ANALELE UNIVERSITĂȚII
"EFTIMIE MURGU" REȘIȚA
ANUL XVIII, NR. 1, 2011, ISSN 1453 - 7397

Emil Zaev, Gerhard Rath, Hubert Kargl

Design of a Hydraulic Damper for Heavy Machinery

A hydraulic unit consisting of an accumulator as energy storage element and an orifice providing friction was designed to damp oscillations of a machine during operation. In the first step, a model for the gas spring was developed from the ideal gas laws for the dimensioning the elements. To model the gas process with a graphical simulation tool it is necessary to find a form of the gas law which can be integrated with a numerical solver, such as Tustin, Runge-Kutta, or other. For simulating the working condition, the model was refined using the van der Waals equations for real gas. A unified model representation was found to be applied for any arbitrary state change. Verifications were made with the help of special state changes, adiabatic and isothermal. After determining the dimensional parameters, which are the accumulator capacity and the orifice size, the operational and the limiting parameters were to be found. The working process of a damper includes the gas pre-charging to a predefined pressure, the nearly isothermal static loading process, and the adiabatic change during the dynamic operation.

Keywords: *Vibration damping, hydraulic damping, gas laws*

1. Introduction

Mining Machines, like the ALPINE Miner of the Sandvik company in Fig. 1, are subject to severe vibrations during their operation. Measures for damping are required. Goal of this project was to investigate the potential of damping oscillation using the hydraulic cylinders of the support jacks (Fig. 2). For this purpose a hydraulic storage element (accumulator), acting as a spring, and a throttling device to provide friction had to be designed. It was desired to test the effect of the system in a simulation model before construction. An existing complex mechanical simulation model of the machine had to be extended with a proper model of the proposed damping system. The main problem to be solved was to model the gas volume in the accumulator providing the elastic property, especially the thermal

behavior. Here the steps, which are essential for the design of the parameters and for the development of a dynamic simulation model, are presented.

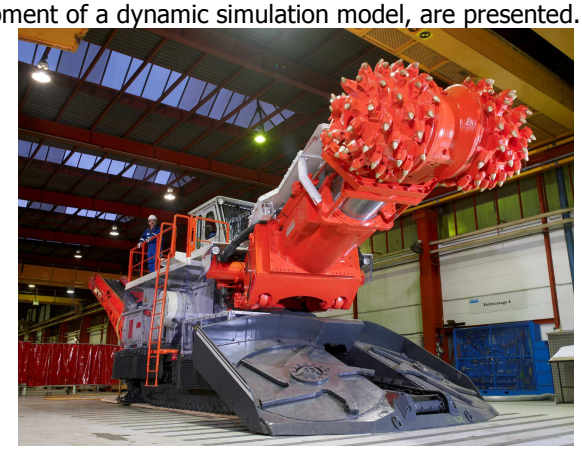

Figure 1. Mining Machine

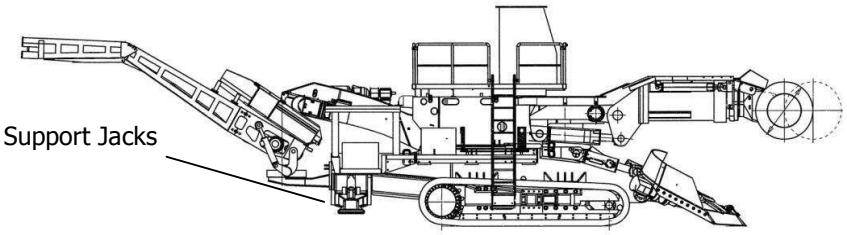

Figure 2. Mining machine with support jacks

Fig. 2 shows the position of the support jacks that are equipped with a hydraulic cylinder for motion. This system was extended with a hydraulic accumulator and an orifice as a throttling device. Fig. 3 displays the basic scheme of this equipment, which is the topic of the presented simulation work.

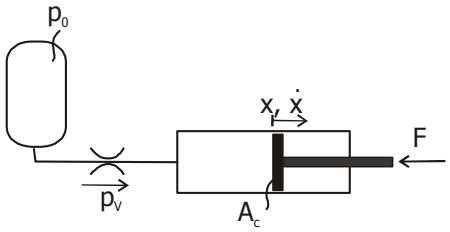

Figure 3. Basic damping circuit in support jacks

2. Nomenclature

p_{00}	Pre-charging pressure of the accumulator
V_{00}	Nominal volume of the accumulator
p_0	Cylinder pressure when the machine is supported during standstill
V_0	Volume (gas) when the machine is supported during standstill
κ	Adiabatic exponent κ=1.4 for nitrogen
A_c, A	Cylinder piston area and area of the gas boundary surface in accumulator
x,y	Displacement of the piston and displacement of the gas boundary
V,v	Gas volume and specific gas volume
p_N, p_v	Nominal pressure drop of the orifice and actual pressure loss
q_N	Nominal flow of the orifice at p_N
k_v	Constant of the orifice
F_s	Spring force of the cylinder (related with piston position)
F_r	Resistive friction force (related with piston speed)
R, R_i	Universal gas constant, specific gas constant
m, n	Amount of gas, mass or moles
T	Absolute temperature
Q,W,E	Heat transfer, mechanical work and energy

3. Hydraulic Accumulator

The hydraulic accumulator provides the spring property to the system. The first design usually is made using the ideal gas equation [4], starting from the ideal gas law

$$pV = mRT\ . \tag{1}$$

State changes of the ideal gas are described with the following equations [5]:

Isochoric state change

$$\frac{p}{T} = c_1 \tag{2}$$

Isothermal state change

$$pV = c_2 \tag{3}$$

Adiabatic state change

$$\begin{aligned} pV^{\kappa} &= c_3 \\ TV^{\kappa-1} &= c_4 \\ T^{\kappa}p^{1-\kappa} &= c_5 \end{aligned} \tag{4}$$

Before an accumulator can be used, a ***filling process*** is performed to preload the device to a certain pressure [6]. It involves an adiabatic expansion from the nitrogen gas bottle in combination with an adiabatic compression of the volume in the accumulator. From experience we know, that the pressure after preloading does not change much, consequently this complex process is not evaluated, and it is simply assumed to start with a given initial pressure p_{00} at a volume V_{00}.

When the machine is standing still and the support jacks are applied the pressure in the accumulator increases again. Temperature of the gas volume will increase also but has some time to come down again. So this is an ***isothermal process*** resulting in a pressure p_0, which is the static operating pressure. With Equ. 3 the volume of gas will be

$$V_0 = \frac{p_{00} V_{00}}{p_0} \ . \tag{5}$$

Oscillations occur when the machine starts working. Since changes are fast, this is an ***adiabatic process***. From Equ. 4 follows for actual the pressure:

$$\begin{aligned} p V^{\kappa} &= p_0 V_0^{\kappa} \\ p &= p_0 \frac{V_0^{\kappa}}{V^{\kappa}} \end{aligned} \tag{6}$$

If someone is interested in the resulting spring constant of the cylinder connected with the accumulator (Fig. 3) in a small neighbourhood of the working point, it can be obtained with

$$c = \frac{\mathrm{d}F}{\mathrm{d}x} = \frac{\mathrm{d}pA_c^2}{\mathrm{d}V} = -\kappa A_c^2 p_0 \frac{V_0^{\kappa}}{V^{\kappa+1}} \ . \tag{7}$$

For a simplified simulation the elastic reaction force of the cylinder piston to an external load only depends on the displacement x of the cylinder piston and follows also from Equ. 6.

$$F = pA_c = p_0 A_c \frac{V_0^{\kappa}}{\left(V_0 + xA_c\right)^{\kappa}} \tag{8}$$

4. Introduction of Friction with Orifice

The accumulator provides the elastic property for the damper. To dissipate kinetic energy to heat, a friction element is necessary. An orifice can be applied for this. The relationship between flow and pressure drop [6] is

$$\frac{q}{q_N} = \sqrt{\frac{p_v}{p_N}}$$
$$q = \frac{q_N}{\sqrt{p_N}}\sqrt{p_v} = k_v\sqrt{p_v} \quad , \tag{9}$$

The pressure loss including both directions of flow is

$$p_v = \mathrm{sgn}\left(\dot{x}\right)\left(\frac{A_c\dot{x}}{k_v}\right)^2 . \tag{10}$$

The reaction force of the support jack is composed of an elastic (spring) and a resistive part:

$$F = f\left(x,\dot{x}\right) = f_1\left(x\right) + f_2\left(\dot{x}\right) = F_s + F_r \tag{11}$$

This simple formalism was used for a first design of the damper. The friction force is proportional to the square of the speed. Using other devices than orifices can improve the heat dissipation [7].

5. Simulation Model for Accumulator with Ideal Gas

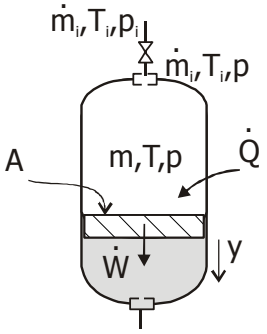

Figure 4. Accumulator as control volume

To simulate a system containing a gas volume it is necessary to find a form of the gas law that can be integrated by a solver. To the control volume (gas volume of the accumulator, Fig. 4) the First Law of Thermodynamics, which is the conservation of energy, is applied.

$$\begin{aligned} Q_{21} - W_{12} + E_{i21} &= E_2 - E_1 \\ \Delta Q - \Delta W + \Delta E_i &= \Delta E \end{aligned} \tag{12}$$

Q_{21} describes the heat exchange over the system surface during the change from state 1 to 2. The mechanical work delivered from the gas volume W_{12} is positive. It is composed of the mechanical 'shaft work' by the motion Δy and the flow work of the incoming gas with the specific volume v_i. The energy of the original volume changes from E_1 to E_2.

$$\Delta Q - p\,A\,\Delta y + p\,v_i\,\Delta m_i + u_i\,\Delta m_i = \Delta u\,m \tag{13}$$

Potential and kinetic energies are ignored, so the fluid energy E only consists of the internal energy. Without a chemical reaction, this energy is the thermal energy. With the definition for constant-volume specific heat

$$\left.\frac{\partial u}{\partial T}\right|_V = c_v \approx \frac{\Delta u}{\Delta T} \tag{14}$$

Equ. 13 becomes

$$\Delta Q - p\,A\,\Delta y + \Delta m_i\left(p\,v_i + u_i\right) = m\,c_v\,\Delta T\ . \tag{15}$$

With the enthalpy collecting the volume work and the internal energy of the incoming gas

$$h_i = u_i + p\,v_i\ . \tag{16}$$

This can be expressed with the specific heat at constant pressure

$$c_p = \left.\frac{\partial h}{\partial T}\right|_P\ . \tag{17}$$

Assuming that c_p is constant and integrating from T=0 to T_i, gives

$$h_i = c_p\,T_i\,, \tag{18}$$

inserting this into Equ. 15 and letting Δt -> 0 yields [8]

$$\dot{Q} - pA\,\dot{y} + \dot{m}_i\,c_p\,T_i = m_0\,c_v\,\dot{T}\ . \tag{19}$$

Solving for the temperature change

$$\dot{T} = \frac{1}{m\,c_v}\left(\dot{m}_i\,c_p\,T - p\,A\,\dot{y} + \dot{Q}\right) \tag{20}$$

gives a differential equation for the temperature T, which can be solved numerically without a singularity. The pressure of the volume can be found with Equ. 1, using the specific gas constant

$$p = \frac{R_i\,T\,m}{V}\ . \tag{21}$$

Fig. 5 shows the implementation of Equ. 19 to 21 in a graphical model. An introduction to modelling with MATLAB®/Simulink™ can be found in [9].

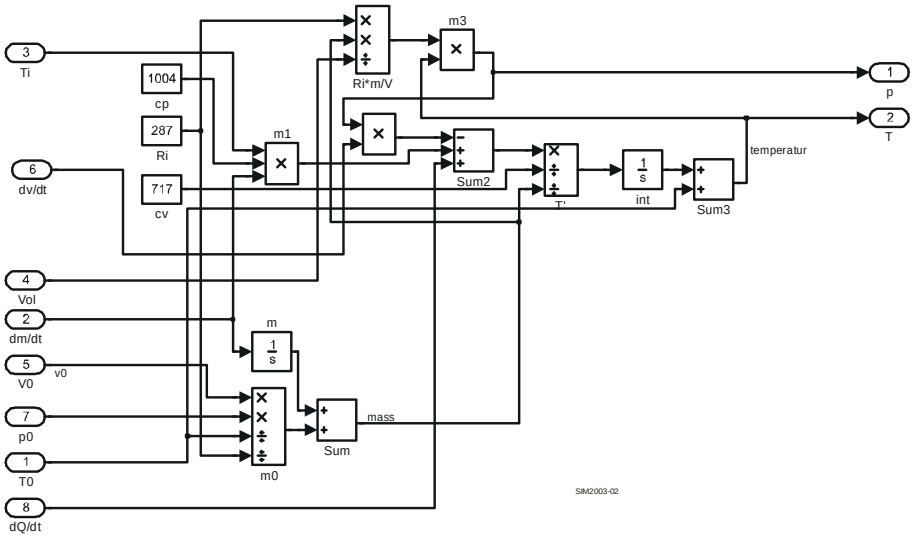

Figure 5. Ideal gas model for arbitrary changes

The model in Fig. 5 is suitable for any arbitrary state change of an ideal gas. It easily can be verified by doing a defined state change and compare the result with Equ. 2,3 or 4. Fig. 6 shows the example for the adiabatic change, Fig. 7 for the isothermal change. Here a little trick is shown. To hold the temperature constant, the heat exchange is controlled appropriately.

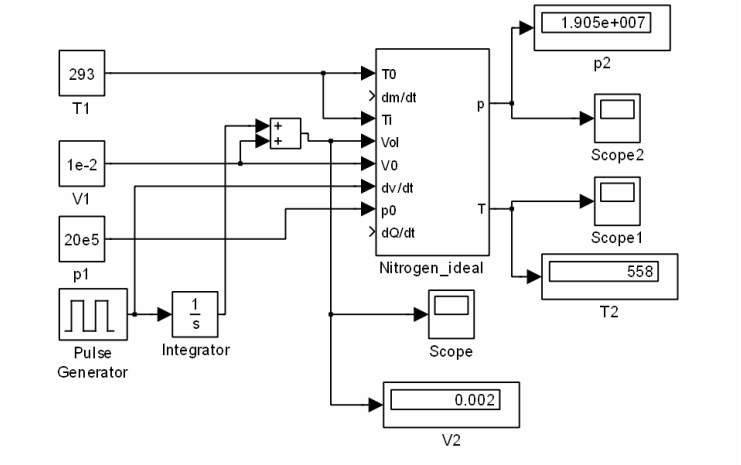

Figure 6. Verification test for adiabatic process

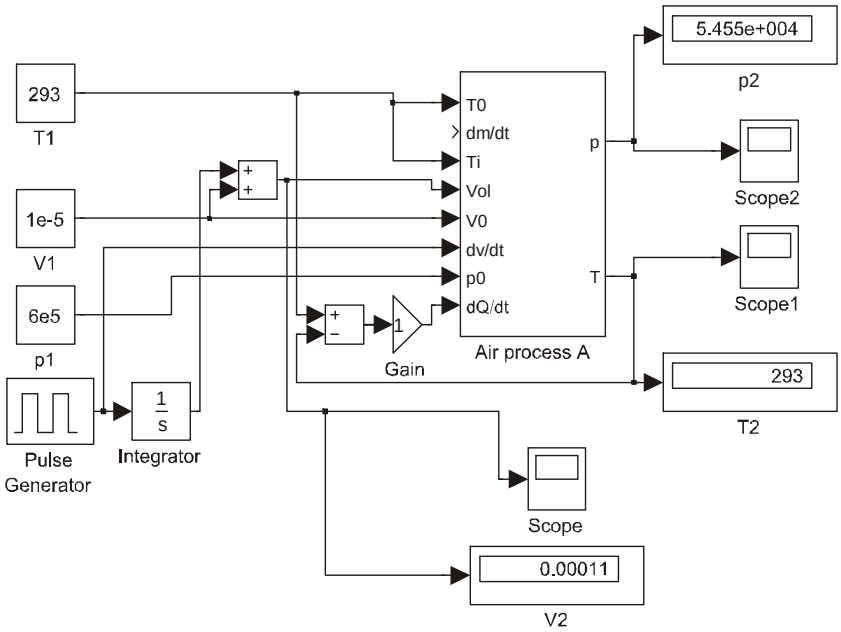

Figure 7. Verification test for isothermal process

6. Simulation Model for the Accumulator with Real Gas

The ideal gas law

$$pV = nRT \tag{22}$$

may be not accurate enough for high pressures that are used in hydraulic accumulators. An improvement can be accomplished with the model of van der Waals. The reason of using the van der Waals equation is that it is mathematically very simple, but, still exhibits critical and phase-transition behavior and predicts a first-order gas-liquid transition [1]. It involves two new constants a and b, unique for each gas, and tables giving these values can be found in many texts [2]. The constant a is introduced to correct the attractive forces among molecules and the constant b is introduced to account for the volume occupied by the molecules. The gas law then becomes

$$\left(p + \frac{n^2 a}{V^2}\right)(V - nb) = nRT\,. \tag{23}$$

To solve for the number of moles, Equ. 23 is of third order, which is difficult to solve. An efficient numerical solution can be established by rearranging to

$$\left(p+\frac{n^2 a}{V^2}\right)(V-nb)-nRT=0 \tag{24}$$

and applying a constraint solver in the MATLAB®/Simulink™ model (Fig. 8). The system is encapsulated into a subsystem block (Fig. 8, right) and is used in this form to calculate the initial mass inside the gas storage.

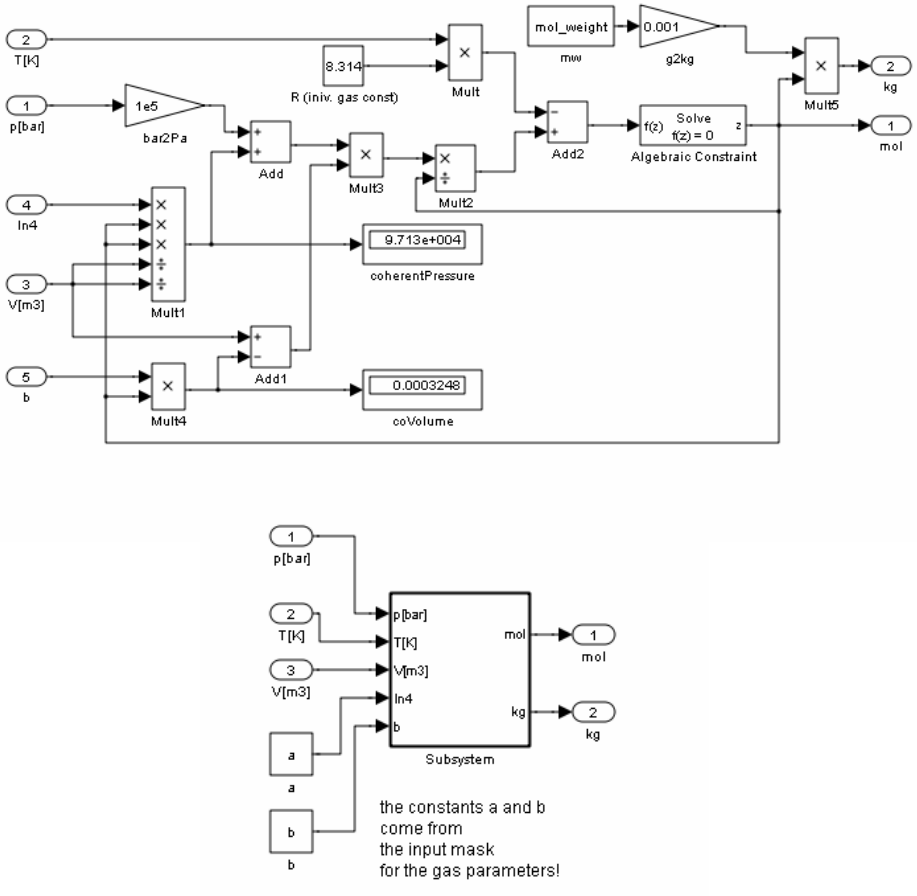

Figure 8. Iterative solution for the gas mass

Since the object is a closed storage element there is no massflow in and out, Equ. 20 becomes

$$\dot{T} = \frac{1}{m c_v}\left(-p\,A\,\dot{y} + \dot{Q}\right) , \tag{25}$$

which again is a differential equation of first order. It is solved with the system in Fig. 9. Equ. 26 works with the specific heat c_v assumed to be constant and shows one of the weakness points of the van der Waals equation.

The complete model for the real gas process using the van der Waals equation is shown in Fig. 9. The system calculates the initial gas mass in the volume and then starts the simulation of the dynamic process.

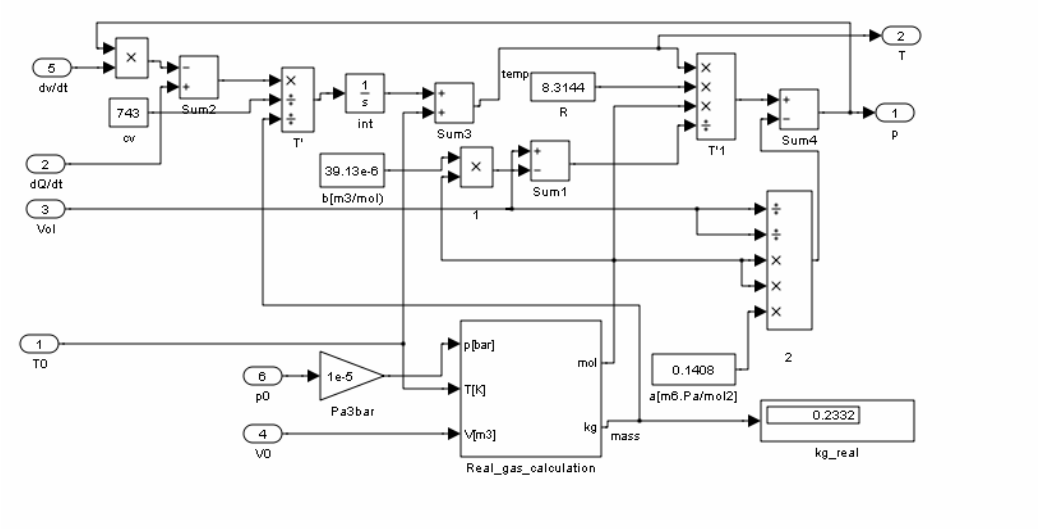

Figure 9. Numerical solution of gas equation for process simulation

Now the model for the gas process is complete and can be encapsulated for application, for example to demonstrate the results for an adiabatic process in Fig. 10. The results compare directly with the ideal gas solution in Fig. 6. The initial volume of 10 liters is comressed to 2 liters. Pressure and temperature are sligthly higher compared to the ideal gas solution.

It should be mentioned, that the purpose of the systems shown here is not to solve the gas equations in a graphical manner, but the simulate an arbitrary process over the time. The goal was to enable co-simulation together with the mechanical system and also to model the heat exchange with the environment.

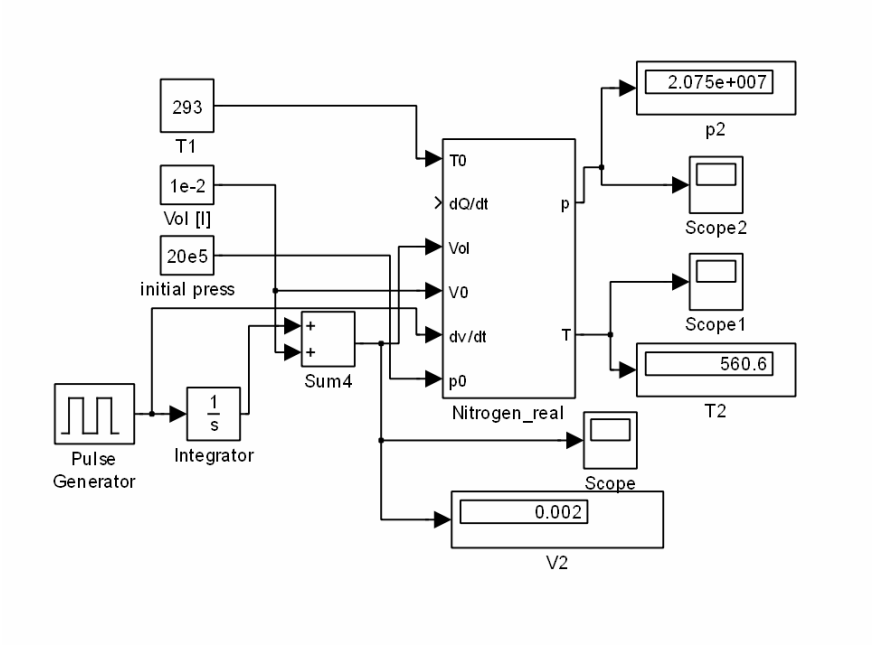

Figure 10. Adiabatic process for model test

7. Conclusions

The presented models worked well to optimise the design of the damping system. It turned out that the achievable energy absorbing effect was limited by the mechanical design of the machine. The practical realisation is pending.

Differences between the ideal and the real gas model were not essential for designing the damping system.

However, the van der Waals model is not appropriate for very accurate calculations, since the individual values of the critical constants are in poor agreement with experiments, though it is useful for qualitative purposes [1]. Another criticism of the van der Waals equation is that it begins to lose accuracy as the density increases (high pressures). For some gases the errors from the van der Waals equation become substantial for pressures above 170 bar [3].

References

[1] Hill T. L., *An Introduction to Statistical Thermodynamics*, Addison-Wesley, 1960, p. 289

[2] Obert E. F., *Concepts of Thermodynamics*, McGraw-Hill, New York, 1963, p.223.

[3] O'Connell J. P., Haile J. M., *Thermodynamics Fundamentals for Applications*, Cambridge University Press, 2005, p.171

[4] Fermi E., *Thermodynamics*, Dover Publications, 1936

[5] Blazeski A., *Thermodynamics 1*, Universitet Sv. Kiril I Metodij, 1994

[6] Drexler, P., Faatz, H., Feicht, F., Geis, H., Morlok, L., Wiesmann, E., *The Hydraulic Trainer Volume 3 - Planning and Design of Hydraulic Power Systems*, *Proportional Hydraulics*, Mannesmann Rexroth AG, 1988

[7] Yamada T., Kurino H., Matsunaga Y., Tagami J., *High performance passive hydraulic damper with semi-active characteristics*, Proceedings of 13th World Conference on Earthquake Engineering, Vancouver, 2004

[8] Richer E., Hurmuzlu Y., *A High Performance Pneumatic Force Actuator System*, ASME Journal of Dynamic Systems, Measurements, and Control, 2000

[9] Colgren, R., *Basic Matlab, Simulink and Stateflow*, American Institute of Aeronautics and Astronautics, 2007

Addresses:

- MSc. Emil Zaev, Ss. Cyril and Methodius University in Skopje, Karpos II bb, Skopje, Macedonia, emil@mf.edu.mk
- Dr. Gerhard Rath, Montanuniversitaet Leoben, Franz-Josef-str- 18, Leoben, Austria, gerhard.rath@unileoben.ac.at
- Dipl.-Ing. Hubert Kargl, Sandvik Mining and Construction, Alpine-Str. 1, Zeltweg, Austria, hubert.kargl@sandvik.com