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Time–Domain Models of the Pneumatic Transmission Lines for the 2 DoF (Two Degree of Freedom) Model Gantry Robot

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ABSTRACT: **This paper presents a theoretical and experimental method for analyzing pneumatic transmission lines in the time-domain for a model pneumatic gantry robot. The objective is to develop pipeline simulators that can be implemented for modeling of fluid servo systems.**

After describe the model, the paper illustrates the experimental tests carried out on several different types of lines, and comparing result with those of the corresponding time-domain simulations.

I. INTRODUCTION

The dynamic performance of a pneumatic servo actuator is typically defined in terms of valve response speed, actuator natural frequency and static loop gain. In most application, little attention is devoted to the lines in this respect, as their dynamics involve frequencies that are far higher than those of the other system components.

In certain cases, however, these considerations are not applicable, e.g., when the device which regulates fluid power – be it a servovalve or a servomotor-pump unit- is located at a significant distance from the device which transforms mechanical power, i.e., the motor. In these cases, long transmission lines have non-negligible effects on the dynamic behavior of fluid servosystems. Other such situations arise when potential vibration and noise transmission between the regulation device and the motor must be limited, and it is thus necessary to assess the effect of the transmission lines in detail [1]. This is the case when the regulating device, rather than being a continuous valve such as a proportional valve with analog spool positioning , is a digital valve whose delivery is regulated through on/off plunger modulation. As plunger opening and closing is practically instantaneous, pressure waves are generated in the line which can lead to system malfunction.

A schematic view of a conventional closed-loop position-controlled pneumatic servosystem with servovalve is shown in Fig. 1. Linearized studies of the dynamics of such systems generally take only the transfer functions of the servovalve and actuator (as well as of the control, obviously) into account, neglecting the effect of the lines.

In cases where this effect must be taken into account by developing a nonlinear model of the entire system, it is necessary to implement a time –domain model of the line.

Though lumped parameter models are traditionally used for this purpose, they yield acceptable result only in case involving analyses of low-dynamic systems.

As the dynamic behavior of transmission lines is described by partial differential equation better result are achieved with distributed parameter models [2] [3].

The examination shown in Fig.1,where the lines connect the servo valve output with the cylinder chambers, shows that each line expresses the relationship between input and output magnitudes, which for line 1 are respectively P_{11} $Q_{11}$and P_{21} Q_{21}. The four pole equation which models the line must be configured so that it can be integrated with the models of the upstream servo valve and the downstream cylinder. In particular, as the servo valve output normally consist of the flow rate delivered by this component, and the actuator's inputs are the flow rates entering the chambers, the line models inputs must be the upstream (or downstream) flow rate and the downstream (or upstream) pressure, while its outputs must be the downstream (or upstream) flow rate and the upstream (or downstream) pressure.

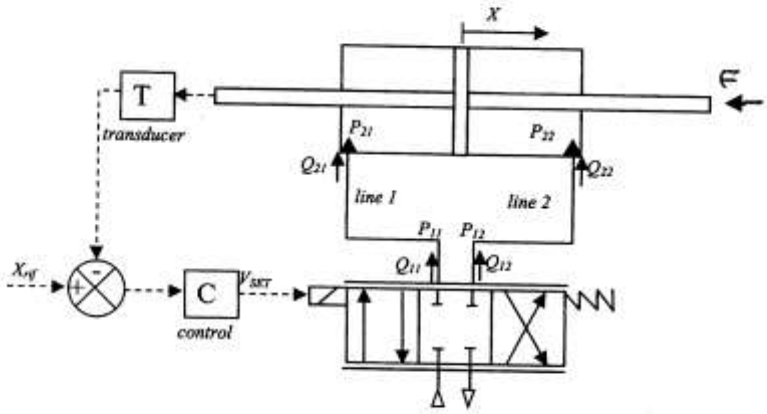

Fig. 1: Schematic view of a pneumatic servo system

Earlier investigations in this field addressed frequency-domain modeling of transmission lines with compressible fluid [4], and the influence of wall viscoelasticity on the dynamic behavior [5]. In time-domain simulations, however, the transfer function associated with the partial differential equations must be approximated.

Of the three most common approximation methods [6], the finite differences, characteristics [7], and modal methods, this investigation employs the latter in a variation formulation developed by Piche and Ellman [8].

The paper will first describe the transmission line model implemented in the Matlab-Simulink environment. After presenting the test bench used for experimental analyses of pneumatic lines, the paper will conclude with a comparison of several experimental and calculated curves for line behavior in the time domain.

II. THE MODEL

The resistance, inductance and capacitance properties of a fluid line are continuously distributed along the entire length of the line.

Let R', L' and C' be respectively the line's resistance, inductance and capacitance per unit length. In a line segment of length *dx* as shown in Fig. 2, the following will apply:

$$\begin{cases} -\dfrac{dp}{dx} = R'Q + L'\dfrac{dQ}{dt} \\ -\dfrac{dQ}{dx} = C'\dfrac{dp}{dt} \end{cases} \qquad [1]$$

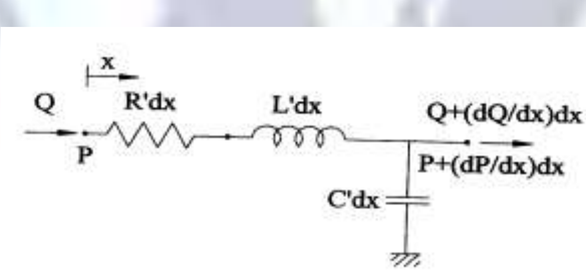

Fig. 2: Model of a line of length dx

Performing a Laplace transform gives:

$$\begin{cases} -\dfrac{dp(x,s)}{dx} = (R' + sL').Q(x,s) \\ -\dfrac{dQ(x,s)}{dx} = sC'.P(x,s) \end{cases} \qquad (2)$$

Considering average pipe friction for laminar flow due to fluid viscosity effects[6], and assuming Newtonian fluid, laminar fluid motion, axially symmetric flow, negligible redial motion, negligible nonlinear convective acceleration, and constant material properties the line resistance, inductance and capacitance per unit length are respectively:

$$R' = \frac{128}{\pi}\frac{\mu}{D^4} \quad (3)$$

$$L' = \frac{\rho}{A} \quad (4)$$

$$C'' = \frac{V}{l\beta} = \frac{V}{\rho c^2 l} \quad (5)$$

were β is the equivalent bulk modulus calculated as :

$$\beta = 1/(1/nP + D/hE) \quad (6)$$

Substituting (3) (4) and (5) in equation (2) gives:

$$\begin{cases} -\dfrac{dp(x,s)}{dx} = -\left(\dfrac{128\mu}{\pi D^4} + s\dfrac{\rho}{\pi r^2}\right).Q(x,s) \\ -\dfrac{dQ(x,s)}{dx} = -s\dfrac{V}{\rho l c^2}.P(x,s) \end{cases} \quad (7)$$

Equations (7) can be rewritten by introducing line impedance Z_0 and the propagation operator Γ, which is given respectively by:

$$Z_0 = \frac{\rho c}{\pi r^2} \quad (8)$$

$$r = \sqrt{\bar{s}^2 + \frac{8vl}{cr^2}\bar{s}} \quad (9)$$

as the normalized Laplace variable is denoted by $\bar{s}$:

$$\bar{s} = T.s \quad (10)$$

where T is the wave propagation time, and is defined as:

$$T = \frac{l}{c} \quad (11)$$

we thus have:

$$\begin{cases} \dfrac{dp(x,\bar{s})}{dx} = -\dfrac{Z_0}{l\bar{s}}\Gamma^2(\bar{s}).Q(x,\bar{s}) \\ \dfrac{dQ(x,\bar{s})}{dx} = -\dfrac{\bar{s}}{Z_0 l}.P(x,\bar{s}) \end{cases} \quad (12)$$

Elimination flow rate Q from equation (12) yields:

$$-l^2\frac{d^2p(x,\bar{s})}{dx^2} + \Gamma^2 P(x,\bar{s}) = 0 \quad (13)$$

To solve the second order differential equation (13), the system boundary conditions must be provided. In other world, at least two of the four variables (pressure P_1 and flow rate Q_1 upstream of the line, and downstream pressure P_2 and flow rate Q_2) must be known.

Of the three possible combinations of boundary conditions (two pressures, two flow rates or one pressure and one flow rate), this investigation used the *Robin boundary conditions*, i.e. the lines upstream flow and downstream pressure:

$$\frac{dp}{dx_{(x=0)}} = -\frac{Z_0\Gamma^2}{l\bar{s}}.Q_1, P_{(x=l)} = P_2 \quad (14)$$

Finding the solution to equation (13) using the robin bounding conditions and the variational method is equivalent to minimizing the functional [8].

$$I(P)=\frac{1}{2}\int_0^1 l^2\left[\left(\frac{dp^2}{dx}\right)+\Gamma^2 p^2\right]dx-lZ_o\frac{\Gamma^2}{\bar{s}}(Q_1P_1) \quad (15)$$

This problem was solved using the Ritz method, i.e., by finding an approximate solution for pressure behavior in the following from:

$$\tilde{P}=P_2+\sum_{i=l}^{n}p_i\varphi_i \quad (16)$$

Where the Ritz parameters p_i are constant with respect to the x coordinate and have the dimension of a pressure, while parameters ϕ_i are continuous functions of the x coordinate that can be differentiated into[0, l],and are dimensionless.

As the boundary condition to be respected is $P(x=l)=P_2$, it is necessary that $\phi_i(x=l)=0$

The trigonometric function:

$$\phi_i=\sin\frac{(2i-1)\pi(l-x)}{2l} \quad (17)$$

satisfies the conditions described above.

Substituting (16) and (17) in equation(15), and imposing stability conditions for functional (15), or in other words that $\frac{\partial I}{\partial p_i}=0[i=1,2,.....,n]$ yields the following values for the Ritz parameters:

$$p_i=-\left[(-1)^i\frac{\Gamma^2}{\bar{s}}Z_0Q_1+\frac{\Gamma^2}{\alpha_i}P_2\right]\frac{2}{\Gamma^2+\alpha_i^2} \quad (18)$$

given that:

$$\alpha_i=\frac{(2i-1)\pi}{2} \quad (19)$$

The Ritz coefficients thus identified enable us to determine pressures and flow rates along the line in the space and frequency domains using equation (16) and system (12), obtaining:

$$\begin{cases}P(x,\bar{s})=P_2+\sum_{i=l}^{n}P_i\sin\frac{(2i-1)\pi(l-x)}{2l}\\ Q(x,\bar{s})=-\frac{l\bar{s}}{Z_0\Gamma^2}\frac{dp}{dx}=\frac{\bar{s}}{Z_0\Gamma^2}\sum_{i=l}^{n}\alpha_ip_i\cos\frac{(2i-1)\pi(l-x)}{2l}\end{cases} \quad (20)$$

By establishing the conditions $p(x=l)=p_2$ and $Q(x=o)=Q_{1,}$ we can calculate the values of p_1 and Q_2 as follows:

$$\begin{cases}P_1=P(x,o)=P_2+\sum_{i=l}^{n}P_i\sin\frac{(2i-1)\pi}{2}=P_2+\sum_{i=l}^{n}(-1)^{i+l}p_i\\ Q_2=-Q(x=l)=-\frac{\bar{s}}{Z_o\Gamma^2}\sum_{i=l}^{n}\alpha_iP_i\end{cases} \quad (21)$$

Equations (21), which were obtained with four Ritz parameters (i = 1 to 4) ware implemented in the Matlab-Simulink environment, Fig. 3. The approximation of the frequency dependent propagation operator Γ and the steady state correction, both described in Pichė [8], were also introduced in the model.

When the downstream pressure and upstream flow are known, the Simulink block can calculate the two unknown magnitudes, upstream pressure and downstream flow rate. Given the line's symmetry, however, it is also possible to calculate downstream pressure and upstream flow rate in cases where the upstream pressure and downstream flow rate are known.

For the line with a downstream plug, the corresponding flow rate is zero. In the case of a line with a downstream nozzle, the corresponding flow rate is linked to down streams pressure by means of load impedance Z_L as follows:

$$Q_2 = \frac{P_2}{Z_L} \quad (22)$$

Parameters for the line include: line length and inside radius, wall thickness the Young's modulus of the line walls, the fluid's temperature, dynamic viscosity and mean pressure, and the poly tropic transformation exponent.

The model was tested for air lines as described in the following paragraphs.

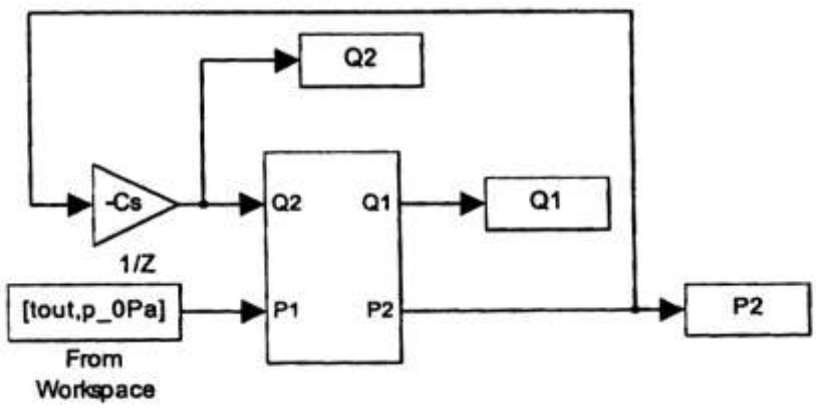

Fig. 3: Simulink model

III. EXPERIMENTAL TESTS

The model described above was validated on a test bench whereby the dynamic behavior of transmission lines of varying structure can be analyzed in the time frequency domains using different load impedances and pressure pulse-reducing where needed. The bench is capable of exciting lines of different geometry while plotting upstream pressure P_1 and down streams pressure P_2.

The transmission line test bench layout is illustrated in Fig. 4.

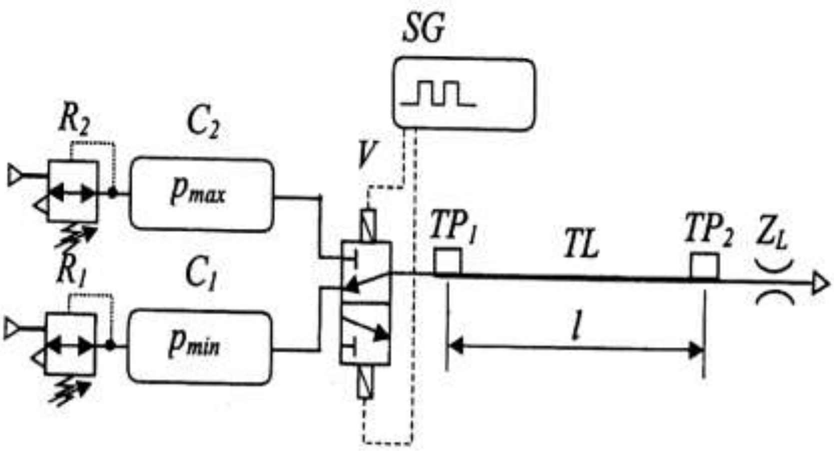

Fig. 4: Transmission line test bench

Tests were carried out with four line configurations, i.e., on a closed a line with load impedance Z_L established by means of a calibrated orifice, a line with two successive segments of different diameter, and a three-line branched system (T-filter).

An example of the experimental upstream pressure P_1 and downstream pressure P_2, recorded in a closed plastic line with length of 1 m described in Fig. 6, is shown in Fig. 5. It can be noticed that the line introduces a delay of above 3 ms that cannot be neglected in a pneumatic servo system study [9].

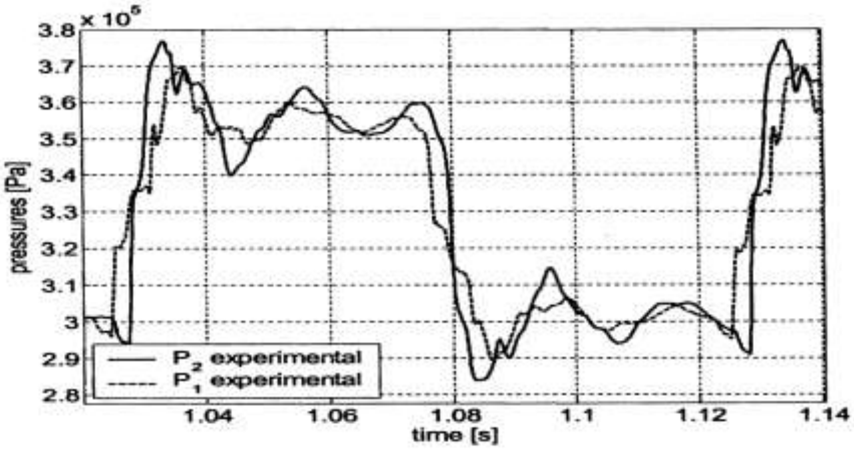

Fig. 5: Line 1 test; upstream pressure P_1 and downstream pr

IV. EXPERIMENTAL - NUMERICAL COMPARISON

The first test (line1 test) was carried out on a plugged plastic line (whose stiffness is 2 GPa) with length of 1 m, outside diameter of 6 mm inside diameter of 4 mm, supplied with an at a temperature of 307 K. The valve was switched at a frequency of 10 Hz, putting the line in communication with two reservoirs maintained at pressures of approximately 3.00 bar and 3.55 bar respectively. Table 1 shows the magnitudes used for simulation testing on line 1. A comparison between the calculated and experimental curves for downstream pressure is shown in Fig. 7.

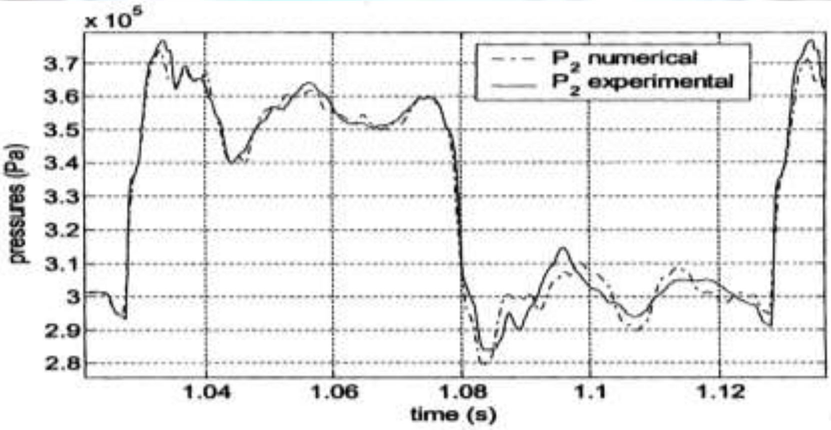

Fig. 7: Line 1 test

TABLE: 1. Model Parameters for Line 1 Testing

Properties			
Length	1 m	Temperature	307 k
Radium	2 mm	Mean Pressure	3.3 bar
Thickness	1 mm	Dynamic viscosity	$1.81.10^{-5}$ Pa's
Young's modulus	2 GPa	Poly tropic Exponent	1.25

Fig.8 shows the variation of flow with input pressure for line of length 5 m and diameter of 5.5 mm, this plotting is done by means of Automation Studio Softwarde V6.(Simulation Software).

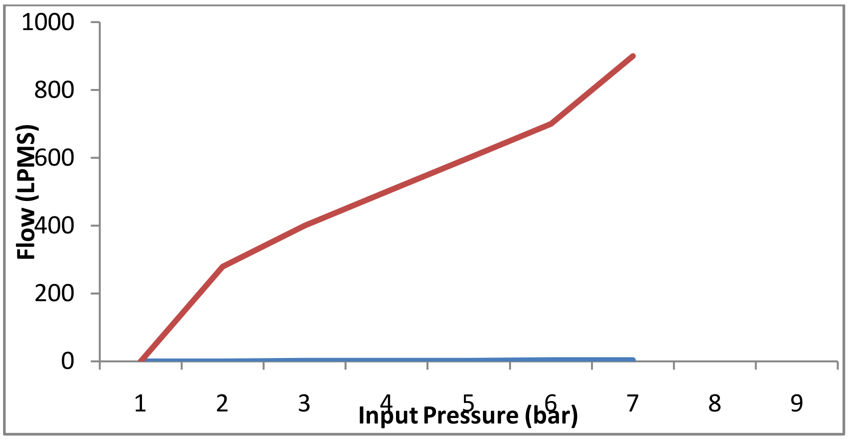

Fig.8: The variation of flow with input pressure for line of length = 5 m and diameter of 5.5 mm

V. CONCLUSIONS

This investigation implemented a line model as described in pichė [8], in the Matlab-Simulink. Experimental testing made it possible to validate the model for simulating the behavior of pneumatic lines in the time domain. Comparisons between theoretical and experimental result indicated that the model is an effective means of simulating pneumatic lines under different operating conditions.

Though this investigation is restricted to the line, the ultimate is to develop models of pneumatic lines that can be used in simulating the time–domain behavior of pneumatic servosystems as shown in Fig.1. Future work will thus focus on integrating the validated line model in a model of the complete servo system. Even at this early stage, however, the model can be said to be well suited to achieving this goal, as the layout of the test bench used in validating the model is that of control systems with digital valve modulation.

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