Numerical research of parameters of interaction of the gas flow with rotary valve of the gas pipeline

A V Boldyrev, D L Karelin, V L Muljukin
Naberezhnye Chelny institute (branch) of Kazan Federal University, 68/19 (1/18), Naberezhnye Chelny, 423812, Russia
E-mail: alexeyboldyrev@mail.ru

Abstract. Conducted numerical research of static characteristics of the rotary gate valve at different angles of its deviation. For this purpose were set different values of pressure differential on the valve depending on which, was determined the mass flow and torque on valve axes. The mathematical model is provided by continuity equations, average on Reynolds, Navier-Stokes and energy, the equation of the perfect gas, the equations of two-layer k-ε model of turbulence. When calculating the current near walls are used Wolfstein's model and the hybrid wall functions of Reichardt for the speed and temperature. The task is solved in three-dimensional statement with use of conditions of symmetry. The structure of the current is analyzed: zones of acceleration and flow separation, whirlwinds, etc. Noted growth of hydraulic resistance of the valve with reduction of slope angle of the valve and with the increase in mass flow. Established increase of torque with reduction of the deviation angle of the valve and with increase in the mass expense.

1. Introduction
Application of slide gate valves with the rotary valve and electromagnetic management was widely adopted in power plant engineering, and the large number of works [1, 2, 3] is devoted to research and optimization of their parameters.

In this connection, the purpose of work is numerical research of the current via the valve installed in the gas pipeline with the rotary valve, and also obtaining static characteristics of this device.

2. Materials and methods
Figure 1 shows a diagram of the computational domain, representing a portion of the tubing disposed therein valve. The course was considered to be symmetrical relative to the plane OZY. The characteristic dimensions were: internal diameter of the pipe 490 mm, the inner diameter of the valve body 330 mm, the wall thickness of the valve is 5 mm, the length of the calculated pipe section 1000 mm, valve length 346 mm.

Below are the equations of the mathematical model, written in tensor form (summation over repeated index, according to the method proposed by Einstein [4]).

Quasi-stationary three-dimensional turbulent flow of a viscous compressible gas (methane CH4) is described:
1) Reynolds averaged continuity equation [4-6]

\[ \frac{\partial (\rho \cdot V_j)}{\partial x_j} = 0, \]  

(1)

Navier-Stokes equations [4-6]

\[ \rho \cdot V_j \cdot \frac{\partial V_i}{\partial x_j} = \frac{\partial P_{ji}}{\partial x_j}, \]  

(2)

and energy equation (according to the first law of thermodynamics) [4-7]

\[ \rho \cdot V_j \cdot \frac{\partial E}{\partial x_j} = \frac{\partial q_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left( P_{ji} \cdot V_j \right); \]  

(3)

2) equation of the ideal gas (Clapeyron's equation) [4-7]

\[ \rho = \frac{p_{atm} + p}{R \cdot T}; \]  

(4)

3) equations of two-layer Realizable of \( k-\varepsilon \) model of turbulence [5, 6, 8, 9]

\[ \rho \cdot V_j \cdot \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right) + G_k - \rho \cdot \varepsilon - \rho \cdot \Psi_M, \]  

(5)

\[ \rho \cdot V_j \cdot \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial x_j} \right) + C_{11} \cdot \rho \cdot s \cdot \varepsilon - \frac{\varepsilon}{k + \sqrt{\varepsilon \cdot \varepsilon}} \cdot C_{12} \cdot \rho \cdot \varepsilon, \]  

(6)

where \( i, j \) – the indexes accepting values 1, 2, 3; \( \rho \) – gas density, kg/m\(^3\); \( V \) – velocity vector of fluid movement, m/s; \( x \) – coordinate, m; \( P \) – stress tensor, Pa; \( E \) – total energy of gas, J/kg; \( q \) – vector of heat flux density, W/m\(^2\); \( p_{atm} \) – atmospheric pressure, 101325 Pas; \( p \) – excessive pressure, Pa; \( R \) – gas constant, 518 J / (kg K); \( T \) – absolute temperature of gas, K; \( k \) – kinetic turbulent energy, J/kg; \( \mu \) – molecular dynamic viscosity of gas, 1.12·10\(^{-5}\) Pa·s; \( \mu_t \) – turbulent dynamic viscosity of gas, Pa·s; \( \sigma_k \) – turbulent number of Schmidt for the equation (5), 1; \( \sigma_e \) – turbulent number of Schmidt for the equation (6), 1; \( G_k \) – the generative component of the equation of transfer of energy \( k \) (5), kg/(m·s\(^3\)); \( \varepsilon \) – speed of dissipation of kinetic turbulent energy \( k \), W/kg; \( \Psi_M \)
– the component of the equation of transfer of energy \( k \) (5), considering influence of compressibility of gas on pulsations of speed, W/kg; \( C_1 \) – coefficient at the generative component of the equation of transfer (6); \( \nu \) – molecular kinematic viscosity of gas (\( \mu / \rho \)), m\(^2\)/s; \( C_\nu \) – coefficient at the dissipative component of the equation of transfer (6), 1.9; \( s \) – the parameter connected with the second invariant of the tensor of speeds of deformations \( S_{ij} \), s\(^{-1}\)

\[
s = \sqrt{2 \cdot \dot{S}_{ij} \cdot \dot{S}_{ij}},
\]

\[
\dot{S}_{ij} = \frac{1}{2} \left( \frac{\partial V_j}{\partial x_i} + \frac{\partial V_i}{\partial x_j} \right).
\]

According to Reynolds's (RANS) approach, components of the stress tensor are defined as follows [4-6] (here the 1st index designates number of the axis perpendicular to which the site, and the 2nd index - number of the axis on which stress is projected is oriented)

\[
P_{ji} = P_{\text{lam}ji} + P_{\text{turb}ji},
\]

where \( P_{\text{lam}} \) – viscous (laminar) stress tensor, Pa;

\( P_{\text{turb}} \) – turbulent stress tensor, Pa.

On the generalized Newton's laws for viscous compressible medium [4-6]

\[
P_{\text{lam}ji} = 2 \cdot \mu \cdot \dot{S}_{ij} - \left( p + \frac{2}{3} \cdot \mu \cdot \text{div} V \right) \cdot \delta_{ij},
\]

where \( \delta \) – the unit tensor (the component is equal 1 at \( i = j \) and it is equal 0 at \( i \neq j \)).

Components of the tensor of turbulent stress are calculated, according to the hypothesis of Boussinesq [4-6]

\[
P_{\text{turb}ji} = 2 \cdot \mu_i \cdot \dot{S}_{ij} - \frac{2}{3} \left( \rho \cdot k + \mu_i \cdot \text{div} V \right) \cdot \delta_{ij}.
\]

The total energy \( E \) represents the sum of internal and kinetic energy of gas [4-6]

\[
E = C_v \cdot T + \frac{|V|^2}{2},
\]

where \( C_v \) – heat capacity of gas at the constant volume, J / (kg K).

According to Mayer [10] formula, we have

\[
C_v = C_p - R,
\]

where \( C_p \) – heat capacity of gas with the constant pressure, 2240 J / (kg K).

Following Reynolds's approach, the vector of heat flux density [4-6]

\[
q = q_{\text{lam}} + q_{\text{turb}},
\]

where \( q_{\text{lam}} \) – viscous (laminar) component of heat flux density, W/m\(^2\);

\( q_{\text{turb}} \) – turbulent component of heat flux density, W/m\(^2\) which are defined as [4-7]

\[
q_{\text{lam}} = -\lambda \cdot \text{grad} T,
\]

\[
q_{\text{turb}} = -\frac{\mu_i \cdot C_p}{\sigma_{\nu}} \cdot \text{grad} T,
\]

where \( \lambda \) – molecular heat conductivity of gas, 0.0348 W / (m·K);

\( \sigma_{\nu} \) – turbulent number of Prandtl, 0.9.

Generation of kinetic turbulent energy depends on the turbulent stress and speeds of deformations [4-6, 8]
\[
G_k = \mu_t \cdot s^2 - \frac{2}{3} \cdot \rho \cdot k \cdot \text{div} \mathbf{V} - \frac{2}{3} \cdot \mu_t \cdot (\text{div} \mathbf{V})^2 .
\] (17)

Taking note of compressibility on turbulent pulsations of speed is made by introduction of the component [5, 8, 11]

\[\Psi_M = \frac{C_M \cdot k \cdot \varepsilon}{a^2} ,\] (18)

where \(C_M\) – Sarkar’s constant, 2.0 [11];
\(a\) – sonic speed in gas depending on its properties and temperature [4, 5], m/s.

Unlike standard \(k-\varepsilon\) model of turbulence the coefficient at the generative component of the equation of transfer of speed of dissipation is not the constant [5, 6, 8]

\[C_{\varepsilon} = \max \left(0.43; \frac{\eta}{5 + \eta}\right) ,\] (19)

where \(\eta\) – the coefficient

\[\eta = s \cdot \frac{k}{\varepsilon} .\] (20)

In this work is used the two-layer approach for the first time offered Rodi [5, 9], allowing to calculate the flow in the viscous underlayer and which is alternative to Low-Reynolds models of turbulence.

The flow in this case is separated into two layers. In the inside layer near walls turbulent dynamic viscosity \(\mu_t\) and speed of dissipation \(\varepsilon\) are determined from the algebraic equations as functions of distance by the normal to the next wall \(y\) and the turbulent Reynolds number

\[Re_y = \frac{y \cdot \sqrt{k}}{\nu} .\] (21)

In particular, in the inside layer the speed of dissipation \(\varepsilon\) is defined as

\[\varepsilon = \frac{\sqrt{k^3}}{l_x} .\] (22)

where \(l_x\) – distance scale, m.

In the external layer is solved input differential equation for transfer of speed of dissipation \(\varepsilon\) (6).

Differential equation for transfer of energy (5) \(k\) (5) in this approach is applied to all area of the flow.

Smooth transition between the decisions received in internal and external layers of the flow will be reached use hybrid wall functions of Jongen [12]

\[\Lambda = \frac{1}{2} \cdot \left(1 + \text{th} \left( \frac{Re_y - Re_y^*}{A} \right) \right) ,\] (23)

where \(Re_y^*\) – the limit value of the turbulent Reynolds number (boundary between two layers of the current), 60;
\(A\) – the constant determining width of the transitional zone between internal and external layers

\[A = \frac{\Delta Re_y}{\text{arth}(0.98)} ,\] (24)

where \(\Delta Re_y\) – the constant, 10.

In the real article to calculation of turbulent dynamic viscosity \(\mu_t\) and speed of dissipation in the inside layer applied Wolfstein’s model [5, 13].
Then the distance scale is defined as [5, 13]

\[ l_e = C_l \cdot y \cdot \left(1 - e^{-\frac{Re_e}{C_l}}\right), \] (25)

where \( C_l \) – the coefficient defined from the following expression

\[ C_l = \kappa \cdot C_{\mu 2}^{0.75}, \] (26)

where \( \kappa \) – the von Kármán constant, 0.42; \( C_{\mu 2} \) – the constant, 0.09.

Turbulent dynamic viscosity \( \mu_i \) depends on hybrid wall function of Jongen [5, 12, 13]

\[ \mu_i = \Lambda \cdot \mu_{i1} + (1 - \Lambda) \cdot \mu_{i2}, \] (27)

where \( \mu_{i1} \) – turbulent dynamic viscosity in the external layer of the current, Pa·s; \( \mu_{i2} \) – turbulent dynamic viscosity in the inside layer of the current, Pa·s.

\[ \mu_{i1} = C_{\mu 1} \cdot \rho \cdot \frac{k^2}{\varepsilon}, \] (28)

where \( C_{\mu 1} \) – the coefficient defined as [5, 8]

\[ C_{\mu 1} = \frac{1}{A_6 + A_8 \cdot U^* \cdot \frac{k}{\varepsilon}}, \] (29)

where \( A_6 \) – the constant, 4.0 [5, 8]; \( A_8 \) – the coefficient calculated on the following formulas (\( i, j, n \) – the indexes accepting values 1, 2, 3)

\[ A_8 = \sqrt{6} \cdot \cos \left(\frac{1}{3} \cdot \arccos \left(\sqrt{6} \cdot \omega\right)\right), \] (30)

\[ \omega = 2\sqrt{2} \cdot \frac{\dot{S}_{ij} \cdot \dot{S}_{mn} \cdot \dot{S}_{ni}}{s^3}; \] (31)

\( U^* \) – the coefficient, s\(^{-1}\) [5, 8].

\[ U^* = \sqrt{\dot{S}_{ij} \cdot \dot{S}_{ij} - \dot{W}_{ij} \cdot \dot{W}_{ij}}, \] (32)

where \( \dot{W} \) – rotation tensor, s\(^{-1}\) [5, 8]

\[ \dot{W}_{ij} = \frac{1}{2} \left(\frac{\partial V_i}{\partial x_j} - \frac{\partial V_j}{\partial x_i}\right). \] (33)

Turbulent dynamic viscosity in the inside layer of the current [5, 13]

\[ \mu_{i2} = \mu \cdot Re_y \cdot C_{\mu 2}^{0.25} \cdot \kappa \cdot \left(1 - e^{-\frac{Re_e}{A_\mu}}\right), \] (34)

where \( A_\mu \) – the constant, 70.

To link the parameters of the flow on the wall and in the inner layer of the flow in this work applied the hybrid wall functions of Reichardt [5, 14, 17] recorded using:

1) dimensionless distance on the normal to the next wall [4]

\[ y^+ = \frac{U^* \cdot y}{v}, \] (35)

2) the dimensionless speed directed parallel to the wall [4]
3) dimensionless temperature [5]

\[ T^* = (T - T_w) \frac{\rho \cdot C_p \cdot u^*}{q_w}, \]  

where \( u \) – the speed directed parallel to the wall, m/s; \( T_w \) – wall temperature, K; \( q_w \) – heat flux density, passing through the wall, W/m²; \( u^* \) – dynamic speed, m/s [4, 5, 14].

\[ u^* = \sqrt{\frac{g^* \cdot u \cdot v}{y} + (1 - g^*) \cdot \sqrt{C_{u2} \cdot k}}, \]  

where \( g^* \) – weight function of the turbulent Reynolds number [5, 14]

\[ g^* = e^{-\frac{Re}{11}}. \]  

Then the generative component of the equation of transfer of kinetic turbulent energy in parietal cells is calculated as:

\[ G_{k \text{ wall}} = g^* \cdot \mu_t \cdot s^2 + (1 - g^*) \cdot \frac{1}{\mu} \left( \frac{\rho \cdot u \cdot u}{u^*} \right)^2 \cdot \frac{\partial u^*}{\partial y^*}, \]  

and the rate of dissipation in near-wall cells \( \epsilon_{\text{wall}} \) is on the formula (22).

The hybrid wall function for dimensionless velocity and temperature have the form [5, 14, 15, 17]

\[ u^* = \frac{1}{k} \cdot \ln \left( 1 + k \cdot y^* \right) + \frac{1}{k} \cdot \ln \left( \frac{E^*}{k} \right) \left( 1 - e^{-\frac{y^*}{y^*_m}} - \frac{y^*}{y^*_m} \cdot e^{-b \cdot y^*} \right), \]  

\[ T^* = e^{-\Gamma} \cdot (T_{\text{lamb}}^* - q_{\text{lamb}}^*) + e^{-\Gamma} \cdot (T_{\text{turb}}^* - q_{\text{turb}}^*), \]

where \( y^*_m \) – coordinate of conditional boundary between viscous laminar and logarithmic turbulent underlayers (is defined during calculations), m; \( E^* \) – the constant, 9.0; \( b \) – the coefficient

\[ b = \frac{1}{2} \left( \frac{y^*_m \cdot \kappa}{1 \cdot \ln \left( \frac{E^*}{k} \right)} + 1 \right) \]

\[ \Gamma \] – the coefficient

\[ \Gamma = \frac{0.01 \cdot (Pr \cdot y^*)^4}{1 + 5 \cdot Pr^3 \cdot y^*}; \]

\( Pr \) – molecular number of Prandtl

\[ Pr = \frac{\mu \cdot C_p}{\lambda}; \]

\( T_{\text{lamb}}^*, T_{\text{turb}}^* \) – dimensionless functions of temperature for viscous and logarithmic underlayers respectively;

\( q_{\text{lamb}}^*, q_{\text{turb}}^* \) – dimensionless functions of viscous dissipation for viscous and logarithmic underlayers respectively.
\[ T_{\text{lam}}^+ = Pr \cdot y^+ , \]  
\[ T_{\text{turb}}^+ = \sigma_t \left( \frac{1}{\kappa} \cdot \ln \left( E^+ \cdot y^+ \right) + P^+ \right) , \]  
\[ q_{\text{lam}}^+ = \frac{\rho \cdot u^+ \cdot u^2}{2 \cdot q_w} , \]  
\[ q_{\text{turb}}^+ = \frac{\rho \cdot u^+}{2 \cdot q_w} \left( u^2 - \left( \frac{Pr}{\sigma_t} - 1 \right) \left( \frac{P^+}{Pr - \sigma_t} \cdot u^+ \right)^2 \right) , \]  
where \( P^+ \) – function of molecular and turbulent numbers of Prandtl \[16\]

\[
P^+ = 9.24 \cdot \left( \frac{Pr}{\sigma_t} \right)^{0.75} - 1 \cdot \left( 1 + 0.28 \cdot e^{0.007 Pr / \sigma_t} \right) .
\]  

In the task are set the following boundary conditions (figure 1) \[4-6\]:

1) for walls – "impermeability" and "not slipping" \((V = 0)\), lack of heat exchange through the wall \((q_w = 0)\), and for kinetic turbulent energy

\[
\frac{\partial k}{\partial y}_{\text{wall}} = 0 ;
\]  

2) for permeable boundaries:

2.1) on the entrance are set full excess pressure \( p_1^* \) (for obtaining static characteristics of the valve several calculations with different value are carried out \( p_1^* \) from 0.05 to 0.6 MPa), full gas temperature \( T_1^* = 283 \) K and turbulence parameters - intensity of turbulence \( J_{1\text{r}} = 0.01 \) and viscosity turbulent dynamic ratio \((\mu / \mu)_1 = 10\).

Excessive static pressure, as well as in the case with firm walls, was in result of calculations, and the module of the vector of local speed of gas flow \( |V_1| \) and static temperature of gas \( T_1 \) were calculated on formulas

\[ |V_1| = \sqrt{2 \cdot C_p \cdot \left( T_1^* - T_1 \right)} , \]  
\[ T_1 = T_1^* \left( \frac{p_1^*}{p_1} \right)^{\frac{k}{C_p}} . \]

2.2) at the exit are set excessive static pressure \( p_2 = 0 \) MPa, and also for inverse flows on this boundary (were used automatically only at emergence of similar flows \[5\]) – static temperature of gas \( T_2 = 283 \) K, intensity of turbulence \( J_{2\text{r}} = 0.01 \) and viscosity turbulent dynamic ratio \((\mu / \mu)_2 = 10\).

During calculations on output boundary were defined the local speed of gas flow \( |V_2| \) and static temperature \( T_2 \).

On the plane of symmetry are equal to zero shearing stresses, heat flux density and gradients of all parameters of the current on the normal to this boundary.

Kinetic turbulent energy and speed of its dissipation are connected with intensity of turbulence \( J \) and viscosity turbulent dynamic ratio \((\mu / \mu)\) following dependences \[5\]

\[ k = \frac{3}{2} \left( J \cdot |V| \right)^2 , \]
\[ \varepsilon = \frac{\rho \cdot C_{\mu_2} \cdot k^2}{\mu \cdot (\mu_i / \mu)} . \]  

(55)

3. Results and discussion

As a result of the executed series of calculations for each corner \( \varphi \) at different values of the total pressure on the entrance to the valve \( p_1^* \) are found: the torsion torque on the valve axis \( M \), resulting from influence of pressure forces and viscous friction, and also - the mass gas rate via the valve \( Q_m \), excessive static pressure on the entrance \( p_1 \), total pressure at the exit \( p_2^* \), losses of the total pressure \( \Delta p^* \) when passing gas via the valve

\[ \Delta p^* = p_1^* - p_2^* . \]  

(56)

The static characteristics of the valve received in calculations are given in figure 2. Growth of hydraulic resistance of the valve with reduction of slope angle of the valve and with increase in the mass expense, and also nonlinear dependence of losses of the total pressure on the mass expense is represented natural. However it is remarkable that square nature of this dependence is broken in the range of corners \( \varphi = 60^\circ - 90^\circ \), what is especially well visible at expenses 15-30 kg/s.

As appears from schedules in figure 2, the torsion torque on the axis of the valve increases with increase in the mass expense and with reduction of flow section of the valve at reduction of the corner \( \varphi \), what will well be coordinated with distribution of pressure in the vicinity of the valve (Figure 2).

At the provision of the valve, parallel in relation to the pipeline axis, value of the torsion torque fluctuates about zero. It is interesting that in the range of expenses of 15-30 kg/s, as well as on dependence for full losses of pressure, there is the excess of schedules.

![Figure 2](image_url)

**Figure 2.** Dependence of losses of the total pressure (at the left) and the torsion torque (on the right) on the valve axis from the mass gas rate via the valve at different corners \( \varphi \).

4. Conclusion

The analysis of the provided results of numerical modeling confirms need of specification of static characteristics of the considered valve by continuation of numerical research of the current in his flowing cavity at other intermediate values of slope angles of the valve \( \varphi \), including, in the full three-dimensional problem definition without application of the plane of symmetry.

Besides, nature of fluctuations of integral characteristics of the flow during the repetitive process, testifies to possible not stationary of the current at some slope angles of the valve \( \varphi \), and, so demands to consider this circumstance in mathematical model and calculations.
References

[1] Edelman A I 1970 Fuel valves of liquid rocket engines (Moscow: Mechanical engineering) p 244
[2] Telenkov A A 2011 Development and deployment of the technique and engineering program of creation of optimum designs of electromagnetic valves of liquid rocket engines. Thesis (Khimki) p 152
[3] Nizhnikov S A Dynamics of the individual electromagnetic drive of the valve of the internal combustion engine. Thesis (Kursk) p 102
[4] Loitsiansky L G 2003 Fluid Mechanics (Moscow: Drofa Publ.) p 840
[5] 2013 User Guide STAR-CCM+ version 8.02.011 (CD–adapco)
[6] Belov I A and Isaev S A 2001 The modeling of turbulent flows (St. Petersburg: Baltic GTU Publ.)
[7] Cebeci T and Bradshaw P 1988 Physical and Computational Aspects of Convective Heat Transfer (New York: Springer)
[8] Shih T - H, Liou W W, Slobber A, Yang Z and Zhu J 1994 A New $\kappa - \epsilon$ Eddy Viscosity Model for High Reynolds Number Turbulent Flows – Model Development and Validation NASA TM 106721
[9] Rodi W 1991 Experience with Two-Layer Models Combining the $\kappa - \epsilon$ Model with a One-Equation Model Near the Wall 29th Aerospace Sciences Meeting January 7–10 Reno NV AIAA 91–0216
[10] Bolgarsky A V, Mukhatchyov G A and Shchukin V K 1975 Thermodynamics and Heat Transfer (Moscow: Vysshaya shkola (Higher school))
[11] Sarkar S and Balakrishnan L 1990 Application of a Reynolds-stress turbulence model to the compressible shear layer ICASE Report 90–18 NASA CR 182002
[12] Jongen T 1998 Simulation and Modeling of Turbulent Incompressible Flows Ph.D. Thesis Lausanne EPFL
[13] Wolfstein M 1969 The velocity and temperature distribution in one-dimensional flow with turbulence augmentation and pressure gradient Int. J. Heat Mass Transfer 12 pp 301–318
[14] Reichardt H 1951 Vollstaendige Darstellung der turbulenten Geschwindigkeitsverteilung in glatten Leitungen Z. Angew. Math. Mech. 31(7) pp 208–219
[15] Kader B A 1981 Temperature and Concentration Profiles in Fully Turbulent Boundary Layers Int. J. Heat Mass Transfer 24 pp 1541–1544
[16] Jayatilleke C L 1969 The influence of Prandtl number and surface roughness on the resistance of the laminar sub-layer to momentum and heat transfer Progress in Heat and Mass Transfer 1 pp 193–330
[17] Launder B E and Spalding D B 1974 The Numerical Computation of Turbulent Flows Computer Methods in Applied Mechanics and Engineering 3 pp 269–289