Modeling of the Flow Comparator as Calibration Device for High Pressure Natural Gas Flow Metering in Modelica

S. Singh[1], G. Schmitz[1], Bodo Mickan[2]

[1] *Institute for Engineering Thermodynamics, Hamburg University of Technology (TUHH), Hamburg, Germany*
[2] *Physikalisch-Technische Bundesanstalt, Braunschweig, Germany*

E-mail (corresponding author): sukhwinder.singh@tuhh.de

Abstract

The German national metrological institute, Physikalisch-Technische Bundesanstalt, is developing a new concept for volumetric primary standard to calibrate high pressure gas flow meters. The TUHH is supporting these R&D activities with its competence to elaborate computational models for detailed analysis of complex electromechanical systems including fluid flow aspects. The new primary standard is called Flow Comparator and uses an actively driven piston prover to measure the gas flow rate using the time the piston needs to displace a defined enclosed volume of gas in a cylinder. In Modelica a computational model is developed to investigate the Flow Comparator's dynamic behaviour and interaction with the other components in the loop. Furthermore, it allows to gather detailed information about pressure and temperature development at arbitrary chosen locations in the system with high time resolution. The validation of the developed model shows good compliance with measured piston velocity and differential pressure at the piston. The model is used to optimize the frequency inverter's control voltage trajectory to increase the available measuring time.

Nomenclature

a, b, c	fitting parameters
f	relative deviation
h	enthalpy
m	mass
p	pressure
s	position
t	time
u	internal energy
$\mathrm{u_s}$	voltage
v	velocity
A	area
F	force
L	inductance
R	resistance
Q	end effect factor
$\dot{Q}$	heat flow
V	volume
$\dot{V}$	volume flow rate
Δ	difference
ω_r	electric frequency
ψ_r	induced part flux
ρ	density
τ_m	polar pitch
ζ	pressure loss coefficient

1. Introduction

Natural gas is one of the most frequently used energy carriers worldwide and is usually transported in high pressure pipelines or as liquefied natural gas in tank vessels. For the trade with natural gas the uncertainty of high-pressure natural gas flow meters is of major importance. The uncertainty depends on the calibration chain and increases with each step. At the top of the calibration chain is the primary standard for high pressure natural gas flow metering. It is a High-Pressure Piston Prover, which is installed and operated at the calibration facility for gas meters pigsar™ in Dorsten, Germany [1,2,3]. The HPPP can be operated with inlet pressures up to 90 bar and flow rates up to 480 m³/h [3].

Due to the increasing size and flow rates of the gas flow meters and the limited operation range of the current national standard, a new concept for calibrating gas flow meters is being developed, the Flow Comparator (FC). For preliminary tests and to investigate the controllability and the usable flow rate a prototype under atmospheric conditions is used. The Flow Comparator prototype in closed loop configuration is shown in Figure 1.

Figure 1: Picture of the Flow Comparator prototype in closed loop configuration

2. Experimental Setup

The two main components of the Flow Comparator are a piston within a cylinder. The cylinder has two layers, one with magnetic properties and the other one acts as an electrical conductor. Furthermore, the piston has an integrated stator core with windings. Together the piston and the cylinder act as an asynchronous linear induction motor (LIM). The electrical power for the LIM is supplied with a cable connected to the piston. The acting force on the piston is controlled by a frequency inverter whose voltage and frequency can be controlled.

A simplified scheme of the experimental setup is shown in Figure 2. The piston has an integrated check valve to limited the pressure drop across the piston. Further more the differential pressure at the piston is measured. A specified leakage in the piston with a flow sensor measures the fluid flowing through it. The measurement and calibration are therefore based on the comparator principle [4]. As transfer standard a turbine meter (TM) is used. The temperature and pressure are measured at the turbine meter. The position of the piston is measured using a distance measuring equipment (DME). The ambient temperature and pressure as well as the temperature and pressure downstream of the cylinder are measured.

A fan is used to set the volume flow rate through the experimental setup. Due to the limited fan control accuracy the bypass valve 1 is installed. It is used to control the set volume flow rate with higher accuracy.

At the beginning the volume flow rate is set by the fan and the opening of valve 1. Valve 2 is open and the flow is bypassing the cylinder to achieve stationary conditions in the loop. When stationary conditions are obtained for a defined time the piston moves slowly upstream. At the starting point the piston is accelerated downstream until the piston velocity is the same as the fluid velocity and Valve 2 is closed. The measurement phase starts when the piston reaches the defined velocity. The volume flow rate can be calculated as shown in Equation 1.

$$\dot{V}_{FC} = \frac{V_{FC}}{\Delta_{FC} t} \tag{1}$$

During the measurement phase the discrete pulses of the turbine meter are counted. The volume flow rate indicated by the turbine meter can be calculated using the discrete pulses per time span and a proportionality factor known from previous calibration or from manufacturer specifications. The result of the calibration is the relative deviation f of the real volume flow rate to the indicated volume flow rate at certain volume flow rate and pressure. The relative deviation f is calculated as stated in equation 2.

$$f = \frac{\dot{V}^{c}_{TM} - \dot{V}^{c}_{FC}}{\dot{V}^{c}_{FC}} \tag{2}$$

The calibration accuracy of the Flow Comparator can be improved by applying corrections methods based on the measured data of the integrated sensors in the piston. In the ideal scenario the piston velocity is the same as the fluid flow velocity for the entire measurement phase and the differential pressure and the velocity across the piston would be zero. But in the experimental setup small differences of the piston velocity to the fluid velocity occur. The first correction method is based on the differential pressure across the piston. A non-zero differential pressure at the piston results into a leakage around the piston. A relationship between the differential pressure and the leakage can be derived by appropriate experiments and can be used to correct the volume displaced by the piston. The relationship is shown in Equation 3.

$$\dot{V}_{leak,\Delta p} = a\Delta p \tag{3}$$

The correction method based on the differential pressure is practical for relatively high leakage flows. For smaller leakage flows the integrated flow velocity sensor is used. The relationship between indicated velocity by the flow sensor and leakage flow is shown in Equation 4.

$$\dot{V}_{leak,v} = bv^2 + cv \tag{4}$$

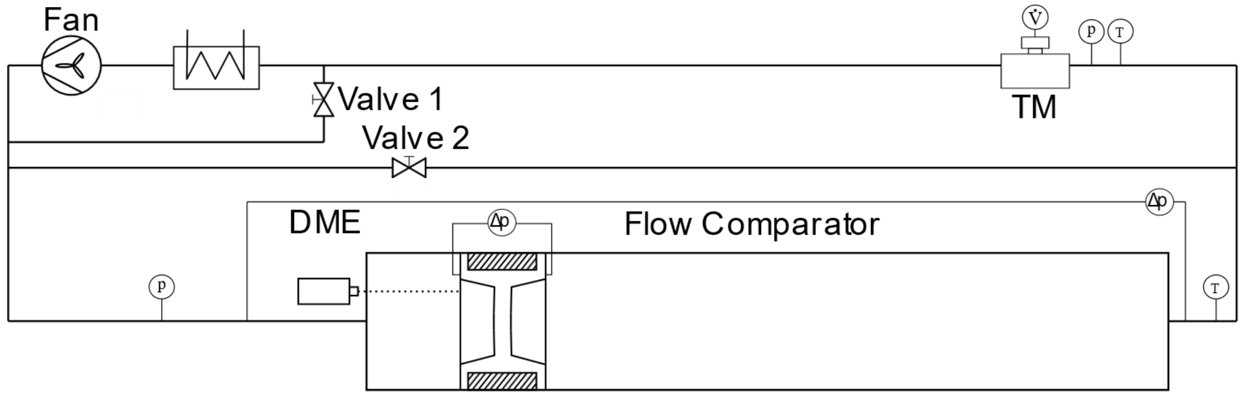

Figure 2: Scheme of the experimental setup with the Flow Comparator in closed-loop configuration

3. Modelica

Modelica is an object-oriented, non-proprietary and equation-based language to conveniently model components of different physical domains which can be connected to model complex physical systems. The models are described by differential, algebraic and discrete equations. The first specification for the Modelica language was released in 1997. It is developed by the non-profit Modelica Association. The Modelica Association is also responsible for the development of the opensource Modelica Standard Library (MSL) which contains about 1600 model components and 1350 functions from many domains [5].

Several components of the MSL containing equations of different physical domains, e.g. fluid flow, thermodynamics, tribology and electrical engineering could be used in the developed model to describe the Flow Comparator's dynamic behavior. The components are easily reusable due to the equation-based and object-oriented modelling approach. The media model used in the models can be easily exchanged allowing to test the model with atmospheric air as well as high pressure natural gas.

Furthermore, the equation-based modelling minimizes the error of rearranging equations and increases the readability of the written code. The physical equations are transformed and rearranged automatically by a compiler into a mathematical equation system which is solved with an algorithm. The symbolic index reduction as well as the solving of the equation system is usually executed by a simulation tool. The simulation environment used to develop and evaluate the Flow Comparator model in Modelica is Dymola® 2019.

4. Model description

A graphical representation of the developed model is shown in Figure 3. The assumptions used in the model are [6]:

- pressure losses are proportional to the dynamic pressure
- one dimensional gas flow
- adiabatic system
- neglection of potential energy
- heat transfer can be neglected in comparison to the convective energy transport

The model extension of the model presented in [7] and therefore the bypass is neglected in the first modelling approach.

In the model, the measuring cylinder is divided into one volume upstream of the piston and one volume downstream of the piston. The enclosed gas volumes of the measuring pipe depend on the position of the piston and change with piston movement. The measuring pipe volumes can store mass m, internal energy $m{\cdot}u$ and momentum $m{\cdot}v$ as described in Equation 5-7.

$$\frac{dm}{dt} = \dot{m}_i + \dot{m}_{i+1} \tag{5}$$

$$\begin{aligned}\frac{d}{dt}mu = \dot{m}_i\left(h_i + \frac{v_i^2}{2}\right) + \dot{m}_{i+1}\left(h_{i+1} + \frac{v_{i+1}^2}{2}\right) \\ + \dot{V}_i\left(\frac{p_{i+1} - p_i + p_{f,i+1} - p_{f,i}}{2}\right) \\ + \dot{Q}_i\end{aligned} \tag{6}$$

$$\begin{aligned}\frac{d}{dt}mv = \dot{m}_i|v_i| + \dot{m}_{i+1}|v_{i+1}| \\ - A(p_{i+1} - p_i) - A(p_{f,i+1} - p_{f,i})\end{aligned} \tag{7}$$

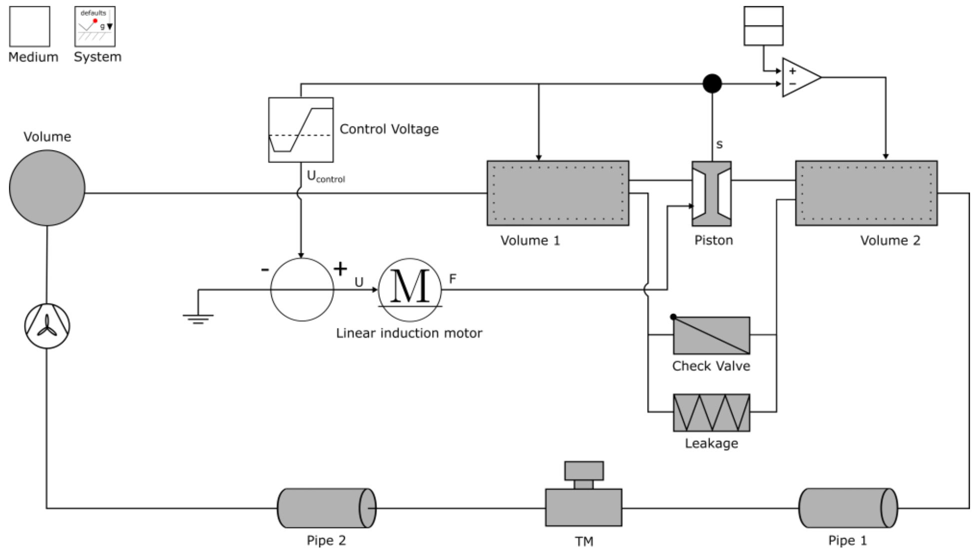

Figure 3: Graphical representation of the computational model

In direction of the fluid flow a spatial discretisation is applied to resolve the fluid flow in the measuring cylinder with higher accuracy. For this the finite volume method with the staggered grid approach is used. In a staggered grid the scalar variables are located in the control volume centre while the velocity and momentum variables are stored on the cell faces. The pressure loss is calculated using a wall friction model from the MSL. A heat port is included in the model which can be connected to a heat port of another model. This could be the environment or the piston. The heat flow is calculated based on the heat transfer model of the MSL.

The piston's motion is determined by Equation 8.

$$m_P \ddot{s}_P = p_1 A_P - p_2 A_P - F_{F,P} - F_{F,C} + F_{LIM} \quad (8)$$

The friction $F_{F,P}$ is calculated using a constant roll resistance coefficient c_R. The resistance due to the connection cable $F_{F,C}$ depends on the position and is calculated using the weight of the connection cable.

The movement of the piston is controlled by the force induced by the linear induction motor. The LIM is modelled using the space-vector equivalent circuit shown in Figure 4. The main differences to the equivalent circuit of a rotatory induction motor is the transversal branch. These are the eddy current resistance and the magnetizing inductance. The differences occur due to the end effect phenomenon in a LIM which influences the magnetizing inductance and the resistance. This is taken into effect by the so-called end effect factor Q which is defined in Equation 9 [8].

$$Q = \frac{\tau_m R_r}{(L_m + L_{\sigma r})v} \quad (9)$$

The end effects increase with the air-gap thickness (results in higher leakage inductance) as well as with higher machine speed and reduces with increasing inductor length [8]. The magnetizing inductance and the resistance vary with the term $f(Q)$:

$$f(Q) = \frac{1 - e^{-Q}}{Q} \quad (10)$$

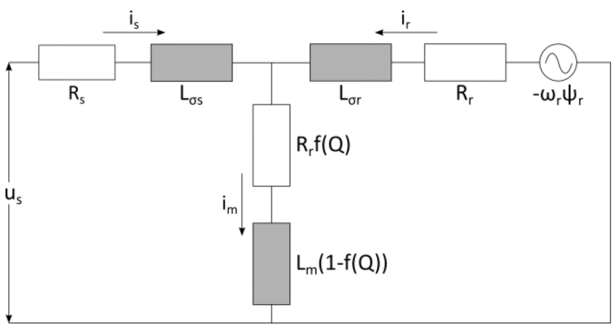

Figure 4: Space-vector equivalent circuit of the LIM

The control voltage depends on the position of the piston and changes sign and value at the starting position. It has a specified stopping time t_s and rising time t_r. The frequency inverter uses the control voltage and a constant U/f characteristic to determine the voltage for the LIM.

The turbine meter uses a constant pressure loss coefficient ζ_{TM} to model the pressure loss occurring in the turbine meter. Furthermore, the model uses a relationship between indicated volume flow rate and real volume flow rate as described in [9].

The pipes are modelled using the DynamicPipe model of the MSL. The model uses balance equations for the mass m, the momentum $m \cdot v$ and the internal energy $m \cdot u$.

5. Validation

The accuracy of the Flow Comparator model is highly relevant for the prediction of the dynamic behavior and interaction of the closed-loop system. It is affected by the aforementioned general assumptions, the accuracy of the given LIM parameters, the assumptions for the friction force, and further simplifications.

Experiments are conducted as described in Section 2 and the measurement data is used for model validation. The model is only validated for the part of the experiment where the piston moves forward and the actual calibration of the meter under test is executed. For the experiment and the simulation, the same control voltage trajectory is used to validate the model of the LIM and the frequency inverter.

In Figure 5 the measured and simulated piston velocity for three different control voltages is shown. For all control voltages the simulation has a lower acceleration than the measurement during the starting phase. After the piston accelerates to the velocity linked with the used control voltage, the simulation and measurement data have a similar value for all control voltages. Also, the decrease in piston velocity due to the increasing resistance force of connection cable shows a similar behavior in simulation and measurement.

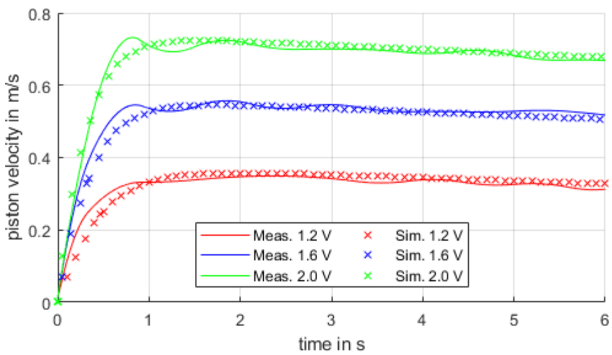

Figure 5: Comparison of the piston velocity over the time in the model and measured data for different control voltages at zero volume flow rate

The piston velocity as well as the differential pressure at the piston for a volume flow rate of $\dot{V}$ = 100 m³/h is shown in Figure 6. The frequency inverter control voltage is set to achieve zero differential at the piston. The simulation is done with the same control voltage. Overall a good accordance of measurement data and simulation is achieved. But the differential pressure at the start of the measurement phase differs in measurement and simulation. This is due to the missing bypass in the simulation and therefore the differential pressure starts with a negative value. Furthermore, in the simulation the differential pressure at the piston increases little more than in the measurement data. A similar behavior is observable for the piston velocity as it decreases faster in the simulation than in the measurement.

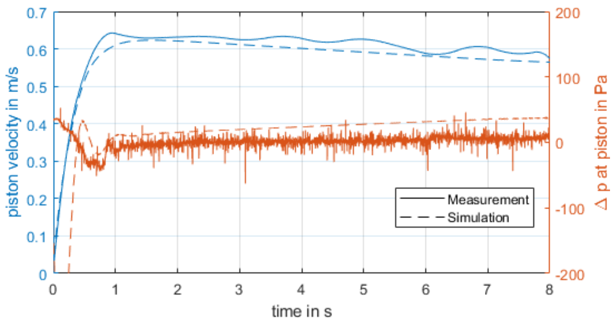

Figure 6: Comparison of velocity and differential pressure at the piston for a volume flow rate of $\dot{V}$ = 100 m³/h and a control voltage of 1.8 V

6. Optimization

The model is used to optimize the control voltage trajectory of the frequency inverter to achieve maximum available measuring time with zero differential pressure at the piston. For this the trajectory in Figure 7 is applied to the model. It has an overshoot of control voltage at the beginning and an increasing slope during the measuring phase to offset the connection cable resistance.

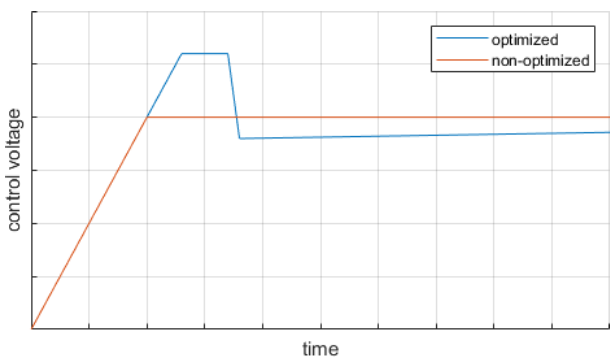

Figure 7: Exemplary control voltage trajectories for non-optimized and optimized case

In Figure 8 the simulation result for the optimized and non-optimized control voltage trajectory is shown. The piston velocity is lower in the optimized case and remains at an almost constant value throughout the simulation. The overshoot in control voltage also appears in the piston velocity. Due to this, the differential pressure initially decreases more than in the non-optimized case. But after the decrease the differential pressure in the optimized simulation decreases to zero and stays at that value for remaining simulation.

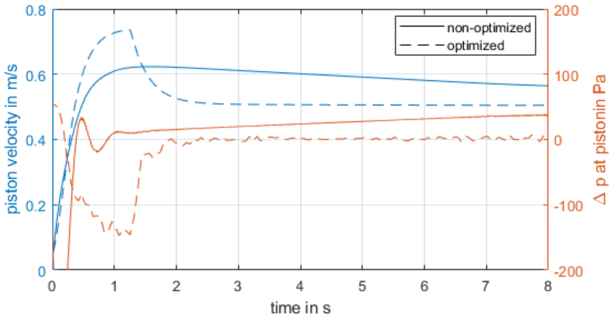

Figure 8: Comparison of velocity and differential pressure at the piston for a volume flow rate of $\dot{V}$ = 100 m³/h with optimized and non-optimized frequency inverter control voltage trajectory

7. Conclusion

The Flow Comparator in loop configuration is modelled using the modelling language Modelica®. The implemented model is successfully validated against measurement data of the actual flow comparator prototype. As application example a simple optimization of the control voltage trajectory to maximize the available measuring time is conducted. This result of optimization will allow to extend the upper limits of flow rate usable for calibrations. Furthermore, the possibility to gather detailed information about pressure and temperature development at arbitrary chosen locations in the system with high time resolution enables much better and more reliable statements about the accuracy of flow rate measurement with this system.

The model of the linear induction motor describes the electromechanical interaction. In future work, it will be essential to extend the model by heat transfer from the motor components to the gas to complete the modelling of the overall thermodynamic performance of the piston prover.

Furthermore, the bypass needs to be considered in the model. This would allow the model to be used for actual predictions of the Flow Comparator in loop configuration with the desired event sequence.

Also, the friction force of the piston needs to be measured more accurately. This includes the position depending friction force, the velocity depending friction force as well as the increase of counter force due to connection cable weight. Based on this a more detailed optimization approach can be used to counteract small deviations which would affect a uniform piston velocity.

References

[1] G. Schmitz and A. Aschenbrenner. „Experience with a Piston Prover as the new primary Standard of the Federal Republic of Germany in High-Pressure Gas Metering." Proceedings of the 18th World Gas Conference.

[2] Physikalisch-Technische Bundesanstalt PTB. Prüfschein der Rohrprüfstrecke. 1991.

[3] Physikalisch-Technische Bundesanstalt PTB. PTB Mitteilungen. Special Issue Volume 119 no.1, 2009.

[4] B. Mickan and R. Kramer: Evaluation of two new volumetric primary standards for gas volume established by PTB, 7th International Symposium on Fluid Flow Measurement (ISFFM) 2009, Anchorage, Alaska.

[5] The Modelica Association, Modelica Language https://modelica.org/

[6] M. von der Heyde, G. Schmitz and B. Mickan. Modeling of the German National Standard for High Pressure Natural Gas Flow Metering in Modelica. In Proceedings of the 11th International Modelica Conference, Versailles, France, September 21-23, 2015.

[7] S. Singh, G. Schmitz and B. Mickan "Modeling of the Flow Comparator Prototype as New Primary Standard for High Pressure Natural Gas Flow Metering" in Proceedings of the 13th International Modelica Conference, Regensburg, Germany.

[8] Sung, Jeong-Hyoun and Kwanghee Nam. „A new approach to vector control for a linear induction motor considering end effects." Conference Record of the 1999 IEEE Industry Applications Conference, 3-7 Oct. 1999, Phoenix, AZ, USA.

[9] B. Mickan, R. Kramer, and V. Strunck. Transient response of turbine flow meters during the application at a high pressure piston prover. In 15th Flow Measurement Conference (FLOMEKO). Linköping University Electronic Press, 2010.