: studies on

the oscillation caused by 1) the valve itself, 2) a system consisting of a valve and a pipeline, and 3) a fluid power system including control valves.

1) Valve itself^{(54)~(57)} Some valves are capable of self-oscillation under some conditions. This kind of instability can be explained as the valve has an equivalent negative damping coefficient⁽⁵⁷⁾.

2) Combination of a valve and a pipeline^{(58)~(61)} The system consisting of a valve and a pipeline is inherently unstable⁽⁶⁰⁾ provided that no friction exists. A method⁽⁶¹⁾ was proposed to stop the self-oscillation.

3) Fluid power system including valve(s)^{(61)~(62)} There are some cases in which the load of the system plays an important role⁽⁶²⁾. In general, the analysis taking into account the whole system is so complicated that a method⁽⁶¹⁾ was proposed in which analysis was carried out only on a part of the system because the rest of the system was considered to have little effect on the oscillation.

7. Conclusions

The recent development of fluid mechanics in fluid power engineering has been described, although there are many other studies which were not referred to in this paper because of space limitations. As stated in Section 1, fluid power engineering is based on both fluid mechanics and control engineering. Fluid mechanics is useful especially in developing fluid power components. It is strongly expected that further development of such components will result from the application of modern fluid mechanics.

References

- (1) Zielke, W., Frequency-Dependent Friction in Transient Pipe Flow, *Trans. ASME, Ser. D*, Vol. 90, No. 1 (1968), p. 109.
- (2) Trikha, A.K., An Efficient Method for Simulating Frequency-Dependent Friction in Transient Liquid Flow, *Trans. ASME, Ser. J*, Vol. 97, No. 1 (1975), p. 97.
- (3) Kitagawa, A., Kagawa, T. and Takenaka, T., High Speed and Accurate Method for Transient Response of Pneumatic Transmission Line Using Characteristics Method (in Japanese), *Trans. Soc. Instr. Cont. Eng.*, Vol. 20, No. 7 (1984), p. 648.
- (4) Kagawa, T., Lee I., Kitagawa, A. and Takenaka, T., High Speed and Accurate Computing Method of Frequency-Dependent Friction in Laminar Pipe Flow for Characteristics Method (in Japanese), *Trans. Jpn. Soc. Mech. Eng.*, Vol. 49, No. 447 (1983), p. 2638.
- (5) Cox, R. H., Wave Propagation through a Newtonian Fluid Contained within a Thick-Walled, Viscoelastic Tube, *Biophysical J.*, Vol. 8 (1968), p. 691.
- (6) Nakano, K. and Yoshimoto, M., The Dynamic Characteristics of a Hydraulic Pipe Line with a Viscoelastic Pipe Wall (in Japanese), *Trans. Soc. Instr. Cont. Eng.*, Vol. 6, No. 3 (1970), p. 201.
- (7) Oguchi, T. and Nakano, K., Oil Hammer in Viscoelastic Pipe Lines Used in Oilhydraulic Circuits (in Japanese), *J. Jpn. Hyd. Pneu. Soc.*, Vol. 9, No. 2 (1978), p. 120.
- (8) Gally et al., An Investigation of Pressure Transients in Viscoelastic Pipes, *Trans. ASME, Ser. J*, 101-4 (1979), p. 495.
- (9) Hashimoto, K., Imaeda, M. and Kikuchi, K., Transient Response of Oilhydraulic Pipelines with Viscoelastic Walls (in Japanese), *J. Jpn. Hyd. Pneu. Soc.*, Vol. 17, No. 1 (1986), p. 66.
- (10) Atabek H. B. et al., Oscillatory Flow near the Entry of a Circular Tube, *ZAMP*, Vol. 12 (1961), p. 185.
- (11) Atabek H. B., Development of Flow in the Inlet Length of a Circular Tube Starting from Rest, *ZAMP*, Vol. 13 (1962), p. 417.
- (12)~(13) Nakano K. et al., Research on Unsteady Flow in a Tube, 2nd and 3rd Reports, (in Japanese), *J. Jpn. Hyd. Pneu. Soc.*, Vol. 8, No. 1 (1977), p. 60, and Vol. 9, No. 3 (1978), p. 190.
- (14)~(16) Nakano, K. and Yokota, S., Oscillatory Flow in the Inlet Region of a Circular Tube, 1st, 2nd and 3rd Reports (in Japanese), *J. Jpn. Hyd. Pneu. Soc.*, Vol. 10, No. 1 (1979), 64, Vol. 11, No. 2 (1980), p. 107, and Vol. 12, No. 3 (1981), 213.
- (17) Washio, S., Konishi, T. and Okamura, K., Research on a Propagation Phenomena in Hydraulic Line (3rd Report, Propagating Characteristics in Tapered Tube), (in Japanese), *Trans. Jpn. Soc. Mech. Eng.*, Vol. 40, No. 329 (1974), p. 208.
- (18) Tanahashi, T. and Ando, T., Distorted Pressure Histories Due to the Step Responses in a Laminar Tapered Tube (in Japanese), *Trans. Jpn. Soc. Mech. Eng.*, Vol. 42, No. 356 (1976), p. 1137.
- (19) Muto, T., Kinishita, Y. and Yoneda, R., Dynamic Response of Tapered Fluid Lines (1st Report, Transfer Matrix and Frequency), (in Japanese), *Trans. Jpn. Soc. Mech. Eng.*, Vol. 46, No. 409 (1980), p. 1705.
- (20) Takahashi, K., Yamashita, Y. and Yamaguchi, R., The Measurement of Laminar Pulsating Instantaneous Flow Rate with a Venturi Meter (in Japanese), *Trans. Soc. Instr. Cont. Eng.*, Vol. 18, No. 6 (1982), p. 589.
- (21) Ikeo, S. and Uzawa, K., Oscillatory Flow through a Convergent Tube (in Japanese), *Trans. Jpn. Soc. Mech. Eng.*, Vol. 51, No. 472 (1985), p. 4230.
- (22) Benedik, P. J., A Laser Doppler Velocimeter to Measure Instantaneous Velocity Profiles, *Flow Its Measurement and Control in Science and Industry*, Vol. 1, Part 2 (1974).
- (23) Nakatani, N., et al., (in Japanese), *Prepr. of 6th Flow Visual. Symp.*, (1978), p. 109.
- (24) Uchiyama, M. and Hakomori, K., A Filter to Estimate Velocity Profiles of Fluid in a Circular

- Pipe (in Japanese), Trans. Soc. Instr. Cont. Eng., Vol. 18, No. 7 (1982), p. 653.
- (25) Nakano, K., Yokota, S. and Ueyama, M., An Approach of Instantaneous Flowrate Measurement in a Circular Tube (in Japanese), Trans. Soc. Instr. Cont. Eng., Vol. 20, No. 5 (1984), p. 421.
 - (26) Nakano, K., Yokota, S. and Miyamoto, T., A Study of Instantaneous Flow Rate Measurement in a High-Frequency Region, Based on Hydraulic Pipe Line Dynamics (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 52, No. 473, B (1986), p. 413.
 - (27) Yokota, S. and Nakano, K., An Approach of Measurement of Unsteady Flowrate through Circular Tubes in Hydraulic Systems (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 17, No. 3 (1986), p. 200.
 - (28) Zhao, T., Kitagawa, A., and Takenaka, T., A Real Time Measuring Method of Unsteady Flow Rate Employing a Centerline Velocity in a Circular Tube (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 52, No. 482 (1986), p. 3484.
 - (29) Uchiyama, M., Liu, F. and Hakomori, K., Real Time Estimation of Transient Flow Velocity Profile Using a Hot Film Velocimeter (in Japanese), Trans. Soc. Instr. Cont. Eng., Vol. 22, No. 9 (1986), p. 962.
 - (30) Zhao, T., Kitagawa, A., Kagawa, T. and Takenaka, T., A Real Time Measuring Method of Unsteady Flow Rate and Velocity Employing a Differential Pressure in a Pipe (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 52, No. 482 (1986), p. 2851.
 - (31) Ohuchi, H. and Ikebe, Y., A Proposal of Measurement of Oscillatory Flow Rate (in Japanese), Trans. Soc. Instr. Cont. Eng., Vol. 18, No. 10 (1982), p. 987.
 - (32) Weston, W. and Lalor, M. J., Determination of the Dynamic Flow Characteristics of an Electro-Hydraulic Servovalve Using a Laser Velocimeter, J. Phys. D: Appl. Phys., Vol. 112 (1979), p. 203.
 - (33) Nakano, K. and Yokota, S., Instantaneous Flowmeter for Hydraulic Systems Utilizing the Dynamic Characteristics of Cylindrical Chokes (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 17, No. 4 (1986), p. 311.
 - (34) Backe, W. and Riedel, H. P., Kavitation in Olhydraulischen Systemen, Industrie-Anzeiger, Vol. 94, No. 8 (1972), p. 153.
 - (35) Ishihara, T., Ouchi, M., Kobayashi, T. and Tamura, N., An Experimental Study on Cavitation in Unsteady Oil Flow (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 44, No. 388 (1978), p. 4215.
 - (36) Lee, I., Kitagawa, A. and Takenaka, T., On the Transient Behavior of Oil Flow under Negative Pressure, Bull. JSME., Vol. 28, No. 240 (1985), p. 1097.
 - (37) Lee, I., Cavitation Inception Caused by Transient Flow in an Oil Hydraulic Pipeline (Effects of Solid Particles and Gas in Liquid), (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 52, No. 476 (1986), p. 1581.
 - (38) Smith, K.J. and Hamrick, J. T., A Rapid Approximate Method for the Design of Hub Shroud Profiles of Centrifugal Impellers of Given Blade Shape, NACA TN 3399, (1965).
 - (39) Katsanis, T., NASA TN D-2546, (1964).
 - (40) Duda, J.L. and Vrentas, J.S., Fluid Mechanics of Laminar Liquid Jets, Chem. Eng. Sci., Vol. 22 (1967), p. 855.
 - (41) Takahashi, K., A Numerical Analysis of Flow Using Streamline Coordinates, Bull. JSME., Vol. 25, No. 209 (1982), p. 1696.
 - (42) Takahashi, K. and Tsukiji, T., Numerical Analysis of a Laminar Jet Using a Streamline Coordinate System, Trans. Can. Soc. Mech. Eng., Vol. 9, No. 3 (1985), p. 164.
 - (43) Tsukiji, T. and Takahashi, K., Numerical Analysis of an Axisymmetric Jet Using a Streamline Coordinate System, Jpn. Soc. Mech. Eng. Int. J., Vol. 30, No. 267 (1987), p. 1406.
 - (44) Takahashi, K., Tsukiji, T. and Sakagami, T., Image Processing of a Jet Flow Using a Streamline Coordinate System, Fluid Control Measurement, Vol. 2, (1985), Pergamon Press, p. 711.
 - (45)~(48) Iboshi, N. and Yamaguchi, A., Characteristics of a Slipper Bearing for Swash Plate Type Axial Piston Pumps and Motors (1st - 4th report), Bull. JSME., Vol. 25, No. 210 (1982), p. 1921; Vol. 26, No. 219 (1983), p. 1589; Vol. 29, No. 254 (1986), p. 2529; Vol. 29, No. 254 (1986), p. 2539.
 - (49) Ishizawa, S., Watanabe, T. and Takahashi, K., Unsteady Viscous Flow Between Parallel Disks with a Time-Varying Gap Width and a Central Fluid Source, Trans. ASME, J. Fluids Eng., Vol. 109 (1987), p. 394.
 - (50) Ishizawa, S., A New Theory of the Multifold Series Expansion for the Boundary Layer Equations and Its Application (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 49, No. 443, B (1983), p. 1345.
 - (51) Oshima, S. and Ichikawa, T., Cavitation Phenomena and Performance of Oil Hydraulic Poppet Valve (2nd Report, Influence of the Chamfer Length of the Seat and the Flow Performance), (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 51, No. 462 (1985), p. 628.
 - (52) Oshima, S. and Ichikawa, T., A Study on the Compensation of Flow Force in a Poppet Valve (2nd Report, Examination by a Half Cut Model), (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 19, No. 2 (1988), p. 169.
 - (53) Nakano, K., Watanabe, H. and Guo, M., Experimental Study for the Compensation of Axial Flow Force in a Spool Valve (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 18, No. 6 (1987), p. 475.
 - (54) Blackburn, J. F., Reethof, G. and Shearer, J. L., Fluid Power Control, (1960), 359, John Wiley & Sons.
 - (55) Funk, J. E., Poppet Valve Stability, Trans. ASME,

- Vol. 86, Ser. D-2 (1964), p. 207.
- (56) Maeda, T., Studies on the Dynamic Characteristics of a Poppet Valve (1st Report, Theoretical Analysis), (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 35, No. 274 (1969), p. 1285.
 - (57) Takahashi, K. and Yamaguchi, R., On the Stability of an Oil-Hydraulic System with a Control Valve (1st Report, Stability Criterion), (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 10, No. 3 (1979), p. 191.
 - (58) Fujii, S., Excitation of Vibration of a Fluid-Column by a Fluttering Valve (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 18, No. 73 (1952), p. 374.
 - (59) Ainsworth, F. W., The Effect of Oil-Column Acoustic Resonance on Hydraulic Valve Squeal, Trans. ASME, Vol. 78, No. 5 (1956), p. 773.
 - (60) Urata, E., Stability Problem on the System Consisting of Spring, Mass and Fluid (in Japanese), Trans. Jpn. Soc. Mech. Eng., Vol. 37, No. 301 (1971), p. 1653.
 - (61) Takahashi, T. and Yamaguchi, R., On the Stability of an Oil-Hydraulic System with a Control Valve (2nd Report, Stabilization of a System Using Volumes and Restrictions), (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 10, No. 3 (1979), p. 200.
 - (62) Kojima, E. and Shinada, M., A Study on Stability of Hydraulic Control Circuits with Counter Balance Valve (in Japanese), J. Jpn. Hyd. Pneu. Soc., Vol. 8, No. 2 (1977), p. 122.
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