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Evaluation of thermodynamic models used for wet gas compressor design

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Norwegian University of Science and Technology
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MASTER THESIS

for

Student Marco Plano
Spring 2014*Evaluation of thermodynamic models used for wet gas compressor design
Evaluering av termodynamiske modeller for våtgass kompressordesign***Background and objective**

Sub-sea wet gas compression is one of the major on-going technology developments in the oil and gas industry. In a wet gas compressor the feed fluid is dominated by natural gas, but can also contain liquid hydrocarbons and a free liquid aqueous phase (water/methanol/glycol).

As part of design and operation of wet gas compressor design it is essential that the fluid phase behaviour calculations are accurate. Compressor internals such as the impeller is an example where we have an insufficient thermodynamic theoretical basis for wet gas and multiphase design.

There is an on-going debate in industrial and scientific community on how to perform thermodynamic design of wet gas compressors. Various modeling approaches have been suggested but these have to be verified. Fundamental phase kinetics and phenomena such as depositions of gas hydrate and scale formation is additional complicating factors.

For the planned Åsgard sub-sea wet gas compression project, gas, oil and water/MEG will be fed to the wet gas compressor. The compressor will normally operate on gas from the sub-sea separator (with entrained oil and water/glycol). Oil and water/glycol will additionally be injected in the feed stream during washing sequences.

NTNU in cooperation with Statoil has during the last decade been involved in the development of advanced new thermodynamic models. One of these models is the CPA-EoS, developed in cooperation with Denmark technical university. However these modern thermodynamic models have not yet been widely tested or applied for wet gas compressor calculations.

In this work comparison of thermodynamic calculations using different equations of state for wet gas compositions in the pressure range 10 to 100 bar and temperature range -10° to 200° C will be done. Various thermodynamic models will be used to calculate compressor outlet conditions based on given inlet temperature and pressure. Finally the effect of water, MEG and heavy hydrocarbons on enthalpy difference and compressor efficiency will be studied.

The following tasks are to be considered:

1. Calculate compressor outlet conditions (power/enthalpy difference, outlet temperatures and efficiency) using various equations of state for selected wet gas compositions in the pressure range 10 to 100 bar and temperature range -10° to 200° C
2. Implementation of a wet gas/multiphase compressor model based on the CPA-EoS

3. Evaluate the effects of liquids (water/MEG/hydrocarbons) on calculated enthalpy difference and compressor efficiency
4. Comparison of developed compressor model to calculations using commercial software such as HYSYS and PROII

-- " --

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- Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
- Field work

Department of Energy and Process Engineering, 14. January 2014



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PREFACE

This report contains the work performed on my master thesis at the Norwegian University of Science and Technology (NTNU) in Trondheim and the Royal Institute of Technology (KTH) in Stockholm.

The work was conducted during the spring of 2014.

I would like to thank my supervisor Even Solbraa for the precious help given during this year. He was a constant source of inspiration, support and motivation both for my preliminary project and thesis compilation. He was undoubtedly a perfect guide through the world of thermodynamic as well as an enjoyable person to share talks, thoughts and coffee breaks.

Thanks to my co supervisor Lars Eirik Bakken for the precious help given during the preliminary work of this thesis and for the thesis set up.

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Thanks to my family, my friends from Italy, from Norway as well as the ones from Stockholm for everything.

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Trondheim, May 30th 2014



Marco Plano

ABSTRACT

Big investments are done in developing wet gas compression technology. The reasons behind these investments are the need to improve the recovery from old reservoirs and the need to make small reservoirs economically profitable.

This work will use a model with the software NeqSim in order to simulate a wet gas compressor and evaluate the sensitivity of performance evaluation to the Equation of State used.

It will be easier to evaluate how much the data provided by vendors will effectively represent the real compressor's performance in the needed operating conditions. Then it is needed to know how much the work required from the compressor is influenced by the Equation of State used for the evaluation.

When it comes to the compressor work estimation, a maximum of 3% of deviation has been identified. According to the API 617 [32], this deviation is considered acceptable.

The deviation in estimating the compressor outlet temperature is maximum 2 K and it depends on the composition of the fluid and the pressure of operation.

For the polytropic efficiency instead, the highest deviation is 2%. Again, according to the standard API 617, a deviation up to 2% is considered acceptable.

Furthermore, this work will highlight the parameters that require higher accuracy when dealing with wet gas compressor performance evaluation. Thus, any equation of state able to give reasonable accuracy in these parameters will be expected to give accurate results in compressor's performance evaluation.

Finally, in conjunction with the preliminary work of this thesis [3], it has been possible to identify the Cubic Plus Association (CPA) equation of state as more suitable for wet gas calculations than Peng-Robinson (PR) and Soave-Redlich-Kwong (SRK) Equations of State.

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NOMENCLATURE

SYMBOLS

a	Speed of sound	(m/s)
b	Covolume	(m ³)
C _p	Isobaric specific heat	(kJ/kg-K)
C _v	Isochoric specific heat	(kJ/kg-K)
D	Diameter	(m)
f	Fugacity	(bar)
g	Gravity acceleration	(m ² /s ²)
H	Work per unit mass of fluid	(kJ/kg)
h	Enthalpy	(kJ/kg)
k	Specific heat ratio	(-)
L	Length	(m)
m	Mass	(kg)
MW	Molar weight	(kg/mol)
n	Polytropic exponent	(-)
p	Pressure	(bar)
Q	Volume	(m ³)
\dot{Q}	Heat flow	(kW)
R	Gas constant	(kJ/mol-K)
s	Entropy	(kJ/kg-K)
T	Temperature	(K)
U	Impeller peripheral velocity	(m/s)
u	Speed of fluid	(m/s)
V	Absolute tangential velocity	(m/s)
v	Specific volume	(m ³ /kg)
W	Work	(kJ)
X	Compressibility constant	(-)
x	Concentration in liquid phase	(-)
Y	Compressibility constant	(-)
y	Concentration in gas phase	(-)
Z	Compressibility	(-)

α	Gas volume fraction	(-)
α	Fugacity coefficient	(-)
β	Angle	(°)
β	Beta ratio	(-)
β	Gas mass fraction	(-)
δ	Slip ratio	(-)
η	Efficiency	(-)
ρ	Density	(kg/ m ³)

ABBREVIATIONS

API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
Comp	Composition
CPA	Cubic Plus Association
DIA	Direct Integration Approach
EoS	Equation of state
GMF	Gas Mass Fraction
GVF	Gas Volume Fraction
IGV	Inlet Guide Vanes
ISO	International Organization for Standardization
MEG	Ethylene Glycol
NCS	Norwegian Continental Shelf
NeqSim	Non Equilibrium Simulator
PR	Peng-Robinson
SGC	Subsea Gas Compression
SRK	Soave-Redlich-Kwong
VLE	Vapour-Liquid Equilibrium
VLLE	Vapour-Liquid-Liquid Equilibrium

SUBSCRIPTS

0	Reference
1	Inlet, suction state
2	Outlet, discharge state
g	Gas
l	Liquid
p	Polytropic
is	Isentropic
θ	Tangential
s	Shaft
TP	Two phase
h	Homogeneous
evap	Evaporated
gas	Gas phase
liq	Liquid phase
aqu	Aqueous phase
i	Element “i”
c	Critical property
m	Molar volume
oil	Oil phase
vap	Vaporization

1. INTRODUCTION

1.1 BACKGROUND

The increase of world energy demand brings new challenges to energy providers in order to match the production with the demand. Most of the biggest oil and gas fields in the Norwegian Continental Shelf are already providing their full capacity. Increasing the life of currently active oil and gas fields is the first step in order to increase overall oil and gas recovery.

This can be achieved by using extraction technologies able to produce oil and gas even at low reservoir pressures. Another way to maximize oil and gas recovery from the Norwegian Continental Shelf is to create extraction facilities with low operation and maintenance costs in order to make economically viable even small wells, which alone will not justify an oil platform.

The technology that has both characteristics is the Subsea Factory. It is possible to go "longer, deeper and colder" thanks to this new concept of extraction,. There is no longer the need for expensive oil platforms. Instead, everything can be managed from distance and small reservoirs can become economically profitable. [1]

The compressor is an important part of the Subsea Factory since the pressure of the gas and oil stream has to be pressurized to the correct pressure for the processes downstream. At the moment two technological solutions are in evaluation: subsea boosting and wet gas compression. [8]

The subsea boosting technology consists in efficiently separating the liquid and gas phase. After the separation a conventional dry gas compressor will increase the pressure of the gas phase while a pump will increase the pressure of the liquid phase. [8]

The wet gas compression consists of the compression of a multiphase flow of gas and liquid (up to 50% in mass of liquid) by using a wet gas compressor. The latter technology is the one studied in this work. [8]

The subsea wet gas compression technology has brought some engineering challenges that are currently involving researchers in different parts of the world and big

investments. One of these challenges is to properly estimate the properties of the wet gas stream during the compression process.

This type of challenge is not new for engineers and the search of a proper Equation of State (EoS) that is able to properly describe a mixture of gas and liquids is still open. During this century different models have been proposed.

Currently available models like Peng Robinson (PR) or Soave Redlich Kwong (SRK) are useful when dealing with gas mixtures. They have good accuracy in the calculation of thermodynamic properties of these mixtures. [3]

However, when dealing with gas mixtures that contain water or other liquids, the polarity of the molecules, as well as the hydrogen bonds, will lower the accuracy of the equations of state.

Various problems are then still unsolved when it comes to the developing of wet gas compressor models. [4]

1.2 SCOPE OF THESIS

The main problem that engineers face when working on Subsea Gas Compression (SGC), is that the ASME PTC10 [2] offers no guidance on wet gas compression testing and the performance of wet gas compressors is difficult to determine. The main reason is that this kind of evaluation is heavily influenced by the multiphase fluid properties, which are difficult to determine.

Unlike conventional compressors, subsea gas compressor will have to compress liquid together with gas. Usually the Gas Volume Fraction (GVF) is between 95% and 99%.

Most of the Equations of State (EoS) used for dry gas performance evaluation loose accuracy when handling liquids. Results become then more sensible to the EoS used to run the evaluation. The focus of this work will be studied the magnitude of this sensitivity and what parameters can affect a performance evaluation.

A good and reliable model is the basis for any engineering work, thus big efforts have been made in order to have a good thermodynamic model for SGC.

A thermodynamic model is crucial to evaluate the performance of a compressor. The main problems that process engineers had to face when modeling SGC is that the fluid composition during the compression process is not constant. Both pressure and temperature rise during compression and if the fluid contains liquid, phase change may occur both as evaporation and as condensation. It is then necessary to model properly these phenomena through a reliable EoS. Unfortunately, it is not yet clear which EoS is more suitable when dealing with a wet gas compression process.

It is important to know how much the results will vary when different EoS are selected or fluid with different compositions are processed. In other words, it is important to do a sensitivity analysis of the SGC evaluation procedure.

Furthermore, the composition of the fluid, pressure and temperature of operation will vary according to the reservoir in which the compressor will operate and the pressure of the process downstream the compressor. It is important then to know how much the sensibility mentioned before changes when the operating conditions are changing.

To summarize, the aim of this work is a sensitivity analysis of a compressor performance evaluation to the Equation of State used.

This sensitivity will be tested at different operating conditions, obtained by changing the following parameters:

1. Fluid composition
2. Pressure of operation

To conclude, this type of analysis can be useful to process engineers for two reasons:

1. Evaluation of performance data from different compressor vendors.

For example, by knowing how much the work required from the compressor is influenced by the EoS used for the evaluation or by the fluid composition, it will be easier to evaluate how much the data provided by vendors will effectively

represent the real performance of the compressor in the required operating conditions.

2. Evaluation of an Equation of State.

This work will try to highlight the parameters that require high accuracy when dealing with wet gas compressor performance evaluation. Thus, any equation of state able to give reasonable accuracy in these parameters will be expected to give accurate results in compressor's performance evaluation.

1.3 STRUCTURE OF THE WORK

In the first part of this thesis the theory behind compressors and the issues related to compression of wet gas is described briefly. Moreover, a brief introduction to the Subsea Compression will be given. This will be followed by an explanation of the evaluation structure and how the model was made.

Finally the results will be commented and analyzed.

In detail, the structure of the work is:

Chapter 1: Introduction and aim of the work

Chapter 2: Theory of compressors

Chapter 3: Introduction of the concept of wet gas

Chapter 4: Theory behind the wet gas compression: technology issues, performance evaluation and current technology characteristics of wet gas compressors

Chapter 5: Description of the wet gas compression modeling process

Chapter 6: Structure of the evaluation

Chapter 7: Validation of the model

Chapter 8: Results discussion

Chapter 9: Comparison with the preliminary project of this work

Chapter 10: Final comments and conclusions

1.4 LIMITATIONS

The main problem related to an analysis of this kind is the huge amount of cases that have to be compared. The time required for obtaining and analyze the results was the main limiting factor for this work.

As will be explained in details in Chapter 6, the model used in this work will calculate the compressor parameters using a total of 3 Equation of State (EoS), 5 fluid compositions, 2 inlet pressures and 3 numbers of steps for implementing the calculation method. This means a total of 13 variables and hundreds of testing points.

Since this kind of detailed analysis is both time consuming and unnecessary, some decision has been made in order to be able to finish the work and have reliable conclusions:

1. According to the preliminary work of this thesis [3] only 3 EoS have been chosen.
2. The sensitivity to operating temperatures has not been studied
3. For the inlet pressure sensitivity, only a "low pressure case" has been done
4. Only the Direct Integration Approach was used as a calculation method. Thus the sensitivity of calculation model used in the evaluation of performance has not been done. Previous works already compared two main calculation procedures used for compressors: Direct Integration Approach and Schultz method. [4]

Regarding the validation of the model used for running the calculation, it is currently difficult to obtain experimental data for wet gas compressors. Thus only a validation using dry gas has been done. This means that is not possible to ensure the accuracy of the model with wet gas compression.

2. COMPRESSOR

Compressors are integral part of petrochemical industry. Centrifugal compressors are the most used ones in this sector since they have smooth operation, large tolerance of process fluctuations and high reliability. [5]

The normal size for such compressors is usually from pressure ratios of 3:1 per stage to 12:1 for experimental models. However most of the centrifugal compressors used in the oil industry do not go over a pressure ratio of 3.5:1.

In the following figure, a schematic of a centrifugal compressor is shown:

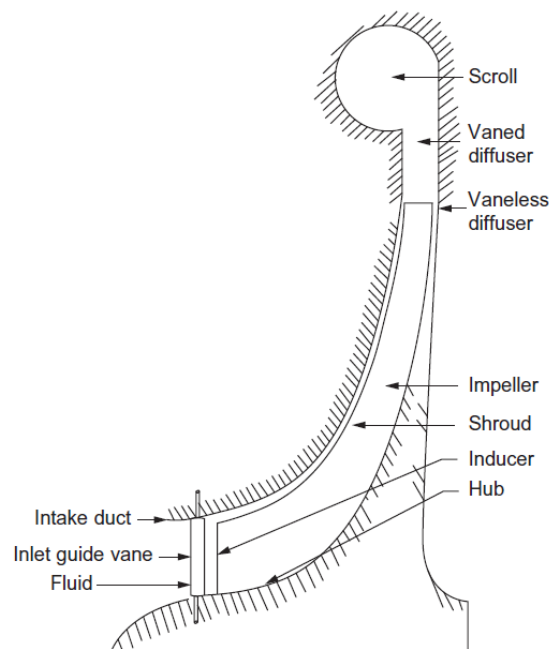


Figure 2.1 Schematic of a centrifugal compressor [5]

The fluid is flowing through the compressor because of the acceleration given by the rotating impeller blades. The velocity is then converted into pressure partially in the impeller and partially in the stationary diffuser. In the diffuser, most of the fluid's velocity is converted into pressure energy. It is common practice to design compressor in such a way that half of the pressure rise takes place in the impeller and half in the diffuser. The diffuser consists of stationary vanes tangential to the impeller. The inlet edge of the diffuser vanes is in line with the resultant airflow from the impeller.

A simple scheme of the velocity triangle for a centrifugal compressor is shown in Figure 2.2

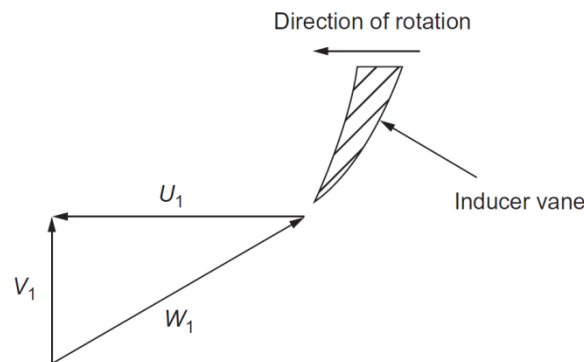


Figure 2.2 Velocity triangle [5]

In this figure the case of a compressor without inlet guide vanes (IGV) is represented; in this case, the fluid has no pre-whirl. The vector " U_1 " stands for the inducer velocity at the mean radial station while the vectors " V_1 " and " W_1 " are the absolute fluid velocity and the relative fluid velocity, respectively. The subscript "1" represents the velocities at the inlet of the inducer. The subscript "2" is commonly used when referring to the impeller outlet velocities.

These velocities (known as "Velocity triangle") are useful in designing compressors since it is possible to link them to other properties, such as the work required per mass unit from the compressor, by using specific equations.

In the case of the work required by the compressor per unit of fluid mass, Euler's equation can be used to link it to the velocity triangle:

$$H = -\frac{1}{g}(U_2 V_{\theta 2}) \quad (2.1)$$

Where:

H = work per unit mass of fluid

g = acceleration of gravity

U_2 = impeller peripheral velocity

$V_{\theta 2}$ = absolute tangential velocity at impeller outlet

Centrifugal compressors are usually used with high pressure ratio and low flow, while axial compressors are used with high flows and low pressure ratio. Centrifugal compressors have a lower rotational speed than axial ones.

2.1 IMPELLER

The aim of the impeller is to give energy to the fluid. It consists of an inducer and the impeller blades that give energy to the fluid.

The aim of the inducer is to increase the fluid's angular momentum without increasing the radius rotation. It has the largest relative velocity in the impeller and if the design is not good it may lead to choking phenomena. In centrifugal compressors, the flow enters in axial direction and leaves through the radial one. The fluid dynamics involved in the impeller is complex and involves three dimensions phenomena. The current state of art allows achieving 90% of efficiency in the impeller section. The main problems related to the impeller fluid dynamic are the stagnation of fluid (when part of the fluid has too low kinetic energy) and separation of the flow.

2.2 DIFFUSER

The aim of the diffuser is to recover as much kinetic energy as possible in order to obtain more pressure energy. It is important then to introduce the concept of stagnation pressure and temperature as follows:

$$p_0 = \frac{1}{2}\rho v^2 + p \quad (2.2)$$

$$T_0 = \frac{V^2}{2C_p} + T \quad (2.3)$$

Where:

p_0 = stagnation pressure

T_0 = stagnation temperature

V = velocity of the fluid

C_p = isobaric heat capacity

As can be seen, the pressure rise in the diffuser is due only to the recovery of kinetic energy.

In a compressor, the diffuser is the part with the lowest efficiency. The matching of the diffuser to the impeller is usually a difficult procedure since the flow regime goes from rotating to stationary.

The main geometric parameters of a diffuser are represented in the following figure:

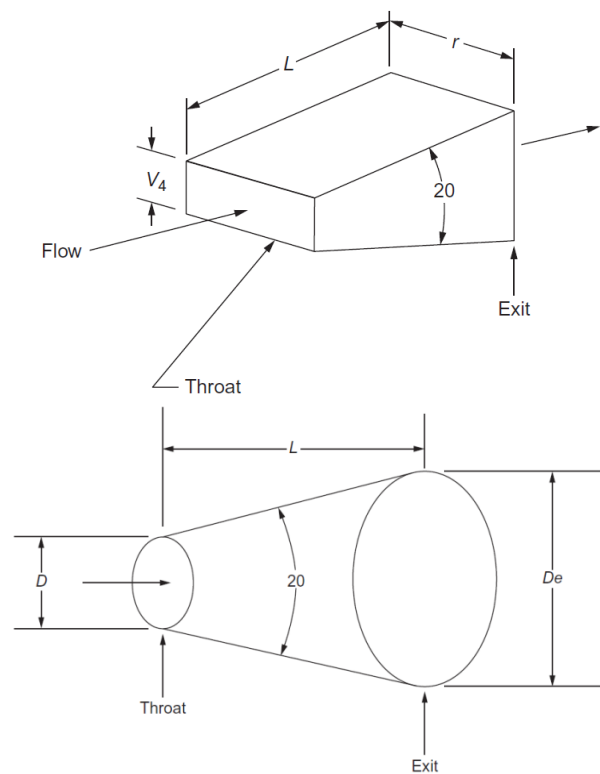


Figure 2.3 Diffuser geometries [5]

The selection of an optimal diffuser is difficult since it is possible to choose among an infinite number of cross-sectional shapes and wall configurations.

Usually the flow in the diffuser is assumed steady in order to obtain the overall geometric configuration. In diffusers the main problems are the separation of the fluid and the

stagnation due to viscous shearing forces. This may lead to eddy losses, mixing losses and changed-flow angles.

2.3 VOLUTE

The aim of the volute is to collect the fluid coming out from the diffuser and send it to the outlet pipe. It plays an important role in the compressor efficiency. A schematic of a volute is represented in Figure 2.3.

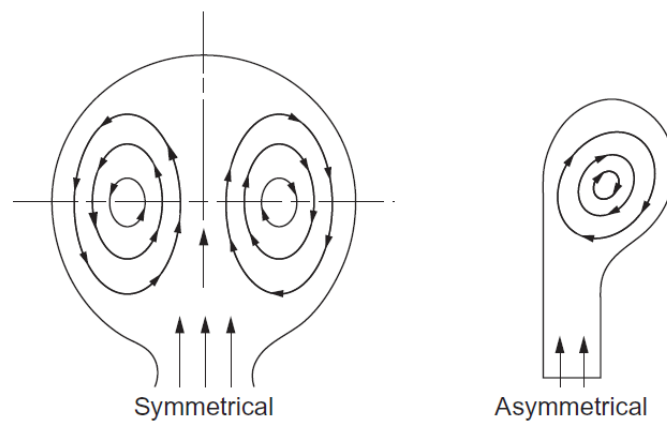


Figure 2.4 Symmetrical and Asymmetrical compressor volutes [5]

The flow pattern in an asymmetrical volute has one vortex instead of two for the symmetrical one. Usually, if the impeller discharges directly in the volute, the intake of the volute is slightly larger in order to allow the flow from the impeller to be bounded with the vortex at the connection between the volute and the impeller casing.

2.4 COMPRESSOR LOSSES

In order to understand better the impact of wet gas on traditional compressors, is important to understand what the main losses in a common compressor are. The introduction of liquid in the compressor will have an impact on those losses, changing the overall compressor performance.

2.4.1 ROTOR

2.4.1.1 SHOCK IN ROTOR LOSSES

These losses are due to the shock occurring at the rotor inlet. The rotor inlet should have wedge-like blades to sustain weak shock and then gradually increase the thickness in order to avoid another shock. If the blades are blunt, the flow may separate from the blade and increase the losses.

2.4.1.2 INCIDENCE LOSS

These losses will be particularly affected by the introduction of liquid. At off-design condition, the flow incidence may be positive or negative. In the case of positive incidence, the flow is reduced. A negative incidence may instead lead to fluid separation. In this case, as explained before, losses will occur.

In the following figure, the positive and negative incidence cases are represented.

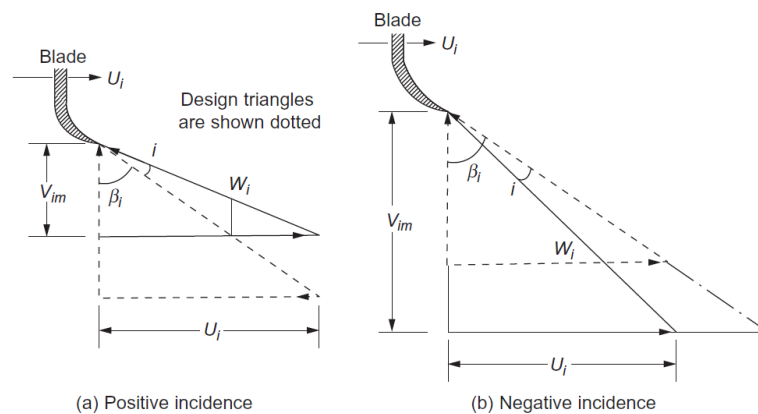


Figure 2.5 Positive and negative airflow incidences on blades [5]

2.4.1.3 DIFFUSION-BLADING LOSSES

These losses are due to the negative velocity gradient in the boundary layer. They are caused by the viscous shearing forces of the fluid. This may lead to separation of the fluid from rotor blades thus increasing losses.

2.4.1.4 CLEARANCE LOSSES

The fluid in a compressor experiences the Coriolis force. This force causes a pressure difference between the driving and trailing faces of an impeller blade. The fluid will try then to flow in the clearance between the impeller and the casing in order to neutralize this pressure difference.

2.4.2 STATOR

2.4.2.1 RECIRCULATION LOSSES

These losses are due to the backflow into the impeller exit of the fluid flowing into the diffuser. They are direct function of the exit angle.

2.4.2.2 EXIT LOSSES

These losses are due to the kinetic energy that is lost with the fluid leaving the vane diffuser. This energy is estimated to be around one half of the total kinetic energy.

2.5 SURGE AND CHOKE

While operating a compressor, there are two main phenomena that are important: surge and choke.

A compressor is in surge when the flow reverse is direction for short time intervals. This irregular operation may bring irreparable damage to the machine. Surge is a phenomenon that is not yet fully understood. The reverse flow may be due to some kind of aerodynamic instability. However is quite evident that the main cause of surge is the aerodynamic stall. When surge take place in the impeller, it may be due to an increase in impeller speed or reduced mass flow. However surge may be initiated at the vane less diffuser. Tests have shown that in the operating conditions typical of oil industry, surge may be initiated in the diffuser.

Different systems exist in order to avoid compressor surge. The most common system are the one that includes a static surge-detector. This type of detector constantly checks that pre-decided values of compressor's conditions are not exceeded. If these limits are

exceeded, is common to implement a fluid recirculation in order to remain in safe operating conditions.

On the other hand, choking occurs when the compressor cannot allow higher mass flow. This is due to the fact that the fluid flowing into the compressor reaches the sonic speed (Mach number equal to one) in any part of the compressor. Prolonged operation in choking condition may lead to damages to impeller blades. [5]

3. WET GAS

There is still some discussion about what the definition of "wet gas" is [6]. A good definition could define wet gas as a "gas with small amounts of liquids" [7]. In the following figure it is possible to see a typical wet gas stream in a pipe. Since Hydrate formation is a big problem in the oil industry, usually Ethylene Glycol (MEG) is added to wet gas in order to avoid Hydrates. Thus the liquid phase is a mixture of Hydrocarbons, Water and MEG.

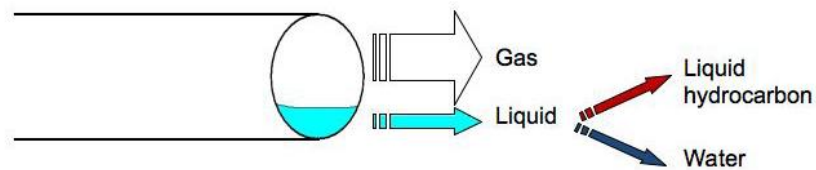


Figure 3.1 Schematic of wet gas flow [7]

Gas Volume Fraction (GVF) and Gas Mass Fraction (GMF) are two important parameters and they are defined as follows: [4]

$$GVF = \alpha = \frac{Q_g}{Q_g + Q_l} \quad (3.1)$$

$$GMF = \beta = \frac{m_g}{m_g + m_l} \quad (3.2)$$

Where:

Q_g = volume of the gas phase at actual conditions

Q_l = volume of the liquid phase at actual conditions

m_g = mass of the gas phase

m_l = mass of the liquid phase

Various tests [4] have shown that GMF is the main parameter that affects the performance of wet gas compressors. The liquid content of wet gas is usually around 1% to 5% in volume. This means that the GVF is usually from 99% to 95%. The liquid content on a mass basis is also important. In wet gas the GMF can go from 99% to even less than 50%.

One of the possible approaches when dealing with wet gas is the "homogeneous approach". In this approach wet gas is considered as a homogeneous fluid. In the following equation the wet gas density is calculated as homogeneous density: [4]

$$\rho_h = \rho_g \alpha + \rho_l (1 - \alpha) \quad (3.3)$$

Where:

ρ_h = homogenous density

ρ_g = density of the gas phase

ρ_l = density of the liquid phase

Another parameter that plays an important role in the multiphase flow calculations is the "slip ratio", which is the ratio between gas and liquid phase densities as shown in the following equation: [4]

$$\delta = \frac{\rho_g}{\rho_l} \quad (3.4)$$

The slip ratio reflects the grade of homogeneity and the slip between the phases and is a useful tool for explaining certain compressor losses when liquid is injected. The topic will be discussed in details in Chapter 4.

The composition of phases depends on local temperature and pressure conditions. As these parameters changes, condensation or evaporation may occur. This leads to a series of problems in processes that involve wet gas. Indeed the composition of the phases (liquid and gas) changes during the process. Wet gas compression or transportation in pipelines from reservoir to process plants can be two examples where the multiphase nature of wet gas may create problems. A reliable and accurate Equations of State (EoS) is required in order to design properly any process equipment that has to handle wet gas.

There are three main flow patterns for wet gas: [7]

STRATIFIED FLOW:

As can be seen in the following figure, stratified flow implies that the gas phase flows in the upper part of the pipe, while the liquid phase is only in the lower part. This is the case of transportation pipes, where there is a stationary flow without relevant changes in orientation or velocity.

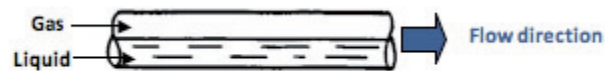


Figure 3.2 Stratified flow [7]

STRATIFIED-WAVY

As can be seen in the figure below, Stratified-Wavy flow is similar to the stratified one but with a wavy gas-liquid interface. This behaviour is caused by big velocity difference between phases. This case is less convenient one since the wavy surface increases the average perceived roughness of the pipe. This means additional friction losses. This flow appears in compressor's impellers, where the entrained liquid forms a slow liquid film compared to the accelerated gas stream. The overall effect on the compressor's performance is modified impeller aerodynamic and increased frictional losses.



Figure 3.3 Stratified-Wavy flow [7]

ANNULAR

This flow pattern also exists with different intensities in the previous patterns. Its magnitude depends on the velocity of the stream: as the velocity increases, annular flow will be more visible. In vertical pipes this flow pattern will be the only possible pattern. As can be seen in the following figure, this pattern is characterized by a thin liquid film on the top and a thicker liquid film on the bottom due to gravity. In vertical flows, the films will theoretically have the same thickness in all the parts of the pipe.



Figure 3.4 Annular flow [7]

In this type of flow the entrainment of liquid droplets in the central gas stream is important. These droplets are the main cause of mass exchange between liquid and gas phase. Droplets will deposit in the liquid film and the liquid will be atomized in small droplets in the gas stream. The overall effect is an increase of entropy and more friction losses. Especially in vertical pipes, it will be possible to notice an annular flow with a considerable amount of droplets entrained in the gas stream. This type of flow is called "Annular-mist". [6] [4]

4. WET GAS COMPRESSION

In this chapter the technology issues related to the compression of wet gas will be explained. The presence of liquid in the compressor causes different problems. When implementing a performance evaluation, one of the problems is that thermodynamic properties have different definitions when dealing with multiphase flows. Furthermore the presence of liquid and the changes in temperature and pressure make phase changes occur.

On the technological side, liquid film may change the aerodynamic efficiency of the impeller and the different density and phase velocities make it necessary to have a specific design in order to optimize the performance.

4.1 SUBSEA COMPRESSION

Oil Industry is currently focusing on Increase reservoir Oil Recovery (IOR). The alternative to achieving this goal are platform compression, which is very capital intensive, or subsea compression, which is less expensive but needs further technological improvements.

The use of platforms is justified only in the case of big reservoirs in acceptable climatic conditions. More reservoirs have been discovered in the Arctic Sea but few of them economically justify the construction of a production platform. The same applies for improving oil recovery from currently operative reservoirs. The increased production not always justifies the construction of a compression platform. [31]

Therefore Subsea Compression seems the most reasonable technology currently available for the purpose previously described. It is divided in wet gas compression or separate gas compression and liquid pumping, as can be seen in the following figures.

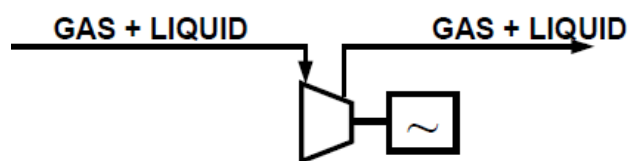


Figure 4.1 Wet gas compression solution [8]

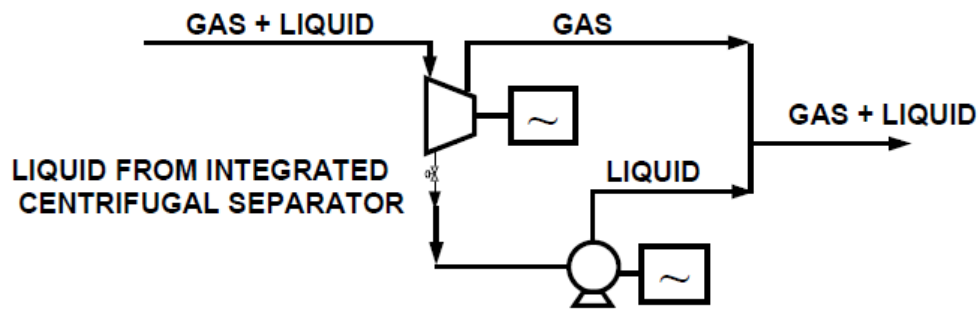


Figure 4.2 Separate liquid pumping and gas compression [8]

It is still uncertain what technology is more suitable for Subsea Compression. Åsgard and Ormen Lange, in the Norwegian Continental Shelf (NCS) are currently the main fields where these technologies are being tested by Statoil ASA.

Any subsea solution must require less maintenance since any operation under the sea level will result in prohibitive costs, eliminating all the advantages of having subsea installations. The need of less maintenance increases the stress on the materials, thus subsea compression needs new materials and new technologies.

Tests on this technology are currently taking place at K-Lab, property of Statoil ASA, in Norway. MAN Turbo is the first vendor contributing at the project and from 2008 also Siemens Demag Delaval is taking part on it.

The main advantages for Subsea Compression will be the increased production flexibility with variable fluid properties and an increase in production from gas/condensate fields, even from small ones. Furthermore Subsea Compression is an invisible production system with less leakage points compared to traditional platforms so it can be used also in harsh or sensitive areas.[8]

4.2 PERFORMANCE ANALYSIS

This chapter, will be explain in details how the performance analysis of a compressor is done. The case of dry gas compressors as well as the case of wet gas compressors will be considered. The main properties that play an important role in the evaluation of the compressor performance will be highlighted.

4.2.1 PROPERTIES OF INTEREST

Regarding the performance analysis, it is important to know what parameters are better to focus on and why. The following paragraphs, it will explain in detail what properties have been chosen and why they are considered important for this work. [4]

4.2.1.1 EFFICIENCY

Efficiency can be defined in different ways. In this work; the compressor efficiency can be defined as:

$$\eta = \frac{W_p}{P_s} \quad (4.1)$$

Where " η " is the efficiency, " W " is the polytropic head and " P_s " is the power to the shaft.

In the case of the evaluation of a thermodynamic model, the efficiency that is important is the Isentropic Efficiency. This efficiency is defined as follows:

$$\eta_{is} = \frac{H_{is}}{H} \quad (4.2)$$

Where " H_{is} " is the isentropic enthalpy difference and " H " is the real enthalpy difference. However this efficiency is not used in compressor performance analysis since it varies at different operating conditions. This is due to the deviation in the isobars: $\left(\frac{dh}{ds}\right)_p = T$.

Thus, it is preferred to use a polytropic analysis, where a polytropic process is considered:

$$pv^n = \text{constant} \quad (4.3)$$

Where:

$$n = \frac{\ln\left(\frac{p_1}{p_2}\right)}{\ln\left(\frac{v_1}{v_2}\right)} \quad (4.4)$$

The polytropic exponent "n" is used for dry gas and it varies along the compressor path, making the exact solution of the polytropic head impossible. It is then considered constant when solving the polytropic head equation.

The polytropic efficiency is defined as follows:

$$\eta_p = \frac{\partial h_{is}}{\partial h} \quad (4.5)$$

This efficiency is constant at different operating conditions because it refers to infinitesimal isentropic processes along a polytropic path. For this reason it is preferred for compressor performance analysis.

The main challenge related to performance evaluation for wet gas is that the fluid composition is not constant during the process, since phase change occurs. It is then not possible to use the calculation method described in the standard ASME PTC-10, known as the Shultz method. This method will be described in detail in section 4.2.2.

In this work, the developed thermodynamic model will calculate the polytropic efficiency based on given Inlet and Outlet pressure and temperature conditions using a particular method of calculation called Direct Integration Approach (DIA). This method has been shown to be more reliable when dealing with wet gas as will be explained in the following paragraph. [4]

4.2.1.2 WORK

The polytropic head of a compressor is a key parameter in the performance evaluation process. For dry gas, the polytropic head is defined as: [4]

$$W_p = \frac{n}{n-1} (p_2 v_2 - p_1 v_1) \quad (4.6)$$

As explained before, the Schultz method is able to give a good approximation of the change in polytropic exponent "n" and it is correct when dealing with dry gas. However it loses accuracy when it comes to wet gas.

Moreover, the definition itself of polytropic head changes for multiphase flow. Indeed, it is possible to define it in two ways: [9]

- Single fluid model, where the multiphase fluid is considered as one fluid. In this case, the polytropic head is defined as follows:

$$W_{ph} = \frac{n_h}{n_h-1} (p_2 v_{h2} - p_1 v_{h1}) \quad (4.7)$$

Where W_{ph} is the polytropic head for wet gas, n_h is the polytropic exponent for wet gas and v_h is the specific volume for wet gas. The subscript "h" stands for the property of the homogeneous mixture.

The polytropic exponent for wet gas is defined as follows:

$$n_h = \frac{\ln\left(\frac{p_1}{p_2}\right)}{\ln\left(\frac{v_{h1}}{v_{h2}}\right)} \quad (4.8)$$

The single fluid model is based on the definition of the two-phase specific volume:

$$v_h = \frac{1}{GVF * \rho_g + (1 - GVF) * \rho_l} \quad (4.9)$$

Where GVF is the gas volume fraction (defined in Chapter 3) and ρ_g and ρ_l are respectively the density of the gas and liquid phase.

- Another approach would be to consider each phase separately. This model is called two fluid model and it defines the polytropic head as follows:

$$W_p = GMF \frac{n}{n-1} \frac{R_0}{MW_{g1}} Z_1 T_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right) + (1 - GMF) v_{l1} (p_2 - p_1) \quad (4.10)$$

Where the fluid quality "GMF" is the gas mass fraction (defined in Chapter 3).

The model selected in this work is the single fluid model as already explained in Chapter 3. This means that the property "y" will be calculated as follows:

$$y_h = y_{gas} \beta + y_{liq} (1 - \beta) \quad (4.11)$$

Where " β " is the fluid quality (or GMF) defined as $\beta = \frac{m_{gas}}{m_{gas} + m_{liq}}$. The terms y_{gas} and y_{liq} are respectively the properties calculated for the gas and liquid phase.

4.2.1.3 OUTLET TEMPERATURE

The outlet temperature is used in this work to determine the polytropic efficiency. As will be shown in Chapter 6, the polytropic efficiency is calculated from an iteration process with the outlet temperature given.

Thus, a possible difference in estimated efficiency for different EoS could be explained with their difference in estimating the outlet temperature.

Furthermore, the outlet temperature is an important data for the process engineers. The design of a process where a wet gas compressor is used, involves the use of coolers in order to control the compressor inlet flow temperature and for the anti-surge control system. [5]

4.2.1.4 DENSITY

Density is a crucial property when it comes to compressors. In this work density has been used to determine the volume flow and the compressibility factor "Z" at the inlet of the compressor.

As explained in the previous section, homogeneous fluid assumption has been done.

In the case of density, it means that the homogenous density at the compressor inlet will be calculated as follows:

$$\rho_h = \rho_{gas}\beta + \rho_{liq}(1 - \beta) \quad (4.12)$$

Since the inlet flow input for NeqSim is in moles per second, it means that the inlet mass flow " \dot{m} " is given. In order to obtain the related volume flow " \dot{V} " it is necessary to use the homogeneous density as follows:

$$\dot{V} = \frac{\dot{m}}{\rho_h} \quad (4.13)$$

However, the main application for density in this work is to know the compressibility factor of the fluid at the inlet of the compressor in order to be able to correlate a certain deviation in performance estimation by different EoS to their ability to calculate the compressibility factor of the fluid.

Density and compressibility factor are strictly correlated by the following equation:

$$Z = \frac{p}{\rho_h R' T} \quad (4.14)$$

Where "Z" is the compressibility factor, "p" the pressure, " ρ_h " the homogeneous density, "R'" is the specific gas constant ($R' = \frac{R}{\text{mole}}$) and "T" is the temperature.

Compressibility is important because it expresses the deviation of a real fluid's behavior from the ideal fluid's one.

It is expected that a certain EoS' deviation in calculating the outlet properties described before (work, outlet temperature and efficiency) can be explained completely or partially with EoS' differences in the calculated density (thus compressibility).

In other words, this work aims at investigating whether compressibility is a parameter that affects the calculated properties.

The reason for considering compressibility as a key parameter, is an article written by Mark R. Sandberg [10] that shows the direct correlation of enthalpy and entropy to compressibility and specific heat solely.

By simple substitution, the author derives the following equation for single-phase gas compression enthalpy and entropy difference:

$$\Delta H = \int C_p dT - \int \frac{RT^2}{P} \left(\frac{\partial Z}{\partial T} \right)_T dP \quad (4.15)$$

$$\Delta S = \int \frac{C_p}{T} dT - \int \left[\frac{ZR}{P} + \frac{RT}{P} \left(\frac{\partial Z}{\partial T} \right)_P \right] dP \quad (4.16)$$

Although these equations have a slightly different role in a multi-phase compression, they clearly show that there is a strict correlation between compressibility and calculation of enthalpy and entropy difference.

A question that may rise is why it is important to calculate enthalpy and entropy in the model of this work.

The detailed answer will follow in Chapter 6, but all the outlet properties previously described (work, outlet temperature and efficiency) are calculated with iteration and flashes that involve the calculation of enthalpy and entropy solely. For example, the deviation between the compressor work calculated from two EoS could be due to how the two EoS calculate enthalpy and entropy of the fluid. According to what was explained before, enthalpy and entropy depend on compressibility and specific heat.

4.2.1.5 SPEED OF SOUND

The speed of sound is a property that is used for compressor design. As the fluid approaches the speed of sound (Mach number =1), compressibility effects and shock waves may occur. This parameter is then of vital importance for compressor design and operation.

If the assumption of homogeneous fluid is done, then the Mach number is defined as follows: [4]

$$M = \frac{u_h}{a_h} \quad (4.17)$$

Where "M" is the Mach number, " u_h " is the speed of the homogeneous fluid and " a_h " is the homogeneous speed of sound for the fluid. The equation to determine the homogeneous properties is Eq. 4.11.

The definition of Mach number for multiphase flow is not as straightforward as for single-phase fluid. When compressing two-phase flows, critical speed and speed of sound may not coincide. This means that choking may occur before the actual speed of sound is reached.

In this work only the speed of sound of the gas phase has been studied and it is defined as follows: [11]

$$a^2 = \left(\frac{C_p}{C_v}\right) \frac{p}{\rho} \quad (4.18)$$

As for density, this work aims to explain a possible deviation in the speed of sound calculated by different EoS in their difference in calculating the specific heat and density. This means that it will be investigated whether big differences in specific heat or density could result in large differences in calculated speed of sound. Thus, this will show the importance of the choice of a specific EoS when dealing with speed of sound calculations.

4.2.1.6 SPECIFIC HEAT

The specific heat is a property that is strictly correlated to enthalpy and entropy calculations as well as to the speed of sound for the reasons as explained before

In this work only the isobaric specific heat will be considered.

4.2.1.7 SOLUBILITY

Knowing how much of a certain element will solute into another one is of vital importance for all the evaluations done in this work.

First, if the solubilities are accurate then the composition of the process fluid at equilibrium conditions will be accurate.

Furthermore, as will be explained in details further on, the evaporation of liquid into the gas phase will have a considerable impact on the compressor work estimation. The following formula defines the enthalpy change due to evaporation of aqueous mass:

$$\Delta H_{vap} = m_{evap} H_{vap} \quad (4.19)$$

According to the simulations done in this work with the model developed, the mass of aqueous phase that will evaporate into the gas phase accounts for less than 0.5% of the total mass flowing into the compressor. However, the heat of vaporization of water and MEG is around 2300 kJ/kg and 1000 kJ/kg respectively, while the enthalpy change of the gas phase is around 250 kJ/kg. Thus it is clear that the enthalpy change due to evaporation of aqueous phase is not negligible. Moreover, since the evaporated mass is a small quantity, it needs higher precision.

Solubility is related to the fugacity coefficient, which expresses the non-ideal behavior of a component and is defined as follows: [12]

$$\alpha = f/P \quad (4.20)$$

Where:

α = is the fugacity coefficient.

f = is the fugacity.

P = is the pressure.

As will be explained in details in Chapter 8, the fugacity coefficient is the basis for implementing the P-T flash calculations. The model used in this work uses P-T flash calculations in order to estimate the polytropic efficiency, outlet temperature and work of the compressor. By knowing how the EoS performs when estimating the solubility for the different components it will be possible to know their accuracy in performing P-T flash calculations.

4.2.1.8 SUMMARY

In order to sum up all the information given, the following graphs represents an overview of the concept expressed in this chapter:

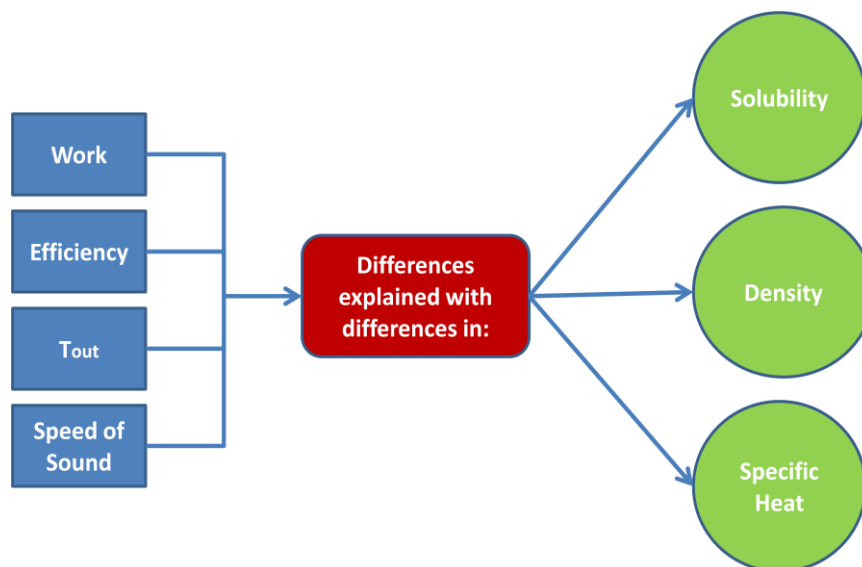


Figure 4.3 Summary of correlation between parameters

This work will attempt to explain the differences in calculated properties (in blue) by using different EoS with their difference in calculating density, specific heat and solubility.

4.2.2 DIRECT INTEGRATION APPROACH

The introduction of liquid in the compressor brings some issues when calculating the compressor's performance. [4]

In dry gas compressors, the isentropic efficiency is not used for performance calculations because it varies when the operating conditions changes. Instead the polytropic approach is used.

As explained before, a polytropic process is defined as: [4]

$$pv^n = \text{constant} \quad (4.21)$$

Where:

$$n = \frac{\ln\left(\frac{p_1}{p_2}\right)}{\ln\left(\frac{v_1}{v_2}\right)} \quad (4.22)$$

"n" is the polytropic exponent for dry gas and it varies along the compressor path, making the exact solution of the polytropic head impossible. It is then considered constant when solving the polytropic head equation: [4]

$$h_p = \int_1^2 v dp = \frac{n}{n-1} Z_1 RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \quad (4.23)$$

In this way the change in polytropic exponent along the compression path is not taken into account. Thus ASME PTC-10 describes the procedure to solve this problem: the Schultz method. [13]

In this method the polytropic exponent "n" is approximated using two compressibility function X and Y: [13]

$$n = \frac{1 + X}{Y \left[\frac{1}{k} \left(\frac{1}{\eta_p} + X \right) - \left(\frac{1}{\eta_p} - 1 \right) \right]} \quad (4.24)$$

The variation of the defined polytropic exponent is small along the compression path and thus it can be assumed to be constant.

Unfortunately, the polytropic exponent defined according to the Schultz method is not reliable anymore when the considered fluid is wet gas. The main reason is that phase changes are more likely to happen in a multiphase flow compression. Since both temperature and pressure changes, it may happen that the change in solubility of the vapor and liquid phase will make some gas to condensate or some liquid to evaporate.

In order to take into account these phenomena, a new approach has been implemented. This approach is called Direct Integration Approach (DIA).

The main concept of this approach is to use the real gas properties along the compression path and not the Schultz averages. This is possible with the development of more reliable EoS and computing power. Even if no standard exist for the performance evaluation of wet gas, the DIA is allowed by ISO standards, where a step-by-step isentropic compression can be implemented with a large number of small steps: [4]

$$\eta_p = \sum_{i=1}^{\infty} \left. \frac{\partial h_{is}}{\partial h} \right|_{\eta_p \text{ constant}} \quad (4.25)$$

In detail, the DIA consists in dividing the polytropic compression path in small steps with constant pressure ratio. Subsequently enthalpy and entropy are calculated for the first step inlet. By assuming a certain polytropic efficiency, the enthalpy and entropy at the outlet of each step are calculated. This assumed polytropic efficiency is kept constant throughout all the compression. The real polytropic efficiency will then be obtained by iterations, in order to get the given outlet temperature. This methodology is particularly suitable for multiphase flows where phase change may occur.

In the following figure is represented the DIA concept:

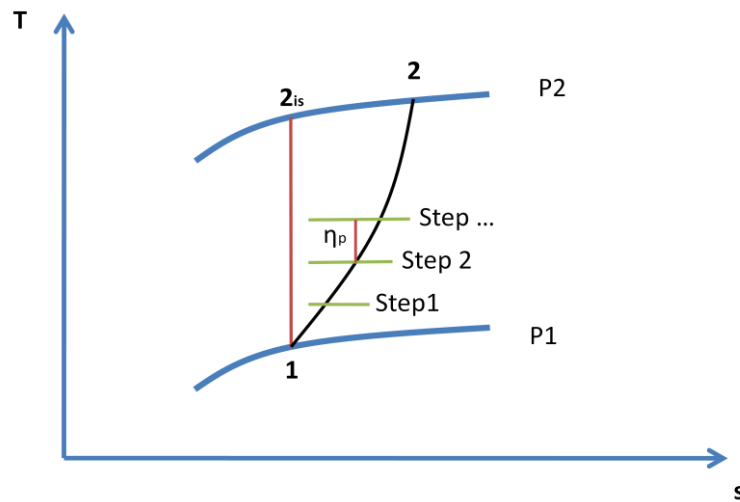


Figure 4.4 Schematic of Direct Integration Approach

This procedure allows updating all the fluid properties during the compression by using a proper EoS. This is possible by assuming thermal and phase equilibrium during the compression. This is not true in real compression, thus a proper model should take into account non-equilibrium phenomena. In this work equilibrium conditions are assumed.

In Chapter 6, it is explained in details how the DIA is implemented in the model used in this work.

An important consideration has to be done regarding the step division of the DIA in this work. According to Bakken et al. [4] the steps have to be such that the pressure ratio is kept constant. However, in order to keep the code as simple as possible and reduce the computing time the author chose to keep a constant pressure difference instead of a constant pressure ratio.

The following figure will show the comparison between the two options:

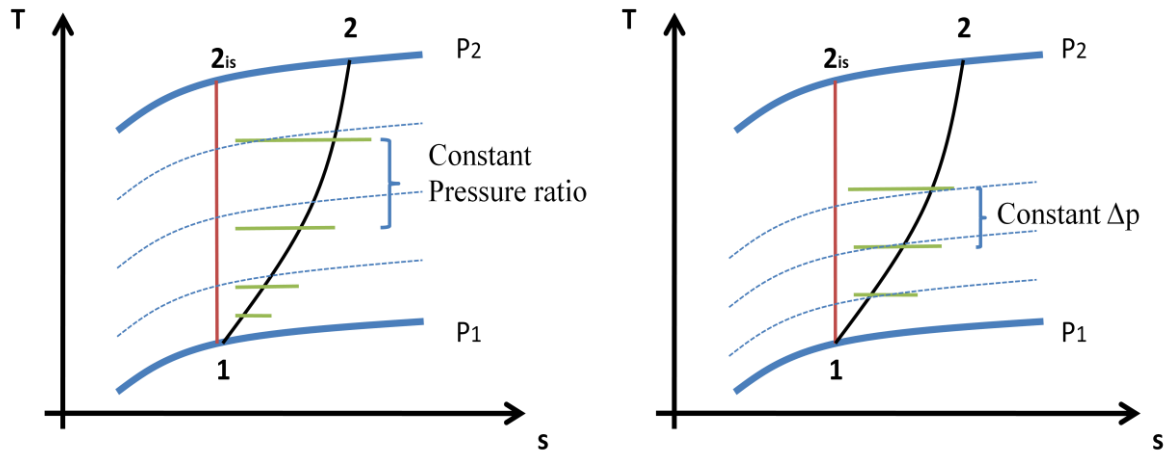


Figure 4.5 Differences in constant pressure ratio and constant pressure difference for each step in Direct Integration Approach

If the pressure ratio is kept constant the pressure difference will be different and vice versa. It has been shown that the results from these two methods are not different for dry and wet gas.

4.3 DESIGN OF WET GAS COMPRESSORS

In this section the consequences on compressor's performance when liquid is introduced will be briefly explained, as well as the technological solutions that have been adopted in order to make possible the subsea wet gas compression.

4.3.1 EFFECT OF LIQUID INJECTION IN COMPRESSOR

The injection of liquid into the compressor has some consequences. They have been properly summarized by Hundseid et al. [14] and reported here together with data from the model used in this work. In this way it is possible both to do a first validation of the model (the behaviour of the compressor follows the expected behaviour) as well as to have a graphic support to the theory.

4.3.1.1 EVAPORATIVE COOLING/CONDENSING HEATING

The liquid phase in the process fluid will evaporate during compression. As result, the outlet temperature of the compressor will be lower. This phenomenon depends mainly from the mass that evaporates and its heat of vaporization. The calculated outlet temperature for the different mixture that will be studied in this work is shown in the following figure. The mixtures will be explained in detail in Chapter 7.

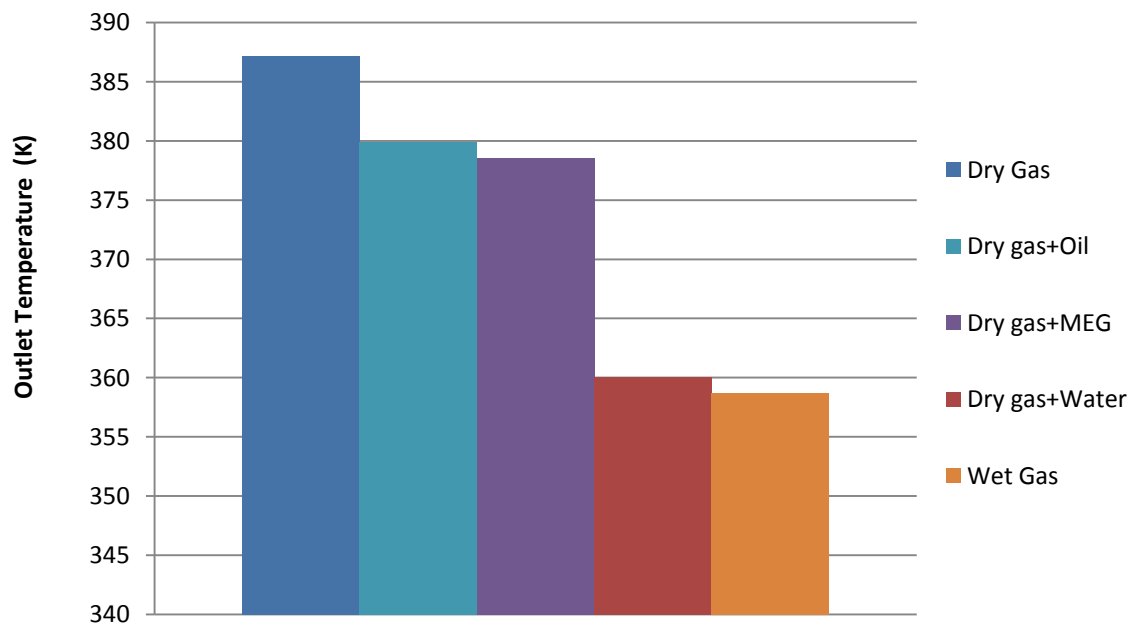


Figure 4.6 Influence of liquid injection on compressor outlet temperature

As the liquid content increases, the outlet temperature decreases. However, the liquid content of the mixture "dry gas + MEG" is higher than the one "dry gas + water", but the outlet temperature for its case is higher. This happens because the amount of liquid flowing into the compressor is important, as well as its heat of vaporization. The heat of vaporization of MEG is around 1000 kJ/kg while the one of water is 2300 kJ/kg at the conditions of operation. Therefore even if less water evaporates, it will have larger impact on the outlet temperature.

The different specific heat of the gas and liquid phase will create a temperature gradient in the fluid. The resulting heat and mass transfer will increase the entropy thus the losses. [14]

4.3.1.2 LIQUID ENTRAINMENT/DEPOSITION

Since the velocities of the liquid and gas phase are different, liquid atomization and droplets deposition will occur. The total kinetic energy is reduced when these phenomena take place. Furthermore, the inter phase heat exchange is strongly dependent on the droplets' diameters. The momentum transfer from gas to liquid phase results in a pressure drop at the compressor inlet. The magnitude is determined by the amount of liquid injected and the slip ratio.

The injection of liquid will also affect the incidence angle in the diffuser and the flow path. The sonic speed will be reduced and the choking criteria change. However it has to be specified that acoustic and critical speed for multiphase flow may not coincide with the single-phase flow ones. [14]

4.3.1.3 LIQUID FILM

The liquid phase flowing into the impeller will create a liquid film that will increase the surface roughness. This is due mainly to the wavy flow pattern of the liquid film and the impact of the droplets. This phenomenon leads to higher frictional losses and higher displacement thickness. The reduction of flow area will also increase the blockage. Shear losses will increase due to changes in liquid phase viscosity. [14]

4.3.1.4 SUMMARY

Injection of liquid into the compressor brings several problems and increases the total losses. These effects bring to higher power consumption and reduced efficiency [9]. However, since this work is an evaluation of thermodynamic models, the mechanical consequences of liquid injection are not taken into account. On the thermodynamic side, the introduction of liquid into the compressor will increase the complexity of the fluid. Thus, it will be more difficult for the available models to estimate properly the fluid parameters such as enthalpy, density, specific heat and solubility. On the other hand, the liquid phase will act as an "intercooler". This will have a positive effect on the compressor performance, if mechanical losses are neglected. The Figure 4.7 shows the results of specific energy consumption estimation from the model used in this work.

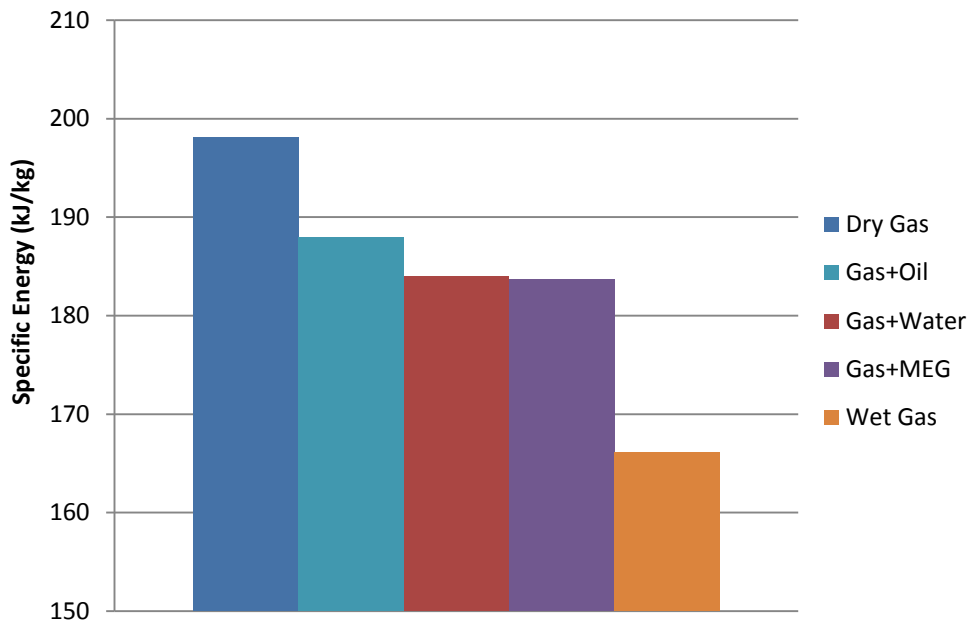


Figure 4.7 Influence of liquid injection on compressor specific work

As seen in the graph, with higher liquid content, the specific energy required from the fluid is lower. Brenne et al. [9] also found out that by increasing the liquid fraction, the specific energy decreased. However, they compared this case with a separate boosting, where the liquid and gas phases are compressed separately by a pump and a compressor, respectively. The result of their comparison was that the specific energy required by wet gas compression was higher than the separate boosting case. A possible explanation of this difference may be that during wet gas compression evaporation occurs.

They also underlined that even if subsea boosting has lower specific energy consumption, from a technological point of view it requires large separators. These components will add pressure drops and thus the overall energy required by a subsea boosting system may be higher than a subsea wet gas compression one.

4.3.2 TECHNOLOGY ISSUES

As already stated, tight technological requirements are applied for Subsea Compression. They are clearly resumed in the work of Brenne et al. [8]. An acceptable efficiency (>70%) for both dry gas and wet gas duty is required, as well as a correct estimation of compressor's performance, as described in the previous chapter. Large compressor units of more than 6 MW are used for subsea applications. As a last requirement, subsea technologies have to be designed for unmanned operation and reduced maintenance. This means that at least 5 years should pass between two maintenance operations.

According to Brenne et al. [4] the main elements to consider when designing a subsea compressor are:

1. *Materials*. Subsea applications, especially with wet gas, increase the mechanical stress on components. The entrained droplets increase drastically the erosion effect of the fluid. New materials have to be tested for this application.
2. *Bearings*. They have to be suitable for remote operation and tolerant to well mixtures. Currently, the most promising technology seems to be the magnetic bearings since no lubricant is required.
3. *Seals*. There is no need of seals since there is only one pressure casing for both motor and compressor. This design avoids contact with seawater and does not require the use of a gearbox. Rotor can be in one piece or in a rigid/flexible coupling.
4. *Motor*. The cooling system will include the compressed fluid. Windage losses become more dominant when the rotational speed is increased.

5. *Design.* There is the need of an assessment of booster operation and maintenance requirements to make control easier. For pumps, the preferred orientation is vertical, while for wet gas compressors there is no preference.

6. *Performance.* Liquid carry over from scrubber will cause fouling. Small particles can appear in the compressor, as the liquid evaporates during compression. Speed and momentum margin is required.

EoS play an important role, especially regarding performance, as previously described. Their accuracy will guarantee optimum design and reliable technical solutions in order to face all the challenges that the subsea environment poses.

Subsea wet gas compressors currently under testing are the Framo WGC 2000, with vertical alignment, one variable speed and pressure limitation of 40 bar, the MAN Turbo Mopico and Hofim, with 7 to 14 impellers, Siemens Demag Delaval ECOII concept, with maximum discharge pressure of 150 bar and 5 impellers and Nuovo Pignone subsea concept. These models are represented in Figure 4.8.

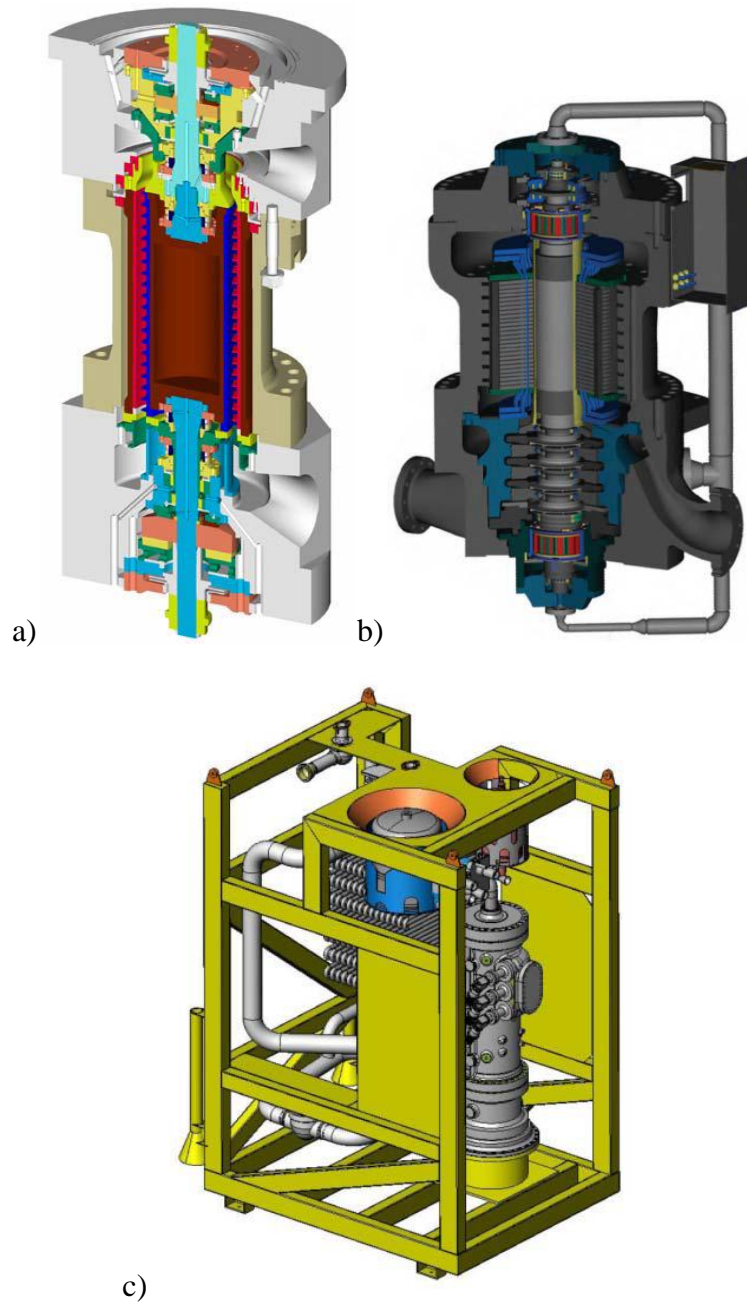


Figure 4.8 WDC 2000 (a), ECO II Concept (b), Ormen Lange subsea compressor module from Nuovo Pignone (c) [8]

5. MODELING OF WET GAS COMPRESSION

This chapter will explain how the wet gas compression process has been modeled. First the structure of the equation of states used in this work (PR, SRK and CPA) will be explained. Subsequently, it will be described in details the modeling of the wet gas compressor using NeqSim and Java.

5.1 EQUATIONS OF STATE

The main processes used in the oil industry are Separation, Dehydration, Fractionation, Acid Gas Removal and Liquefaction. All of these processes need precise temperature and pressure, and they involve continuous evaporation and condensation. Important data is the Vapour-Liquid Equilibrium, since in almost all the processes it is important to know how much gas is in the liquid phase and vice versa in order to size the different part of the process. Finally, in particular for LNG, thermal and caloric properties of gases are key elements for proper compressors and heat exchangers design.

Other applications of gas modelling are the design and performance evaluation of compressors. Compressors are used to maintain the correct pressure along the Natural Gas chain from reservoirs. Correct evaluation of thermodynamic properties allows a good estimation of work required from the compressor, thus sensibly reducing cost uncertainties. As mentioned in the introduction, this work is directed to wet gas compression. Wet gas, compared to dry gas, is a mixture of natural gas, oil, water and MEG. The most common EoS currently available have been developed for Natural Gas and not for wet gas. When dealing with wet gas compression, a proper equation of state becomes of vital importance. The main reason is that phase equilibrium has a great impact on performance, and any mistake in the mass fraction calculation will result in to sensible mistakes in efficiency and power needed [15]. At the current status, neither ASME nor ISO standards specify what Equation of State should be used for the compressor's performance calculations. This underlines the importance of a proper investigation of currently available EoS and the need to develop a reliable EoS also for wet gas applications.

All the equations of state are improvements of the well-known Ideal Gas Law, that is expressed in the following equation: [16]

$$PV = nRT \quad (5.1)$$

Where:

P = Pressure

V = Volume

T = Temperature

n = Moles number of gas

R = Ideal gas constant (8.3144621 J/K-mol)

The Ideal gas law is valid only for extremely low pressures, where the molecules can be considered mono-atomic and their volume neglected, and high temperatures, where the high kinetic energy make negligible the interactions between molecules. It is obvious that an equation like this has few applications in oil industry, where the operating pressure and temperature range is usually large and the molecules are complex carbon chains. Thus since several decades, effort has been made to build an improved equation able to describe real gases.

Real gas behaviour is expressed by a real gas law that includes the so called “compressibility factor”, which takes into account the non-ideal behaviour of a gas. The real gas law is reported in the following equation: [17]

$$PV = ZnRT \quad (5.2)$$

Where "Z" is the compressibility factor.

The equations currently used in oil industry are improvements of the ideal gas law equation of state and real gas law.

5.1.1 SOAVE-REDLICH-KWONG (SRK) AND PENG ROBINSON (PR)

In 1972, Soave proposed an important modification to the Redlich-Kwong EoS and since that time it has been the main modification to the Van Der Waals equation. The resulting equation, called Soave-Redlich-Kwong equation of state (SRK) is the following: [18]

$$p = \frac{RT}{v - b} - \frac{a(T)}{v(v + b)} \quad (5.3)$$

The term $a(T)$ is defined as follows:

$$a_i(T) = a_{ci} \alpha_i(T) \quad (5.4)$$

$$a_{ci} = \frac{0.42747R^2T_{ci}^2}{p_{ci}} \quad (5.5)$$

$\alpha_i(T)$ is an adimensional factor that becomes unity at $T = T_{ci}$, which is the critical temperature of the element "i".

Peng and Robinson proposed the following modification for a new equation of state (PR EoS): [19]

$$p = \frac{RT}{v - b} - \frac{a(T)}{v(v + b) + b(v - b)} \quad (5.6)$$

The term $a(T)$ is defined as proposed by Soave in 1972.

PR offer the largest range of operating conditions and greatest variety of systems, and thus is the most commonly used EoS for gas processing applications. Both SRK and PR directly generate all required equilibrium and thermodynamic properties. In HYSYS, they contained enhanced binary interaction parameters for hydrocarbon-hydrocarbon pairs (they are a combination of fitted and generated interaction parameters), and for several hydrocarbon-non hydrocarbon binaries. In the case when pseudo components or non-library components are used, HYSYS generates automatically interaction parameters in order to improve the VLE data. However, this is not the case for this work since all the components used were present in HYSYS library. [15]

It is important to underline that PR applies some specific component-component interaction parameters. In this work, the components used that have these special parameters are: Nitrogen, Water, Carbon Dioxide and Ethylene Glycol (MEG).

A comparison of formulation for PR and SRK EoS as implemented in HYSYS is reported in the following Figure:

	Soave Redlich Kwong	Peng Robinson
	$P = \frac{RT}{V-b} - \frac{a}{V(V+b)}$ $Z^3 - Z^2 + (A-B-B^2)Z - AB = 0$	$P = \frac{RT}{V-b} - \frac{a}{V(V+b) + b(V-b)}$ $Z^3 + (1-B)Z^2 + (A-2B-3B^2)Z - (AB-B^2-B^3) = 0$
where		
$b =$	$\sum_{i=1}^N x_i b_i$	$\sum_{i=1}^N x_i b_i$
$b_i =$	$0.08664 \frac{RT_{ci}}{P_{ci}}$	$0.077796 \frac{RT_{ci}}{P_{ci}}$
$a =$	$\sum_{i=1}^N \sum_{j=1}^N x_i x_j (a_i a_j)^{0.5} (1 - k_{ij})$	$\sum_{i=1}^N \sum_{j=1}^N x_i x_j (a_i a_j)^{0.5} (1 - k_{ij})$
$a_i =$	$a_{ci} \alpha_i$	$a_{ci} \alpha_i$
$a_{ci} =$	$0.42748 \frac{(RT_{ci})^2}{P_{ci}}$	$0.457235 \frac{(RT_{ci})^2}{P_{ci}}$
$\alpha_i^{0.5} =$	$1 + m_i (1 - T_{ri}^{0.5})$	$1 + m_i (1 - T_{ri}^{0.5})$
$m_i =$	$0.48 + 1.574\omega_i - 0.176\omega_i^2$	$0.37464 + 1.54226\omega_i - 0.26992\omega_i^2$ When an acentric factor > 0.49 is present HYSYS uses following corrected form: $0.379642 + (1.48503 - (0.164423 - 1.016666\omega_i)\omega_i)\omega_i$
$A =$	$\frac{aP}{(RT)^2}$	$\frac{aP}{(RT)^2}$
$B =$	$\frac{bP}{RT}$	$\frac{bP}{RT}$

Figure 5.1 Parameters for SRK and PR equations used in HYSYS [15]

For both NeqSim and HYSYS, the mixing rule used for PR and SRK is the classic Van Der Waals mixing rule.

5.1.2 CUBIC PLUS ASSOCIATION (CPA)

Cubic Plus Association Equation of State was created to develop a new EoS able to extend the current EoS for compounds containing polar/hydrogen bonds, and thus that can be able to cover almost all the needs of the oil industry. CPA equation is the following: [20]

$$P = \frac{RT}{V_m - b} - \frac{\alpha(T)}{V_m(V_m + b)} - \frac{1}{2} \left(\frac{RT}{V_m} \right) \left(1 + \frac{1}{V_m} \frac{\partial \ln g}{\partial \left(\frac{1}{V_m} \right)} \right) \sum_i x_i \sum_{A_i} (1 - X_A) \quad (5.7)$$

Where V_m is the molar volume, X_A is the fraction of A-sites of molecule "i" that are not bonded with other active sites, and x_i is the mole fraction of the component "i". As can be seen the equation is the sum between SRK EoS and the contribution of association term as given by Michelsen and Hendriks. [21]

The CPA is reduced to a cubic EoS when no compounds with hydrogen bonds are involved. Normal cubic EoS however, do not give satisfactory results when dealing with mixtures containing hydrogen bonds, in particular when VLE and LLE data are of interest.

CPA combines the simplicity of a cubic EoS and the association term of Wertheim theory. This term takes into account the specific site-site interaction of hydrogen bonding between similar molecules (self-association) and unlike molecules (cross-association). It is well known that SRK gives interesting results when dealing with simple gas mixtures or simple heavy hydrocarbon compounds. it was thus necessary to perform an adjustment to the current cubic EoS available, in order to extend their application to almost all the processes involved in the oil industry nowadays. Especially when dealing with wet gas processing, polar/hydrogen bonds play an important role. Wet gas is for definition a mixture of gas hydrocarbons (for which cubic EoS have proven good accuracy in properties estimation) and liquid phase composed of water, glycols and heavy hydrocarbons. The presence of water and glycols require an EoS that is able to handle the molecules with hydrogen bonds. [20] In this work, it will be studied in more details if increased accuracy of CPA with components that have hydrogen bonds will have a

sensible effect on a compressor's performance analysis. In NeqSim, for CPA, a temperature dependent mixing rule has been used.

5.2 MODELLING OF WET GAS COMPRESSION ON NEQSIM

The backbone of this work is the Thermodynamic model used for calculating all the properties needed, such as compressor work and outlet temperature.

The software used in this work is called NeqSim and is an open source project for thermodynamics and fluid-mechanics simulations. With this program, it is possible to simulate the most common operations in petroleum industry where any users can contribute with their own modules and models to the code. NeqSim is an abbreviation for Non-EQuilibriumSIMulator. The development of the software started in 1998 by Even Solbraa.

Using Java on the platform NetBeans it was possible to use the software NeqSim in an algorithm able to solve all the necessary equations. The model is divided in two main parts: Polytropic efficiency calculator and Compressor Simulator.

5.2.1 POLYTROPIC EFFICIENCY CALCULATOR

This first part of the model is the simplest one. It is intended to analyze the difference of the different Equation of State (EoS) used in estimating the polytropic efficiency. In other words, it would be possible to know how the differences between the EoS selected in this work will affect the final result.

In the oil industry, when designing a compressor for a certain reservoir, some parameters can be obtained from Reservoirs Engineers. These parameters are the reservoir pressure and temperature. They are the inlet pressure and inlet temperature of the compressor (P_{in} and T_{in}). It will also be possible to obtain the gas composition, which usually is in moles fraction. On the other hand, Process Engineers will provide the required pressure for the downstream processes. This pressure will be the outlet pressure of the compressor (P_{out}).

When the compressor is selected and designed, tests can be run in a laboratory in order to get the outlet temperature (T_{out}).

At this point, all the needed parameters are available. By using the Direct Integration Approach (DIA) it is possible to obtain the correct polytropic efficiency by iterating. The iteration method chosen in this case is the Newton method, since it is simple and easily converges. The iteration process consists in guessing a first value of polytropic efficiency

(usually 0.8 is considered a good guess) and run the DIA in order to obtain a rough estimation of the outlet temperature (T_{out}). If the difference between the calculated T_{out} and the experimental one is bigger than an arbitrary error, then a new guess of polytropic efficiency is selected. In Appendix A.1 it is possible to see the code used for this part of the method. The method is represented in Figure 5.2 using a simple conceptual map.

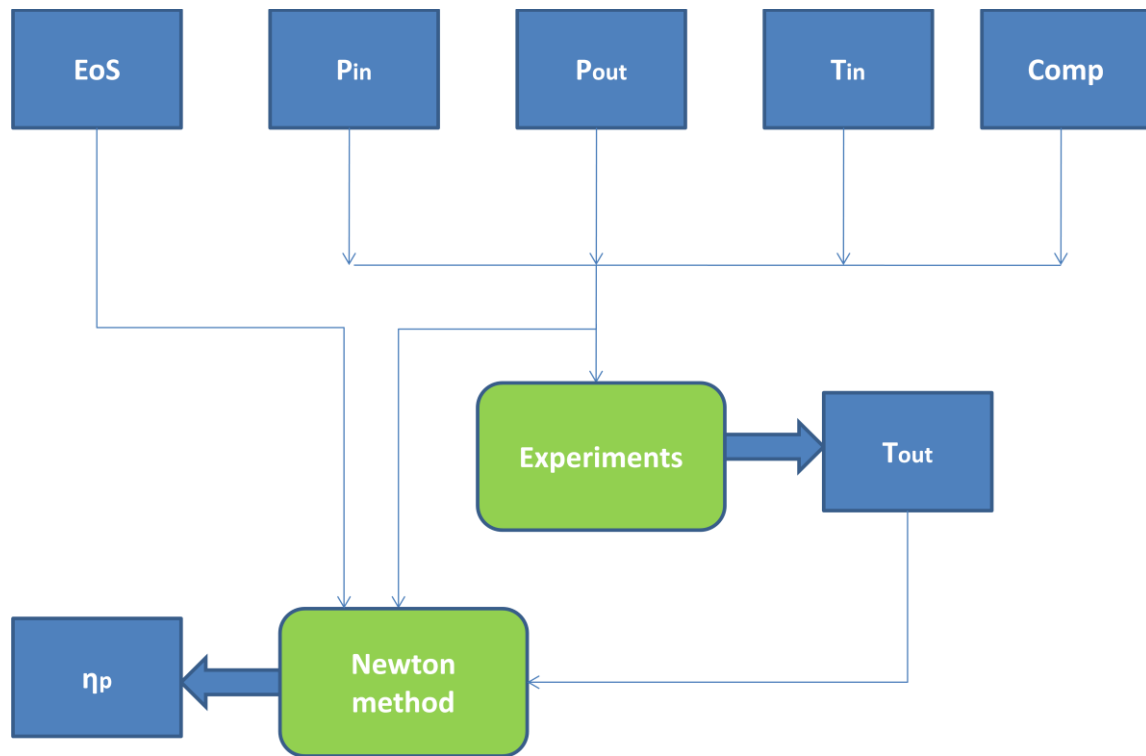


Figure 5.2 Block diagram of polytropic efficiency calculator

5.2.2 COMPRESSOR SIMULATOR

This part of the model is made in order to simulate the compression of wet gas. The simulation is purely thermodynamic, this means that only the compression of the fluid is considered and not all the mechanical parameters of the compressor. It takes the inputs and run the Direct Integration Approach (DIA), as described in Chapter 4. The main objective of the simulator is to show what happens to the compressor performance evaluation if different Equations of State (EoS) or different fluids are used. This means that the outcome of the model will be the sensitivity of the compressor parameters to the EoS used and fluid considered. Figure 5.3 represent the block diagram of the compressor simulator.

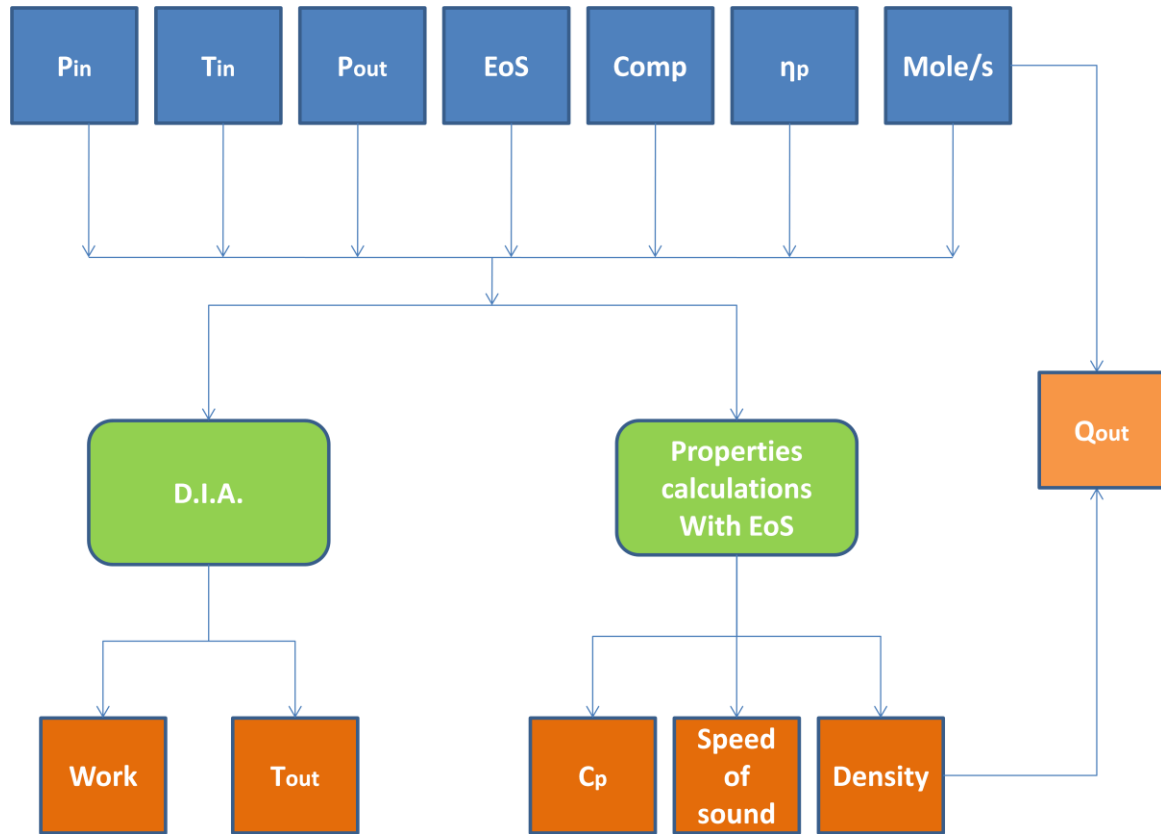


Figure 5.3 Block diagram of compressor simulator

5.3 IMPLEMENTATION OF THE DIRECT INTEGRATION APPROACH

As explained in Chapter 4, Bakken et al. [4] showed in their article that the use of the Schultz method in the polytropic analysis of a compressor is not effective when dealing with wet gas. The efficiency is often overestimated and this is due mainly to phase changes during compression. Temperature and pressure are rising along the compression path and the liquid present in the fluid can either evaporate or condensate. It is important to point out that since temperature and pressure of the gas will change, also its liquid solubility will do so. This means that, depending on the conditions, condensation or dilution of liquid can occur.

The Direct Integration Approach will take all the changes in the fluid into account by continuously updating the fluid properties as the compression proceed. This is done by dividing the compression path in small compression steps, with equal pressure ratio.

Current Equations of State (EoS) linked with powerful calculation tools allow precise estimation of enthalpy and entropy. This makes the direct integration of polytropic head possible based on the following formula:

$$\eta_p = \frac{\partial H_{is}}{\partial H} \quad (5.8)$$

Where " H_{is} " is the isentropic enthalpy, H is the real enthalpy and η_p is the polytropic efficiency.

In this way the real gas properties along the compression path are used, instead of average ones as in Schultz method. The polytropic efficiency is assumed constant along the compression path and the ISO standards allow the isentropic compression approximation by using a large number of steps:

$$h_p = \sum_{i=1}^n \partial h_s \quad (5.9)$$

Where $\sum_{i=1}^n \partial h_s$ is the sum of all the enthalpy differences in all the compression steps.

Referring to Figure 5.4, the model receives the arbitrary number of steps (N_{steps}), the pressure at the step inlet ($P_{in,step}$) and the overall pressure outlet from the compressor (P_{out}) and it calculates the delta pressure of each step and their pressure outlet (ΔP and $P_{out,step}$). By using the temperature and pressure at the step inlet ($T_{in,step}$), the method uses the selected Equation of State (EoS) in order to calculate both the entropy and enthalpy at the step inlet (S_{in} and H_{in}).

In this way the method has all the data to perform an isentropic compression by simply performing a Pressure-Entropy flash (PS flash) at the pressure of the step outlet and the entropy calculated before. The outcome of the PS flash is the isentropic enthalpy at the step outlet ($H_{out,is}$). By using the polytropic efficiency (η_p) the method calculates the related real enthalpy at the step outlet ($H_{out,step}$). At this point, using Eq. 5.8 it is possible to obtain the total head of the compression process. The method will then perform a Pressure-Enthalpy flash (PH flash) at the step outlet pressure and enthalpy. The outcome

of this PH flash is the temperature of the step outlet ($T_{out,step}$). The outlet temperature of the final step is the required compressor outlet temperature.

In Figure 5.4 is possible to see the whole process represented.

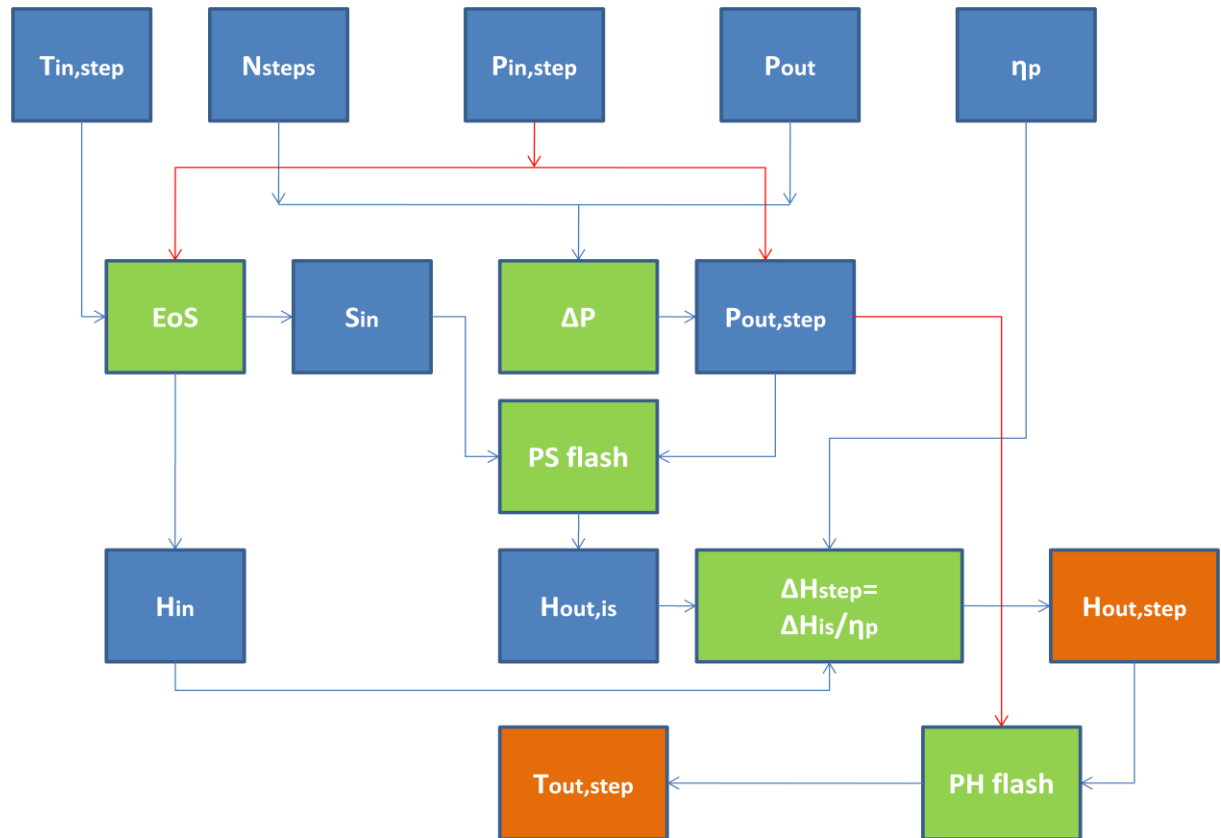


Figure 5.4 Block diagram of Direct Integration Approach implementation in NeqSim

6. IMPLEMENTATION OF EVALUATION

In this chapter it will be explained how the evaluation has been done and what is its structure. It will be explained in detail what parameters have been changed in order to verify if the results are affected considerably by their variation.

6.1 STRUCTURE OF EVALUATION

First, as described previously in Chapter 4, the main parameters for a compressor of wet gas have been identified. These properties can be summarized as "Inlet Parameters", that are fluid density, isobaric specific heat, speed of sound, Gas Volume and Mass Fractions (GVF and GMF respectively), and "Outlet Parameters", that are the outlet temperature, the polytropic efficiency and the polytropic head.

The next step is to understand what influences these parameters. However the sensitivity to different EoS could change according to the operating conditions. Operating conditions may change, for example, by changing the operating pressure and the composition of the mixture. Furthermore, the sensitivity may also change according to the number of steps used in the Direct Integration Approach (DIA). The overall procedure can be summarized in the following scheme:

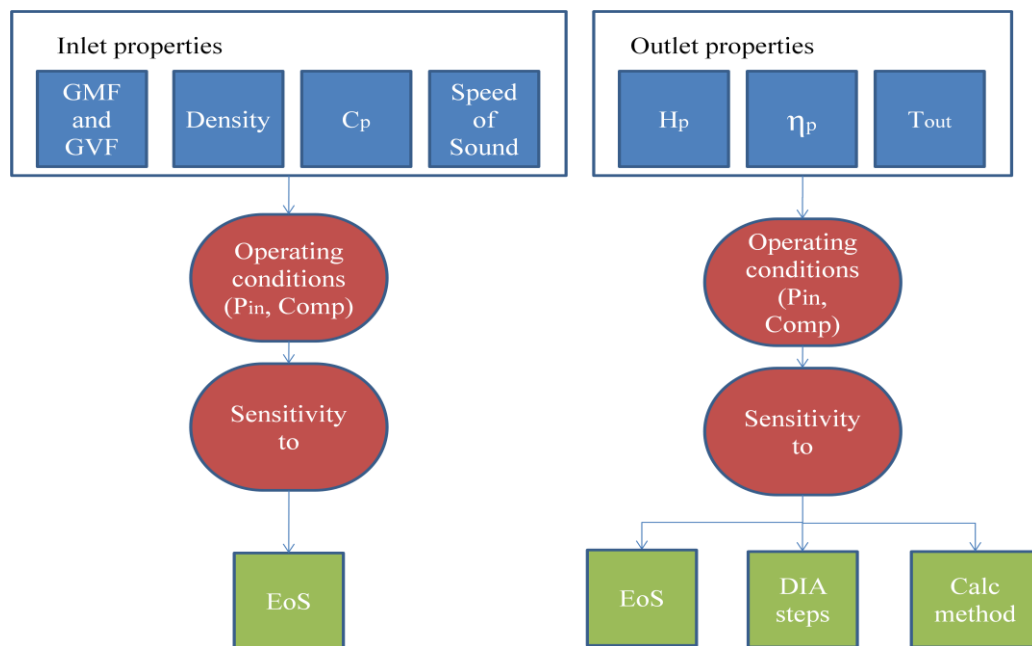


Figure 6.1 Schematic of evaluation's structure

As it can be seen, the sensitivity of "Inlet Properties" to the EoS used and the sensitivity of "Outlet Properties" to EoS used will be studied in order to find any correlation between them. It will be also studied how this sensitivity is affected by the operating conditions. This will be done by changing the operating pressure to the compressor and the fluid composition.

Regarding the calculation method used, in this thesis only the DIA will be implemented since the time available does not allow repeating all the work for each method used in gas compression. Furthermore Bakken et al. [4] already investigated what method is expected to be more accurate for wet gas compression. The outcome of their analysis was that the DIA is more accurate for wet gas calculation, since it takes into account the phase changes during compression.

6.2 VARIATION OF PARAMETERS

Once the structure of the evaluation is decided, it is important to know how the parameters have been changed in order to implement the evaluation.

6.2.1 EQUATION OF STATE

The selected Equations of state (EoS) are:

1. Soave Redlich Kwong (SRK)
2. Peng Robinson (PR)
3. Cubic Plus Association (CPA).

The reason for this choice comes from the preliminary work of this thesis [3] where the accuracy of different EoS has been evaluated in order to determine which EoS may be more suitable for wet gas compression. The result of this comparison was that SRK, PR and CPA might be the most suitable EoS for wet gas since they are able to handle water, MEG and oil.

6.2.2 COMPOSITION

6.2.2.1 MIXTURES

As explained in Chapter 3, wet gas is a mixture of different elements like water, natural gas, MEG and oil.

However, the proper definition for "wet gas" is Natural Gas with a Gas Volume Fraction (GVF) between 0.99 to 0.95 %. This means that the liquid fraction can be any type of liquid, like water, MEG, oil or all these three liquids mixed together.

A simple straightforward procedure has been chosen to implement this evaluation. The reference case is the dry gas, which is a mixture of hydrocarbons from methane to n-pentane plus Nitrogen and Carbon Dioxide.

Afterwards, water, MEG and oil have been separately injected in the dry gas and the properties of the compressor have been calculated.

As the final case, a complete wet gas mixture has been considered.

To summarize, the different mixtures are:

1. Dry Gas (Methane to n-Pentane, Nitrogen and Carbon Dioxide)
2. Dry Gas + Water
3. Dry Gas + MEG
4. Dry Gas+ Oil
5. Wet Gas (Dry Gas + Water + MEG + Oil)

The composition of the mixture has been taken from data from Statoil about the wet gas coming from Åsgard reservoir provided by Professor Lars Eirik Bakken. This composition can be seen in Table 6.1:

Table 6.1 Wet gas composition used in this work

Composition	Mole Fraction [-]
Methane	0.88316
Water	0.039944
MEG	0.0173083
N2	0.00296682
CO2	0.00209667
ethane	0.028915
propane	0.0116637
i-butane	0.00222488
n-butane	0.00315726
i-pentane	0.00097763
n-pentane	0.00104928
C6	0.00137641
C7	0.0025341
C8	0.0023069
C9	0.00031771
C10	1.88549E-06

6.2.2.2 IMPLEMENTATION OF COMPOSITION IN NEQSIM AND RELATED PROBLEMS

In NeqSim the units for inlet fluid are mol/sec, thus the different mixtures have been obtained by simply multiplying the mole fraction from the composition above by a factor 10, in order to obtain the mol/sec for NeqSim. For example, methane has a concentration of 0.88 mole fraction, thus in NeqSim will have an inlet flow of 8.8 mol/sec.

In Table 6.2 it is reported the total mass and volume flow (at 44 bar and 298 K) for each mixture:

Table 6.2 Mass and volume flow for the different mixtures used in this work

	Mass flow [kg/s]	Volume flow [m³/s]
Dry Gas	1.610E-01	4.263E-03
Gas+ Water	1.682E-01	4.270E-03
Gas+ MEG	1.718E-01	4.272E-03
Gas + Oil	1.681E-01	4.262E-03
Wet gas	1.860E-01	4.279E-03

These volume flow is calculated at equilibrium condition using CPA EoS.

Furthermore, the different mixtures have been obtained by deleting the components not of interest from the list above. So the case of mixture number 2 (dry gas + water) has been obtained by deleting MEG and oil from the list of components in NeqSim.

In this way the GVF and GMF change from case to case, as well as the total volume and mass flow.

For example, the GVF for the "dry gas + water" case is 0.99855 (calculated using CPA), while in the case "dry gas + MEG" is 0.9979. This difference will affect the results and has to be taken into account when evaluating the results.

The ideal case would have been to keep the same GVF and GMF for all the cases in order to see what could be the influence of the liquid phase composition on the evaluation. The reason for choosing this other way to proceed is that there was no way to set up an "equal" comparison, where both GMF and GVF are constant. In fact, since the liquids considered

(water, MEG and oil) have different densities (respectively 1000 kg/m³, 1100 kg/m³ and 600-700 kg/m³), if the same GVF is considered, the GMF would have been inevitably different and vice-versa.

The conceptual model of the inlet stream is the following:

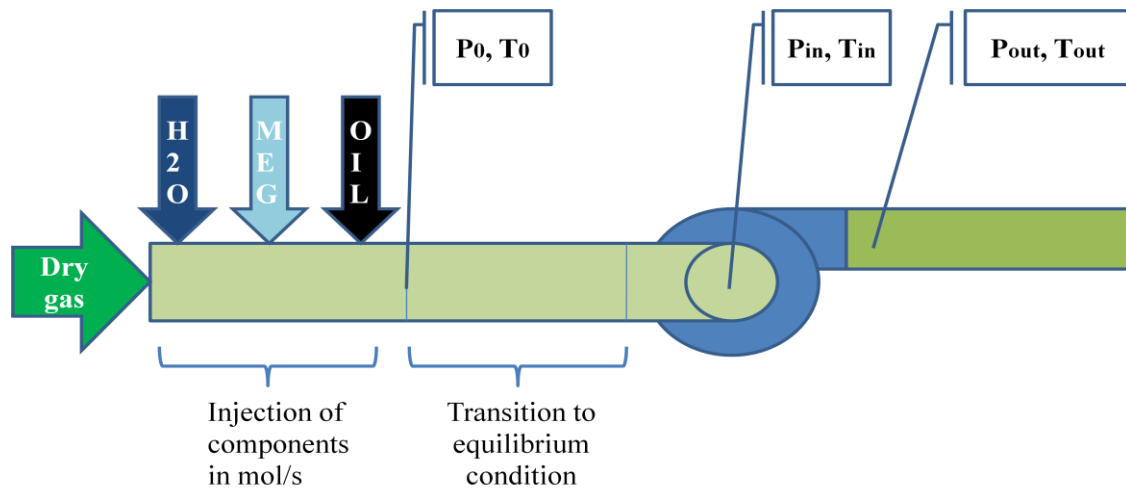


Figure 6.2 Schematic of components injection in compression process

In this figure there is a schematic representation of the compressor system. The inlet pipe (pale green), the compressor (blue) and outlet pipe (green). The inlet pipe has originally only dry gas at the inlet. Subsequently, according to what composition is needed in the evaluation, are injected water, MEG and/or oil. After the injection, the stream reaches equilibrium conditions and thus some gas will go into the liquid phase and some liquid will saturate the gas phase. This equilibrium happens at a certain pressure and temperature, here called T_0 and P_0 . After mixing, the fluid is at P_{in} and T_{in} , which are defined in the model.

6.2.3 OPERATION PRESSURE OF COMPRESSOR

As reported in the preliminary work of this thesis [3], accuracy of the Equations of State changes when temperature and pressure change.

A proper evaluation should take into account also these variations, thus it is necessary to analyze low, medium and high pressure and temperature cases.

The operating inlet pressure and temperature will change for different reservoirs as well as when a reservoir is discharging. The outlet pressure is depending on the process upstream of the compressor, which is always different since it depends on the distance to the reservoir from the process plant and the type of process receiving the gas extracted. A simplified scheme of the process is represented in the figure below:

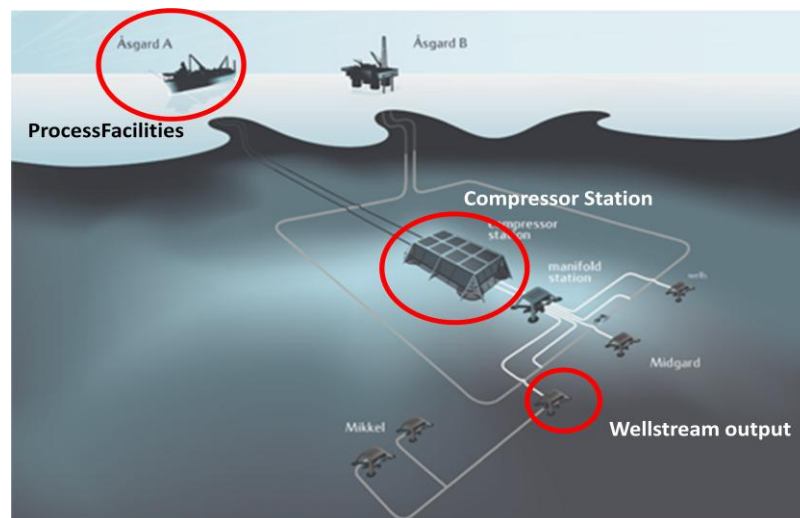


Figure 6.3 Representation of extraction-compression-processing process in Åsgard [1]

Different compressor design and operation conditions (i.e. different required volume flows) will change the operation temperatures.

Because of time limitations, a complete evaluation is not possible in this work, thus only two cases have been chosen. These cases are the medium pressure and low pressure ones.

In the medium pressure case, the actual values of Åsgard wet gas compression from Statoil have been used. In the low pressure case, temperature has been kept constant while the operating pressure of the compressor has been lowered in order to study the behavior of the EoS at lower pressures.

It is possible to see the values in the following table:

Table 6.3 Medium and low pressure case conditions

	Inlet Pressure [bar]	Pressure outlet [bar]	Pressure ratio [-]	Temperature inlet [°C]
Medium Pressure case	44	117	2.659	15
Low Pressure case	10	26.6	2.659	15

The two compression paths are represented in a PT diagram in Figure 6.4. The compression inlet and outlet conditions for both cases are far from the boiling point of water and MEG thus the only evaporation that will occur will be due to solubility of water and MEG in the gas phase.

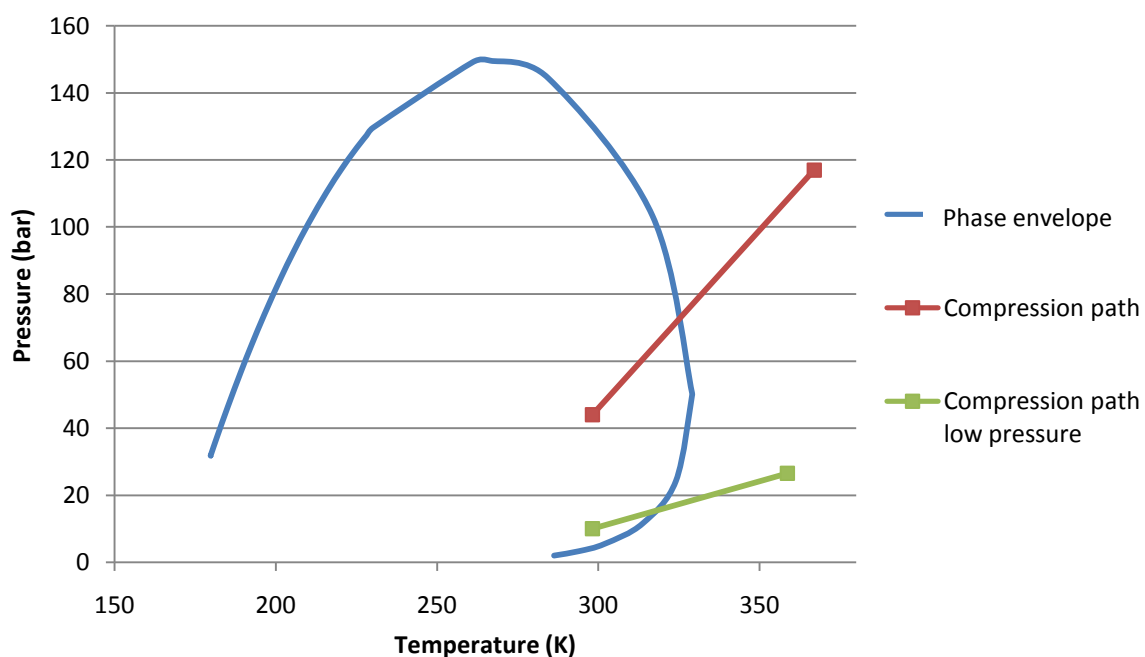


Figure 6.4 Phase envelope of wet gas calculated using SRK EoS in HYSYS.

6.2.4 DIRECT INTEGRATION APPROACH STEPS

As explained in Chapter 6, the Direct Integration Approach (DIA) consists in dividing the compression path in "small" steps with constant pressure difference. This process is shown in the figure below:

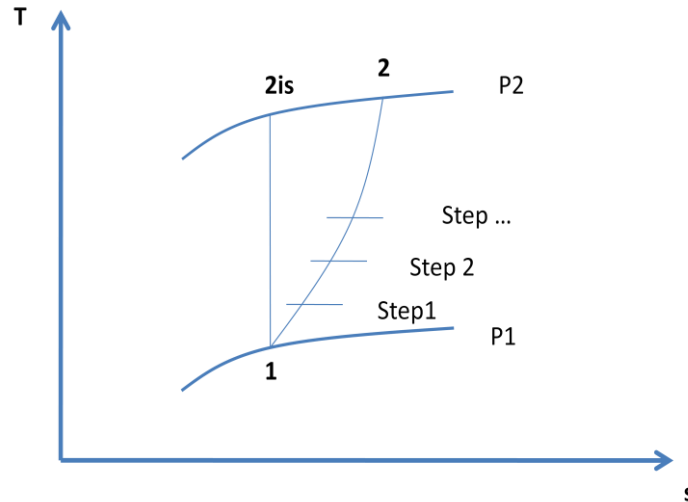


Figure 6.5 Steps subdivision in Direct Integration Approach

It is expected that if less steps are taken into account, the phase change during the compression will be approximated with bigger errors.

When instead the number of steps is increased, the phase changes during the compression will be calculated with higher accuracy but the mistake relative to the EoS selected (since no EoS gives accuracy of 100%) are expected to sum up and have negative effect on the overall accuracy of the method.

The chosen number of steps to verify this phenomenon is 4, 40 and 400 steps.

7. VALIDATION OF THE MODEL

The aim of the validation is to be sure that the model used gives realistic results and that it does not contains substantial errors.

Thus, a direct comparison between the results of the model in this work and a well-proven commercial software has been selected. The software selected is HYSYS, a chemical process simulator widely used in the oil industry.

The comparison was made for the case of dry gas compression. The operation fluid is the dry gas mixture described in this work and the compression happened in the same conditions as for the medium pressure case with dry gas explained in this work. However, an evaluation of the differences between NeqSim and HYSYS when wet gas is used as operation fluid was also be implemented.

The results of the two software depend on different factors:

- How the equation of state are implemented in the software
- The calculation method used
- The fluid properties database used

One of the objectives of model validation could have been evaluating how much the calculation method effectively influenced the calculations. The calculation method used in NeqSim was the direct integration approach while in HYSYS the Schultz method was used. However, the two software implement in different ways the equations of state and they have different fluid databases, which do not allow making reliable evaluations. In other words, the deviation between the two software could be due to the combined effect of the previously described differences and not only to the calculation method selected. Furthermore on HYSYS it is not possible to use CPA EoS.

This validation is intended as confirmation that the NeqSim model can be assumed realistic and the results are compatible with the ones of HYSYS.

7.1 DRY GAS

In Table 7.1 the composition of the dry gas used for the comparison is shown. As can be seen, the dry gas is the same as the one used for running the simulations with the model.

Table 7.1 Dry gas composition for validation in HYSYS

Component	Mole Fraction	Mole flow (mol/s)
Methane	0.88316	8.8316
N2	0.0029668	0.029668
CO2	0.0020967	0.020967
ethane	0.028915	0.28915
propane	0.0116637	0.116637
i-butane	0.0022249	0.022249
n-butane	0.0031573	0.031573
i-pentane	0.0009776	0.009776
n-pentane	0.0010493	0.010493

The compression process in a P-T diagram together with the phase envelope of the dry gas mixture is shown in Figure 7.1. There is no liquid at the inlet and outlet of the compressor as well as no liquid appears during the compression process.

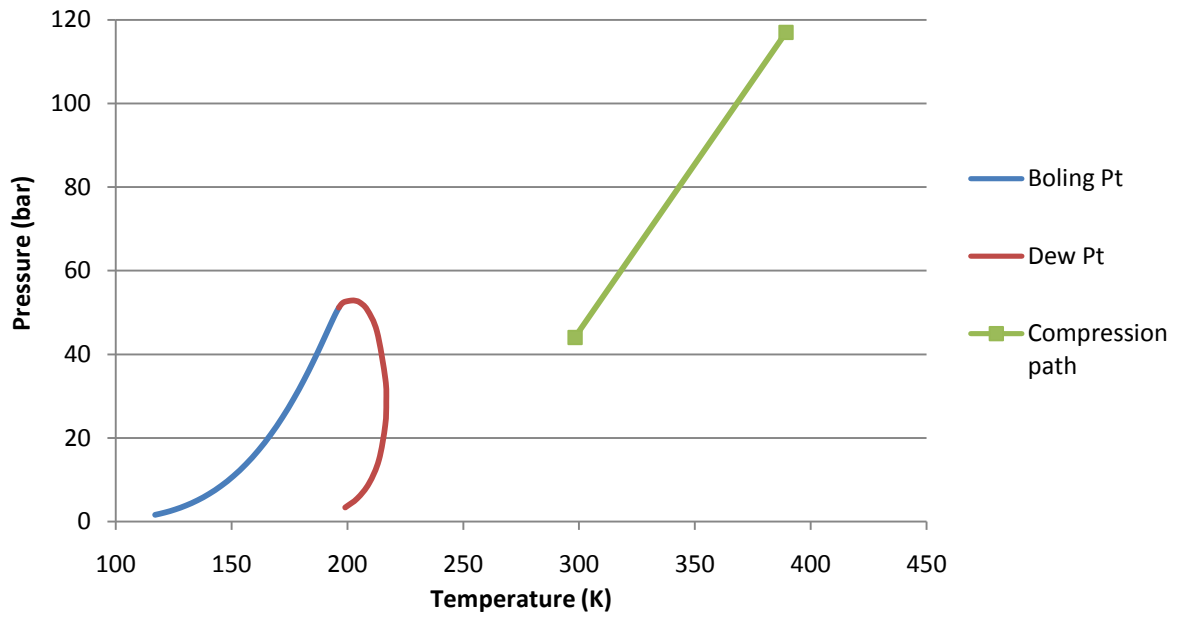


Figure 7.1 Phase envelope of dry gas calculated in HYSYS using PR EoS. In addition is represented the compression path

The phase envelope has been calculated in HYSYS using PR EoS since it is the most common equation of state when dealing with natural gas.

7.1.1 COMPRESSION OPERATION CONDITIONS

The conditions of the compression process are summarized in the following table:

Table 7.2 Compressor operation conditions in HYSYS

Pressure inlet	44 bar
Temperature inlet	298.15 K
Pressure outlet	117 bar
Polytropic efficiency	0.8

The compressor conditions are the same as the medium pressure ones described in this work.

7.1.2 CALCULATION METHOD

As mentioned before, the model developed in this work utilizes the Direct Integration Approach as calculation method for the compressor performance analysis. HYSYS instead utilize the Schultz method. This method has been described in Chapter 4.

7.1.3 RESULTS

The results of the validation with the dry gas mixture are summarized in the following tables. The comparison has been done using both PR and SRK EoS.

7.1.3.1 PR EoS

The following table summarizes the results for the validation of the model developed in this work with the PR EoS.

Table 7.3 PR EoS results summary

	HYSYS	NeqSim	Deviation from HYSYS
Outlet Temperature	391.7 K	389.3 K	- 2.4 K
Work	28.97	29.623 kW	+ 2.25 %

There is a non-negligible difference between the estimated compressor outlet temperatures, while 2.5% in compressor work estimation is considered low according to API 617. According to Hundesid et al. [22], the differences in work estimation when using the Schultz method and the direct integration approach are around 0.06%. When dealing with dry gas the difference in work estimation when using the Schultz method or the Direct Integration Approach is negligible, thus this does not explain the difference between the results from the model used in this work from HYSYS.

7.1.3.2 SRK EoS

The following table summarizes the results for the validation of the model developed in this work with SRK EoS.

Table 7.4 SRK EoS results summary

	HYSYS	NeqSim	Deviation from HYSYS
Outlet Temperature	391.9 K	389.4 K	- 2.5 K
Work	29.77 kW	30.5 kW	+ 2.49 %

The same considerations done for PR apply to this case since the differences are the same.

From the validation with SRK and PR EoS results it is possible to conclude that the model developed in this work is acceptable. Indeed the deviation in outlet temperature and work can be explained with a different implementation of the EoS. Thus, it seems that no mistakes have been done in the model development and structure, because such errors would bring results in differences "out of scale", for example 50 K of difference in outlet temperature and 20% in work estimation.

7.2 WET GAS

In the following table the composition of the wet gas used for the comparison is shown. As can be seen, the wet gas is the same as the one used for running the simulations with the model used in this work.

Table 7.5 Wet gas composition used in HYSYS for comparison with NeqSim

Component	Mole Fraction	Mole flow (mol/s)
Methane	0.88316	8.8316
Water	0.039944	0.39944
MEG	0.0173083	0.173083
N2	0.0029668	0.029668
CO2	0.0020967	0.020967
ethane	0.028915	0.28915
propane	0.0116637	0.116637
i-butane	0.0022249	0.022249
n-butane	0.0031573	0.031573
i-pentane	0.0009776	0.009776
n-pentane	0.0010493	0.010493
C6	0.0013764	0.013764
C7	0.0025341	0.025341
C8	0.0023069	0.023069
C9	0.0003177	0.003177
C10	1.89E-06	1.89E-05

The components C6, C7, C8, C9 and C10 have been implemented in HYSYS as Hexane, Heptane, Octane, Nonane and Decane. This can be considered an approximation since the element C6, for example, is not only Hexane but is a mixture of different components. This may lead to some deviations in the validation.

In Figure 7.2 the compression process in a P-T diagram together with the phase envelope of the wet gas mixture is shown. There is aqueous phase both at the inlet and at outlet of

the compressor, while there is an oil phase only at the inlet of the compressor. The compression process is far from the boiling point of water and MEG thus only the oil phase will evaporate completely.

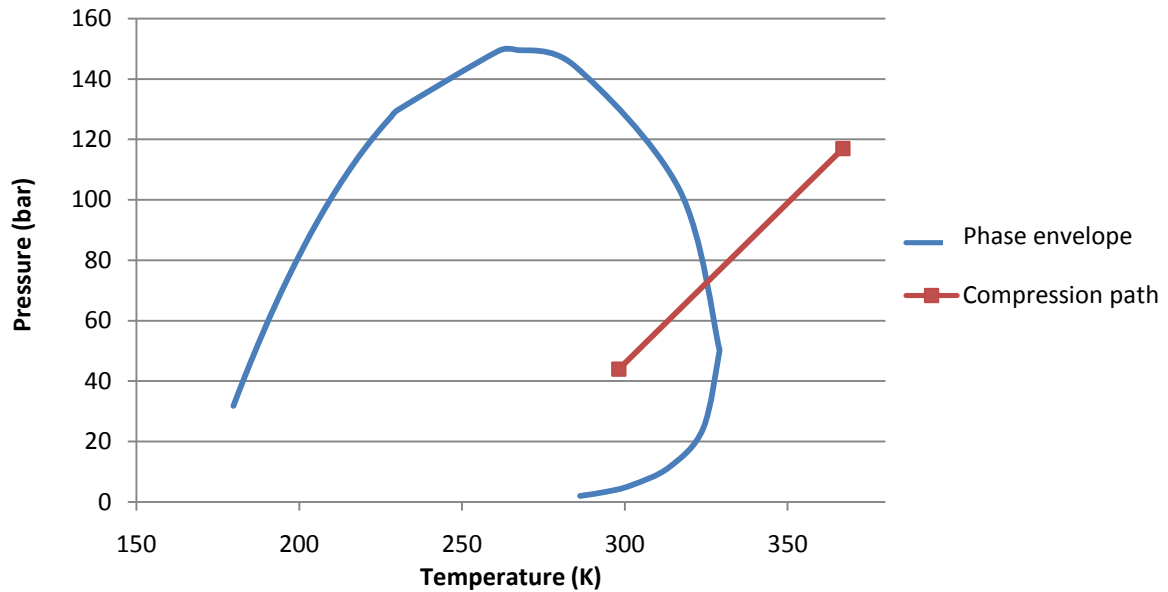


Figure 7.2 Phase envelope of wet gas calculated in HYSYS using SRK EoS. In addition is represented the compression path.

The phase envelope has been calculated in HYSYS using SRK EoS, since the preliminary project of this thesis [3] showed that is more suitable for wet gas calculations than PR. The ideal case would have been the use of CPA but unfortunately is not available in HYSYS.

7.2.1 COMPRESSION OPERATION CONDITIONS

The conditions of the compression process are summarized in the following table:

Table 7.6 Compressor operation condition used in HYSYS for wet gas calculation

Pressure inlet	44 bar
Temperature inlet	298.15
Pressure outlet	117 bar
Polytropic efficiency	0.8

The compressor conditions are the same as the medium pressure case ones.

7.2.2 RESULTS

The results of the validation with the dry gas mixture are summarized in the following tables. The comparison has been made using both PR and SRK EoS.

7.2.2.1 PR EoS

The results for the validation of the model developed in this work with the PR EoS are summarized in the following table:

Table 7.7 PR EoS results summary

	HYSYS	NeqSim	Deviation from HYSYS
Outlet Temperature	370 K	368.9 K	- 1.1 K
Work	28 kW	28.592 kW	+ 2.1 %

There is a non-negligible difference between the estimated compressor outlet temperatures while 2.1% in compressor work estimation is considered low according to API 617. It seems instead that the difference is lower for the outlet temperature (from 2.4 K to 1.1 K). This is due to the differences between how PR is implemented in NeqSim and HYSYS. HYSYS indeed contains enhanced binary interaction parameters for hydrocarbon-hydrocarbon pairs (they are a combination of fitted and generated interaction parameters), as well for lots of hydrocarbon-non hydrocarbon binaries. This may explain the similarity between the model of this work and HYSYS. Another reason for the low differences between the software may be that fluid phase change is taken into account with average properties in Schultz method as implemented in HYSYS.

7.2.2.2 SRK EoS

The following table summarizes the results for the validation of the model developed in this work with the SRK EoS.

Table 7.8 SRK EoS results summary

	HYSYS	NeqSim	Deviation from HYSYS
Outlet Temperature	369.75 K	367 K	- 2.75 K
Work	28.83 kW	29.412 kW	+ 2 %

In this case there is an higher deviation between the model used in this work and HYSYS when SRK EoS is used. The same considerations done for PR apply in this case. The lower deviation between the model of this work and HYSYS in the case of PR EoS can be caused by the fact that PR is currently the most utilized EoS in this type of simulators, thus it could be that it has been particularly optimized for processes that involves hydrocarbons.

8. RESULTS

In this chapter the main results achieved in this work will be reported and explained. First, the outlet properties deviations will be illustrated, and then the same will be done with the inlet properties. An interpretation of results as well as a parallel analysis in order to identify the causes of the behavior will be given.

In this analysis, CPA is taken always as reference.

8.1 WORK

When calculating the work required by the compressor, it is possible to study in details what properties require the highest precision.

In order to do this, the NeqSim model previously described will be used together with the following formula for a cross comparison:

$$\Delta H = \Delta H_{gas} + \Delta H_{aqueous} + \Delta H_{oil} + \Delta mH_{vap} \quad (8.1)$$

Where:

- ΔH_{gas} is the enthalpy change due to the gas phase.
- $\Delta H_{aqueous}$ is the enthalpy change due to the aqueous phase.
- ΔH_{oil} is the enthalpy change due to the oil phase.
- ΔmH_{vap} is the enthalpy change due to the evaporation of the aqueous phase.

In this case, the gas phase contains methane, ethane, propane, butane and pentane. The oil phase contains the elements here described as C6, C7, C8, C9 and C10. The aqueous phase contains water and MEG.

From Eq. 8.1 it is possible to go deeper in detail:

$$\Delta H = [(m_{gas} + \Delta m)h_2 - m_{gas}h_1] + [(m_{aqu} - \Delta m)h_2 - m_{aqu}h_1] + m_{oil}(h_2 - h_1) + [\Delta m_{water} H_{vap_{water}} + \Delta m_{MEG} H_{vap_{MEG}}] \quad (82)$$

Where:

- m_{gas} , m_{aqu} , m_{oil} are the mass of gas, aqueous phase and oil at the compressor inlet.
- Δm is the mass of aqueous phase that evaporates during compression.
 $\Delta m = \Delta m_{water} + \Delta m_{MEG}$. So the mass of aqueous phase that evaporates is the sum of the mass of water and MEG that evaporates.
- H_{vap} is the heat of vaporization calculated as an average between the compressor inlet and outlet temperature: $H_{vap} = \frac{H_{vap}(T_1) + H_{vap}(T_2)}{2}$.

From this formula is clear that only two parameters have influence in the work calculations: mass and enthalpy.

A detailed analysis of each term has to be done in order to understand the relevant parameters that affect the enthalpy calculations.

8.1.1 MASS ANALYSIS

In this section the contribution of the different masses to the work calculation will be studied in order to determine what is important and what is not.

It is possible to calculate the VLE data for wet gas by using the NeqSim model described in this work since one of the assumptions in this analysis is the Vapor-Liquid Equilibrium (VLE).

By looking at the results, it is possible to take away the terms that are negligible in order to keep the equations as simple as possible.

8.1.1.1 GAS PHASE

The mass of gas is mainly determined by the original gas input (m_{gas_0}) plus the mass of oil that dissolves into the gas.

$$m_{gas} = m_{gas_0} + m_{oil_{gas}} \quad (8.3)$$

In this work, the notation " m_{1_2} " will be used to indicate the mass of element "1" that dissolves in the element "2". In the case already mentioned of " $m_{oil_{gas}}$ " it indicates the mass of oil that dissolves into the gas phase.

8.1.1.2 OIL PHASE

The mass of oil is determined by the original mass of oil input minus the oil that dissolves into the gas and aqueous phase plus the mass of aqueous and gas phase that dissolves into the oil.

$$m_{oil} = m_{oil_0} - m_{oil_{gas}} - m_{oil_{aqu}} + m_{aqu_{oil}} + m_{gas_{oil}} \quad (8.4)$$

The results from the model show that the only significant terms are the mass of gas that dissolves in the oil phase and vice versa:

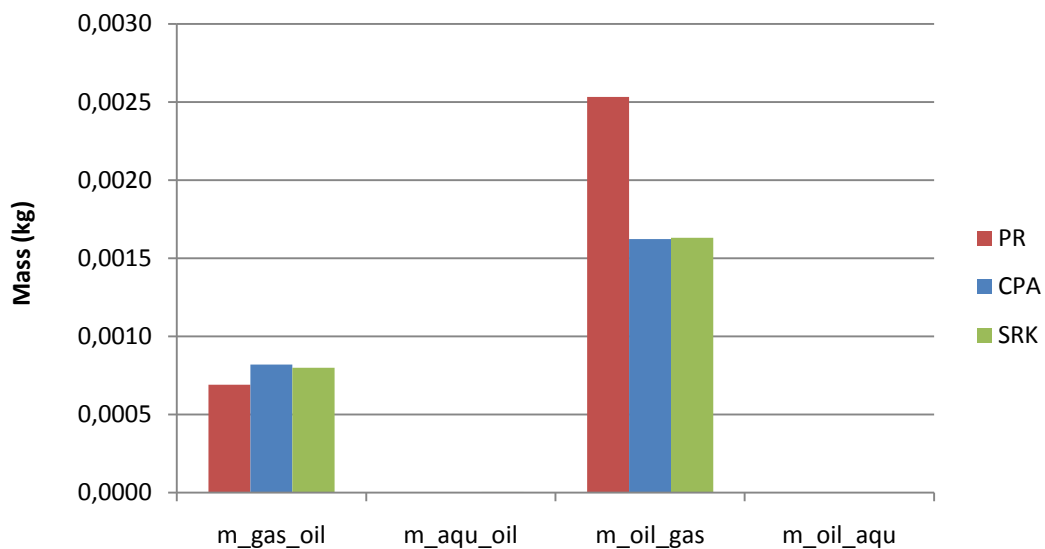


Figure 8.1 Solubility results from PR, SRK and CPA EoS for oil phase

Thus the equation for the oil mass can be rewritten as:

$$m_{oil} = m_{oil_0} - m_{oil_{gas}} + m_{gas_{oil}} \quad (8.5)$$

8.1.1.3 AQUEOUS PHASE

From the results of the model, the mass of aqueous phase at the compressor inlet is not influenced by the mass of gas and oil that dissolves into it. Thus it can be considered equal to the input value.

$$m_{aqu_{in}} = m_{aqu_0} \quad (8.6)$$

The mass of aqueous phase that evaporates can be obtained by simply doing the difference between the compressor input and output mass of aqueous phase.

$$\Delta m = m_{acq_{out}} - m_{acq_{in}} \quad (8.7)$$

The main solubility terms that contribute to the evaluation of the total aqueous phase mass at the compressor outlet are shown in the following graph:

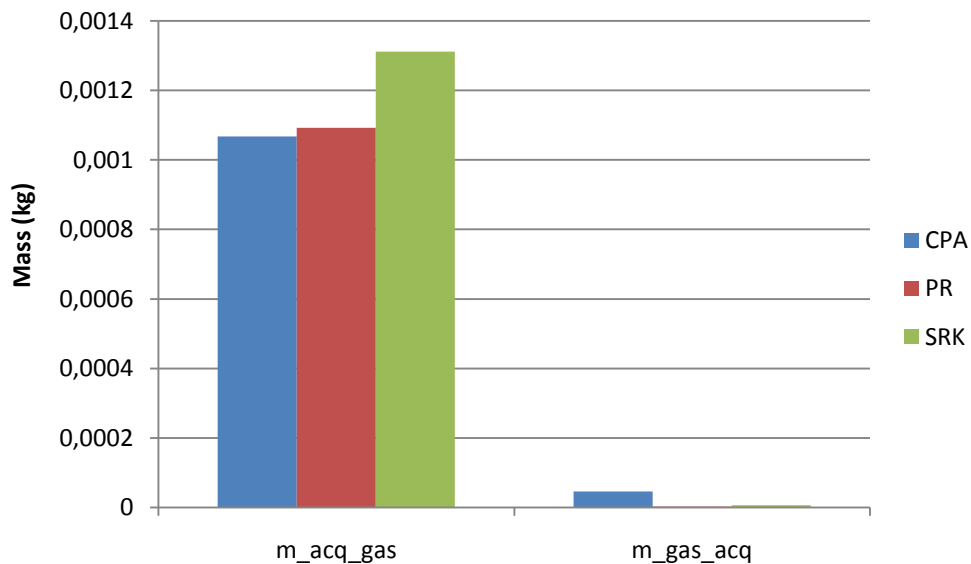


Figure 8.2 Solubility results for PR, SRK and CPA EoS for aqueous phase

As said before, the mass of aqueous phase at the inlet is not affected by the solubility of gas and oil. However is clear that the mass of aqueous phase at the compressor outlet is

determined by the original aqueous phase mass input minus the mass of aqueous phase that dissolves into the gas phase.

$$m_{acq_{out}} = m_{acq_0} - m_{acq_{gas_{out}}} \quad (8.8)$$

The mass of aqueous phase that evaporates can be rewritten as:

$$\Delta m = (m_{acq_{gas}})_{out} \quad (8.9)$$

At the compressor outlet, the mass of aqueous phase that dissolves in the gas phase is not negligible. The reason can be that at the outlet of the compressor the oil phase does not exist anymore since all the oil has evaporated. Thus, the bigger volume of gas makes the water solubility into gas more relevant to the mass balance.

8.1.1.4 CONCLUSIONS

When dealing with wet gas the EoS used should have precise estimation of VLE data. In particular, the EoS used needs to calculate precisely the solubility of oil in gas and gas in oil as well as the solubility of water and MEG in gas.

8.1.1.5 EQUATION OF STATE SENSITIVITY

In this work, as previously described, three EoS have been used: PR, SRK and CPA. It is possible to determine how much the final mass results will be affected by the chosen EoS.

From the results, the overall compression work is the following:

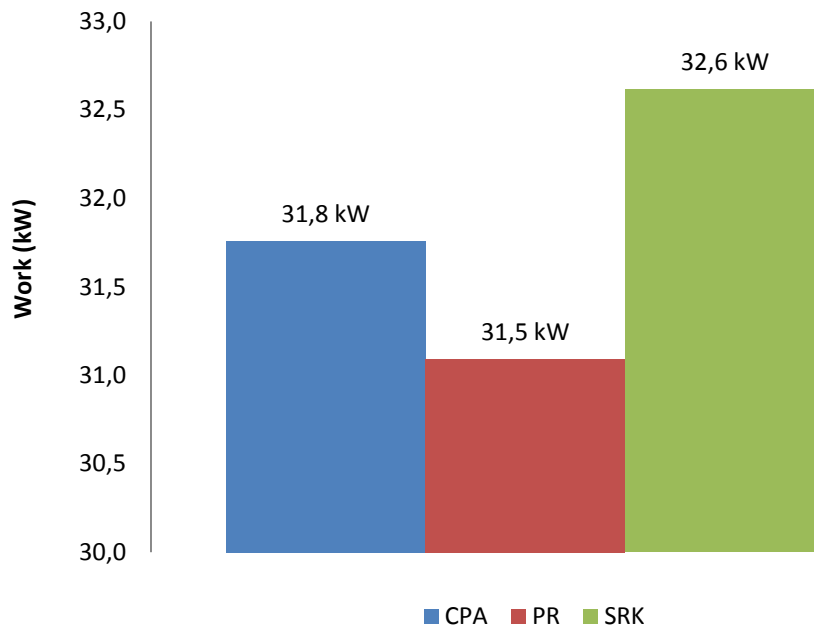


Figure 8.3 Comparison of EoS results for compressor work estimation

In order to highlight the influence of a different EoS in VLE mass calculations, the same enthalpy estimation (from CPA) has been assumed for the three EoS. In other words, the only parameter that changes in the three results are the VLE masses and not the enthalpies.

By taking CPA as reference case, the deviation in percentage points is the following:

Table 8.1 Deviation of PR and SRK from CPA EoS in work estimation

EoS	Deviation (%)
CPA	0
PR	-0.1
SRK	2.6

This difference is explained with the "radar graph" in Figure 8.4, which represents the variation in estimation of gas, aqueous and oil masses. Radar graphs are useful tools for simultaneously comparing the performance of EoS with different parameters. In the case of Figure 8.4 the parameters are the changes in evaporated, oil, gas and aqueous mass estimation. If the lines of the equations (red for PR and green for SRK) are on the 0% line (in the middle), then they have the same estimation as CPA.

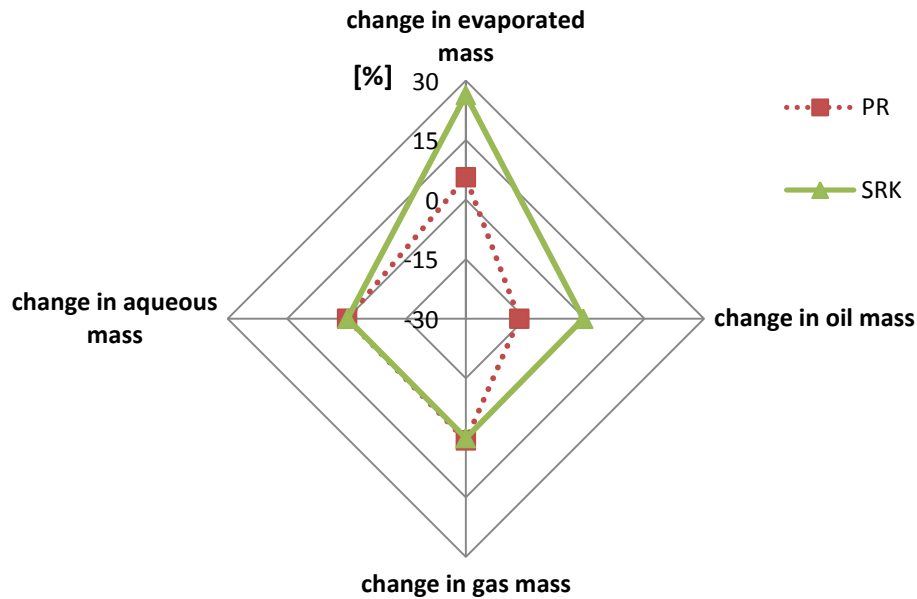


Figure 8.4 Summary of PR and SRK deviation from CPA in various parameters

SRK has the biggest difference compared to CPA (26%) in evaporated mass estimation while PR estimates a 17% lower mass of oil. The estimated mass of gas and aqueous phase are the same for all the three EoS.

However, since PR estimates also an evaporated mass different by 6% from CPA, the results have to be adjusted in order to make conclusions. Thus, the case where PR estimates the same evaporated mass as CPA (thus 0% difference) has been considered. Furthermore the case in which PR estimate the oil mass with a difference equal to the one of SRK in estimating the evaporated mass (26%) has been considered. Since both evaporated mass and oil mass would have the same deviation (26%), would be easy to see which brings more deviation in the final work estimation.

In this case the estimated work has been the following:

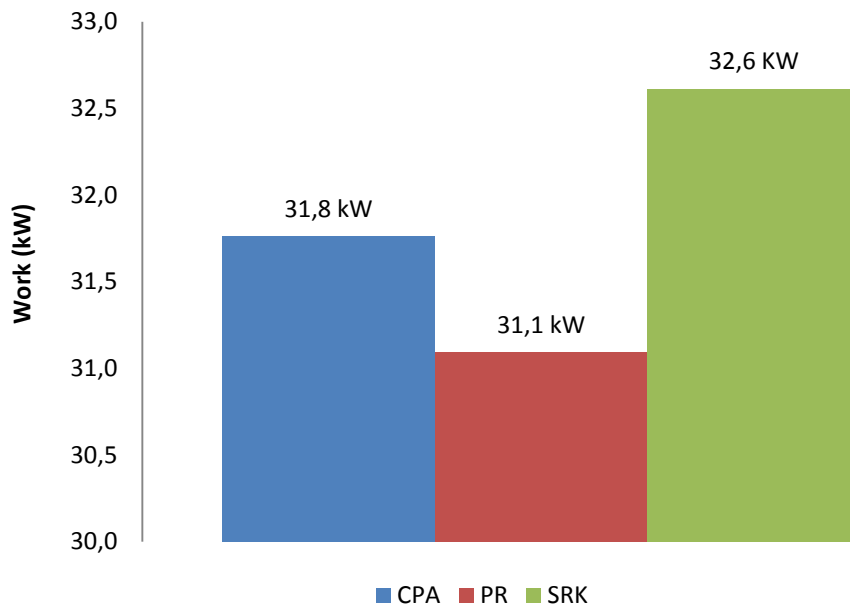


Figure 8.5 Comparison of EoS results for compressor work estimation

In order to highlight the influence of a different VLE results from the different EoS, the same enthalpy estimation (from CPA) has been assumed for the three EoS. In other words, the only parameter that changes in the three results is the VLE mass and not the enthalpy.

By taking CPA as reference case, the deviation in percentage points in this case is the following:

Table 8.2 Deviation of PR (modified) and SRK from CPA EoS in work estimation

EoS	Deviation (%)
CPA	0
PR (modified)	-2.1
SRK	2.6

It can be seen that a 26% of deviation from CPA in estimating the oil mass (PR modified) will give a work deviation of 2,1%. On the other hand, the same deviation (26%) in evaluating the evaporated mass (SRK) will give a work deviation of 2,6%. It is possible

to conclude that the oil mass estimation is slightly more important than the estimation of evaporated mass.

However another consideration has to be done. By observing the following graph it is possible to see the mass share of each component in the overall fluid mass used to calculate the work:

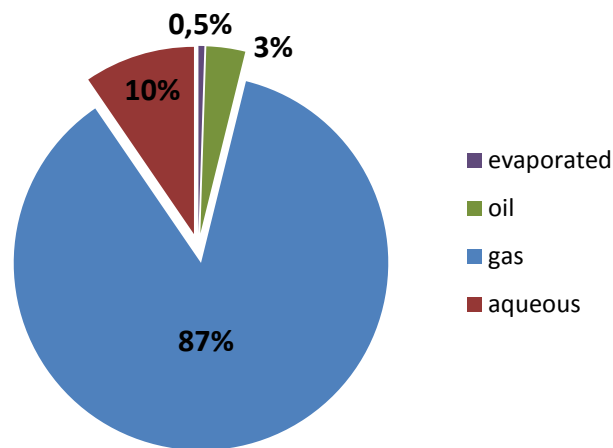


Figure 8.6 Mass shares between phases for wet gas

Thus, the gas phase accounts for 87% of the total mass while the mass that evaporates during compression is the 0,5% of the overall mass.

From the considerations done before, the evaporated mass has almost the same influence on the final work estimation as the oil mass but it accounts only for the 0,5% on the total mass while the oil mass is 3% of the overall mass. This means that the 26% variation in the evaporated mass is an absolute difference of 0,00025 kg while 26% variation in oil mass is an absolute difference of 0,0016 kg. Thus, even if the oil mass has slightly more influence on the work estimation, the evaporated mass requires higher precision due to the smaller quantity.

The reason for this will be more clear after the analysis of enthalpies. In fact the enthalpy of vaporization of water and MEG is around 2000 kJ/kg and 1000 kJ/kg respectively while the enthalpy change of the oil phase is around 500 kJ/kg.

8.1.1.6 CONCLUSIONS FOR MASS ANALYSIS

When dealing with wet gas, the EoS used should have a precise estimation of VLE data. In particular the EoS used needs to calculate precisely the solubility of oil in gas and gas in oil as well as the solubility of water and MEG in gas. These last ones need an higher accuracy.

8.1.2. ENTHALPY ANALYSIS

When evaluating what enthalpy has the largest impact, it is important to study how much is the difference between the enthalpy changes that contributes to the equation for work estimation (Eq. 8.1). The following graph reports this comparison:

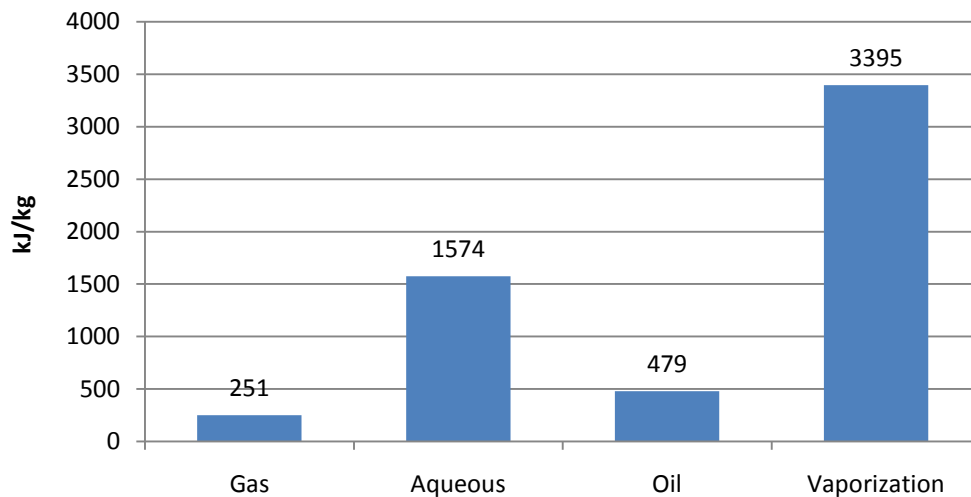


Figure 8.7 Enthalpy change value per unit mass for the different component of compressor work

From Figure 8.7 it is clear that liquid and vaporization enthalpies play an important role in the work estimation. However, these data have to be linked with the mass share of each component in order to make some conclusions. Figure 8.8 reports the enthalpy change (that is in kJ and not kJ/kg) share for each component.

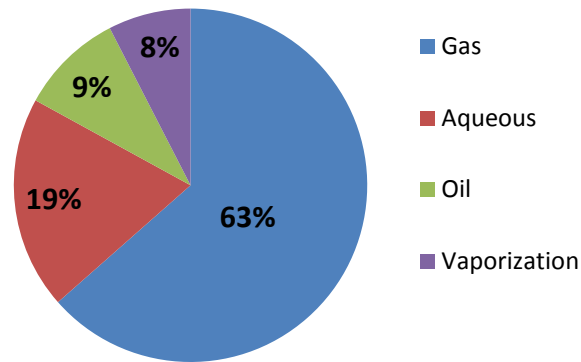


Figure 8.8 Enthalpy change share

The gas phase has a smaller enthalpy change but it consist of 87% of the total mass, thus it contributes of more than 60% in the total enthalpy change.

This means that gas enthalpy have to be estimated with higher precision. Small mistakes in enthalpy estimation will be enhanced by the fact that the error is multiplied for a big number (the gas mass). This will bring to higher deviation of total work estimation.

It can be concluded then that the enthalpy estimation that requires higher absolute precision is the gas enthalpy.

8.1.2.1 EQUATION OF STATE SENSITIVITY

In this work, as previously described, three EoS have been used: PR, SRK and CPA. It is possible then to know how much the fact to choose the PR, SRK or CPA EoS affects the final results.

In order to evaluate the difference in estimated compressor work due to different enthalpy estimation, it has been assumed that all the EoS calculate the same mass for each of the components previously described. CPA was used as reference in this case in order to calculate all the masses.

The following graph shows the calculated compressor work for each EoS under the previously mentioned assumptions:

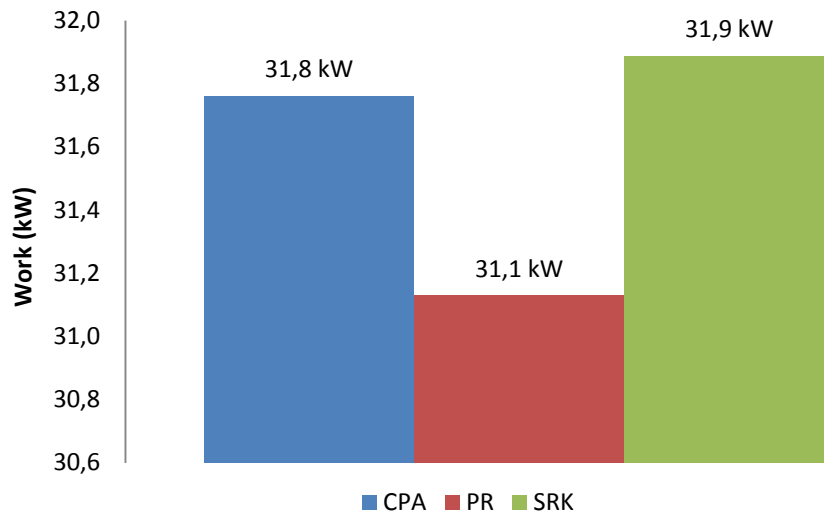


Figure 8.9 Comparison of EoS results for compressor work estimation

By taking CPA as reference, it is possible to estimate the deviation for PR and SRK from CPA:

Table 8.3 Deviation of PR and SRK from CPA EoS in work estimation

EoS	Deviation (%)
CPA	0
PR	2
SRK	1.2

This difference can be explained by looking at the enthalpy estimation deviation for each component for PR and SRK:

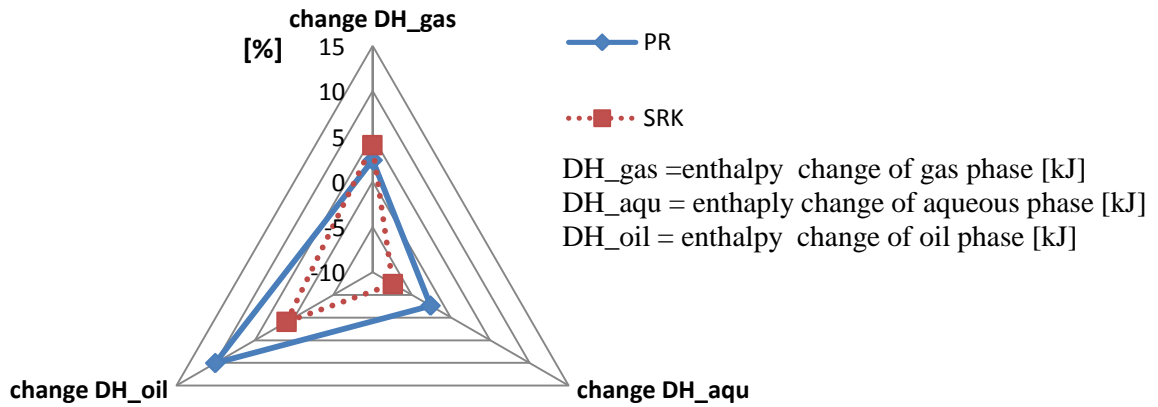


Figure 8.10 Summary of PR and SRK deviation from CPA in various parameters

The combined deviation of enthalpy change estimation for oil and aqueous phase gives a 2% deviation for PR and a 1.2% deviation for SRK.

As explained before, the oil mass share is just 3% of the total mass while the aqueous phase accounts for a 10%. Furthermore, the oil phase has an enthalpy change of around 500 kJ/kg while the aqueous phase has around 1500 kJ/kg, thus the aqueous phase has more influence on the overall work estimation. These deviations have a combined effect on compressor work estimation. In the case of SRK, even if the aqueous enthalpy estimation is higher than for PR (therefore a higher deviation from CPA is expected) it has also a 4% deviation in the gas enthalpy change estimation. As said before, the gas phase plays an important role in the total enthalpy difference (60%) thus the combined effect of this deviation and the aqueous one makes the SRK results similar to CPA than PR.

8.1.2.2 CONCLUSIONS FOR ENTHALPY ANALYSIS

As shown before, the enthalpy that requires higher accuracy is the gas enthalpy. However the influence of the enthalpy estimation of the oil and aqueous phases is not negligible.

8.1.3 CONCLUSIONS FOR MASS AND ENTHALPY ANALYSIS

The analysis done shows that when evaluating the compressor work, it is important to require from the EoS used a particular accuracy in calculating the following properties:

- Solubility of gas in oil (P=44 bar and T=298 K)
- Solubility of oil in gas (P=44 bar and T=298 K)
- Solubility of water and MEG in gas (P=117 bar and T=370 K)
- Enthalpy of gas (P=44 bar and T=298 K; P=117 bar and T=370 K)
- Enthalpy of water and MEG (P=44 bar and T=298 K; P=117 bar and T=370 K)
- Enthalpy of oil (P=44 bar and T=298 K; P=117 bar and T=370 K)

8.1.4 RESULTS OF SIMULATIONS FOR THE COMPRESSOR WORK

SRK and CPA differ by only 0,5% while PR has a constant deviation of 3% from CPA. According to API617 the maximum tolerated deviation is 4% for compressor's work, thus the work estimation is not sensible to the EoS used. In Figure 8.11, it is shown the deviation of SRK and PR compressor work estimation from the CPA one for both normal and low pressure case (see Chapter 6 for the definition of "medium" and "low" pressure case).

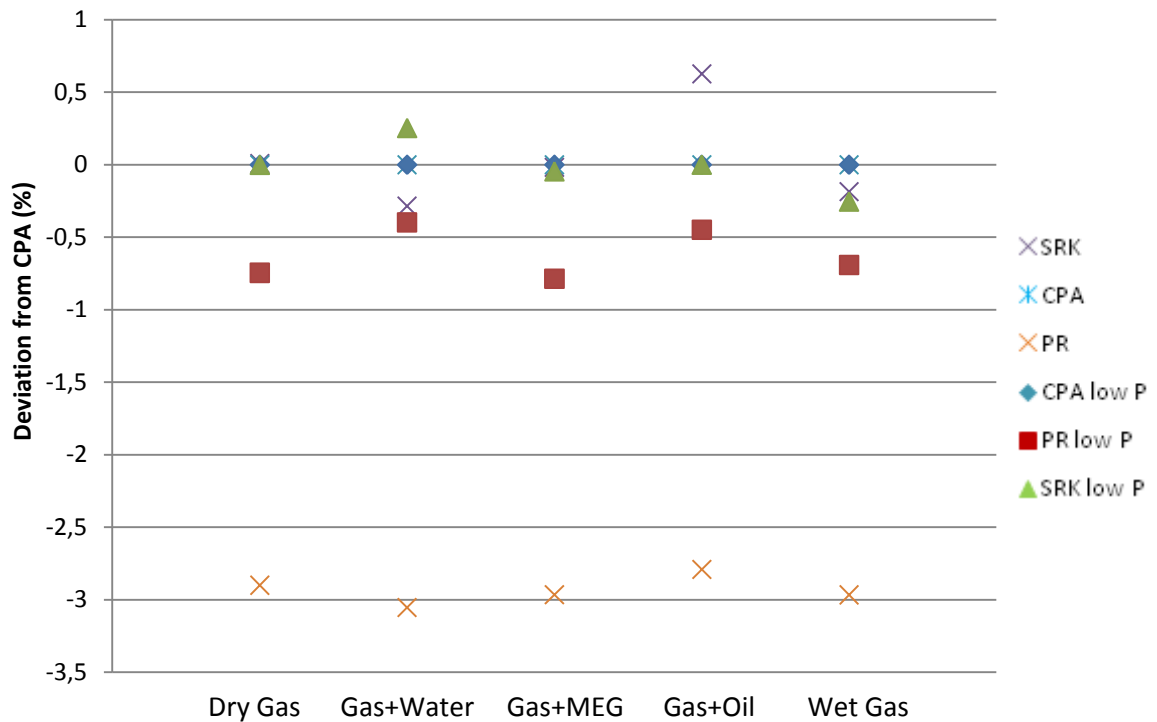


Figure 8.11 Simulation results for compressor work for the different mixtures, pressures and EoS

As the pressure is reduced the deviation of SRK and PR from CPA it is also reduced. This can be explained by the fact that the fluid is going more to the ideal conditions (that is low pressure and high temperature) when the pressure is reduced. Near to ideal conditions, the EoS are able to estimate with more precision the properties of the selected fluid, thus the deviation in the results will be lower.

It is difficult to explain the behavior of PR. The result of this deviation is due to the combined effect of deviation in enthalpy and mass estimation.

However, it is important to understand better the results given from the model in order to know how the parameters affect the final results.

When estimating the compressor work, the model calculates the enthalpy and entropy at the inlet of each step. Then, by performing a PS-flash, it calculates the enthalpy at the step outlet. The process is summarized in Figure 8.12:

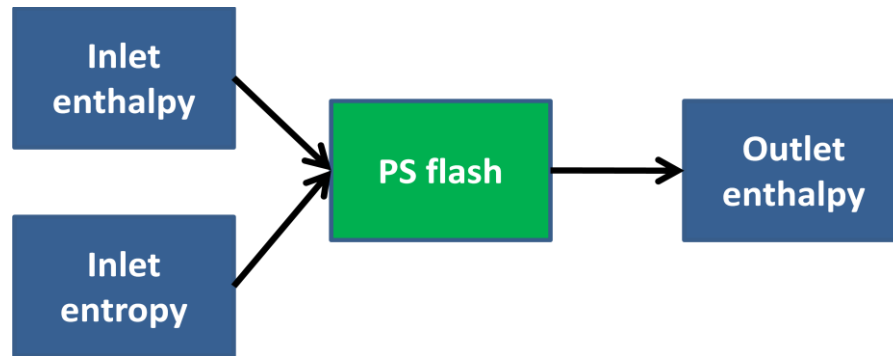


Figure 8.12 Schematic of work estimation process in the NeqSim model

As explained in Chapter 4, the only parameters that affect enthalpy and entropy calculation are the compressibility and specific heat.

Regarding the PS-flash it is important to understand how it is performed in the model.

In NeqSim both PS-flash and PH-flash are performed with an iterative process that involves a PT-flash. In the case, for example, of the PS-flash, the model will guess a temperature and then, by performing a PT-flash, will calculate the entropy. Through various iterations, the model will arrive at better temperature guess until the entropy does not match the entropy required. The same procedure will apply also to PH-flash.

It is necessary to know how a PT-flash is implemented in NeqSim.

A PT-flash involves iterative calculations, until the fugacity "f" of the element "i" of both vapor and liquid phase is not the same:

$$f_i^g = f_i^l \quad (8.10)$$

The fugacity is defined as follows:

$$f_i^g = x_i \alpha_i P \quad (8.11)$$

$$f_i^l = y_i \alpha_i P \quad (8.12)$$

Where:

" x_i " is the concentration in mole fraction of the element "i" in the liquid phase.

" y_i " is the concentration in mole fraction of the element "i" in the vapor phase.

" α_i " is the fugacity coefficient of the element "i" at the specified pressure and temperature.

"P" is the pressure.

According to this definition, Eq. 8.10 becomes:

$$y_i^g \alpha_i^g P = x_i^l \alpha_i^l P \quad (8.13)$$

The fugacity coefficients " α_i " are calculated using equations of state. High accuracy in calculating the fugacity coefficient is required in order to have good estimations of compressor work.

To summarize, in order to calculate the work required from the compressor, it is important to require that the EoS used has high accuracy in the following parameters:

- Specific heat, for estimation of enthalpy and entropy
- Compressibility factor, for estimation of enthalpy and entropy
- Fugacity coefficients, for solubility calculations and PT flashes

It is possible to conclude that the behavior shown in Figure 8.11 is due to variation of these three parameters. However, it is difficult to identify with precision which parameter has the highest influence since the behavior shown is the results of each parameter deviation.

8.2 OUTLET TEMPERATURE

For the outlet temperature, it is better to evaluate the absolute difference instead of the relative one.

There is no standard that defines what the maximum acceptable deviation for the outlet temperature is, thus it depends on the single project specifications. In Chapter 10, an evaluation of outlet temperature deviation will be done. The case of Åsgard will be considered and it will be studied how much 1,7 K of difference will impact on the sizing of the heat exchangers after the compressor. In Figure 8.13, it is possible to notice that the sensibility of the outlet temperature to the EoS used depends on the mixture composition.

The introduction of water and MEG enhance the sensitivity.

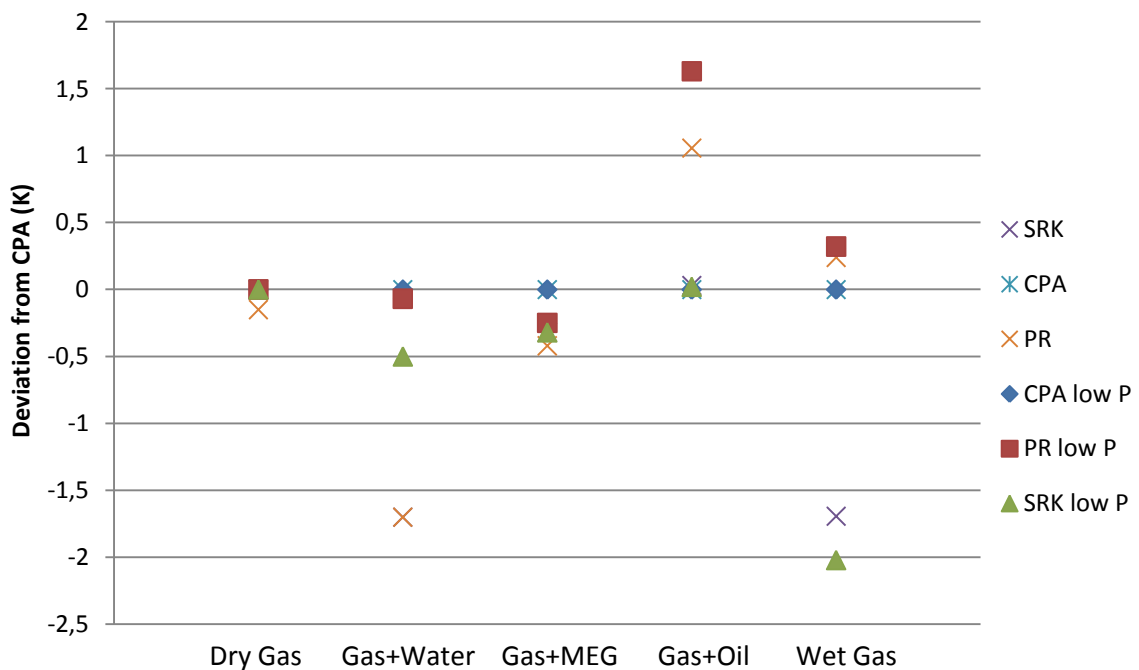


Figure 8.13 Simulation results for compressor outlet temperature for the different mixtures, pressures and EoS

For the estimation of Outlet Temperature from the compressor, it is not possible to identify a trend when the operating pressures are changing. The results do not follow the

rules of "lower pressure better performance". This may be caused by the "compensation" phenomena of parameters deviation.

As it has been done for the compressor work, it is important to understand in details what parameters play an important role in the estimation of the outlet temperature.

The first part of the model is the same as for the compressor work estimation. By calculating the inlet enthalpy and entropy, the model performs a PS-flash in order to calculate the enthalpy at the outlet of the step. Lastly, the model performs a PH-flash in order to calculate the outlet temperature of the step.

The process can be summarized in the following graph:

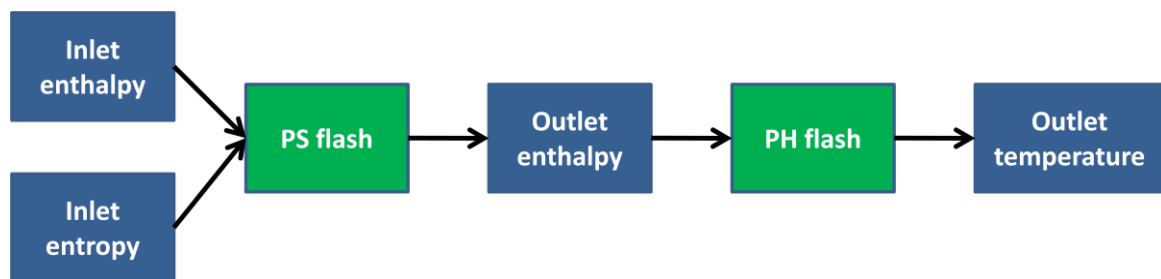


Figure 8.14 Schematic of compressor outlet temperature estimation process in the NeqSim model

As explained before, PH-flash and PS-flash are performed in the same way in NeqSim. Thus the same comments of the work estimation apply also to the outlet temperature estimation.

To summarize, in order to calculate the outlet temperature of the compressor, it is important to require that the EoS used has high accuracy in the following parameters:

- Specific heat, for estimation of enthalpy and entropy
- Compressibility factor, for estimation of enthalpy and entropy
- Fugacity coefficients, for solubility calculations and PT flashes

Since in the outlet temperature estimation process two flashes are performed, the accuracy in estimating the fugacity coefficient has more importance than the case of compressor work estimation, where only the PS-flash were performed.

It is possible to conclude that the behavior shown in Figure 8.13 is due from variation of these three parameters. However, it is difficult to identify with precision which parameter has the highest influence since the behavior shown is the results of each parameter deviation.

8.3 POLYTROPIC EFFICIENCY

As already explained in Chapter 5, the model used in this work estimates the polytropic efficiency in the same way as it estimates the compressor outlet temperature.

By calculating the inlet enthalpy and entropy, the model performs a PS-flash in order to calculate the enthalpy at the outlet of the step. Lastly, the model performs a PH-flash in order to calculate the outlet temperature of the step. Then, the model performs the Newton Method (see Chapter 5) in order to find the right polytropic efficiency until the calculated outlet temperature match the specified outlet temperature.

The process can be summarized in the following graph:

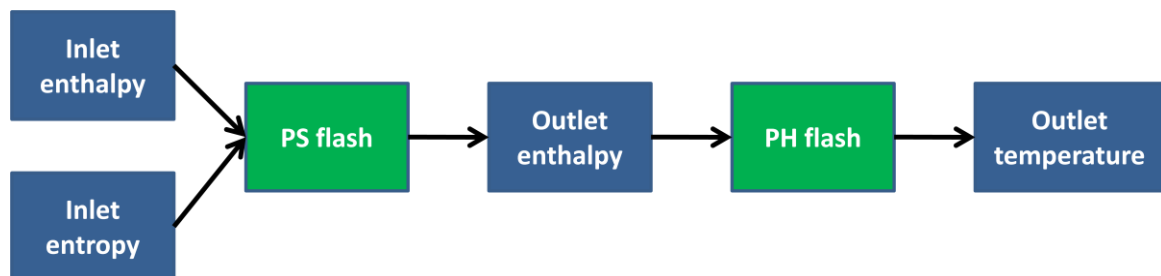


Figure 8.15 Schematic of compressor polytropic efficiency estimation process in the NeqSim model

As explained before for the outlet temperature, PH-flash and PS-flash are performed in the same way in NeqSim. Thus the same comments of the compressor work estimation apply also to the polytropic efficiency estimation.

To summarize, in order to calculate the polytropic efficiency of the compressor, it is important that the EoS used has high accuracy for the following parameters:

- Specific heat, for estimation of enthalpy and entropy
- Compressibility factor, for estimation of enthalpy and entropy
- Fugacity coefficients, for solubility calculations and PT flashes

It is important to underline that also in order to obtain the correct polytropic efficiency, NeqSim implements the Newton Method (explained in Chapter 5), thus the convergence of the model will depend also on how Newton Method is implemented this method, i.e. what is the first guess, how the derivative of the function is defined etc..

The result of the simulations are reported in the Figure 8.16 and it is important to underline that the results for this case are in absolute percentage points since the polytropic efficiency is already expressed in percentage (e.g. the deviation in absolute percentage points between 48% and 50% is 2%):

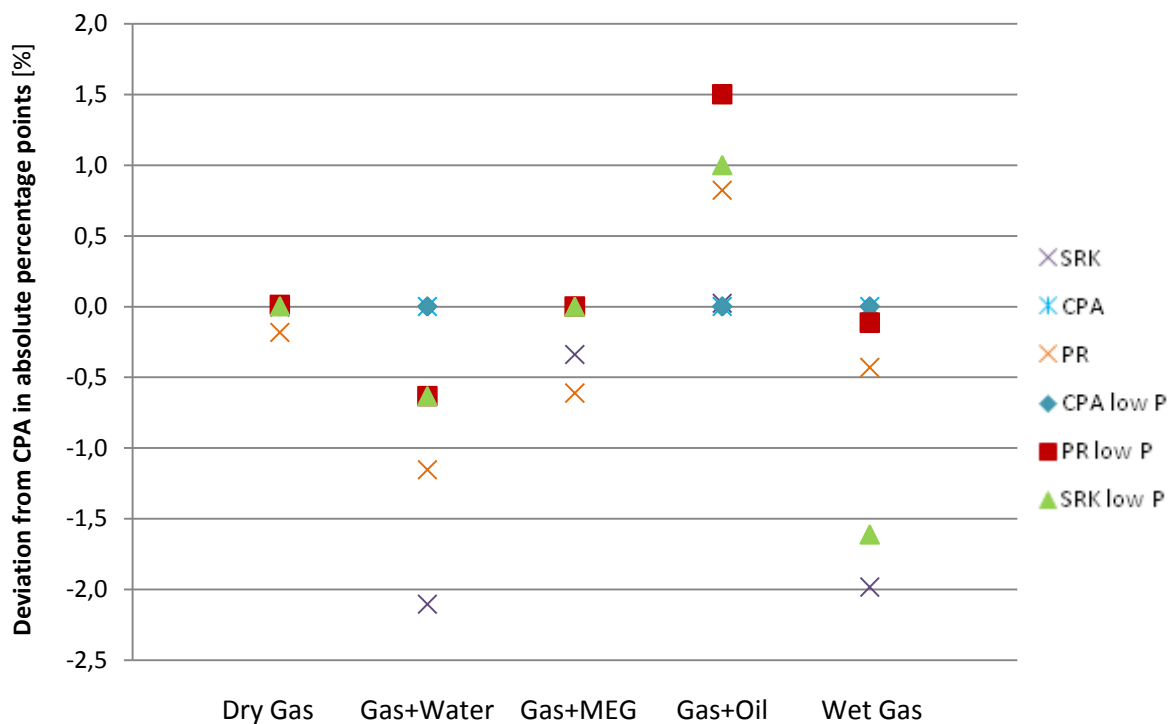


Figure 8.16 Simulation results for compressor polytropic efficiency for the different mixtures, pressures and EoS

As it can be seen, the deviation of the polytropic efficiency is dependent on the equation of state used and the fluid composition. However, according to the standard API 617, a deviation of 2% in polytropic efficiency estimation is considered acceptable. As stated by Bakken [4], some applications may have lower tolerance on polytropic efficiency deviation. Furthermore, too high deviation may bring to improper compressor design.

The results for polytropic efficiency have a similar behavior to the outlet temperature. The reason is that they are obtained in the same way in the NeqSim model.

8.4 DENSITY

Density, thus compressibility factor, seems to be sensible to the EoS selected and to the composition. In Eq. 4.15 and 4.16, enthalpy and entropy calculation depend only on the specific heat and compressibility factor. It is possible to conclude then that the contribution of the compressibility factor to enthalpy and entropy is affected by the EoS selected and fluid composition. Especially it seems that the introduction of water and MEG enhance the deviation from CPA.

The following graph shows the results of the simulation for the estimation of fluids density for different EoS at medium and low pressure:

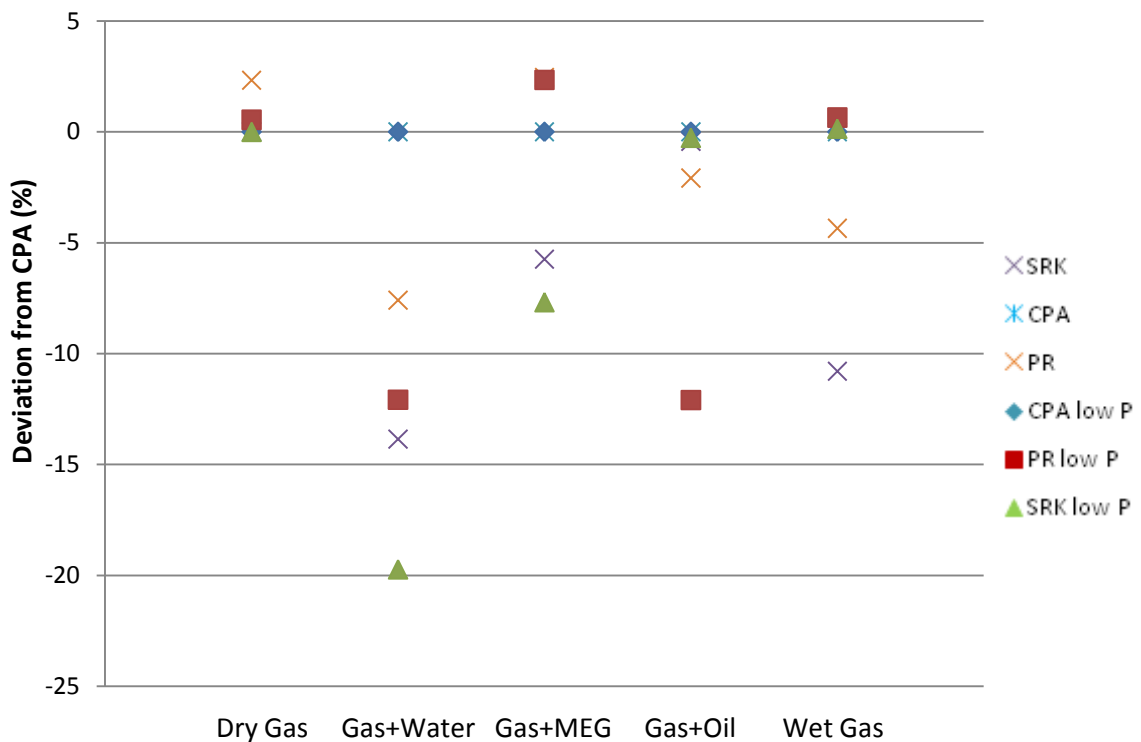


Figure 8.17 Simulation results for wet gas density for the different mixtures, pressures and EoS

As it can be seen, it is difficult to define a trend and results do not follow the rules of "lower pressure better performance". This is due to the "compensation" phenomena. It is evident that density (thus compressibility factor) is heavily affected by the equation of state used. In the case of "dry gas + water" the deviation can be up to 20%.

8.5 SPECIFIC HEAT

Specific heat does not have considerable sensitivity to the EoS selected and to the composition. In Eq. 4.15 and 4.16 it is shown that enthalpy and entropy depend only on specific heat and compressibility factor. It is possible to conclude then that the contribution of specific heat to enthalpy and entropy is not affected by the EoS selected and fluid composition in this case. It seems that the introduction of oil slightly enhance the deviation from CPA.

The following graph shows the results of the simulation for the estimation of fluids specific heat for different EoS at medium and low pressure:

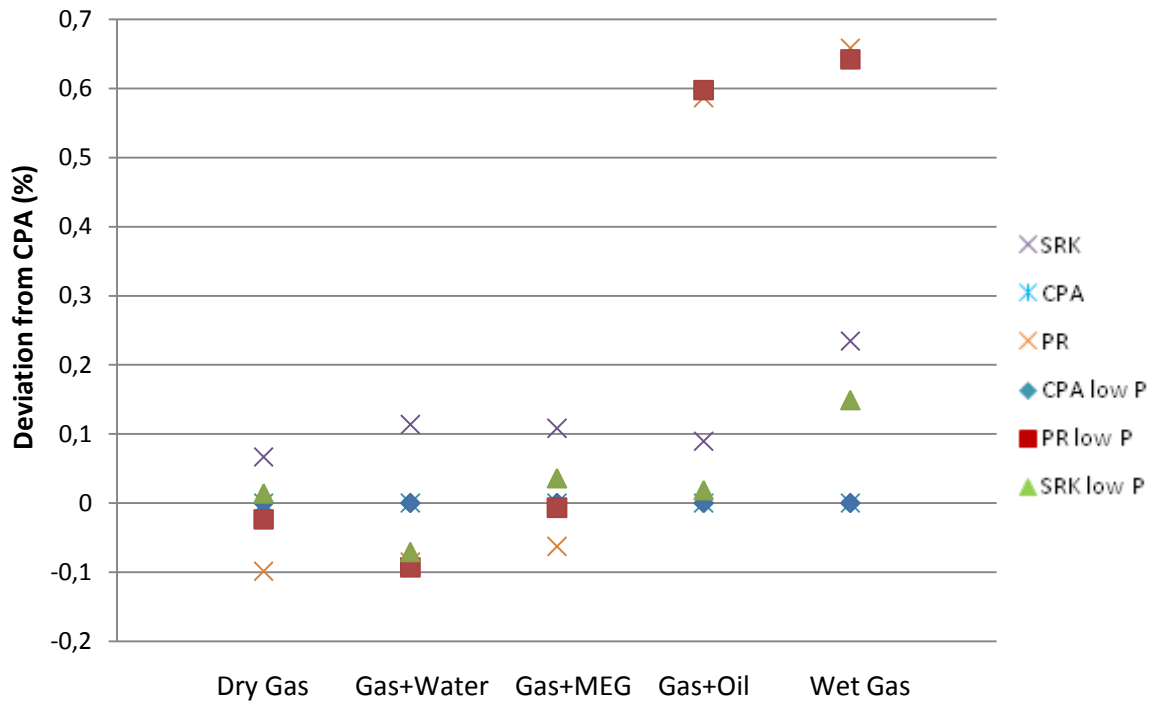


Figure 8.18 Simulation results for wet gas isobaric specific heat for the different mixtures, pressures and EoS

8.6 SPEED OF SOUND

Regarding the speed of sound, there is no considerable sensitivity to the selected EoS and to the composition apart for PR. PR shows a constant deviation of around 2% from CPA. Furthermore, the behavior is the same of the compressor work.

The following graph shows the results of the simulation for the estimation of fluids specific heat for different EoS at medium and low pressure:

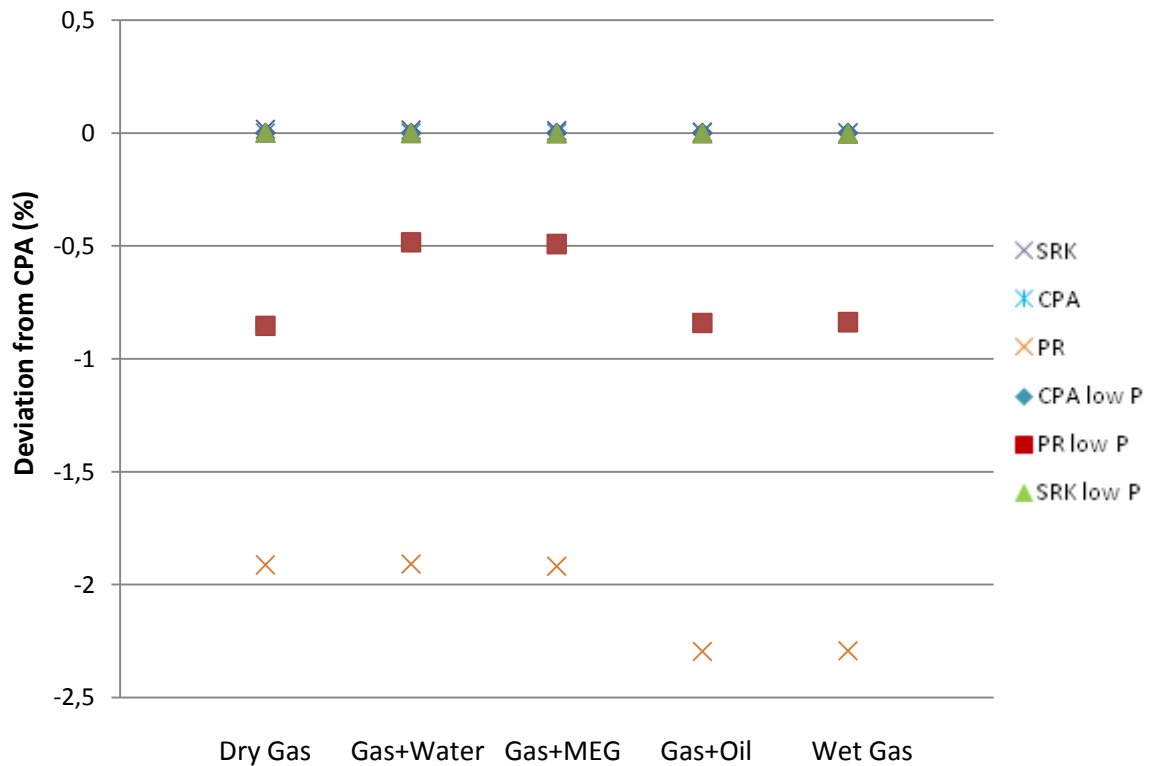


Figure 8.19 Simulation results for gas phase speed of sound for the different mixtures, pressures and EoS

It is important to notice that PR has a constant deviation from CPA and SRK (around 2%). This recalls the behavior of PR in the work estimation, where PR had a constant deviation from SRK and CPA of about 3%. Since the speed of sound in this case is calculated only for the gas phase, it is possible to make some conclusions. The gas phase brings a constant deviation in the work since it is common to all the mixtures. It is difficult to explain a constant deviation in other ways, no matter the composition. The only explanation is then that PR has a deviation of 3% in estimating the compressor work

for the gas phase. Apparently the gas phase is the only phase that has influence on final compressor work.

After analysis on the speed of sound equation, it can be concluded that deviations in specific heat does not bring any difference in estimating the speed of sound. On the other hand, density estimation has a great influence on the final value of speed of sound.

Thus the 3% deviation of PR from CPA in compressor work estimation and 2% in speed of sound estimation are due to the difference in density estimation for the gas phase. Furthermore, SRK has similar results to CPA for compressor work and speed of sound. This behavior is explained by the fact that, for the gas phase property estimation, CPA and SRK have the same results, since CPA is built upon SRK. The gas phase has a great influence in the wet gas calculations.

8.7 GAS MASS FRACTION

For the calculation of the mixture's Gas Mass Fraction (GMF), there is not considerable sensitivity of the EoS selected and of the composition apart for PR. PR have a larger deviation when calculating the GMF for a mixture with an oil phase.

The following graph shows the results of the simulation for the estimation of fluids GMF for different EoS at medium and low pressure:

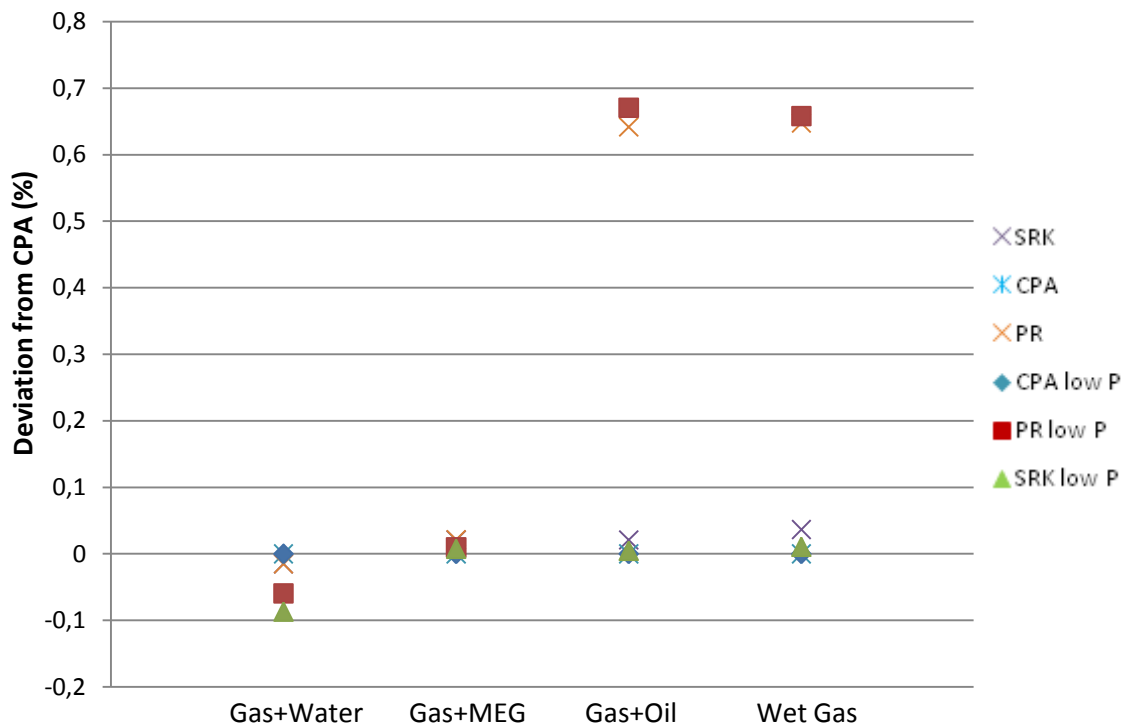


Figure 8.20 Simulation results for gas mass fraction for the different mixtures, pressures and EoS

The behavior seems not affected by the operating pressure.

Even if it seems that the deviation in estimating the GMF for SRK and PR compared to CPA is low, the analysis done for the compressor work has shown that the work is extremely sensible to small changes in estimated mass, especially for the oil phase and the evaporated mass.

8.8 NUMBER OF DIA STEPS SENSITIVITY

As explained in the Chapter 6, a sensitivity analysis on the number of steps used in the Direct Integration approach has been implemented in this work. The aim of this analysis was to determine how much the number of DIA steps affects the final results of the model.

A number of steps considered acceptable is around 40. All the results reported in this work are indeed obtained with 40 DIA steps. However, it is interesting to see what happens with higher and lower number of steps. In this analysis, it has been chosen to compare the results with 4, 40 and 400 DIA steps.

8.8.1 STEPS SENSITIVITY ON COMPRESSOR WORK ESTIMATION

According to the results, there is no difference (from 0.05 to 0.5%) between 40 and 400 DIA steps when estimating compressor work. This is valid for both the medium and low pressure case in this work, for all the mixtures. The only exceptions are:

- Medium pressure case, dry gas mixture and CPA EoS. The difference between 40 steps and 400 steps is 54% for the estimation of compressor work that is unacceptable.
- Low pressure case, gas + MEG mixture and PR EoS. The difference between 40 and 400 steps is out of scale (around 300%).

There are two possible explanation of this behavior:

- The "accumulation of error". Indeed by increasing the number of steps, the error in the model and in the EoS is amplified.
- A bug in the model. After many iterations the model has an improper implementation of the Newton Method.

The difference in estimating compressor work between 40 and 4 DIA steps are also negligible since they are around 0.5% for all the cases, mixtures and EoS. The only exception is for the medium pressure case, gas and oil mixture using SRK EoS, where the deviation between 4 and 40 steps is 1.3%. According to API 617 this is not a considerable difference.

A possible explanation of this can be that 4 DIA steps are a too rough approximation of the fluid properties changes during compression. This seems to have a non-negligible impact when dealing with mixtures of gas and oil and when using SRK EoS.

8.8.2 STEP SENSITIVITY ON COMPRESSOR OUTLET TEMPERATURE ESTIMATION

When estimating compressor outlet temperature, the results show that there is no difference between 40 and 400 DIA steps for all mixtures, all cases and EoS apart of:

- Medium pressure, gas + water, PR EoS. The deviation between 40 and 400 DIA steps estimated outlet temperature is 1 K. This cannot be considered negligible.
- Medium Pressure, wet gas, CPA EoS. The deviation between 40 and 400 DIA steps estimated outlet temperature is 53 K. This shows a considerable problem in this case when increasing the DIA steps as already confirmed for the work estimation.
- Low pressure case, gas + MEG, PR EoS. The deviation between 40 and 400 DIA steps estimated outlet temperature is 151 K. This shows a considerable problem in this case when increasing the DIA steps as already confirmed for the work estimation.

The same considerations as for the work estimation apply in this case. Either the increasing of DIA steps may cause the accumulation of errors inside the model and the EoS, resulting in a considerable difference for a lower number of DIA steps or a bug in the model results in high deviations.

The difference between 4 and 40 DIA steps for the outlet temperature are all around 0.3 to 0.5 K. As will be explained in details in Chapter 9, it is difficult to say if 0.5 K is a considerable difference. Since the mass of gas that flows into the compressor is high, 0.5 K of difference may be a considerable difference thus it cannot be neglected.

This difference is particularly high for the case of "dry gas + water" mixture at low pressure. In this case the deviation between 4 and 40 DIA steps is around 1 K for PR and SRK EoS. Since with CPA EoS this difference is reduced to 0.3%, this is a further confirmation that CPA is able to make a better estimation of mixture properties when water is involved.

The difference between 4 and 40 DIA steps case is indeed due to a worst approximation of fluid properties changes during compression when the number of step is lower.

8.8.3 STEP SENSITIVITY ON POLYTROPIC EFFICIENCY ESTIMATION

When estimating the compressor polytropic efficiency, the results shows that there is no difference between 40 and 400 DIA steps for all mixtures, all cases and EoS.

However, while the data for medium pressure are complete and reliable, it was not possible to obtain results for some cases at low pressure due to problems in the software.

These cases were:

- Dry Gas + Water
- Dry Gas + Oil
- Dry Gas + MEG

In order to make conclusions on polytropic efficiency estimation sensitivity of the equation of state used (section 8.3), a manual iteration process has been implemented in order to obtain the polytropic efficiency estimation for the 40 DIA steps case. However, since the manual iteration has higher uncertainty in the final results, it is not reliable to evaluate the direct integration approach steps sensitivity.

To summarize, no step sensitivity has been implemented for the low pressure case.

In the medium pressure case, the difference between 4 and 40 DIA steps for the polytropic efficiency are larger and may be up to 1 absolute percentage point. However API 617 defines all deviation in polytropic efficiency lower than 2% as acceptable.

This difference is particularly high for the case of "dry gas + water" mixture at medium pressure. In this case, the deviation between 4 and 40 DIA steps is around 1 absolute percentage point for PR and SRK EoS. Since with CPA EoS this difference is slightly lower, this is a further confirmation that CPA is able to make a better estimation of mixture properties when water is involved.

The difference between 4 and 40 DIA steps case is indeed due by a worst approximation of fluid properties changes during compression when the number of step is lower.

8.8.4 STEP SENSITIVITY CONCLUSIONS

From the results, it is clear that not necessarily more DIA steps will give better results. Often the results do not change when the DIA steps are increased. What changes instead is the time required for the simulation. Indeed with 40 DIA the time required to run the model is around 1 minute on average, while with 400 steps can be up to 10-15 minutes. Furthermore, by increasing the number of DIA steps, some problems may occur in the model as showed before. In the cases where it has been detected a big deviation between 40 and 400 DIA steps, apart from the accumulation of error, also a strange behavior of the model's code may bring more errors when dealing with such high number of steps.

It has been shown also that a too low number of DIA steps results in a non-negligible difference when estimating compressor outlet temperature. This, as already explained, is due to a worse approximation of the fluid properties changes during compression when the number of step is lower.

8.9 RESULTS CONCLUSION

The most interesting result is that when estimating the compressor work, the gas phase seems to be the only one that has influence on the final compressor work estimation. This however has to be verified with other analysis. From the results in this work, PR has a constant deviation from CPA and SRK no matter what the composition is. A possible explanation may be that the deviation in density (thus compressibility) estimation of PR for the gas phase brings a 3% deviation in work estimation. Since the gas phase has the biggest influence in the overall work estimation, this error remains constant even if water MEG and oil are added.

This is confirmed also by the fact that SRK and CPA have similar results for work estimation. This should not happen when water, MEG or oil are added to the fluid since SRK has lower accuracy in estimating the properties of this components. However the results of work estimation for SRK and CPA remains similar (with the highest deviation of 0.5% for oil and water case) no matter what is the composition. SRK and CPA gives exactly the same results for the dry gas case since CPA reduces to SRK for dry gas, thus again their estimation of compressor work is only affected by their ability to estimate the dry gas phase work required. Only for a small percentage, the result is affected by the introduction of water and oil.

This is not true instead for the estimation of outlet temperature and polytropic efficiency. The reason can be that both are property independent from the mass, thus it does not matter anymore that the gas phase has the highest mass share. In the outlet temperature and polytropic efficiency it has been already explained an increased importance for PT-flash thus of solubility estimation.

Unfortunately, the simulation shows that the deviation in work, outlet temperature and polytropic efficiency cannot be explained with the calculated EoS deviation in estimating the density and specific heat. More test points are needed to identify a trend and to state a clear correlation. However, even with more data it may be difficult to link deviation in density and specific heat to work, outlet temperature and polytropic efficiency since the deviations contribute together (and can compensate each other) to the final results.

However the relation between solubility and enthalpy calculation to the compressor work estimation is more clear. It seems that only some solubilities play an important role in this process while others can be neglected.

According to this analysis, it is not possible to describe in detail the role of each property in wet gas calculations, however it is possible to identify what group of properties are more important and what can be neglected. This analysis will help to understand in what properties an EoS needs to have higher accuracy when dealing with wet gas. This can be helpful both for a possible EoS tuning and even for new EoS developing.

9. WHICH EOS IS MORE SUITABLE FOR WET GAS?

The conclusion of the preliminary project of this thesis is that CPA is the only equation of state able to handle properly water and MEG thus was assumed a good selection when evaluating processes involving wet gas. [3]

However, according to the outcomes of this thesis, it is possible to implement a more accurate evaluation.

After all the considerations done in the previous chapter, the most important properties for wet gas calculations are the following:

- Specific heat
- Compressibility factor
- Fugacity coefficients
- Solubility:
 - Solubility of gas in oil (P=44 bar and T=298 K)
 - Solubility of oil in gas (P=44 bar and T=298 K)
 - Solubility of water and MEG in gas (P=117 bar and T=370 K)

Thus, from the data collected in the preliminary project of this thesis, it will be possible to see what EoS have the highest accuracy in these properties.

9.1 SPECIFIC HEAT

In the preliminary project, it has been evaluated the accuracy of various EoS (included PR, CPA and SRK) in calculating the isobaric specific heat of a natural gas mixture with the following composition: [23]

Table 9.1 Gas composition for specific heat experimental data

Component	Molar fraction
Methane	0,89569
Nitrogen	4,96E-03
Ethane	8,35E-02
Propane	1,20E-02
Isobutane	1,49E-03
n-Butane	2,26E-03
2-Methyl-Butane	1,50E-04

This composition is similar to the composition of the gas phase of the fluid studied in this thesis.

The operating conditions at which the calculations have been made are:

- Pressure=150 bar
- Temperature: from 30 to 140 °C

No data has been calculated for lower pressures.

The following graph shows the deviation from experimental data of the calculated isobaric specific heat of CPA, SRK and PR:

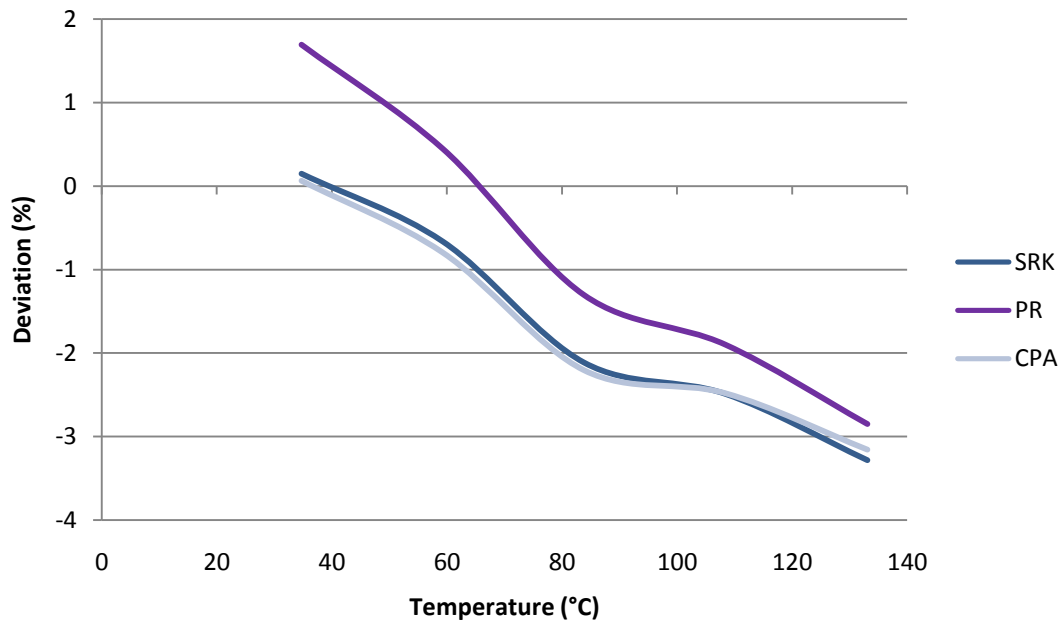


Figure 9.1 Isobaric specific heat deviation of EoS from experimental data at P=150 bar

SRK and CPA have the same behavior since CPA is built upon SRK.

In order to make some conclusions, it is important to repeat that the operating conditions of the compressor studied in this work are:

1. For the medium pressure case:
 - $P_{inlet} = 44$ bar and $T_{inlet} = 298$ K
 - $P_{outlet} = 117$ bar and $T_{outlet} = 370$ K
2. For the lower pressure case:
 - $P_{inlet} = 10$ bar and $T_{inlet} = 298$ K
 - $P_{outlet} = 26.6$ bar and $T_{outlet} = 360$ K

Since the minimum pressure at which the evaluation has been done is 150 bar, it is possible to evaluate only the performance at a condition similar to the compressor outlet of the medium pressure case.

At this point there is no significant difference between CPA, SRK and PR, however it seems that PR, in the range of temperature of interest ($T=298$ K to 370 K) has a slightly

better performance. At these conditions the isobaric specific heat was 2.94 kJ/K-kg according to the experimental data [23] and PR estimate an isobaric specific heat of 2.94 kJ/K-kg, thus a deviation of 1.3%.

The following graph shows the deviation of calculated Cp for pure water by SRK, PR and CPA EoS from the experimental data: [24]

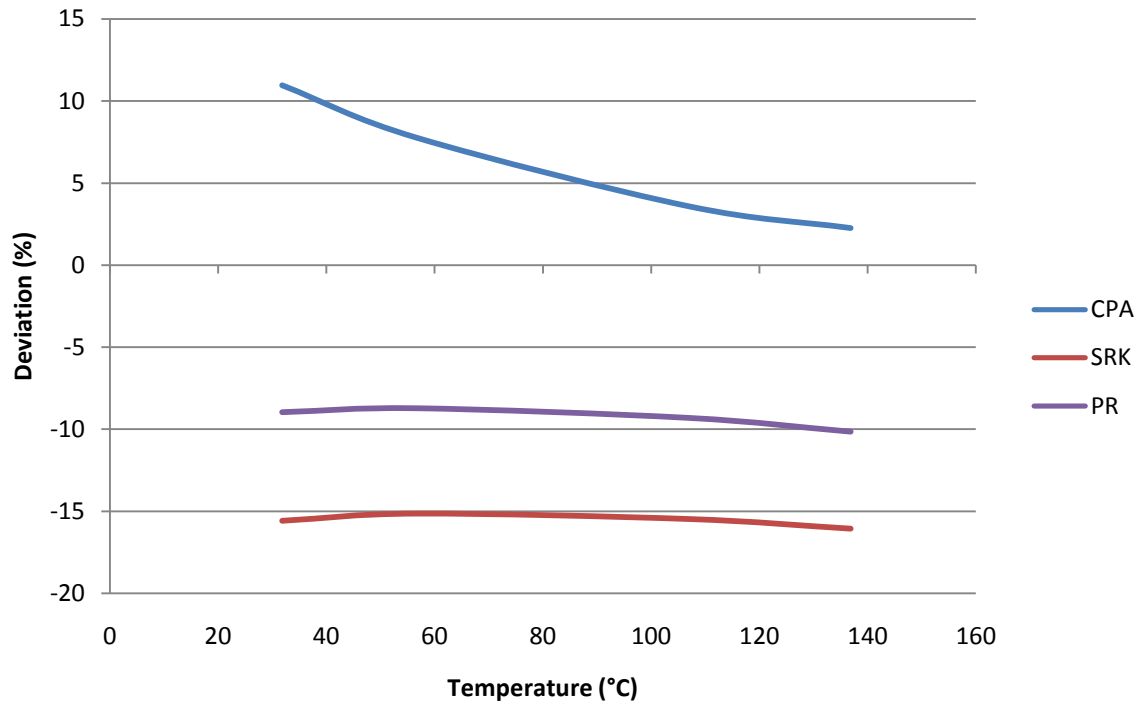


Figure 9.2 Water isobaric specific heat deviation of EoS from experimental data at P=150 bar

It is clear that, as expected, CPA has a higher accuracy when estimating the specific heat of water, in particular at conditions similar to the ones at the compressor outlet (P=117 bar and T=370 K). Water isobaric specific heat from experimental data at P=150 bar and 380 K was estimated as 4.19 kJ/K-kg while CPA estimate an isobaric specific heat of 4.56 kJ/K-kg that is 4% of deviation from the experimental one.

9.2 DENSITY

The density is strictly correlated to the compressibility factor from the following formula:

$$Z = \frac{p}{\rho_h R' T} \quad (9.1)$$

Where “Z” is the compressibility factor and “ ρ_h ” is the density of the homogeneous fluid, according to Eq. 4.12.

It is then possible to conclude that an EoS is able to properly estimate the density of a fluid at certain conditions because it is able to estimate a correct value for the compressibility factor.

In the preliminary project, it was evaluated the accuracy of various EoS (included PR, CPA and SRK) in calculating the density of a natural gas mixture with the following composition: [25]

Table 9.2 Gas mixture composition for experimental density data

Component	Mole Fraction
CH4	0,90991
C2H6	0,02949
C3H8	0,01513
i-C4H10	0,00755
n-C4H10	0,00755
i-C5H12	0,00299
CO2	0,00403
N2	0,02031
n-C5H12	0,304

This composition is similar to the gas phase of the fluid studied in this thesis.

The following graph shows the deviation from experimental data of the calculated Density of CPA, SRK and PR: [25]

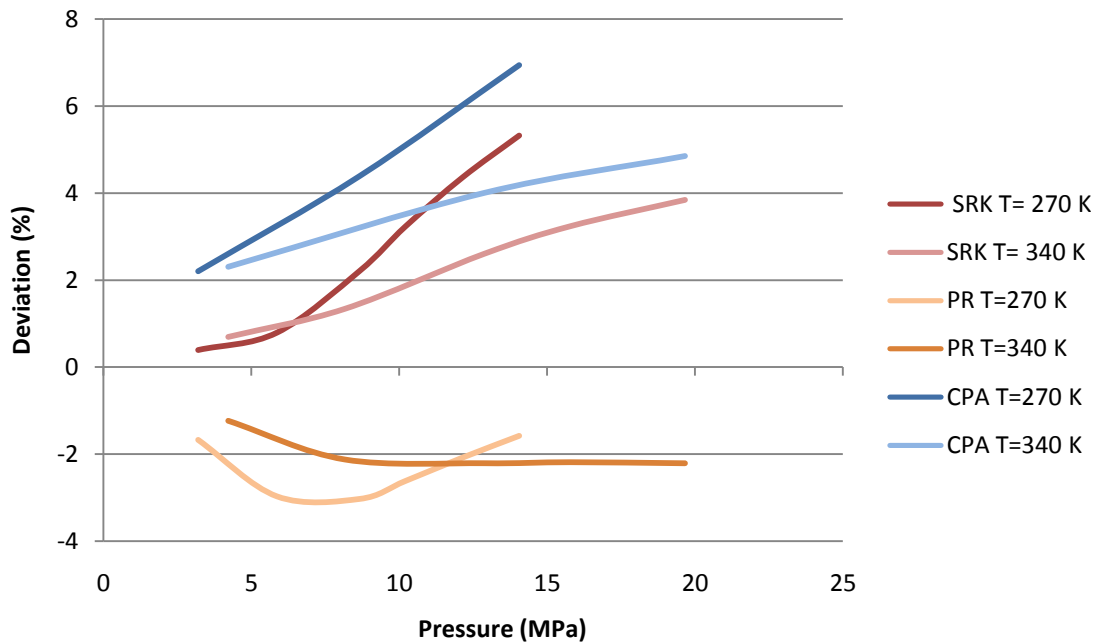


Figure 9.3 Density deviation of EoS from experimental data for various temperatures and pressures

As it can be seen, the evaluation has been made at conditions similar to the operating conditions of this work for the medium pressure case:

- $P_{inlet} = 44 \text{ bar}$ and $T_{inlet} = 298 \text{ K}$
- $P_{outlet} = 117 \text{ bar}$ and $T_{outlet} = 370 \text{ K}$

CPA and SRK were expected to behave the same since CPA is built upon SRK. However, since SRK has been implemented in HYSYS, and CPA in NeqSim, there could be some variations. Generally, it is possible to say that at the compressor inlet, SRK has a higher accuracy when evaluating the gas phase density (and so the compressibility factor).

At conditions similar to the compressor inlet the fluid has a density of 58.1 kg/m^3 according to experimental data while SRK estimated 57.5 kg/m^3 which is 1% of deviation. Thus SRK has a higher accuracy at compressor inlet conditions.

PR has a higher accuracy at the compressor outlet. At conditions similar to the compressor outlet the fluid has a density of 94.6 kg/m^3 while PR estimates a density of 96.7 kg/m^3 , which is a 2.2% of deviation.

By comparing this data with the data for isobaric specific heat, it is possible to conclude that PR will have a higher accuracy when evaluating the isobaric specific heat and the compressibility factor at the compressor outlet, thus a better estimation of the outlet enthalpy of the gas phase. No conclusions can be made about the compressor inlet, since no data are available for the isobaric specific heat accuracy for the EoS studied at the compressor inlet condition.

The following graph shows the deviation of calculated density of pure water by SRK, PR and CPA EoS from the experimental data: [24]

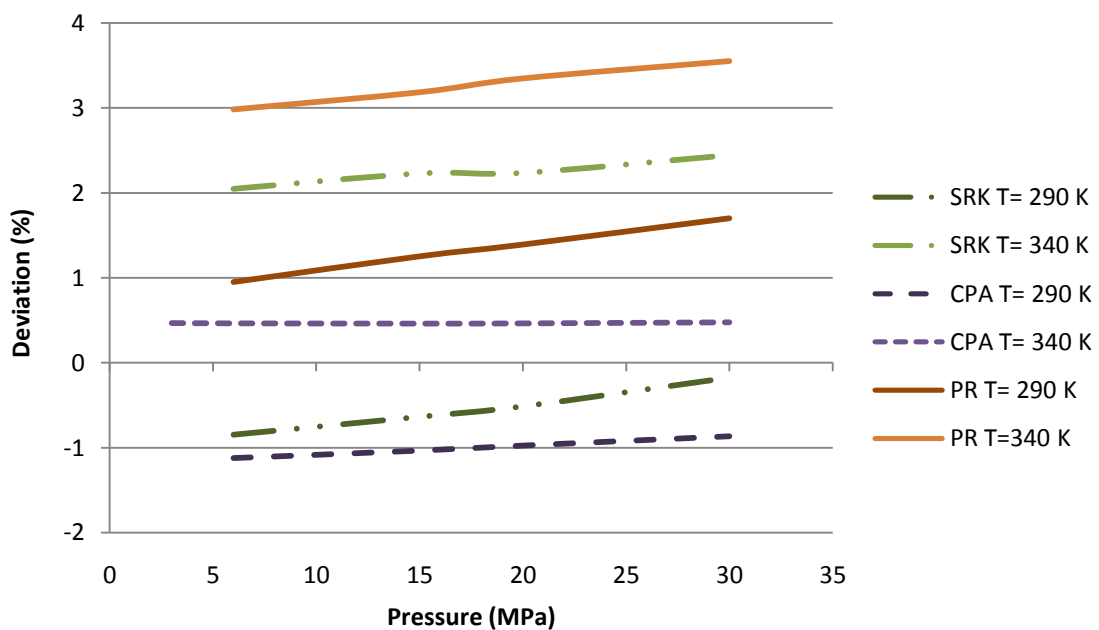


Figure 9.4 Water density deviation of EoS from experimental data for various temperatures and pressures

As it can be seen, CPA has the highest accuracy at conditions similar to the compressor outlet ($P=117$ bar and $T=370$ K). Water density from experimental data at $P=150$ bar and $T=340$ K was estimated as 986 kg/m³, while CPA estimated a water density of 981.1 kg/m³, that is 0.46% . Regarding the compressor inlet conditions ($P=44$ bar and $T=298$ K), SRK and CPA have a similar deviation in predicting water density. Water density from experimental data at $P=60$ bar and $T=290$ K is estimated as 1002 kg/m³ while SRK and CPA estimate a water density of 1010 kg/m³ and 1012.7 kg/m³ respectively that correspond to a deviation of 0.85% and 1.12% .

9.3 SOLUBILITY OF GAS IN OIL AND OIL IN GAS

Unfortunately, no data has been collected for solubility of gas in oil. However the ability of each EoS to predict the solubility of oil in gas has been studied. The results for the solubility of n-Decane in Methane for an n-Decane-Methane mixture are reported in the following graph: [26]

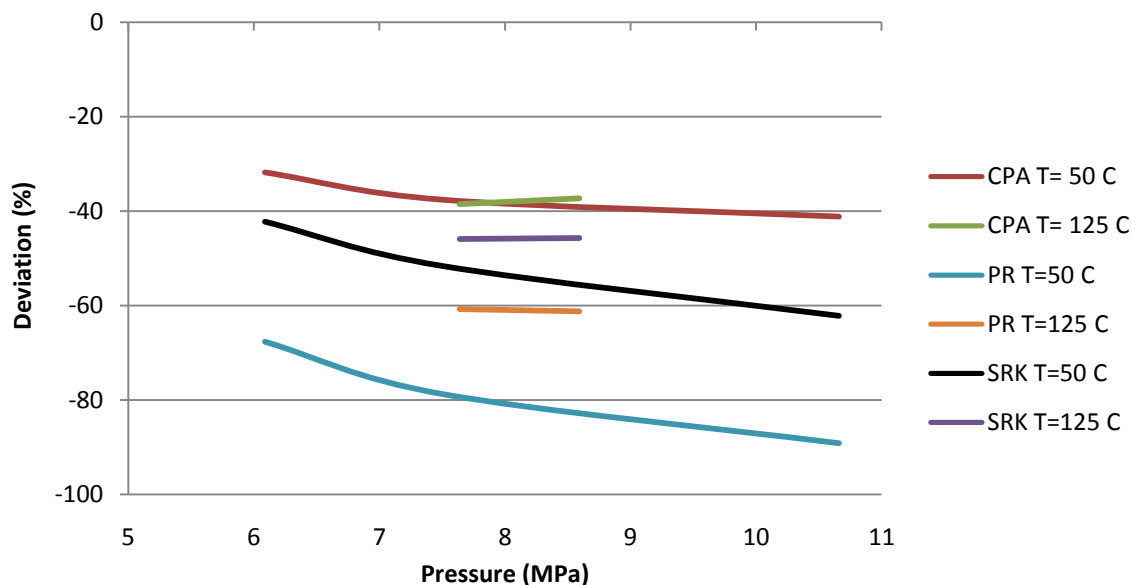


Figure 9.5 Solubility of n-Decane in Methane deviation of EoS from experimental data for various pressures and temperatures

As it can be seen, CPA is the most accurate in predicting the amount of n-Decane that dissolves into the gas phase at the compressor inlet and outlet conditions.

For example, at 50 bar and 50 C, the amount of n-Decane in Methane for an n-Decane-Methane mixture is 0.00056 (in mole fraction) while CPA estimates a mole fraction of 0.00074. The analysis done for the work estimation shows that even small molar fractions can affect considerably the results of work estimation.

Other cases have been studied (2,2,5-Trimethylhexane,tert-Butylbenzene, n-Dodecane) but unfortunately not for CPA EoS since the components were not available in NeqSim. However the results for these cases show that SRK is more accurate than PR at predicting the solubility of oil in the gas phase at all the pressure and temperature studied: [26]

Table 9.3 Average deviation of EoS from experimental data for solubility of heavy hydrocarbons into methane

EoS	Average deviation for 2,2,5-Trimethylhexane [%]	Average deviation for tert-Butylbenzene [%]	Average deviation for n-Dodecane [%]
SRK	-68	-61	-43
PR	-93	-92	77

9.4 SOLUBILITY OF WATER AND MEG IN THE GAS PHASE

In the preliminary project of this thesis, the accuracy of PR, SRK and CPA in evaluating the solubility of water and MEG in Methane in a Water-MEG-Methane mixture was estimated. In the following graphs, it is possible to see the results of this evaluation: [27]

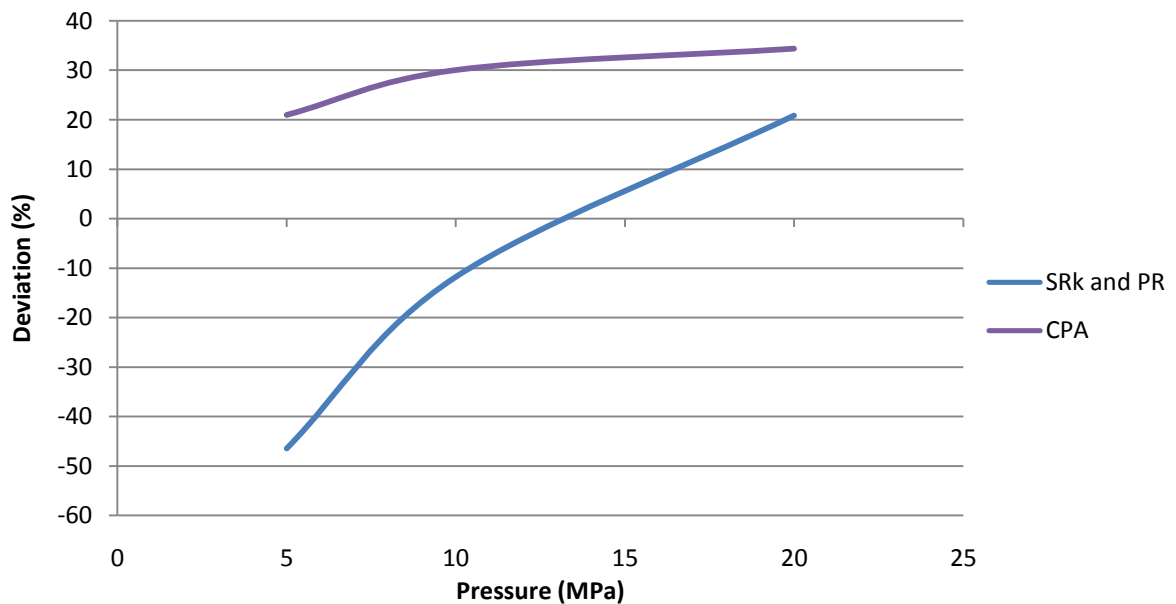


Figure 9.6 Water solubility in gas phase deviation of EoS from experimental data at T=298 K

As can be seen, when estimating the solubility of water in Methane, all the three EoS perform similar. CPA perform slightly better at the compressor inlet conditions (P=44 bar and T=298 K). At these conditions the mole fraction of water in methane is 0.00045 according to experimental data, while CPA estimates a mole fraction of water in methane of 0.00035, which is a deviation of 21%. According to the analysis done for the

compressor work estimation, these quantities are not negligible and can result in around 1% of deviation in compressor work estimation.

Furthermore, according to the results, CPA is the only equation able to estimate the solubility of MEG in Methane with an error of less than 200%.

In the fluid considered in this thesis (Åsgard wet gas), according to the model developed in this work the share of water and MEG in the evaporated aqueous mass during the compression is the following:

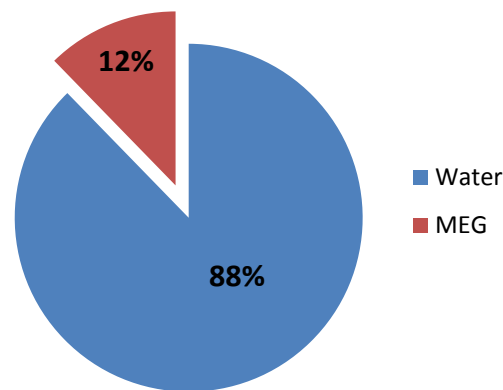


Figure 9.7 Mass share between water and MEG of the total evaporated mass from the aqueous phase

The water that dissolves into the gas phase accounts for 88% of the total evaporated aqueous phase mass. As explained before however, the MEG contribution is not negligible, since the evaporated mass required extremely high precision. In the case studied in this work, a 26% of deviation in estimated evaporated mass would result in a 2% of difference in the estimated compressor work. The 26% of the evaporated mass, in this case, was equal to 0,00025 kg.

9.5 FUGACITY COEFFICIENT

The fugacity coefficient is related to the solubility estimations. Thus if an equation has high accuracy when estimating the VLE compositions, it means that it has an accurate estimation of fugacity coefficients. It is possible to state that if an equation of state is able to predict properly the solubility in the cases described before (thus has a good estimation of the fugacity coefficients for that components) it is then able to perform accurate PT-flash calculations. As explained previously, the PT-flash is the basis upon which the other flashes calculations are built, like PH-flash and PS-flash.

From the comparison done in the preliminary project work of this thesis CPA was the most accurate when calculating the solubility of interest for the compression of wet gas:

- Solubility of gas in oil (P=44 bar and T=298 K)
- Solubility of oil in gas (P=44 bar and T=298 K)
- Solubility of water and MEG in gas (P=117 bar and T=370 K)

Thus, it can be expected that CPA would perform more accurate PT-flash calculations than PR and SRK.

9.6 CONCLUSIONS

PR seems the best EoS regarding Natural Gas with no water and no MEG. However CPA is the EoS with the highest accuracy when estimating solubilities of heavy hydrocarbons (oil) , water and MEG.

From the results, CPA can be considered the equation most suitable for wet gas. However, some considerations have to be done. First, due to the time available, was not possible to estimate the performance for the solubility of gas in oil. Thus, a complete analysis should include also these evaluations. Furthermore, CPA has proven to be more accurate but the difference in the wet gas calculations of work and polytropic efficiency shows that there is not considerable variation if CPA, SRK or PR are selected. This is because the deviations in estimating the previously described properties for wet gas can have a "compensating" effect. This means that for example PR can have a bad estimation of solubility of MEG in gas phase and water compressibility factor but the two effects can compensate each other and give a final result similar to the one of CPA.

This is the reason why even if CPA has a better performance in the properties important for wet gas compression, PR and SRK can have similar results.

10. CONCLUSION AND RECOMMENDATIONS

10.1 CONSIDERATION ABOUT DEVIATIONS

From the results obtained in this work, if the equation of state (EoS) is changed among PR, SRK and CPA, then work, efficiency and temperature may change by 1% to 3%.

The standard API 617 fortunately sets the limit value for compressor performance evaluation.

In the case of constant speed compressor, a deviation of 4% in the estimation of polytropic head is considered acceptable, as well as 7% in power. When estimating polytropic efficiency, it can have a deviation up to 2%.

However, according to Bakken et al. [4] some compressors may require a deviation within 2% (as in the case of Snøhvit LNG plant). Furthermore a deviation of 4% may bring to improper design of the compressor.

Thus there is not a specific value within the sensitivity of the performance analysis that can be considered "low", but a case-by-case consideration has to be done.

However, referring to the API 617, it can be stated that the performance analysis is not considerably affected by the choice of one of the three proposed EoS (SRK, PR and CPA).

API 617 does not specify any restriction about the outlet temperature calculation. However, in some cases, a difference of 1 or 2 Kelvin can be considerable. Since the volume of gas processed by the subsea compressor is large, an overestimation of one or two degree may lead to an over sizing of cooler heat exchangers in the compressor.

10.2 COST OF DEVIATION

Money is the only practical parameter that can give a clear idea of the real value of the deviations mentioned previously.

In the following paragraph a rough estimation of the economic value of the deviation will be done.

The parameters considered are work and outlet temperature. The first for obvious reasons: every Watt consumed by the compressor has a cost so a 3% of difference in work evaluation will represent an under or over estimation of the operating electricity consumption.

The outlet temperature instead is important for the reasons explained in Chapter 6. The prediction of a correct outlet temperature may be involved in different cases. The sizing of the intercooler heat exchangers is one of those.

10.2.1 COST OF COMPRESSOR WORK DEVIATION

The following data about the total expected energy recovery from Åsgard by using subsea gas compression have been collected from different sources:

Table 10.1 Conversion from boe to Sm³ of the total expected gas recovery from Åsgard thanks to subsea wet gas compression

Value	Unit	Source
280000000	boe	[1]
42'420	Million Sm ³	[28]
1 boe	= 151.5 Sm ³	[28]

Thus, the use of subsea compression will allow an increased recovery of an equivalent of 42'420 million of standard cubic meters of gas in 15 years of operation.

Using the model developed for this work, the following work required by the wet gas compressor (in kJ/Sm³) has been calculated for each EoS:

Table 10.2 Deviation from CPA of SRK and PR EoS in estimation of compressor work and related operation cost

	Deviation from CPA [%]	kJ/Sm³	NOK/Sm³
CPA	0	145'231	12.99
SRK	0.23	144'895	12.96
PR	3.3	140'435	12.56

Taking CPA as the reference case, with simple calculations it is possible to calculate the overall cost of compression operation during the 15 years of Åsgard subsea compression activity:

Table 10.3 Difference of SRK and PR from CPA EoS for operation cost estimation in million NOK

EoS used	Difference in operation costs for wet gas compressor in 15 years [Million NOK]
CPA	0
SRK	- 1.27
PR	- 18.2

Although this is a rough calculation it shows that in 15 years a 3% deviation in work estimation could result in 18.2 million NOK of budget underestimation.

10.2.2 COST OF OUTLET TEMPERATURE DEVIATION

Process engineers have to determine the outlet temperature from the compressor in order to size the compressor cooler (as part of the anti-surge system) and the cooler used to bring the wet gas stream at the desired temperature.

In the following figure the process scheme of a subsea wet gas compressor can be seen. [29]

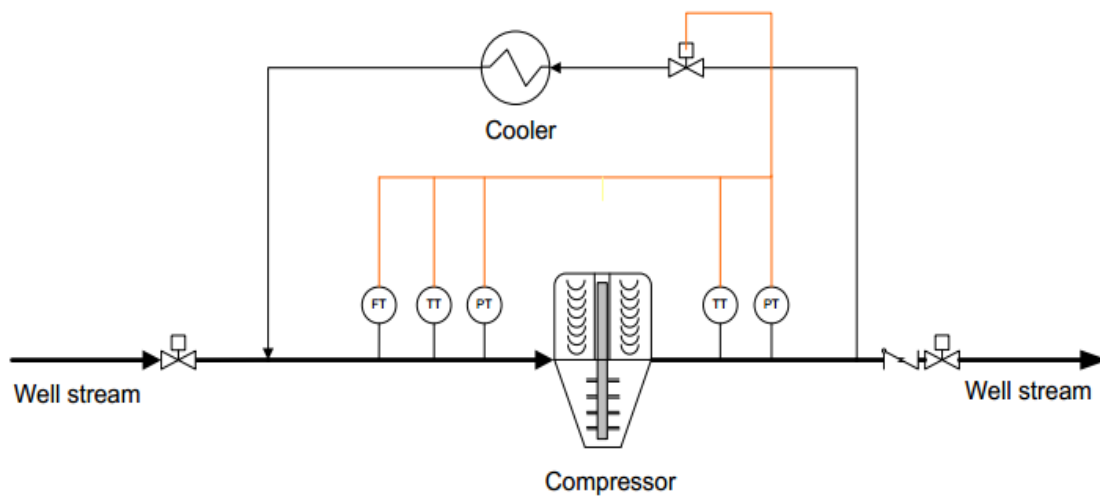


Figure 10.1 Schematic of anti-surge system in a subsea compressor [29]

In the Åsgard project, a cooler after the compressor is also planned. It is clear that in order to size properly the cooler at the compressor outlet, a precise estimation of the temperature is necessary.

When dealing with temperature is not useful to consider the percentage deviation but the absolute value in Kelvin.

According to the results from the model developed in this work, the following deviations have been calculated (CPA is again taken as reference point):

Table 10.4 Deviation of SRK and PR from CPA EoS in compressor outlet temperature estimation

EoS used	T_{out} [K]	Deviation from CPA [K]
CPA	368.7	0
SRK	367	+ 1.7
PR	368.9	- 0.24

These data is valid for the previously described compressor operation conditions and fluid composition.

As it can be seen, using SRK instead of CPA result in a difference of 1.7 K in the estimated outlet temperature. This difference will bring a difference in estimated cooling power required in the process.

It has to be underlined that this is a rough estimation since data are difficult to collect for such projects.

The case considered is again the Åsgard subsea wet gas compression project. Two compressors tested for the Åsgard project are from Dresser-Rand and Framo. During testing at the Statoil laboratory (K-Lab) an operating inlet gas flow of around 2000 Am³/h was specified. According to the process scheme in Figure 10.1 the stream going into the cooler is just part of the stream processed. It is not possible to quantify the amount of gas passing through the cooler. However in the Åsgard project there is a cooler right after the compressor in order to keep the stream at a desired temperature. In the following calculation the case of a 2000 Am³/h flow passing through the cooler was assumed. [30]

No data about the operation conditions of this test are available thus they are assumed equal to the case studied in this work (that is similar to the Åsgard expected operating conditions).

The model developed in this work has been used in order to determine density and isobaric specific heat of the compressor inlet stream.

The following table sum up the data collected for this evaluation:

Table 10.5 Parameters used for the evaluation

Volume Flow	2000	Am ³ /h
Density	148.82	kg/Am ³
<i>m</i>	82.7	kg/s
Isobaric Specific Heat	382.97	J/kg-K
$\Delta T = T_{outCPA} - T_{outSRK}$	1.7	K

Using the well-known formula below, the required cooling power " \dot{Q} " has been calculated:

$$\dot{Q} = \dot{m} * c_p * \Delta T = 54 \text{ kW} \quad (10.1)$$

Since SRK is used instead of CPA will result in over sizing of the cooler's heat exchangers by around 54 kW.

It is almost impossible to quantify how much 54 kW are in NOK for subsea applications since the price varies according to various factor such as the type of heat exchanger chosen, the temperatures of the fluid and the installation costs.

However, it it is possible to say that in all subsea applications, the size of the equipment is of vital importance since the cost and complexity of installing objects at the bottom of the sea is extremely high.

10.3 FINAL COMMENTS AND FURTHER WORK

As explained in Chapter 8, a detailed analysis on what properties require the highest precision in wet gas compression analysis has been done. Then, using the data from the preliminary project of this work, the accuracy of PR, SRK and CPA in these important parameters has been shown. PR had higher accuracy for dry gas properties estimation but lost reliability when water, MEG or oil was considered. Instead, CPA had a slightly worse performance in dry gas properties estimation but was the only equation of state that was able to handle water, MEG and oil with a reasonable accuracy. This can be considered a first, rough analysis, since more data have to be compared at different conditions in order to do proper considerations.

The other outcome of this thesis is the sensitivity of wet gas compressor performance evaluation on the equation of state used. As seen from the results, for compressor work estimation, the deviation between CPA and SRK was negligible (below 0.5%), while PR had a constant 3% of deviation, independently of the composition. This deviation was even lower if the compressor operating pressures were reduced. According to the API 617, up to 4% deviation is considered acceptable. However, this limit depends on different factors and the case where only deviations lower than 2% are considered acceptable is not rare. [4]

When estimating the compressor outlet temperature, the deviation of SRK and PR from CPA was depending on the fluid composition and operation pressure. However the maximum difference detected was 2 K for SRK in the low pressure case for the wet gas mixture.

The estimation of the polytropic efficiency showed a similar behaviour to the estimation of compressor outlet temperature since they were modelled in NeqSim in the same way. In this case the maximum deviation from CPA was 2 absolute percentage point for the medium pressure case for wet gas mixture and "dry gas + water" mixture.

Unfortunately, no proper considerations could be done about the correlation between deviation in estimating inlet properties (speed of sound, density, GMF, isobaric specific heat) and deviation in estimating outlet properties. This highlights the need for a more detailed sensitivity analysis that should go deeper into the mathematical structure of the equations of state and the model in order to eliminate the "compensation effect". This

effect arises when the errors in estimating two parameters are compensating each other and thus the final deviation is lower than the absolute sum of the separate parameter's deviation. This effect makes any comment done by looking at the final deviation of work or outlet temperature useless. Instead, a rigorous analysis on the sensitivity of the equation of state with regards to density, specific heat and solubility estimation has to be done.

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APPENDIX

A.1 MAIN CODE

```
1 package test.neqsim;
2
3 import thermo.system.*;
4 import thermodynamicOperations.*;
5 import processSimulation.*;
6 import processSimulation.processEquipment.*;
7 import processSimulation.processEquipment.stream.Stream;
8 import processSimulation.processEquipment.compressor.*;
9
10 public class TestNeqsim {
11
12     public static void main(String[] args) {
13
14         SystemInterface testFluid = new SystemPrEos(298.15, 10);
15
16         testFluid.addComponent("methane", 8.8316);
17         testFluid.addComponent("nitrogen", 0.0296682);
18         testFluid.addComponent("CO2", 0.0209667);
19         testFluid.addComponent("ethane", 0.28915);
20         testFluid.addComponent("propane", 0.116637);
21         testFluid.addComponent("i-butane", 0.0222488);
22         testFluid.addComponent("n-butane", 0.0315726);
23         testFluid.addComponent("i-pentane", 0.0097763);
24         testFluid.addComponent("n-pentane", 0.0104928);
25
26         testFluid.addTBPfraction("C7", 0.0137641, 86.18/1000.0, 0.664);
27         testFluid.addTBPfraction("C8", 0.025341, 96.46/1000.0, 0.7217);
28         testFluid.addTBPfraction("C9", 0.023069, 124.66/1000.0, 0.7604);
29         testFluid.addTBPfraction("C10", 0.0031771, 178.2/1000.0, 0.8021);
30         testFluid.addTBPfraction("C11", 0.0000188549, 263.77/1000.0, 0.8416);
31
32         testFluid.addComponent("water", 0.39944);
33         testFluid.addComponent("MEG", 0.173083);
34
35         testFluid.createDatabase(true);
36         testFluid.setMixingRule(10);
37         testFluid.setMultiPhaseCheck(true);
38         testFluid.initPhysicalProperties();
39
```



```

40     double[] temperature = newdouble[10];
41     double[] work = newdouble[10];
42     double[] Cp_Vapour = newdouble[10];
43     double[] Cp_liquid = newdouble[10];
44     double[] Cp = newdouble[10];
45     double[] Density_Vapour = newdouble[10];
46     double[] Density_liquid = newdouble[10];
47     double[] Density = newdouble[10];
48
49     Stream stream1 = newStream(testFluid);
50     Compressor compressor1 = newCompressor(stream1);
51     compressor1.setOutletPressure(26.59090909);
52     compressor1.setUsePolytropicCalc(true);
53     compressor1.setPolytropicEfficiency(0.8);
54     Stream stream2 = newStream(compressor1.getOutputStream());
55
56     processSimulation.processSystem.ProcessSystem operations =
newprocessSimulation.processSystem.ProcessSystem();
57     operations.add(stream1);
58     operations.add(compressor1);
59     operations.add(stream2);
60     operations.run();
61
62     doubleCpVapour = stream1.getThermoSystem().getPhase(0).getCp();
63     doubleCpliquid = stream1.getThermoSystem().getPhase(2).getCp();
64     doubleCpOil = stream1.getThermoSystem().getPhase(1).getCp();
65     doubleCpone = stream1.getThermoSystem().getPhase(0).getBeta() * CpVapour +
stream1.getThermoSystem().getPhase(1).getBeta() * CpOil +
stream1.getThermoSystem().getPhase(2).getBeta() * Cpliquid ;
66
67     doubleDensityVapour = stream1.getThermoSystem().getPhase(0).getDensity();
68     doubleDensityliquid = stream1.getThermoSystem().getPhase(2).getDensity();
69     doubleDensityOil = stream1.getThermoSystem().getPhase(1).getDensity();
70     double Density1 = stream1.getThermoSystem().getWtFraction(0) *
DensityVapour + stream1.getThermoSystem().getWtFraction(1) * DensityOil +
stream1.getThermoSystem().getWtFraction(2) * Densityliquid ;
71
72     System.out.println("work " + compressor1.getTotalWork() + " density " +
Density1 + " Cp " + Cpone + " SPEED OF SOUND " +
stream1.getThermoSystem().getPhase(0).getSoundSpeed());
73
74     compressor1.solvePolytropicEfficiency(390.15); % use this line of the code only
for polytropic efficiency calculations
75     compressor1.getOutputStream().displayResult();

```

```

76     System.out.println("Hvap " +
stream1.getThermoSystem().getHeatOfVaporization() + " POLI " +
compressor1.getPolytropicEfficiency() + " dentity " +
stream1.getThermoSystem().getDensity() + " cp " +
stream1.getThermoSystem().getCp());
77     stream1.getThermoSystem().display();
78
79     double massFlowGas = stream1.getThermoSystem().getPhase(0).getBeta() *
stream1.getThermoSystem().getPhase(0).getMolarMass();
80     double massFlowLiq = stream1.getThermoSystem().getPhase(2).getBeta() *
stream1.getThermoSystem().getPhase(2).getMolarMass();
81     double massFlowOil = stream1.getThermoSystem().getPhase(1).getBeta() *
stream1.getThermoSystem().getPhase(1).getMolarMass();
82
83     double volFlowGas = massFlowGas /
stream1.getThermoSystem().getPhase(0).getDensity();
84     double volFlowLiq = massFlowLiq /
stream1.getThermoSystem().getPhase(2).getDensity();
85     double volFlowOil = massFlowOil /
stream1.getThermoSystem().getPhase(1).getDensity();
86
87     double GMF = massFlowGas / (massFlowGas + massFlowOil + massFlowLiq);
88     double GVF = volFlowGas / (volFlowGas + volFlowOil + volFlowLiq);
89     System.out.println("inlet stream - GMF " + GMF + " GVF " + GVF + " Z IN " +
stream1.getThermoSystem().getZ() + " Z OUT " +
compressor1.getOutStream().getThermoSystem().getZ());
90
91 }
92 }
93

```

A.2 COMPRESSOR CODE

```
1
2 package processSimulation.processEquipment.compressor;
3
4 import processSimulation.processEquipment.*;
5 import thermo.system.*;
6 import thermodynamicOperations.*;
7 import javax.swing.*;
8 import processSimulation.processEquipment.stream.*;
9
10 import java.awt.*;
11 import java.text.*;
12
13
14 public class Compressor extends ProcessEquipmentBaseClass implements CompressorInterface {
15     protected String name = new String();
16     SystemInterface thermoSystem;
17     ThermodynamicOperations thermoOps;
18     Stream inletStream;
19     Stream outputStream;
20     double dH = 0.0;
21     double inletEnthalpy = 0;
22     double pressure = 0.0;
23     private double isentropicEfficiency = 1.0, polytropicEfficiency = 1.0;
24     private boolean usePolytropicCalc = false;
25     boolean powerSet = false;
26
27     public Compressor() {
28     }
29
30     public Compressor(Stream inletStream) {
31         setInletStream(inletStream);
32     }
33
34     public Compressor(String name, Stream inletStream) {
35         this.name = name;
36         setInletStream(inletStream);
37     }
38
39     public void setName(String name) {
40         this.name = name;
```

```

41 }
42
43 public void setInletStream(Stream inletStream) {
44     this.inletStream = inletStream;
45
46     this.outStream = (Stream) inletStream.clone();
47 }
48
49 public void setOutletPressure(double pressure) {
50     this.pressure = pressure;
51 }
52
53 public double getEnergy() {
54     return dH;
55 }
56
57 public double getPower() {
58     return dH;
59 }
60
61 public void setPower(double p) {
62     powerSet = true;
63     dH = p;
64 }
65
66 public Stream getOutStream() {
67     return outStream;
68 }
69
70 public double solvePolytropicEfficiency(double outTemperature) {
71     double funk = 0.0, funkOld = 0.0;
72     double dfunkdPoly = 100.0, dPoly = 100.0, oldPoly = outTemperature;
73     run();
74     int iter = 0;
75     boolean useOld = usePolytropicCalc;
76     System.out.println("use polytropic " + usePolytropicCalc);
77     do {
78         iter++;
79         funk = thermoSystem.getTemperature() - outTemperature;
80         dfunkdPoly = (funk - funkOld) / dPoly;
81         double newPoly = polytropicEfficiency - funk / dfunkdPoly;
82         if (iter <= 1) {
83             newPoly = polytropicEfficiency + 0.01;
84         }

```

```

85     oldPoly = polytropicEfficiency;
86     polytropicEfficiency = newPoly;
87     isentropicEfficiency = newPoly;
88     dPoly = polytropicEfficiency - oldPoly;
89     funkOld = funk;
90     run();
91     System.out.println("temperature compressor " +
thermoSystem.getTemperature() + " funk " + funk + " polytropic " +
polytropicEfficiency);
92     } while (Math.abs((thermoSystem.getTemperature() - outTemperature) /
outTemperature) > 1e-5 || iter < 3);
93     usePolytropicCalc = useOld;
94     return thermoSystem.getTemperature();
95 }
96
97 public void run() {
98     System.out.println("compressor running..");
99     thermoSystem = (SystemInterface) inletStream.getThermoSystem().clone();
100    thermoOps = new ThermodynamicOperations(thermoSystem);
101    thermoSystem.init(3);
102    double presinn = thermoSystem.getPressure();
103    double hinn = thermoSystem.getEnthalpy();
104    double densInn = thermoSystem.getDensity();
105    double entropy = thermoSystem.getEntropy();
106    inletEnthalpy = hinn;
107
108    if (usePolytropicCalc) {
109        int numbersteps = 400;
110        double dp = (pressure - thermoSystem.getPressure()) / (1.0 * numbersteps);
111        for (int i = 0; i < numbersteps; i++) {
112            entropy = thermoSystem.getEntropy();
113            hinn = thermoSystem.getEnthalpy();
114            thermoSystem.setPressure(thermoSystem.getPressure() + dp);
115            thermoOps.PSflash(entropy);
116            double hout = hinn + (thermoSystem.getEnthalpy() - hinn) /
polytropicEfficiency;
117            thermoOps.PHflash(hout, 0);
118        }
119
120    } else {
121        thermoSystem.setPressure(pressure);
122
123
124        System.out.println("entropy inn.." + entropy);

```

```

125     thermoOps.PSflash(entropy);
126     double densOutIdeal = thermoSystem.getDensity();
127     if (!powerSet) {
128         dH = (thermoSystem.getEnthalpy() - hinn) / isentropicEfficiency;
129     }
130     double hout = hinn + dH;
131     isentropicEfficiency = (thermoSystem.getEnthalpy() - hinn) / dH;
132     dH = hout - hinn;
133     thermoOps.PHflash(hout, 0);
134 }
135 outStream.setThermoSystem(thermoSystem);
136 }
137
138 public void displayResult() {
139
140     DecimalFormat nf = new DecimalFormat();
141     nf.setMaximumFractionDigits(5);
142     nf.applyPattern("#.#####E0");
143
144
145     JDialog dialog = new JDialog(new JFrame(), "Results from TPflash");
146     Container dialogContentPane = dialog.getContentPane();
147     dialogContentPane.setLayout(new FlowLayout());
148
149     thermoSystem.initPhysicalProperties();
150     String[][] table = new String[50][5];
151     String[] names = {"", "Phase 1", "Phase 2", "Phase 3", "Unit"};
152     table[0][0] = "";
153     table[0][1] = "";
154     table[0][2] = "";
155     table[0][3] = "";
156     StringBuffer buf = new StringBuffer();
157     FieldPosition test = new FieldPosition(0);
158
159     for (inti = 0; i < thermoSystem.getNumberOfPhases(); i++) {
160         for (int j = 0; j < thermoSystem.getPhases()[0].getNumberOfComponents();
j++) {
161             table[j + 1][0] =
thermoSystem.getPhases()[0].getComponents()[j].getName();
162             buf = new StringBuffer();
163             table[j + 1][i + 1] =
nf.format(thermoSystem.getPhases()[i].getComponents()[j].getx(), buf, test).toString();
164             table[j + 1][4] = "-";
165         }

```

```

166     buf = newStringBuffer();
167     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 2][0] =
"Density";
168     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 2][i + 1] =
nf.format(thermoSystem.getPhases()[i].getPhysicalProperties().getDensity(), buf,
test).toString();
169     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 2][4] =
"[kg/m^3]";
170
171     buf = newStringBuffer();
172     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 3][0] =
"PhaseFraction";
173     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 3][i + 1] =
nf.format(thermoSystem.getPhases()[i].getBeta(), buf, test).toString();
174     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 3][4] = "[-
]";
175
176     buf = newStringBuffer();
177     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 4][0] =
"MolarMass";
178     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 4][i + 1] =
nf.format(thermoSystem.getPhases()[i].getMolarMass() * 1000, buf, test).toString();
179     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 4][4] =
"[kg/kmol]";
180
181     buf = newStringBuffer();
182     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 5][0] =
"Cp";
183     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 5][i + 1] =
nf.format((thermoSystem.getPhases()[i].getCp() /
thermoSystem.getPhases()[i].getNumberOfMolesInPhase() * 1.0 /
thermoSystem.getPhases()[i].getMolarMass() * 1000), buf, test).toString();
184     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 5][4] =
"[kJ/kg*K]";
185
186     buf = newStringBuffer();
187     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 7][0] =
"Viscosity";
188     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 7][i + 1] =
nf.format((thermoSystem.getPhases()[i].getPhysicalProperties().getViscosity()), buf,
test).toString();
189     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 7][4] =
"[kg/m*sec]";
190

```

```

191     buf = newStringBuffer();
192     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 8][0] =
"Conductivity";
193     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 8][i + 1] =
nf.format(thermoSystem.getPhases()[i].getPhysicalProperties().getConductivity(), buf,
test).toString();
194     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 8][4] =
"[W/m*K]";
195
196     buf = newStringBuffer();
197     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 10][0] =
"Pressure";
198     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 10][i + 1] =
Double.toString(thermoSystem.getPhases()[i].getPressure());
199     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 10][4] =
"[bar]";
200
201     buf = newStringBuffer();
202     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 11][0] =
"Temperature";
203     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 11][i + 1] =
Double.toString(thermoSystem.getPhases()[i].getTemperature());
204     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 11][4] =
"[K]";
205     Double.toString(thermoSystem.getPhases()[i].getTemperature());
206
207     buf = newStringBuffer();
208     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 13][0] =
"Stream";
209     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 13][i + 1] =
name;
210     table[thermoSystem.getPhases()[0].getNumberOfComponents() + 13][4] = "-";
211 }
212
213 JTableJtab = newJTable(table, names);
214 JScrollPaneScrollPane = newJScrollPane(Jtab);
215 dialogContentPane.add(scrollpane);
216 dialog.pack();
217 dialog.setVisible(true);
218 }
219
220 public String getName() {
221     return name;

```



```

222 }
223
224 public double getTotalWork() {
225     return thermoSystem.getEnthalpy() - inletEnthalpy;
226 }
227
228 public void runTransient() {
229 }
230
231 /**
232  * @return the isentropic efficiency
233  */
234 public double getIsentropicEfficiency() {
235     return isentropicEfficiency;
236 }
237
238 /**
239  * @param isentropicEfficiency the isentropic efficiency to set
240  */
241 public void setIsentropicEfficiency(double isentropicEfficiency) {
242     this.isentropicEfficiency = isentropicEfficiency;
243 }
244
245 /**
246  * @return the usePolytropicCalc
247  */
248 public boolean usePolytropicCalc() {
249     return usePolytropicCalc;
250 }
251
252 /**
253  * @param usePolytropicCalc the usePolytropicCalc to set
254  */
255 public void setUsePolytropicCalc(boolean usePolytropicCalc) {
256     this.usePolytropicCalc = usePolytropicCalc;
257 }
258
259 /**
260  * @return the polytropic efficiency
261  */
262 public double getPolytropicEfficiency() {
263     return polytropicEfficiency;
264 }
265

```

```
266 /**
267  * @param polytropicEfficiency the polytropic efficiency to set
268  */
269 public void setPolytropicEfficiency(double polytropicEfficiency) {
270     this.polytropicEfficiency = polytropicEfficiency;
271 }
272 }
```