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Validation of 1D flow model for high pressure offshore natural gas pipelines

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ABSTRACT

Transportation of natural gas through high pressure large diameter offshore pipelines is modeled by numerically solving the governing equations for one-dimensional compressible pipe flow using an implicit finite difference method. The pipelines considered have a diameter of 1 m and length of approximately 650 km. The influence of different physical parameters which enter into the model are investigated in detail. These include the friction factor, equation of state and heat transfer model. For high pressure pipelines it is shown that the selection of the equation of state can have a considerable effect on the simulated flow results, with the recently developed GERG 2004 being compared to the more traditional SRK, Peng–Robinson and BWRS equations of state. Also, including heat accumulation in the ground is important in order to model the correct temperature at the outlet of the pipeline. The flow model is validated by comparing computed results to measured values for an offshore natural gas pipeline.

1. Introduction

Natural gas may be transported over long distances through high pressure transmission pipelines. An overview of the Norwegian natural gas transport system lying in the North Sea is shown in Fig. 1. The network is operated by the Norwegian state owned company Gassco. After the gas has been processed and unwanted components are removed it is fed into long export pipelines and transported from Norway to continental Europe and the UK. The pipelines have a diameter of approximately 1 m and can be between 600–800 km in length. Measurements of the state of the gas such as pressure, mass flow, temperature and composition are available only at the inlet and outlet. To know the state of the gas between these two points one has to rely on computer models.

Transmission of natural gas through high pressure pipelines can be modeled by numerically solving the governing equations for one-dimensional compressible viscous heat conducting flow. Such mathematical models have several important applications in the gas industry. These include designing, operating and monitoring natural gas pipelines and predicting the pipeline hydraulic capacity. High accuracy in transport capacity is important to ensure optimal utilization of the network, as failure to deliver the forecasted capacity can result in penalties and a poor reputation as a gas network operator (Langelandsvik et al., 2009). They also play an integral part in software based leak detection systems. It is therefore crucial that these models are as accurate as possible, but at the same time fast and efficient as conditions in the pipeline are usually transient.

An overview of different numerical techniques used to solve the governing flow equations can be found in base literature articles (Thorley and Tiley, 1987). These include the method of characteristics, finite difference, finite volume and finite element methods. Finite difference methods have commonly been used to model the flow of natural gas through pipelines (Abbaspour and Chapman, 2008; Chaczykowski, 2010; Kiuchi, 1994), with implicit methods being preferred to explicit, as these are stable for any choice of time and spatial step.

In order to accurately model the flow through high pressure pipelines one has to solve the full non-isothermal model (Osiadacz and Chaczykowski, 2001), which implies solving the continuity, momentum and energy conservation equations for the flow. As well as solving the governing flow equations, several physical processes have to be modeled in appropriate ways. These include

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Nomenclature

A	pipe cross section [m^2]
c_p	heat capacity at constant pressure [J/(kg K)]
c_v	heat capacity at constant volume [J/(kg K)]
D	pipe diameter [m]
Fo	Fourier number
f	friction factor
g	gravitational constant [m/s^2]
h	film transfer coefficient [W/(m^2 K)]
k	heat transfer coefficient [W/(m K)]
$\dot{m}$	mass flow rate [kg/s]
p	pressure [Pa]
p_c	critical pressure [Pa]
p_r	reduced pressure
Q	heat flow [W]
R	gas constant [J/(kg K)]
Re	Reynolds number
r	pipe radius [m]
T	Temperature [K]
T_a	Ambient temperature [K]
T_c	critical temperature [K]
T_r	reduced temperature
t	time [s]
U	total heat transfer coefficient [W/(m^2 K)]
u	gas velocity [m/s]
x	spatial coordinate [m]
Z	compressibility factor
ϵ	equivalent sand grain roughness [m]
λ	thermal conductivity [W/(m K)]
ρ	density [kg/m^3]
ρ_m	molar density [kg mol/m^3]
θ	pipe inclination angle

the friction factor, equation of state and heat exchanges between the gas and the surrounding environment. Previous research has looked into the sensitivity of the pipeline gas flow model to the selection of the equation of state (Chaczykowski, 2009) and the effect of the pipeline thermal model (Chaczykowski, 2010). However, in both these cases the investigated pipeline had an inlet pressure of 8.4 MPa, which is typical that of an on-shore distribution network. Offshore pipeline like those in Fig. 1 can have an inlet pressure of up to 20 MPa, well above that typically considered in the literature.

The objective of this study is to validate the one-dimensional flow model for high pressure offshore natural gas pipelines. An implicit finite difference method is used to solve the governing flow equations. The model is validated by running simulations on an offshore natural gas pipeline and comparing numerical results to measured values. Different physical processes which enter into the one-dimensional flow model will be investigated and discussed in detail. These include the friction factor, equation of state and heat transfer between the gas and the surrounding environment. For the equation of state the recently developed GERG 2004 will be compared to the more traditional SRK, Peng–Robinson and BWRS equations of state currently used today. The heat exchange between the gas and the surroundings will be modeled using both a steady and unsteady external heat transfer model. The main difference between these two approaches is that the unsteady heat transfer model takes into account heat accumulation in the ground surrounding the pipeline.

2. Theory

2.1. Governing equations

The governing equations for one-dimensional compressible viscous heat conducting flow are found by averaging the three-dimensional versions across the pipe cross section. The result is:

Continuity

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

Momentum

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho u^2+p)}{\partial x}=-\frac{f\rho u|u|}{2D}-\rho g\sin\theta \tag{2}$$

Energy

$$\rho c_v\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}\right)+T\left(\frac{\partial p}{\partial T}\right)_\rho\frac{\partial u}{\partial x}=\frac{f\rho u^3}{2D}-\frac{4U}{D}(T-T_a) \tag{3}$$

In the momentum equation the first term on the right hand side is the friction term, where f is the friction factor. The final term is the gravity term where θ is the pipe inclination angle. In the energy equation the second term on the left hand side represents the Joule–Thomson effect, which is cooling upon expansion. On the right hand side the first term is the dissipation term, which is breakdown of mechanical energy to thermal energy. The final term represents the heat exchange between the gas and the surroundings.

The density can be related to pressure and temperature by using a real gas equation of state

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

where $Z=Z(p,T)$ is the compressibility factor. When working with natural gas pipelines, one is often interested in knowing the pressure and mass flow at the inlet and outlet. By replacing the density with pressure and introducing the mass flow rate $\dot{m}=\rho uA$, where A is the pipeline cross section, Equations (1)–(3) can be developed into partial differential equations for p, $\dot{m}$ and T (Chaczykowski, 2010). The result is:

$$\begin{aligned}\frac{\partial p}{\partial t}=&\left[\frac{1}{T}+\frac{1}{Z}\left(\frac{\partial Z}{\partial T}\right)_p\right]\left[\frac{1}{p}-\frac{1}{Z}\left(\frac{\partial Z}{\partial p}\right)_T\right]^{-1}\frac{\partial T}{\partial t}\\&-\frac{ZRT}{pA}\left[\frac{1}{p}-\frac{1}{Z}\left(\frac{\partial Z}{\partial p}\right)_T\right]^{-1}\frac{\partial\dot{m}}{\partial x}\end{aligned} \tag{5}$$

$$\begin{aligned}\frac{\partial\dot{m}}{\partial t}=&\frac{\dot{m}ZRT}{pA}\left(-2\frac{\partial\dot{m}}{\partial x}+\dot{m}\left[\frac{1}{p}-\frac{1}{Z}\left(\frac{\partial Z}{\partial p}\right)_T\right]\frac{\partial p}{\partial x}\right.\\&\left.-\dot{m}\left[\frac{1}{T}+\frac{1}{Z}\left(\frac{\partial Z}{\partial T}\right)_p\right]\frac{\partial T}{\partial x}\right)-A\frac{\partial p}{\partial x}-\frac{fZRT\dot{m}|\dot{m}|}{2DAp}-\frac{pA}{ZRT}g\sin\theta\end{aligned} \tag{6}$$