EFFECT OF DYNAMIC PROPERTIES ATTRIBUTABLE TO HYDRAULIC LINES ONTO OPERATION OF AVIONIC HYDRAULIC DRIVE SYSTEM

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Abstract

The present paper is dedicated to discuss how dynamic parameters of a hydraulic line intended to transfer hydraulic power from a source of hydraulic power (a hydraulic pump) to an actuating device (a hydraulic motor) can affect operation of an avionic hydraulic drive. Avionic hydraulic drives are operated with fast-varying waveforms of flow intensities and pressures. It is why analysis of most hydraulic drive systems must take account of compressibility of working fluid and elasticity of hydraulic lines. It leads to a wave model for propagation of energy variation down a hydraulic line (a pipe). The paper deals with flow of a compressible liquid via a hydraulic line with flexible walls. Theoretical deliberations were successfully verified by experimental research studies that were carried out on a dedicated test bench, therefore the presented amplitude vs. frequency characteristic curves of the hydraulic line could be obtained from both theoretical considerations and experimental research studied. The experimental research studies were carried out for hydraulic line terminated with a fixed flow restrictor alone and a flow restrictor combined with a hydraulic accumulator as well as for a hydraulic line supplied from a source of constant pressure and terminated with a valve with adjustable flow together with a hydraulic accumulator. The theoretical calculations for the amplitude vs. frequency characteristic curve are based on a model for a hydraulic line with its resistance depending on frequency.

Keywords: *fluid power transmission, pump delivery, volume flow (rate), hydrostatic (forcing) pressure, compressibility effect, viscous friction effect*

1. Introduction

Avionic hydraulic drive system subject to various input forces, usually originated from variation of external loads or operation of steering devices and equipment dedicated for generation of hydraulic power. A hydraulic line can be considered as a rigid module capable to transfer hydraulic power with no distortions is justified only in cases when frequency values of input functions are low. For aircraft driving systems, particular in aircraft steering assist systems, values of flow intensities and pressure subject to rapid changes in time. It is why for supplying lines with their lengths about 1 m one has to take account of compressibility of working liquids and elasticity of hydraulic lines (pipelines), which leads to a wave model for propagation of energy variation down a hydraulic line. It is extremely important to consider frequencies of input impacts that may cause phenomena when hydraulic resonance or a standing wave occurs in a supplying pipeline. Taking account of wave phenomena that may occur in a hydraulic line makes it possible to precisely predict behaviour of the driving system under various operating conditions, different from the ones when assumption of the perfect stiffness of the line is justified. Such phenomena as hydraulic resonance, momentary overloads or undesired vibrations can only be explained when the limited propagation speed of pressure changes down hydraulic lines is taken into account.

In former studies [1-4] the author presented description of dynamic properties attributable to hydraulic lines of avionic hydraulic driving system, where the line is intended to transfer hydraulic power from the source of hydraulic power (a hydraulic pump) to the actuating device (a hydraulic motor). The hydraulic line is considered as a four-terminal hydraulic network with two inputs and two outputs, where the network is described by its specific transmittance matrix that can be easily introduced to flowchart diagrams that describe dynamic properties of the system. The hydraulic line is characterized by means of components of hydraulic impedance (complex resistance), i.e. serial impedance that includes specific inertance and resistance per unit length, where these two parameters reflect the effect of inertia and viscous friction of liquid. The subsequent parameters include shunting admittance of liquid per unit length (reflecting the circuit capacitance) that takes account of the liquid compressibility. Therefore two basic parameters are obtained, suitable to characterize models of hydraulic lines with distributed parameters: the propagation operator and the characteristic impedance. The first parameter defines the time of delays in transmission of signals down a hydraulic line as well as attenuation and dissipation of waveforms representing pressure and flow rate, whereas the second one is the internal impedance of the line as seen from the side of the line load. Theoretical analyses already completed by the author have demonstrated that among models with distributed parameters it is the model of a hydraulic line with variable resistance that should dynamic properties of hydraulic circuits in the best way. However, verification of the foregoing hypothesis needs to perform a comparison between results from theoretical computations and the ones obtained from experimental researches. Experiments had to be carried out for hydraulic lines terminated with use of a permanent flow restrictor alone and a flow restrictor combined with a hydraulic accumulator as well as for a hydraulic line supplied from a source of constant pressure and terminated with a valve with adjustable variable cross-section or with a valve combined with a hydraulic accumulator.

2. Flow of compressible liquid down a hydraulic line with flexible walls

In [1, 2] differential equations are derived for a hydraulic line with variable resistance. These equations are expressed in the following matrix form:

$$\frac{\partial}{\partial x}\begin{bmatrix} P(x,s) \\ Q(x,s) \end{bmatrix} = -\begin{bmatrix} 0 & Z(s) \\ Y(s) & 0 \end{bmatrix}\begin{bmatrix} P(x,s) \\ Q(x,s) \end{bmatrix}, \tag{1}$$

where:

P(x,s) - Laplace transform for instantaneous pressure in a hydraulic line,
Q(x,s) - Laplace transform for the instantaneous flow rate in a hydraulic line,
Z(s) - serial impedance per unit length of the line,
Y(s) - shunting admittance per unit length of the line.

When an input function is a sine waveform (periodical variations) with the frequency of ω, such a type of input impacts is the most hazardous for avionic hydraulic drive system. For such a case the waveforms for pressures and flows can be obtained by the substitution $s = j\omega$. The solution of the equation (1) on the plane of complex variables can be written as the following expression:

$$\begin{bmatrix} P(0,s) \\ Q(0,s) \end{bmatrix} = \begin{bmatrix} ch\Gamma(s) & Z_c(s) sh\Gamma(s) \\ \frac{1}{Z(s)} sh\Gamma(s) & ch\Gamma(s) \end{bmatrix}\begin{bmatrix} P(l,s) \\ Q(l,s) \end{bmatrix}. \tag{2}$$

where:

$$\Gamma(s) = \frac{T_0 s}{\sqrt{1 - \frac{2}{j\sqrt{\frac{s}{\omega_0}}} \frac{J_1\left(j\sqrt{\frac{s}{\omega_0}}\right)}{J_0\left(j\sqrt{\frac{s}{\omega_0}}\right)}}}$$

is the propagation operator, whilst:

$$Z_c(s)=\frac{Z_{c0}}{\sqrt{1-\frac{2}{j\sqrt{\frac{s}{\omega_0}}}\frac{J_1\left(j\sqrt{\frac{s}{\omega_0}}\right)}{J_0\left(j\sqrt{\frac{s}{\omega_0}}\right)}}}$$

is the wave impedance,
where:

$$T_0=l\sqrt{L_0C_0}=\frac{l}{a}\ ,$$

$$Z_{c0}=\sqrt{\frac{L_0}{C_0}}=\frac{a\,\bar{\rho}}{\pi r_w^2}\ ,$$

where:
T_0 - standard (rated) delay time for a wave that is propagated along the hydraulic line,
J_0, J_1 - the Bessel function for the first kind, rank 0, 1,
ω_0 - characteristic frequency of the hydraulic line,
Z_{c0} - rated characteristic impedance of the hydraulic line,
s - Laplace operator,
L_0 - inertance per unit length of the hydraulic line,
C_0 - capacitance per unit length of the hydraulic line,
l - total length of the hydraulic line,
a - sound velocity in a non-viscous liquid,
r_w - internal radius of the hydraulic pipeline (conduit).

Hydraulic lines are meant to connect generators (sources) of hydraulic power (a hydraulic pump) with a receiver of hydraulic power (a hydraulic motor) via various appliances designed for control, adjustments and protections. In order to easily formulate relationships that bind inlet and outlet pressured of the line with values of flow rate it is recommended to represent a hydraulic line as a four-terminal hydraulic network. The equation (2) enables to derive three expressions for transmittance of a line with the impedance load:

$$\frac{P(0,s)}{Q(0,s)}=\frac{1}{K_z}\frac{ch\Gamma(s)+K_zZ_c(s)sh\Gamma(s)}{ch\Gamma(s)+\frac{1}{K_zZ_c(s)}sh\Gamma(s)}\ , \tag{3}$$

$$\frac{P(l,s)}{P(0,s)}=\frac{1}{ch\Gamma(s)+K_zZ_c(s)sh\Gamma(s)}\ , \tag{4}$$

$$\frac{Q(l,s)}{Q(0,s)}=\frac{1}{ch\Gamma(s)+\frac{1}{K_zZ_c(s)}sh\Gamma(s)}\ , \tag{5}$$

where:
$P(0,s)$, $P(l,s)$ - Laplace transform for pressure at the inlet and outlet of the hydraulic line,
$Q(0,s)$, $Q(l,s)$ - Laplace transform for flow rate at the inlet and outlet of the hydraulic line,
K_z - equivalent conductivity of the flow restrictor,
Z_c - characteristic impedance of the hydraulic line.

3. Comparison of experimental amplitude-frequency response curves of the hydraulic line against the ones obtained from theoretical computations

Experimental research studies were carried out on a dedicated test bench provided with a hydraulic positive-displacement pump of the plunger type, hydraulic accumulators with capacity of 360 cm^3, an electrohydraulic servomechanism as well as with pressure gauges and centrifugalhead flow meters. Disturbances to flow rates and pressure values were introduced with use of an electrohydraulic servomechanism supplied with signals from the sine waveform generator with constant amplitude and adjustable frequency as well as of a valve with adjustable (variable) section installed at the pipeline output. The tests were performed on a steel pipeline with the diameter of 80 mm and length of 10 m. The hydraulic oil AeroShell Fluid 41 at temperature of 25ºC was used as a hydraulic fluid (kinematic viscosity of oil is 23ºCSt, specific gravity 0.85G/cm^3 and compression modulus 14000 kG/cm^2).

The test scope covered four configuration arrangements of the hydraulic line: (i) the line with a fixed flow restrictor at the line end, (ii) the line terminated with a flow restrictor and a hydraulic accumulator, (iii) the line supplied from a source of constant pressure and terminated with a valve with adjustable (variable) cross-section and (iv) the line supplied from a source of constant pressure and terminated with a valve with adjustable (variable) cross-section and a hydraulic accumulator. In the case (i) the flow of fluid was forced via a pipeline with a fixed flow restrictor at the line end, where the flow rate was 8 dm^3/min and the pressure at the line output was 5 MPa. For the case (ii) the flow of fluid was forced via a pipeline with a fixed flow restrictor at the line end shunted with a hydraulic accumulator, with the same flow rate of 8 dm^3/min and the pressure at the line output of 5 MPa. The gaseous part of the hydraulic accumulator was filled with nitrogen at the pressure of 2.0 MPa. For the cases (i) and (ii) the input flow was disturbed by sine waveforms. For the case (iii) the flow of fluid was forced by a source of constant pressure whilst the pipeline end was terminated with a valve with adjustable (variable) cross-section, where the flow rate via the pipeline was 8 dm^3/min whilst the pressure at the line output was 10 MPa. For the case (iv) the flow of fluid was forced by a source of constant pressure whilst the pipeline end was terminated with a valve with adjustable, variable cross-section and the valve was shunted with a hydraulic accumulator. The flow rate via the pipeline was 8 dm^3/min, whilst the pressure at the line output was 10 MPa. The gaseous part of the hydraulic accumulator was filled with nitrogen at the pressure of 5.0 MPa.

The acquired measurement data made it possible to determine the following characteristics: for the cases (i) and (ii) – the relationship between input and output pressured for the pipeline as well as between the input pressure of the pipeline and the flow rate at the line input. The experiments (iii) and (iv) were intended to find out the relationship between the output pressure of the line and the opening cross-section of the electrohydraulic servomechanism as well as between the flow rate at the line input and the opening cross-section of the servomechanism.

Figure 1-7 present amplitude-frequency response curves for the hydraulic line acquired from experiments and from theoretical calculations. The theoretical calculations for the amplitude-frequency response curves were carried out with use of equations (3), (4) and (5) for the model of the hydraulic line with the frequency-dependant hydraulic resistance. Fig. 1 and 2 present both

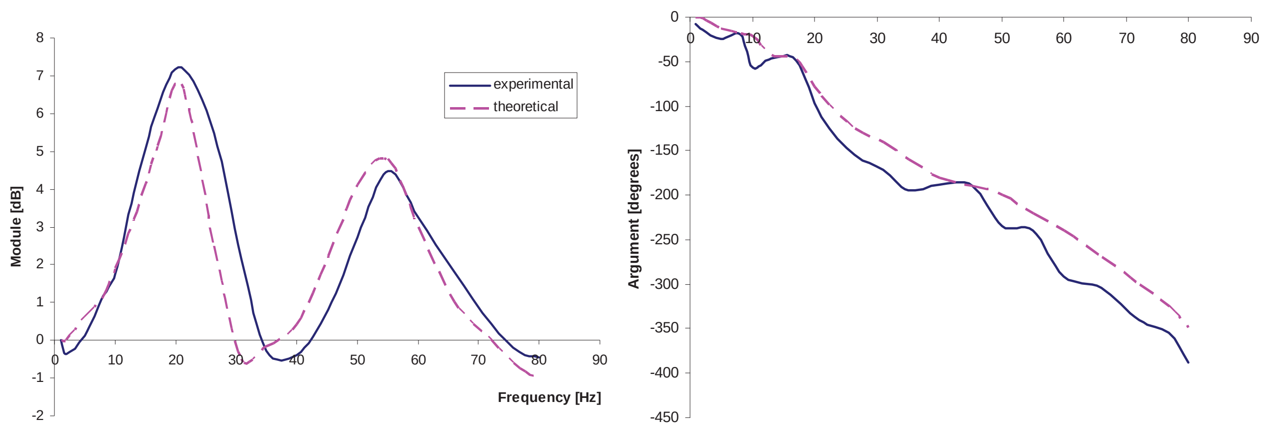

Fig. 1. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (i), with the transmittance function P(l,jω) / P(0, jω)

theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (i), with the transmittance functions $P(l,j\omega)\,/P(0,\,j\omega)$ and $P(0,j\omega)\,/\,Q(0,\,j\omega)$. Fig. 3 and 4 present both theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (ii), with the transmittance functions $P(l,j\omega)\,/P(0,\,j\omega)$ and $P(0,j\omega)\,/\,Q(0,\,j\omega)$. Fig. 5 and 6 present both theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (iii), with the transmittance functions $P(l,j\omega)\,/\,F(j\omega)$ and $Q(0,\,j\omega)\,/\,F(j\omega)$, where $F(j\omega)$ is the Laplace transform for deviations in throttling surface of the valve. Fig. 7 present both theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (iv), with the transmittance function $P(l,j\omega)\,/\,F(j\omega)$.

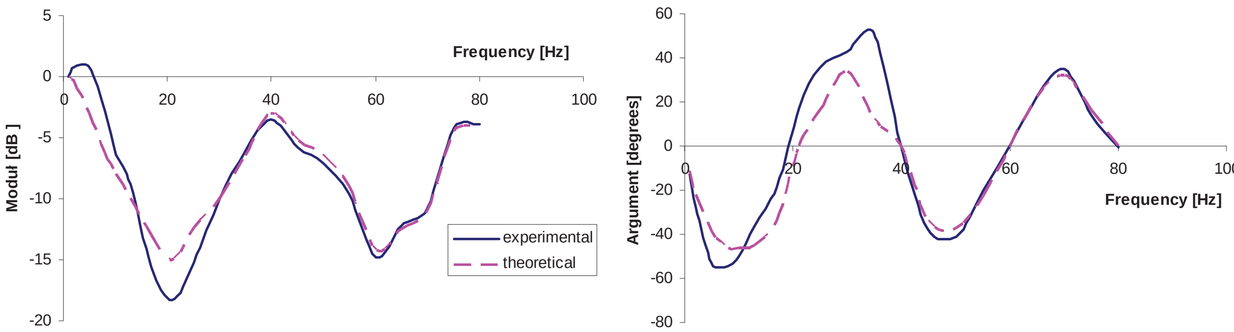

Fig. 2. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (i), with the transmittance function P(0,jω) / Q(0, jω)

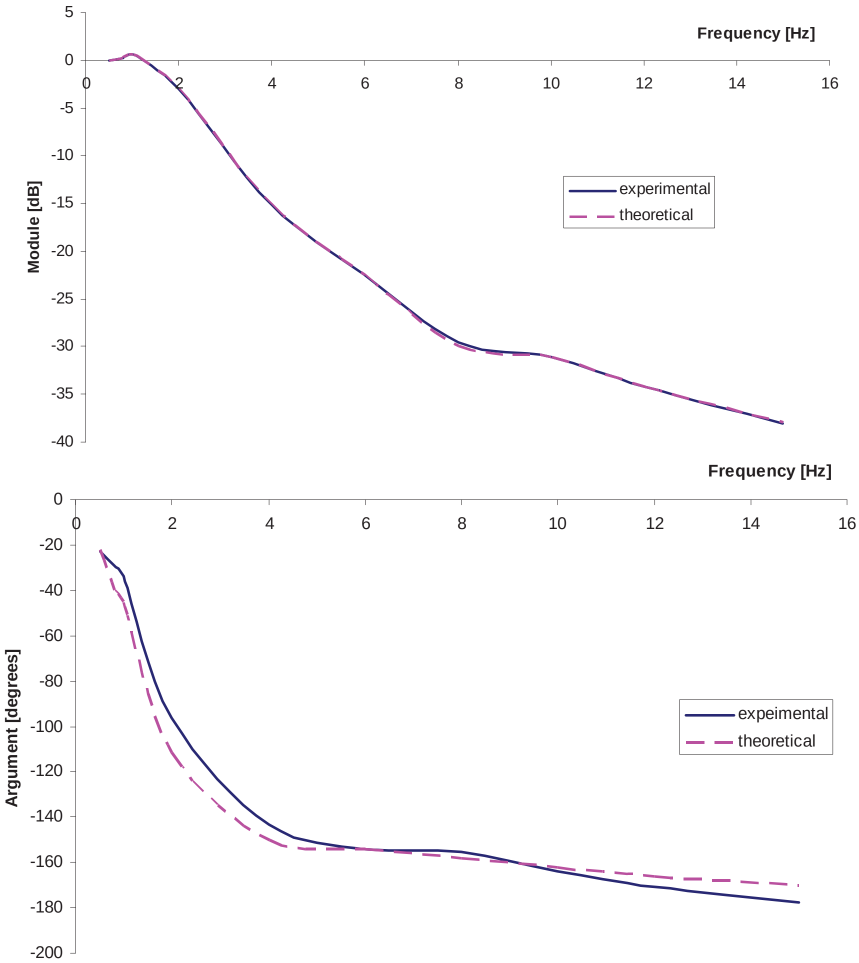

Fig. 3. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (ii), with the transmittance function P(l,jω) / P(0, jω)

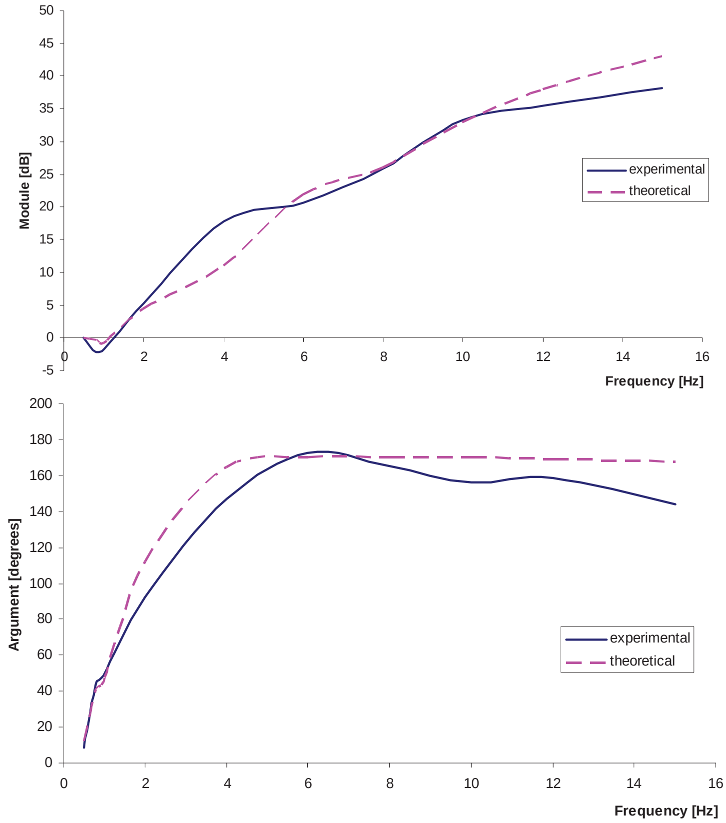

Fig. 4. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (ii), with the transmittance function P(0,jω) / Q(0, jω)

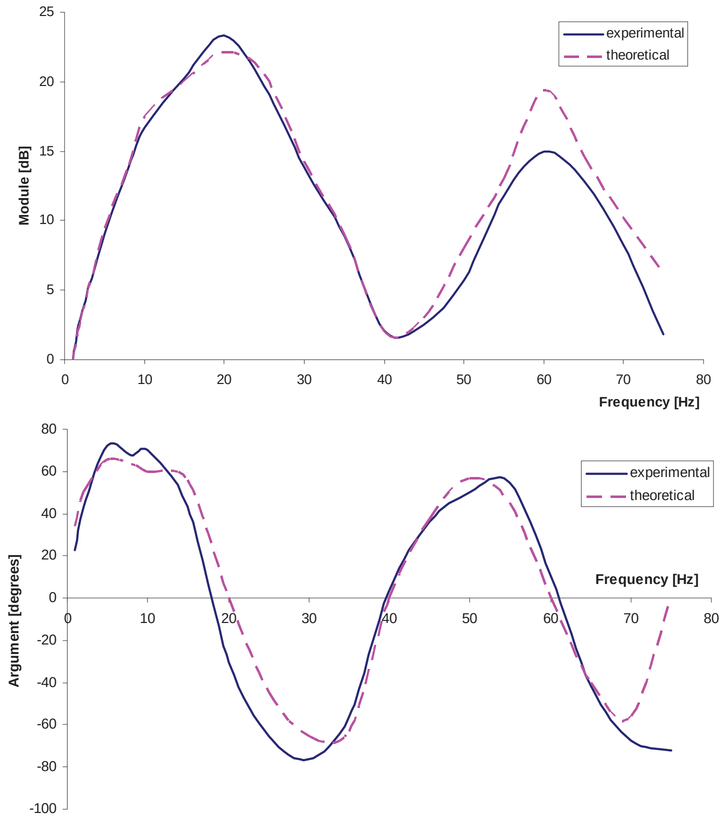

Fig. 5. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (iii), with the transmittance function P(l,jω) / F(jω)

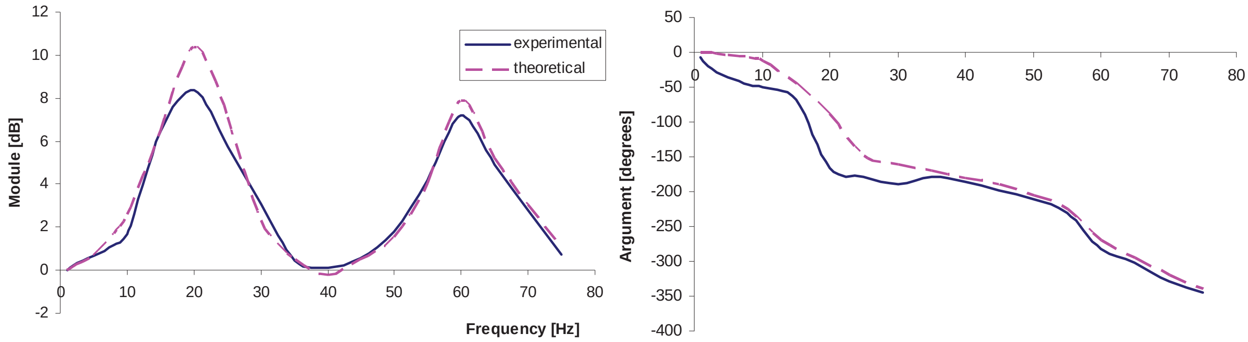

Fig. 6. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (iii), with the transmittance function P(0,jω) / F(jω)

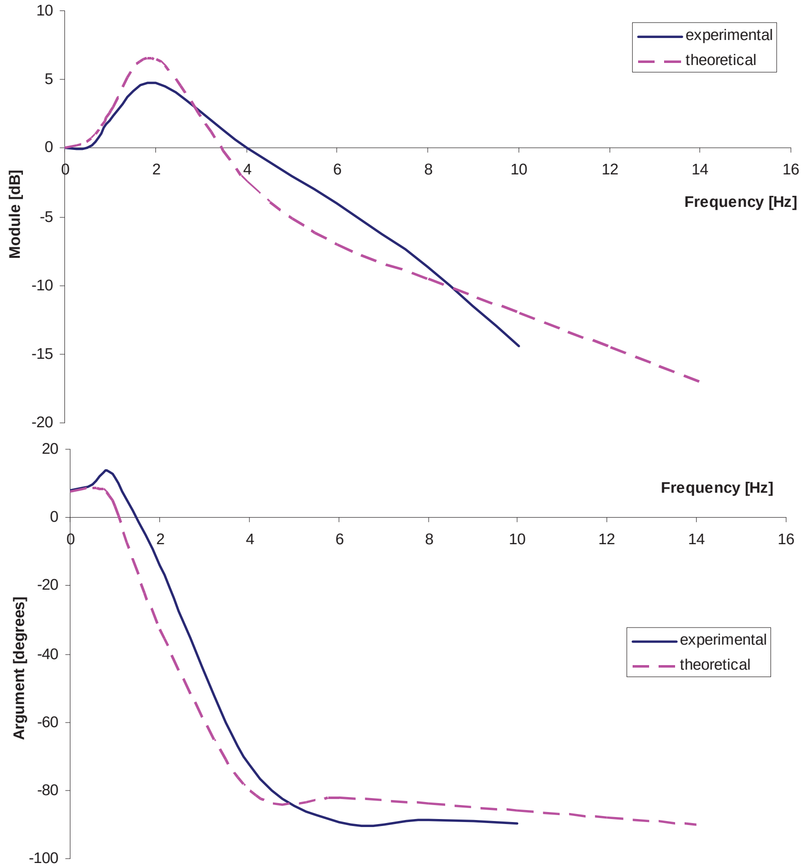

Fig. 7. Theoretical and experimental amplitude-frequency response curves of the investigated hydraulic line for the case (iv), with the transmittance function P(l,jω) / F(jω)

4. Recapitulation

Comparison between results from experimental research studies and output of computations that were carried out for the adopted model of a hydraulic line with variable resistance serve as the proof that the adopted model is good enough to depict dynamic properties of hydraulic lines. As the description of fluid flow via hydraulic lines employs a model with variable resistance, it is capable to take into account wave phenomena that may occur in the lines. Such phenomena, as hydraulic resonance, temporary overloads or undesired vibrations can be only explained when the assumption about limited propagation speed of pressure variations down hydraulic lines is made. Taking account of wave phenomena in hydraulic lines makes it possible to more precisely predict behaviour of hydraulic drive systems under various operating conditions.

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