Improvement in Dynamic Properties of a Pilot-Operated Gas Pressure Control Valve

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Abstract

In this paper, dynamics of a pilot-operated gas pressure control valve are studied through measurement and mathematical modeling for the purpose of obtaining high accuracy and stability over a wide range of flow rate. The pilot stage helps to increase accuracy. However, fluid-born noise and vibration often occur in such type of valves and pressure controllers running at supersonic pressure drop and high flow rate value. These phenomena are caused by instability of balance of the valve in a flow, instability of damping and friction forces. In the paper, the analytical and experimental research is carried out to reveal the most essential factors influencing stability and dynamic properties of the valve. The nonlinear and simplified linear models based on the perturbation technique are developed to predict the stability domain in the space of structural and operational parameters. The stability criterion for the system is deduced using D-decomposing method. CFD software is employed to study the effect of the poppet geometry on aerodynamic lifting force. Simulation is carried out with MatLab/Simulink, considering factors that influence on the dynamic properties of the valve such as lifting force, nonlinear friction and pilot dynamics. The analysis and simulations show general agreement with experimental data. Effective means for obtaining stable operation of the system are proposed.

KEYWORDS: gas pressure control valve, stability, self-exited oscillation, damping

1. Introduction

Fluid-born noise and vibration are a common problem in many types of valves and pressure controllers. These phenomena are caused by instability of the valve balance in a fluid flow. Difficulties of its stabilization are connected with complexity of physical processes of acousto-vortex interaction of the mobile structure of the valve with a flow and attached systems. Besides, small viscosity of gas makes impossible creation of stable forces of a contact friction in guiding surfaces. In many cases steady functioning of valves is provided with forces of coulomb friction which are instable and considerably change while in service under the influence of many factors. Therefore unstable behavior of gas valves can arise unexpectedly even in certified units at any changes of system configuration or service conditions or manufacturing process (change of input impedances of the attached pipelines, change of parameters of vibration of the case, change of guiding surface condition).

Many researchers have examined the stability of systems consisting of valves connected to pipelines and other components. Various mechanisms of flow induced instability in valves are described in [1]. Guidance for design against instability is given for each case: jet flow-inertia mechanism, turbulence, acoustic resonance. Results obtained in [2] indicate that negative hydraulic stiffness at the control valve due to fluid-structure interaction may cause the self-excited oscillation in the system. The interaction of safety valves with supersonic gas flow is studied in [3] and feedback between valve's vibration and stagnation-pressure oscillation is found and examined. Stability problem for control valves functioning in pressure charging systems is considered in [4], where an influence of air forces on system stability is analyzed. Many issues of safety valves design are presented in [5] including comprehensive data about coefficients of flow and flow forces for various valve's geometry.

The study of a safety valve dynamics is carried out in [6] and the method of stability analysis in terms of a relative positioning of impedance performances of the valve and the attached pipe lines is offered. It is shown that it is possible to provide stability of the simplest safety valves both by change in the mobile unit damping and by change in an upstream pipe line dynamic characteristic. Some structures and principals of design of correcting devices (or compensators) for providing stability of the control valves as part of a system are considered in [7].

However relevant data on structure and valve parameters are necessary for designing of correcting devices.

In this paper a pilot operated gas pressure control valve is considered. On the basis of theoretical and experimental research effective means for providing stability both at the expense of the valve damping, and at the expense of impact on the attached system characteristic are offered.

2. Theoretical study

2.1. Modeling

As a result of preliminary experimental research of gas safety valves shown in **Figure 1** the valves inclined to unstable work have been selected.

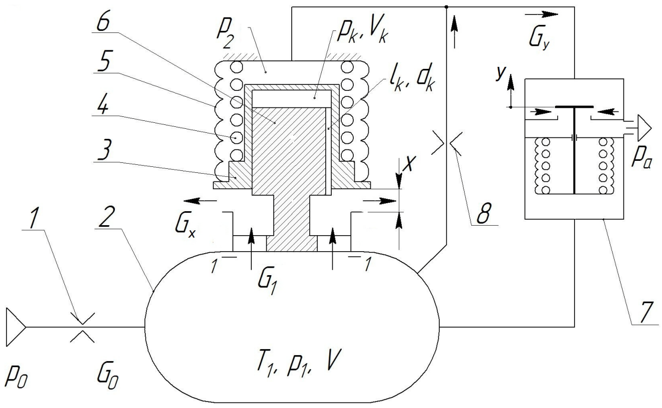

Figure 1: Schematic Diagram of the Control System: 1, 8 – restrictor; 2 – vessel; 3 – main valve poppet; 4 – spring; 5 – bellow; 6 – guide rod; 7 – pilot valve

It is established that vibrations of the valves occur at the frequency close to natural frequency of the main valve. Parameters of self-excited oscillations don't change when the pilot valve 7 is switched off and replaced with a source of constant pressure. It allows assuming that instability of the system is connected with dynamics of the main valve.

To simplify mathematical model following assumptions are made: a fluid is an ideal gas; pressure drop through the valve is supercritical; nonlinear friction are negligible; input impedance of the downstream system is equal to zero ($p_a = const$); hydraulic losses at the inlet can be neglected; the cross sectional area 1-1 is great enough compared to the maximum area of the throttling section, that is Mach number M_1<<1 even at the critical condition in the throttling section.

Assuming polytropic process for the gas in volume 2, one obtains

$$C\frac{dp_1}{dt} = G_0 - G_1, \quad (1)$$

where $C = \dfrac{V}{nRT_0}$ - pneumatic capacity.

Continuity equation for the flow between cross-section $1-1$ and the valve opening area is:

$$G_1 = G_x + S_1\rho_1 \frac{dx}{dt}. \qquad (2)$$

Flow velocity at $1-1$

$$U_1 = \frac{G_1}{S_1\rho_1} = \frac{G_x}{S_1\rho_1} + \frac{dx}{dt}. \qquad (3)$$

Mass flow rate at the critical pressure drop is

$$G_x = Axp_1, \qquad (4)$$

where $A = \mu\pi d\sqrt{\frac{k}{RT_1}}\left(\frac{2}{k+1}\right)^{\frac{k+1}{2(k-1)}}$.

Equilibrium equation for the main valve is:

$$M\frac{d^2x}{dt^2} + D\frac{dx}{dt} + Jx + F_0 - F_\Sigma = 0. \qquad (5)$$

Consider the main valve balance when pilot valve is switched-off, i.e. $p_2 = p_a = const$.

Assume that flow at the outlet section forms right angle to axis x. Then the summary flow force F_Σ, including the change in axial momentum flux through the control volume limited to the inlet channel wall, section 1-1, the valve's opening area and the poppet plane, can be expressed as:

$$F_\Sigma = \varphi S_1(p_1 - p_a) + G_1(U_1 - \dot{x}). \qquad (6)$$

Then the equation of balance of the valve (5) taking into account expressions (2), (3), (4) and (6) will become

$$M\ddot{x} + (D - Ap_1x)\dot{x} + Jx - \frac{A^2p_1RT_1}{S_1}x^2 - S_1(\varphi p_1 - p_{10}) = 0. \qquad (7)$$

Similarly the equation (1) can be deduced to

$$C\frac{dp_1}{dt} = G_0 - Axp_1 - S_1\frac{p_1}{RT_1}\frac{dx}{dt}. \qquad (8)$$

2.2. Nonlinear Simulation

Simulation study of nonlinear equations (7) and (8) is carried out using software MatLab/SimuLink at the parameters of valve and operating conditions given in Table 1.

Parameter	Notation	Value
Mass of main valve, kg	M	0,5
Spring stiffness, N/m, max	J	20600
min		13000
Area of section 1-1, m^2	S_1	$7{,}085 \cdot 10^{-3}$
Seat diameter, m	d	0.095
Flow coefficient	μ	0,6
Opening pressure, bar	p_{10}	4,6
Max mass flow rate, kg/s	G_0	1,8
Stagnation temperature, K	T_1	293
Specific gas constant, J/kg/K	R	287

Table 1: Initial Data

As appears from the equation (7), the account of dynamic component of the flow force leads to decrease in total damping factor proportionally to the valve lift x and to pressure p_1 at the valve input (i.e. to flow rate G_0). Besides, influence of static component of the flow force is directed to a reduction in total stiffness proportionally to square of the valve stroke x^2 and to inlet pressure p_1. Decrease in total stiffness due to flow forces action results in reduction of frequency of transient response and leads to reduction of a static error more than twice in a range of flow rates from ***0*** to G_{max} [10].

Taking into account the characteristic of the pilot valve, expression (6) for the flow force will become

$$F_\Sigma = \varphi S_1 p_1 - S_2 p_2 - (S_1 - S_2) p_a + G_1 (U_1 - \dot{x}). \tag{9}$$

The flow force value can be corrected by changing factor φ that allows adjusting mathematical model. Calculating the flow force factor φ for the various valve's geometry is carried out by simulation of flow field and pressure fluctuations in the throttling area using CFD-solver ANSYS.

Figure 2 shows the flow and pressure fields for the valve with flat and cone poppet at the opening $x=1mm$. The calculated steady state flow force F_Σ versus lift x is presented in **Figure 3**. The curve in **Figure 3 (a)** is not monotonous - the flow force increases at small openings (up to 4 mm) and then drops down. It can be the reason of static type instability resulted in self-exited oscillation at small openings, when increment of flow force exceeds increment of spring force. To avoid such mechanism of instability numerical simulations are carried out for several valve's configurations.

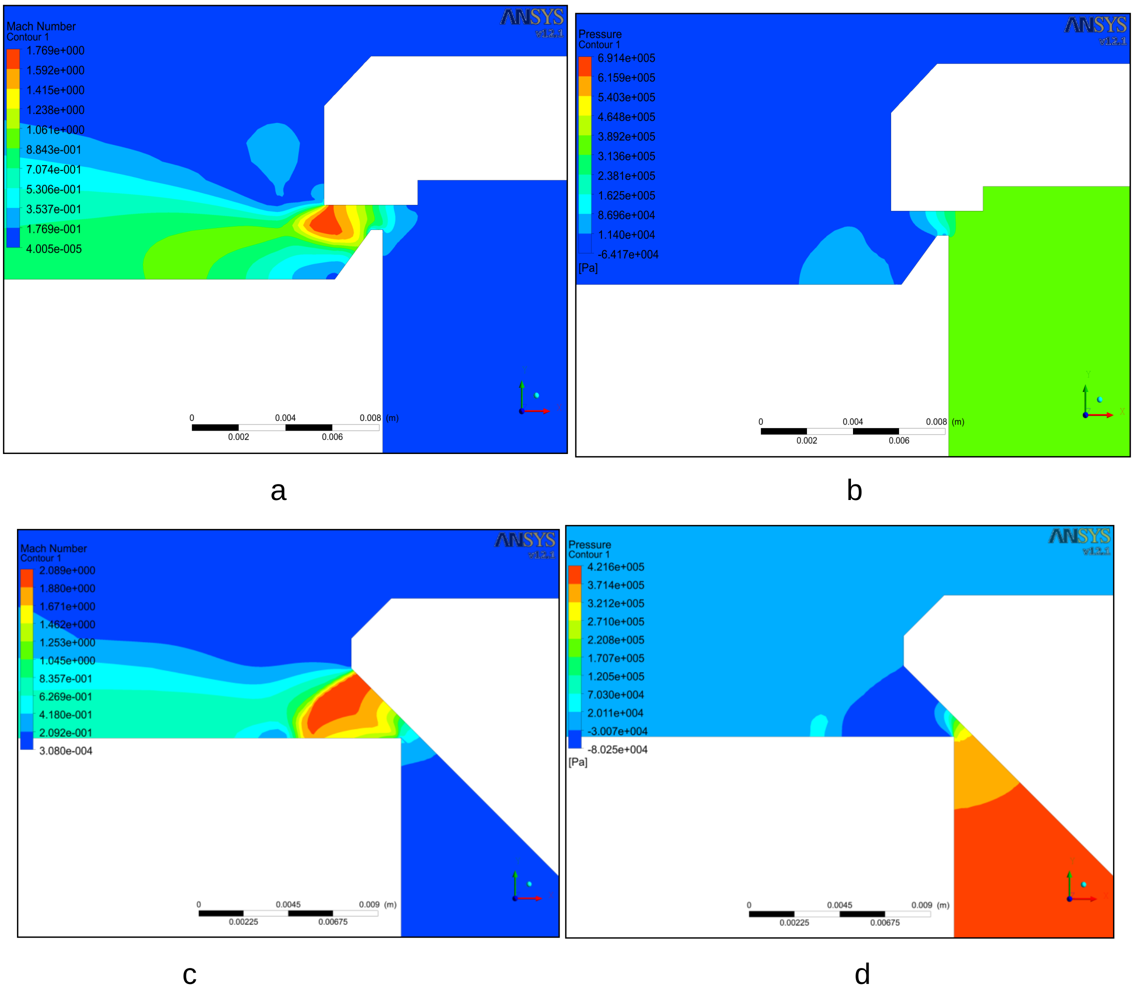

Figure 2: Simulated Mach number (a, c) and pressure (b, d) fields for the flat and cone poppet at *x=1 mm*

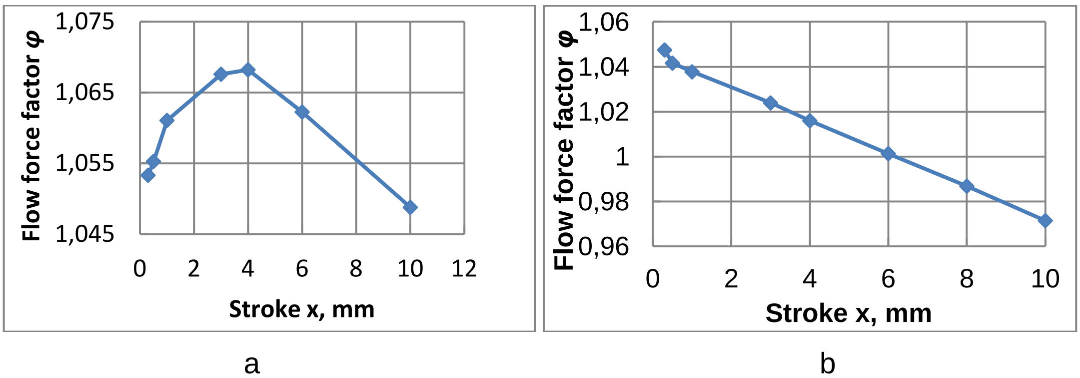

Figure 3: Calculated flow force characteristics for the flat (a) and cone (b) poppet

The force characteristic for the cone poppet shown in **Figure 3 (b)** has negative angle of slope over the whole range of openings. In this case the main valve obtains self-aligning feature and can keep stability with a spring of a smaller stiffness.

2.3. Linear Modeling

In order to obtain more general regularities research of the simplified model of the valve is conducted. The initial equations are linearized about steady-state operating conditions using method of small perturbations, and stability conditions are formulated in terms of requirements to damping factor D in the whole range of flow rates from 0 to G_{max}. Characteristic equation is derived in form $D_e(s)=0$, where $D_e(s)$ is eigenoperator of the system. The stability criterion for the system is deduced using D_e-decomposition method [6], [7]. According to this method, a space of system parameters can be divided into stability (if exists) and instability domains. This way, stability domain can be plotted on two-axis plane for any pair parameters linearly occurred in characteristic equation. **Figure 4** shows the stability domain on the plane *V-D*. Here the simplest model of the pilot valve as proportional circuit is presented by the following expression $p_2 = p_{20} + K_p(p_1 - p_{10})$.

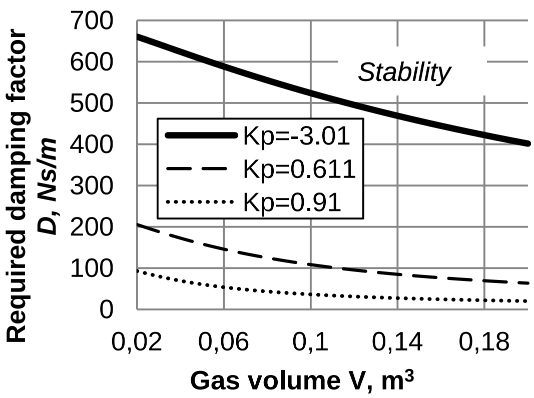

Figure 4: Effect of gain K_p on required damping factor D

The results obtained from linear analysis totally correspond with the data of the nonlinear model research:

• The worst conditions for the valve stability are realized at G_{max}. In this case the greatest damping D is required.

• The required damping D increases with reduction of the tank volume V and the feedback gain K_p.

Since feedback gain K_p depends on permissible static error, to stabilize similar system it is necessary to be guided by the maximum flow rate and the least tank volume.

On the basis of theoretical study next means are proposed to improve dynamics of the system and to increase stability margin.

2.4. Stabilizing the System

2.4.1. Resistance at the Valve Inlet.

As shown in work [7], it is possible to provide stability of a safety valve by increase in the real component of upstream piping impedance. In our case it can be realized by means of a throttle attached to the valve inlet. Made of some porous material or a package of mesh elements the throttle has characteristic close to linear in a wide range of flow rates. Efficiency of such approach is demonstrated in [8]. It is shown there that the system stability over all range of operating conditions is obtained at a small value of hydraulic resistance Z_R=*20 kPas/kg.*

2.4.2. Damping of the moving unit.

It is possible to provide a required damping by a choice of parameters of the channel connecting cavity between top of the guiding (6) and the poppet (3) (**Figure 1**). Such damping is widely used and much explored relating to hydraulic valves [9]. For our case, the valve model included resistance and inductance of the channel is examined in previous study [8] using MatLab/Simulink software to define parameters of the damping device - the length l_K and diameter d_k of the channel, the volume V_K - *from* the requirements of the valve stability and transient quality.

3. Experimental research

Tests of the valve are conducted on the special test rig presented in **Figure 5**.

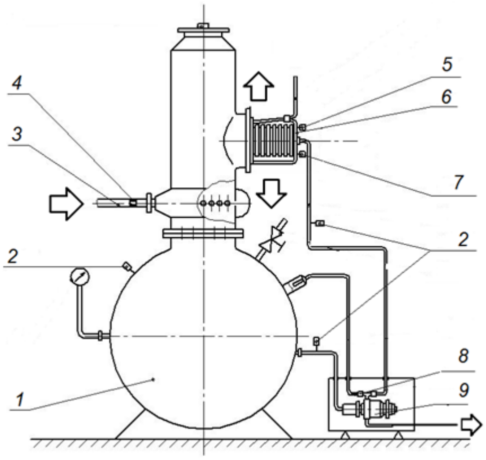

Figure 5: Test rig: 1 – vessel; 2 – pressure sensor; 3 – pipeline; 4 – flow metering restrictor; 5 – valve position sensor; 6 –valve tested; 7 – vibro acceleration sensor; 8 – flow restrictor; 9 – pilot valve

Variation of the flow rate is carried out by supply pressure $\boldsymbol{p_0}$ (up to 200 bar) upstream of a throttle 4 (or throttle 1 in **Figure 1**). Static and dynamic pressures in the vessel and pipes are measured with transducers 2, valve displacement and acceleration – by means of gages 5 and 7. The measuring frequency range is up to 2500 Hz.

In **Figure 6** typical behaviour of the instable valve is presented at slow increase and decrease of pressure p_0 upstream metering restrictor 4. One can see instability occurred at small lift of the valve.

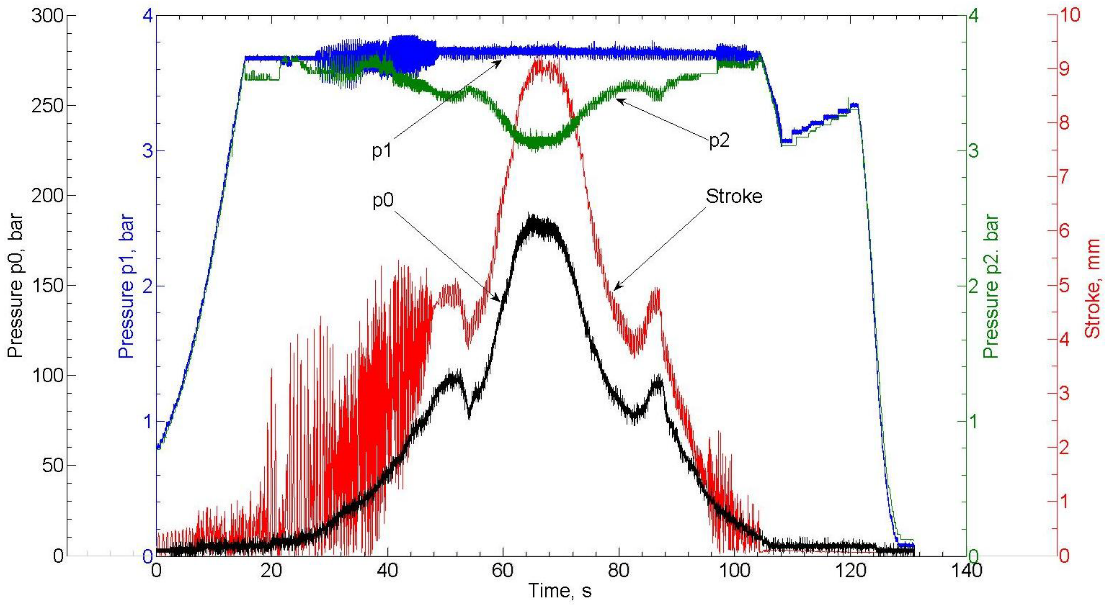

Figure 6: Instable behaviour of the valve at slow variation of supply flow rate G_0 from 0 to 1.8 kg/s

Self-exited oscillations of the valve have been registered at frequency about 30 Hz that corresponds to the frequency received in numerical experiments. Some difference in the process attenuation can be explained by nonlinear friction in guiding surfaces.

Efficiency of a throttle at the valve inlet for system stabilization is confirmed. Oscillation amplitudes decrease with growth of a throttle resistance. At resistance of 20 kPa*s/kg oscillations aren't exited.

In **Figure 7** behaviour of the valve stabilized by the proposed damping device is presented at the same operating conditions.

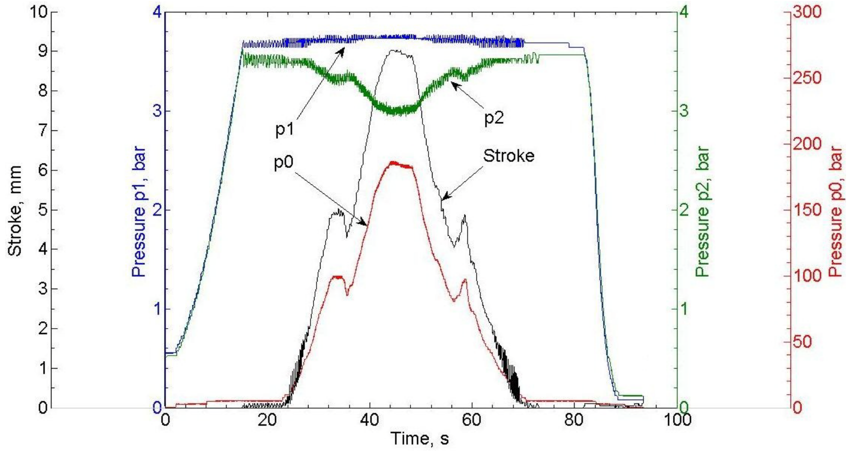

Figure 7: Measurement for the stabilized valve at slow variation of supply flow rate G_0 from 0 to 1.8 kg/s

4. Conclusion

An unstable behaviour of a pilot-operated gas pressure control valve at supercritical pressure ratios is studied through measurement and mathematical modeling. The analytical and experimental research is carried out to reveal the most essential factors influencing stability and dynamic properties of the valve. The linearized model and the perturbation technique are used to predict the stability domain in the space of structural and operational parameters. The analysis shows general agreement with experimental data.

Three effective means to improve system stability are offered:

- installing a throttle at the valve inlet;
- inserting additional damping into the main valve unit;
- advanced design of the main poppet geometry.

The mathematical model for definition of parameters of correcting devices (the throttle and the damper) proceeding from requirements of static accuracy and quality of transients is received.

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7. Nomenclature

D	viscous drag coefficient	N*s/m
d	diameter	m
F	force	N
G	mass flow rate	kg/s
J	spring stiffness	N/m
k	adiabatic exponent	
l	length	m
M	mass of main valve	kg
n	polytropic exponent	
p	pressure	N/m2
R	specific gas constant	J/kg/K
S	cross-sectional area	m2
T	temperature	K
U	velocity	m/s
V	volume	m3
x	displacement	m

Z	impedance	1/m/s
μ	flow coefficient	
ν	kinematic viscosity	m2/s
ρ	fluid density	kg/m3
φ	flow force factor	