Development of Accumulator for High Frequency Ripple Absorption*

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A new accumulator is developed which is possible to absorb ripples by pump and cavitation in valve, ranging from several hundred to several thousand c/sec in the hydraulic system. (Acc: abbreviation of Accumulator)

The attenuation of Acc in the case of constant flow source and constant pressure source when the pipe is terminated with a thrutting load, was computed and compared with experimental values.

Observation of the bladders of hat and belows type used in this experiment, was made and by life test, the endurance limit of them was investigated.

By improving the bladder's profile and smoothing stress distribution, the author affirmed that this Acc had good endurance limit under ripple pressure of about ±15 kg/cm².

1. Introduction

Heretofore, Acc of bladder type has been used for the ripple absorption of a piston pump with 2 or 3 cylinders(1)-(3).

But, the ripples of high speed pump and cavitation of valve and throttle at the pipe in the hydraulic system, are ranging from several hundred to several thousand c/sec, and are source of noise in the hydraulic system.

The Acc's of present commercial uses are not designed for high frequency ripple absorption and the effect is unsatisfactory.

The author paid attention to it, and made tentatively several types of Acc's for high frequency ripple absorption. Also, he affirmed the effectiveness of them, and could construct a practical type by considering the endurance limit.

In the design of Acc for high frequency use, the author aimed at the attenuation of 10~30 dB for 1~4th components of high speed pump ripples, viz. 200~1 000 c/sec.

This paper states the attenuation characteristics of the Acc for high frequency ripple absorption, experimental result for the pump ripples and cavitation ripples of random nature, and the deformation and endurance limit of the Acc which is made tentatively.

Main nomenclatures in this report are as follows.

- TL: transmission loss dB
- P, Q: pressure kg/cm², flow rate cm³/sec
- Pₚ₀, Pₚₐ: n denotes position, 0 means the pressure without Acc and a means the pressure with Acc. Similar for Q
- S: sectional area of pipe cm²
- Sₚ: sectional area of Acc neck cm²
- ρ: density of oil kg/cm³
- c: sound velocity of oil cm/sec
- Rₜ: throttle resistance kg/cm²
- R: characteristic impedance of pipe (=ρc/S) kg/cm²
- ε, μ: frequency ratio ω/ωₑ and ωₑ/ωₚ
- ωₑ: corner angular frequency rad/sec
- R: fluid friction of pipe (=8νγr²)sec⁻¹
- ν: kinematic viscosity of oil cm²/sec
- r: radius of pipe cm
- mₑ, cₑ, Rₑ, ωₑ: equivalent mass, damping coefficient, spring constant and angular natural frequency of Acc
- Vₑ: gas volume of Acc cm³
- n: polynome index of gas
- lₑ: length of Acc neck cm
- ζ: damping ratio of Acc

2. Attenuation characteristics of Acc for high frequency ripple absorption

The causes of a noise which is originated in the pipe lines of the hydraulic system are classified as follows (i) pump ripple which has a fundamental frequency of vane number x revolution and its harmonics, stationary vibration ranging from 100~1 000 c/sec, (ii) pressure ripple by cavitation which
has random nature, ranging from 100 ~ 2500 c/sec.

There are two methods of absorbing these pulsations, viz. (a) muffler of expansion cavity and (b) Acc type. (a) has demerits of large dimensions and heavy weight when the hydraulic system is high pressure and large capacity; on the contrary, type (b) has merits of being compact and light weight.

But, generally, it is difficult to take attenuation band wide and it is a problem how to make it broader. In order to absorb broad band high frequency ripple, it is necessary to make the natural frequency high as helmholtz resonator and the attenuation band wide.

2.1 Calculation method of the attenuation of Acc (In case of constant flow source)

Flow fluctuation of the hydraulic pump is caused, mainly, by geometrical structure. Consequently, it is regarded as a constant flow source when the pump is regarded as the source of ripple.

Constant flow source is defined as one in which the internal resistance of the sound source itself is infinity and constant flow rate is produced from the source independent of external pressure. In this case, it is supposed that the pipe pressure near the Acc neck changes uniformly.

In the Acc system of Fig 1, stationary pulsation wave is generated at the constant flow source of the right end, and the left end is terminated with a throttling resistance of value $R_s$. Supposing that $I_1 \sim I_3$ are transfer functions of I \sim III parts respectively,

$$\frac{P_s}{Q_s} = I_1 \times (P_1/Q_1), \quad \frac{P_s}{Q_s} = I_2 \times (P_2/Q_2), \quad \frac{P_s}{Q_s} = I_3 \times (P_3/Q_3)$$

where, $I_1, I_2$ are transfer function of straight pipe,

$$I_{1,2} = \frac{\cosh (\beta l/c)}{\left(\frac{S_s}{S} \sinh (\beta l/c)\right), \quad \cosh (\beta l/c)}$$

$I_2$ is a transfer function of connecting part of Acc. Supposing $P_s = \hat{P}_s$,

$$I_2 = \begin{pmatrix} 1 & 0 \\ -S_s^2 & 1 \end{pmatrix}$$

If the values of $I_1 \sim I_3$ are known, as shown above,

$$\frac{P_s}{Q_s} = I_1 \times I_2 \times I_3 \times \frac{P_1}{Q_1} = \left(2e_{z_1} z_2\right) \frac{P_1}{Q_1}$$

where, $z_1 \sim z_2$ are final matrix elements obtained by the matrix multiplication.

Relation between $P_s$ and $Q_s$ is,

$$P_s = R_s Q_s$$

If the pump is regarded as the source of ripple and denoting mean pressure and mean flow rate as $P_M, Q_M$, and equal fluid resistance toward mean flow acts on the pulsation flow, $R_s = 2P_M/Q_M$ may be regarded as the resistance to the pulsating flow.

When $Q_1$ is input and is made a reference,

$$Q_1 = \frac{z_2 z_3 - z_1 z_4}{z_2 z_1 - z_4} \quad P_1 = R_s Q_1$$

If $R_s = R_{cp}, |P_s(\omega)| = 1$ or $|Q_s(\omega)| = 1$ is known to hold. That is, in the pipe III, no reflection of sound wave occurs, and phase change only happens.

In this case, the pressure without Acc, for example, the pressure at the resistance termination $P_{so}$ is $|P_{so}| = R_{cp} |Q_{so}| = R_{so} Q_1$, by the condition of no reflection. With Acc, $|P_{so}| = R_{so} Q_1$ and $|P_{so}(\omega)|/|Q_{so}(\omega)| = |Q_s(\omega)/Q_1(\omega)|$.

Consequently, flow response is equal pressure response of the same point with and without Acc. Generally, $R_{so} > R_{cp}$, and flow response and pressure response defined here are not equal.

2.2 Fundamental nature of attenuation characteristics of Acc (In case of constant flow source)

As the fundamental expression of Acc characteristics, the author treats the case of $R_s = R_{cp}$ and takes the distance between input source and Acc sufficiently short compared with wave length of sound. That is,

$$I_1 = \begin{pmatrix} 1 & 0 \\ 0 & 1 \end{pmatrix}$$

The length of pipe III has no meaning, and it is sufficient to assume that resistance termination without reflection is connected to Acc output port,

$$P_s = R_s Q_s$$

From Eqs. (7) (8) and (3), the ratio of output flow rate $Q_s$ to input flow rate $Q_1$ is,

$$\frac{Q_s}{Q_1} = \frac{\omega^2 - \omega_s^2 - j(C_s/m_s) \omega}{\omega^2 - \omega_s^2 - j(R_s S_s^2/m_s + C_s/m_s) \omega}$$

Assuming that $\hat{P}_s$ is the mean value of feed gas pressure of Acc, the angular natural frequency $\omega_n$ of Acc is,

$$\omega_n = \sqrt{\frac{\hat{P}_s}{m_s}} = \sqrt{n \hat{P}_s S_s}{\rho \nu \alpha}$$

Furthermore, the corner angular frequency $\omega_c$ of Eq. (9) is defined as follows.

$$\omega_c = \frac{n \hat{P}_s}{R_s \nu^2} = \frac{n \hat{P}_s S_s}{\rho \nu \alpha}$$
Transmission loss $TL_\text{Q}$ is expressed as follows by the parameters $\varepsilon, \mu, \zeta$.

$$TL_\text{Q}=20\log_{10}\frac{|Q_1|}{|Q_2|}=20\log_{10}\frac{\varepsilon}{\mu+2\zeta}$$

$$\frac{(\varepsilon^2-1)^2+(\varepsilon/\mu+2\zeta)^2}{(\varepsilon^2-1)^2+4\zeta^2}$$

where, $\zeta=\frac{\omega_s}{\omega_\alpha}$, $\varepsilon=\frac{\omega_s}{\omega_\alpha}$, $\mu=\frac{\omega_s}{\omega_\alpha}$.

$TL_\text{Q}$ is drawn by taking $\varepsilon$ as abscissa and taking $\mu$ as parameter in Fig. 2. $TL_\text{Q}$ reaches its maximum at $\varepsilon=1.0$ (where, the case of $\zeta=0.01$ is drawn). In this case, clearly, pressure response $|P_{\text{in}}(\omega)/P_{\text{out}}(\omega)|$ is equal flow rate response $|Q_s(\omega)/Q_1(\omega)|$.

The attenuation band can be taken wider by setting $\omega_\alpha$ small, that is, setting $\mu$ small, under the condition of $\omega_\alpha$ constant. That is,

$$\mu=\frac{S}{S_0}=\sqrt{\frac{\alpha n}{\beta c^2}}, \quad \alpha=\frac{S_0}{\lambda_0 V_0}$$

That is, in order to make the attenuation band wider under the condition of $\omega_\alpha$ constant ($\omega_\alpha=$ constant), it is necessary to make $S/S_0$ small, when the value of the root of Eq. (13) is constant.

It is concluded that in order to make $\omega_\alpha$ larger and the attenuation band wider, $S_0$ and $V_0$ must be chosen large and $\lambda_0$ chosen small.

### 2.3 Example of calculation of attenuation value of Acc in the case of throttling resistance load (constant flow source)

In Fig. 1, the influence of resistance $R_\alpha$ on the attenuation characteristics is calculated, the object of which is the example of experiment shown afterward.

That is, conduit dimensions are, $l=25$ cm, $l_2=300$ cm, $e=1000$ m/sec, $S=3.1$ cm², $\rho=0.9 \times 10^{-7}$ kg sec/cm², $R=8$ sec⁻¹, characteristic impedances of pipe $R_p=2.9 \times 10^{-7}$ kg sec/cm², Acc dimensions are $e_0=2.1 \times 10^{-5}$ kg sec/cm², $\zeta=0.1$, $m_\nu=5.6 \times 10^{-5}$ kg sec cm⁻¹, $S_0=9.6$ cm², $\omega_\alpha=1.88 \times 10^4$ rad/sec ($f_\alpha=300$ c/sec).

In Fig. 3(a), the calculated value of $TL_\text{P}=20\log_{10}|P_{\text{in}}/P_{\text{out}}|$ is shown taking $R_\alpha$ as parameter. The curve of $TL_\text{P}$ is nearly equal to $TL_\text{Q}$. That is, the curve of $R_\alpha=R_\text{sp}$ is influenced by the reflection between Acc and input source distance $l_1$ only, and $TL_\text{P}$ become negative from about 900 c/sec, and this is the upper limit of the attenuation band.

In proportion to $R_\alpha$ becoming larger than $R_\text{sp}$, the influence of reflection between Acc and resistance appears and unevenness appears in the curve of $R_\text{sp}$.

The frequency corresponding to the cancave part of the figure corresponds to anti-resonance frequency and the frequency corresponding to the convex part of the figure corresponds to resonance frequency.

In Fig. 3(b), $TL_\text{Q}=20\log_{10}|Q_1/Q_2|$ is drawn by the parameter $R_\alpha$, and the convex part of the attenuation curve is becoming larger in proportion to $R_\alpha>R_\text{sp}$, when $R_\text{sp}$ is taken as reference.

The convex part spreads with the resonance point of the pipe without Acc as its centre. As shown above, in the case of $R_\alpha>R_\text{sp}$, pressure response and flow rate response differ, the former is the characteristic of fluctuation taking $R_\alpha=R_\text{sp}$ as its centre, and the latter is the characteristic of fluctuation taking the backbone curve of $R_\text{sp}$ as its minimum line.

### 2.4 Attenuation characteristic of Acc in case of constant pressure source

In case of cavitation pulsation, it may be adequate to assume that the pressure itself produced by the formation and collapse of air bubbles is the source of pulsation.

That is, in this case, when the sound source is considered to be the generating position of cavitation, the pressure produced at the position is assumed to be constant, no matter what sort of load is connected, under the condition that the nature of cavitation remains constant. This is a case of zero impedance of the sound source.
In this case, the pressure response \( \frac{P_1}{P_s} \) (\( P_1 \): input source) is attained similarly. This response is evidently equal to \( \frac{P_{a1}}{P_{a1} \text{ in the case of } R_b = R_{1b}} \). The response is different from the flow rate response mentioned in the former section, and the upper and lower limits of the attenuation band are determined from the distance between Acc and constant pressure source.

In Fig. 4, \( TL_p = 20 \log_{10} \left| \frac{P_i}{P_s} \right| \) is shown, whose dimensions are equal to those in the former section, \( l_1 \) being changed 10, 25, 50 cm under the condition of \( R_b = R_{1b} \).

If \( \omega l_1/c \) is small, the following Eq. is deduced easily.

\[
\frac{P_1}{P_s} = \frac{m_{p}}{m_{s}} \left[ 1 - \left( \frac{Q_s}{\omega} \right)^2 \right]
\]

\[
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\]

\[
Q_s = \sqrt{\frac{k_s}{m_{ps}}}
\]

That is, the lower limit of the attenuation band is determined by \( Q_s \), and the attenuation amount is determined by \( m_{ps}/m_s \).

Also, the upper limit of it is determined by \( \omega = \pi l_1/c \). This value is twice the upper limit of angular frequency of the constant flow source.

3. Structure and deformation of Acc for high frequency ripple absorption

In order to develop practical Acc for high frequency ripple absorption, prerequisite condition is its sufficient durability.

As the present aim, the author set 3 conditions, namely, (1) Acc can endure about 100 kg/cm\(^2\) of line pressure, (2) its rubber bladder has the strength which can bear the external pressure twice the feed pressure of Acc; and (3) it has sufficient capacity for pulsation absorption.

In the first place, it is necessary that rubber bladder doesn’t rupture by over-expansion, when initially gas is impregnated to Acc. For this purpose, usually, a protection for rubber bladder is provided by making Acc neck narrow.

But in order to be effective for high frequency, the neck of Acc must be large, as evident from the above calculation.

For this purpose, a metallic plate is attached at the lower part of Acc rubber bladder and at the same time, protrusion of rubber bladder is prevented by conforming the vessel to its profile.

As the structure of the rubber bladder, the author adopts a cylindrical form (abbreviation of which is hat type) which is simplest in structure and a single row bellows.

From the necessity of neck diameter to be large, it is difficult to screw Acc vessel to pipe, and we must consider an Acc structure which is integral with pipe.

Some examples of trialy manufactured structure of Acc vessel meeting these conditions are shown in Fig. 5.

In the figure, (a) is similar to the conventional type, and named separate or S type, (b) is the hat type Acc of one body structure, named I\(_h\) type, (c) is the bellows type Acc of one body structure, named I\(_b\) type. Natural volume of Acc bladder is about 35～50 cc.

First, we observed the states of deformations of...
these Acc rubber bladder by transparent acril vessel. Deformed states of hat type Acc are shown in Fig. 6.

The length of acril vessel is twice the length of rubber bladder.

So, rubber bladder expands about twice when gas is fed (a). From this state, by gradual expansion, the rubber bladder shrinks axially (b). When the rubber length becomes about 1.6 times, pleats begin to form (c). This is the point where the elongation of rubber settles by creep.

By further expansion from this state, the radial pleat grows larger and the rubber bladder deforms, as if it were buckled unsymmetrically.

Consequently, when the difference between feed pressure and line pressure is too large, over stress of the rubber membrane is expected.

Deformed states of the rubber bladder of bellows type Acc are shown in Fig. 7. By pressurizing Acc, rom the initial feed in Fig. (a), bellows begins to deform almost axially.

But, when the line pressure is too large compared with feed pressure, the bellows begins to deform unsymmetrically, and the metal plate begins to encroach the rubber part.

From these results, it was concluded that generally, the bellows type was more stable than the hat type.

4. The experiment of Acc attenuation characteristics

4.1 The effect on pump ripple

For experiment, we used two kinds of pumps, (a) gear pump whose tooth number is 16 and discharge flow rate is 200l/min (at 1 000 rpm) and (b) variable piston pump whose piston's number is 7, and maximum discharge flow rate 55 cc/rev.

In both cases, the pressure is regulated by the relief valve or throttle valve, connected to the output
port of the pump.

(i) The experiment in the case of small reflection of the Acc output conduit (gear pump is used)

The rotation of the pump is maintained at the constant speed of 1000 rpm. The pulsation of oil flow and noise of the pump is most influenced by the 1st~3rd harmonics (266, 532, 798 c/sec), and in order to reduce noise, Acc must operate effectively for the three components at least by the absorption of the pump ripple.

The inserting position of Acc is located about 25 cm apart from the discharge point of the pump, and the discharge pipe is mainly constituted by a high pressure rubber hoses whose length is about 10 m.

We ascertained that each component of the ripple was uniformly distributed along the pipe in the case of no Acc.

The resistance of the throttle valve \( R_t = 2P_M/Q_M \) is 0.015~0.03 (\( P_M = 25~50 \text{ kg/cm}^2 \)) for the characteristic impedance of the conduit \( R_Z = 0.01 \) (we ascertained that sound velocity of the conduit was about 1000 m/sec), and both values were comparatively close. We assumed that the conduit was terminated with no reflection. Consequently, consideration was given only to the influence of reflection of the conduit between pump and Acc.

(a) The case of S type Acc

In this case, \( S_n/S \) was 0.33, and the natural frequency of Acc was varied with 300 c/sec as its centre.

We could get the attenuation of 20~30 dB to 1st ripple component, but the attenuation band was narrow because of small neck area.

For that reason, we could hardly recognize the effect on 2nd harmonic or above. The comparison between the experimental and the calculated values were shown in Fig. 8(a). Solid line corresponds to the case when the reflection between Acc and pump isn't considered, and dotted line to the case when the reflection is taken into account.

(b) The case of I type Acc

Experiment was done, under the condition of \( S_n/S = 1 \), by enlarging the neck diameter to 35ø.

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(a) State of expansion

(b) Equilibrium state

(c) Pressurized state

(d) State of large pressurization

Fig. 7 Deformed state of rubber bladder of bellows type
This experimental apparatus is shown in Fig. 9. The natural frequency of Acc was varied with 400 c/sec as its centre.

First, we ascertained that variation of TL didn’t occur by the change of measuring stations and the state of the attenuation of the ripple was uniform along the conduit.

We also found that (i) the influence of metal plate of the rubber bladder was small, (ii) TL decreased slightly when the feed pressure was too small compared with external pressure and (iii) TL was independent of the structure of rubber bladder, viz. hat or bellows type.

TL became 20~30 dB for the ripples of 1~3 harmonics and the purpose of the experiment was achieved.

In Fig. 10, the measured result by the Brown tube is shown in the case with and without Acc at the line pressure 40 kg/cm². We found that the ripple decreased about 1/10 in total amplitude.

These data were analyzed by the fourier spectrum as before and were compared with calculated values as shown in Fig. 8(b). We acknowledged that the experimental values agreed fairly well with the calculated values which considered the reflection between Acc and pump.

(ii) Case of reflection in Acc output pipe (use of axial piston pump)

In order to investigate the influence of pipe reflection on the Acc attenuation characteristic, an experiment was conducted in the case that the pipe is terminated with a throttling resistance.

The main pipe was 3/4" and conduit conditions were similar to the case of 2.3 section. Throttle resistance was given by relief valve.

The pump rotation was changed from 750 to 1000 rpm by adjustment of the electric source frequency of the driving induction motor. The natural angular frequency of Acc was about 300 c/sec and the line pressure was 25 kg/cm².

This pump had the 1~10th harmonics whose principal harmonic was piston number × rotation cycle and the frequency response was obtained by the fourier analysis of the ripple.

\[ TL_{28} = 20 \log_{10} \left| \frac{P_{o}}{P_{a}} \right| \] and \[ TL_{24} = 20 \log_{10} \left| \frac{P_{o}}{P_{a}} \right| \]

Fig. 8 Comparison of experiment and calculation of high frequency Acc (Calculated value: \( \zeta = 0.1 \), solid line is the ideal characteristic of \( \zeta = 0 \), \( R_e = R_p \), dotted line is the case of \( \zeta = 0.1 \), \( R_e = 0 \), .□. etc. are experimental values.)

(a) Without Acc

(b) With Acc

Fig. 10 Effect of absorption of ripple pressure of gear pump by Acc
which were the ripple responses of Acc insertion position and resistance terminating position with and without Acc when the pump discharge flow rate was about 0.25 l/sec (1,000 rpm), were shown in Fig. 11.

In this case, $R_\infty = Z_\infty / Q_M$ was about 0.2, and its calculated value was drawn in the figure. The position of the peak and the minimum points of the attenuation curve agreed fairly well, but the values differed considerably.

This may be partly attributed to the lack of exactness of the experiment itself, and perhaps, main reason is supposed to be that larger damping acts on the wave motion than the resistance value given by the mean flow rate. But detailed reason is uncertain now.

4.2 The effect of Acc on the cavitation ripple

Cavitation ripple originates at the throttling section of the fluid conduit and becomes one cause of fluid noise.

This is considered the constant pressure source of the ripple approximately, as described in section 2.4. In this case, the effect of Acc was obtained by experiment and comparison with calculated values was made.

Experimental valve is the flow control valve of counter flow type, shown in literature(4). At the sharp edge orifice of the valve, negative pressure is produced in the center part of the rotating flow, and in the range of 0.2~2.0 kg/cm$^2$ of the back pressure, random ripple pressure is produced by cavitation and it propagates along the conduit.

The spectrum of the ripple differs according to the orifice profile and jet velocity and has a wide band from 10~ several thousand c/sec.

The effect of Acc was measured in the case that the cavitation ripple was severe when the back pressure was 1.5 kg/cm$^2$ by feeding the flow rate of 200 l/min to the valve by test stand.

In Fig. 12, oscillogram of the ripple with and without Acc is shown when Acc of I type is used. It is observed that the ripple is random in nature and the effect of Acc was small for the low frequency component.

The spectrum of Fourier analysis is the mean value and has no clear meaning, but it may give us some information on the mean nature of Acc effect. In this case, the comparison between experimental and calculated values is shown in Fig. 13.

In the comparatively low frequency domain below 1,000 c/sec, both agree in tendency, but in the high frequency domain, tendency becomes different.

This reason may be considered that pump is connected to the Acc output port about 10 m apart, so, the calculated condition of $R_\infty = R_{cg}$ isn't satisfied.
and the ripple has random nature.

But judging from the fact that the experimental value agrees with the calculated value to some extent, the hypothesis of constant pressure source holds approximately well, conversely.

In this case, the attenuation characteristic of Acc, different from the pump ripple, is wide and smooth, and it has the feature of small amount of attenuation.

5. Experiment of the endurance of Acc

Every type of Acc of trial construction has a metal plate for protection at its lower part, because the neck diameter is made large.

Consequently, at the state of compressional deformation, the freedom of deformation is constrained, and this point differs from the usual Acc.

The rubber bladder of hat type is simple in the form and easy for production of its vessel, but when it is given large compressional deformation, it deforms unstably as shown in Fig. 6 (d) ~ (f), large stress originates at the connection part of the rubber and metal plate.

In contrast with this, the rubber bladder of bellows of one row type has a complicated structure of the rubber itself and the internal profile of the vessel becomes special form too.

The rubber bladder itself easily deforms axially, if the external pressure is considerably higher than feed pressure, and it is stable from the structural point of view.

In order to get guide for the practical use of these Acc's, we investigated the endurance limit, giving hydraulic pulsation of 5 ~ 20 kg/cm² in total amplitude at 5 ~ 15 c/sec by using several Acc.

Pulsation generator as shown in Fig. 14, drives multiplying gear train by DC motor and double-eccentric cam apparatus (maximum total amplitude 1 mm) which is connected directly to it.

Double eccentric cam is connected to the spool of a 4-way directional valve, and by switching of oil path by the spool, pulsation is generated.

When the oil path is disconnected, the pressure in the high pressure line rises nearly to the relief pressure, and when the oil path is connected, the pressure in the high pressure line decreases. At the low cycle region, nearly rectangular pressure pulsation and at the high cycle region, nearly simple sinuous pulsation is generated.

By the life test of ±2.5 kg/cm², about ten million cycles, we couldn't discover anything unusual at both hat and bellows type Acc. When pulsation pressure rose to ±5.0 kg/cm², in some cases, crack was generated at the metal plate and the flange in both Acc.

This happened because of insufficient initial design of Acc bladder, but the frequency of the crack generations was numerous at the hat type.

Consequently, when the ripple pressure was large, we judged that the hat type Acc was inappropriate, and we tried the improvement of the bellows type.

The main improvements were two points, these are, (a) the prevention of over compressional force by the increase of the thickness of flange and taking the fastening ratio about 20%, (b) the thickness of the convex part making thinner and the metal part thicker.

As the result, we could get a rubber bladder which was fully possible to endure the cyclic ripple of 5 million cycles of ±15 kg/cm² (line pressure 40 kg/cm²).

6. Discussion

6-1 Fundamental attenuation characteristic of Acc

On the assumption that Acc operated as one degree of freedom system, which had equivalent mass (sum of the mass of Acc rubber bladder itself and the equivalent mass of oil which moved in a body with it), air spring and the damping of oil and rubber bladder, we calculated the attenuation characteristic of Acc of constant flow source and constant pressure.

![Fig. 13 Effect of Acc on cavitation ripple pressure $\left(t_1=25 \, \text{cm}, \, f_n=560 \, \text{c/sec}, \, S_a=5\times9.6 \, \text{cm}^2\right)$](image)

![Fig. 14 Life test apparatus of Acc](image)
source.

First, the problem is, whether the actual Acc operates as in this simple model or not. If the neck diameter is small to some extent, compared with the conduit diameter, this assumption may be true.

But when the neck diameter becomes equivalent to the conduit diameter or the more, as shown in Fig. 5 (b), (c), the assumption of uniformity of pressure of the connecting part of the neck and the pipe will not hold true strictly.

Consequently, the equivalent mass of the oil to the rubber changes by the frequency, Q characteristic of the attenuation curve is apparently more influenced by it than the effect of the damping of Acc itself and the curve is supposed to become smooth \( \varepsilon = 1 \) as its centre.

Because the neck diameter is large, the internal damping of the rubber influences most by calculation. Stiffness of the rubber bladder itself is about 10%, of the stiffness of air, loss factor of nitrile rubber is about 0.2, so \( \zeta \) is about 0.01~0.02.

The measured value doesn’t show any sharp peak corresponding to above data (Fig. 8, Fig. 13).

One reason for this discrepancy is assumed that the one dimensional calculation method is adopted for the Acc branch flow. Further investigation is needed about this point.

6-2 Attenuation characteristic of Acc in the case of constant flow source

In the case of constant flow source, the ideal characteristic of Acc which satisfies two conditions of \( l_1 = 0, R_e = R_{sp} \), is given in Fig. 2. But when \( l_1 \) is not equal to 0, input pipe is resonant at the frequency determined by \( l_1 = \lambda/4 (\lambda: \text{wave length of sound}) \), and the upper limit of the attenuation band is limited. The lower limit isn’t determined.

In Fig. 8(b), which is supposed to satisfy this condition, the experimental values of 1~4 components are plotted and the calculated value which considers the influence of the reflection \( l_1 \), is comparatively in good agreement with them.

In the case of \( R_e = R_{sp}, l_1 \neq 0 \), the situation becomes complex, as the effect of reflection appears.

As shown in Fig. 11, the agreement between calculation and experiment was not good. In this case, the error of taking \( R_e \) was supposed to become dominant because \( l_2 \) was short. Further survey is need about this point.

6-3 Attenuation characteristic of Acc in the case of constant pressure source

The upper and lower limits of Acc attenuation band are determined by \( l_1 \) in the case of \( R_e = R_{sp}, l_2 \neq 0 \).

That is, as shown in Fig. 4, the lower limit is determined by \( Q_s \) and the upper limit is given by \( l_1 = \lambda/2 \).

The upper limit extends twice wider than the case of section 6-2, on the contrary, the lower limit is produced.

The calculated value which assumes cavitation ripple as constant pressure source, is comparatively in good agreement with experimental value in low frequency domain, as shown in Fig. 13. Consequently, the above supposition may hold approximately.

6-4 Comparison with cavity type muffler

Large attenuation is easily attained by muffler by taking the expansion ratio large and the length appropriate.

But in the case of high pressure and large flow rate, it has the drawback of large weight and dimension. Acc has the merit of being compact and small weight.

In Fig. 15, comparison between muffler whose expansion ratio is 50 and length is 50 cm, and Acc of \( \mu = 0.05 \), is shown.

In the case of muffler, the distance between source and muffler input is taken \( l_1 \), and muffler output is assumed to terminate with no reflection condition (similar in Acc).

Figure (a) is the comparison under the condition of \( l_1 = 0 \) in the case of constant flow source, Fig. (b) is the comparison under the condition of \( l_1 = 50 \text{ cm} \) in the case of constant pressure source.

In Fig. (a), both Acc and muffler have common nature that the attenuation characteristic becomes smooth by taking \( l_1 \) short.

In Fig. (b) of constant pressure source, muffler and Acc have common feature that they show no effect, unless \( l_1 \) is taken large, they show equal tend-
ency, in spite of apparent difference between in line and branch filter.

But essentially, muffler shows cyclic attenuation characteristic and is effective up to high frequency region. But fundamentally, the attenuation characteristic of Acc is limited to around the centre of its resonance point.

So, we must use both types properly.

7. Conclusions

(1) The author developed a small Acc which could widely absorb high frequency ripple of hydraulic system.

(2) For pump ripple, calculation of attenuation of constant flow source was made and comparison with experiment was made.

(3) For cavitation ripple, calculation of attenuation of constant pressure source was made and comparison with experiment was made.

(4) Life test of the Acc rubber bladder's endurance was done and the author confirmed that it could endure ripple pressure of \( \pm 15 \text{ kg/cm}^2 \) sufficiently.

In closing this paper, the author expresses his deep gratitude to Mr. Matsuzaki and Mr. Fujisawa senior researchers for their guidances and Mr. Takenoshita for his assistance in the experiments.

References


Discussion

T. Ichikawa (Shizuoka University):

(1) How do you treat the neck length \( l_n \) in Eq. (10) of natural frequency \( \omega_n \) in case of Acc in Fig. 5? (b) (c)?

You don't consider the masses of rubber bladder and metal plate. Can this effect be ignored?

(2) The experiment is mainly conducted by gear pump of large flow rate pulsation. Pulsation of instantaneous discharge flow rate of gear pump is paraborid and different from sinus wave of calculation.

When instantaneous discharge flow rate of the pump is expressed by Fourier series, the amplitude of fundamental wave is about 19% smaller than the total amplitude of instantaneous discharge flow rate.

How do you treat this point, in comparison of calculated values and experimental values?

K. Shiomi (Kawasaki Heavy Industries Co., Ltd.):

In this field with little publications of literatures in this country and abroad, it is very interesting for you to make analysis and to conduct experiment on the damping of fluid vibration of high frequency region. I pay deep respect for your labor. I wish to ask you a few questions.

(3) Generally speaking, in the case of treating these problems, it is a problem how to suppose end condition of load side.

How, do you think, to treat the cases which are different from simple relief valve and throttle valve load as treated in this report, that is, (a) the case of driving positive displacement motor (wide band ripple originates from the load side two) (b) the case of driving cylinder of large inner volume such as press etc. (oil volume of load side changes widely by the movement of ram)?

(4) In this report, relationship between transmission loss and difference between feed pressure of rubber bladder and mean line pressure, is little mentioned. In our study, in case of resonant cavity eliminating apparatus based on the principle of Helmholtz, when line pressure is considerably lower than feed pressure, we experienced that we could acknowledge scarcely any eliminating effect. Didn't you experience such phenomenon in the Acc of your test?

(5) For the general high pressure specification of recent years in which mean line pressure fluctuates between 0~250 kg/cm\(^2\) frequently, beside, ripple component of about 15 kg/cm\(^2\) superposes over it, I think that Acc of your test has some deficiency in endurance. What do you think about this point?

Author's closure

(1) Acc for present investigation has a considerably large neck diameter, compared with pipe diameter and vessel diameter, so, Eq. (10) which is deduced under the assumption of one dimensional flow, may be considered as an approximate formula.

Consequently, the value of equivalent neck length can't be attained without exact calculation. In this report, the author couldn't describe in detail, but, approximate neck length \( l_n \) is the sum of geometrical length neck length \( l_n' \), and open end corrections
$\alpha$ and $\alpha'$ of pipe side and vessel side respectively, that is $l_a = l'_a + \alpha + \alpha'$ as shown in Append.-Fig. 1.

Usually, open end correction $\alpha$ for helmholtz resonator under the condition of small value of $l'_a$ and with flange, is assumed 0.785 $r_a$. In this case, the author adopts the value $\alpha = 0.785 r_a$. Consequently,

$$l_a = l'_a + 1.57 r_a$$

is adopted.

Next, the influence of the masses of rubber bladder and metal plate can't be ignored, generally, because initially resonance point is designed high.

So, as the equivalent mass, we should write $m_e = m_f + m_R$, $m_f = \rho l_a S_a$ (fluid mass), where $m_R$ is the equivalent mass of rubber bladder and metal plate.

By the numerical check for the Acc under test, in case of neck profile $35\phi \times 5L$ (length), $m_f = 30$ gram is obtained. By supposing that mass distribution along the rubber bladder is uniform and it moves like a piston, $m_R = m_{R0}/3$ ($m_{R0}$: total masses of rubber bladder and metal plate) is attained. $m_{R0}$ is about 30 grams, so, $m_R$ is nearly 10 grams.

Consequently, natural frequency decreases about 15%. In the calculated curves of attenuation shown in Fig. 8 and so on, the natural frequency is taken, considering these effects.

(2) As you mentioned here, instantaneous discharge flow rate of the gear pump under test is paraboloid.

In this report, the pump's flow rate is analyzed to Fourier spectrum by direct current, fundamental and its harmonics, and is assumed to be superimposes of them.

For example, the analyzed results of pump ripple wave form with and without Acc by continuous frequency analyzer of B & K Co. is shown in Append.-Fig. 2. TL by Acc is as shown in the figure and wave form is analyzed to 5th harmonics.

I think, it is not unnatural to discuss by Fourier analysis when direct current and wave harmonics are superimposed as this. But it is a question if it is right to take equal throttle resistance $R_e$ to direct current flow and wave flow components.

In this report, I take equal values. Further
investigation is needed about these points.

(3) This time, I considered fluid resistance load by throttle valve $Z = P/Q = R_e$ as the load condition, in order to grasp fundamental characteristics of Acc for high frequency ripple absorption.

If load conditions are (a) hydraulic motor or (b) cylinder, equal treatment is possible by the change of load impedance.

For example, in case of (b) of cylinder load, as shown in reference (3) it is sufficient to define cylinder impedance $Z = (mS^2)/(s)$ (s: Laplacian operator) and connect it to pipe. Same treatment is applied to case (a). (see Append.—Fig. 3).

In these cases, problem is that the influence of reflection of Acc output pipe $l_2$ comes out anew. That is, the effect of Acc becomes small to the ripple frequency, at which output pipe $l_2$ resonances in the mode of Acc as free end.

I applied this Acc to the case of the large volume change by cylinder movement, but, the effect doesn't change substantially and the effect of $l_1$ is rather large.

Generally, moving velocity of cylinder is sufficiently small compared with wave propagation speed, so, the effect of movement can be ignored.

In case (a), I suppose you consider the circuit as shown in Append.—Fig. 4. In this case, generators of ripple are two. It is possible to analyze Acc's effect considering that each generator is approximately terminated with fixed end. Exactly, it is necessary to consider each load impedance.

(4) In case of rubber bladder type's Acc, relationship between line pressure and feed pressure is very important.

When line pressure becomes lower than the feed pressure, rubber bladder expands to the full and stops at stopper and the effect is lost.

When line pressure becomes higher than feed pressure, rubber bladder's volume becomes too small and together with rise of resonance frequency, attenuation band becomes narrow and the effect substantially becomes small.

I experienced the phenomena mentioned above.

(5) As you point out, I think it is impossible to some extent in the case that mean line pressure is high and fluctuates widely.

But practically, when you are concerned with the ripple pressure of high pressure region only, by taking Acc feed pressure as about 150 kg/cm², it is possible to absorb ripple to the line pressure above it. Specially it isn't problem whatever ripple component is large.

In this case, it seems rather good to consider application of expansion type muffler.

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