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Verification of Transient Gas Flow Simulation Model

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ABSTRACT

Predictions of the gas flow rate, temperature and pressure profiles along the pipeline under transient conditions are vital to the operation of gas transmission pipelines. Available simplified models for the calculation of these profiles are evaluated. Numerical solution with the method of lines is adopted to allow for estimation of the magnitude of the model terms. Sensitivity of pipeline flow model to the choice of heat transfer model, and to the accuracy of compressibility factor, heat capacity and friction factor calculations is investigated. The influence of the selection of different equations of state on pipeline line-pack results is also demonstrated. Equations of state commonly used in the gas industry were investigated, i.e.: Soave-Redlich-Kwong, Benedict-Webb-Rubin, AGA-8 and SGERG-88. The predictions from the numerical solution are compared to the field data from the Yamal-Europe pipeline.

KEYWORDS: pipeline gas flow, transient model, heat transfer model, equation of state

INTRODUCTION

Predicton of the gas flow-rate, temperature and pressure profiles along the pipelines under transient conditions requires adequate mathematical models from the class of systems with distributed parameters. Numerical methods rather than analytical ones are used for their solution. Discretization of the models is usually carried out through the finite difference methods, leading to the systems of "stiff equations", which need specific numerical methods of solution. In this article, we focus on the accuracy of a non-isothermal transient gas flow model. The impact of heat transfer model on the accuaracy flow parameters is demonstrated. The effect of the selection of different equations of state is also discussed. The results of the model solution are compared to the field data from the Yamal-Europe pipeline.

TRANSIENT GAS FLOW MODEL

The unsteady non-isothermal compressible flow in gas pipelines is described by a set of partial differential equations expressing mass, momentum and energy conservation laws, as follows

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho w)}{\partial x} = 0 \tag{1}$$

$$\frac{\partial(\rho w)}{\partial t} + \frac{\partial\left(p + \rho w^2\right)}{\partial x} = -\frac{2 f \rho w |w|}{D} - g \sin\alpha \tag{2}$$

$$\frac{\partial}{\partial t}\left[\left(u + \frac{w^2}{2}\right)\rho\right] + \frac{\partial}{\partial x}\left[\left(h + \frac{w^2}{2}\right)\rho w\right] = \rho\left(q - wg\sin\alpha\right) \tag{3}$$

where u is the internal energy per unit mass of gas (specific internal energy), h is the specific enthalpy and q is the rate of heat transfer per unit time and unit mass of gas.

Eqs. (1)−(3) may be rewritten in terms of pressure and the 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 equation of state for a real gas

$$\frac{p}{\rho} = zRT \tag{4}$$

and the thermodynamic identity

$$\mathrm{d}u = c_v \mathrm{d}T + \left[T\left(\frac{\partial p}{\partial T}\right)_v - p \right]\mathrm{d}v \tag{5}$$

the following set of hyperbolic equations with pressure, flow and temperature as a function of time and distance is obtained [1]

$$\left[\frac{1}{p}-\frac{1}{z}\left(\frac{\partial z}{\partial p}\right)_T\right]\frac{\partial p}{\partial t}-\left[\frac{1}{T}+\frac{1}{z}\left(\frac{\partial z}{\partial T}\right)_p\right]\frac{\partial T}{\partial t}+\frac{\rho_n zRT}{pa}\frac{\partial Q_n}{\partial x}=0 \tag{6}$$

$$\begin{aligned}&\frac{\partial Q_n}{\partial t}+\frac{\rho_n Q_n zRT}{pa}\frac{\partial Q_n}{\partial x}-Q_n\left[\frac{1}{p}-\frac{1}{z}\left(\frac{\partial z}{\partial p}\right)_T\right]\\&\times\left(\frac{\partial p}{\partial t}+\frac{\rho_n Q_n zRT}{pa}\frac{\partial p}{\partial x}\right)+Q_n\left[\frac{1}{T}+\frac{1}{z}\left(\frac{\partial z}{\partial T}\right)_p\right]\\&\times\left(\frac{\partial T}{\partial t}+\frac{\rho_n Q_n zRT}{pa}\frac{\partial T}{\partial x}\right)+\frac{A}{\rho_n}\frac{\partial p}{\partial x}+\frac{2fzRT\rho_n Q_n\left|Q_n\right|}{DAp}=0\end{aligned} \tag{7}$$

$$\begin{aligned}&\frac{\partial T}{\partial t}+\frac{\rho_n Q_n zRT}{pa}\frac{\partial T}{\partial x}+\frac{RT}{c_v}\frac{\rho_n Q_n zRT}{pa}zT\left[\frac{1}{T}+\frac{1}{z}\left(\frac{\partial z}{\partial T}\right)_p\right]\\&\times\left\{\frac{1}{Q_n}\frac{\partial Q_n}{\partial x}-\left[\frac{1}{p}-\frac{1}{z}\left(\frac{\partial z}{\partial p}\right)_T\right]\frac{\partial p}{\partial x}+\left[\frac{1}{T}+\frac{1}{z}\left(\frac{\partial z}{\partial T}\right)_p\right]\frac{\partial T}{\partial x}\right\}\\&-\frac{2f}{c_v D}\left(\frac{zRT\rho_n\left|Q_n\right|}{Ap}\right)^3-\frac{q}{c_v}=0\end{aligned} \tag{8}$$

The heat transfer from the gas to the surroundings has a significant effect on the gas parameters obtained from the solution of the above gas flow model. Based on a steady-state non-isothermal gas flow model in both onshore and offshore pipelines Gersten et al. [2] showed that considering heat transfer reduces uncertainties in planned transport capacities and pressure losses. In our previous work [3] isothermal and non-isothermal pipeline gas flow models were compared. It has been shown that there exists a significant difference in the pressure profile along the pipeline between isothermal and non-isothermal process. The use of an isothermal model may lead to significant errors in calculation of the energy consumption of the drivers of the compressors. The work of Modisette [4] concluded that the accuracy of the heat transfer model affects both linepack and pressure loss in gas pipelines. In generally, various methods are used for estimation of the heat transfer term q, most of which assume modification of steady-state flow expression.

Real-gas model

According to AGA8/1992 (Compressibility Factor of Natural Gas and Related Hydrocarbon Gases, AGA Report No. 8, American Gas Association, Arlington, VA.) and ISO 12213-3:1997 (Natural gas — calculation of compression factor — Part 3: Calculation using physical properties), the equation of state for the calculation of compressibility factor of natural gas is in the form of the virial expansion

$$z = 1 + B\rho_m + C\rho_m^2 \tag{9}$$

where ρ_m – molar density of the gas.

It is convenient to rewrite Eq. (9) as a series in powers of pressure instead of molar density, which would be somewhat better form considering the dependent variables of the system of Eqs. (6-8) [5]. An equivelent form used for calculation of the derivatives of compressibility factor is

$$z = 1 + B'p + C'p^2 \tag{10}$$

Virial coefficients in Eq. (10) are calculated from the original virial coefficients by equating (9) and (10) and solving the original virial expansion for p. The new virial coefficients in terms of B and C are

$$B' = \frac{B}{RT} \tag{11}$$

$$C' = \frac{C - B^2}{(RT)^2} \tag{12}$$

The AGA8/1992 and ISO 12213-3:1997 standards give constants, gas parameters and mixing rules for the calculation of the virial coefficients in Eq. (9). Soave-Redlich-Kwong (SRK) equation of state [6] and Benedict-Webb-Rubin (BWR) equation of state [7] were taken for comparison of the flow models in this study.

Heat-transfer model

In the energy equation, the heat transfer term q represents the amount of heat exchanged between unit mass of gas and the surroundings per unit time. Application of Fourier's law to calculate the overall heat-transfer between the gas and the ground for a discretization section of a pipeline yields

$$q\rho = -\frac{4U}{D}\left(T - T_{ambience}\right) \tag{13}$$

where U is an overall heat transfer coefficient. There exists an analytical steady-state solution for k for a cylinder near a half-plane, which corresponds to the geometry of a buried pipeline [8]. Nevertheless, it is a common practice to calculate k as for a set of concentric cylindrical layers with the distance between the boundary of an outer layer and the pipe equal to the burial depth of the pipe. The ambient temperature is fixed and equal to the ground temperature at the same horizontal level as the pipe axis, and at a sufficient lateral distance from the pipe. This technique for simplified heat transfer modelling and its applicability to calculate accurate temperature profiles in gas pipeline has been evaluated in the case study presented in this work.

The process of heat transfer from the gas to the surrounding environment is described using unsteady heat transfer model so that the description of heat flux could take into consideration the effect of heat capacity of the surroundings of a pipeline. Using the finite difference method, one-dimensional axial-symmetric heat exchange process can be expressed by the following set of equations, representing thermal balances of the elements – coaxial cylindrical surfaces (Fig. 1).

$$
\begin{aligned}
& q\rho = -\frac{4k_0}{D}\left(T - T_1\right) \\
& \frac{m_1 c_{p1}}{\mathrm{d}x}\frac{\partial T_1}{\partial t} = k_0\left(T - T_1\right) - k_1\left(T_1 - T_2\right) \\
& \frac{m_2 c_{p2}}{\mathrm{d}x}\frac{\partial T_2}{\partial t} = k_1\left(T_1 - T_2\right) - k_2\left(T_2 - T_3\right) \\
& \vdots \\
& \frac{m_n c_{pn}}{\mathrm{d}x}\frac{\partial T_n}{\partial t} = k_{n-1}\left(T_{n-1} - T_n\right) - k_n\left(T_n - T_{ground}\right)
\end{aligned}
\tag{14}
$$

where n is the number of discretization sections of heat-transfer area (equal to number of elements), m_i is element mass (i = 1, ... n), c_{pi} is the specific heat of element i, $m_i \bullet c_{pi}$ is the element heat capacity per pipeline unit length, dx is the discretization section of a pipeline, T_i is the element temperature and k_i is the heat transfer coefficient between elements (i –1) and i (k_0 denotes heat transfer coefficient between the gas and the first element). In case of a one-dimensional approach, the process should be modelled by a minimum of two cylindrical layers as heat capacitors of substantially different capacity; so that their time constants were significantly different (the near and distant surroundings of the pipeline could respond to temperature changes quickly and slowly, respectively). It has been assumed for the purpose of heat-transfer area discretization that every element has the same thermal resistance. Thus, in steady state, the temperature difference between consecutive ground sections are equal and the initial condition can be accurately modelled.

Solution method

Method of lines (MOL) was used for the numerical solution of the system of nonlinear partial differential equations (6)-(8). MOL proceeds with two separate steps:

- Spatial derivatives approximation. In this study finite difference technique with second order central-difference interpolation for all internal points was used.

- Time integration of the resulting ordinary differential equations (ODE). In this work, the implicit multistep Gear's method was used [9].

The system (6)-(8) is converted to the ODE system by approximation of the spatial derivatives $\frac{\partial T}{\partial x} = \Delta_X\left(T\right)$ $\frac{\partial p}{\partial x} = \Delta_X\left(p\right)$ $\frac{\partial Q_n}{\partial x} = \Delta_X\left(Q_n\right)$ with the three-point differentiation formula. Approximation of spatial derivative of pressure is given below as an example

$$
\Delta_X(\mathbf{p}) = \begin{bmatrix} \mathrm{d}p(x_0)/\mathrm{d}x \\ \mathrm{d}p(x_1)/\mathrm{d}x \\ \mathrm{M} \\ \mathrm{d}p(x_{N-1})/\mathrm{d}x \\ \mathrm{d}p(x_N)/\mathrm{d}x \end{bmatrix} =
$$

$$
= \frac{1}{2\Delta x}\begin{bmatrix} -3 & 4 & -1 & 0 & \mathrm{L} & 0 \\ -1 & 0 & 1 & 0 & \mathrm{K} & 0 \\ \mathrm{M} & \mathrm{O} & \mathrm{O} & \mathrm{O} & \mathrm{K} & \mathrm{M} \\ 0 & \mathrm{L} & 0 & -1 & 0 & 1 \\ 0 & \mathrm{L} & 0 & 1 & -4 & 3 \end{bmatrix}\begin{bmatrix} p(x_0) \\ p(x_1) \\ \mathrm{M} \\ p(x_{N-1}) \\ p(x_N) \end{bmatrix} + \mathrm{O}(\Delta x^2)
\tag{15}
$$

Above approximation is second-order correct, i.e. the truncation error is proportional to Δx^2. Apart from boundary points, the derivative of $p(x)$ approximated at point x_j is based on function values at grid points x_{j-1}, and x_{j+1}.

The implicit multistep Gear's method was used for the time integration, because the system of equations (6)-(8) require the solver which is appropriate for stiff systems of ODEs. The integrator works with a variable stepsize procedure, and controls both the global error (the error that propagates from previous steps) and the local error (introduced at the current step). Detailed discussion of the variable stepsize integrators recommended for the simulation of gas transmission networks can be found in the paper by Chua and Dew [10].

Model verification

In the case study, we present the verification of the gas flow model under transient conditions based on the field data from the Yamal-Europe pipeline on Polish territory. The 66 miles (107 km) pipeline section between national grid delivery point in Lwówek and Malnow compressor station near German border was selected for the analysis (Fig. 2). In the numerical calculations the following data were used: *Gas*: the gas is a mixture with a molar composition of CH_4 = 98.3455, C_2H_6 = 0.6104, C_3H_8 = 0.1572, i-C_4H_{10} = 0.0299, n-C_4H_{10} = 0.0253, i-C_5H_{12} = 0.0055, n-C_5H_{12} = 0.0040, N_2 = 0.0303 and CO_2= 0.7918. The density ρ_n = 0.0433 lb/ft^3 (0.695 kg/m^3); *Pipeline*: The pipeline length L = 66 miles (107 km), the pipe diameter d_o = 56 in (1422 mm). The properties of the pipe wall are listed in Table 1; *Soil*: the thermal conductivity k_{soil} = 1.18 Btu/hr·ft·°F (2.05 W/m·K), the density 125 lb/ft^3 (2000 kg/m^3), specific heat capacity c_p = 1.18 Btu/lb·°F (1420 J/kg K) and the pipe depth z = 4.92 ft (1.5 m). The soil temperature was in the range of 37.2-40.6°F (2.9-4.8°C).

The functions $p(0,t) = f_1(t)$ $T(0,t) = f_2(t)$ and $Q_N(L,t) = f_3(t)$ (boundary conditions) are shown in Figures. 3, 4 and 5, respectively.

Impact of unsteady heat transfer condition

The effect of pipeline heat transfer model on pressure and temperature values at the delivery node is presented in Figs. 6 and 7, respectively. The results show that unsteady heat transfer model with the effect of heat accumulation in the surroundings of the pipeline produces smaller amplitude of temperature fluctuations The changes of temperature are also more spread in time due to the response of the ground temperature resulting from heat accumulation. The accuracy of gas temperature prediction in the pipeline can be significantly improved in comparison with the model containing steady-state heat transfer term.

Sensitivity of pipeline flow model to selection of the equation of state

We investigate the influence of different equations of state, including AGA-8 and SGERG-88, which are frequently used in gas and petroleum industry on the results of transient gas flow modelling. For comparison, models with more universal equations of state i.e. SRK and BWR are solved to illustrate the overall gas flow model inaccuracies.

The effect of the selection of different equations of state on the flow parameters and pipeline line-pack is demonstrated in Figs. 8, 9 and 10. The results of comparison show relatively small influence of the type of the equation of state on flow parameters. The form of the equation of state might be regarded as a contributory factor in leak detection systems based on volume balance methods, in which pipeline line-pack must be accurately evaluated.

Tuning of friction factor

Assuming that Fanning friction factor is calculated using Colebrook-White equation [11], the absolute roughness of the inner surface of the pipe must be estimated. For steel pipes, the roughness from the range of 0.002-0.004 in (0.05-0.1 mm) is typical, however, in the case of pipes with internal wall surfaces covered with epoxide coatings, much lower values are appropriate. The average value of pipe roughness obtained from the measurements of the selected pipes during the construction of the Yamal-Europe pipeline was 0.00014 in (0,0035 mm). Tuning of the simulation model proved that the value of 0.0004 in (0.01 mm) provide the most accurate results (Fig. 11).

CONCLUSIONS

The study have shown that the most influential factor for the accuarcy of the model is a friction factor. Heat transfer process described by unsteady state model gives much better approximation of real precess in comparison to steady state model. Comparison of selected equations of state applied to pipeline modeling reveals that the type of the equation of state is a minor contributory factor to the accuracy of the results.

Generally, mathematical description of pipeline fluid flow requires verification of many parameters, for which it is difficult to determine appropriate values. Measured data from the pipeline SCADA system is used to validate the model with estimated values of these parameters and provide a means of establishing a reliable pipeline simulator.

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NOMENCLATURE:

A - cross-section area of the pipe,
B - second virial coefficient,
C - third virial coefficient,
c_p - specific heat at constant pressure,
c_v - specific heat at constant volume,
D - pipe diameter,
f – Fanning friction factor,
g - the net body force per unit mass (the acceleration of gravity),
h - specific enthalpy,
k - heat transfer coefficient,
L - pipeline length,
m - heat-transfer element mass,
N - number of pipeline discretisation sections,
n - number of discretisation sections of heat transfer area,
p - gas pressure,
q - rate of heat transfer per unit time and unit mass of the gas,
Q - volumetric flow rate,
R - specific gas constant,
t - time,
T - gas temperature,
u - specific internal energy,
U - overall heat transfer coefficient,
v - specific volume,
w - flow velocity,
x- spatial coordinate,
z - compressibility factor,
Z – pipeline depth.

Greek symbols
α - angle between the direction x and the horizontal,
ε - roughness of inner pipe surface,
λ - thermal conductivity,
ρ - density of the gas.

TABLES

Pipe wall structure	Thickness, in (mm)	Thermal conductivity, Btu/hr•ft•°F (W/m•K)
Internal coating	0.019 (0.5 mm)	0.30 (0.52)
Steel L480MB (X 70)	0.757 (19.22 mm)	26.2 (45.3)
External coating (polyethylene)	0.118 (3.0 mm)	0.23 (0.4)

Table 1 – Properties of pipe wall

FIGURES

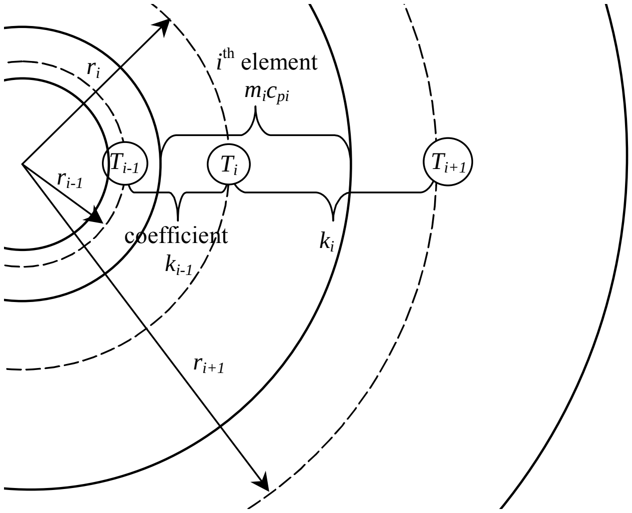

Figure 1 – Heat transfer area discretization scheme (pipeline cross section)

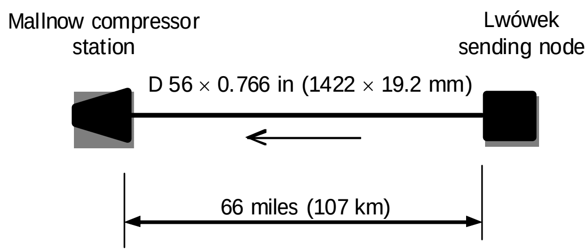

Figure 2 – Gas transmission system investigated in this study

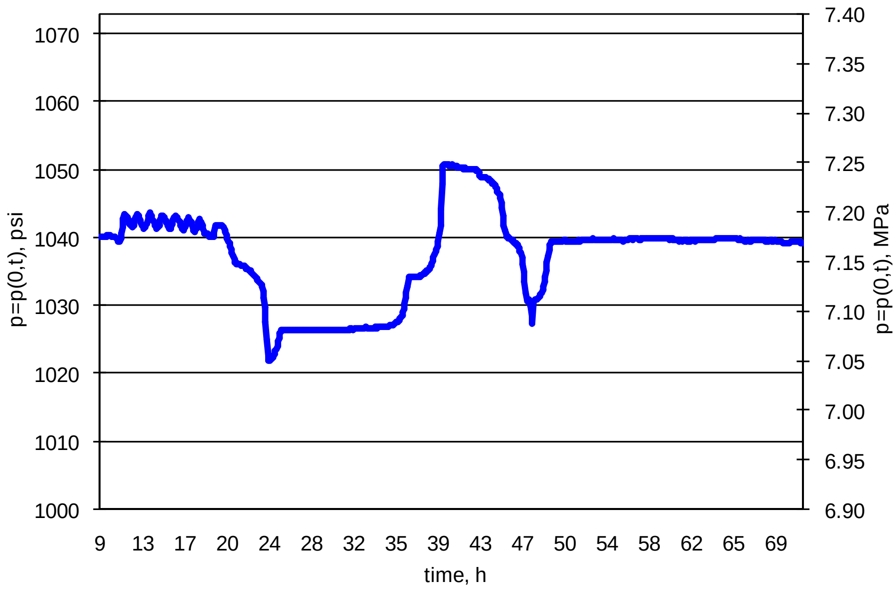

Figure 3 – Changes in pressure with time at sending node (boundary condition)

Figure 4 – Changes in temperature with time at sending node (boundary condition)

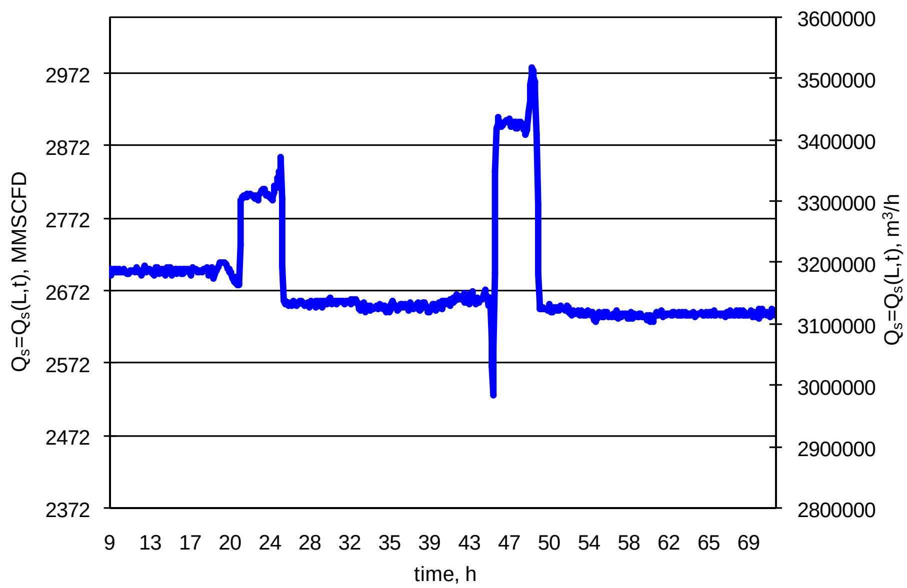

Figure 5 – Changes in flow rate with time at delivery node (boundary condition)

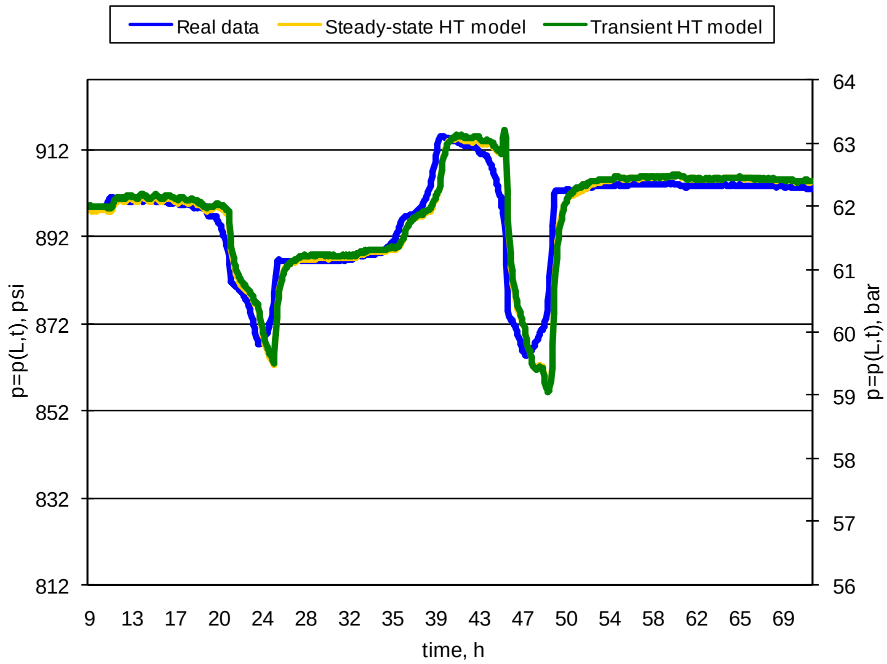

Figure 6 – Effect of heat transfer model on pressure at the delivery node

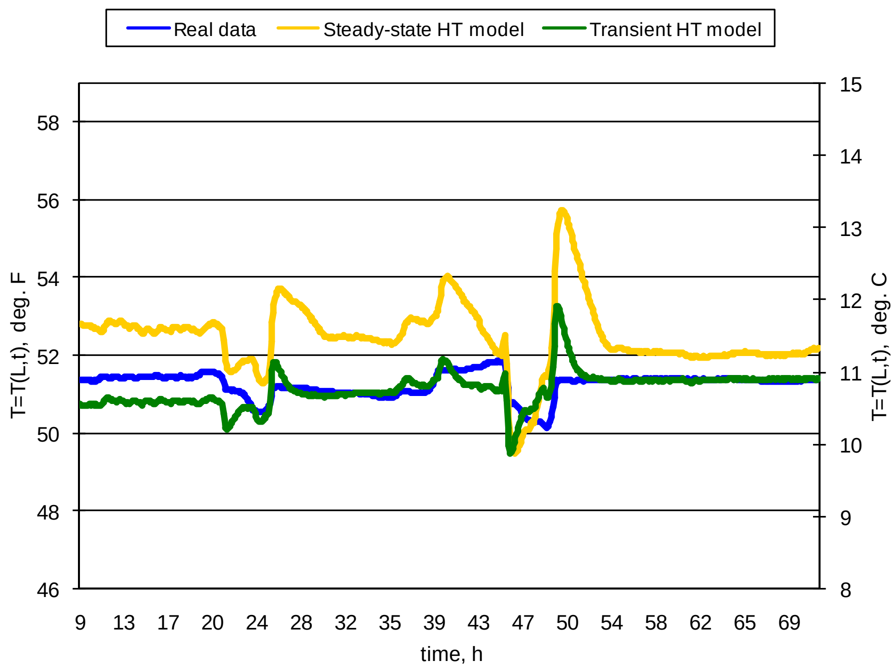

Figure 7 – Effect of heat transfer model on temperature at the delivery node

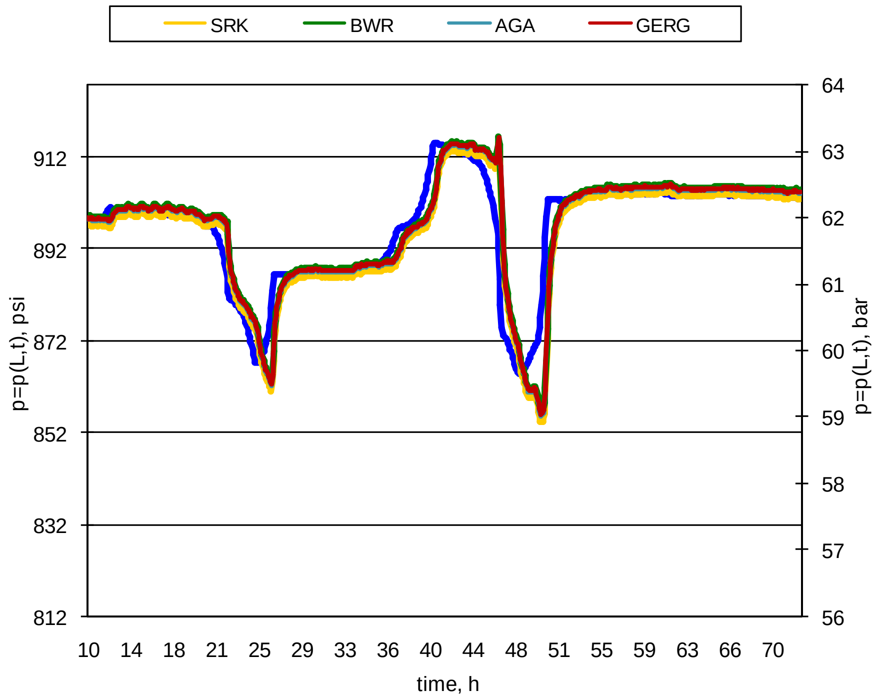

Figure 8 – Influence of different equations of state on pressure at the delivery node

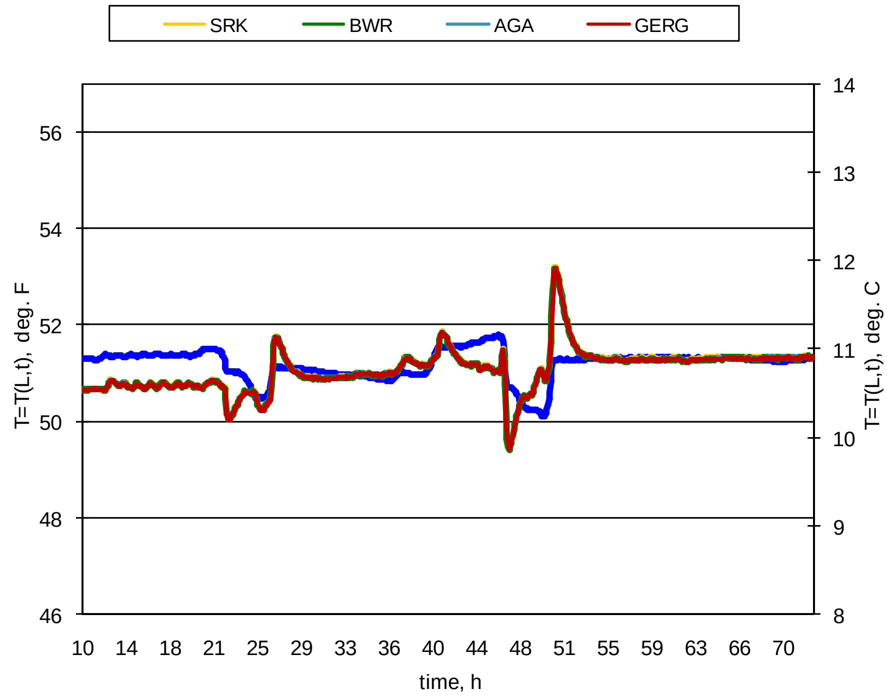

Figure 9 – Influence of different equations of state on temperature at the delivery node

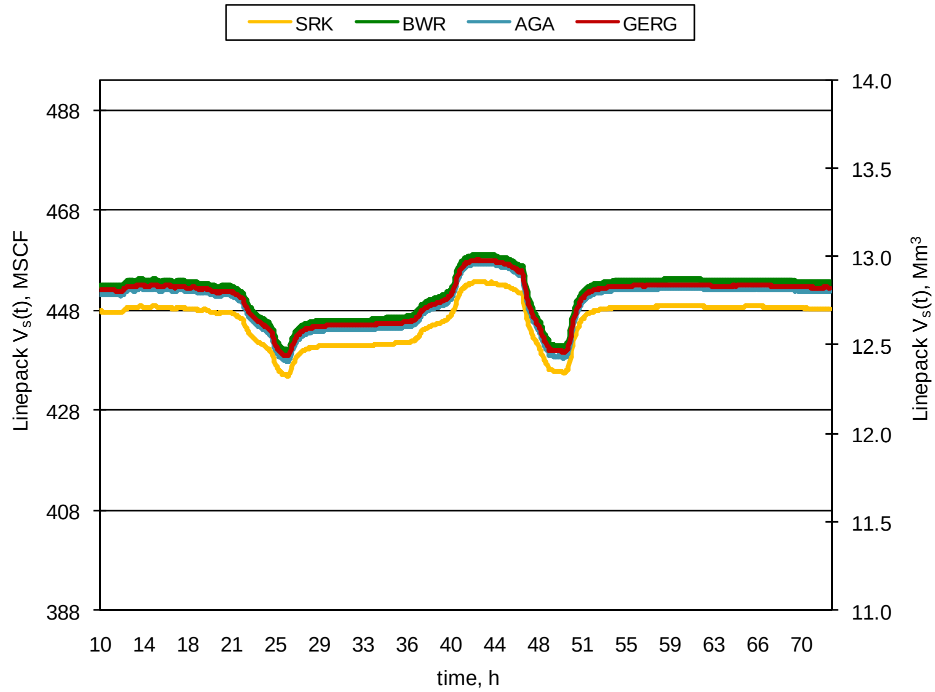

Figure 10 – Influence of different equations of state on pipeline line-pack

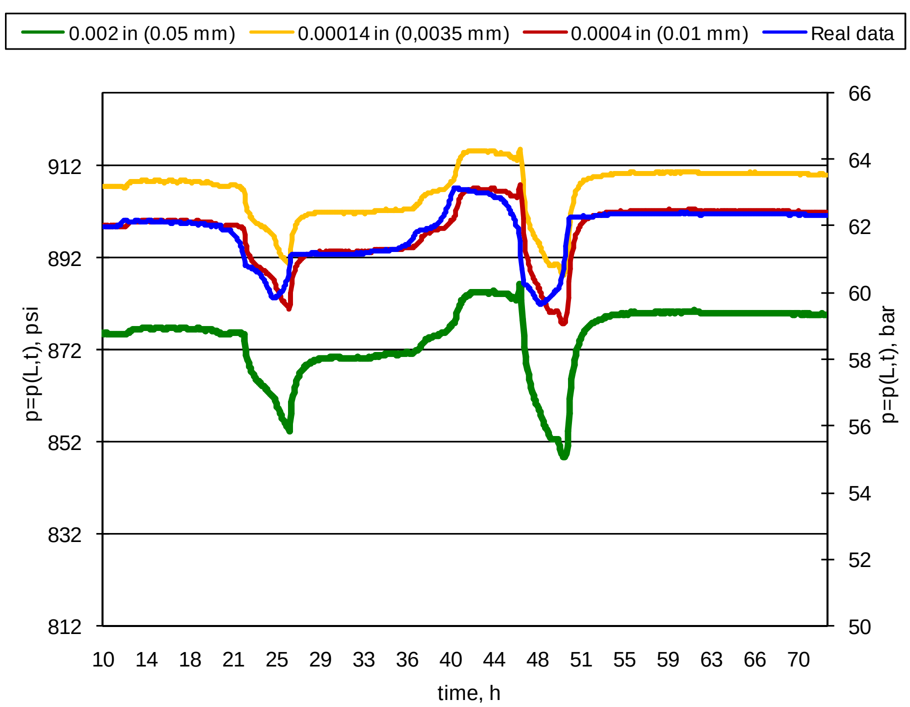

Figure 10 – Friction factor tuning - effect on pressure at the delivery node