OPTIMISING THE EFFICIENCY OF NATURAL GAS TRANSPORT SYSTEMS BY ACCOUNTING FOR THE REAL GAS COMPRESSOR WORK

František RIDZOŇ, Vojtech MOLNÁR

Faculty of Mechanical Engineering, Slovak University of Technology in Bratislava,
Nám. slobody 17, 812 31 Bratislava, Slovakia,
frantisek.ridzoň@stuba.sk, vojtech.molnar@stuba.sk

ABSTRACT

Enthalpy equations are employed to consider how pressure influences the state of natural gas. It was shown, that the analysis of a gas compressor working under typical conditions that occurs in natural gas transportation systems must be based on a thermodynamic real gas relationship. Comparing the results of the calculations of the work of the compressor with the experimental results of the work done by the compressor, shows very good correlation. This method can be also applied to the case of expansion turbines used in the gas industry for reducing high pressure to an intermediate level.

Key words: enthalpy, absorbed power, aerodynamic power, gas compressibility factor, real gas residual function

Nomenclature

Symbols
c – absolute flow velocity
c_p – isobaric specific heat capacity
h – enthalpy
$\dot{m}$ - mass flow
N - speed
p - pressure
P – absorbed power
P_v – aerodynamic power
Q – volumetric flow
r – gas constant
s – entropy
t - temperature
T – absolute temperature
v – specific volume
z – gas compressibility factor
κ - isentropic exponent
μ_m - mechanical efficiency
ρ - density

Subscripts
0 – total value
1 – suction
2 – discharge
ideal – ideal gas
real – real gas
s – isentropic
std – standard conditions

Superscripts
$-$ - averaged value

INTRODUCTION

In the gas industry gas compressors are used to compress high flow volume rates of natural gas at high pressure levels, and with a low pressure ratio. This results, obviously, in high compression work. The compressors are of a radial type and usually powered by gas turbines or electro drives. The analysis of the power required is based on the change of the state and transport properties of the compressed gas. The thermodynamic properties of natural gas differ significantly from the properties of ideal gas, mainly at high pressure levels. In such cases, the equation describing the state of an ideal gas does not reflect the thermodynamic properties of natural gas with sufficient accuracy. The exact determination of the work of the compressor for natural gas transportation is of crucial significance in optimising the efficiency of transportation systems. Therefore a computational method was developed to support the control system of natural gas transportation installations. The contribution of the new method is demonstrated by comparing it with experimental data obtained from natural gas compressor testing.

IDEAL GAS RELATIONSHIP FOR A GAS COMPRESSOR

The state of any gas mixture consisting of known components can be described using any two of the following: pressure, density or temperature. State equations approximate these relationships. The equation can also be used to calculate enthalpy and entropy from the pressure and the temperature. The most simple equation of state is the equation for an ideal gas

$$p = \rho r T \tag{1}$$

The actual head (isentropic work) for a gas compressor (Fig. 1) and of an ideal gas consisting of known components is basically calculated from overall conditions, but can usually be approximated by its static conditions:

$$\Delta h_0 = h(T_{02}) - h(T_{01}) = \bar{c}_p (T_{02} - T_{01}) \approx \bar{c}_p (T_2 - T_1) = \Delta h \tag{2}$$

Similarly the entropic head of the compressor in the case of an ideal gas is given by:

$$\Delta h_{0s} = \bar{c}_p T_{01} \left[\left(\frac{p_{02}}{p_{01}} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \approx \bar{c}_p T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] = \Delta h_s \tag{3}$$

where the average value of isobaric specific heat capacity is calculated from:

$$\bar{c}_p = \frac{1}{T_2 - T_1} \int_1^2 c_p(T)\,dT \tag{4}$$

The isobaric specific heat capacity at the temperature T, and ideal gas is assumed to be defined as follows:

$$c_p(T) = A + BT + CT^2 + DT^3 \tag{5}$$

where A, B, C a D are gas dependent constants.

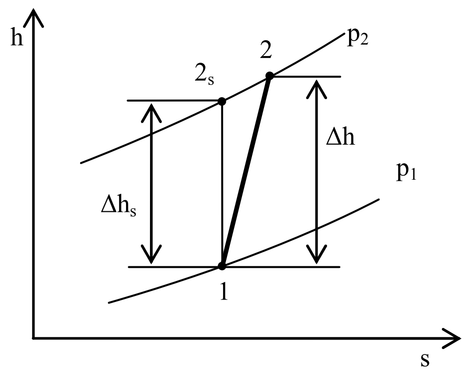

Fig. 1 Static thermodynamic change behaviour in gas compressor.

REAL GAS BEHAVIOUR FOR A GAS COMPRESSOR

The real gas behaviour for a gas compressor can be expressed by the residual function of the real gas mixture. Real gas thermodynamic properties such as enthalpy and entropy can be derived from the relationship between the isobaric specific heat capacity $c_{p,}$ as a function of temperature at standard pressure conditions, and an equation of state expressed in terms of the compressibility [1]. Assuming that the enthalpy is a function of both temperature and pressure

$$h = h(T, p) \tag{6}$$

From I. thermodynamic theorem we obtain

$$dh = Tds + vdp \tag{7}$$

Along an isotherm, this may be expressed as

$$\left(\frac{\partial h}{\partial p}\right)_T = T\left(\frac{\partial s}{\partial p}\right)_T + v \tag{8}$$

Substituting Maxwell relations

$$\left(\frac{\partial s}{\partial p}\right)_T = -\left(\frac{\partial v}{\partial T}\right)_p \tag{9}$$

in the equation (8) gives

$$\left(\frac{\partial h}{\partial p}\right)_T = \left[v - T\left(\frac{\partial v}{\partial T}\right)_p\right] \tag{10}$$

Total derivative of the enthalpy

$$dh = \left(\frac{\partial h}{\partial T}\right)_p dT + \left(\frac{\partial h}{\partial p}\right)_T dp \tag{11}$$

By definition, the isobaric specific heat capacity is defined as

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p \tag{12}$$

Substitution of equation (12) and equation (10) into the enthalpy total derivative yields:

$$dh = c_p dT + \left[v - T\left(\frac{\partial v}{\partial T}\right)_p\right] dp \tag{13}$$

For an ideal gas $(pv = rT)$, the second term vanishes:

$$\left[v - T\left(\frac{\partial v}{\partial T}\right)_p\right]_{ideal} = 0 \tag{14}$$

and it gives the well known simplified relation for enthalpy. Substituting the real gas equation of state $(pv = zrT)$, this term is often referred to as the residual function. It is related to the derivative of the compressibility and may be expressed as:

$$\left[v - T\left(\frac{\partial v}{\partial T}\right)_p\right]_{real} = \frac{-rT^2}{p}\left(\frac{\partial z}{\partial T}\right)_p \tag{15}$$

This results in a more general expression for the enthalpy, which is shown now to be a function of both pressure and temperature:

$$dh = c_p dT - \frac{rT^2}{p}\left(\frac{\partial z}{\partial T}\right)_p dp \tag{16}$$

This equation determines a change in enthalpy between two states. It is accomplished along an isobar for the first term and then along the isotherm for the second term. The change in static enthalpy between two state points is given by

$$\Delta h = \int_1^2 c_p dT - \int_1^2 \frac{rT^2}{p}\left(\frac{\partial z}{\partial T}\right)_p dp \tag{17}$$

A similar analysis for the entropy s = s(p,T) can be done, and results in the relationships for the total derivative of entropy

$$ds = \frac{c_p dT}{T} - \left[\frac{zr}{p} + \frac{rT}{p}\left(\frac{\partial z}{\partial T}\right)_p\right] dp \tag{18}$$

Finally, integration is used to obtain the entropy difference between two state points

$$\Delta s = \int_1^2 \frac{c_p dT}{T} - \int_1^2 \left[\frac{zr}{p} + \frac{rT}{p}\left(\frac{\partial z}{\partial T}\right)_p\right] dp \tag{19}$$

Once again, the residual function was demonstrated here to be a function of compressibility. The integration of equation (18) to determine the change in entropy is accomplished also by isobarically integrating the temperature dependent first term and isothermally integrating the pressure dependent second term.

The gas compressor head (Fig. 1) can be determined from the measurement of suction and discharge pressure and temperature. The relationship between the pressure, temperature, and the enthalpy is defined by the equations of state described previously.
Using the derived equations, the relevant enthalpies for the suction, the discharge, and the isentropic discharge state can be computed [2].
The actual head is

$$\Delta h = h(T_2, p_2) - h(T_1, p_1) \tag{20}$$

The isentropic head is calculated from:

$$\Delta h_s = h[p_2, s_1(T_1, p_1)] - h(T_1, p_1) \tag{21}$$

ABSORBED COMPRESSOR POWER

The actual flow Q can be calculated from standard flow Q_{std} (t_{std} = 15°C, p_{std} = 101325 Pa) or mass flow

$$Q = \frac{\rho_{std}}{\rho} Q_{std} = \frac{\dot{m}}{\rho} \tag{22}$$

where the density is $\rho = \dfrac{p}{z(T, p)rT}$

Aerodynamic (internal) power of the compressor

$$P_v = \rho Q \Delta h = \frac{p}{zrT} Q \Delta h = \rho_{std} Q_{std} \Delta h \tag{23}$$

The typical amount of mechanical losses is about $1 \div 2\%$ of the absorbed power. By introducing mechanical efficiency, the absorbed compressor power becomes:

$$P = \frac{P_v}{\mu_m} = \frac{\rho_{std} Q_{std} \Delta h}{\mu_m} \tag{24}$$

For directly driven compressors, the absorbed power P_m is exactly equal to the power at the power turbine output shaft. Using special equipment, these properties can be experimentally determined.
The relative derivation of the absorbed compressor power P calculated from equation (24) and from measured values P_m is given by:

$$\Delta P = \frac{P - P_m}{P_m} .100\% \tag{25}$$

COMPARISON OF CALCULATED AND MEASURED COMPRESSOR POWER

A computational code was developed to perform the calculation of equations (22), (23), (24) and (25). Included are subroutines for calculation of the thermodynamic properties (equations (20), (21)) of all natural gas components. The actual natural gas composition is input into the code. A Nuovo Pignone's radial compressor installation in a natural gas transportation system was selected for the present experiments. Figure 2 shows the schematic location of the measurement points for the thermodynamics properties. Measurement of the absorbed compressor power P is achieved by means of a torque meter. The static pressure and temperature p_1, T_1 was measured near the compressor inlet. The static pressure and temperature p_2, T_2 was measured at the compressor outlet. The location of the point of measurement of the volumetric flow was determined by the possibilities offered by the pipeline installation and the static pressure and temperature measuring equipment.

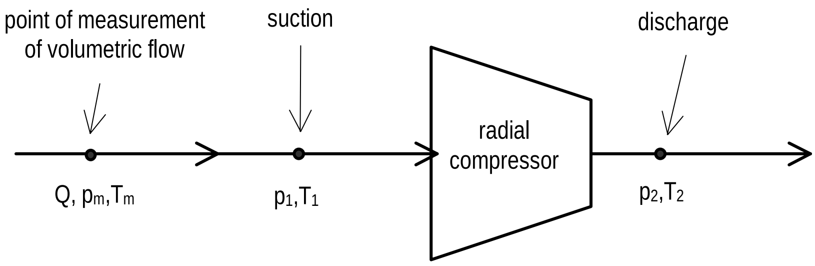

Fig. 2 Points of measurement.

Figures 3, 4 and 5 show in absolute scale the comparisons of the measured absorbed compressor power P_m with the calculated power P. The mechanical efficiency is intended in the calculation to be $\mu_m = 0{,}98$. The calculation was carried out for ideal gas assumption and real gas assumption. The natural gas composition used for the calculation is given in table 1. The agreement between the measured values and calculated values with real gas assumption is within 2,5%. Discounting the pressure relation in calculation (ideal gas) increases the deviation up to 15%.

Table 1. Natural gas composition as % volume fraction

Methane	97,23%
Ethane	1,07%
Propane	0,38%
Butane	0,22%
Nitrogen	0,77%
Carbon dioxide	0,33%

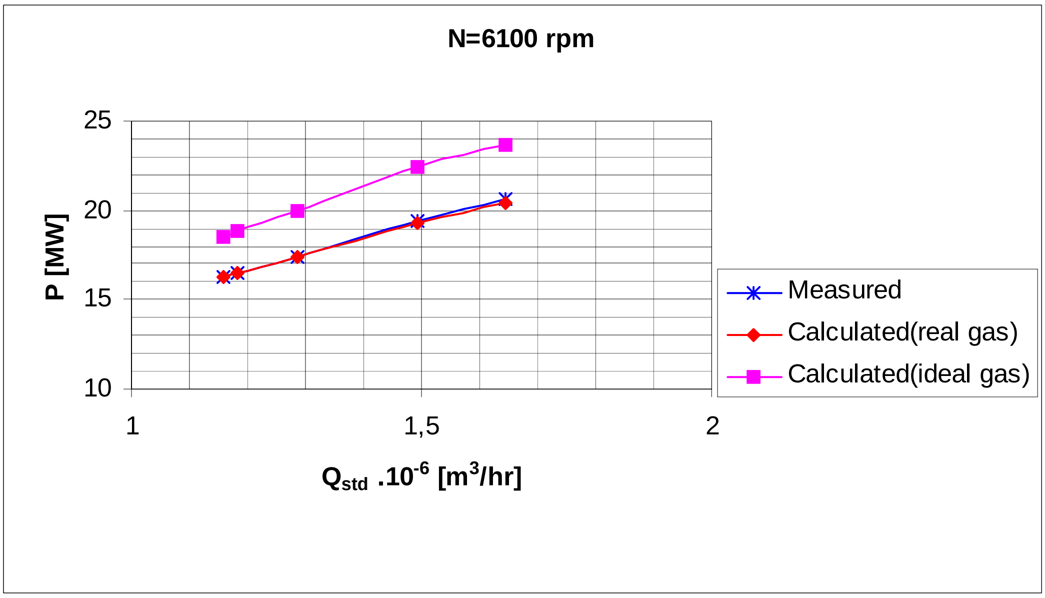

Fig. 3 Absorbed compressor power (N=6100 rpm, t_1 ~13,8°C, p_1 ~4,9MPa, p_2/p_1 = 1,48-1,50263).

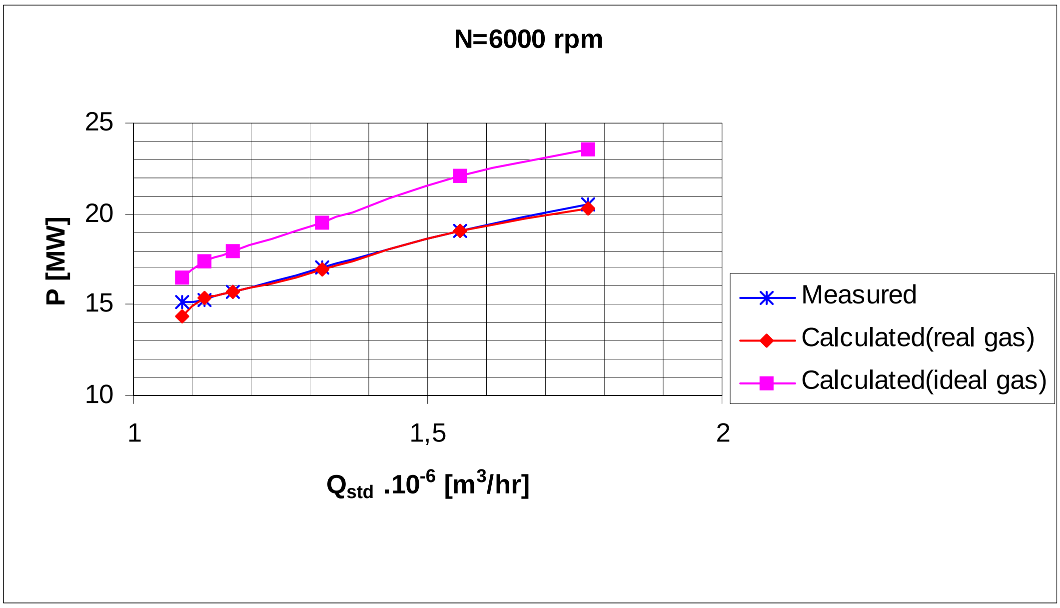

Fig. 4 Absorbed compressor power (N=6000 rpm, t_1 ~13,8°C, p_1 ~4,9MPa, p_2/p_1 = 1,436-1,4865).

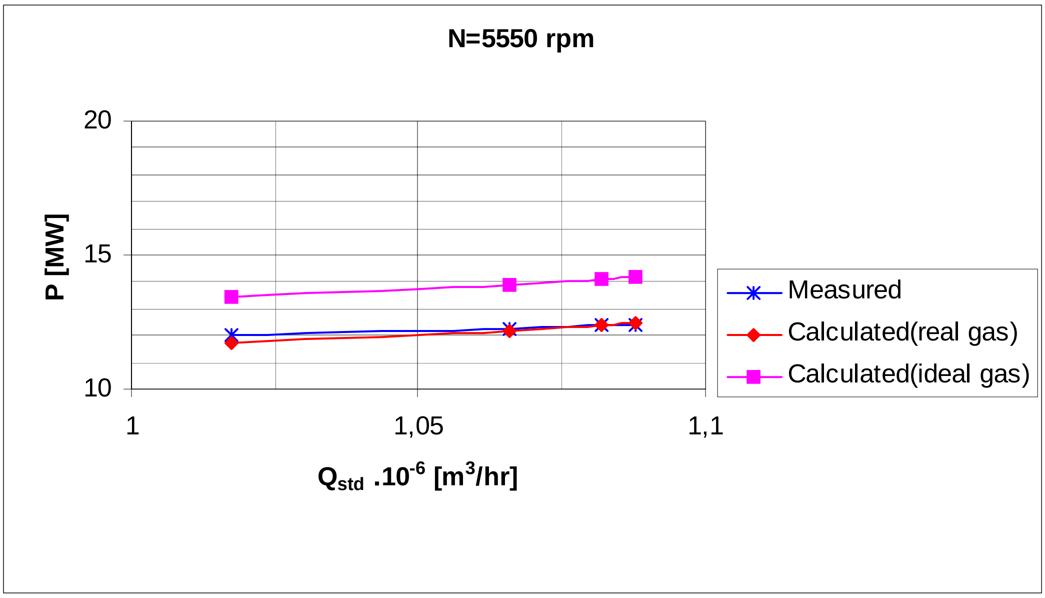

Fig. 5 Absorbed compressor power (N=5550 rpm, t_1 ~13,8°C, p_1 ~4,97MPa, p_2/p_1 = 1,406-1,40926).

CONCLUSION

We have demonstrated that the analysis of a gas compressor working under the typical conditions that prevail in natural gas transportation systems must be based on a thermodynamic real gas behaviour. In this paper the method of residual equation was derived and applied to a computational code. The comparison of the calculated results with the experimental results of the absorbed compressor work shows very good agreement. This method can be also applied to the case of expansion turbines used in the gas industry for reducing the high pressure to an intermediate level.

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