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COMPARISON OF VIBRATION TRANSMISSION MECHANISM IN HYDRAULIC HOSES

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SUMMARY

When considering the isolation of vibrations and pressure fluctuations of pumps and other sources by means of hydraulic hoses, it is necessary to consider several transmission mechanisms, namely bending waves, torsional waves and waves involving coupled longitudinal motion of the hose and fluid, the latter transmitting pressure variations. Using measurements of the outputs of a variety of pumps, and typical hose properties measured in resonance tests, this paper shows that, for typical pump outputs, the high attenuation of bending waves means that lateral motion and bending rotation at the pump end of the hose give rise to very little force or moment at the other end.

The most difficult pump output to be isolated is pressure ripple. In addition to problems of pressure fluctuations in subsequent parts of the circuit, the force exerted at the hose termination due to input pressure ripple exceeds that due to axial input motion and exceeds by an even greater amount that due to lateral input.

HİDROLİK HORTUMLARDA VİBRASYON İLETİM MEKANİZMASININ KARŞILAŞTIRILMASI

ÖZET

Hidrolik hortumlar vasıtasıyla, pompaların ve diğer kaynakların titreşimlerinin ve basınç dalgalanmalarının izolasyonu dikkate alındığı zaman, çeşitli iletim mekanizmalarının gözönüne alınması gerekir. Bunlar esas olarak bükülme dalgaları, burulma dalgaları ve hortum ve akışkan çiftini ihtiva eden, basınç değişimlerini ileten hortum ve akışkanın boyuna hareketidir. Bu çalışma, değişik pompaların ölçülen çıktılarını ve rezonans testlerinde ölçülen tipik hortum parametrelerini kullanarak, tipik pompa çıktıları için, bükülme dalgalarının yüksek sönümü nedeniyle hortumun pompa ucundaki lateral hareket ve bükülme rotasyonunun hortumun diğer ucunda çok küçük kuvvet veya moment meydana getirdiğini göstermektedir.

İzalasyonu en zor olan pompa çıktısı basınç dalgacıklarıdır. Devrenin takip eden bölümlerindeki basınç dalgacıkları nedeniyle hortum çıkış ucuna etkiyen kuvvetler, eksenel hareket girdisini geçtiği gibi lateral girdisini de çok daha fazla geçer.

1- INTRODUCTION

The level of acoustic radiation from hydraulic systems due to pump excitation is a matter of some concern from environmental as well as legal considerations.

Measurements of pump noise have been reported by McCandlish and Linley [1] for twenty gear pumps and by McCandlish and Petrusewicz [2] for a vane pump.

Crook and Heron [3] measured sound power levels from an axial piston and a gear pump as well as the sound power levels radiated from connected hoses and pipes. Further work was reported by Heron and Hansford [4] . Sound power levels of 55 to 65 dBA/m were found for hoses. However a pipe of 25mm o.d. connected directly to the pump produced about 90 dBA/m when the pump power level was 83 dBA.

Sound radiation from control panels [5] was found to give sound pressure levels of up to 85 dBA at 0.25mm, while a particularly noisy circuit reported by Curry [6] produced a sound pressure level of 99dBA even though the axial piston pump employed was known to be a quiet one when used in other systems.

The indications are that circuits can produce undesirably high sound levels even when pump noise is acceptable or controllable.

It is accordingly valuable to examine the levels of fluid and mechanical input from typical pumps and to compare the attenuation of each kind of input for the types of hoses commonly employed in hydraulic circuits.

In the axial direction, a pump inputs fluid pressure and wall vibration. Two waves are generated in the hose, each involving axial wall and fluid motion.

Motion of the pump at right angles to the hose produces bending waves, as does a rotational input about an axis at right angles to the hose. Rotations about the hose axis will induce torsional waves in it.

2- TRANSMISSION OF PRESSURE FLUCTUATION AND AXIAL MOTION

It has been shown [7] that pressure fluctuations within a hose and axial wall motions of the hose are transmitted by two different axial waves can be characterised by their complex propagation contants γ_1 γ_2 by modal ratios N_1, N_2 and by characteristic impedances Z_{p1}, Z_{p2} such that at a frequency w the axial wall motion $\bar{w}$ and pressure p at time t and distance x along a hose are given by

$$\bar{w} = \text{Re}\ \{ We^{jwt} \} \quad , \quad p = \text{Re}\{ P\ e^{jwt} \} \tag{2.1}$$

where
$$W = \sum_{i=1}^{2} N_i \left[C_{ai}\ e^{-\gamma_i x} + C_{bi}\ e^{\gamma_i x} \right] \tag{2.2}$$

$$P = \sum_{i=1}^{2} Z_{pi} \left[C_{ai}\ e^{-\gamma_i x} - C_{bi}\ e^{\gamma_i x} \right] \tag{2.3}$$

C_{ai} and C_{bi} are constants depending on boundary conditions.

The total axial force at any point along a fluid-filled hose is the difference between the tension in the wall and the pressure acting over the fluid cross section. If μ_h is the mass per unit length of hose, ρ is the density of the fluid and A_0 is the cross sectional are of the fluid it my be shown that the complex amplitude of the total axial force F is given by

$$F = \sum_{i=1}^{2} \frac{w}{\gamma_i} (\mu_h N_i + \rho A_0) \left[C_{ai}\ e^{-\gamma_i x} - C_{bi} e^{\gamma_i x} \right] \tag{2.4}$$

In order to examine the amounts of axial isolation that can be achieved with different types of hoses, two situations are considered, namely pressure input and velocity input. In the first case the input end of the hose is taken to be fixed, and excitation is by a pressure fluctuation at that end.The output end of the hose is assumed to be both rigidly fixed and, as far as fluctuating pressures are concerned,blocked.Clearly real systems will differ from this, but it provides a simple basis for comparison. The assumption of a blocked end will give a realistic estimate of the end force when a valve substantially constricts the flow at the hose outlet, or when there is a sharp bend in the pipe close to the end of the hose. The extent to which the hose is isolating the downstream circuit from the effects of the pressure fluctuation is indicated by the ratio of outlet to inlet complex pressure amplitudes Y_{pp} and the ratio, as complex amplitudes, of the output force at the end of the hose to the inlet pressure, Y_{fp}.

In the second case considered, that of axial velocity input, it is assumed that at the inlet the hose and the fluid within it vibrate axially with the same motion. The outlet end is again assumed to be fixed and blocked.The amount of isolation achieved in this case is measured by the ratio of outlet pressure to inlet axial velocity Y_{pw}. and by the ratio of axial force at the outlet to inlet velocity Y_{fw}. The four transmission ratios just described can be obtained by putting the appropriate boundary conditions into equations 2.2, 2.3 and in order to evaluate the constants. For a particular hose and particular elastic properties, the propagation constants are proportional to frequency [7] and this is also true of Z_{pi}. It is therefore convenient to work in terms of these quantities divided by frequency w.This will be indicated by a bar, i.e. $\bar{\gamma}_i$, $\bar{Z}_{pi}$. For a hose of length ℓ boundary conditions in the case of pressure input give the following values for the constants

$$C_{a1p} = - C_{b1p} = \left(\frac{N_2}{N_2 - N_1} \right) \left(\frac{1}{e^{-\bar{\gamma}_1 w\ell} - e^{\bar{\gamma}_1 w\ell}} \right) \quad (2.5)$$

$$C_{a2p} = - C_{b2p} = \left(\frac{N_1}{N_1 - N_2} \right) \left(\frac{1}{e^{-\bar{\gamma}_2 w\ell} - e^{\bar{\gamma}_2 w\ell}} \right) \quad (2.6)$$

In the case of velocity input the constants are

$$C_{a1w} = - C_{b1w} = \left(\frac{N_2 - 1}{N_2 - N_1} \right) \left(\frac{1}{e^{-\bar{\gamma}_1 w\ell} - e^{\bar{\gamma}_1 w\ell}} \right) \quad (2.7)$$

$$C_{a2w} = - C_{b2w} = \left(\frac{N_1 - 1}{N_1 - N_2} \right) \left(\frac{1}{e^{-\bar{\gamma}_2 w\ell} - e^{\bar{\gamma}_2 w\ell}} \right) \quad (2.8)$$

Equation 2.3 and 2.4 then give

$$Y_{pp} = \frac{2(\bar{Z}_{p1}C_{a1p} + \bar{Z}_{p2}C_{a2p})}{\bar{Z}_{p1}C_{a1p}\,(e^{-\bar{\gamma}_1 w\ell} + e^{\bar{\gamma}_1 w\ell}) + \bar{Z}_{p2}C_{a2p}(e^{-\bar{\gamma}_2 w\ell} + e^{\bar{\gamma}_2 w\ell})} \qquad (2.9)$$

$$Y_{fp} = \frac{2(\bar{Z}_{f1}C_{a1p} + \bar{Z}_{f2}C_{a2p})}{\bar{Z}_{p1}C_{a1p}(e^{-\bar{\gamma}_1 w\ell} + e^{\bar{\gamma}_1 w\ell}) + \bar{Z}_{p2}C_{a2p}(e^{-\bar{\gamma}_2 w\ell} + e^{\bar{\gamma}_2 w\ell})} \qquad (210)$$

$$Y_{pw} = 2(\bar{Z}_{p1}C_{a1w} + \bar{Z}_{p2}C_{a2w}) \qquad (2.11)$$

$$Y_{fw} = 2(\bar{Z}_{f1}C_{a1w} + \bar{Z}_{f2}C_{a2w}) \qquad (2.12)$$

where $\bar{Z}_{pi} = -\dfrac{\rho w}{\gamma_i}$, $\bar{Z}_{fi} = \dfrac{\mu_h N_i + \rho A_o}{\gamma_i}$ $(i=1,2)$ (2.13)

It can be seen that the above expressions for the axial transmission ratios involve frequency and hose length only in the form of the product $w\ell$. In figures 1 to 8 the magnitudes of the transmission ratios typical of two different types of 10 mm bore hose are shown plotted against this product. The values of the wave properties $\bar{\gamma}_i$ and N_i were obtained from axial resonance tests on 2 m lengths of hose at frequencies between 60 and 300 Hz. The method of testing and analysis has been described elsewhere [8] . When interpreting the abscissa as a variation of frequency it should be borne in mind that variation of elastic properties with frequency will give a corresponding variation in $\bar{\gamma}_i$ and N_i. This is likely to be most pronounced in the nylon hose, but in all cases the transmission is likely to be somewhat lower than that shown in the figures if the frequency is higher than that at which the wave properties were measured. The wave properties are likewise dependent on temperature, mean fluid pressure and vibration amplitude.

It can be seen from figures 1 to 8 that at low values of $w\ell$ the transmission ratios are dominated by resonances. In this region, hoses are not beneficial for isolating pressure ripple and axial motion unless it is possible to

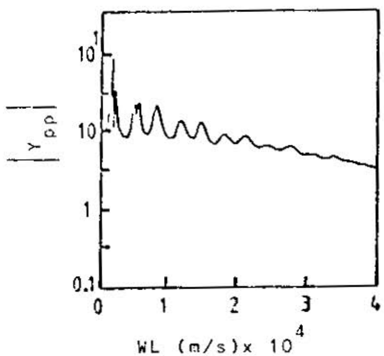

Fig.1-Output pressure/
Input pressure
Steel braid hose

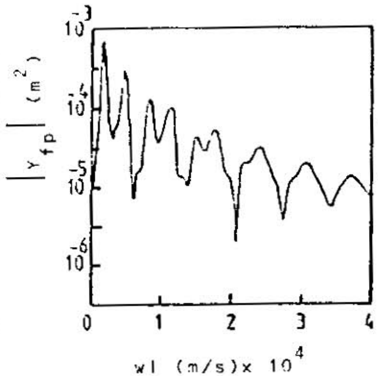

Fig.3-Axial force/Input
pressure
Steel braid hose

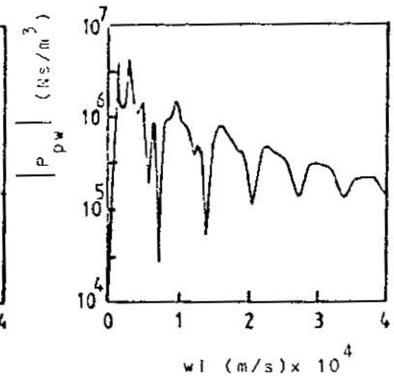

Fig.5-Output pressure/
Input axial velocity
Steel braid hose

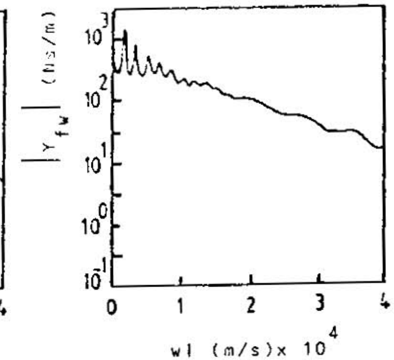

Fig.7-Axial force/Input
axial velocity
Steel braid hose

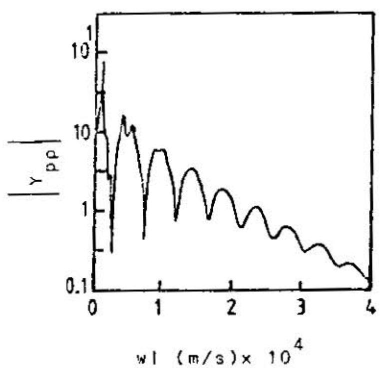

Fig.2-Output pressure/
Input pressure
Nylon braid hose

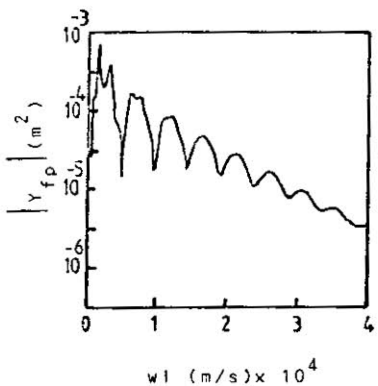

Fig.4-Axial force/
Input pressure
Nylon braid hose

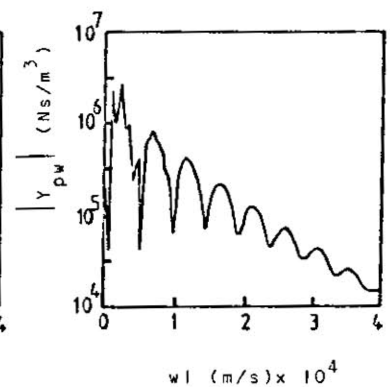

Fig.6-Output pressure/
Input axial velocity
Nylon braid hose

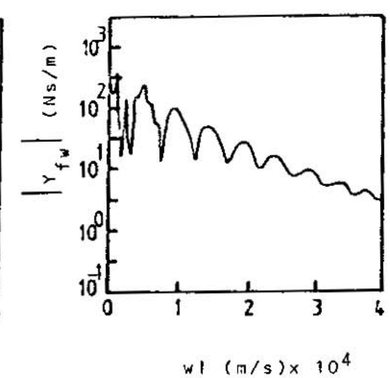

Fig.8-Axial force/Input
axical velocity
Nylon braid hose

employ tuning of circuit lengths to reduce components of vibration at particular frequencies. The problems of attempting to do this have been discussed [9] .

3. TRANSMISSION OF LATERAL FORCES

Lateral forces and moments are transmitted by bending distortions of the hose. The hose is treated as an initially straight uniform beam using the well known beam equation

$$- \mu_o \frac{\partial^2 v}{\partial t^2} = EI \frac{\partial^4 v}{\partial x^4} \qquad (3.1)$$

where v is the lateral displacement at distance x along the hose and EI is the effective flexural rigidity of the composite structure. For sinusoidal motion of all elements at a frequency w, v is given by

$$V = R_e \; \{V \;\; \exp(jwt)\} \qquad (3.2)$$

Including the damping by making EI complex and combining equations 3.1 and 3.2 gives the following equation for the variation of V with x:

$$V = C_1 \sin \lambda x + C_2 \cos \lambda x + C_3 \sinh \lambda x + C_4 \cosh \lambda x \qquad (3.3)$$

V is a complex function of x so that equation 3.2 gives both amplitude and phase variation along the hose. C_1 to C_4 are constants depending on the boundary conditions while λ is the complex constant given by

$$\lambda = \left(\frac{\mu_o w^2}{|EI|} \; e^{-j\Phi} \right)^{1/4} \qquad (3.4)$$

Bending waves propagate at a speed $w/R_e(\lambda)$. Equation 3.4 shows this propagation speed to be propotional to $\sqrt{w}$.

The bending moment at any point along the hose and the shear force is given by

$$EI \frac{d^2 V}{dx^2} = EI \lambda^2 (-C_1 \sin \lambda x - C_2 \cos \lambda x + C_3 \sinh \lambda x + C_4 \cosh \lambda x) \qquad (3.5)$$

$$EI\ \frac{d^3V}{dx^3} = EI\ \lambda^3(-C_1\cos\lambda x+C_2\sin\lambda x+C_3\cosh\lambda x+C_4\sinh\lambda x) \qquad (3.6)$$

As in the case of axial forces described in Section 2, it is convenient for the purpose of comparing different hoses to do so by considering the ratios of the forces and moments exerted by the hose when one end is rigidly fixed, to the exciting motions at the input end. Taking the hose to be rigidly fixed at the output end is quite realistic since in most practical situations the attenuation of lateral motions is high. The exciting motion generally consist of a combination of lateral and rotational motions so that four transmission ratios occur, relating output force S and output moment M to input displacement δ and input rotation θ.

Taking x=0 at the fixed end, the boundary conditions of zero slope and displacement there, and slope θ and displacement δ at $x=\ell$ give the values of the constants. The output moment and force are then the values given by equations 3.5 and 3.6 with x=0:

$$M = \frac{EI\ \lambda^2\ [\delta(\cos\lambda\ell-\cosh\lambda\ell) + (\theta/\lambda)\sin\lambda\ell-\sinh\lambda\ell)]}{\cos\lambda\ell\ \cosh\lambda\ell - 1} \qquad (3.7)$$

$$S = \frac{EI\ \lambda^3[\delta(\sin\lambda\ell+\sinh\lambda\ell) + (\theta/\lambda)(\cos\lambda\ell-\cosh\lambda\ell)]}{\cos\lambda\ell\ \cosh\lambda\ell - 1} \qquad (3.8)$$

Referring back to equation 3.4 the argument $\lambda\ \ell$ is

$$\lambda\ell = \ell\sqrt{w}\left(\frac{\mu_0}{|EI|}\right)^{1/4} e^{-j\ \Phi/4} \qquad (3.9)$$

so that for a given hose type the forces transmitted depend on the product $\ell\sqrt{w}$. When the other terms in λ in equations 3.7 and 3.8 are replaced using equation 3.4 the four transmission ratios can be identified as follows:

$$|Y_{S\delta}| = \left|\frac{S}{\delta w^{1/2}}\right| = \left|\frac{\sin\lambda\ell + \sinh\lambda\ell}{\cos\lambda\ell\cosh\lambda\ell - 1}\right|\ |EI|^{1/4}\ \mu_0^{3/4} \qquad (3.10)$$

$$|Y_{s\theta}| = \left| \frac{S}{\theta w} \right| = \left| \frac{\cos\lambda\ell - \cosh\lambda\ell}{\cos\lambda\ell\,\cosh\lambda\ell - 1} \right| |EI|^{1/2} \mu_o^{1/2} \quad (3.11)$$

$$|Y_{\delta}| = \left| \frac{M}{\delta w} \right| = |Y_{s\theta}| \quad (3.12)$$

$$|Y_{\theta}| = \left| \frac{M}{\delta w^{1/2}} \right| = \left| \frac{\sin\lambda\ell - \sinh\lambda\ell}{\cos\lambda\ell\cos\lambda\ell - 1} \right| |EI|^{3/4} \mu_o^{1/4} \quad (3.13)$$

The definition of each of the transmission ratios includes the appropriate power of w to give a single unique relationship, for a given hose type, between that transmission ratio and the length-frequency parameter $\ell\sqrt{w}$. Figures 9 to 12 show these transmission ratios plotted against $\ell\sqrt{w}$ for the two hose types already considered in connection with axial forces, the values of the hose wave properties used being shown in Table 1.

Table-1: Bending and torsion wave properties used to produce Figures 9 to 14

Reinforcement	$\|EI\|$ (Nm^2)	μ_o (kg/m)	$\tan\phi$	k_t (Nm^2/rad)	J (kg m)	$\tan\phi_t$
Double braid steel	7.68	0.560	0.113	59.3	29×10^{-6}	0.15
Double braid nylon	2.38	0.346	0.065	16.6	13×10^{-6}	0.13

It can be seen from Figures 9 to 12 that at low values of $\ell\sqrt{w}$ the transmission ratios show strong resonance effects, as for the axial ratios, but in this case there is a more regular succession of resonances as only one type of wave is present. For higher values of ℓ or w the reflections that give rise to the resonances become small and the fluctuations disappear, the transmission ratios then becoming steadily attenuated with increasing $\ell\sqrt{w}$. The first terms on the r.h.s. of equations 3.10, 3.11 and 3.13 all tend to the same value as $\ell\sqrt{w}$ is increased, this value being $2(\exp(-\ell\sqrt{w}b))$, where b, the imaginary part of $\lambda \ / \sqrt{w}$, is given by

$$b = \left(\frac{\mu_h}{|EI|}\right)^{1/4} \sin \ \Phi/4 \qquad (3.14)$$

Thus the four transmission ratios tend to:

$$|Y_{S\delta}|_a = 2 \ e^{-\ell \sqrt{w} \ b} \ |EI|^{1/4} \quad \mu_o^{3/4} \qquad (3.15)$$

$$|Y_{s\theta}|_a = |Y_{M\delta}|_a = 2 \ e^{-\ell \sqrt{w} \ b} \ |EI|^{1/2} \ \ \mu_o^{1/2} \qquad (3.16)$$

$$|Y_{M\theta}|_a = 2 \ e^{-\ell \sqrt{w} \ b} \ |EI|^{3/4} \quad \mu_o^{1/4} \qquad (3.17)$$

All the curves on Figures 9 to 14 tend to straight lines of slope -b.log(e).

4- TRANSMISSION BY TORSION

Rotational vibrations of the input end of the hose about its axis cause torsional waves to be propagated along the hose, resulting in vibrating torques being exerted by the hose at its output end. The one dimensional wave equation

$$\frac{\partial^2 \Psi}{\partial t^2} = \frac{k_t}{J} \quad \frac{\partial^2 \Psi}{\partial x^2} \qquad (4.1)$$

describes the motion. Ψ is the rotation of the cross-section at distance x along the hose, J is the moment of inertia of unit length of the hose, and k_t is the torsional stiffness of unit length, taken to be complex to allow for the damping. For sinusoidal motion of all elements at a frequency w, Ψ is given by

$$\Psi = \mathrm{Re}\{ \ \overline{\Psi} \ e^{jwt} \} \qquad (4.2)$$

Combining equations 4.1 and 4.2 leads to the solution

$$\overline{\Psi} = C_5 e^{-\gamma_t x} + C_6 e^{\gamma_t x} \qquad (4.3)$$

for the variation of the complex amplitude along the hose. C_5 and C_6 are constants depending on the boundary conditions while γ_t is the complex propagation constant for torsional waves, given by

$$\gamma_t = jw \left(\frac{J}{k_t}\right)^{1/2} \qquad (4.4)$$

For given elastic properties, γ_t is seen to be proportional to frequency and hence it is convenient to define $\overline{\gamma}_t$ as $\gamma_{t/w}$.

Different hoses can be compared by comparing the ratio for each hose of torque by the hose at a fixed output end to the torsional motion at the input end. The end conditions $\overline{\Psi}=0$ at $x=0$ and $\overline{\Psi}=\overline{\Psi}_\ell$ at $x=\ell$ yield the values of constants. The torque Γ exerted at the fixed end is the value of $k_t\left(\frac{d\,\overline{\Psi}}{dx}\right)$ at $x=0$;

$$\Gamma = \frac{2k_t\,\overline{\gamma}_t w\,\overline{\Psi}_\ell}{e^{\overline{\gamma}_t w\ell} - e^{-\overline{\gamma}_t w\ell}} \qquad (4.5)$$

It can be seen that, for given hose properties, there is a unique relationship between the transmission ratio defined as torque exerted per input torsional velocity, and the product $w\ell$. This ratio is given by

$$|Y_{r\Psi}| = \left|\frac{\Gamma}{w\,\overline{\Psi}_l}\right| = \left|\frac{2k_t\,\overline{\gamma}_t}{e^{\overline{\gamma}_t w\ell} - e^{-\overline{\gamma}_t w\ell}}\right| \qquad (4.6)$$

Figures 13 to 16 show this ratio plotted against $w\ell$ for the two hose types previously considered. The values of the hose torsional wave properties used are shown in Table 1. The graphs show strongly resonant behaviour over most of the range of $w\ell$ covered. The eventual steady attenuation with $w\ell$ can be found by considering equation 4.6 when w becomes very large; $\exp(\overline{\gamma}_t w\ell)$ becomes negligible and we find

$$|Y_{r\Psi}|_a = 2\,(J|k_t|)^{1/2}\, e^{-w\ell\overline{\alpha}}\, t \qquad (4.7)$$

where $\bar{\alpha}_t$, the real part of γ_t , is

$$\bar{\alpha}_t = \left(\frac{J}{|k_t|}\right)^{1/2} \sin\left(\Phi_t/2\right) \tag{4.8}$$

$\tan \Phi_t$ being the loss factor for torsion. The final slopes of the lines on Figures 15 and 16 are then $-\bar{\alpha}_t \log(e)$.

5- RELATIVE IMPORTANCE OF TRANSMISSION MECHANISMS

In order to consider the relative importance of the various transmission mechanisms from a pump to the rest of a hydraulic circuit, it is necessary to have information on typical outputs from pumps. However, the wide diversity of pumps makes it difficult to obtain data which has any general validity. The difficulty is made worse by the fact that the method of supporting the pump and driving motor, and variations in outlet arrangements, also have a major influence on the vibration at the point where the upstream end of the hose is connected. It is therefore not possible to get more than likely relative orders of magnitude of the input motions and pressures. To this end, measurements have been made on several axial piston and external gear pumps. Two small accelerometers (mass 0.65 gr) were mounted on each of two small aluminium blocks which were fixed on opposite sides of the hose input end fitting. One accelerometer on each side was mounted axially and the other tangentially relative to the end fitting, so that the sums and differences of the outputs gave the axial and lateral translation of the fitting and the torsional and bending angular inputs to the hose. Mean delivery pressures were set at 50 bar. Typical pressure ripple outputs were already available [10] [11] ; these have been supplemented with further measurements using a piezoelectric pressure transducer.

Certain common features can be discerned in the results obtained for most or all of the pumps and these are described below.

The frequancy content of the acceleration signals was obtained up to 4 kHz by a Fourier Transform analyser. Over this range the level of random

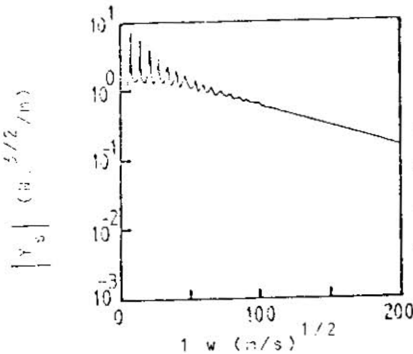

Fig.9-Lateral force/Input lateral motion Steel braid hose

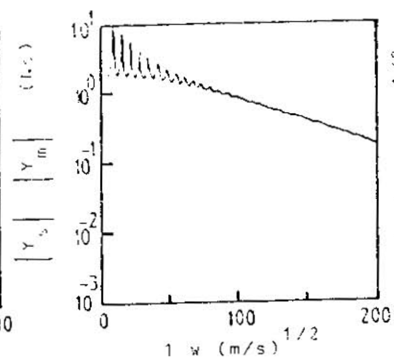

Fig.11-Input bending rational motion and bending moment/Input lateral motion Steel braid hose

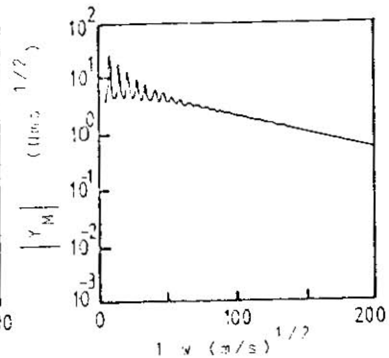

Fig.13-Bending moment/ Input banding rational motion Steel braid hose

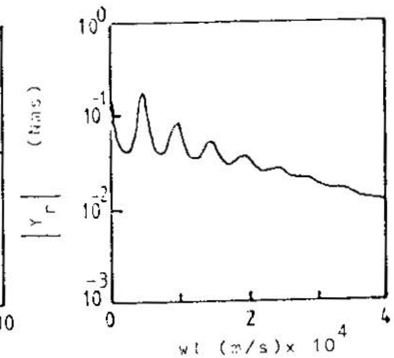

Fig.15-Torque/Input torsional velocity Steel braid hose Steel braid hose

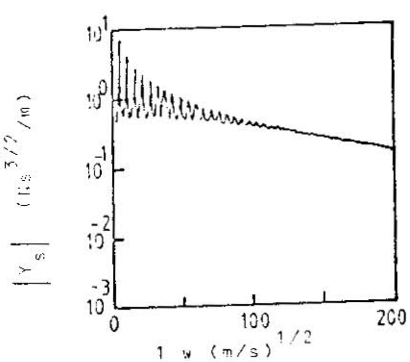

Fig.10-Lateral force/Input lateral motion Nylon braid hose

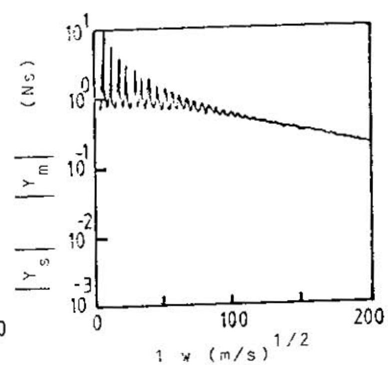

Fig.12-Lateral force Input bending rotational motion and bending moment/Input lateral motion Nylon braid hose

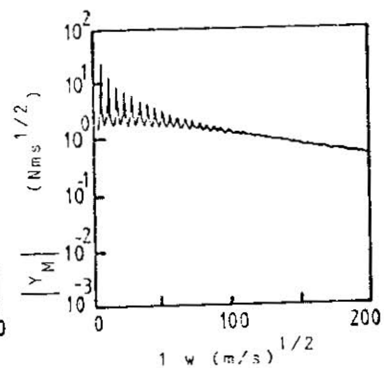

Fig.14-Bending moment/Input bending rotational motion

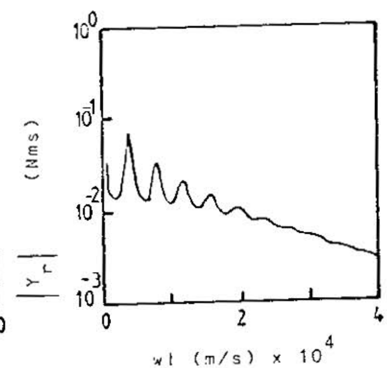

Fig.16-Torque/Input torsional velocity Nylon braid hose

motion was very small; the spectra consisted of discrete lines at multiples of the pump shaft revolution frequency which in all cases was close to 25 Hz. Components at multiples of the pumping frequency tended to be the highest, but this was not invariably so. Components at other multiples of shaft speed were in cases of significant amlitude and in some cases they were higher than the pumping harmonics. In general the highest velocity amplitudes were at the lower frequencies but for the axial piston pumps particularly, significant amlitudes occurred up into the region of 3 kHz. A test on one axial piston pump showed that vibration was greatly reduced when the mean delivery pressure was reduced to zero.

Frequency analysis of pressures over the same range indicates that only multiples of the pumping frequency are significant.For gear pumps it has been shown [10][12] that amplitudes of successive harmonic components decrease rapidly, whereas axial piston pumps generate substantial higher frequency components.

Direct comparison of lateral and axial vibration at the point where the outlet hose is connected to the pump shows that at lower frequencies, up to about 1 kHz, axial motion tends, on the whole, to be somewhat greater than lateral. At higher frequencies, 2 to 3 kHz, the opposite is true. There is also some tendency at higher frequencies for torsional motion to be greater than bending angular inputs. These differences may be the result of the length of cantilevered pipe and fittings between the body of the pump and the hose. The minimum length which this can have is that of the hose end fitting but it could be much longer. For purpose of comparison, however, it seems reasonable to assume the input motion to a hose to have no preferred orientation, and to take a typical figure for the ratio of angular to linear motion of 30 rad./m, bearing in mind that there are wide variations between pumps and between different frequency components.

Looking in the same way at typical levels of output pressure from pumps without any special pressure ripple reducing mechanisms, feeding into high impedance circuits, it is possible to obtain a typical order of magnitude for the ratio of pressure amplitude at a pump outlet to the velocity of the hose end fitting at that point. The results referred to above suggest about 10^8 Ns/m^3 over a wide range of frequency, but it must again be remembered that this will vary widely from pump to pump and between individual harmonics.

It is now possible to compare the relative importance of the various transmission mechanisms from the pump to the rest of the hydraulic system on the basis of pressure, force and moment at the termination of the hose. This can be done by using the relative pump outputs given above, together with the data on transmission ratios given if Figures 1 to 16.

In order to simplify comparison a hose length of 2 m is assumed, and a frequency of 1600 Hz. The lateral velocities at the pump end of the hose are taken to have an arbitrary reference level of 1 mm/s. The input angular velocities in torsion and bending, and the pressure amplitude at the pump are then taken to be those given by the typical ratios previously mentioned. Combining these with the transmission ratios for the hoses given the pressures, forces and moments shown in Table 2 at the hose termination.

The particular length and frequency chosen might be close to a condition of resonance in some cases and close to antiresonance in others. To avoid obtaining misleading values as a result of this, the transmission ratios used correspond to mean lines drawn throught the resonant fluctuations shown in Figures 1 to 16. The high attenuation of bending waves means that resonant fluctuations in the corresponding transmission ratios become unimportant above comparatively short hose lengths and frequencies. Most of the other transmission ratio plots show significant resonant behaviour over the whole of the plotted range. In the case of longitudinal waves, this can be the result of interference between the two forward waves as well as interference between forward and reflected waves.

The striking feature of Table 2 is the importance of the pump fluctuation. This has by far the greatest influence on the output pressure amplitude, which would be expected, but it also causes the hose to exert the greatest force on the subsequent circuit. The force, compounded of tension in the hose wall and the fluid pressure acting over the bore area, is much greater than that caused by an axial motion input to the over bore area, is much greater than that caused by an axial motion input to the hose, which in turn is substantially greater than the lateral force produced by a lateral motion input for the steel reinforced hose. Reduction of pressure fluctuations therefore merits most attention. It can be seen from Figure 1 that the steel reinforced hose does not give much attenuation. The type of hose which is most beneficial in this respect is the one reinforced with nylon. Nylon hoses capable of higher mean working pressures could not be expected to give as much attenuation as that plotted in Figure 2 but measurements have shown them to be much better than steel reinforced hoses for the same pressure rating [13] . The benefit of nylon hoses for reducing fluid borne noise has been reported previously [9][14].

Table-2: Force, moments and pressure at the output end of 2 m of hose, based on transmission ratios at 1600 Hz taken from smoothed versions of the curves shown in figures 1 to 16, and assuming the following input amplitudes:linear velocity (arbitrary ref.) 1 mm/s, angular velocity 0.03 rad/s, pressure 10^5 N/m^2.

		Type of Hose	
		Double Braid Steel	Double Braid Nylon
Force (Newtons) caused by	axial input	0.11	0.018
	lateral input	0.011	0.015
	bending angular input	0.0066	0.0074
	pressure input	1.9	2.1
Moment (Nm) caused by	lateral input	0.00022	0.00025
	bending angular input	0.00013	0.00012
	torsional angular input	0.00089	0.00028
Pressure (N/m^2) caused by	axial input	310	93
	pressure input	80 000	11 000

When examining Table 2 it must be remembered that pressure ripple can be greatly reduced by design features in pumps and by using hydraulic silencers. Under these circumstances the pump body motion inputs, for which flexible hoses are the only remedy, assume greater importance.

For the typical ratio of rotational to translational motion assumed, it can be seen that rotation of the input end of the hose about a transverse axis, and lateral input motion, both have much the same effect on the force and moment at the termination. However, an axial input motion has a greater effect on the output force, and a torsional input rotation has a greater effect on the output moment.

As reported by Hughes and Sanders [14] the spiral wound construction gives very good attenuation of bending waves. It also effectively isolates the more significant axial input. It is, however, the least effective in coping with torsion. These effects can be attributed to the low axial modulus and damping of spiral wound hose together with high circumferential shear stiffness in the wall.

It has been shown [15] that curvature of a hose centreline introduces some coupling between axial and bending transmission, but that this is usually only important in so far as it affects resonant behaviour at the lower frequencies. At higher frequencies, where the wavelengths are short compared with the radius of curvature, measurements have shown that the effect of curvature on all the transmission ratios is very small. In spiral wound hose some coupling has been observed between axial motion and torsion. This is assumed to be a static effect due to the inherent asymmetry of the construction. It has not been allowed for in the results that have been presented.

Another assumption which is implicit in the preceding discussion is that the fluid and mechanical impedances which the different types of hose present to the pump output are not so different and so significant that they have an important effect on the ranking of the hoses. The entry fluid impadances of long lengths of the steel braid and nylon hoses are similar, and that of the spiral hose is approximately half. This is a small difference compared with the gross differences in some of the transmission ratios. Differences in mechanical impedance are likely to have even less importance except possibly in the lateral direction when a hose is connected to a fairly long cantilevered pipe.

The hose properties used to calculate the transmission ratios can only be regarded as typical values because in many cases there is an appreciable variation with vibration amplitude as well as with mean pressure and temperature. This is particularly true of loss factor, which has a major influence on transmission. For example the loss factor for the nylon hose in bending varied by a factor of nearly 3 as a result of a seven to one variation in amplitude. The corresponding variation for the steel braid hose was approximately $1\frac{1}{2}$. Mean pressure had a considerable stiffening effect on the steel braided hose as discussed by Sanders and Hughes [16].

The general conclusions drawn in the foregoing discussion have been qualitatively supported by measurements at the termination of 4m lengths of the two types of hose when connected to an axial piston pump on a power pack. The only discrepancy of any significance was that the output pressure fluctuations produced by an input axial velocity were higher than expected. This may have been due to the end conditions differing somewhat from those assumed in the transmission ratio calculations.

The importance of pressure fluctuations from a pump compared with the mechanical vibration given to the hose has been demonstrated by Lipscombe [17] using a gear pump in which the normal pressure ripple could be doubled or virtually eliminated.

6- CONCLUSIONS

Within the frequency range examined, up to 4 kHz, the vibration generated at the outputs of several axial piston and external gear pumps has been found to be almost entirely at multiples of the revolution frequency, with multiples of the piston or gear tooth frequency often, but not invariably, giving the highest levels. Pressure fluctuations tend to be only at pumping harmonics. Although the characteristics of the vibrations varied somewhat between axial and lateral directions, it is considered reasonable to neglect this when drawing conclusions for pumps in general. Typical orders of magnitude have been presented for the ratio of angular motion to linear motion, and for the ratio of pressure amlitude to linear velocity amplitude, namely 30 radians per metre and 10^8 Ns/m^3. They appear to be reasonable values for the whole frequency range.

Using dynamic properties measured by resonance and natural frequency tests, transmission ratios have been presented for representative steel reinforced hose of double braid construction, and for a medium pressure hose with nylon braid and lining. These take the form of amplitudes of fluid pressure, axial force, shear force, bending moment and torsional moment at the termination of the hose, as ratios of the relevant input amplitudes from the pump, i.e. pressure, axial and lateral motion, and bending and torsional motion. Combining these transmission ratios with the relative pump outputs enables the following conclusions to be drawn.

Both the pressure and force amplitudes at the termination of a hose are likely to be dominated by the pressure amplitude produced by the pump, rather than by its vibratory motion, unless special pressure ripple reducing techniques are employed.

An axial vibration input to a hose will normally produce more force at the termination than a lateral input, although for the naylon hose there was not much difference. The spiral wound hose gave very high attenuation of lateral and bending input, and also gave good attenuation of axial input.

The most important angular input is torsion, particularly with the spiral wound hose which gave poor torsional attenuation.

Pressure fluctuations and also torsion were attenuated best by the nylon hose.

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