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COMPARATIVE ASSESMENT OF STEADY-STATE PIPELINE GAS FLOW MODELS

ANALIZA PORÓWNAWCZA MODELI PRZEPŁYWU GAZU W RUROCIĄGU W STANACH USTALONYCH

One-dimensional, non-isothermal flow of gas in a straight pipe has been considered to predict pressure and temperature profiles along the horizontal pipeline under steady-state conditions. Selected analytical models for the simplified calculation of these profiles are evaluated on the basis of the numerical solution of the accurate model, which incorporates the convective term in the momentum equation and the kinetic energy term in the energy equation, while treating the enthalpy as a function of pressure and temperature. For closure of the system of the conservation equations, the GERG 2004 equation of state was chosen. In order to present the discrepancies introduced by the models, the results of the numerical and analytical solutions are compared with the field data. The results show that in the case of the high pressure gas transmission system, the effects of the convective term in the momentum equation and the kinetic energy term in the energy equation are negligible for pipeline pressure and temperature calculation accuracies. It also indicates that real gas effects play an important role in the temperature distribution along the pipeline and cannot be neglected from the calculation when approximate analytical equations are used.

Keywords: pipeline gas flow, steady-state flow model, convective term, kinetic energy term, design flow equation, GERG 2004 application

W artykule analizowano jednowymiarowy, nieizotermiczny przepływ gazu w stanie ustalonym w celu określenia zmian ciśnienia i temperatury w poziomym gazociągu. Wyniki uproszczonych obliczeń za pomocą wybranych modeli analitycznych zostały porównane z wynikami obliczeń uzyskanych za pomocą numerycznego całkowania modelu dokładnego, zawierającego człon konwekcyjny w równaniu pędu oraz człon energii kinetycznej w równaniu energii, jednocześnie przyjmując entalpię jako funkcję ciśnienia i temperatury. W celu zamknięcia układu równań zachowania zastosowano równanie stanu zgodnie z metodą GERG 2004. Dla przedstawienia niedokładności związanych z zastosowaniem różnych modeli, przeprowadzono weryfikację wyników na zbiorze danych rzeczywistych. Wyniki pokazują, że w przypadku systemów przesyłowych wysokiego ciśnienia, wpływ członu konwekcyjnego w równaniu pędu oraz członu energii kinetycznej w równaniu energii jest pomijalny z punktu widzenia dokładności

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obliczeń wartości ciśnienia i temperatury w gazociągu. Ponadto, wyniki pokazują, że prawidłowy opis właściwości gazu rzeczywistego odgrywa ważną rolę w obliczeniach zmian temperatury wzdłuż gazociągu i powinien być stosowany w przypadku przybliżonych metod analitycznych.

Słowa kluczowe: transport rurociągowy, ustalony przepływ gazu, człon konwekcyjny, człon energii kinetycznej, równanie przepływu, metoda GERG 2004

Nomenclature:

A – area, m^2
c_p – specific heat at constant pressure, J/(kg K),
D – pipe diameter, m,
g – the acceleration of gravity, m/s^2,
h – specific enthalpy, J/kg,
k – pipe roughness, mm,
L – pipeline length, m,
M – mass flow rate, kg/s,
P – power, W,
p – gas pressure, Pa,
Q – volumetric flow rate, m^3/s,
R – specific gas constant, J/(kg K),
T – gas temperature, K,
T_s – temperature of the surroundings of the pipeline, K,
U_L – overall linear heat transfer coefficient, W/(m K),
w – flow velocity, m/s,
x – distance coordinate, m,
z – compressibility factor.

Greek symbols:

α – angle between the direction x and the horizontal,
η_{is} – Joule-Thomson coefficient,
λ – Darcy friction factor,
ρ – density of the gas, kg/m^3.

Subscripts:

S – isentropic state,
N – normal conditions.

Note: Flow rate Q_N is shown in normal conditions of 101.325 kPa, 273.15 K

1. Introduction

Prediction of the pressure and temperature profiles along the pipeline under steady-state conditions belongs to many pipeline simulation and optimization problems, which require less adequate mathematical models, compared to the transient gas flow analysis. Prediction of pressure drop is a major objective in pipeline planning, in which steady state operation of the system is usually assumed. Similarly, in the field of pipeline optimization, pressure drop calculations are usually performed based on explicit analytical flow equations, since the expressions describing the relation between the pressure and the flow rate constitute the constraints in the optimization problem. Some examples of applications, usually related to design activities, include the selection of the network structure and the sizing of the diameters of the pipes. Furthermore, optimal control of gas networks may involve complex pipeline network analysis. In such cases, the explicit analytical expressions describing the relationship between pressure, temperature and flow rate for single pipelines are required to handle network operations efficiently due to the scale and complexity of the network.

As a general principle, the analysis scheme that is used to model the behavior of the pipeline must not deteriorate the accuracy of the project design calculations, i.e. must be detailed enough to include all relevant aspects of the pipeline flow. For example, consideration of the real gas effects and the non-isothermal flow behavior will be of importance when dealing with compressor station analysis. Similarly, the pressure drop at isothermal conditions in the pipeline section located upstream of compressor station can be inadequate to handle compressor station optimization problems, since both pressure and temperature values at the suction node of the compressor station play an important role in the fuel consumption analysis.

There is a vast literature on the analytical pipeline flow equations and different empirical correlations for the friction factor, a review of which can be found elsewhere, e.g. Hyman et al. (1975), Finch and Ko (1988), Ouyang and Aziz (1996), and more recently in the paper by Abdolahi et al. (2007). In simulating the steady-state flow of gas in pipes, most investigators neglect some terms in the conservation equations, which result in a loss of the accuracy of the simulation results.

Tian and Adewumi (1994) derived an analytical expression for compressible fluid flow in pipelines by assuming that temperature and compressibility factor are constant throughout the pipeline, while keeping the convective term in the momentum equation. The excellent agreement between the results predicted with this equation and the field data was obtained. In high-pressure cases, the relative error caused by neglecting the convective term was very small.

A more detailed study on the significance of the convective term in the momentum equation under steady-state conditions has been presented by Demneh and Mesbah (2008). The isothermal model coupled with the equation of state (EOS) for real gas (explicit in density) showed that the contribution of pressure drop due to the convective term to the total pressure drop is negligible at high operating pressures. At pressures above 3.5 MPa, the ratio of the pressure drop due to kinetic energy change to the total pressure drop in the pipeline was below 0.5% for all studied flow conditions (gas velocities). It is worth mentioning that under transient conditions the importance of the convective term in momentum equation is significant, as demonstrated by Abbaspour and Chapman (2008). The comparison shows that to the contrary of steady-state modeling, there exist significant difference in the flow distribution between solution with the convective term and without the convective term, and this difference increases when the mass flow rate increases in pipe.

In addition to the research on the effect of the convective term, studies on the heat transfer between the gas and the surroundings of the pipeline were undertaken, since a detailed design calculations for pipeline systems, i.e. gas pipelines and compressor stations sizing problems, require knowledge of the temperature profiles along the pipelines.

Schorre (1954) derived an equation which allows for the calculation of the temperature of the gas flowing in a horizontal pipeline. The energy equation was applied by superimposing the Joule-Thomson effect on the temperature changes due to heat transfer in the pipeline. The equation has been widely used in the gas industry, see the study by Fasold and Wahle (1998) for an example of this approach.

Coulter (1979) presented a modification of the Schorre's equation to account better for the real gas effects, resulting in an incorporation of the compressibility factor, while keeping the heat capacity along the pipeline unchanged. To mitigate this limitation, Edalat and Mansoori (1988) developed an analytical technique through the corresponding states principle, in which both the Joule-Thomson coefficient and the heat capacity at constant pressure are functions of pressure and temperature.

Based on a more rigorous analysis of the thermodynamic behavior of the flowing fluid, Alves et al. (1992) formulated the analytic equation describing the temperature distribution in pipes under steady state conditions. The equation included the effect of kinetic and potential energy on the temperature, and allowed for variable ambient temperature as a function of depth. This model is applicable to both pipeline and well bore temperature calculations.

The effect of ambient temperature change on the pipeline gas temperature was studied in the paper by Gersten et al. (2001). The steady-state, non-isothermal gas flow model was solved, coupled with different analytic heat transfer models for an on-shore and off-shore pipelines. The study showed that the realistic heat transfer model turns out to be essential, as the heat exchange with the environment has a strong influence on the temperature distribution along the pipeline. Furthermore, the real gas thermodynamics plays an important role in the gas flow analysis, and the effect of pressure on the enthalpy, i.e. the decrease in the temperature associated with the gas expansion was seen in the solution.

The method for solving one-dimensional model for steady-state flow of gas in a horizontal pipeline without neglecting any terms in conservation equations was developed by Buthold et al. (1971). As a relationship for the gas density, the Redlich-Kwong EOS was used. The integration technique used in the study was the fourth-order Runge-Kutta method. Similar approach, based on the numerical integration of ordinary differential equations for steady-state flow of steam in a horizontal pipeline, coupled with a dedicated equation of state (explicit in specific volume), has been performed by Białecki and Kruczek (1996). For an example of a numerical integration of the conservation equations with pressure and enthalpy as the dependent variables and the Peng-Robinson EOS, see the paper by Barragán-Hernández et al. (2005).

More recently Abdolahi et al. (2007) characterized different steady-state pipeline flow models with analytical and numerical approaches. The results from a variety of analytic pipeline flow equations were compared with the results of the accurate model based on conservation equations, coupled with the EOS explicit in compressibility factor. Comparison of the results with the field data showed that the numerical method featured much higher level of accuracy. The results of the study also showed that the pressure profile under steady-state conditions was essentially insensitive to the convective term when compared to the effect of the frictional term in the momentum equation.

The present paper concentrates on two aspects of the steady-state pipeline flow modeling: (i) the kinetic energy contribution to the energy equation, and (ii) the effect of simplified analytical integration of the differential equations on the accuracy of the results. The later is aimed at identifying the simplified analytical method that provides relatively high level of accuracy in the prediction of the pressure and temperature profiles, with the immanently low computational resource requirements compared to numerical integration techniques. The solution's compromise computational cost is expected to provide robust solutions for network simulation problems and a workable constraints handling in pipeline optimization problems. The predicted results of the numerical method with the accurate model, i.e. without neglecting any terms in the conservation equations, and the two explicit analytical methods, obtained by solving the energy equation under different sets of simplifying assumptions, were compared with the field data from the Yamal-Europe pipeline, taking the accuracy of calculations of the pressure and temperature distribution as a criterion. The numerical method with the accurate model adopted throughout this study uses the GERG 2004 model (Kunz et al., 2007) for taking into consideration the real gas effects, which is more contemporary and higher accuracy EOS compared to the previous investigations.

2. Steady-state pipeline gas flow models

2.1. Numerical approach

The steady-state non-isothermal compressible flow of gas in pipelines is described by a set of the equations expressing mass, momentum and energy conservation laws

$$M = \rho w A = \rho_N Q_N = \text{const} \tag{1}$$

$$\frac{\mathrm{d}p}{\mathrm{d}x} + \rho w \frac{\mathrm{d}w}{\mathrm{d}x} = \frac{\lambda}{D} \frac{\rho w |w|}{2} + \rho g \sin\alpha \tag{2}$$

$$M \frac{\mathrm{d}}{\mathrm{d}x}\left(h + \frac{w^2}{2}\right) = -U_L (T - T_S) - Mg\sin\alpha \tag{3}$$

Eqs. (1)–(3) may be rewritten in terms of pressure and volumetric flow rate under standard conditions (instead of density and velocity, respectively). This is a matter of convenience, since these quantities are commonly measured and used in the gas industry. By using the thermodynamic identity

$$\mathrm{d}h = c_p \mathrm{d}T + \left[\frac{1}{\rho} + \frac{T}{\rho^2}\left(\frac{\partial \rho}{\partial T}\right)_p\right]\mathrm{d}p \tag{4}$$

the following set of equations describing the non-isothermal, steady-state pipeline gas flow is obtained

$$A\frac{\mathrm{d}p}{\mathrm{d}x} + B\frac{\mathrm{d}T}{\mathrm{d}x} = C \tag{5}$$

$$D\frac{\mathrm{d}p}{\mathrm{d}x} + E\frac{\mathrm{d}T}{\mathrm{d}x} = F \tag{6}$$

where coefficients A, B, C, D, E, F are known functions of temperature and pressure:

$$A=\frac{1}{\rho}-\left(\frac{\rho_N Q_N}{A}\right)^2\frac{1}{\rho^3}\left(\frac{\partial\rho}{\partial p}\right)_T,\quad B=-\left(\frac{\rho_N Q_N}{A}\right)^2\frac{1}{\rho^3}\left(\frac{\partial\rho}{\partial T}\right)_p,\quad C=-\frac{\lambda}{2D}\left(\frac{\rho_N Q_N}{\rho A}\right)^2-g\sin\alpha,$$

$$D=\frac{T}{\rho^2}\left(\frac{\partial\rho}{\partial T}\right)_p,\quad E=c_p,\quad F=\frac{\lambda}{2D}\left(\frac{\rho_N Q_N}{\rho A}\right)^2-\frac{U_L}{\rho_N Q_N}(T-T_S)$$

Solving the set of the equations (5) and (6) for derivatives of pressure and temperature results in

$$\frac{\mathrm{d}p}{\mathrm{d}x}=(CE-BF)(AE-BD)^{-1} \tag{7}$$

$$\frac{\mathrm{d}T}{\mathrm{d}x}=(AF-CD)(AE-BD)^{-1} \tag{8}$$

The values of ρ, $(\partial\rho/\partial p)_T$ and $(\partial\rho/\partial T)_p$ are determined from the equation of state.

2.1.1. Effect of kinetic energy term on temperature distribution

It has already been recalled that for high-pressure gas transmission pipelines under steady-state conditions, the effect of the convective term in momentum equation is negligible (Tian & Adewumi, 1994; Abdolahi et al., 2007; Demneh & Mesbah, 2008). Furthermore, removing the kinetic energy term from the energy equation, the set of conservation equations has the form

$$\frac{\mathrm{d}p}{\mathrm{d}x}=\frac{\lambda}{D}\frac{\rho w|w|}{2}+\rho g\sin\alpha \tag{9}$$

$$M\frac{\mathrm{d}h}{\mathrm{d}x}=-U_L(T-T_S)-Mg\sin\alpha \tag{10}$$

The solution of Eqs (9) and (10) is analogous to the previous one except the coefficients A and B in Eqs. (5) and (6) are now $1/\rho$ and 0, respectively.

2.2. Analytical approach

The first integral of the momentum equation relates the flow with pressure changes along the pipeline. Assuming the flow in a horizontal pipe, Eq. (9) takes the form

$$\frac{\mathrm{d}p}{\mathrm{d}x}=\frac{\lambda}{D}\frac{\rho w|w|}{2} \tag{11}$$

Substituting the mass flow rate from Eq. (1) and integrating over spatial coordinate, the following end result is obtained

$$p_1^2-p_2^2=\frac{zR\overline{T}\lambda L}{A^2D}M|M| \tag{12}$$

Eq. (12) is known as the flow equation governing one dimensional, compressible fluid flow, and is readily available in text books on engineering applications of fluid mechanics. The compressibility factor, which is a function of pressure and temperature, and the friction factor should be evaluated at average conditions for the pipe, which causes the solution to be an iterative process. Depending on the needs of a given application, Eq. (12) can be corrected to account for elevation and kinetic energy change. The detailed development of the most general integrated flow equation (which accounts for the effects of elevation, friction, and kinetic energy change) is given in the paper by Ouyang and Aziz (1996).

The simplest approach to calculating average gas temperature in the pipeline is to assume that the thermal equilibrium with constant-temperature surroundings is reached, which leads to an isothermal process. However, when considering long distance pipeline systems with series of compressor stations causing the gas temperature to rise, a certain amount of heat transfer between the gas in the pipe and the surroundings will occur in reality, although thermal equilibrium may not always be reached. Therefore, for high-pressure gas transmission systems, the isothermal flow process will often provide unacceptable accuracy of the results. It is worth mentioning that the second extreme case is an adiabatic approach. The adiabatic process is typically adopted when flows with higher velocities are considered, for example in short piping elements, like nozzles, where there is no time for the heat transfer between the gas and its surroundings to take place.

For flow in a horizontal pipe, Eq. (10) takes the form

$$M\frac{\mathrm{d}h}{\mathrm{d}x} = -U_L\left(T - T_S\right) \tag{13}$$

Before integration of the above equation, the assumption regarding the thermodynamic behavior of the flowing gas must be made. Two cases, corresponding either to the ideal gas model or the real gas model in the thermodynamic description of the gas flow, are considered herein for the purpose of energy equation integration.

2.2.1. Energy equation integration: ideal gas model

Assuming ideal gas model with constant heat capacity rate, Eq. (13) becomes a separable differential equation

$$\frac{\mathrm{d}T}{T - T_S} = -\frac{U_L}{c_P M}\mathrm{d}x \tag{14}$$

By integrating between T_1 at $x = 0$ and T_2 at $x = L$ we get

$$T_2 = T_S + \left(T_1 - T_S\right)e^{-\beta x} \tag{15}$$

where: $\beta = U_L/(c_P M)$

Eq (15) allows for temperature profile prediction in a pipeline. This profile asymptotically approaches the temperature of the surroundings. The average gas temperature in Eq. (12) is calculated form the following equation

$$\overline{T} = \frac{1}{L}\int_0^L T\mathrm{d}x = \frac{1}{L}\left[T_S L + \frac{T_1 - T_S}{\beta}\left(1 - e^{-\beta L}\right)\right] \tag{16}$$

2.2.2. Energy equation integration: real gas model

In order to take into account the effect of pressure and temperature on the enthalpy of the real gas, the total differential and the cyclic relation provide the appropriate thermodynamic relationships

$$\mathrm{d}h = \left(\frac{\partial h}{\partial T}\right)_p dT + \left(\frac{\partial h}{\partial p}\right)_T dp \tag{17}$$

$$\left(\frac{\partial h}{\partial p}\right)_T = -\left(\frac{\partial T}{\partial p}\right)_h \left(\frac{\partial h}{\partial T}\right)_p \tag{18}$$

Substituting Eq. (18) into the Eq. (17) and combining this with Eq. (13) we obtain

$$c_P \frac{\mathrm{d}T}{\mathrm{d}x} - \eta c_P \frac{\mathrm{d}p}{\mathrm{d}x} + \frac{U_L}{M}(T - T_S) = 0 \tag{19}$$

where $c_p = (\partial h/\partial T)_p$ and $\eta = (\partial T/\partial p)_h$. Combining this with Eq. (11) gives the following derivation

$$\frac{\mathrm{d}T}{\mathrm{d}x} - \eta \frac{\lambda}{D} \frac{\rho w|w|}{2} + \frac{U_L}{c_P M}(T - T_S) = 0 \tag{20}$$

The second term in Eq. (20) represents the changes of the gas temperature along the pipeline as a result of real gas effects, being a consequence of enthalpy dependence on pressure. This term includes the work done against attractive forces between gas molecules during expansion, i.e. the Joule-Thomson effect when the processes are isenthalpic. In the range of pressure and temperature values typical for gas transmission systems this work causes the gas to cool. The third term on the left hand side of Eq. (20) reflects the changes of the gas temperature along the pipeline as an effect of heat transfer with the surroundings of the pipeline.

Substituting $\rho = p/(zRT)$ and rearranging we obtain

$$\frac{\mathrm{d}T}{\mathrm{d}x} + \left(\frac{U_L}{c_P M} + \frac{\eta \lambda z R M|M|}{2pDA^2}\right) T = \frac{U_L}{c_P M} T_S \tag{21}$$

The solution of the above linear differential equation yields

$$T_2 = \frac{\beta}{\beta + \gamma}\left[T_S - T_S e^{-(\beta+\gamma)L}\right] + T_1 e^{-(\beta+\gamma)L} \tag{22}$$

where: $\gamma = \dfrac{\eta \lambda z R M|M|}{2pDA^2}$

The average gas temperature in Eq. (12) is calculated form Eq. (23).

$$\overline{T}=\frac{1}{L}\int_0^L T\mathrm{d}x=\frac{1}{L}\left[\frac{\beta}{\beta+\gamma}T_S L+\left(\frac{T_1}{\beta+\gamma}-\frac{T_S}{(\beta+\gamma)^2}\right)\left(1-e^{-(\beta+\gamma)L}\right)\right] \tag{23}$$

By setting $\eta = 0$, Eqs. (22) and (23) reduce to the equations derived from the ideal gas model, i.e. Eqs. (15) and (16), respectively.

3. Field application: example of a gas transmission system DN1400, MOP = 8.4 MPa

Two case studies were conducted in this section in order to evaluate the predictability of the models in terms of pressure and temperature errors. The results of the numerical and analytical solutions are compared to the field data from the three sections of the Yamal-Europe pipeline on Polish territory (Fig. 1). The geometrical data were: internal diameter 1383.6 mm, length of section (1-7) 177 km, length of section (8-9) 36 km, and length of section (9-13) 112 km. A trial and error analysis under steady-state conditions was conducted to determine the pipe roughness of 1.0 µm, that produced a reasonable match with the measured field data, and a fairly steady-state flow periods in the pipeline were used to obtain an average overall linear heat transfer coefficient of 3.0 W/(m K). Two scenarios under different operating conditions of the gas transmission system, which we refer to as case study 1 and case study 2, have been considered. The initial conditions were the pressure and temperature values at the inlet nodes of the pipeline sections, and the volumetric flow rate at reference conditions (101.325 kPa, 273.15 K) in the respective sections of the pipeline. The gas composition was (mole fractions) 98.29% methane, 0.64% ethane, 0.23% propane, 0.81% nitrogen, and 0.03% carbon dioxide. The test data for the two operating scenarios are summarized in Table 1.

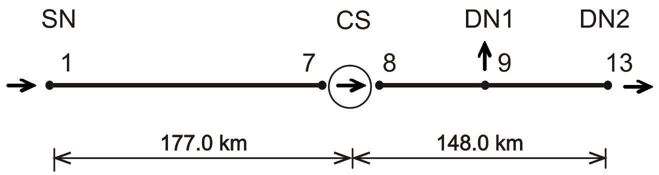

Fig. 1. Structure of the gas transportation system, SN-sending node. CS-compressor station, DN1, DN2-delivery nodes

Much of the research work has been done in the area of simplified friction factor calculations, for the simple reason that an implicit relationships were impractical when there were no personal computers or even desk-top calculators (Schroeder, 2010). It is not within the scope of this study to discuss those empirical flow formulas, and the Darcy friction factor was obtained from the iterative solution of the Colebrook – White equation (Colebrook, 1939) using Newton's method

$$\frac{1}{\sqrt{\lambda}} = -2.0 \log\left(\frac{k}{3.7D} + \frac{2.51}{Re\sqrt{\lambda}}\right) \tag{24}$$

The differential equations in the numerical models were solved with the fourth order Runge–Kutta integration method.

TABLE 1

Summary of test data (boundary conditions)

Operating conditions		Case study 1	Case study 2
Parameter type	**Node/Section**	**Parameter value**	
p (MPa)	1	7.411	7.660
	8	7.497	7.587
	9	7.174	7.226
T (°C)	1	21.0	23.8
	8	25.8	25.8
	9	20.9	20.9
Q_N (Mscm/h)	1-7	3515	3733
	8-9	3524	3749
	9-13	3189	3415
T_S (°C)	1-7	5.0	5.0
	8-13	5.5	5.6

3.1. Results and discussion

The results of the calculations of pressure and temperature distribution in the gas transmission system for the two case studies are presented in Tables 2 and 3, respectively. Table 2 shows the comparison of pressure and temperature differences between simulated and measured values at each node of the gas transmission system in case study 1, while Table 3 provides comparisons of the simulated pressure and temperature values in case study 2. These results are also given in the graphic form in Figs. 2 and 3 for pressure and temperature profiles under the first operating scenario, and in Figs. 4 and 5 for pressure and temperature profiles at the second scenario. The differences in simulated pressures, obtained from the accurate numerical model (Eqs. (2) and (3)) and the simplified numerical model (Eqs. (9) and (10)) are negligible in both case studies for the present field network, which confirms the previously reported results concerning the effect of the convective term in the momentum equation on pressure distribution in high pressure gas transmission pipelines. The simulated temperatures from the solution of the accurate and simplified numerical models are essentially the same (with the accuracy of 0.1 deg. C.) Therefore, for the case studies investigated here, the contribution of the kinetic energy term in the energy equations is found to be insignificant as well.

TABLE 2

Simulation results in case study 1

Node	Chainage (km)	Field measurements		Method/Model							
				Numerical				Analytical			
				Accurate: Eqs. (2) and (3)		Simplified: Eqs. (9) and (10)		Real gas: Eqs. (12) and (22)		Ideal gas: Eqs. (12) and (15)	
		p (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)
1	0	7.411	21.0	7.411	21.0	7.411	21.0	7.411	21.0	7.411	21.0
2	38	7.023	15.9	7.045	16.7	7.045	16.7	7.044	16.8	7.044	18.0
3	61	6.775	14.0	6.818	14.4	6.818	14.5	6.816	14.6	6.814	16.5
4	91	6.510	11.5	6.513	11.8	6.513	11.8	6.508	12.2	6.503	14.7
5	124	6.177	9.5	6.163	9.3	6.164	9.3	6.155	9.9	6.146	13.1
6	154	5.897	7.0	5.830	7.3	5.831	7.3	5.818	8.2	5.805	11.9
7	177	5.637	6.3	5.562	5.9	5.563	5.9	5.548	7.1	5.530	11.1
8	0	7.497	25.8	7.497	25.8	7.497	25.8	7.497	25.8	7.497	25.8
9	36	7.174	20.9	7.145	21.0	7.145	21.0	7.144	21.0	7.143	22.2
10	60	6.991	16.9	6.977	18.1	6.977	18.1	6.976	18.1	6.976	18.8
11	97	6.647	14.5	6.666	14.4	6.666	14.4	6.664	14.5	6.663	16.1
12	137	6.349	11.3	6.319	11.2	6.319	11.2	6.313	11.5	6.311	13.8
13	148	6.198	10.9	6.220	10.4	6.221	10.4	6.214	10.8	6.211	13.3

TABLE 3

Simulation results in case study 2

Node	Chainage (km)	Field measurements		Method/Model							
				Numerical				Analytical			
				Accurate: Eqs. (2) and (3)		Simplified: Eqs. (9) and (10)		Real gas: Eqs. (12) and (22)		Ideal gas: Eqs. (12) and (15)	
		p (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)	*p* (MPa)	*T* (°C)
1	0	7.660	25.0	7.660	23.8	7.660	23.8	7.660	23.8	7.660	23.8
2	38	7.227	18.7	7.258	19.1	7.258	19.1	7.257	19.1	7.256	20.5
3	61	6.947	15.7	7.008	16.6	7.008	16.6	7.005	16.7	7.002	18.7
4	91	6.646	11.9	6.671	13.6	6.672	13.6	6.665	14.0	6.660	16.8
5	124	6.270	9.3	6.285	10.8	6.286	10.8	6.274	11.5	6.264	14.9
6	154	5.949	6.5	5.916	8.4	5.917	8.4	5.900	9.5	5.884	13.5
7	177	5.658	5.6	5.618	6.8	5.619	6.8	5.598	8.2	5.577	12.6
8	0	7.587	25.8	7.587	25.8	7.587	25.8	7.587	25.8	7.587	25.8
9	36	7.226	20.9	7.195	21.0	7.195	21.0	7.194	21.1	7.193	22.4
10	60	7.017	16.7	7.003	18.1	7.003	18.1	7.002	18.2	7.002	19.0
11	97	6.627	14.0	6.649	14.5	6.649	14.5	6.647	14.6	6.645	16.4
12	137	6.279	10.7	6.250	11.2	6.251	11.2	6.245	11.6	6.240	14.2
13	148	6.108	10.2	6.137	10.3	6.138	10.3	6.131	10.8	6.124	13.7

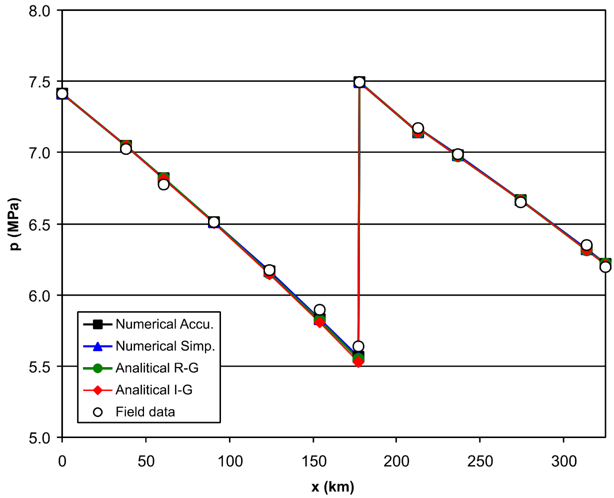

Fig. 2. Pressure distribution along the pipelines in case study 1

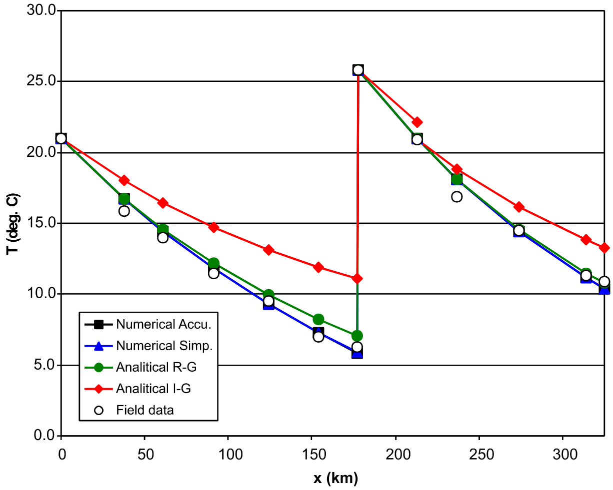

Fig. 3. Temperature profile along the pipelines in case study 1

The results of the calculations of the temperature gradients by the analytical integration show considerable differences between real gas and ideal gas models. While the real gas model shows relatively good approximation capability of temperature distribution, the ideal gas model fails to provide accurate results, with average absolute deviations in temperature much higher than those obtained from the other models. Table 4 summarizes the percentage deviations in pressure and temperature, therefore provides more insight into the accuracy of the models. The average absolute deviations in pressure (AADp) and temperature (AADT) were calculated as

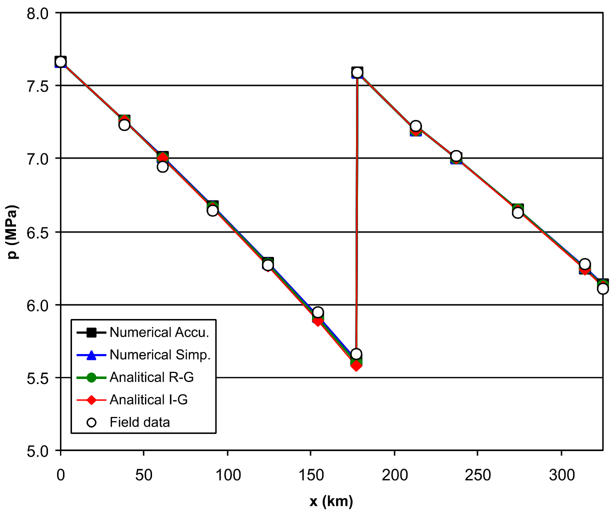

Fig. 4. Pressure distribution along the pipelines in case study 2

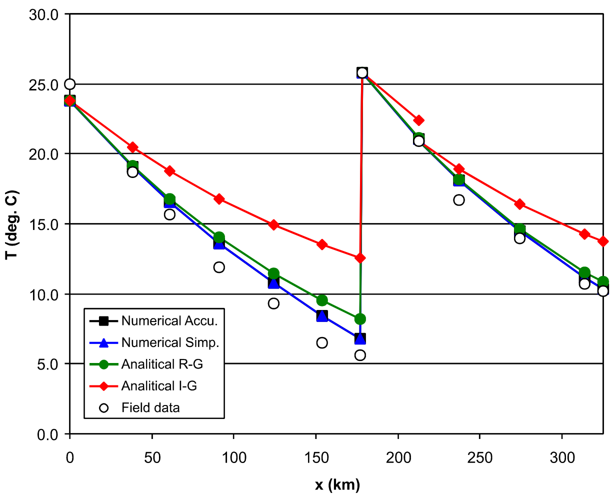

Fig. 5. Temperature profile along the pipelines in case study 2

follows: $\text{AADx} = \frac{1}{N}\sum_{i=1}^{N}\left|\left(x_i^{\text{measurement}} - x_i^{\text{model}}\right)/x_i^{\text{measurement}}\right|$, where N is the number of nodes, in which measurement data are available. The results show that while both numerical and analytical models achieve excellent results in terms of pressure gradients, this cannot be said about temperature gradients, and the temperature gradient obtained from the analytical model with ideal gas assumption in particular. For the case studies under consideration, the average deviations of temperature in this data set were 28.7% and 43.0%, respectively.

TABLE 4

The average absolute deviation in pressure (AADp) and temperature (AADT)

Operating conditions	Method/Model	*AADp* (%)	*AADT* (%)
case study 1	Numerical, Accurate: Eqs. (2) and (3)	0.5	3.4
	Numerical, Simplified: Eqs. (9) and (10)	0.5	3.4
	Analytical, Real gas: Eqs. (12) and (22)	0.5	5.5
	Analytical, Ideal gas: Eqs. (12) and (15)	0.6	28.7
case study 2	Numerical, Accurate: Eqs. (2) and (3)	0.5	10.1
	Numerical, Simplified: Eqs. (9) and (10)	0.5	10.1
	Analytical, Real gas: Eqs. (12) and (22)	0.5	16.1
	Analytical, Ideal gas: Eqs. (12) and (15)	0.5	43.0

Table 5 contains the results of the calculations of power demand in compressor station CS (Fig. 1) based on the compressor isentropic head. Apart from compressor pressure ratio, the suction temperature is a major influence on the compressor power, which is directly linked to the fuel consumption. The enthalpy values were determined from the suction pressure and temperature data resulting from the solution of the numerical and analytical flow models, and a constant discharge pressure (CS setpoint). The isentropic compression power was calculated from the expression $P = M_{8-9}(h_{8S} - h_7)$, where $h_{8S} = h(p_8, T_8^{is})$, and T_8^{is} is the temperature corresponding to the isentropic discharge state. Figures from Table 5 clearly indicate that, compared to the numerical integration approaches, the approximate calculations performed by the analytical integration resulted in a greater overestimation of the driving power necessary to compress the gas, above and below 5%, respectively. In particular, the analytical model with the simplified assumption regarding temperature distribution should be considered as less suitable for the prediction of power demand.

TABLE 5

The isentropic compression power in CS

Operating conditions	Method/Model	*P* (MW)	Deviation *DP* (%)
case study 1	Field data	26.569	–
	Numerical, Accurate: Eqs. (2) and (3)	27.844	4.8
	Numerical, Simplified: Eqs. (9) and (10)	27.825	4.7
	Analytical, Real gas: Eqs. (12) and (22)	28.290	6.5
	Analytical, Ideal gas: Eqs. (12) and (15)	29.227	10.0
case study 2	Field data	27.251	-
	Numerical, Accurate: Eqs. (2) and (3)	28.147	3.3
	Numerical, Simplified: Eqs. (9) and (10)	28.124	3.2
	Analytical, Real gas: Eqs. (12) and (22)	28.727	5.4
	Analytical, Ideal gas: Eqs. (12) and (15)	29.792	9.3

4. Conclusions

The simultaneous prediction of pressure and temperature requires coupling of the momentum and energy conservation equations. This leads to an iterative computational procedure, which fail to handle efficiently many pipeline simulation and optimization problems, such as transport capacity and fuel consumption considerations. Earlier attempts to simplify this procedure assumed either a constant temperature profile in the pipeline or an ideal gas heat capacity rates in the thermodynamic description of the pipeline flow.

This article briefly outlined issues involved in the analysis of energy equation simplifications aimed at providing explicit expressions for temperature profile along the pipeline under steady-state conditions. A realistic temperature model requires taking into consideration real gas thermodynamics, since the cooling associated with the gas expansion shows considerable influence on temperature distribution in high pressure gas transmission pipelines. This study has demonstrated that the simplified approach based on ideal gas model can be treated only as a very rough estimate of the temperature profile. The accurate prediction of the temperature gradient is essential when the goal of the optimization is to minimize the temperature dependent expression of the compressor power.

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