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SIMULATION MODEL OF A SHIPS PNEUMATIC VALVE

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ABSTRACT
In this paper movement of the one direct acting spring relief valve used in ship pneumatic system is analyzed. Using differential equations, for dynamic and thermodynamic subsystems, a continuous dynamic model is described in MATLAB Simulink. As simulation results, piston valve movement and control volume pressure changing, relevant for construction and exploitation of spring relief valve, is showed and analyzed.

1. INTRODUCTION
Importance of direct spring relief valve acting in ship pneumatic system is significant for corrected and safe system operation. It is confirmed that the valve structure has a significant influence on undesirable vibrations and noises appearance, caused by pressure and flow fluctuation, in hydraulics and pneumatics systems [4,5]. However, other causes that can influence, such as fluid compressibility and energy accumulation ability, are not sufficient researched. Knowledge of this manifestation and its causes are very important for proper relief spring valve construction and elements selection. In this paper directly operated relief spring valve in ship start air system [3] is analysed and valve piston movements is described using differential equations. Based on the continuous dynamics model, movement of valve piston is derived in MATLAB-Simulink program package. Valve piston movement and fluid state changing in real situation is analyzed, and results are showed by diagrams.

2. DESCRIPTION AND VALVE DESIGN
Directly operated relief spring valve is an important element of the ships pneumatic system, Figure 1. Compared with hydraulic systems, fluid in pneumatic systems has energy accumulation ability, which presents additional danger not only for regular and safe system work, but what is more dangerous for personal safety. While system pressure is below acceptable value, spring mechanism of valve keep valve piston on its seating (valve is closed). If system pressure rises above acceptable value, pressure force overcomes the spring force, and then the valve moves from its seating. At that moment fluid begins to flow from system through valve, causing the system pressure drop. As a consequence of pressure drop, the valve piston should begin movement toward its seating, causing a flow reduction through the valve. Due to this fact, the control volume pressure increased, valve piston moves from its seating again. Possible pressure and flow changes may have directly influence on piston valve movement to (and) cause vibration and noise during its operation. In order to research this undesirable appearance, it is necessary to make an analysis of fluid and valve piston interference during movement.

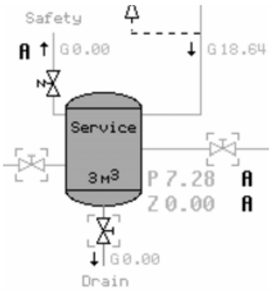

Figure 1. Part of the ship's pneumatic system

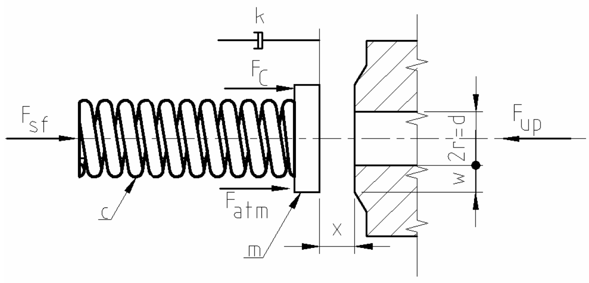

Figure 2. Back pressure valve dynamic model

3. MATHEMATICAL MODEL OF THE VALVE

The two subsystems is observed, mechanical which described movement of the valve piston and thermodynamically which described state of fluid in control volume.

3.1. Mechanical subsystem

Referring to Figure 2., valve piston movement differential equation can be written:

$$m \cdot \frac{d^2x}{dt^2} + k\frac{dx}{dt} + F_{sf} + F_C + F_{atm} - F_{up} = 0 \tag{1}$$

where is: m – moving mass [kg], x – valve displacement [m], k – dumping coefficient [N/(m/s)], F_{sf} – spring force [N], F_C – Coulomb friction force [N], F_{atm} – atmospheric pressure force [N], F_{up} – upstream pressure force [N].

The spring force which acts on the valve piston is:

$$F = c(x + x_0) \tag{2}$$

where is: c – spring rate [N/m], x_0 – precompressed spring length [m].

Coulomb friction force acts in direction opposite to that of the piston valve motion:

$$F_C = (F_s, F_d) \cdot \left(-sig\left(\frac{dx}{dt}\right)\right) \tag{3}$$

where is: F_s – static friction force [N], F_d – dynamic friction force [N].

3.2. Thermodynamically subsystem

Assuming that the fluid in system behaviour as a perfect gas, equation of state is written:

$$p_i\, V_i = m_i\, R\, T_i \tag{4}$$

where is: i – 1,2 for defined field (incoming pipe to the valve, control volume), p – pressure [Pa], V – control volume [m^3], m – fluid mass [kg] i T – fluid stagnation temperature [K].

Differentiating equation (4) and then rearranging with energy equation and the heat transfer [3], pressure changing equation is:

$$\frac{dp_i}{dt} = \kappa\frac{RT}{V_i}\frac{dm_i}{dt} - \kappa\frac{p_i}{V_i}\frac{V_i}{dt} \tag{5}$$

where is: dm – mass flow rate through defined field [kg/s].

The mass flow rate through defined field depends on pressure ratio [1,2], and in this case is:

$$\frac{dm_i}{dt} = \begin{cases} C_d A_i \sqrt{\frac{2\kappa}{R(\kappa-1)}} \frac{p_u}{\sqrt{T_u}} \left(\frac{p_d}{p_u}\right)^{\frac{1}{\kappa}} \sqrt{1-\left(\frac{p_d}{p_u}\right)^{\frac{\kappa-1}{\kappa}}} & \text{za} \left(\frac{p_d}{p_u}\right) > \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa\text{-}1}} \\ C_d A_i \sqrt{\frac{\kappa}{R\left(\frac{\kappa+1}{2}\right)^{\frac{\kappa+1}{\kappa-1}}}} \frac{p_u}{\sqrt{T_u}} & \text{za} \left(\frac{p_d}{p_u}\right) \leq \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa\text{-}1}} \end{cases} \tag{6}$$

where is: C_d – discharger coeficient, A – cross-sectional area of fluid flow [m^2], κ – ratio of specific heat, R – ideal gas constant [J/kg K], p_u – valve upstream pressure [Pa], p_d – valve downstream pressure [Pa], T_u – upstream stagnation temperature [K].

Static fluid force acting on piston valve specifies maximum system pressure and based on it spring constant is choose. The value of this force is:

$$F = \begin{cases} \pi r^2 \Delta p & \text{za x} = 0 \\ \pi (r+w)^2 \Delta p & \text{za x} > 0 \end{cases} \tag{7}$$

where is: r – upstream pipe radius [m] and w – land width of nozzle [m].

With static force which acts on valve, there is a flow force caused by fluid force. For a very small piston valve displacement, using Navier – Stoke's equation [1], flow force which acts on piston valve is:

$$F = \frac{\pi\left((r+w)^2 - r^2\right)}{2\ln\left(1+\left(\frac{w}{r}\right)\right)} \Delta p \,. \tag{8}$$

4. CONTINUOUS SIMULATION MODEL

Based on the differential equation of the valve motion and fluid state change, mathematical model in MATLAB-Simulink has been obtained. The model is shown in Figure 3.

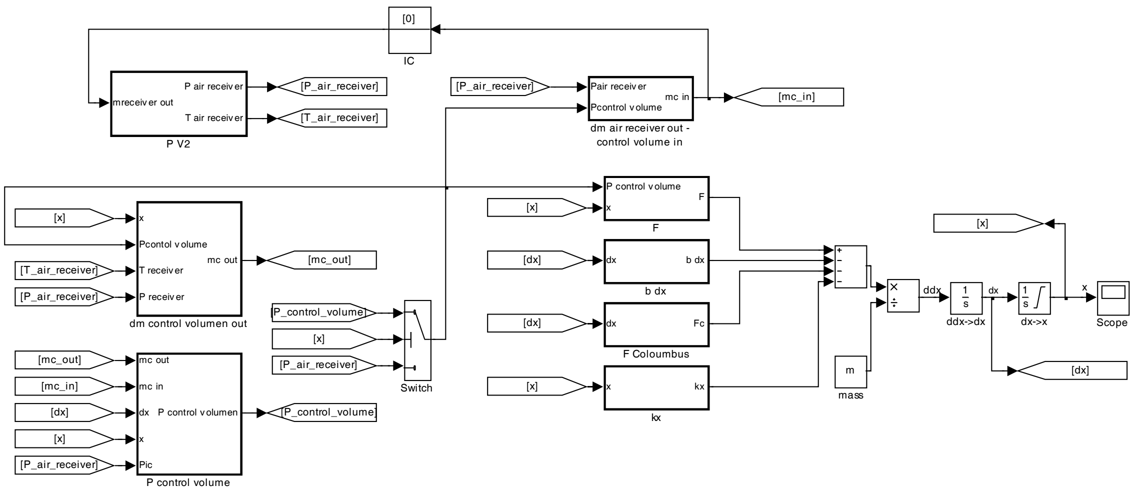

Figure 3. Simulation model of back pressure spring valve in MATLAB-Simulink

5. SIMULATION RESULTS AND CONCLUSION

Applying dynamic computer simulation on the relief valve, control volume pressure and valve displacement is observed. Analyses and comparisons of piston valve movement and control volume pressure changing are made for critical (opening) pressure of 710 kPa, varying spring rate and its precompression length. The results of simulations are shown in Figures 4, 5, 6 and 7.

It can be seen that in the pneumatic system the movement of the piston valve is stable, without huge oscillations, even at initial opening of the relief valve where the oscillations are significant, Figure 4. and Figure 5. On the other hand, control volume pressure at initial opening of valve oscillate, and after short period it is stabilized and decreasing until spring force become equal with pressure forces, Figure 6. and Figure 7. Accordingly, it can be concluded that the spring rate and its precompression are not too significant for stabile valve operation. Moreover, smaller spring rate and bigger precompression length gives quicker response, thereby quick establish of allowed system pressure. Furthermore, mentioned piston valve and fluid interference stabilize valve operation. The explanation for this can be that system pressure and the fluid energy also increases, during the time the valve was closed. In the moment when the valve has opened, the pressure of fluid energy drop, so that the difference of total energy which acts on piston valve is smaller. This energy change and portion energy transfer from fluid to valve spring causing valve operation stability, and conclusion can be made that the movement of the piston valve in pneumatic system, in comparison with hydraulic, is significantly stable, and vibrations and noise become negligible. It could be interested to give into the consideration an appliance of valve with gas chamber in ship hydraulic system, thus obtaining the vibrations and noise caused by possible malfunction of relief valve operation negligible.

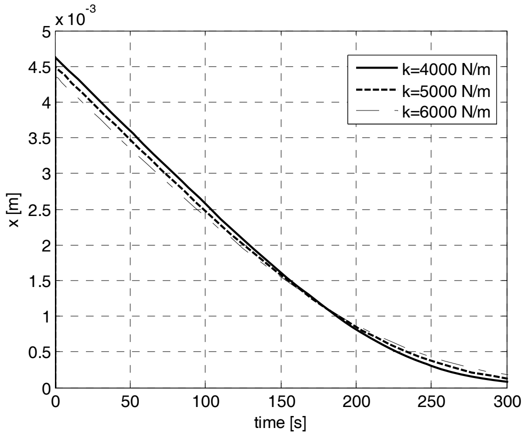

Figure 4. Valve displacements (t=0-300s)

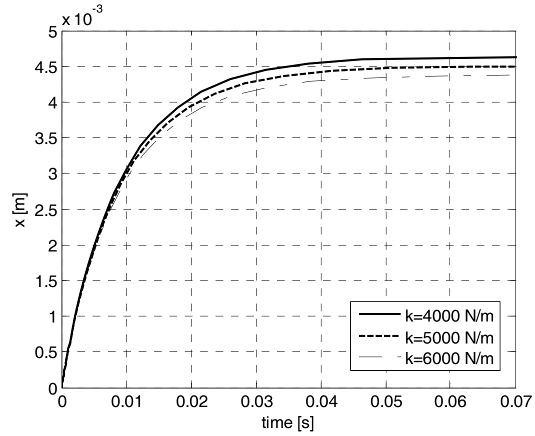

Figure 5. Valve displacements (t=0-0.07s)

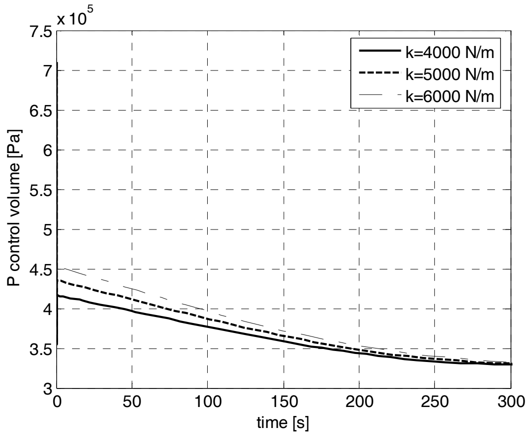

Figure 6. Control volume pressures (t=0-300s)

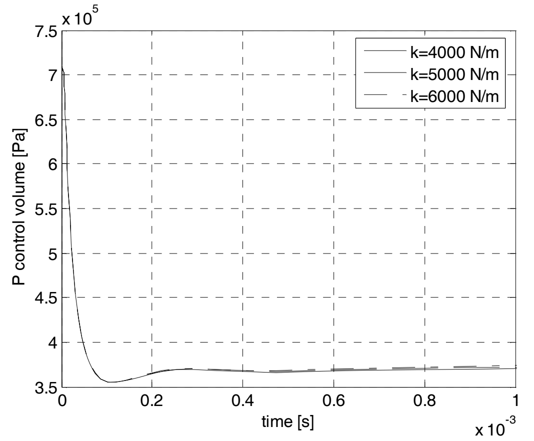

Figure 7. Control volume pressures (t=0-0.001s)

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