

On Stability of the Two Stage Piloted Gas Pressure Control Unit

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This paper presents the modeling and model validation of a gas pressure control system with a two stage pressure reducing unit. Such regulator structures are in routine use if requirements of high accuracy are coupled with a large pressure drop. Dynamic behavior of such systems drastically depends on performances of the pipe section in between two control valves. In the paper, two system modifications are considered. The initial system has a heating unit installed between two control stages and connected with them by long pipe lines. The field operation of the system at low temperature requires moving the heating unit upstream the first control valve. So the section volume between two stages decreases, and the system is prone to unstable behavior. The both modifications of the system are compared, its dynamic performances are studied on the simulation model and the critical volume of the intermediate section is defined. The experimental results revile reasonable agreement with theoretical analysis.

Keywords: Gas pressure control system, piloted pressure reducing valve, stability

Target audience: Pneumatics, Control System Design

1 Introduction

The gas pressure reducing valves are widely used in industry for various applications. They are main control units of the natural gas transportation and distributing stations in the network from gas extraction to the consumer end-user. The pressure reducer is a normally open valve that keeps outlet pressure nearly constant at a wide range of higher inlet pressure. To provide a high quality of static performances, the pilot operated regulators are widely used. The dynamic behaviour of such complicated units becomes very sensitive to the system configuration and operation conditions. In some cases instabilities are observed downstream the pressure regulator: the outlet pressure oscillates around the set point with very high amplitude.

Many studies were carried out in the past in order to characterize the regulators behavior. The procedure of forming nonlinear dynamic model of a gas pressure regulator was studied in [1]. The model showed a possibility of self-exciting oscillations in the system without outside disturbances. The results reported the effect of each design parameter for self-exciting oscillations and suggested methods to correct them. It was proposed that a linear model is sufficient for evaluation of stability and transient response if flow through the valve is laminar, dry friction is negligible, and the motion of valve and diaphragm is not constrained.

Some experimental and numerical simulations were performed in [2 and 3] to identify the relative influence of several parameters on stability and amplitude of oscillations.

Dynamic simulation of high pressure regulator was studied in [4] with the assumption of adiabatic process in the pressure regulator. The model was simulated using numerical tools and verified experimentally. It was found that as outlet volume increases, the stability of outlet pressure increases too.

The stability characteristics were investigated in [5]. It was found out the cause of vibration and proposed possible design modification for eliminating the unstable vibrations. It was concluded that damping coefficient, the diaphragm area, and upper and lower volumes are most important design parameters that affect the stability.

The comprehensive review of the past works in the area is presented in [6]. A nonlinear mathematical model of pressure regulator used to pressurize liquid propellant rocket tanks was developed. An analytical solution based on constant mass low rate assumption was compared with nonlinear formulation and the results demonstrated a high degree of confidence.

A mathematical model of the piloted gas regulator was developed in [7]. On the basis of modeling, experimental analysis and measured performances the operating conditions were found that increase stability. The numerical and experimental results were in good agreement with respect to each other. It was concluded that oscillations in downstream pressure increase for small volumes and higher upstream pressure.

As shown in the past works no generalized model of the gas pressure regulator exists. The structure and parameters of the regulator and connected network have a differential effect on the system stability. In this paper the two-stage piloted gas pressure regulator is considered. Dynamic behavior of such systems drastically depends on performances of the section in between two control valves. In the paper, two system modifications are considered. The initial system has a heating unit installed between two control stages and connected with them by long pipe lines. The field operation of the system at low temperature requires moving the heating unit upstream the first control valve. So the volume between two stages decreases, and the system is prone to unstable behavior. The both modifications of the system are compared, its dynamic performances are studied on the simulation model and the critical volume of the intermediate section is defined.

2 Modelling and simulation

A diagram of the gas pressure control system is shown in Figure 1. It includes two similar reducing unites installed up- and downstream the heater. The heater is located in a separate housing at a distance around 15 m connected by long pipes with the gas distributing station where both reducing unites are mounted. In this initial structure of the system, a cool gas is admitted to the input of the first reducing unite then goes out to the heater unit and comes back to the second reducer input. In the modified system both reducing valves are connected in series and the heater is located upstream the first stage. So the volume between two stages became much smaller being formed by a short connecting pipe.

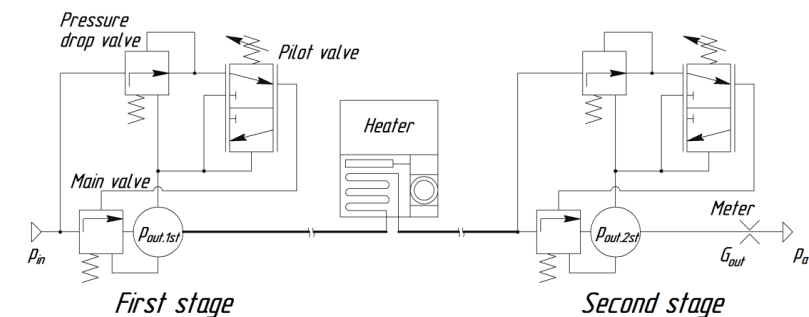


Figure 1: Diagram of the gas pressure control system

Each pressure reducing unit presents a pilot controlled regulator. It is composed of three control valves. The diagram of the regulator is shown in Figure 2. The main valve is positioned in the flow path to restrict the flow rate. The valve is driven by an actuator – a diaphragm, dividing a casing into two chambers K and L. Chamber K is connected to the downstream volume V_{out} through a pressure transmitting pipe. The pressure in other chamber L is controlled by the pilot valve. So the net force required to move the actuator is supplied by the pilot. The pressure induced force exerted on the diaphragm is balanced by the set value of the downstream pressure. The pilot compares downstream pressure p_{out} with a preset value and operates on the main valve by using an auxiliary pressure from the pressure drop valve. The pilot valve moving up and down can respectively decrease

or increase the main valve opening. The role of the pressure drop valve is to provide an independency of the pressure differential across the diaphragm from the upstream pressure p_{in} and the main valve displacement x_m . As the outlet pressure increases to the preset value, the pilot valve moves up such closing lower throttling orifice. The main valve opening interrupts. If pressure p_{out} continues to increase the pilot is moving upper and the upper throttling orifice is opening. The chamber L is connected with the outlet volume V_{out} , the main valve starts closing to decrease the flow rate and to prevent increasing of pressure p_{out} . At a steady state operation the pilot and pressure drop valves are closed; the main valve opening corresponds to the flow rate required to keep constant preset p_{out} value.

2.1 Mathematical model

To simplify mathematical model following assumptions are admitted: a fluid is an ideal gas; dry friction in mechanical units is negligible, lifting forces are independent of the lift, and the motion of valve and diaphragm is not constrained. Some of these assumptions are significantly contradictory to actual operations and are only acceptable in preliminary studies.

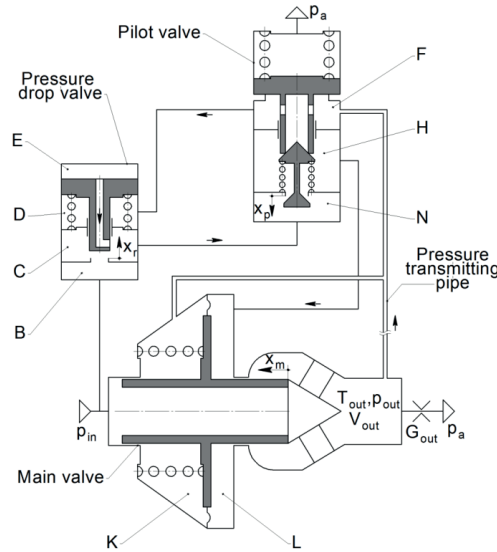


Figure 2: Diagram of the gas pressure reducing valve (one stage)

Mathematical model consists of differential equations including force balance equations for each moving shaft, gas equations for each chamber and mass flow rates for each throttle.

2.1.1 Main valve

Equilibrium equation for the main valve is

$$M \frac{d^2 x_m}{dt^2} + D \frac{dx_m}{dt} + Jx_m + F_0 + (p_K - p_L) A_m = 0, \quad (1)$$

where F_0 is a spring pretension at $x_m = 0$, A_m is an equivalent area of the membrane.

Assuming polytropic process for the gas in corresponding volumes, one obtains

$$\frac{V_K}{\gamma RT_K} \frac{dp_K}{dt} = A_m \frac{p_K}{RT_K} \frac{dx_m}{dt} + G_{pipe},$$

$$\frac{V_L}{\gamma RT_L} \frac{dp_L}{dt} = G_{HL} - A_m \frac{p_K}{RT_K} \frac{dx_m}{dt},$$

$$\frac{V_{out}}{\gamma RT_{out}} \frac{dp_{out}}{dt} = G_m - G_{pipe} - G_{out},$$

where G_{HL} is a mass flow rate from volume H to L, G_{pipe} is a mass flow rate in the pressure transmitting pipe from V_{out} to the branch point of pipes to volumes F and K.

Mass flow rate at the valve lift x_m at subcritical flow

$$G_m = C_d \pi d_{seat} x_m p_{in} \sqrt{\frac{2}{RT_{in}} \frac{\gamma}{\gamma-1} \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{\gamma+1}{\gamma}} \right]}$$

and at critical pressure drop

$$G_m = C_d \pi d_{seat} x_m p_{in} \sqrt{\frac{\gamma}{RT_{in}} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{2(\gamma-1)}}}.$$

2.1.2 Pressure transmitting pipe

For a laminar flow

$$(p_{out} - p_K - Z_{pipe} G_{pipe}) \frac{A_{pipe}}{l_{pipe}} = \frac{dG_{pipe}}{dt},$$

$$\text{where impedance } Z_{pipe} = \frac{RT_{out}}{p_{out}} \frac{32 \mu l_{pipe}}{A_{pipe} d_{pipe}^2}$$

for a turbulent flow

$$(p_{out} - p_K - Z_{pipe} G_{pipe}^2) \frac{A_{pipe}}{l_{pipe}} = \frac{dG_{pipe}}{dt},$$

where impedance

$$Z_{pipe} = f \frac{RT_{out}}{p_{out}} \frac{l_p}{2 d_{pipe} A_{pipe}^2}, \quad f = \left(-1.8 \log_{10} \left(\frac{6.9}{Re} + \left(\frac{e}{3.7 d_{pipe}} \right)^{1.11} \right) \right)^{-2}$$

- friction factor, e – surface roughness.

2.1.3 Pressure drop valve

Equilibrium equation for the pressure drop valve

$$M \frac{d^2 x_r}{dt^2} + D \frac{dx_r}{dt} + Jx_r + F_0 + (p_E - p_D) A_{piston} = 0. \quad (9)$$

Assuming polytropic process for the gas in volumes V_C , V_D and V_E one obtains:

$$\frac{V_C}{\gamma RT_C} \frac{dp_C}{dt} = G_r - G_{CN}, \quad (10)$$

$$\frac{V_D}{\gamma RT_D} \frac{dp_D}{dt} = G_{FD} - A_{piston} \frac{p_D}{RT_D} \frac{dx_r}{dt}, \quad (11)$$

$$\frac{V_E}{\gamma RT_E} \frac{dp_E}{dt} = A_{piston} \frac{p_E}{RT_E} \frac{dx_r}{dt} - G_{EC}. \quad (12)$$

For a subcritical and supercritical flow, consequently

$$G_r = C_d \pi d_{seat} x_r p_{in} \sqrt{\frac{2}{RT_{in}} \frac{\gamma}{\gamma-1} \left[\left(\frac{p_C}{p_{in}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_C}{p_{in}} \right)^{\frac{\gamma+1}{\gamma}} \right]}, \quad (13)$$

$$G_r = C_d \pi d_{seat} x_r p_{in} \sqrt{\frac{\gamma}{RT_{in}} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{2(\gamma-1)}}}. \quad (14)$$

2.1.4 Pilot valve

Equilibrium equation for the pilot valve

$$M \frac{d^2 x_p}{dt^2} + D \frac{dx_p}{dt} + Jx_p + F_0 + (p_F - p_a) A_{piston} = 0. \quad (15)$$

For a polytropic processes in the valve volumes

$$\frac{V_N}{\gamma RT_N} \frac{dp_N}{dt} = G_{CN} - G_p, \quad (16)$$

$$\frac{V_H}{\gamma RT_H} \frac{dp_H}{dt} = G_p - G_{HL}, \quad (17)$$

$$\frac{V_F}{\gamma RT_F} \frac{dp_F}{dt} = G_{pipe} - A_{piston} \frac{p_F}{RT_F} \frac{dx_p}{dt} - G_{FD}. \quad (18)$$

2.2 Simulation results

Simulation study of nonlinear equations (1) - (18) is carried out using software MatLab/SimuLink at the parameters of the regulator and operating conditions given in Table 1. The parameters presented in Table 1 correspond to the initial system. The modified system is different from the initial one by the intermediate section

between two regulators. Instead the heater, a chamber is installed, and the length of each pipe is reduced to 3.25 m. The effect of the chamber volume is studied during simulation.

Parameter	Value	Parameter	Value
Spring stiffness J , kN/m		Mass M , kg	
• Main valve	25	• Main valve	2
• Pressure drop valve	18	• Pressure drop valve	0.2
• Pilot valve	90.3	• Pilot valve	0.2
Spring pretention F_0 , N		Seat diameter d , mm	
• Main valve	500	• Main valve	50
• Pressure drop valve	270	• Pressure drop valve	2
		• Pilot valve	3
Pipe upstream the heater:		Pipe downstream the heater:	
• length l_{p1} , m	18.25	• length l_{p1} , m	24.2
• diameter d_{p1} , m	0.057	• diameter d_{p1} , m	0.057
Specific gas constant R , J/kg/K	519	Adiabatic exponent γ	1.4
Volumetric flow rate Q , nm/h	0...2000	Kinematic viscosity ν , m ² /sec	3.5*10 ⁻⁴
Input pressure p_{in} , MPa	8.0	Temperature T , K	306
Output pressure p_{out} , MPa		Volume	
• first stage	5.5	• heater V_{heater} , l	100
• second stage	1.2	• downstream the main valve V_{out} , l	25

Table 1 Simulation input data

Time responses of the regulator to a step change in the output flow rate are shown in Figures 3 and 4. The process starts with a slow increasing the system input pressure up to the nominal value at the meter's diameter 13mm. When the controller reaches standard conditions, the flow rate changes by the meter's diameter step increase from 13 to 15 mm at the 30th second. A step input - the volumetric flow rate - is presented in Figure 5.

The Figures 3 and 4 illustrates instability of the modified system. As can be seen, the initial system shows a stable behaviour but the modified system having a small volume (10 l) between two stages of the controller, is unstable from the very beginning. The amplitude of oscillations of the main valve (first stage) is equal to the steady state lift value for initial system. The oscillation amplitude of the second stage regulator is much smaller.

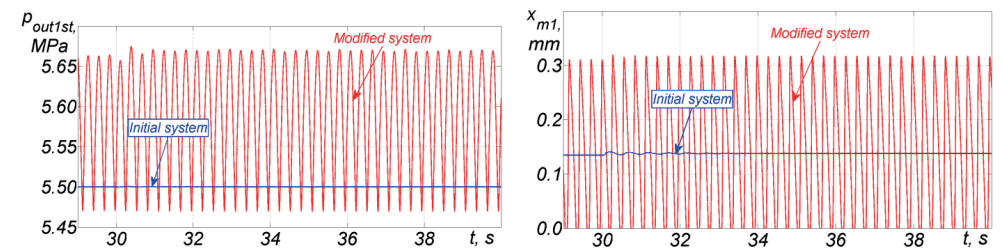


Figure 3: Output pressure and the main valve displacement for the first stage

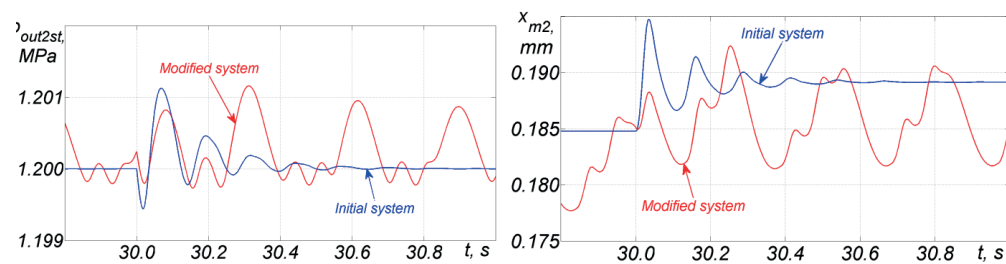


Figure 4: Output pressure and the main valve displacement for the second stage

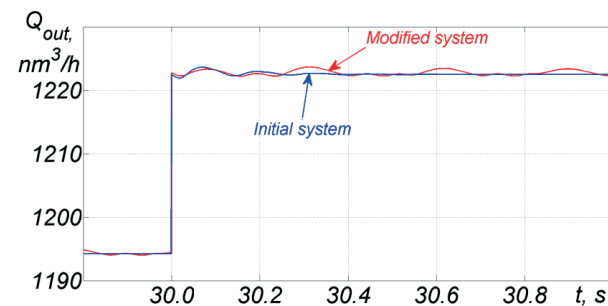


Figure 5: Time response of volumetric flow rate at the system output

An effect of chamber volume downstream of the first stage on the system stability is shown in Figures 6 and 7. A step disturbance of a volumetric flow rate at the system output is presented in Figure 8.

As the chamber volume increases from 20 l to 35 l the oscillations disappeared, and the system becomes stable.

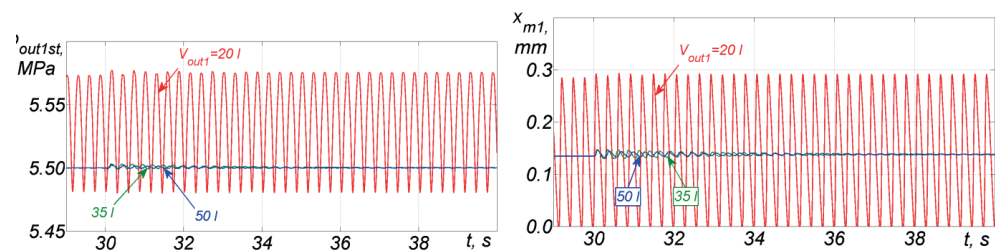


Figure 6: Output pressure and the main valve displacement for the first stage at various volumes (modified system)

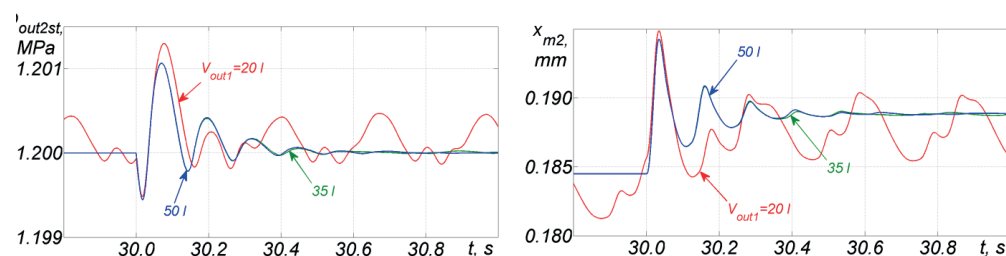


Figure 7: Output pressure and the main valve displacement for the second stage at various volumes (modified system)

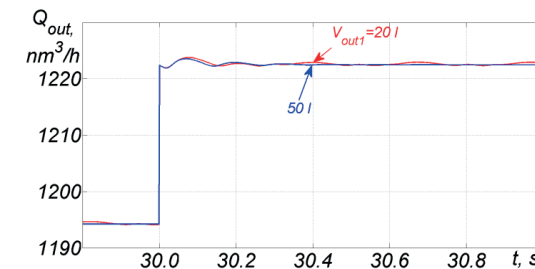


Figure 8 – Time response of volumetric flow rate at the system output

3 Experimental Results

The results of simulations were generally confirmed during operation in the framework of the gas distributing station. The initial system having the heater unit between two reducing stages demonstrated stable behaviour over all range of flow rates. After modification of the system the pressure oscillations downstream the first stage were registered. The range of oscillations was around 10 bars at the set point 50 bars. The oscillations were accompanied with noise. The instability was cut off with a decrease of the input pressure and an increase of the chamber volume. The tests will be continued in the laboratory conditions to find out stability domains in the regulator parameters.

4 Summary and Conclusion

Mathematical model of the two-stage gas pressure reducer was developed and simulated in the paper. The reference system with the heater unit installed between two stages and modified system without heater were compared and their dynamic behaviour was considered. The research revealed the definite and strong effect of the chamber volume in between two reducing stages on the system stability. The critical volume required for a stable behaviour was determined as result of simulation. The results were generally confirmed during operation in the framework of the gas distributing station. The research will be continued with the goal to improve the dynamic performances of the reducer and to widen its stability margin. It will be pointed out to updating the reducer design, its mathematical model and to carrying out laboratory tests of the regulator and its components.

5 Acknowledgements

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Nomenclature

Variable	Description	Unit	Variable	Description	Unit
A	Area	$[m^2]$	Q	Volumetric flow rate	$[m^3/s]$
C_d	Discharge coefficient		R	Specific gas constant	$[J/kg/K]$
D	Viscous drag coefficient	$[N*s/m]$	T	Temperature	$[K]$
d	Diameter	$[m]$	V	Volume	$[m^3]$
F	Force	$[N]$	x	Displacement	$[m]$
G	Mass flow rate	kg/s	γ	Adiabatic exponent	

J	Spring stiffness	N/m	μ	Dynamic viscosity	[kg/m/s]
l	Length	m	ν	Kinematic viscosity	[m ² /s]
M	Mass	kg	ρ	Fluid density	[kg/m ³]
p	Pressure	[bar]	ω	Frequency	[1/s]

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