



NTNU – Trondheim
Norwegian University of
Science and Technology

Subsea Wet Gas Compressor Dynamics

Luber Carlui Perez Aguilera

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Supervisor: Lars Eirik Bakken, EPT

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MASTER THESIS

for

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Subsea Wet Gas Compressor Dynamics
*Dynamisk respons til havbunnskompressor***Background and objective**

Current major challenges are related to efficient utilisation of existing infrastructure and exploration of new fields in the North Sea. Increased production of gas demands new field development based on sub-sea production, including bulk separation and wet gas compression. The wet gas compression represents an energy-intensive production segment in which limited research has been invested. Specifically this relates to the wet gas compression fundamentals. Wet gas compressors form a key element in future North Sea developments and it is of vital importance to understand and develop a consistent know-how basis for the dynamic behaviour under different operating conditions. The model fundamentals will be based on investigations of typical subsea wet gas compressor concepts

The following tasks are to be considered:

The objective is to form a consistent basis for a typical wet gas compressor behavior at different dynamic operating conditions. Based on available data from literature a dynamic model representing the wet gas compressor behavior should be developed.

Focus areas

1. Establish integrated wet gas compressor performance variation at different operating conditions, including internal motor and bearing cooling flows.
2. Evaluate and document Hysys Dynamic system functionality in order to model variation in operating conditions.
3. Establish an optimum Hysys Dynamic model based on results from part 1 and 2, and document compressor response at variation in gas volume fraction.

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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 2013



Olav Bolland
Department Head



Lars E. Bakken
Academic Supervisor

Research Advisor: Håvard Nordhus, Statoil

Preface

The present thesis represents the final work in the MSc in Natural Gas Technology, in the Department of Energy and Process at NTNU.

First of all I want to thanks my supervisor Professor Lars Bakken for its support and patient. He was very open and always willing to guide me during my work. I also want to thank Mr. Håvard Nordhus from Statoil for its technical support and time.

Thanks to all my friends, family and everyone has supported me during these months of hard work. Thanks.

Abstract

In this Thesis the Aspen HYSYS dynamic functionalities were explored in order to build a dynamic Wet Gas Compressor model. In particular the Automation feature was used to implement two different correction methods, interpolation and Wood's correction, that accounts for wet gas impact on compression performance, in dynamic-state. This was done through the creation of a VBA script in Microsoft Excel. The implementation of the correction methods showed to be fast and effective.

The HYSYS dynamic model and the VBA script were used to explore the performance of the compressor and the system under different operating conditions including wet gas showing that the total compressor power decreases when the GVF decreases. Additionally, the pipeline inlet pressure signal registered a maximum overshoot of 10% which seems to be acceptable. The major drawback of the Wood's correction method is it can report higher efficiencies at wet gas conditions than a dry gas conditions, which is inconsistent with experimental tests

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Nomenclature

List of symbols

| Symbol | Description | Unit |
|---------------|------------------------------|----------------------|
| D | Diameter | [m] |
| $E(t)$ | Error Signal | [-] |
| f | Polytropic head factor | [-] |
| H | Head | [m] |
| h | Enthalpy | [kJ/kg] |
| k | Isentropic exponent | [-] |
| K_c | Process gain | [-] |
| \dot{m} | Mass flow rate | [kg/s] |
| MW | Molecular weight | [kg/kmol] |
| N | Rotational speed | [rpm] |
| η | Efficiency | [-] |
| p | Pressure | [bara] |
| P | Compressor power | [MW] |
| Q | Volumetric flow rate | [m ³ /s] |
| R | Gas constant | [kJ/kgK] |
| T | Temperature | [K] |
| V | Gas velocity | [m/s] |
| v | specific volume | [m ³ /kg] |
| W | Wood's factor | [-] |
| X | Compressibility function X | [-] |
| Y | Compressibility function Y | [-] |
| Z | Compressibility factor | [-] |
| ε | Correction routine parameter | [-] |
| ρ | Density | [kg/m ³] |
| \dot{Q} | Electric motor heat | [kW] |

List of subscripts

| Subscript | Description |
|------------------|--------------------|
| 1 | Inlet |
| 2 | Outlet |
| c | compressor |
| $frac$ | fraction |
| g | gas |
| l | liquid |
| liq | liquid |
| mp | multiphase |

| | |
|------------|------------|
| n | new |
| o | old |
| p | polytropic |
| s | suction |
| <i>vap</i> | vapor |

List of abbreviations

| Abbreviation | Description |
|---------------------|----------------------------------|
| API | American Petroleum Institute |
| GMF | Gas Mass Fraction |
| GVF | Gas Volume Fraction |
| IC | Interpolation correction |
| LVF | Liquid Volume Fraction |
| ODEs | Ordinary Differential Equations |
| OP | Operating Parameter |
| PDEs | Partial Differential Equations |
| PFD | Process Flow Diagram |
| PI | Proportional Integral |
| PID | Proportional-Integral-Derivative |
| PV | Process Variable |
| SCS | Subsea Compression Station |
| SGC | Subsea Gas Compression |
| SGCS | Subsea Gas Compression Station |
| SGCT | Subsea Gas Compression Train |
| SI | International System |
| SP | Set Point |
| USGPM | U.S. Gallon Per Minute |
| VB | Visual Basic |
| VBA | Visual Basic for Applications |
| VSD | Variable Speed Drive |
| WC | Wood's correction |
| WGC | Wet Gas Compression |

1 INTRODUCTION

In the offshore oil and gas industry is intended to place the production facilities as near as possible to the hydrocarbon reservoir. However new resources are being discovered in more remote places, deeper waters and challenging climate conditions. Therefore, the industry is looking for new technologies that can unlock the, until now, stranded sources of energy.

In order to face this challenge one the most promising technology developments is Subsea Gas Compression (SGC). Whether is a new remote gas field (green field) or a depleted gas reservoir (brown field), SGC can extend the life of the field and by consequence increase revenues. Additionally, multiphase transport issues can be avoided by increasing the transport pressure.

The idea of SGC goes back to 1985 when Kjell Olav Stinessen presented the first known sketch (Lima, Storstenvik et al., 2011) and nowadays there are three ongoing projects in Norway: Åsgard, Gullfaks and Ormen Lange. One of the key elements of these developments is the compressor and a lot effort is put on creating a solid know-how basis on its operation and correct performance prediction.

In this work, it will be presented a study of the dynamic behavior of subsea wet gas compressor. The commercial simulation software HYSYS will be used to build a representative dynamic model of a typical Wet Gas Compression Train (SGCT). The development will be based on an integrated motor compressor or MOtor Pipeline Compressor, MOPICO, (Gilon, 1991). The field data that has been used is be based on the Åsgard field characteristics however the developed model is slightly different.

In particular, the HYSYS dynamic functionalities will be explored in order to build a dynamic WGC (Wet Gas Compressor) model that accounts for wet gas impact on compression performance. The Automation feature of HYSYS will be used in the dynamic-state to implement two different correction methods: Interpolation correction and Wood's correction. After, The HYSYS dynamic model will be used to explore the performance of the compressor and the system under different operating conditions including wet gas.

2 THEORY FUNDAMENTALS

In this chapter, it will be introduced the main theoretical foundations that support the work carried out. It will be briefly presented the existing compressor types with special emphasis on the centrifugal compressor. Focus will be also put on compressibility phenomenon's such as surge and stonewall, and different approaches for centrifugal compressor performance analysis including wet gas compression.

Additionally, it will be explored the basis in process control theory and the principal compressor control systems. At the end of the chapter, a short introduction is made on electric motor and drivers and control theory

The content of this Chapter is rather wide as it tries to cover all the topics of the Thesis. For more specific review of each matter the reader is advised to consult (Bequette, 2003; Lüdtke, 2004; Forsthoffer, 2005; Peng, 2008; Saravanamuttoo, Cohen et al., 2009; Romagnoli and Palazoglu, 2012).

2.1 Type of compressors

In the industry there are a vast number of compressor types with different characteristics and principles. However, they can be grouped in two main groups: 1) positive displacement compressors and 2) dynamic compressors.

Positive displacement compressors work under the principle of increasing the pressure of a gas on a fixed volume or confined space. The characteristics of these types of compressors are fixed volume, variable pressure capability in terms of energy or head, and not self-limiting. This means that the compressors will stall unless a device to limit the pressure is included. Examples of these types of compressors are (Forsthoffer, 2005)

- Reciprocating
- Sliding-vane
- Rotary-liquid piston
- Rotary lobe
- Screw

On the other hand, dynamic compressors use rotating blades to increase the pressure of the gas by increasing its velocity. The performance of a dynamic compressor is essentially determined by the inlet conditions of the gas, the blade tip speed velocity and the relative velocity of the gas between the blades. Two types of dynamic compressors are (Forsthoffer, 2005):

- Centrifugal
- Axial

Different from positive displacement compressors, the dynamic compressor has fixed energy capacity or head for a given (variable) suction volume flow and its performance curve is very

different from the positive displacement curves. The Figure 2.1 shows a comparison between both types of compressors.

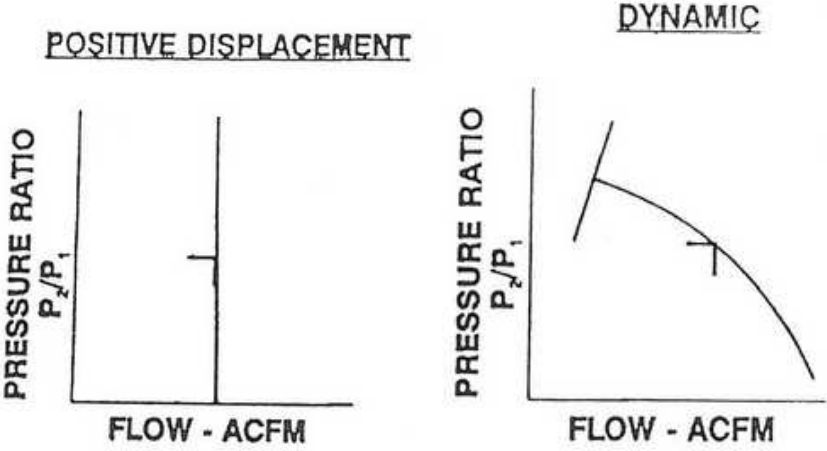


Figure 2.1 Performance curve for positive displacement and dynamic compressor (Forsthoffer, 2005)

2.2 The centrifugal compressor

As mentioned in the previous section, the centrifugal compressor increases the pressure of the gas by increasing its tangential velocity. There are two main components of a centrifugal compressor, impeller and diffuser. The impeller is mounted on a rotating shaft and placed inside of a stationary casing. The diffuser, is a set of diverging passages which surrounds the impeller (Forsthoffer, 2005; Saravanamuttoo, Cohen et al., 2009). The gas is sucked into the compressor by an inlet nozzle which avoids shock or turbulence and routes the gas to the impeller. In the impeller, the gas increases its energy level by increasing its velocity. Next, the gas goes through the diffuser which decelerates the gas by increasing the radial area and converts the kinetic energy of the gas in static pressure. The Figure 2.2 presents the cross section of a centrifugal compressor.

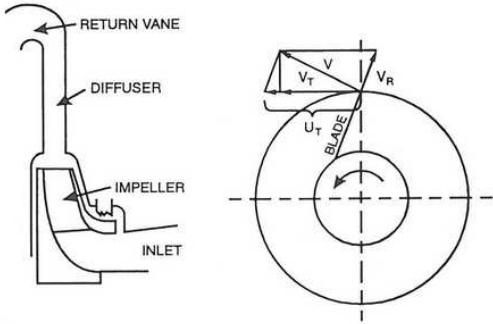


Figure 2.2 Centrifugal compressor cross section (Forsthoffer, 2005)

In a single-stage compressor the compressed gas leaves the diffuser through a volute casing which collects and directs the gas to the discharge nozzle, see Figure 2.3. The volute has the functionality of further reduce the velocity of the gas and gain additional static pressure. In a multistage compressor, the gas leaving the first diffuser goes to a return vane and a return channel which route the gas to the next impeller. After the gas has passed through the last diffuser it is finally routed to the discharge volute.

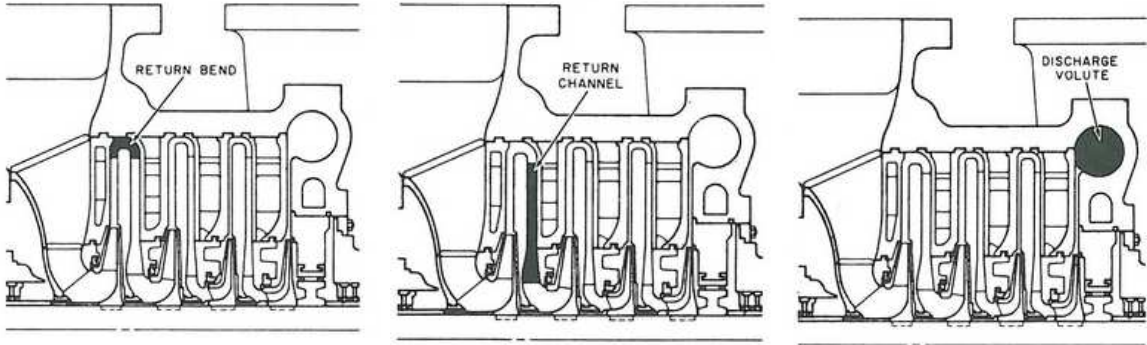


Figure 2.3 Centrifugal compressor (Lapina, 1982)

A compression stage usually refers to the combination of an impeller, a diffuser, a return vane and return channel; while the term compressor section refers to a combination of stages. In this work, we will refer to impeller also as compression stage.

2.3 Compressor phenomena’s

Shock waves

In centrifugal compressors, shock waves can occur when the velocity of the gas relative to the surface which it is moving approaches the speed of sound in the fluid (Saravanamuttoo, Cohen et al., 2009). In this case the Mach number, see Equation 2.1(Lapina, 1982), is close to the unity.

$$M_a = \frac{V}{\sqrt{kgZRT}} \tag{2.1}$$

The shock waves inside the compressor can cause flow breakdown and rapid increment of the internal losses. It is then of high importance to take care of the Mach number, both at the impeller tip and at the leading edges of the diffuser. As a rule of thumb, the Mach numbers can be restricted to around 0.8.

Surge

Surge is a periodic phenomenon produced by the interaction between the compressor and the system. It is a result of flow separation originated by low gas velocity anywhere in the compressor. It could be at the inlet guide vane, the suction section of the impeller, in the middle or discharge of the impeller or even at the diffuser.

At surge conditions the compressor is not able to generate the head required by the process system, see Figure 2.4. The theoretical surge flow point can be calculated by Equation 2.2. However, the exact location of the surge point cannot be predicted due to manufacturing tolerances and specific characteristics of the compressor and the system (Tveit, Bakken et al., 2005).

$$\left[\frac{\partial H_p}{\partial Q_s} \right]_n = 0 \quad 2.2$$

Some of the consequences are that back flow can occur, pressure turns unstable, noise and vibrations increase producing damage due to mechanical stress. Additionally, the rapid movement of the gas moving back and forth inside the compressor generates large quantities of energy which increase the temperature of the flow. Hence, the compressor is more susceptible to thermal stress (Forsthoffer, 2005).

The impact of surge varies from one application to another and it is more severe in high pressure system and in process gases with high molecular weight. In compression process working with low density gases, as hydrogen for instance, the mechanical damage due to surge might remain hidden and is detectable only during physical inspection. Several characteristics of surge are (Forsthoffer, 2005):

- It is a high speed phenomenon. Back flow might happens is less than 150 ms.
- Reversal rate is from 30 to 120 cycles/sec.
- Pressure turns unstable.
- Noise and vibrations are generated.
- Rapid increase of temperature.
- Compressor could trip.
- The severity varies with the application.

Rotating stall

Rotating stall is another important compressibility effect which can cause instabilities and bad compressor performance. It can generate surge but can also appear in normal stable operation range. Rotating stall in the product of non-uniform flow or geometry between vanes or blades and it passes from channel to channel. It could cause aerodynamically vibrations and hence fatigue failures (Saravanamuttoo, Cohen et al., 2009).

Stonewall (Choke)

Stonewall is also referred as choke and it is defined as the maximum volume flow that the impeller, o compressor, can handle, see Figure 2.4. It is generated by an excessive volume flow (proportional to the gas velocity) through the impeller. At this condition, the gas velocity at the impeller inlet tends to the sonic velocity and the relative Mach number is equal to the one.

Stonewall it is not a damaging phenomenon but it constraints the energy delivery of the compressor and it affect the overall process. Export compressors connected to pipelines have to produce large amounts of head and stonewall is very critical in this case. However, in low pressure systems the compressors could be operated at stonewall conditions.

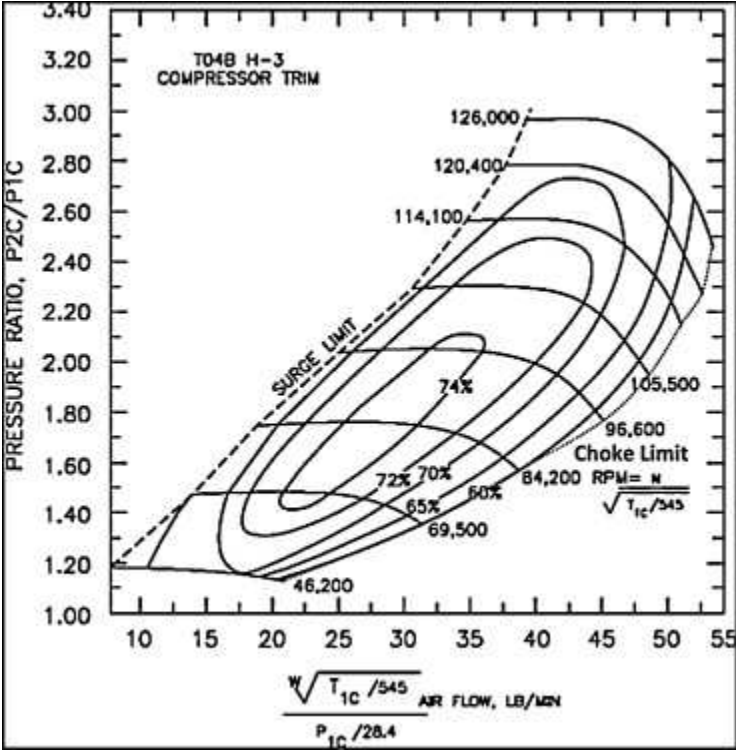


Figure 2.4 Typical compressor map showing surge and choke lines. (Razib, Hasan et al., 2012)

2.4 Performance relationships

The performance of an existing compressor can be predicted based on several parameters that relate basic fluid dynamics with the geometry of the machine. In this section, it will be described the use of compressor maps and the so-called fan laws.

2.4.1 Performance characteristics

The performance characteristics or so-called compressor maps are used to relate the different compression parameters by using non-dimensional numbers. In (Saravanamuttoo, Cohen et al., 2009) it is shown that when the gas composition and the impeller diameter are both constant the Buckingham Pi theorem can be used to find four non-dimensional numbers that describe the performance of a centrifugal compressor:

$$f\left(\frac{p_2}{p_1}, \frac{T_2}{T_1}, \frac{\dot{m}\sqrt{T_1}}{p_1}, \frac{N}{\sqrt{T_1}}\right) = 0 \quad 2.3$$

By using these numbers it is possible to plot the performance curve of the compressor. The curves can be used then to predict the behavior of the compressor under different conditions and implement the corresponding regulation methods.

Moreover, for a better understanding the temperature ratio can be expressed in terms of the adiabatic efficiency. So often the performance curves are expressed in terms of pressure ratio and adiabatic efficiency versus $m\sqrt{T_1}/P_1$ with $N/\sqrt{T_1}$ as parameter. A typical compressor characteristic curve for a centrifugal compressor is shown in Figure 2.5 and Figure 2.6.

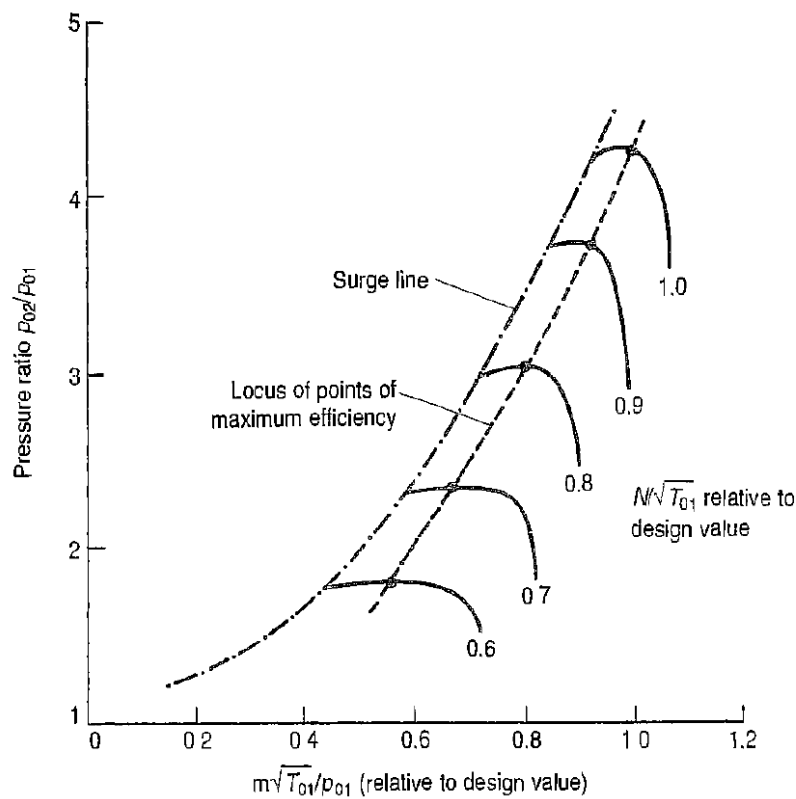


Figure 2.5 Typical pressure ratio curve for a centrifugal compressor (Saravanamuttoo, Cohen et al., 2009)

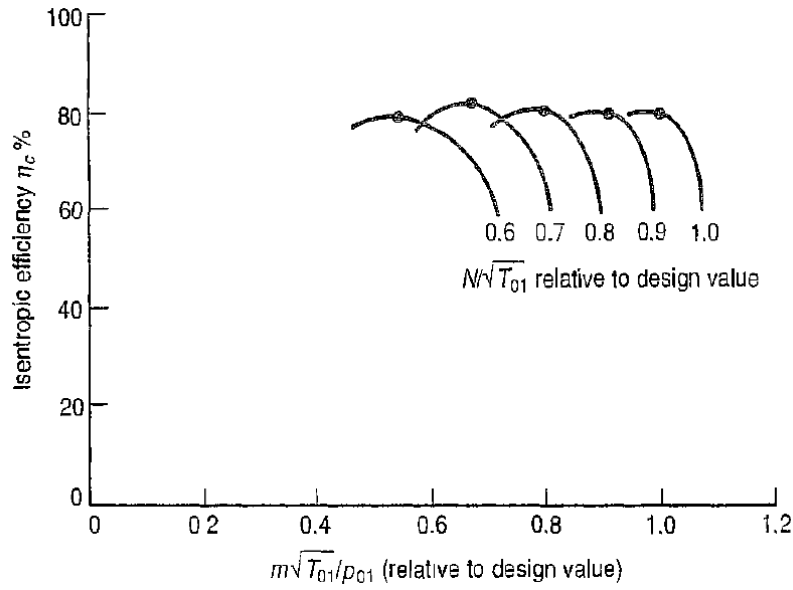


Figure 2.6 Typical efficiency curve for centrifugal compressor (Saravanamuttoo, Cohen et al., 2009)

Additionally, the performance curves can be expressed in terms of head vs. suction volume flow in order to take into account changes in inlet gas temperature, inlet pressure or molecular weight. In this work, such types of curves are used.

2.4.2 Fan laws

The fan laws or similarity laws relate different non-dimensional parameters and can be used to predict the performance of a machine operating under different conditions. They can also be used to predict the performance of other geometrically similar machines operating at the same conditions (Peng, 2008).

In order to apply the similarity laws all the linear dimensions of two compressors have to be in constant proportion and all the angular dimensions are equal for every element of the compressors. Also, the flow patterns need to be the same in both cases (Peng, 2008).

The fan laws directly relate the suction volume flow, impeller diameter, rotational speed, compressor head and compressor power:

$$\frac{Q_o}{D_o^3 \cdot N_o} = \frac{Q_n}{D_n^3 \cdot N_n} \quad 2.4$$

$$\frac{H_o}{D_o^2 \cdot N_o^2} = \frac{H_n}{D_n^2 \cdot N_n^2} \quad 2.5$$

$$\frac{P_o}{D_o^5 \cdot N_o^3} = \frac{P_n}{D_n^5 \cdot N_n^3} \quad 2.6$$

In particular, in compressors and identical impellers diameters:

$$\frac{Q_o}{N_o} = \frac{Q_n}{N_n} \quad 2.7$$

$$\frac{H_o}{N_o^2} = \frac{H_n}{N_n^2} \quad 2.8$$

$$\frac{P_o}{N_o^3} = \frac{P_n}{N_n^3} \quad 2.9$$

2.5 Compression analysis

In this section it will be presented three different thermodynamic approaches to estimate the compressor performance.

2.5.1 Isentropic analysis

In an isentropic process the change of state is determined by:

$$pv^k = \text{constant} \quad 2.10$$

When applying the ideas gas law the relation between pressure and temperature becomes:

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k}{k-1}} \quad 2.11$$

The isentropic head, H_s , is defined by:

$$H_s = \int_1^{2s} v dp \quad 2.12$$

In order to solve the integral it is necessary to assume the isentropic exponent as constant when applying combining with Eq. 2.10. The isentropic head is then given by:

$$H_s = \frac{k}{k-1} [p_2 v_{2s} - p_1 v_1] \quad 2.13$$

The isentropic head equation can be al re-written by applying $pv = ZRT$:

$$H_s = \frac{k}{k-1} Z_1 R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1 \right] \quad 2.14$$

The isentropic efficiency is then defined as the ratio of the isentropic head to the total actual head:

$$\eta_s = \frac{H_s}{H} \quad 2.15$$

The total compressor power is then defined as:

$$P = H \cdot \dot{m} = \dot{m} \frac{H_s}{\eta_s} \quad 2.16$$

2.5.2 Polytropic analysis

In a Polytropic process the change of state is determined by:

$$pv^n = \text{constant} \quad 2.17$$

When applying the ideas gas law the relation between pressure and temperature becomes:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n}{n-1}} \quad 2.18$$

The polytropic head, H_p , is defined by:

$$H_p = \int_1^2 v dp \quad 2.19$$

In order to solve the integral it is necessary to assume the polytropic exponent as constant when applying combining with Eq. 2.17. The isentropic head is then given by:

$$H_p = \frac{n}{n-1} [p_2 v_2 - p_1 v_1] \quad 2.20$$

The polytropic head equation can be al re-written by applying $pv = ZRT$:

$$H_p = \frac{n}{n-1} Z_1 R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \quad 2.21$$

The polytropic efficiency is then defined as the ratio of the polytropic head to the total actual head:

$$\eta_p = \frac{H_p}{H} \quad 2.22$$

The total compressor power is then defined as:

$$P = \dot{m} \frac{H_p}{\eta_p} \quad 2.23$$

The polytropic head exponent can be calculated with the Eq. 2.24 when considering ideal gas:

$$\frac{n-1}{n} = \frac{1}{\eta_p} \frac{k-1}{k} \quad 2.24$$

2.5.3 Schultz analysis

The Schultz analysis is based on the polytropic analysis however it considers two individual polytropic exponents, polytropic temperature exponent and polytropic volume exponent (Schultz, 1962). In particular the polytropic volume exponent defines the change of state:

$$pv^{n_v} = \text{constant} \quad 2.25$$

The polytropic temperature exponent and the polytropic volume exponent, at constant efficiency, are defined as:

$$n_T = \frac{1}{1 - \frac{p}{T} \left(\frac{\delta T}{\delta p} \right)_{\eta_p}} \quad 2.26$$

$$n_v = - \frac{v}{Tp} \left(\frac{\delta p}{\delta v} \right)_{\eta_p} \quad 2.27$$

In this analysis, two additional compressibility functions, X and Y, are added:

$$X = \frac{T}{v} \left(\frac{\delta v}{\delta T} \right)_p - 1 \quad 2.28$$

$$Y = - \frac{p}{v} \left(\frac{\delta v}{\delta p} \right)_T \quad 2.29$$

Furthermore the polytropic exponent can be expressed in terms of the compressibility functions:

$$n_v = \frac{(1 + X)}{Y \left[\frac{1}{k} \left(\frac{1}{\eta_p} + X \right) - \left(\frac{1}{\eta_p} - 1 \right) \right]} \quad 2.30$$

$$\frac{n_T - 1}{n_T} = \frac{\left(\frac{k-1}{k} \right) \left(\frac{1}{\eta_p} + X \right) Y}{(1 + X)^2} \quad 2.31$$

Similarly, the polytropic head can be calculated when solving the integral given by the Eq. 2.19. However, the Schultz procedure includes a correction factor f to take into account the changes in the polytropic volume exponent along the compression path.

$$H_p = f \frac{n_v}{n_v - 1} [p_2 v_2 - p_1 v_1] \quad 2.32$$

The polytropic head equation can be re-written by applying $p v = ZRT$:

$$H_p = f \frac{n_v}{n_v - 1} Z_1 R T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_v - 1}{n_v}} - 1 \right] \quad 2.33$$

The polytropic volume exponent can be calculated at the given discharge conditions as:

$$n_v = \frac{\ln \left(\frac{p_2}{p_1} \right)}{\ln \left(\frac{v_1}{v_2} \right)} \quad 2.34$$

The polytropic head correction factor is defined as:

$$f = \frac{h_{2s} - h_1}{\frac{k_v}{k_v - 1} (p_2 v_{2s} - p_1 v_1)} \quad 2.35$$

Where the isentropic volume exponent is defined as:

$$k_v = \frac{k}{Y} \quad 2.36$$

Or as:

$$k_v = \frac{\ln \left(\frac{p_2}{p_1} \right)}{\ln \left(\frac{v_1}{v_{2s}} \right)} \quad 2.37$$

The method of calculating the average properties for k , X and Y , is not specifically defined and its calculation will directly influence the performance estimation (Hundseid, 2008)

Finally, the polytropic efficiency is also given by:

$$\eta_p = \frac{H_p}{H} \quad 2.22$$

2.6 Wet gas compression

The understanding of wet gas compression comprises many phenomenon's that are not present in single phase compression. In multiphase compression the interaction between the phases in terms of mass, momentum and heat exchange, contribute to the compressor performance. Interest on this matter is rapidly increasing due to represent one of the key elements in the development of subsea wet gas compressors. The cited works present an extend study of this matter based on actual lab tests: (Brenne, Bjørge et al., 2005; Brenne, Bjørge et al., 2008; Hundseid, Bakken et al., 2008).

The liquid content in the gas phase has a significant impact on the compressor aerodynamics and thermodynamics of the fluid. It is possible to find: mass and heat transfer (evaporation and condensation) even at the same time, liquid entrainment and deposition in the gas phase, and liquid film formation (Hundseid, Bakken et al., 2008). All this effects are present at the same so it makes difficult the development of correct performance prediction methods.

The consulted references have shown through experimental data that the specific polytropic head decreases as the GVF decreases and the liquid content increases in the gas phase. Likewise, experiments have reported that the polytropic efficiency also decreases as the GVF decreases. Therefore, prediction of the compressor performance under wet gas conditions is extremely important. In this work, available real test data have been used to predict the necessary compressor characteristic curves under wet gas conditions.

2.6.1 The single fluid model approach

In the case of wet gas compression, it is possible to use a single-fluid model as expressed below. The multiphase polytropic exponent can be calculated as:

$$n_{mp} = \frac{\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{v_{mp1}}{v_{mp2}}\right)} \quad 2.38$$

The multiphase polytropic head is the given by:

$$H_{p,mp} = \frac{n_{mp}}{n_{mp} - 1} [p_2 v_{mp2} - p_1 v_{mp1}] \quad 2.39$$

Where the multiphase specific volume is defined as:

$$v_{mp} = \frac{1}{GVF\rho_G + (1 - GVF)\rho_l} \quad 2.40$$

And the Gas Volume Fraction is:

$$GVF = \frac{Q_g}{Q_g + Q_l} \quad 2.41$$

This approach is used for HYSYS when liquid is present in the centrifugal compressor unit.

2.7 Introduction to control theory

In Dynamic simulations, as in the actual process operations, the control design of the process is a key element which needs special attention. The present section is not intended to be a deep review of process control but just an overview of the most important concepts that are needed for a clear understanding of the dynamic model developed in this thesis. For a full review of process control the reader is advised to consult (Marlin, 2000; Bequette, 2003; Romagnoli and Palazoglu, 2012).

Process Variable

The Process Variable (PV), also referred as controlled variable or output variable, is the variable that needs to be controlled in the process or system. The process variable is closely related to the process product specification, process safety or specific operating conditions.

Set point

It is the desired value for the controlled variable.

Error signal

It is the difference between the set point and the current value of the process variable, at any time t .

Steady-state offset

Offset is the difference between the steady-state values of the set point and the process variable. In most processes, the steady-state offset is required to be zero.

Operating Variable

The Operating Variable, OP, also referred as manipulated variable must be a process variable which can be changed independently making the controlled variable to achieve its set point. The manipulated variable should have enough range of coverage, e.g. % valve opening, so it can bring the process variable to its set point under unforeseen system disturbances.

Disturbances

Disturbances are events that affect the process system and make the controlled variables to change from their desired values or set points. These events or disturbances cannot be changed or manipulated therefore the control system should be able to handle all the potential disturbances and return the process variables to their set points.

Overshoot

The overshoot can be defined as the difference between the highest value of the process response and its steady-state value. In some literature it is referred to the manipulated variable (Marlin, 2000) while in others for the controlled variable (Bequette, 2003). Often, it is expressed as the ratio of change (overshoot ratio) of the highest value to the steady-state value.

Open loop control

In an open loop control, the corrective action is made periodically according certain principles such as statistical data compilation of the process variable values. In this case, the value of the measured variables is displayed to the operator who is able to manipulate directly, by a use of controller or by hand, the final control element, e.g. a valve (Marlin, 2000). A typical open loop control process is shown in Figure 2.7. In the open loop control the controller sets the input to the process without any feedback from the process variable.

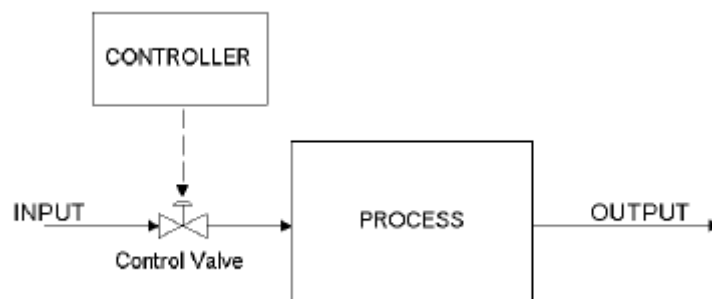


Figure 2.7 Typical open loop control system (Aspen Technology, 2011)

Closed loop control

A closed loop or feedback control uses the process output as input signal to the controller which, according the given algorithm, computes the required action to apply to the control element. The function of the closed control loop is to maintain a small deviation between the controlled variable and the specified set point by adjusting the manipulated variable (Marlin, 2000). The Figure 2.8 shows a typical closed loop control system.

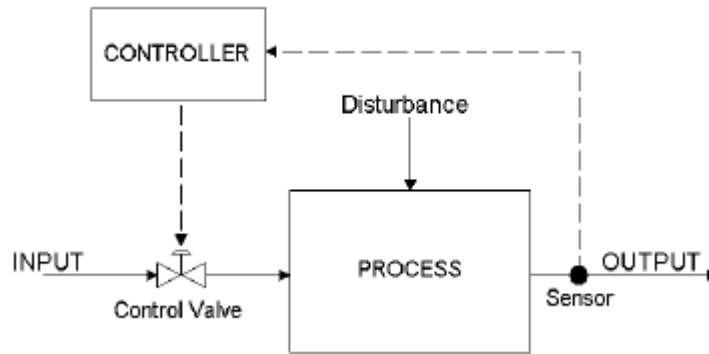


Figure 2.8 Typical closed loop control system (Aspen Technology, 2011)

Proportional control

In the proportional algorithm the control action, e.g. % valve opening, is proportional to the error signal. As the error increases, the adjustment of the operating variable also increases, and vice versa. The P-only algorithm can be written as:

$$OP(t) = OP_{SS} + K_c E(t) \quad 2.42$$

Where:

OP_{SS} , bias. Initialization term

K_c , process gain

$E(t)$, signal error

Although the proportional controller can damp out the oscillations of the process variable under disturbances it cannot reach a zero steady-state offset in a feedback control system. This is one of the major drawbacks of the P-only controller and it should be considered only when a quick response is desired or in systems where steady-offset are not important.

Proportional Integral control

The Proportional-Integral (PI) controller is able to dampen out the process variable oscillations but also the return it to its set point thus achieving zero steady-state offset. The PI controller includes both a proportional and integral term, see Equation 2.43.

$$OP(t) = K_c E(t) + \frac{K_c}{T_i} \int E(t) \quad 2.43$$

Where:

T_i , integral time

The integral time, T_i , is defined as the time taken for the controller output signal to move an amount equivalent to the error. As it can be seen from the Equation 2.43 the integral action

will decrease when the integral time increases, and the opposite. It is important to take into account this fact due to large reductions in integral time (increasing the integral effect) will prolong the response time.

The drawback of this controller type is that the dynamic response is slower so the oscillation period is higher. This makes the process variable to take more time to stabilize in comparison with the Proportional controller. A graphic comparison between both controllers response is shown in the Figure 2.9. If it is tuned improperly the PI controller could cause system instabilities.

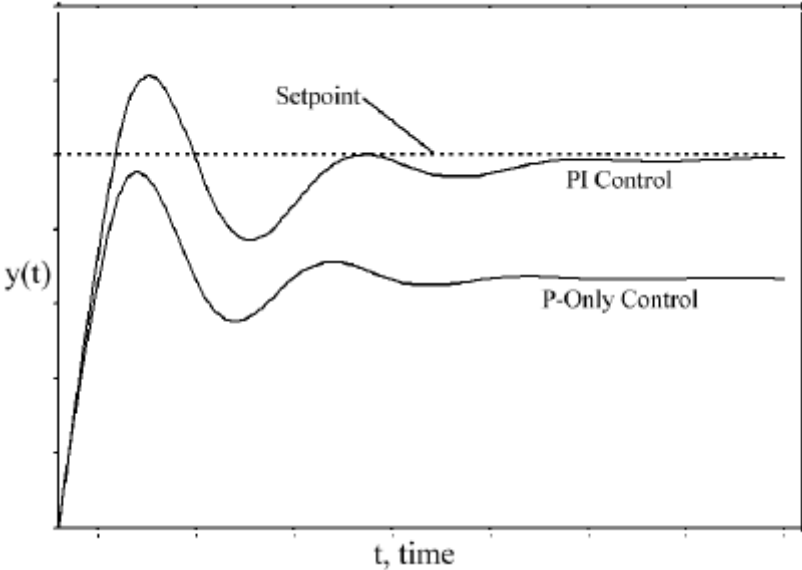


Figure 2.9 Proportional and PI control response (Aspen Technology, 2011)

The PID control

The Proportional-Integral-Derivative (PID) algorithm adds and derivative term which can reduce the natural period of oscillation in comparison to the PI controller, see Equation 2.44. The derivative term of the PID controller offers the possibility to anticipate the direction of the error and act quicker than the P- or PI- controller.

$$OP(t) = K_c E(t) + \frac{K_c}{T_i} \int E(t) + K_c T_d \frac{dE(t)}{dt} \tag{2.44}$$

Where:
 T_d , derivative time

The derivative time, T_d , is defined as the time required for the proportional action to reach the same level as the derivative action (Aspen Technology, 2011). The integral term does not affect the final steady-state value of the error but it gives a quick response based on the rate of change of the process variable (Marlin, 2000). Hence, the response time of the PID controller can be compared to the P-controller. The Figure 2.10 shows a comparison between the answer

of the PI and PID controller. Note that PID response presents less oscillation while achieving the same steady-state value as the PI controller, zero offset.

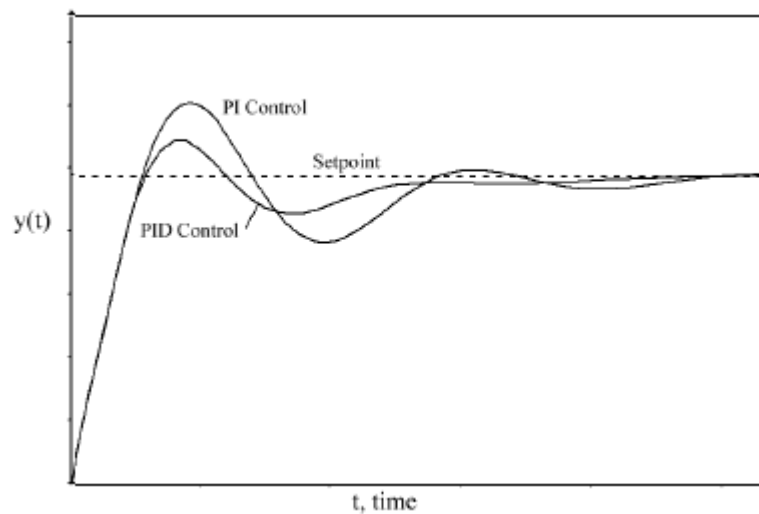


Figure 2.10 PI and PID control response (Aspen Technology, 2011)

2.8 Compressor Control

During compressor off-design operation the inlet conditions are not equal to design and by consequence the compressor performance is affected. Therefore, it is necessary to implement regulation or control systems in order to mitigate possible undesired effects and operate under the process specification. There are many proven mechanism for compressor control: variable speed, suction throttling, adjustable inlet guide vanes, adjustable diffuser vanes and/or bypass regulation (discharge throttling). Among them, the rotational speed control is implemented in most processes due to its high capabilities.

Additional to operation regulation methods it is important to protect the compressor from unforeseen situations like power shortage which can lead the compressor into the surge area. In this case, an anti-surge control system can be implemented as safety barrier in order to prevent the compressor going into surge area.

In the following sub-sections it will be described the main features of the compressor speed and anti-surge control. For a total review of other compressor regulation methods the reader is advised to consult (Lapina, 1982; Lüdtke, 2004; Forsthoffer, 2005; Peng, 2008).

2.8.1 Variable speed

When controlling the compressor speed it is possible to operate in such way that it only produces the necessary head which conduces to relatively high efficiencies at part load. In the same way, it is possible overload operation by operating above the design compressor speed. In multistage compression running on the same shaft, all the compressor stages or impellers are driven at the same speed. This entails an intrinsic relation between the compressor speed and the head capacity and efficiency of every stage.

In industry operation the compressor speed is adjusted using a Variable Speed Drive (VSD). The great flexibility and large operability range of this control mechanism makes it wide used in the industry and it is a key element in subsea applications.

The speed control is based on varying the synchronous speed of the stator magnetic field. The speed of the field is given by:

$$N = \frac{120f}{p} \quad 2.45$$

Where:

f , Supply frequency

N , Synchronous speed

p , Number of magnetic poles

Since the number of poles must be an even integer, the best approach to speed control over a wide range is to provide a variable-frequency supply (Hughes, 2006).

2.8.2 Anti-surge

As mentioned before surge is a cyclic or periodic phenomenon due to the interaction of the compressor with the system hence it will always occur and it cannot be totally eliminated. However, it is possible to avoid it by a so called anti-surge control. In an anti-surge control a fraction of the discharge flow is re-circulated and sent back to the compressor inlet (see Figure 2.11).

An anti-surge controller, normally a Proportional Integral (PI) controller, determines the operating point of the compressor and its position with respect to the control line. If the operating point is between the surge and control line, the controller activates the recycle flow by means of the anti-surge control valve. In this way, the suction volumetric flow increases and the operating point is moved to the right of the control line (see Figure 2.12). The control line is typically 10 to 15 % of the surge line but it varies with the size of the compressor.

The upstream and downstream flow transmitter, pressure transmitters and temperature transmitters send a signal to the anti-surge controller where it is analyzed. The controller normally uses a polynomial equation to approach the control line and determine the set point using the inlet and discharge pressure. The set point, which lies on the control line, is then compared to the operating point and the controller sends a signal to close or open the control valve. Depending of the operating conditions, the compressor can move towards the surge line in fraction of seconds. Therefore the control mechanism, including transmitters, logic solver and actuators must be fast enough to avoid surging.

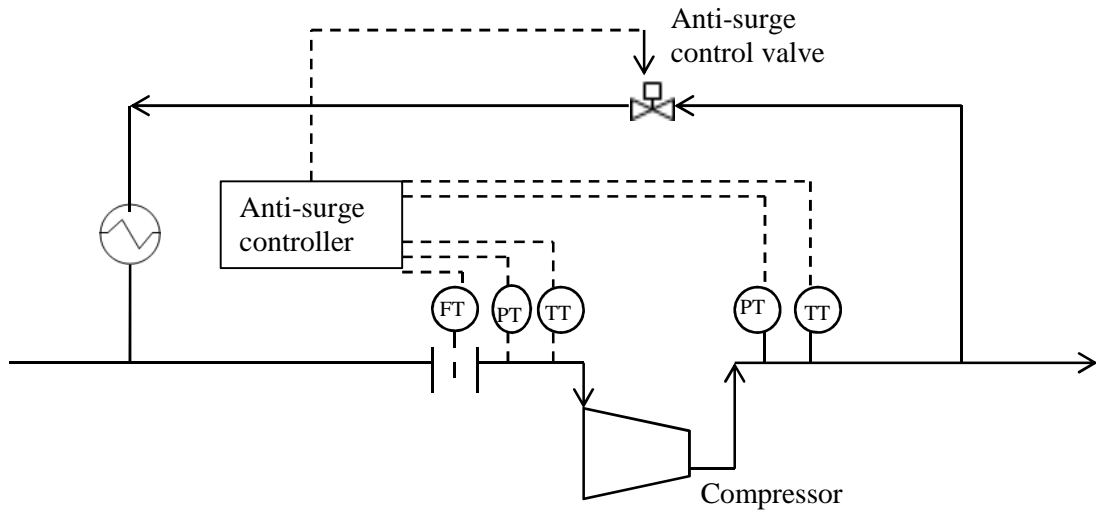


Figure 2.11 Typical anti-surge control system

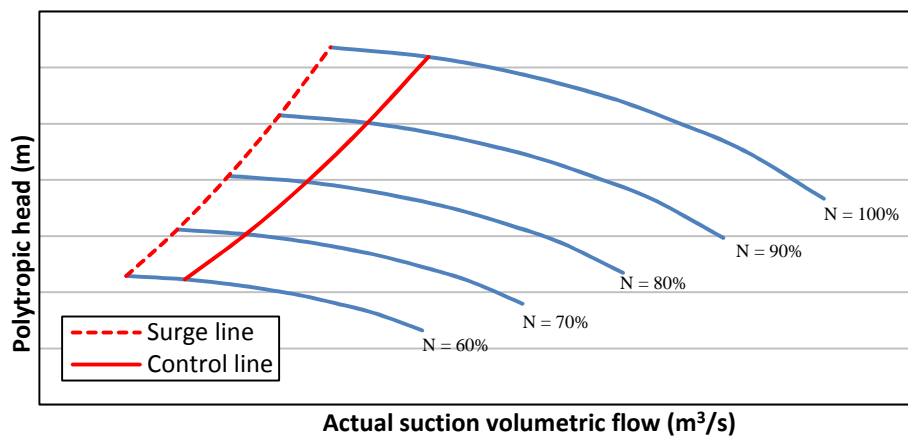


Figure 2.12 Typical compressor operating map

3 SUBSEA GAS COMPRESSION

The main benefits of subsea gas compression compared to traditional platform compression can be listed as (Bjerkreim, 2004; Lima, Storstenvik et al., 2011):

- Increase recovery. Since the boosting is placed closer to the production wells it has a higher positive impact on production which can be maintained longer.
- Lower capital cost.
- Potential reduce of operational cost.
- Due to its unmanned character it can be considered as safer than topside activities.
- Less environmental impact. SCS are expected to be smaller and lighter than offshore platforms.
- Reduce flow assurance problems due to higher transport pressure.

3.1 Subsea Gas Compression station main components

Despite the common believe subsea gas compression is not a young technology but it has been developed for many years. The first idea of subsea compression was presented by Kjell Olav Stinessen in 1985 (Lima, Storstenvik et al., 2011):

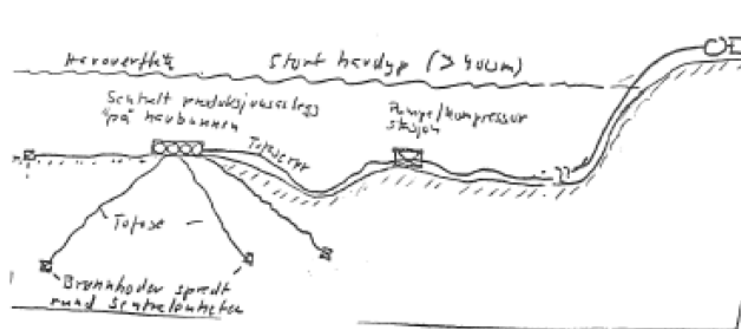


Figure 3.1 First sketch of subsea gas compression (Lima, Storstenvik et al., 2011)

Since then, several prototypes have been developed implementing different approaches with respect gas liquid separator, electric motor and compressor integration, configuration and marinization of topside components. The current projects for subsea gas compression are based in two different visions. The first and classical concept uses two phase upstream separation and further single phase compression and boosting. The Åsgard and Ormen Lange projects belong to this type of subsea gas compression. On the other hand, the Gulfaks project belongs to a more novel concept where the well stream is compressed without prior separation. This last ends in a development which is more compact and tolerant to gas liquid content.

The Table 3.1 summarizes the main features of the three current projects for subsea gas compression (Skoftealand, Hilditch et al., 2009; Davies, Bakke et al., 2010; Hjelmeland, Olsen et al., 2011; Lima, Storstenvik et al., 2011; Statoil, 2012).

Table 3.1 Characteristic data from the current subsea gas compression projects

| Project | Åsgard | Ormen Lange | Gullfaks |
|---------------------------------|---|---|---|
| Characteristic | | | |
| Country | Norway | Norway | Norway |
| Planned start-up | 2015 | 2016 | 2015 |
| Design lifetime | 30 years | 30 years | 20 years |
| Water depth | 250 – 325 m | 850 m | 135 m |
| Step out | 40 km | 120 km | 15 km |
| Design gas flow rate | 25 MSm ³ /d | 60 MSm ³ /d | 11 MSm ³ /d |
| Additional gas recovery | 30 billion Sm ³ gas | 30-50 billion Sm ³ gas | 2.5-3 billion Sm ³ gas |
| ΔP | 60 bar | 60-70 bar | 30 bar |
| Design pressure | 210 bar | 255 bar | 30 bar |
| Max. LVF into compressor | 0.46 | API standard | 0.25-1.92 |
| Number of trains | 2 + 1 spare | 2 | 2 |
| Compressor power | 2 x 11.5 MW | 2 x 12.5 MW | 2 x 5 MW |
| Structure size | 75 m x 45 m x 20 m | 60 m x 38 m x 12 m | 34 m x 20 m x 12 m |
| Structure weight | 4 800 tons | 3300 tons | 950 tons |
| Type and number of compressors* | 2 integrated motor centrifugal compressor | 2 subsea + 2 onshore vertical integrated motor centrifugal compressor | 2 multiphase axial counter-rotating compressors |
| Type and number of pumps* | 2 centrifugal | 2 subsea + 2 onshore multistage centrifugal liquid pump | none |
| Type and number of coolers* | 2 anti-surge + 2 after passive coolers | 2 anti-surge passive coolers | 2 passive after coolers |
| Type and number of separators* | 2 two-phase vertical separator-scrubbers | 2 two-phase vertical separator-scrubbers | none |
| VSD and Power distribution | Topside VSD Åsgard A platform | Subsea VSD Nyhamna (onshore) | Subsea VSD Gulfaks C platform |

* Per compression train

In this work, focus is put on the first and classical type of developments for subsea gas compression. In Figure 3.1 it is shown a classical PFD of for a SGCS. The well stream (gas, condensate, water, and other production fluids) are routed into the gas/liquid separator where the phases are separated. The gas volume fraction (GVF) entering into the compressor is dependent on the efficiency of the separator. The gas is compressed while the liquid pressure is boosted by the pump. Subsequently the compressed gas and the boosted liquid are transported together through the transport pipeline to the host facility, onshore or offshore. A typical anti-surge control loop is included in the PFD.

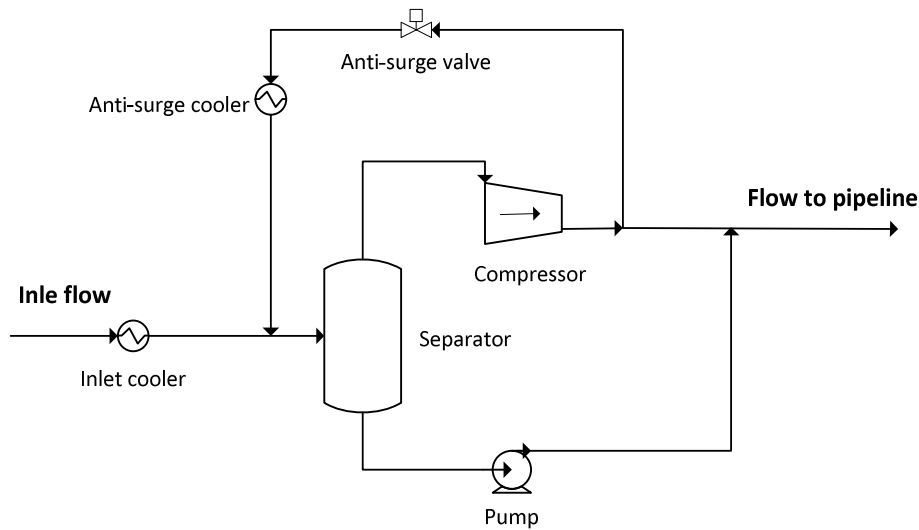


Figure 3.2 Typical PFD for subsea gas compression

The main components of the SGCS are: wet gas or gas compressor, liquid/gas separator, pumps and coolers. Additionally, electric components such as VSD and voltage transformers are of high importance. Following, it will be shortly described the key features of main components in the PFD and some of the existing prototypes.

Compressor

The compressor is the principal components of the SGCS and it has to be developed in such way that offers high reliability. Intervention procedures for maintenance imply high cost and production lost due to shut-down therefore it is intended to design a more compact and simpler compressor than topside. It is intended to discard non-essential components and some of the developments compromise active magnetic bearing and motor integrated compressor. These new configurations area able to decrease the Mean Time Between Failures (Baggerud, Sten-Halvorsen et al., 2007).

Another concern in subsea gas compression is the robustness of the compressor and its flexibility under different operational conditions. The compressor must be developed in such way that it can stand wet gas if the upstream separation is not very efficient. Therefore, there is a strong interaction between the upstream separation efficiency and the compressor performance.

It is also necessary to take into account the weight and size of the compressor in the development phase in order to reduce the environmental foot print.

The Åsgard compressor is based on the motor integrated compressor philosophy where the compressor and motor are hermetically sealed in the same casing. The integrated motor compressor concept is explained more in detail in section 0. The Ormen Lange compressor is based on the same concept but with the difference that the arrangement is vertical, reducing the footprint, and a sea water cooler is used to cool down the motor cooling gas.

As mentioned before, the Gullfaks project applies a more novel technology where the well stream is not processed before compression. The Wet Gas Compressor (WGC) is based on a counter-rotating principle and the mechanical design is divided in three main parts: upper electric motor, compressor section and lower electric motor (Hjelmeland, Olsen et al., 2011).

In Figure 3.3 and Figure 3.4 is shown the compressor prototype for the Ormen Lange and Gullfaks project, respectively. For a full revision of the different subsea compressor prototypes please refer to (Baggerud, Sten-Halvorsen et al., 2007; Brenne, Bjorge et al., 2008).



Figure 3.3 Ormen Lange: Integrated motor compressor casing (Skofteland, Hilditch et al., 2009)



Figure 3.4 Gullfaks: Wet gas compressor prototype WGC4000 (Hjelmeland, Olsen et al., 2011)

Separator

As mentioned before the separation efficiency upstream of the compressor is extremely important for the performance of the compressor and of the system as a whole. The separator must be designed to provide a high a constant GVF to the compressor. This enhances the compressor performance and avoids rapid deterioration of the compressor impellers and other components. There have been different developments of subsea gas/liquid separator; from traditional vertical gravity vessels until compact inline cyclonic separators.

A typical subsea separator for sub gas compression applications is shown in the Figure 3.5. This separator is the one developed for the Åsgard project and it works as a bulk separator and scrubber at the same time. Also, it is able to handle sands and slugs; slug buffer volume is 12.5 m³. This type of separator is based on a Statoil patented spinlet inlet arrangement and axial flow cyclones. The flow is distributed in such way that liquid is guided downwards on the separator wall while the gas is routed upwards in the center of the separator. This special design is very robust and efficient.

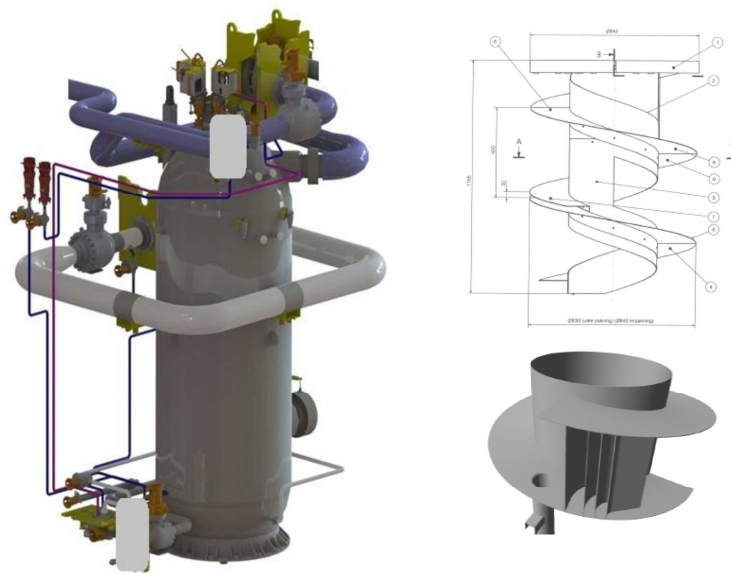


Figure 3.5 Typical subsea separator and internal parts

Pump

The pump is the most mature technology of all the SGCS' components. Subsea single phase pumps, multiphase pumps and hybrid pups have been extensively used in the offshore industry and are available on the market. In principle, the required pump in the SGCS could be a single phase pump but the possibilities of gas entrainment in the liquid phase would affect its performance. Instead, multiphase pumps could be used. Again, the pump selection is dependent of the separation efficiency upstream.

The majority of the available pumps are based in two different principles: 1) positive displacement and, 2) rotodynamic pumps. Twin-screw pumps belong to the first category while helicon-axial and centrifugal pumps belong to the second. The Table 3.2 summarizes some of the main features of these pumps:

Table 3.2 Subsea boosting pumps characteristics

| Helico-axial pumps | Twin-screw pumps |
|-------------------------------------|---|
| Limited ΔP at high GVF | High ΔP at high GVF |
| Robust for solid materials handling | Highly affected for solid materials |
| Extensively used in the industry | Not totally proven in subsea applications |
| | Potential leak through the screws |

Coolers

In a SGCS there are several alternatives of where to position the subsea coolers, e.g, inlet cooler, anti-surge cooler, outlet cooler (not shown in Figure 3.2). The number of coolers is specific for every system configuration and problem solution. For instance the anti-surge loop could be route upstream of the inlet cooler. This option eliminates the necessity of the anti-surge cooler. The outlet cooler might be also included to eliminate the potential of pipeline deterioration due to thermal stress.

In a subsea environment is natural to use sea water as the cooling medium and avoid using other process utilities which might complicate the system configuration. The decision to be made then is whether to use active coolers or passive coolers based.

In passive cooling, the pipes are in direct contact with the sea water and the heat exchange is based on purely natural convection. It offers high reliability due to no active components (e.g., sea water pumps) are included but it has limitations in terms of controllability. Also, the thickness and material of the wall pipes constraint the heat exchange.

Active coolers are based on the shell and tube concept and it can be controlled by means of the sea water pumped into it. The inclusion of active pumps brings reliability, power distribution and configuration challenges (Lima, Storstenvik et al., 2011). The final selection is subjected to the overall system integration.

3.2 Integrated motor compressor

The integrated motor compressor is now an available technology which can simplify enormously the compressor configuration as well to increase the process safety. The first unit of this type was commissioned in 1991 and has proven to be both economical and reliable (Gilon and Boutriau, 1998). Since then, several prototypes have been commercialized and current developments combine high-speed electrical motors and active magnetic bearings.

In the new developments, traditional oil bearings can be replaced by active magnetic bearings while the high speed motor eliminates the gear needed to run the centrifugal compressor at optimum speed. It is eliminated the need for cooling utilities to cool the compressor motor due to part of the process gas is used as cooling gas. Additionally, the compressor and the electric motor are placed in the same case and therefore dry gas seal are not required any longer (Alban, Pellerin et al., 2009).

For its simplicity and high reliability, these types of high-speed compressors and configurations are very suitable for subsea systems where high reliability and zero environmental impact are demanded.

In the Figure 3.6 it is presented a typical integrated motor compressor. A proportion of the process gas is taken at the compressor outlet, used to cool down the electric motor and the magnetic bearings, and routed back to the compressor inlet. This configuration is very efficient energy wise due to the high cooling flow available, the high pressure of the cooling flow and the high heat capacity in case of natural gas (Gilon and Boutriau, 1998). In the case of multistage compression, the cooling gas can be taken also from the first stage outlet. The amount of cooling gas is dependent of the process conditions and it could vary from 5 to 15% of design flow (Alban, Pellerin et al., 2009).

The literature in the area is extent and for further information the reader is suggested to consult: (Gilon, 1991; Gilon and Boutriau, 1998; Kleynhans, Pfrehm et al., 2005; Beer, Tessaro et al., 2006; Alban, Pellerin et al., 2009)

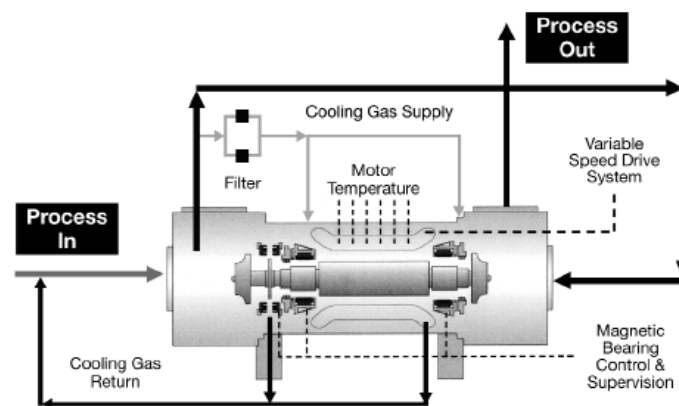


Figure 3.6 Typical integrated motor compressor (Kleynhans, Pfrehm et al., 2005)

3.3 Control system and power supply. VSD.

In long step-out subsea developments the power distribution is one of the major challenges hence the controllability of the rotating machines such as multiphase pumps and compressors. The implementation of VSD in subsea compressor systems is a technology challenge and the industry is joining efforts to overcome the technology gap. Traditionally, and it is the case for the Åsgard SGCS, the VSD have been placed topside from where the boosting or production pumps are driven. However, in the case of the Ormen Lange SGCS the VSD will be placed subsea. This subsea VSD are the first in its class and should be able to run the high speed

motor compressor and pump motor with high levels of reliability and availability in order to minimize intervention or stop production within at least five years (Bjerkreim, Haram et al., 2007).

4 HYSYS Dynamics

This chapter has the intention to introduce the main features of HYSYS dynamics and all the considerations needed when developing a dynamic model. It also introduces the most important available functionalities in HYSYS dynamics that were used in this work. Special focus will be put in the Automation functionality of HYSYS which was extensively used in this thesis.

The information presented in this chapter comes mainly from the HYSYS user guides (Aspen Technology, 2011; Dissinger, Levine et al., 2012)

4.1 Conservation relationships

The mathematical modeling in HYSYS Dynamics is based on conservation relationships for mass, components and energy. In steady-state none of these quantities change in time but in dynamic-state it is necessary to include an accumulation term in the balance that takes into account changes in time.

A simple mass, component and energy balance can be represented by Equations 4.1, 4.2 and 4.3. The specific equations depend on each individual equipment and the method selected to perform the calculations.

Mass balance

$$\text{Rate of accumulation of mass} = \text{mass flow into the system} - \text{mass flow out of the system} \quad 4.1$$

The rigorous equations in HYSYS dynamics can take into account phenomena effects such as vaporization, chemical reactions, density changes, etc. In HYSYS the mass balance is also referred as pressure-flow balance.

Component balance

$$\text{Rate of accumulation of component } j = \text{Flow of component } j \text{ into the system} - \text{Flow of component } j \text{ out of the system} + \text{Rate of formation of component } j \text{ by reaction} \quad 4.2$$

Energy balance

$$\text{Rate of accumulation of total energy} = \text{Flow of total energy into the system} - \text{Flow of total energy out of the system} + \text{Heat added to the system across its boundaries} + \text{Heat generated by reaction} - \text{Work done by the system on surroundings} \quad 4.3$$

The energy flow coming into the system and going out of the system can be due to convection or conduction while the heat added to the system across its boundary is by conduction or radiation (Aspen Technology, 2011).

4.2 Calculation method

HYSYS uses a lumped model, this is that all the physical properties are considered to be equal in space so there are not thermal or component concentration gradients in the coordinates (x, y, z). Only time gradients are considered in the lumped model. In this way, the system can be represented and solved as a set of Ordinary Differential Equations (ODEs).

The solution obtained in the lumped model is an approximation of the actual solution which could be found by considering the space gradients. This kind of model is so-called Distributed model and it can be represented by a set of Partial Differential Equations (PDEs). The advantage of the lumped model is that it is much easier to solve than the distributed system while it offers, depending of the process to model, acceptable results.

The conservation equations can be expressed by non-linear ODEs however an analytical solution is rather extremely complicated; therefore numerical integration is used to solve them. The solution reached by the integration method is more closely to the analytical solution as the time step is smaller. However, a small time step requires additional calculation time for the same elapsed real time thus it is important to balance the accuracy with the speed of the calculation.

The ODEs of the lumped model in HYSYS Dynamic are solved by the implicit Euler Method which uses rectangular integration to compute the value of Y_{n+1} , see Equation 4.4 (Aspen Technology, 2011):

$$Y_{n+1} = Y_n + hf_{n+1}(Y_{n+1}) \quad 4.4$$

where:

Y, property Y

n, current time step

h, step size

The three conservation balances are not computed at the same time step since it is time consuming. Instead, they are solved at different time step frequencies. In HYSYS Dynamics the mass, energy and component balance are solved every first, second and 10th time step respectively. In general, when there is not chemical reaction involved, composition tends to change more gradually than other properties so it can be solved less frequently. These values can be modified by the user.

4.3 Equipment: Sizing and specifications

An important issue in dynamic simulation is that pressure and flow are directly related and one cannot be calculated without the other. It must be a pressure gradient across every operation unit so the flow through it can be calculated. All the unit operations can be defined

as pressure node or resistance equation operation. Pressure node operations compute a pressure drop taking into account the vapor hold up in the unit, e.g. vessels. On the other hand, the resistance equation operations calculate a pressure drop based on a given resistance equation, e.g. valves, pumps, compressors, etc. (Dissinger, Levine et al., 2012).

In contrast to steady-state, in the dynamic mode it is necessary to specify rating details of each operation unit in the process flow sheet so it can match the actual equipment performance. The information needed usually involved the equipment size, geometry, relative position with respect to the ground (static head), nozzle placement, etc. Once the rating information of each equipment have been specified then HYSYS uses an advanced Holdup model to calculate heat losses, product compositions, separator liquid levels, static head contributions, etc.

In the following, it will be discussed the main rating parameters needed for each operation unit. Special focus is put on the operation units which form part of the developed wet gas compression model. Units such as distillation columns, expanders, pumps, etc., are not discussed in this section. For more information the reader is advised to consult the Unit Operations and Dynamic Guide provided by (Aspen Technology, 2011).

Valves and actuators

In Dynamic-state a valve can be specified by either a fixed pressure drop or a pressure flow relation. The pressure-flow relation is recommended since it is more realistic than a fixed pressure drop.

If a fixed pressure drop is selected on the valve one pressure and one flow specification are needed at the inlet stream of the valve. Instead, if the option of pressure-flow is preferred then two pressures are specified or one pressure and one flow. In the pressure-flow tab, it is possible to specify the flow coefficient (C_v or C_g) of the valve or a simple resistance equation.

In the case, that the C_v and C_g method is chosen as pressure-flow specification, HYSYS offers the possibility to size the valves at the given conditions by using the Fisher Universal Sizing Method. This method can predict accurately the flow for either high or low recovery valves, for any gas and for any service condition (Aspen Technology, 2011):

$$Q_{vap} = v_{fracfac} \cdot C_g \cdot \sqrt{\rho \cdot p_1} \cdot \sin \left(\frac{59.64}{C_1} \sqrt{1 - \frac{p_2}{p_1} \cdot cp_{fac}} \right) \quad 4.5$$

$$Q_{liq} = (1 - v_{fracfac}) \cdot 63.338 \cdot C_v \cdot \sqrt{\rho \cdot (p_1 - p_2)} \quad 4.6$$

$$C_1 = \frac{C_g}{C_v} \quad 4.7$$

$$Km = 0.001434C_1 \quad 4.8$$

$$Cp_{fac} = \frac{0.4839}{\sqrt{1 - \left(\frac{2}{1 + \gamma}\right)^{\frac{\gamma}{\gamma-1}}}} \quad 4.9$$

$$\gamma = \frac{C_p}{C_v} \quad 4.10$$

where:

C_v Liquid flow sizing coefficient

C_g Gas flow sizing coefficient

p_1 = inlet stream pressure

p_2 = outlet stream pressure

$v_{fracfac} = 1$ if outlet molar vapor fraction > 0.1

0 if outlet molar vapor fraction = 0

molar vapor fraction/0.1, otherwise

In the case of control valves it is important to specify the operating characteristics of the valve. This is the relation between the actual valve position (% Valve Opening) and the corresponding flow capacity (% C_v). This selection will dictate the opening and closing performance of the valve. In HYSYS dynamics there are four different possibilities to select (Aspen Technology, 2011):

- Linear: The flow through the valve is directly proportional to the % of valve opening.

$$\%C_v = \% \text{ Valve opening} \quad 4.11$$

- Quick opening: This type of control valves will allow high flow rates at low openings while, as the valve opens further the rate at which the flow increases is smaller.

$$\%C_v = 100 \cdot \left(\frac{\% \text{ Valve Opening}}{100}\right)^{0.5} \quad 4.12$$

- Equal percentage: In this type, the flow through the valve is small at lower opening percentages while the flow increases rapidly as the valve opens. This type of valve acts in the opposite way as the quick opening valve.

$$\%C_v = 100 \cdot \left(\frac{\% \text{ Valve Opening}}{100}\right)^3 \quad 4.13$$

- User table: The fourth method allows the user to input a table with the valve position and the associated percentage flow capacity.

A comparison between the three first control valve flow characteristics is shown in the Figure 4.1.

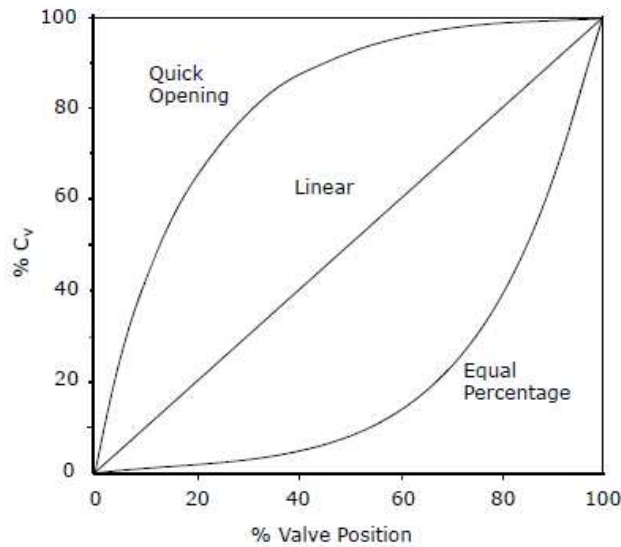


Figure 4.1 Control valve characteristics (Aspen Technology, 2011)

It is also important to specify the valve mode which defines the relationship between the desired actuator position and the current actuator position. For example, the desired actuator position can be set by and PID controller or from a spreadsheet calculation. In the actuator page of the valve dynamic tab it is possible to select between three modes: Instantaneous, first order and linear.

In the instantaneous mode the actuator moves instantaneously to the desired position set by the PID controller. The relationship between the current and desired actuator position can be expressed as:

$$Act \% = Act_{Desired}\% \quad 4.14$$

Compressors

In dynamic-state as well as in steady-state off-design operation it is necessary to input the compressor characteristic curves. Likewise, in dynamic-state it is possible to input more complete information about the compressor:

- Surge and stonewall curve
- Impeller and rotating shaft inertia information.
- Friction loss term
- Electric motor speed vs. torque curve
- Electric motor inertia and friction factor

In the dynamic specifications tab only two dynamic specifications shall be selected. The characteristic curve is advised to be always used hence it remains only one free specification; among them we found the compressor power, compressor speed or total power loss, etc. The selection of the second specification depends on the configuration of the control system.

The pressure-flow specifications at the compressor surrounding should be two pressures specification for the inlet and outlet stream or one pressure and flow specification.

Mixers/Tees

The dynamic performance of the mixer and tee operation units is very similar. In the mixer dynamic tab it is possible to either equalize the pressure of all the inlet streams or to set the outlet pressure as the lowest inlet stream pressure as dynamic specification. It is recommended to choose the first option due to it is more realistic. In this way, the pressures of all the surrounding streams are equal if static head contributions are not considered.

In the case of the tee, it is possible to select the split ratios as dynamic specification, however this is not advised. Instead, the flow ratios should be calculated from the tee surrounding pressures and all resistances through the flow sheet. If a given flow ratio has to be active controlled it is possible to add and PID controller which operates the internal flow ratio valve inside the tee.

Cooler/Heater

In dynamic-state the cooler/heater can be modeled with either a fixed pressure drop or pressure-flow equation. Similarly to the valve operation, a fixed pressure drop in the cooler/heater is not recommended since this is unlikely in an actual process. Hence, it is recommended to use a pressure-flow equation which related the pressure drop with the flow across the cooler/heater see Equation 4.15 . The overall k-value can be calculated in steady-state at the design conditions and the used as dynamic specification. The Equation 4.15 does not consider the possible static head contributions.

$$flow = \sqrt{density} \cdot k \cdot \sqrt{p_1 - p_2} \quad 4.15$$

Additionally, the duty applied to the holdup fluid inside the cooler/heater has to be specified. Three methods are available:

- Supplied duty. If selected, the duty supplied to the heater (or extracted from the cooler) has to be specified. Normally, this option is selected when using a PID controller to calculate the duty based on a process variable specification.
- Product temperature specification. In this case the temperature of the product is specified and HYSYS computes the corresponding duty to achieve the specified temperature.
- Duty fluid. This option is used when the user has knowledge about the utility used as heating/cooling media. In this case, information about the utility is required as input.

4.4 Controllers

In HYSYS dynamics it is possible to select between five different control schemes: split range controller, ratio controller, PID controller MPC controller and DMCplus controller. Among all these controllers, the PID controller is the most common used.

The PID controller algorithm (see Equation 2.44) can be solved in HYSYS by two different sub-algorithms: PID velocity form and PID positional form.

In the PID velocity form the Equation 2.44 can be written as (Aspen Technology, 2011):

$$u(t) = u(t - 1) + K_c \left[e(t) - e(t - 1) + \frac{1}{T_i} \right] e(t)h + T_d \frac{(e(t) - 2e(t - 1) + e(t - 2))}{h} \quad 4.16$$

where:

$u(t)$, controller output at time t

h , sampling period

The PID positional form can be expressed as (Aspen Technology, 2011):

$$u(t) = K_c \left[e(t) + \frac{1}{T_i} \sum_{i=1}^n e(i)h + T_d \frac{(e(t) - e(t - 1))}{h} \right] \quad 4.17$$

In HYSYS, the velocity form is recommended when there is an integral term and the positional form when there is no integral term and. This is due how HYSYS solves the positional form of the integral term which is summed up continuously implying a continuous summation of errors, so-called integral wind-up. The effect of integral wind-up makes the system to oscillate more (Hansen, 2008). In HYSYS, the velocity form is set as default subtype algorithm.

The process of tuning the PID parameters can very comprehensive and it is out of the scope of this work to intend any optimization around them. The consulted material of HYSYS dynamics recommends the following values for the PID parameters:

Table 4.1 Recommended PID tuning parameters (Dissinger, Levine et al., 2012)

| System | K_c | T_i (minutes) | T_d (minutes) |
|-------------|-------|-----------------|-----------------|
| Flow | 0.1 | 0.2 | 0 |
| Level | 2 | 10 | 0 |
| Pressure | 2 | 2 | 0 |
| Temperature | 1 | 20 | 0 |

4.5 Dynamic tools

HYSYS-dynamics has several powerful tools that make easier to create, modify or improve a dynamic model. In this section three important tools will be described: dynamic assistant, integrator and event scheduler.

Dynamic assistant



The HYSYS dynamics assistant is a very useful tool when changing from steady-state to dynamic-state, or when building a new dynamic case. It makes suggestions about the required system boundary conditions as well the pressure-flow specification in the unit operations. It has to be noticed that despite the practicality of the dynamic assistant it is just a guide and not its suggestions are always applicable. It is important to use own engineering judgment.


The dynamic assistant also checks the degrees of freedom of the system and suggests which variables need to be added or dismissed so the system is fully specified before starting the simulation. It has also the capability to add valves where are necessary.

In the case of existing un-sized equipment like valves, heat exchangers or vessels, the dynamic assistant also offers the possibility to size them using the corresponding method for each unit. Therefore, it is important that when changing from steady-state the model is at the design or normal operating conditions. The performance of units such as valves, vessels, etc. will depend on the sizing parameters calculated in the steady-state.

The control scheme of the process cannot be created through the dynamic assistant and it depends only on the expertise and knowledge of the user.

Integrator

Firstly, the integrator serves to activate  and deactivate  the simulation case. Different from the steady-state solver the dynamic integrator cannot be activated before the simulation flow sheet is complete and the degrees of freedom rule is satisfied.

It is also possible to access to the integrator properties and the important information related to the numerical scheme of the dynamic simulation. From the integrator general tab it is possible to select automatic or manual integration. In the manual integration the user has to specify the number of time steps while in the automatic integration the simulation runs until the user turns it off, . The manual integration is often used when debugging or trouble shooting a case.

The integration time step is set to 0.5 seconds by default. This value can be reduced to improve the accuracy of the result however this will lead to higher computation time, and vice versa. In general, the default value represents a good trade-off between accuracy and computation time.

Additionally, the execution tab of the integrator allows manipulating the execution rates for the conservation relationships discussed in the section 4.1. The execution rate for the logical operations (controllers, transfer function, etc.) block can be also manipulated.

Other options such as enabling the static head contributions, modeling choking of liquid inside valves, etc. are also possible to select.

Event scheduler

The event scheduler is a powerful tool in HYSYS dynamics that allows the user to automatically execute specific tasks once the integrator has been started. The event scheduler can be used to simulate special case scenarios as shut-down, start-up and system disturbances.

The pre-programmed tasks can be started by pre-determined conditions. These could be a specific elapsed time or simulation time, evaluation of logical expression or stabilization of a certain variable within a set tolerance during a certain amount of time.

The logic behind programming each event is shown in the Figure 4.2. Each Schedule compromises a set of sequences which are composed of Events. At the same time, an Event is formed of Conditions with a subsequent set of Actions.

Each individual schedule should be managed as a separate independent set of sequences. Two or more sequences cannot interact when they belong to different schedules. For each sequence, the pre-determined conditions are evaluated and when they are met the action list, within the event, is executed. Each sequence, in the schedule, is executed one per time step as they appear.

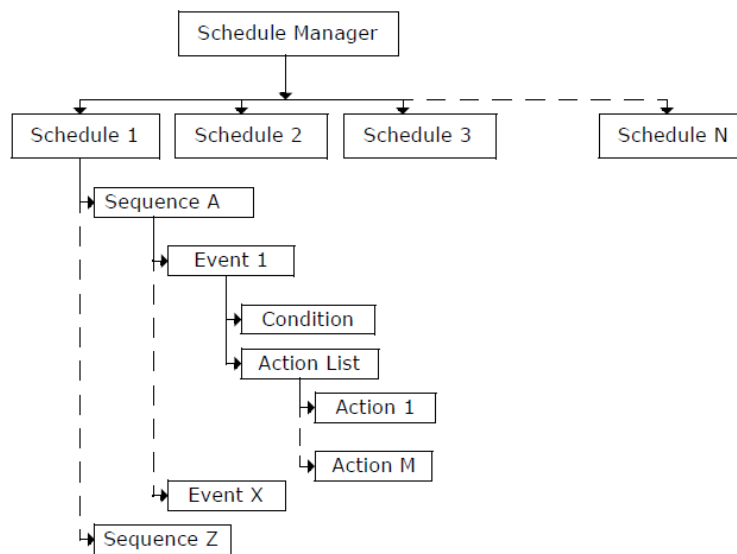


Figure 4.2 Event Scheduler programming logic

It is advised to develop as many steps as possible within a sequence rather to stop and start new sequences. In the Chapter 6, it will be shown how the event scheduler was used to set up different cases of study.

4.6 HYSYS Automation

Automation is maybe the functionality of more importance for the developing of this thesis and the presented dynamic model. Automation makes possible that HYSYS is accessed and manipulated from third party software in a server-client framework. The server provides a service that can be used by the client if knowing the right protocols. This means that the user

is able to access and exchange all the information in the HYSYS flow sheet from tools like Visual Basic, Visual Basic for Application (VBA) in excel, and other programming languages such as MATLAB and C++. HYSYS also has its own Macro Language Editor which allows developing scripts inside the simulation environment.

In this sense, by combining HYSYS with other tools makes possible to perform tasks that are not trivial in HYSYS or to focus in particular flow sheet information. In the subsequent, it will be tried to explain briefly the main characteristics of the Automation concept. For more information regarding this subject please refer to the Aspen HYSYS Customization Guide (Aspen Technology, 2011).

Automation is based on the object hierarchy concept. An object contains a set of functions, so-called methods, and variables, so-called properties that can be accessed from the third party software. In the Figure 4.3 it is possible to appreciate better this concept. In this example, the object is the car and it has a set of properties and methods. The properties, color, make, engine, etc. are variables that have an associated value. The methods, drive, refuel, etc. are functions or subroutines that define the performance of the object. An object could contain one or more sub-objects with their own properties and methods.

The method to call or get a specific object property or method is through the object hierarchy system. In VB and VBA the sub-objects, properties and methods are accessed using the dot (.) operator. Using Automation it is possible to access to over 340 objects and over 5000 combined properties and methods in HYSYS.

In the car example, defining the color (property) of the car (object) would be:

Car.colour = blue

In the same way, to define the size of the car engine would be:

Car.engine.size = 6

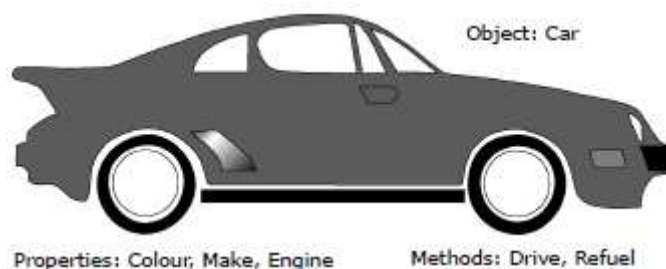


Figure 4.3 Object hierarchy example in Automation (Aspen Technology, 2011)

The syntax used in Automation is similar to the one used in VB and VBA. The Table 4.2 summarizes the main syntax and their functionalities.

Table 4.2 Main syntax in HYSYS Automation

| Syntax | Function |
|--------|---|
| Dim | Used to declare objects and variables |
| Get | Used to start communication with an existing object |
| Create | Used to create an object |
| Set | Used reference to an object variable (sub-object, property or method) |
| Let | Used to modify the value of a property or method |
| Item | Used to access to a specific object within a collection of objects |

Example: Accessing to an open HYSYS case

```
Set hyApp = CreateObject("HYSYS.Application")
```

```
Set hyCase = hyApp.ActiveDocument
```

Example: Accessing to a stream in the previous flow sheet

```
Set hyFlowsheet = hyCase.Flowsheet
```

```
Set hyStream = hyFlowsheet.MaterialStreams.Item("well_stream")
```

Example: Changing the temperature of the "well_stream"

```
Let hyStream.TemperatureValue = 300
```

By default HYSYS used the SI units to all the calculations and values exchanged through Automation. This has to be taken into account when developing any code. The automation functionality will be exposed again in the Section 5.5.

In this chapter, it was described the main aspects and characteristics of HYSYS dynamics and the main tools that are important for development of the wet gas compression model proposed in this thesis. In the next chapter, it will be explained how the different functionalities of HYSYS were used for the creation of the model.

5 The dynamic model

In this chapter it will be explained in detail how the dynamic model was developed from the steady-state and its main features such as boundary conditions, equipment specifications and system controllability.

A key goal of this thesis is to implement within the dynamic model correction methods that account for wet gas compressor performance. The optimal implementation through HYSYS Automation of two different correction methods is also discussed in this chapter.

5.1 Model background

In this section it will be briefly presented the steady-state model developed last semester during the project thesis (Perez, 2012). The steady-state model is the starting point for the dynamic model so it is important to have a clear idea of how it was developed from the beginning. It will be also discussed the selection of number of impellers and calculation of characteristic curves.

The HYSYS dynamic model was developed using available field data corresponding to the Åsgard field; however the configuration of the Subsea Gas Compression Station (SGCS) presented in this work differs from the actually intended in the Åsgard Subsea Gas Compression (SGC) project. The main difference is that the developed model here represents a wet gas compressor and no gas/liquid separation is performed upstream of the centrifugal compressor.

In the project thesis it was studied the pressure boosting requirements of the system (from well head to platform) and several alternatives were proposed for the designing and configuration of the compressor. It was concluded that the operating life of the SGCS would be divided in two consecutive cases: constant gas production and constant compressor power (Perez, 2012).

In the first case, the production of the field is maintained constant while the reservoir, so the well head pressure, declines. In this case, the compressor power increases gradually with the well head pressure reduction until it reaches the assumed maximum design value of 10 MW (the last revision of Åsgard project facts shows that the maximum design compressor power is 11.5 MW, (AkerSolutions, 2013)). This study permitted to create the total compressor power and production profile for the well head pressure decline, see Figure 5.1 and Figure 5.2. In the given profiles, the first operation scenario is represented with a blue line.

From this first study it was also possible to estimate the number of impellers for the centrifugal compressor. The total number of impellers was calculated to be two with a maximum polytropic head of 3000 m per impeller. The assumed maximum impeller head is in accordance with the typical value for estimating purposes (Lapina, 1982).

In the second case, after the maximum design compressor power is reached, the plateau gas production cannot be further maintained and it is necessary increment the boosting capacity of

the compressor. At this point of the field operation life, it is proposed to make a rewind of the compressor to change the existing two impellers and add four more for a final total of six impellers. In this scenario the compressor speed is adjusted to keep the compressor power at its maximum design value of 10 MW, while the well head pressure declines and the head requirement maintains increasing. In HYSYS the available gas production is calculated when specifying the well head pressure, the maximum compressor power and the required platform inlet pressure.

In this case, the available gas production decreases gradually with the well head pressure reduction until it reaches a minimum value of 40% of the plateau production. At this this stage the operation the SGCS is considered to be not profitable anymore, meaning the end of the life of the field. In the Figure 5.1 and Figure 5.2 the case for constant compressor power is represented with a red line.

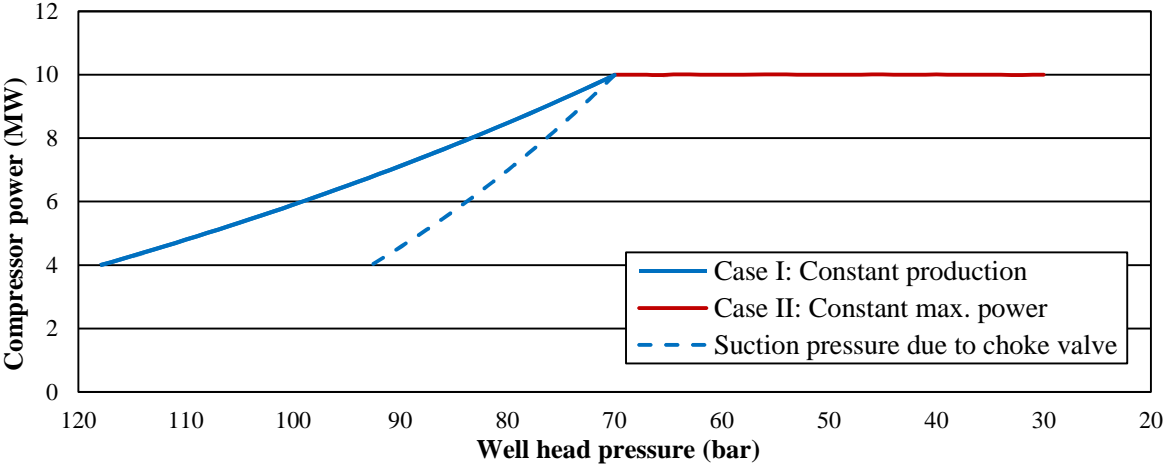


Figure 5.1 Steady-state compressor power profile

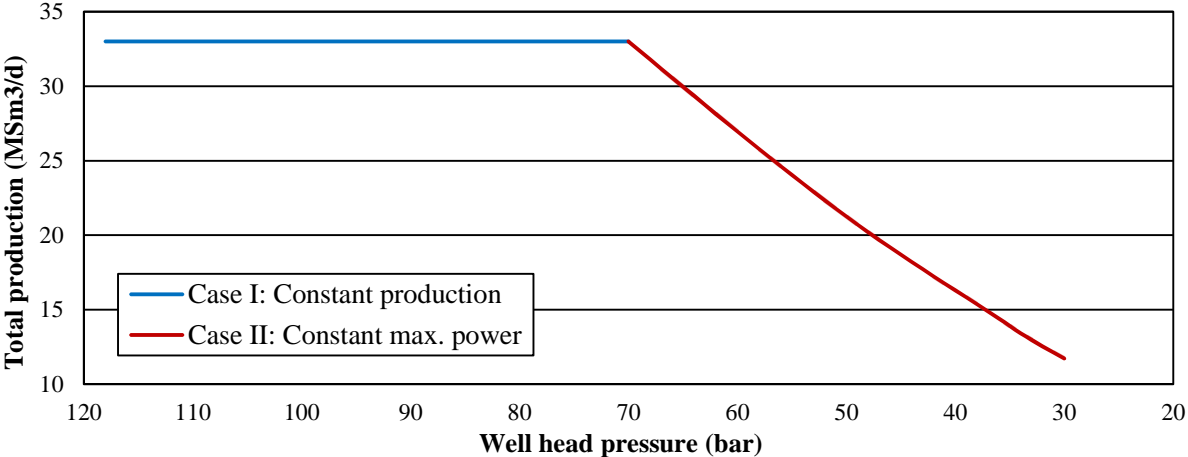


Figure 5.2. Steady-state production profile

The profiles shown above can be also presented in terms of polytropic head (m) versus actual inlet volumetric flow (m³/s). The resulting head requirement profile is then used to select the design point for each impeller and based on experimental curves extrapolate the typical

performance and generate the characteristic curve for each impeller. The Figure 5.3 shows the characteristic curves (constant production case) of the first and second impeller at the maximum design speed of 10 000 rpm, N =100%.

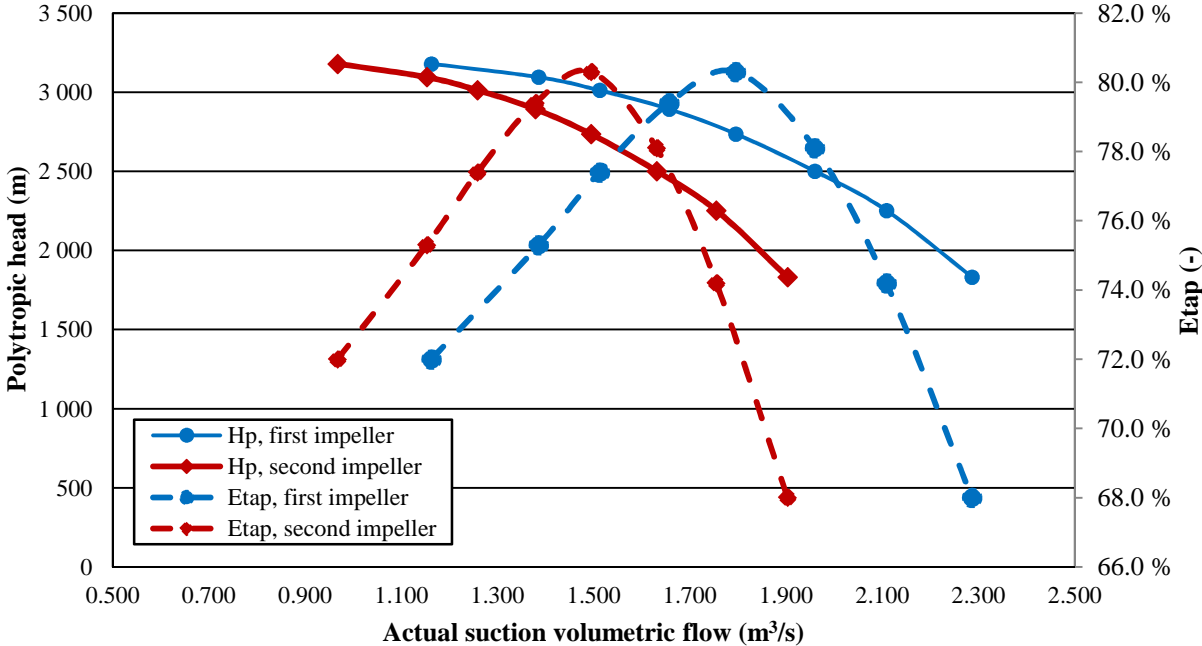


Figure 5.3 Impellers characteristic curves, case I

Once the characteristic curve for each impeller has been determined it is possible to use HYSYS to explore the operability of the SGCS both in steady and dynamic-state. In the project thesis (Perez, 2012) the steady-state operation was widely discussed while in the current work the main focus is on the dynamic mode. The next sections will address the developing of the dynamic model and the transition from the steady-state.

5.2 From steady to dynamic-state

First of all, it is extremely important to point out that the developed dynamic model corresponds to the first case exposed in the steady-state study, constant gas production, Case I. Thus, in the dynamic model, a fixed gas flow rate is sent from the compression station to the pipeline (see boundary conditions, Section 5.3.1).

Also, as it was mentioned before the developed model does not pretend to be an exact representation of the Åsgard SGCS but what is intended is to study the main fundamentals of these types of systems. Both the steady-state and dynamic model are based on the concept and include the main processing equipment described in the Chapter 3.

In order to include the motor and bearing cooling flows it is necessary to model the entire centrifugal compressor as two individual impellers. The cooling flow is extracted after the first impeller and sent to the electric motor and magnetic bearings. Then, the warm cooling flow is sent back to the inlet of the first impeller where it is pre-mixed with the coming compression flow. The magnetic bearings and electric motor cooling demand were modeled as a single heater where the cooling flow is heated up according the cooling demand.

In HYSYS, each impeller was modeled as an individual compressor and their characteristic curves were set into two separate HYSYS-spreadsheets. HYSYS has the capability to export a given variable value from the spreadsheet to a selected unit in the PFD environment. This functionality was used to export the individual characteristic curves from the spreadsheet to the corresponding compressor unit.

The Figure 5.4 and Figure 5.5 show the steady-state and dynamic-state models PFD, respectively. From the figures it can be seen the evolution from the steady-state to the dynamic model, which includes the anti-surge control loop that is of high importance during compressor dynamic simulation.

Since the main focus is on the compressor dynamic performance and how it is affected under different operating conditions including wet gas performance, it was convenient to make the following simplifications:

- Elimination of inlet cooler. Instead, the temperature of the stream SCGS inlet is set to be as it was already cooled down in the inlet cooler hence it does not represent the actual well head temperature.
- Elimination of inlet separator. This simplification plus the addition of a water/condensate stream allows manipulating directly the GVF at the inlet of the first impeller. In the actual SGCS the GVF at the inlet of the compressor is given by the separator efficiency.
- The dynamic model does not include the transport pipeline. This is maybe the main simplification of the model. The inclusion of the transport pipeline will affect the responses of the system making them slower than if it is not included. This is due to the large volume and resistance that it represents. However, this simplification can be considered acceptable since the main focus is on the compressor performance. In the steady-state study, it was determined that at constant production rate the compressor outlet pressure remains nearly constant at 120 bar under different conditions. In the dynamic model, this value was used to set up a compressor outlet pressure controller (see Section 5.3.3)

After doing these simplifications and building the flow sheet the next step was to set the stream and unit specifications and the rest of the needed information. The next section will address these aspects.

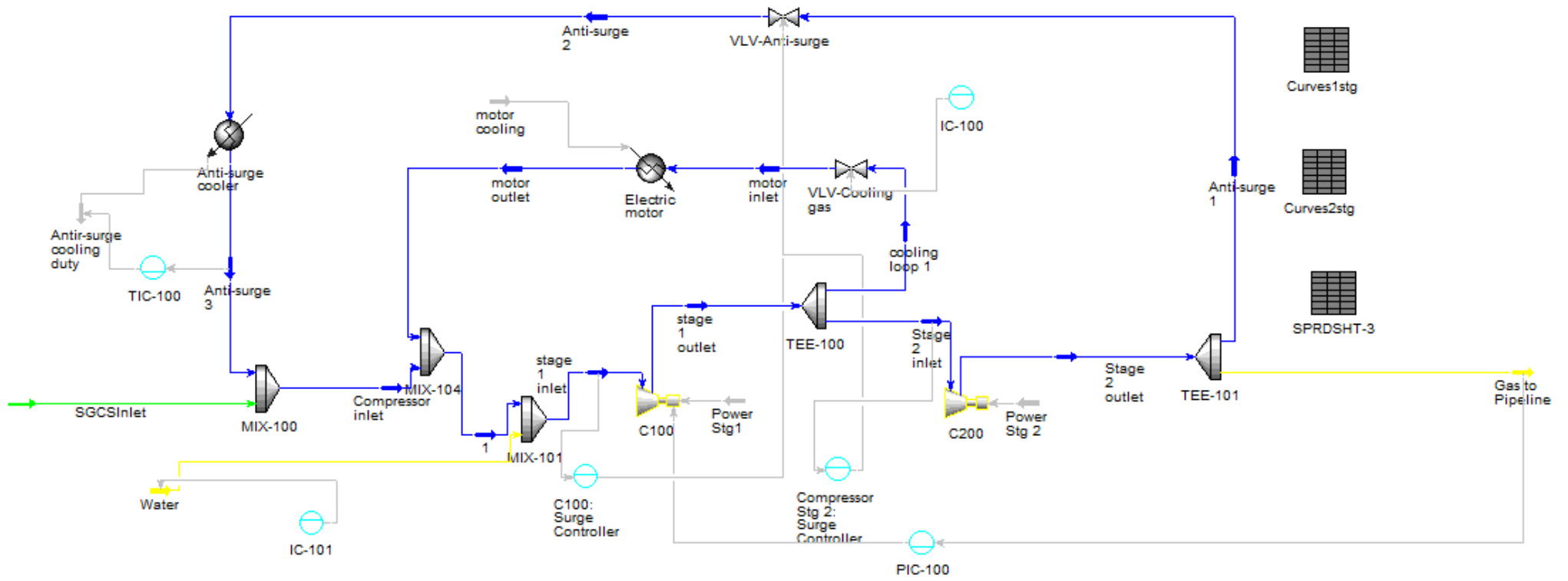


Figure 5.5. PFD: Dynamic model

5.3 Main features

In order to start the dynamic simulation, first it is necessary to complete the model by specifying the boundary conditions of the system, the pressure-flow specification of the equipment and include the process control scheme. In this section, all these issues will be addressed.

5.3.1 Boundary conditions

In HYSYS dynamics the boundary conditions of the system are fully specified when specifying one pressure or flow specification for each inlet and outlet stream. In our model, there are two inlet streams and one outlet stream therefore three boundary conditions are needed. Additionally, the temperature has to be specified in each of the inlet streams. The boundary and temperature specifications are summarized in the Table 5.1:

Table 5.1 Boundary conditions

| Stream name | Specification | Value |
|------------------------------|---------------|--------------------------|
| SGCS Inlet ¹ | Pressure | 90 bara |
| Water ² | Molar flow | 0 – 5 kmol/s |
| Gas to pipeline ³ | Molar flow | 6.76 kmol/s (13.8 MSCMD) |
| SGCS Inlet ⁴ | Temperature | 15 °C |
| Water | Temperature | 15 °C |

¹ The SGCS inlet pressure (equal to the compressor suction pressure) can be varied to study the performance of the compressor and the system response.

² The water flow rate is used to control the GVF at the inlet of the compressor (first impeller inlet). In a specific case of study, this stream is also used to represent a condensate stream.

³ The flow specification is the molar flow rate fixed at 6.76 kmol/s. The mass flow rate will depend on the molecular weight of the resulting mixture at wet gas conditions (gas and water/condensate)

⁴ The SGCS inlet temperature represents the discharge inlet cooler temperature, 15 °C.

5.3.2 Unit operations sizing

As it was exposed in the Section 4.3 it is necessary to make the sizing of the equipment such as control valves, heat exchangers and other unit operations in order to make a correct dynamic modeling and control of the process. In the following, it will be shown the sizing criteria and parameters used for each unit in the flow sheet.

Valves and Cooler/Heater

The valve flow coefficient (Cv) and the overall k-values of the heat exchangers were calculated taking as reference the steady-state conditions at a compressor inlet pressure of 90 bara. The sizing parameters used as dynamic pressure-flow specifications are summarized in the Table 5.2.

Table 5.2. Sizing conditions and parameters

| Unit | Sizing conditions | Sizing parameter | Value |
|--------------------|--|------------------|--|
| Anti – surge valve | 100% reflux when fully open | Cv | 400 USGPM (60 F, 1 psi) |
| Cooling gas valve | 5% of total flow when fully open | Cv | 50 USGPM (60 F, 1 psi) |
| Anti-surge cooler | 100% reflux and $\Delta p = 5$ bara | Overall k | 74 [kg/hr/sqrt(kPa·kg/m ³)] |
| Electric motor | 5% of total flow and $\Delta p = 5$ bara | Overall k | 157.4 [kg/hr/sqrt(kPa·kg/m ³)] |

Additionally, the mode and operating characteristics of each valve are summarized in the Table 5.3.

Table 5.3 Operating characteristics of valves

| Valve | Operating characteristics | Mode |
|--------------------|---------------------------|---------------|
| Anti – surge valve | Linear | Instantaneous |
| Cooling gas valve | Linear | Instantaneous |

In order to model the cooling of the electric motor and the magnetic bearings a heater was introduced to simulate the heat absorbed by the cooling gas flow. From a consulted reference (Crowley, Bansal et al., 2003) it was possible to find a direct experimental relation between the motor electrical losses and the total consumed power, see Eq. 5.1. The cited reference corresponds to an electric motor driven compressor with similar design parameters as the one studied in this work. The referenced data was obtained for a compressor of 10 MW design power and 12000 rpm design speed.

$$\dot{Q}_{motor} = 23.959 \times P_c - 3.6264 \quad [\text{kW}] \quad 5.1$$

This equation was set into one spreadsheet, linked with the total compressor power and the result exported to the heater (electric motor) duty. In this way, a new electric motor duty is calculated for any change in the total compressor power.

The equation presented is only a steady-state approach for the electric motor thermal transient behavior. A complete transient thermal analysis will need complete data of the actual electric motor and the use of theoretical models of heat transfer mechanisms. Studies on this matter are presented by (Pinto, Antunes et al., 1994; Asaii, Moghtaderi et al., 1996; Rajagopal, Seetharamu et al., 1998; Arshad, Chin et al., 2001).

Compressors

As it was seen in the Section 4.3, in HYSYS dynamics, a compressor unit is completely specified when at least one compressor curve and an additional specification are input.

In the model, a full compressor map was generated by applying the fan laws to the characteristic curve found during the steady-state study, see Section 5.1. The compressor map was generated for the rotational speeds of 60, 70, 80, 90, 100 and 105% of the design speed, see Appendix A.

Additionally, the compressor speed was used as specification. In the control scheme the speed is further manipulated by the compressor outlet pressure controller.

The Schultz analysis was selected for the compressor performance method.

Mixers/Tees

Following the discussed in the Section 4.3, in all the mixers the “equalize pressures” option was selected as dynamic specification and in the tee units the split ratio specification was not enabled.

5.3.3 Controllers

A total of six controllers were added in the model flow sheet in order to handle potential disturbances in the system. The Table 5.4 summarizes the functionality and handled variables of each controller. According with the discussed in the Section 4.4, the velocity algorithm was selected for all the PI controllers.

Table 5.4. Summary of controllers

| Controller | Type | Operating Parameter (OP) | Process Variable (PV) | Set Point (SP) |
|------------------------|------|-----------------------------------|--|---------------------|
| IC-101 | PI | Water molar flow rate | GVF - First impeller inlet | Variable |
| PIC-100 | PI | Compressor rotational speed | Pressure - Gas to pipeline | 120 bar |
| IC-100 | PI | % opening – Cooling gas valve | Cooling gas ratio | 5 % |
| TIC-100 | PI | Control valve – Anti-surge cooler | Temperature – Anti-surge 3 | 15 ° C |
| C100: Surge controller | PI | % opening – Anti-surge valve | Actual volume flow – First impeller inlet | $H_p = C \cdot q^2$ |
| C200: Surge controller | PI | % opening – Anti-surge valve | Actual volume flow – Second impeller inlet | $H_p = C \cdot q^2$ |

In the case of the anti-surge controllers the set point is the so-called surge flow and in HYSYS can be calculated based on the expression:

$$H_p = A + B \cdot q + C \cdot q^2 + D \cdot q^3 \quad 5.2$$

where:

H_p = Polytropic head (m)

q = surge flow (m^3/s)

A, B, C, D = surge line parameters

The expression above is used to characterize the surge line of the corresponding compressor map. In our case, the compressor map is generated when applying the fan laws to the compressor characteristic curve at 100% rotational speed, hence the relationship between the polytropic head and the surge flow is quadratic and the expression is simplified to:

$$h_p = C \cdot q^2 \quad 5.3$$

The quadratic constant C can be calculated by simple manipulation of the fan laws equations or by polynomial regression of the surge line in the compressor map. This method implemented within HYSYS is a simplification of the Eq. 2.2 showed in the Section 2.3.

Once the constant C is specified in the anti-surge controller then HYSYS will calculate the surge flow at the given suction conditions. The surge margin is also specified in the anti-surge controller. For both controllers, the control line was set in 10% higher than the calculated surge flow. The anti-surge valve will remain close unless any of the two anti-surge controllers computes that the actual suction volume flow is lower than the corresponding control flow. It is important to note that both anti-surge controllers have as OP the same valve. In the case that both compressors reach the surge control line, HYSYS selects the lower surge flow to calculate the valve opening.

The controller IC-100 is used to control the molar flow ratio to the cooling gas loop by operating the internal valve of the tee unit, TEE-100. In normal operation conditions this valve is manipulated manually and no active control is made. However, active control was included in the simulation in order control the cooling gas flow and be able, later on, to compare the results between the different cases of study.

Finally, for the configuration of the controllers it is necessary to specify the tuning parameters. The values of these parameters were taking from the HYSYS Dynamics manual (Aspen Technology, 2011) and are shown in the Table 5.5.

It is important to notice that optimization with respect to the tuning parameters was not intended and it is out of the scope of this work. Hence it is for further study to assess the impact of the tuning parameters on the controllability of the system.

Table 5.5. PID tuning parameters

| Controller | Gain, Kc | Integral time , Ti (min) | Derivate time , Ti (min) |
|------------------------|----------|--------------------------|--------------------------|
| IC-101 | 0.1 | 0.2 | - |
| PIC-100 | 0.1 | 0.2 | - |
| IC-100 | 0.1 | 0.2 | - |
| TIC-100 | 1 | 20 | - |
| C100: Surge controller | 0.2 | 0.1 | - |
| C200: Surge controller | 0.2 | 0.1 | - |

5.4 Wet gas impact

One of the main goals of the thesis is to develop a model which can take into account the wet gas impact on the compressor performance. In order to do so, two different methods were implemented within the dynamic model, the first based on experimental data interpolation and the second based on empirical relations.

Method 1: GVF interpolation correction (IC)

The first method uses experimental compressor characteristic curves at GVF = 1 and GVF = 0.95 to correct the behavior of the compressor performance under wet gas conditions. The variations in polytropic head, volume flow and polytropic efficiency from the experimental curves at GVF =1 and GVF = 0.95 are used as interpolation factors to interpolate/extrapolate point by point the characteristic curves of our compressor impellers to the given GVF. This procedure is fully explained in the Appendix B.

The final result of the interpolated curves can be better illustrated in Figure 5.6:

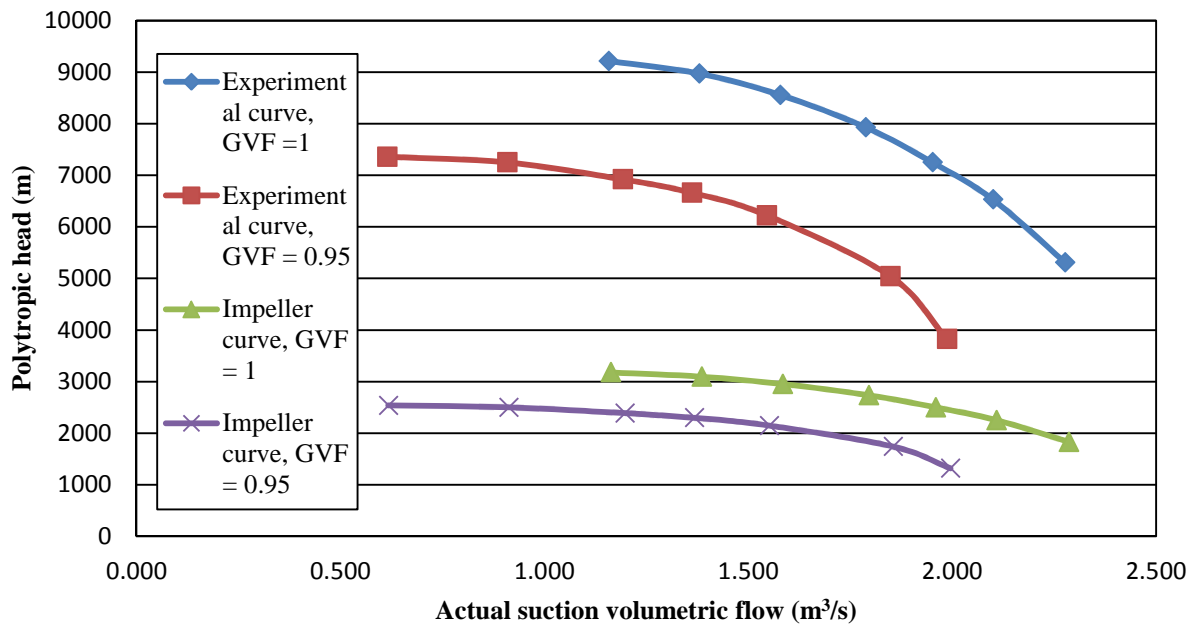


Figure 5.6. Comparison between experimental curves and the first impeller curve

Once the impeller curve at $GVF = 0.95$ has been calculated it is also possible to calculate a new curve for any GVF value. The same procedure also applies to the polytropic efficiency vs. volume flow curve. The Figure 5.7 shows the characteristic curve of the first impeller at different GVF . The Appendix C shows a complete example of the interpolation procedure.

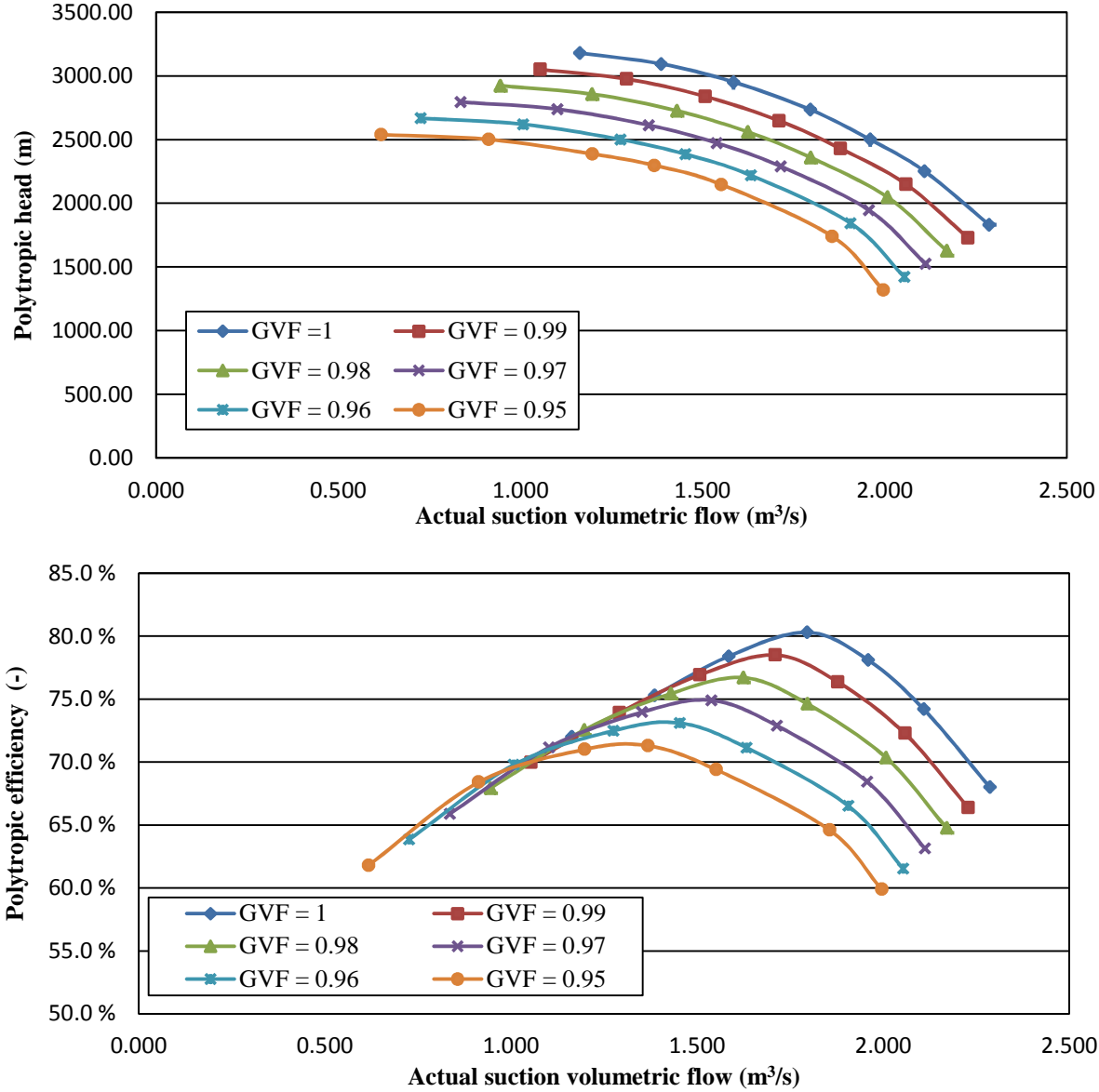


Figure 5.7. First impeller characteristic curve at different GVF

Method 2: Woods’s correction (WC)

The second method involves the correction of the compressor performance based on the GMF calculated from the GVF , gas and liquid densities at the compressor inlet conditions. The correction method used in this thesis is presented by (Hundseid, 2008) in his Doctoral Thesis and it is a modification of the Wood’s multiphase speed of sound model for two-phase flow.

The polytropic head at wet gas conditions is corrected based on the dry gas conditions for the corresponding volumetric flow, suction pressure and compressor speed. The volumetric and the polytropic efficiency are not corrected with this method.

The empirical relations proposed by (Hundseid, 2008) can be manipulated resulting in the following equations:

$$h_{p,w} = h_p \cdot W \tag{5.4}$$

$$W = \frac{1}{f \cdot \sqrt{f \cdot GVF}} \tag{5.5}$$

$$\frac{1}{GMF} = f = \frac{[GVF \cdot \rho_g + (1 - GVF) \cdot \rho_l]}{\rho_g} \tag{5.6}$$

From these relations, each point of the polytropic head characteristic curve at dry conditions can be corrected to the corresponding value at the given wet gas conditions, and the only information needed are the GVF, the gas and liquid densities at the compressor inlet (see Appendix C).

Any of the two methods can be implemented within HYSYS (both, in steady and dynamic state) to take into account the compressor performance variation at wet gas conditions. Once the characteristic curve at N = 100% is corrected the full compressor map is generated by applying the fan laws. A comparison between the two methods is shown in the Figure 5.8 below. In the next section it will be explained how these two correction methods were implemented in HYSYS Dynamics to correct the performance of both impellers.

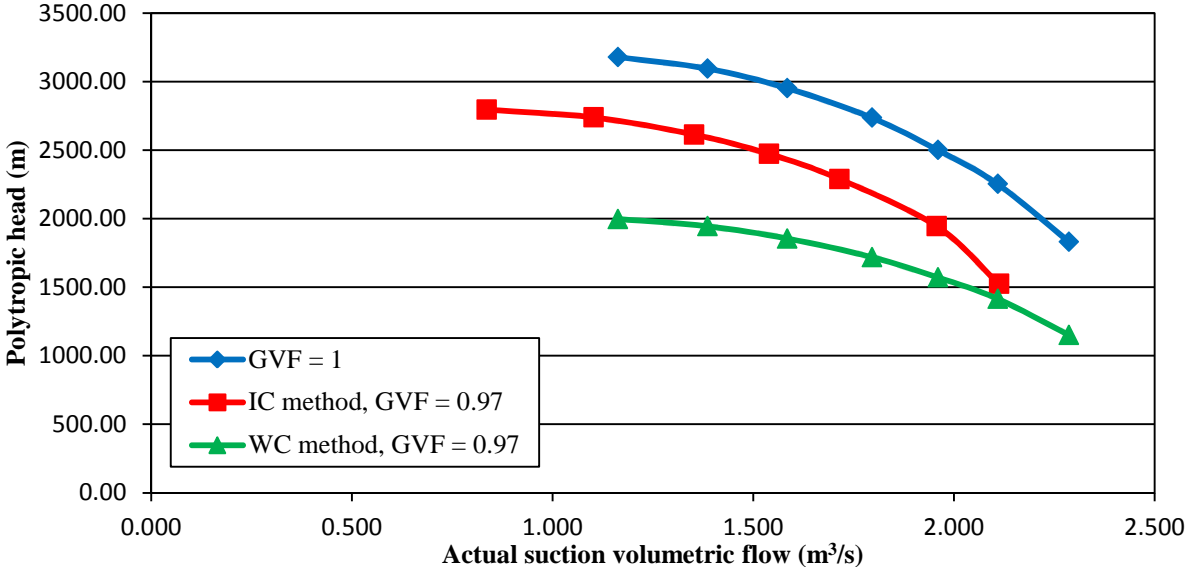


Figure 5.8. Corrected characteristic curve of First impeller

In the figure it can be seen that the interpolation method corrects the polytropic head but also the inlet volumetric flow while the second method only corrects the polytropic head. Again, both correction methods are full illustrated in the Appendix C.

5.5 Correction methods implementation

In the Section 5.2 it was explained that in the HYSYS model the compressor map of each impeller were set into two different spread sheets and these are linked directly with the compressor units. The goal is to develop a procedure that applies the already discussed correction methods to the dry-gas conditions curve (set in the spread sheets) and automatically put the new curves (at wet-gas conditions) in the compressor units. However, and this is a major drawback of HYSYS, the compressor curves are automatically deactivated when it is attempted to modify them. This means, that the solver/integrator will stop when the corrected new compressor curves are sent from the spreadsheet to the compressor unit.

In a steady-state simulation, one possibility is to simply activate manually each of the compressor curves and then activate the solver. However in the case of having any internal loop, as the cooling gas loop, the flow sheet needs to iterate several times before it converges and the manual activation of the curves becomes tedious. In dynamic simulation the situation is more critical and a manual re-activation of the curves is not possible. Indeed, if any change is made on the compressor curves while the integrator is running the simulation automatically fails and HYSYS shuts down. In order to overcome this issue it was necessary to create a code in VBA for Microsoft Excel which was connected with HYSYS dynamics to automatize the procedure of correcting and activating of the compressor curves.

Following the explained in the Section 4.6 (HYSYS Automation) the code created in VBA is able to work as an interface between HYSYS Dynamics and Excel, see Figure 5.9. The VBA code continuously checks the flow properties at the inlet of the compressors and when any change, with respect to the previous recorded time step, is detected the new curve and surge flow constant C are automatically computed in excel, sent to the HYSYS spread sheets and from the spread sheets to the respective compressor unit.

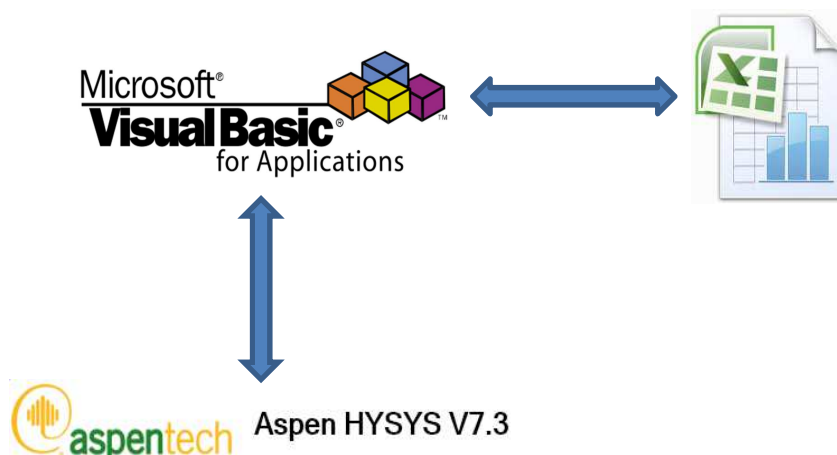


Figure 5.9. Schematic framework of VBA, Excel and HYSYS

The manner of how both correction methods are implemented on VBA is very similar however they use different inputs and generate different outputs. The interpolation method needs only the GVF at the inlet of each compressor and corrects the dry gas curve in polytropic head, polytropic efficiency and suction volume flow. Alternatively, the Wood's correction needs the GVF, gas and liquid densities and corrects the characteristic curve only in terms of head. Therefore, individual codes were created for each particular method but they can still be applied to the same HYSYS case. The complete script for each correction method is attached in the Appendix D.

The implementation and correction procedure are presented in the Figure 5.10 and Figure 5.11 as flow chart forms. The subscripts 1 and 2 stand for inlet of first and second impeller, respectively. The VBA script uses the hierarchy system to connect first with the HYSYS case, with process flow sheet and then with all the items and operation units of importance.

Note from the flow charts that the interpolation method corrects the curve only if the GVF changes while the Wood's method corrects the curve only if the parameter W (Eq. 5.4) changes. Since the WC method comprises three input parameters that will change simultaneously, the direct comparison of only one variable makes the interaction between HYSYS and VBA more numerically favorable. As example, in some cases the GVF reaches its steady-state value but the phases densities change for one or two time steps more. Hence, the W factor accounts for this numerical issue.

The VBA script stops the integrator when it detects that the ε condition is met for any of the two impellers and the compressor curves have to be corrected. As mentioned before, HYSYS will shut down if it is attempted to edit the compressor curves while the integrator is running. Then, Excel computes the corrected compressor curves and the VBA sets the new values in the HYSYS spread sheets where they are sent directly to the respective compressor. After, the integrator is started and the whole procedure is repeated.

It is important to notice that the VBA code and the HYSYS dynamics flow sheet have different and variable computational speeds. During the transient behavior the VBA code tends to be slower and faster when the steady-state is reached. Hence, it could be attempted to couple both speeds but that would imply that the simulation in HYSYS has be stopped when it is faster than the VBA script, and vice versa. However, the tests done showed that the VBA code is faster than response of the process controllers and this should be enough for a valid performance.

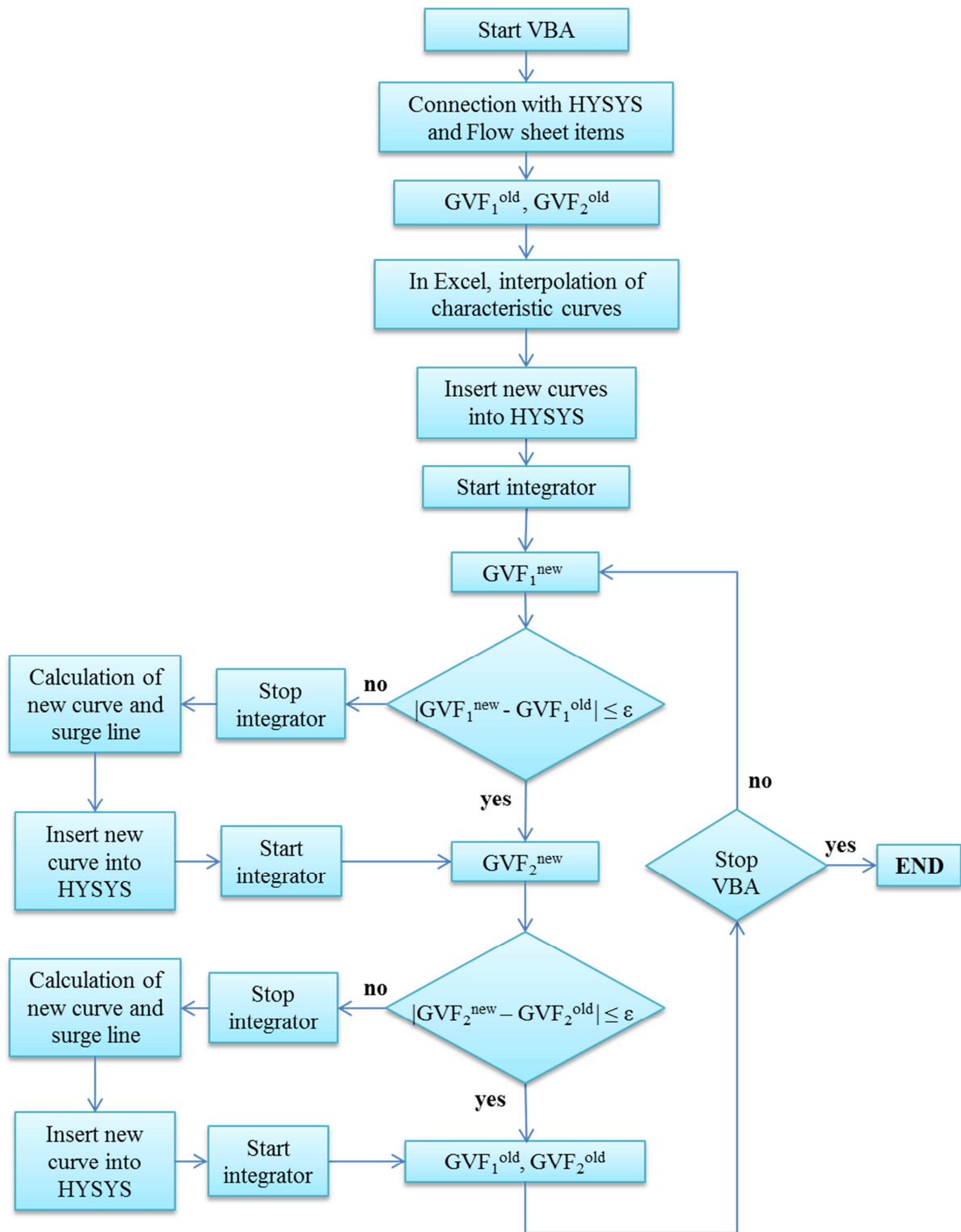


Figure 5.10. Flow chart: Interpolation correction method

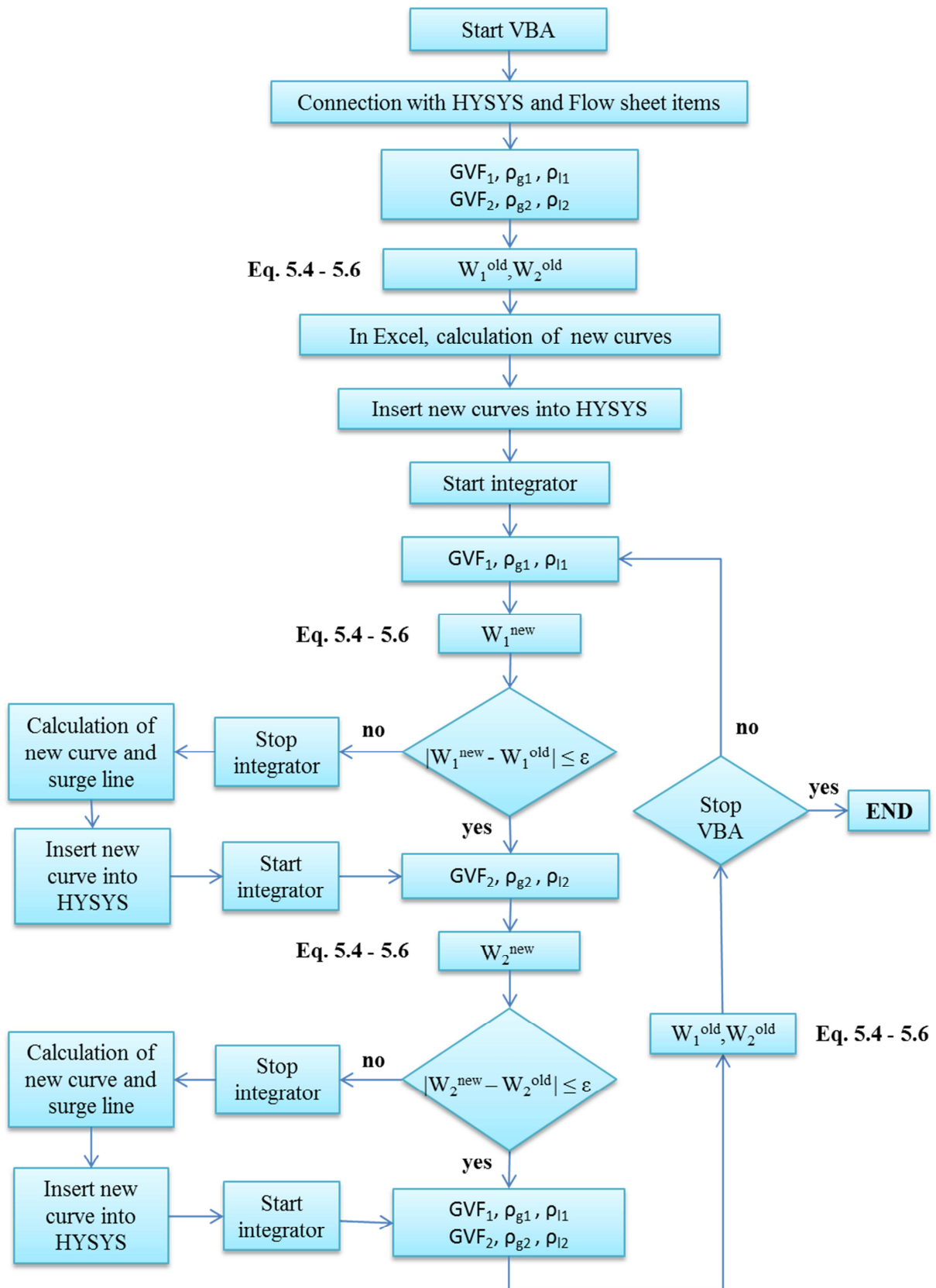


Figure 5.11. Flow chart: Wood's correction method

The VBA code was developed in such way that a user dialog window allows the user to start, stop or resume the application (see Figure 5.12). The interaction between HYSYS dynamics and the VBA correction script is only stopped when the user desires so. In this way the VBA application is in continuous and permanent communication with HYSYS dynamics and there is no need to specify the integration time in HYSYS. Additionally, it is possible to change any specification value in the flow sheet (pressures, temperatures, flow rates, etc.) and the VBA script will be still online and monitoring the GVF at inlet of both impellers.

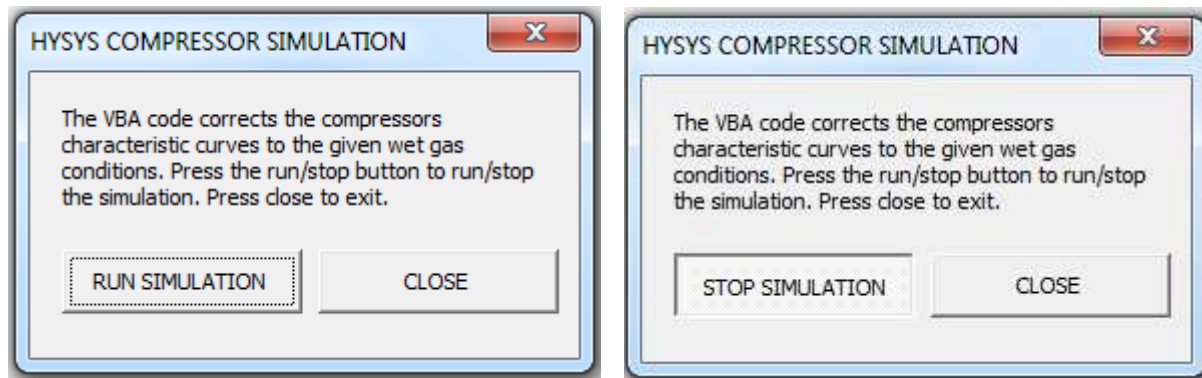


Figure 5.12 Run/Stop simulation button

In this chapter it was described the development of the dynamic model in HYSYS and all the required considerations. It was shown that in any dynamic simulation special care has to be put on the boundary conditions, pressure-flow specification of the operating units and the control design of the process. Also, it was explained how the Automation functionality of HYSYS was used to implement corrections methods that account for wet gas compressor performance. In the next chapter several cases of study will be presented.

6 Results and discussions

Once the HYSYS dynamics functionalities were explored and used to build the dynamic subsea wet gas compression model it was necessary to study the performance of the system under different operating conditions. In particular, it is of interest to study the compressor response under wet gas conditions. In order to do this, it was first necessary to test and tune the implementation of the correction methods discussed in the Section 5.5. The tuning of the correction methods application will be exposed in this chapter.

The cases of study described in this chapter are:

- **Case 1:** GVF impact on compressor performance
- **Case 2:** Molecular weight impact on compressor performance
- **Case 3:** Suction pressure impact on compressor performance

6.1 Model tuning

With the objective of improving the implementation of the wet gas corrections methods it was necessary to run several pre-tests. As seen in the Figure 5.10 and Figure 5.11 the correction, or not, of the impeller characteristic curve is subjected to a pre-defined value ε . Hence it was necessary to explore the real effect of the parameter ε in the application of the correction methods.

In this sense, a Sequence was set up in the Event Scheduler which automatically changed the set point in the GVF controller (IC-101) from 1 to 0.97. This procedure was repeated for different ε values and for both correction methods. The conditions of this study are summarized in the Table 6.1.

Table 6.1 Model tuning: Main case parameters

| | |
|---------------------------|---|
| Action | Reduce SP of IC-101 from 1 to 0.97 |
| Adjusted variable | Water flow rate |
| Monitored variable | Compressor speed Pipeline inlet pressure GVF at compressor inlet (first impeller) |

IC method:

In the case of the interpolation correction the values tested for ε were:

- $\varepsilon = 1 \cdot 10^{-2}$
- $\varepsilon = 1 \cdot 10^{-3}$

As result, the compressor speed and pipeline inlet pressure profile are shown in the Figure 6.1 and Figure 6.2 respectively.

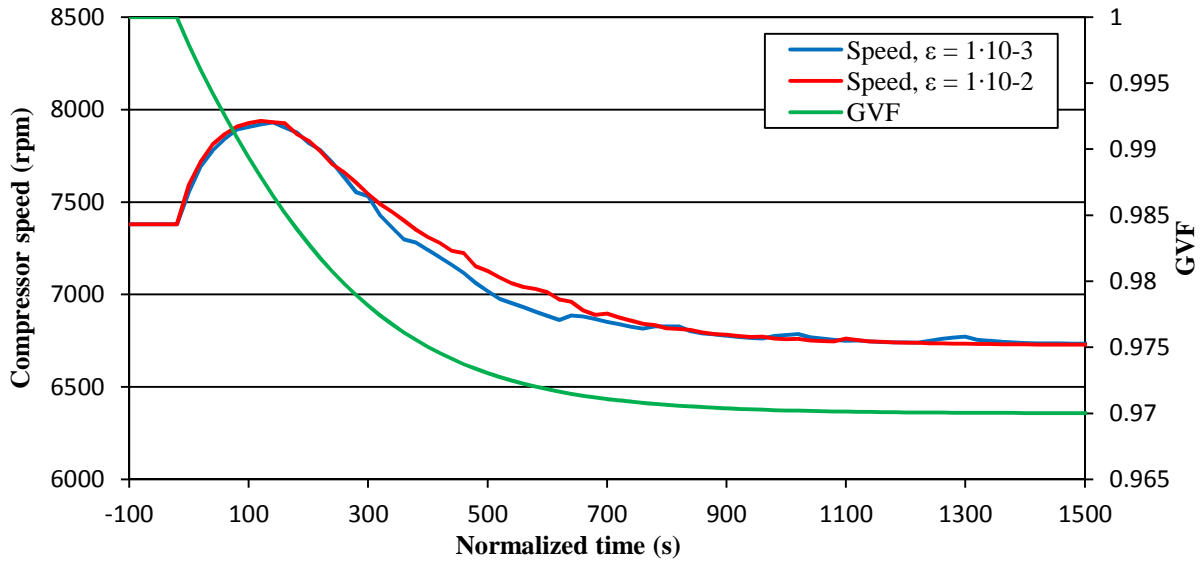


Figure 6.1 Model tuning: Compressor speed profile, IC method

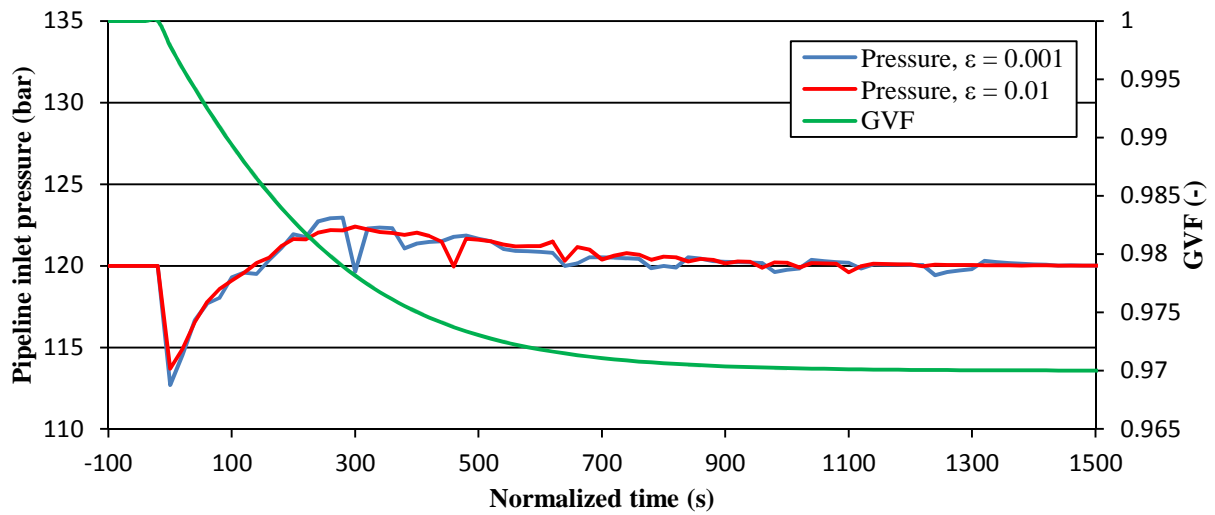


Figure 6.2 Model tuning: Pipeline inlet pressure profile, IC method

WC method:

In the case of the Wood's correction the values tested for ϵ were:

- $\epsilon = 1 \cdot 10^{-2}$
- $\epsilon = 1 \cdot 10^{-3}$
- $\epsilon = 1 \cdot 10^{-4}$

As result, the compressor speed and pipeline inlet pressure profile are shown in the Figure 6.3 and Figure 6.4 respectively.

It can be seen how the correction routine is implemented in different time steps making the compressor speed profile to oscillate until it reaches definitely the steady-state. For the different ϵ values the steady-state speed vale is around the same.

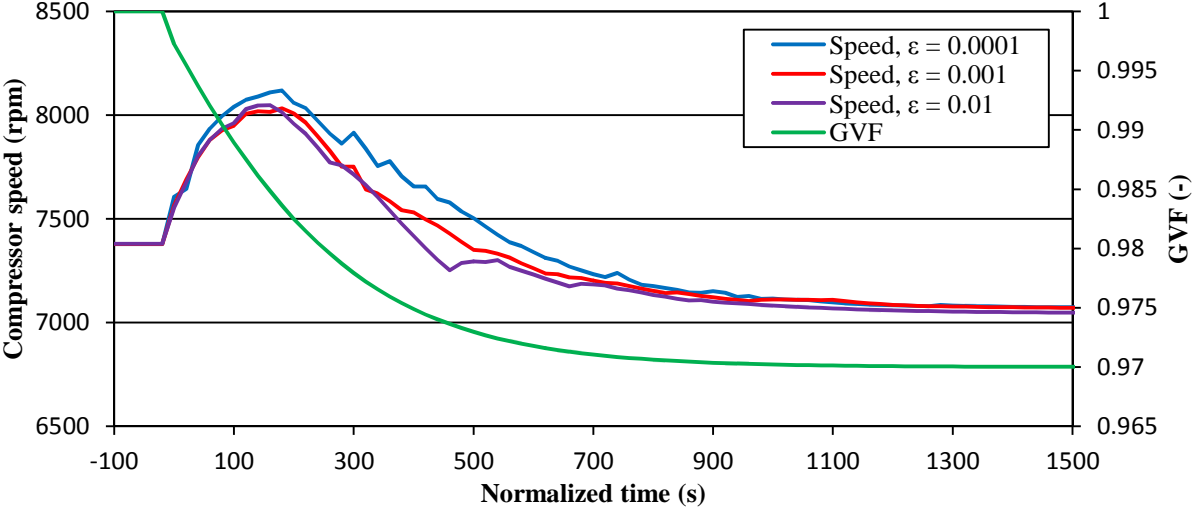


Figure 6.3 Model tuning: Compressor speed profile, WC method

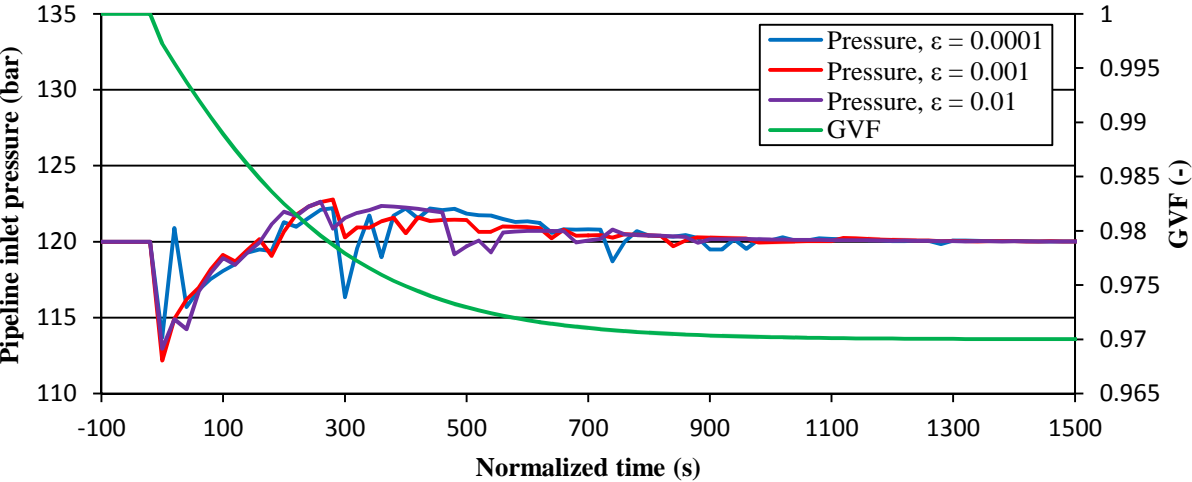


Figure 6.4 Model tuning: Pipeline inlet pressure profile, WC method

Discussion

From the profiles it can be seen that the ϵ value affects the stability and dynamic response of the compressor, e.g. a large ϵ will activate more times the correction routine making the system to iterate more before it can reach the steady-state. Therefore, it is essential to find an

appropriate value for ε which can guaranty a good result without compromising the stability of the system.

It can be also seen that for the IC method a value of $\varepsilon = 0.01$ offered good stability to the system and it is still possible to reach the same steady-state compressor speed as for $\varepsilon = 0.001$. In the same way, the WC method at $\varepsilon = 0.001$ reported smooth speed and pressure profiles, while resulting in the same steady-state compressor speed as for $\varepsilon = 0.0001$.

For all the following cases of study the ε values were fixed in 0.01 and 0.001 for the first and second correction method respectively, see Table 6.2

Table 6.2 Correction routine parameter ε : Values for cases of study

| IC | WC |
|----------------------|-----------------------|
| $\varepsilon = 0.01$ | $\varepsilon = 0.001$ |

The VBA script was created in such way that it is possible to keep the count of how many times an impeller curve is changed. In this sense, the number of changes in each correction methods were compared for the given ε values (0.01 and 0.001) and both methods computed nearly the same number of changes. Specifically, the characteristic curve of the first impeller was adjusted 15 times and the second impeller 21 times. At higher ε values the number of changes can be up to the double or more. This issue limits the real time online monitoring of the process.

6.2 Case 1: GVF impact

A first issue to consider is the response of the compressor and the correction routine under different GVF changes. The first sub-case corresponds when the IC-101 controller is set in automatic mode and the GVF at the inlet of the first impeller is adjusted progressively from 1 to 0.97; whereas in the second sub-case the GVF changes within one time step. For this last it was necessary to change the mode of the controller IC-101 from automatic to manual and set directly the water molar flow rate. For both sub-cases the Event Schuler was used to set up the particular cases.

In the following the results of the two sub-cases will be presented.

Case 1.a: Smooth GVF change

Table 6.3 Case 1.a: Main case parameters

| | |
|---------------------------|---|
| Action | Reduce SP of IC-101 from 1 to 0.97 |
| Adjusted variable | Water molar flow rate |
| Monitored variable | Compressor speed Pipeline inlet pressure Total compressor power GVF at compressor inlet (first impeller) |

As result, the compressor speed, pipeline inlet pressure and total compressor power profile are shown in the Figure 6.5, Figure 6.6 and Figure 6.7 respectively (the GVF profile is presented in each figure). In this sub-case, the GVF reaches the set point after 1100 seconds.

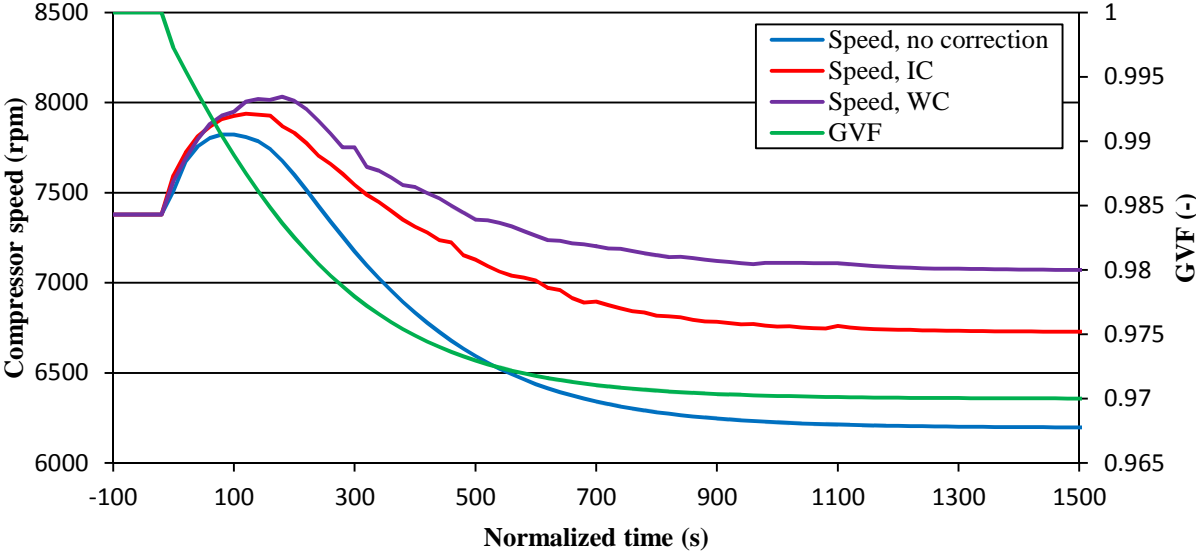


Figure 6.5 Case 1.a: Compressor speed profile

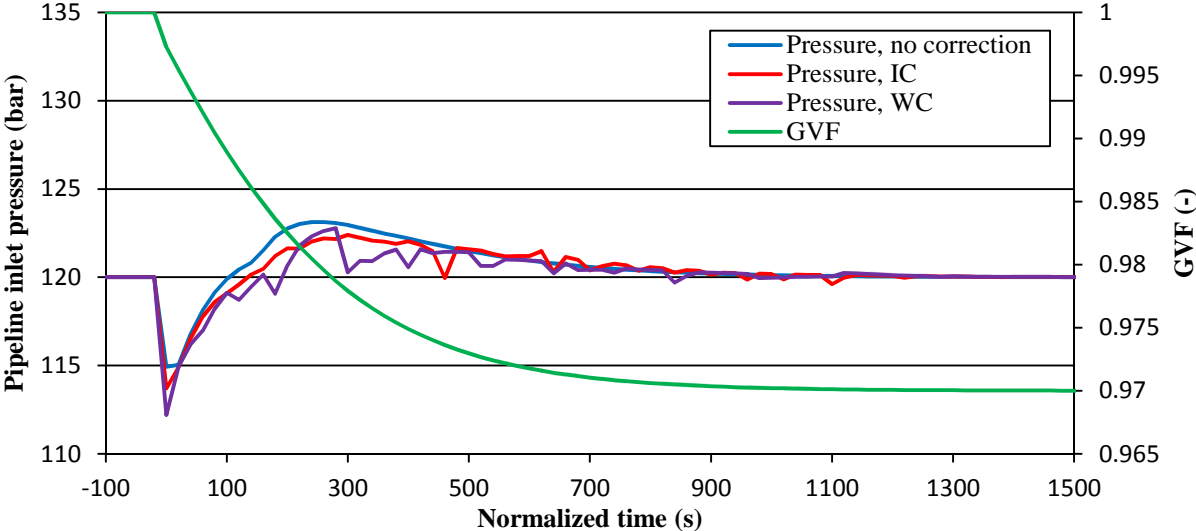


Figure 6.6 Case 1.a: Pipeline inlet pressure profile

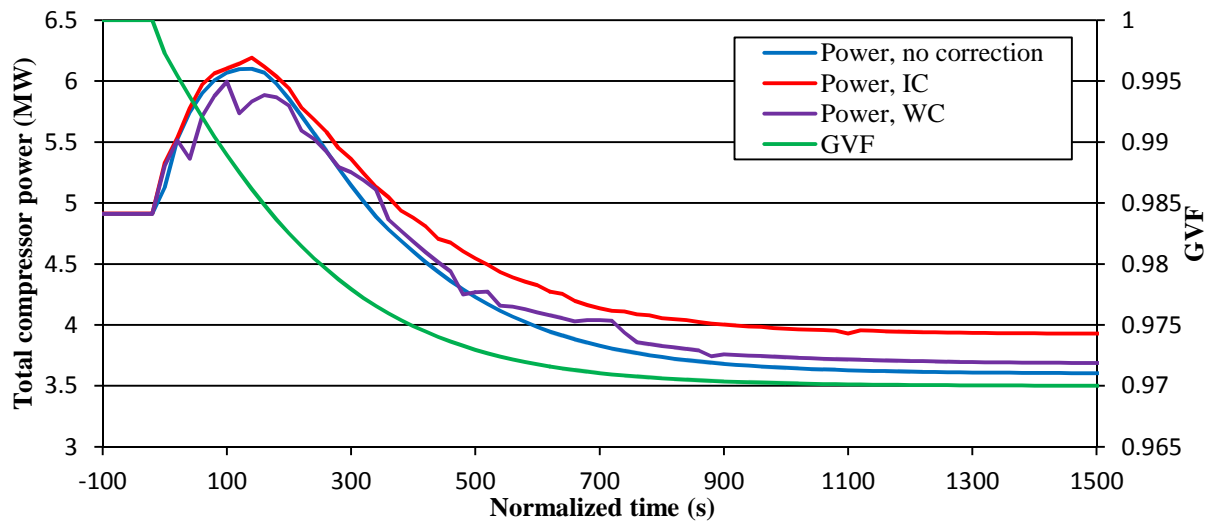


Figure 6.7 Case 1.a: Total compressor power profile

Case 1.b: GVF change in a time step

Table 6.4 Case 1.b: Main case parameters

| | |
|---------------------------|---|
| Action | Set OP of IC-101 equal to 57.85% of Max. molar flow* |
| Adjusted variable | Water molar flow rate |
| Monitored variable | Compressor speed Pipeline inlet pressure Total compressor power GVF at compressor inlet (first impeller) |

* The maximum molar flow is set in 3 kmol/s.

As result, the compressor speed, pipeline inlet pressure and total compressor power profile are shown in the Figure 6.8, Figure 6.9 and Figure 6.10 respectively.

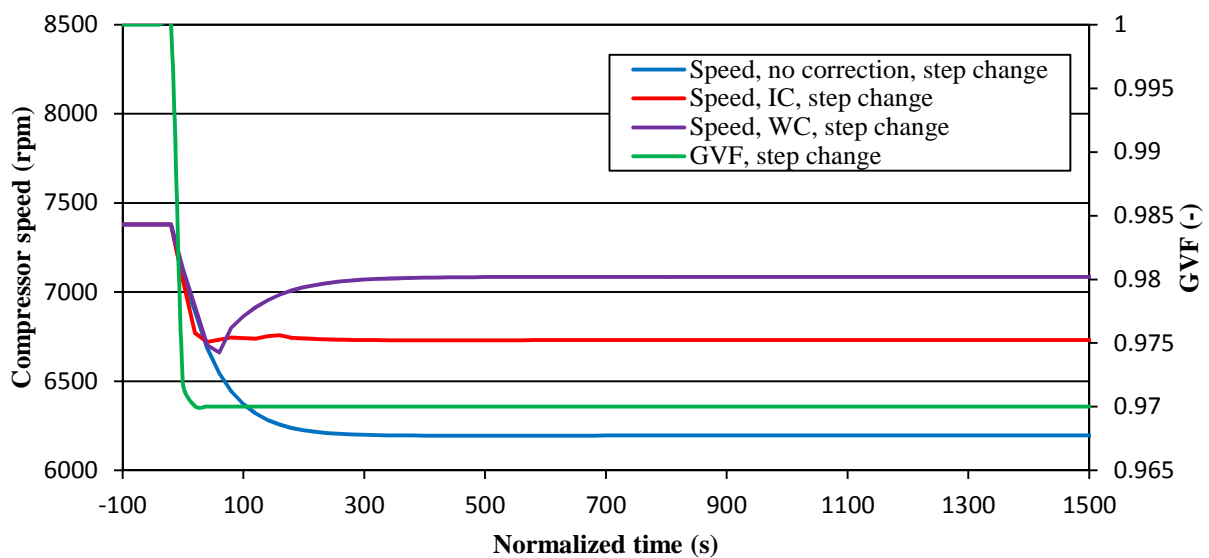


Figure 6.8 Case 1.b: Compressor speed profile

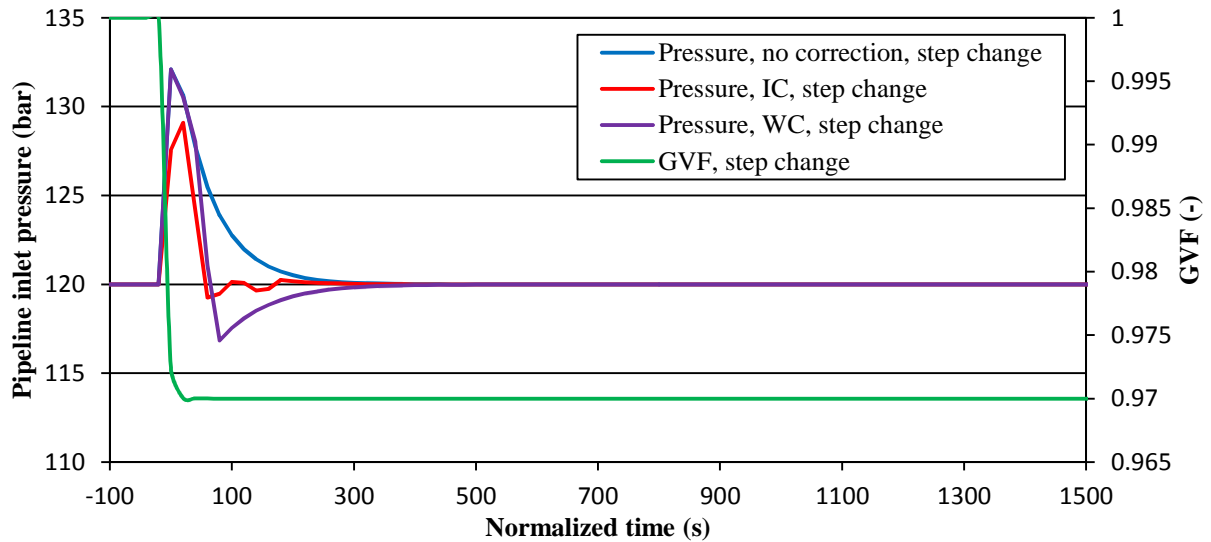


Figure 6.9 Case 1.b: Pipeline inlet pressure profile

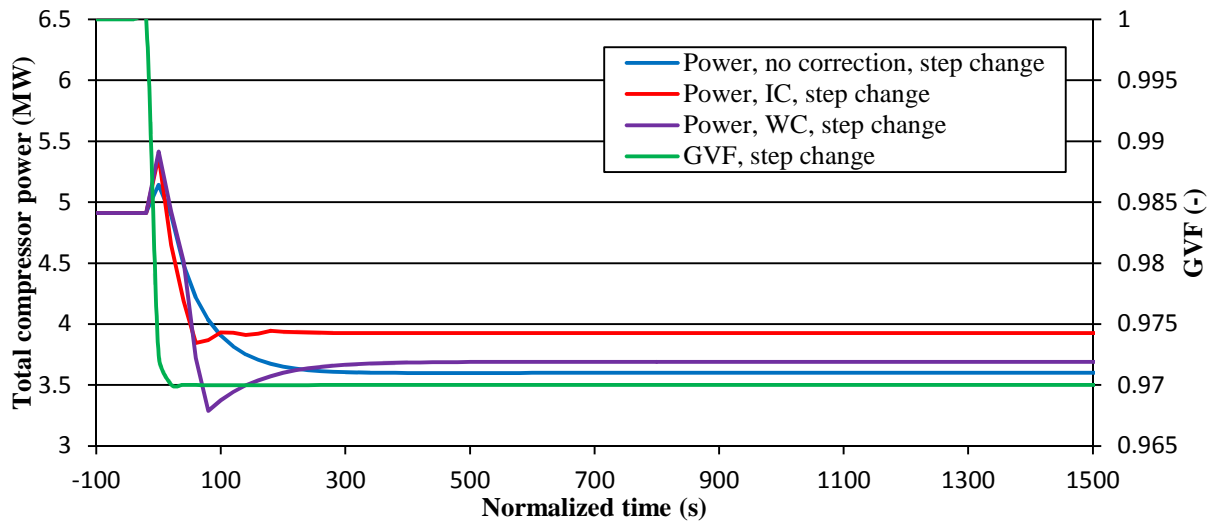


Figure 6.10 Case 1.b: Total compressor power profile

In the Figure 6.5 and Figure 6.8 it is possible to see how the compressor speed is adjusted by the controller PIC-100 during the GVF variation. It can be seen that the compressor speed decreases as the GVF decreases until it reaches again the steady-state. For both sub-cases the steady-state speed values were identical, being highest for the WC method (7069 rpm) and lowest for the simulation without correction (6197 rpm).

In the Figure 6.8, Case 1.b, it can be seen that for the three simulations the compressor speed reaches its steady-state value after 200 seconds and before that, the two correction methods are already executed. This evidences the quick response of the correction methods even when the GVF changes within one step. In the Case 1.a the compressor speed reaches the steady-state after 1300 seconds, see Figure 6.5.

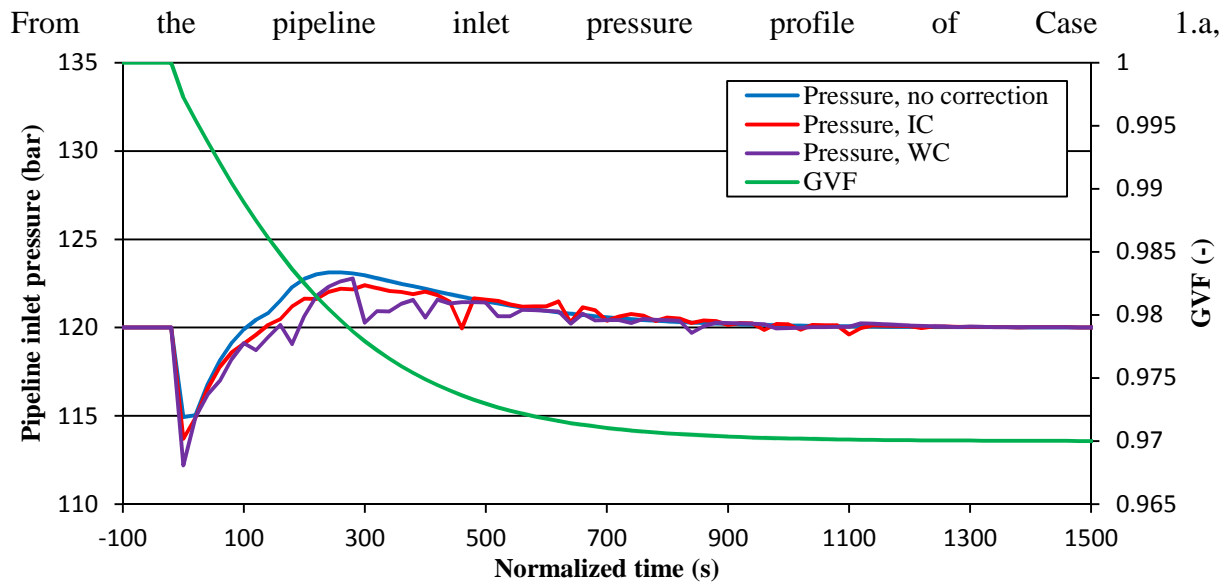


Figure 6.6, it can be observed that the simulation with no correction reaches the steady state after approximately 900 seconds, whereas the profile by interpolation and Wood's correction achieve the steady-state at around the same time after 1200 seconds. In the Case 1.b, the pressure reaches (all the methods) the steady-state after 400 seconds.

It can be noticed from the

Figure 6.7 and Figure 6.10 that the total compressor power decreases when the GVF decreases. The steady-state values are also the same between correction methods in the two sub-cases. The interpolation correction method computed the highest compressor power, 3.93 MW, while the simulation without correction computed the lowest, 3.6 MW.

Discussion

In the case 1.b, all the simulations reach the steady-state pressure value after 400 seconds, see Figure 6.9, which is much faster than in the Case 1.a. This is due to the drastic change in GVF and the quick response of the system. Again, the fact that the pressure when no correction method is applied stabilizes at the same time as when the correction methods are applied is an indication of the good interaction between the HYSYS flow sheet and the VBA script.

An important matter to consider is the overshoot ratio in the pressure signal. The Table 6.5 presents the overshoot values in the pressure signal taking as reference the maximum deviation from the SP of 120 bar. In the Case 1.b, the pipeline inlet pressure presents a maximum overshoot of 10.1% (no correction and WC method) whereas the maximum overshoot in the Case 1.a is 6.5% (WC method). The overshoot is higher for the Case 1.b due to the step change in the GVF. However there is not big difference with respect to the Case 1.a.

An overshoot of 10% seems to be acceptable but it is necessary to make a more rigorous and specific study to determine if these pressure fluctuations could cause vibrations and system deteriorations.

Table 6.5 Case 1: Overshoot ratio in the controlled pressure

| | Case 1.a | | Case 1.b | |
|---------------|----------------|-----------|----------------|-----------|
| | Max. deviation | Overshoot | Max. deviation | Overshoot |
| | bar | % | bar | % |
| No correction | 114.9 | 4.2 | 132.1 | 10.1 |
| IC | 113.7 | 5.3 | 129.1 | 7.6 |
| WC | 112.2 | 6.5 | 132.1 | 10.1 |

In the Table 6.6 it is displayed the main performance parameters in steady-state, both, at the beginning of the simulation and after the GVF disturbance has been introduced in the system.

As it can be seen from the table, the experimented reduction in compressor power is due to the reduction in polytropic head and reduction in suction mass flow (since the inlet molar flow remains constant and the low molecular weight of the water). It can be noticed that the reduction in polytropic head is similar among the three cases but the polytropic efficiency variation presents a larger difference. From the three simulations, the IC method reports the highest compressor power due to the resulting low efficiency in both impellers. The opposite occurs with the simulation with no correction which reports the highest efficiencies and thereby the lowest compressor power. This difference is a direct consequence of correcting the efficiency curve in the IC method.

Table 6.6 Case 1: Steady-state performance values

| | First impeller | | | Second impeller | | | Total compressor power | | |
|---------------|----------------|--------|----------|-----------------|---------|----------|------------------------|------------|-----------|
| | \dot{m} | H_p | η_p | \dot{m} | H_p | η_p | HYSYS | Calculated | Deviation |
| | kg/s | m | % | kg/s | m | % | MW | MW | % |
| Initial | 144.74 | 1406.2 | 78.21 | 137.50 | 1332.3 | 76.35 | 4.91 | 5.00 | 1.83 |
| No correction | 140.50 | 1076.5 | 79.82 | 133.47 | 1057.9 | 79.64 | 3.60 | 3.67 | 1.94 |
| IC | 140.50 | 1093.8 | 70.07 | 133.47 | 1045.54 | 72.63 | 3.92 | 4.00 | 2.04 |
| WC | 140.50 | 1068.8 | 77.71 | 133.47 | 1066.58 | 78.15 | 3.69 | 3.75 | 1.63 |

The corrected characteristic curves can be also represented in one single graph in order to have a clear idea of how they are corrected. The Figure 6.11 and

Figure 6.12 show the characteristic curve of the first impeller after the steady-state has been reached. The efficiency characteristic curve for the WC method is the same as for dry gas conditions.

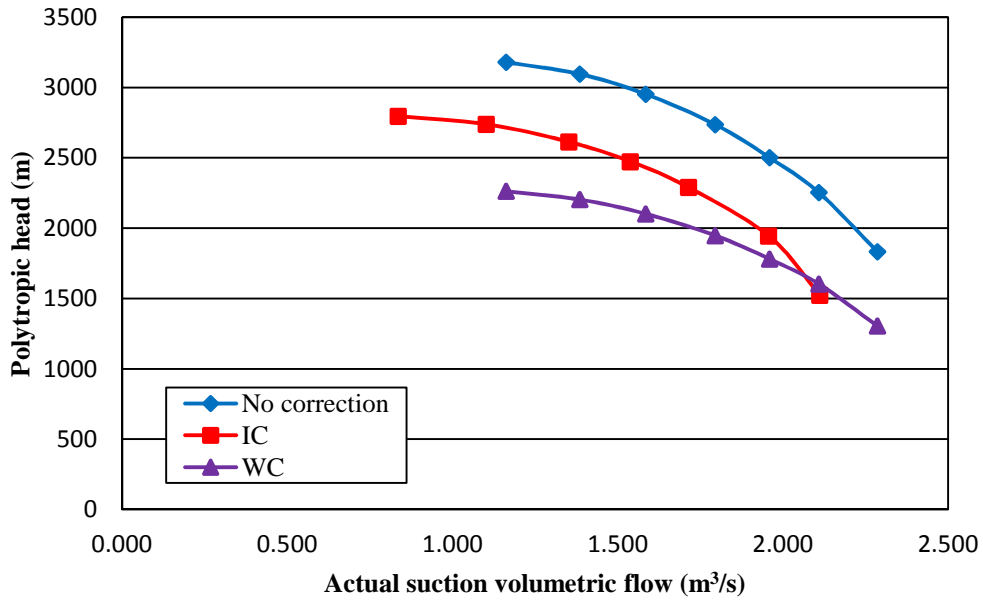


Figure 6.11 Case 1: Corrected first impeller characteristic head curve at GVF = 0.97

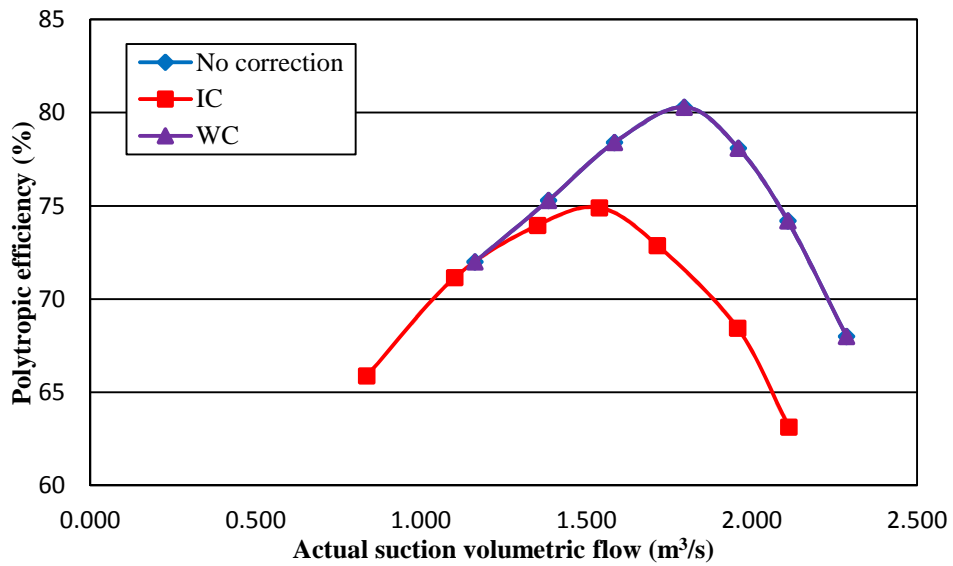


Figure 6.12 Case 1: Corrected first impeller characteristic efficiency curve at GVF = 0.97

The full compressor map can be also represented with the trajectory of the corresponding operating point. A different compressor map and operating point trajectory will be computed for each impeller and for each correction method. As an example, the compressor map of the first impeller without correction is displayed in the Figure 6.13. The figure also shows the trajectory of the operating point for both sub-cases 1.a and 1.b.

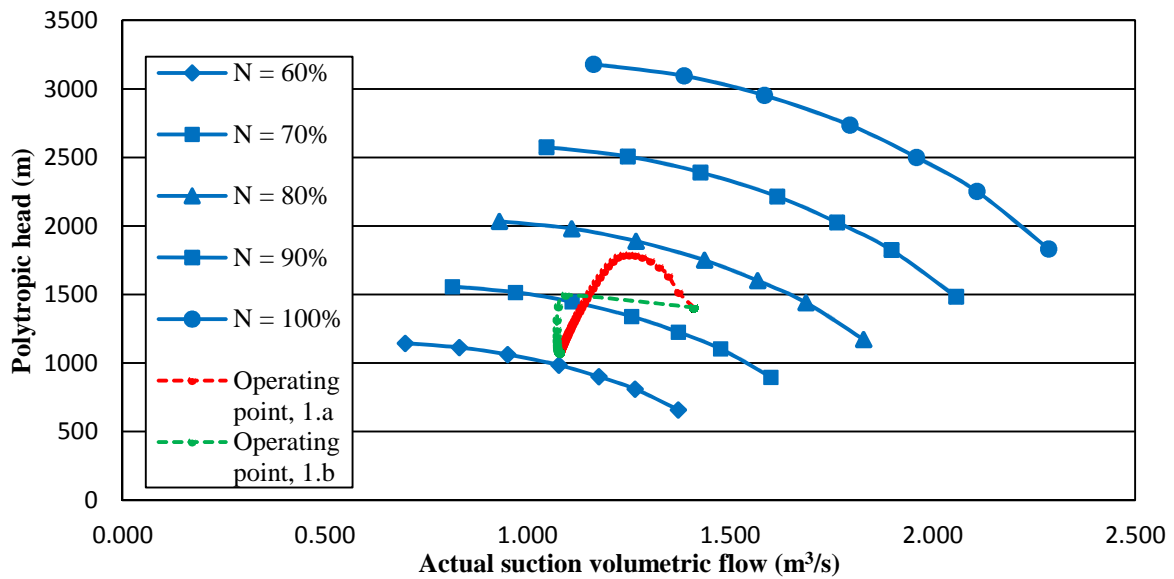


Figure 6.13 Case 1.a: First impeller compressor map without correction

As it can be seen in the compressor map, and also from the compressor speed profiles, the Case 1.b presents a drastic drop in speed during the GVF step change. This is a direct consequence of the quick variation in the inlet volumetric flow which due to the high water mass density is reduced in few seconds. Furthermore, the polytropic head is reduced but maintaining a relative high efficiency. All these produce that the compressor power response has a smaller pick than in the Case 1.a.

Considering again the pressure pulse observed in the Case 1.b it can be seen in the Figure 6.13 that the operating point is far from the surge line. This means that during normal operation a pressure overshoot of 10% should be handled by the compressor magnetic bearings and the entire system. This specific issue must be considered during the designing and testing of the whole compression system.

6.3 Case 2: Molecular weight impact

The inherent consequence of manipulating the GVF at the suction side of the compressor is the variation in the fluid molecular weight. Therefore, it is important to assess the real impact of the molecular weight on the compressor performance and system response. In this sense, a case study was set up where the water stream was replaced for a condensate stream of molecular weight 67.94 kg/kmol, see composition in Appendix E. The Table 6.7 summarized the main parameters for this case.

Table 6.7 Case 2: main case parameters

| | |
|---------------------------|---|
| Action | Reduce SP of IC-101 from 1 to 0.97 |
| Adjusted variable | Condensate molar flow rate |
| Monitored variable | Compressor speed Pipeline inlet pressure Total compressor power GVF at compressor inlet (first impeller) |

As result, the compressor speed, pipeline inlet pressure and total compressor power profile are shown in the Figure 6.14, Figure 6.15 and Figure 6.16, respectively.

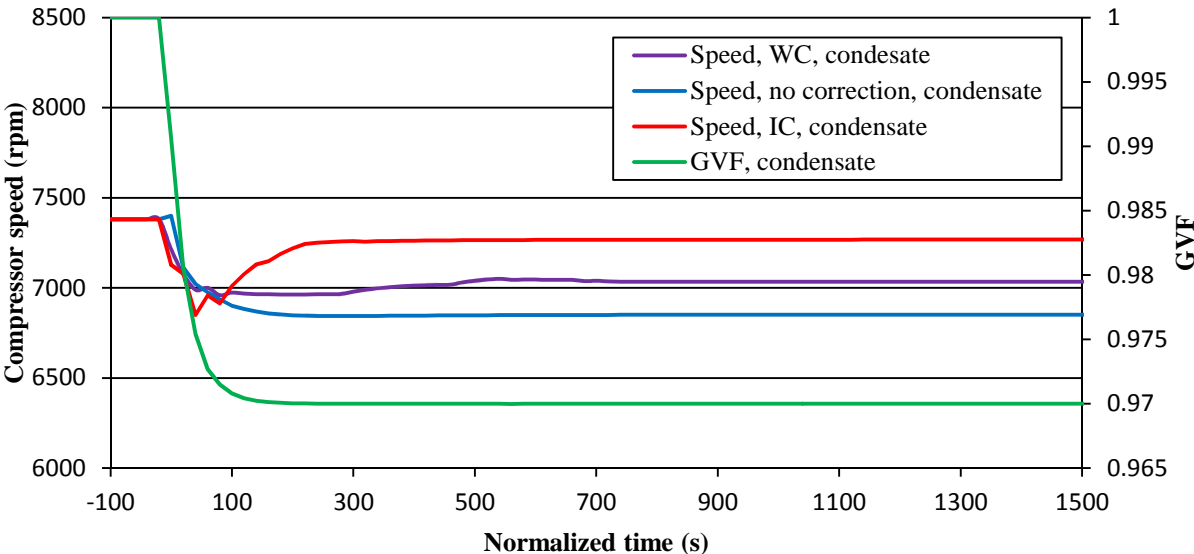


Figure 6.14 Case 2: Compressor speed profile

The speed profile shows that the compressor speed decreases when the GVF decreases and for this specific case the GVF reaches the steady-state value after 200 seconds. The IC method computes the highest compressor speed (7268 rpm) whereas the WC method the lowest, 6851 rpm.

In the simulation without correction the speed reaches the steady-state after 200 seconds, the IC method after 300 seconds and the WC method after 800 seconds.

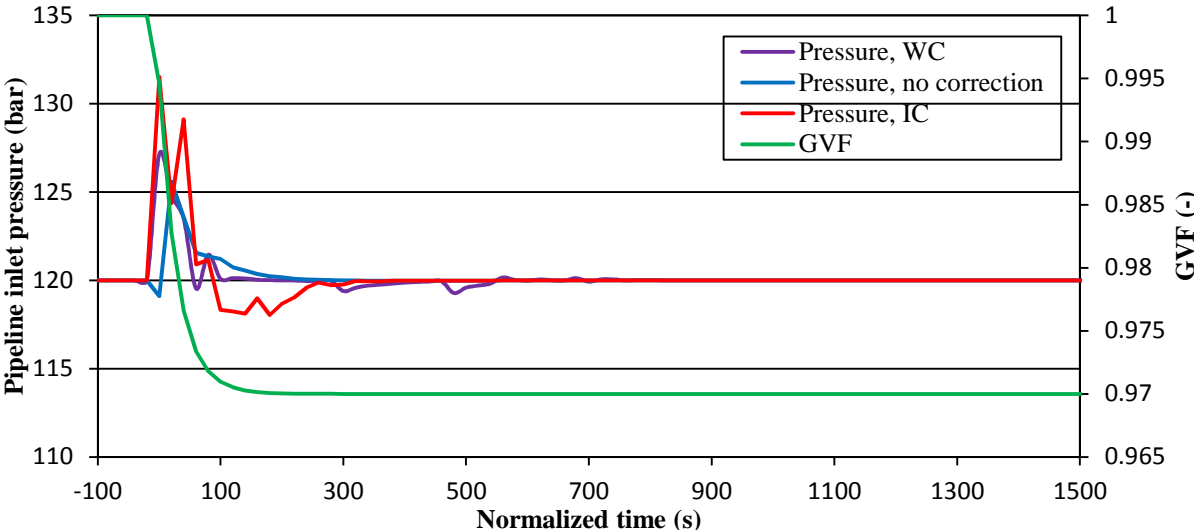


Figure 6.15 Case 2: Pipeline inlet pressure profile

From the pipeline inlet pressure profile it can be observed that the no correction and the IC method reach the steady-state around after 300 seconds and the WC method after 800 seconds.

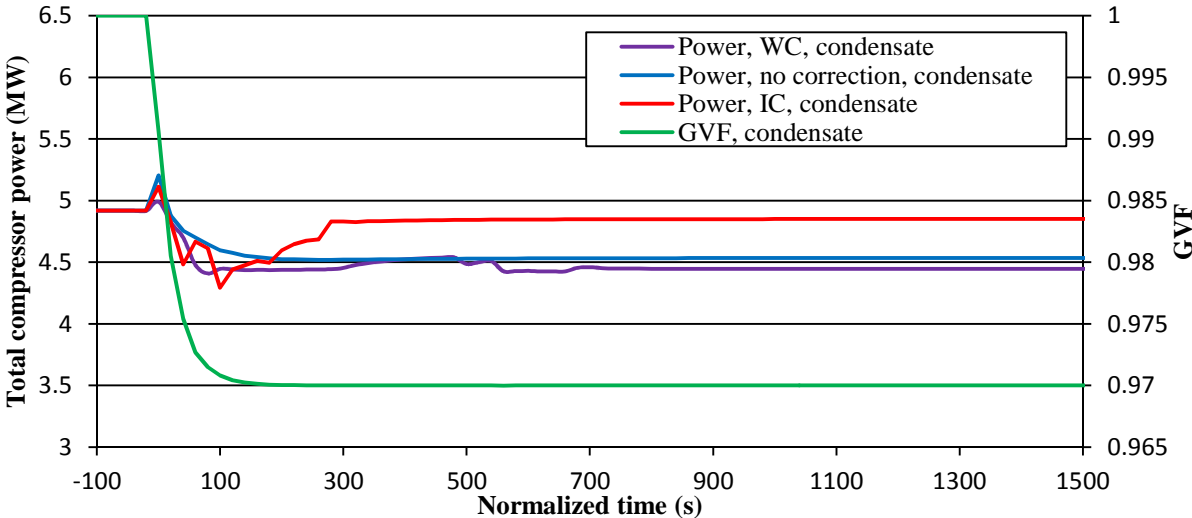


Figure 6.16 Case 2: Total compressor power profile

In the total compressor power profile can be seen how it decreases with the reduction in GVF. The IC method computes again the highest compressor power, 4.85 MW, whereas the WC method calculates the lowest, 4.45 MW.

Discussion

The GVF stabilization time in the Case 2 (200 seconds) is much less than the one needed in the case 1.a where the GVF steady-state time is 1100 seconds. This first difference is due to the low density of the condensate mixture (occupies more volume) which will have a quicker impact on the GVF. This quick impact on the GVF makes the system to behave more like the Case 1.b when the GVF changes within one time step. However the steady state values for the compressor speed and total power are quite different.

Since the WC method uses the GMF as correction parameter it is more sensible for changes in the fluid conditions and thus it is possible that it corrects the characteristic curve after the GVF has reached the steady-state. Once the GVF at the inlet of the first impeller has reached the steady-state the rest of the flow properties (gas and liquid density) are more likely to remain constant. However, this is not the case for the second impeller. In particular, the characteristic curve of the second impeller is corrected more times than the first impeller due to the inlet condition are continuous fluctuating. This is a direct effect of the motor cooling gas loop.

Additionally, it is possible to compute the overshoot ratio in the pressure signal. The overshoot for the Cases 1 and 2 are presented together in the Table 6.8. In contrast to the Case 1, all the overshoot values in the Case 2 are lower than 10% which is a good measure of the

stability of the system under GVF changes. It can be concluded that the system is more stable for fluids of higher molecular weight.

Table 6.8 Case 2: Overshoot ratio in the controlled pressure

| | Case 1.a | | Case 1.b | | Case 2 | |
|---------------|----------------|-----------|----------------|-----------|----------------|-----------|
| | Max. deviation | Overshoot | Max. deviation | Overshoot | Max. deviation | Overshoot |
| | bar | % | bar | % | bar | % |
| No correction | 114.9 | 4.2 | 132.1 | 10.1 | 125.6 | 4.7 |
| IC | 113.7 | 5.3 | 129.1 | 7.6 | 131.5 | 9.6 |
| WC | 112.2 | 6.5 | 132.1 | 10.1 | 127.1 | 5.9 |

Even though the total compressor power also decreases in the Case 2, the rate of reduction is less than in the Case 1 and the final steady-state values are higher. The reason is that in this case there are two opposite effects: the polytropic head reduction and the mass flow increment. Similarly to the Case 1, the polytropic head decreases with the GVF however, the mass flow increases due to the high molecular weight of the condensate stream (the molar flow is constant). Hence, these two effects work against each other causing the small reduction in compressor power. The Table 6.9 summarizes the main performance values at the steady-state conditions. As mentioned before, the IC method computes the highest steady-state compressor power. This is due to the low efficiency resulting from the efficiency curve interpolation.

It is important to note that WC method reports the lowest compressor power as a result of the high efficiency in both impellers. However, in the Case 1 the simulation using the dry gas curve reported the highest efficiencies and hence the lowest total compressor power. A direct consequence of no correcting the efficiency curve is that, as in this case, the compressor model will report higher efficiencies at wet gas conditions which is inconsistent with experimental tests (Brenne, Bjørge et al., 2005; Hundseid, 2008)

Table 6.9 Case 2: Steady-state performance values

| | First impeller | | | Second impeller | | | Total compressor power | | |
|---------------|----------------|--------|----------|-----------------|--------|----------|------------------------|------------|-----------|
| | \dot{m} | H_p | η_p | \dot{m} | H_p | η_p | HYSYS | Calculated | Deviation |
| | kg/s | m | % | kg/s | m | % | MW | MW | % |
| Initial | 144.74 | 1406.2 | 78.21 | 137.50 | 1332.3 | 76.35 | 4.91 | 5.00 | 1.83 |
| No correction | 156.43 | 1178.3 | 77.50 | 148.61 | 1159.1 | 76.95 | 4.53 | 4.62 | 1.99 |
| IC | 156.43 | 1116.1 | 70.48 | 148.62 | 1228.7 | 74.20 | 4.85 | 4.94 | 1.86 |
| WC | 156.43 | 1131.1 | 78.9 | 148.61 | 1204.6 | 78.34 | 4.45 | 4.53 | 1.80 |

Additionally, like in the Case 1, it is possible to plot the three characteristic curves together in order to check that the correction routine actually worked and corrected the curves. The

Figure 6.17 and Figure 6.18 display the characteristic curve of the first impeller after the steady-state has been reached. It can be noted the small difference between the curve corrected by the Wood’s method and the curve at dry gas conditions. This is product of the low density of the condensate stream (442 kg/m^3) which makes the GMF to be closer to the unity than in the case of the water stream. The GMF in the Case 1 was 0.79 while in the case 2 was 0.91, at the suction conditions of the first impeller. The similar effect will be found at the inlet of the second impeller.

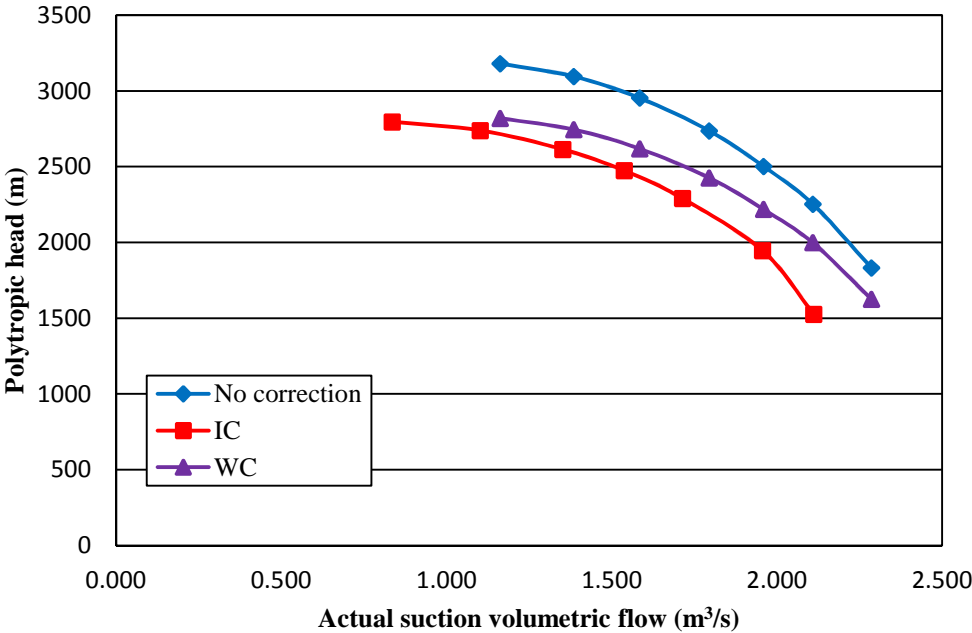


Figure 6.17 Case 2: Corrected first impeller characteristic head curve

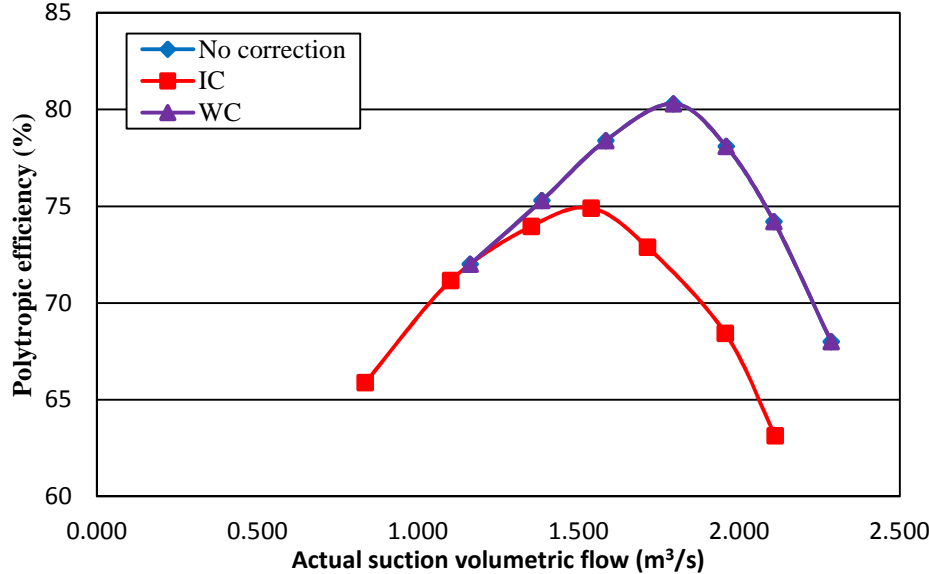


Figure 6.18 Case 2: Corrected first impeller characteristic efficiency curve

Likewise, the full compressor map with the operating point trajectory can be plotted. As explained before, each simulation will generate a different compressor map and operating point trajectory. As illustration, the compressor map of the first impeller at dry gas conditions is shown in the Figure 6.19.

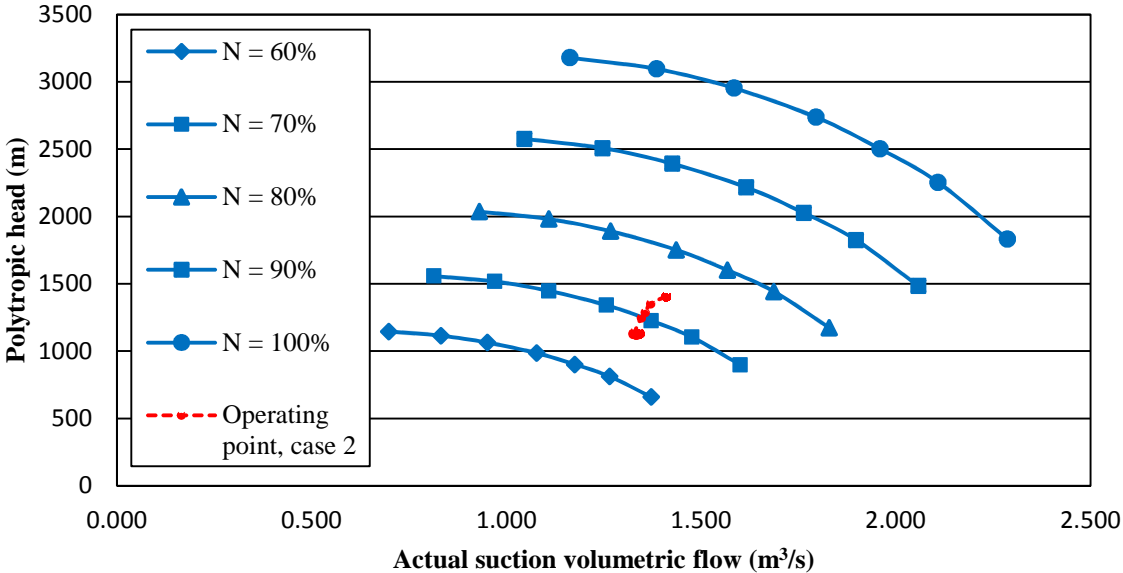


Figure 6.19 Case 2: First impeller compressor map without correction

From the compressor map it can be noticed the small change in volumetric flow which is product of the low density of the condensate stream. As consequence, the compressor speed is reduced but in small percentage and thus the final operating speeds for this case are higher than in the Case 1. This can be also illustrated when comparing the Figure 6.14 with the Figure 6.5 and Figure 6.8.

Once more, the operating point remains distant from the surge area. A similar performance will be found for the simulations using the correction methods.

6.4 Case 3: Suction pressure impact

During normal production it is possible to have sudden pressure drops in the compressor suction side. Therefore, a Sequence was set up in the Even Scheduler in order to study the system behavior under these situations. The characteristics of this case study are summarized in the Table 6.10.

Table 6.10 Case 3: main case parameters

| | |
|---------------------------|---|
| Action | Reduce inlet suction pressure from 90 to 80 bar |
| Adjusted variable | SGCS Inlet pressure specification |
| Monitored variable | Compressor speed Pipeline inlet pressure Total compressor power Suction pressure |

It is important to note that in this case scenario was executed under dry gas conditions hence the correction methods are not carried out. Please note that for better appreciation the figures' scale is not the same as in the previous cases.

The monitored variables, compressor speed, pipeline inlet pressure and total compressor power are shown in the Figure 6.20, Figure 6.21 and Figure 6.22, respectively.

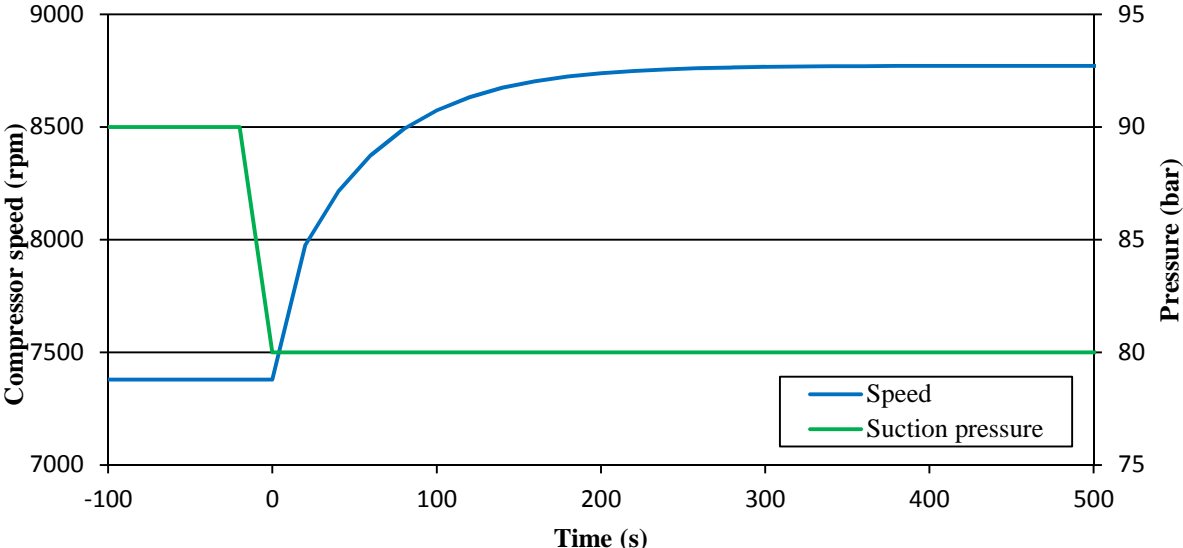


Figure 6.20 Case 3: Compressor speed profile

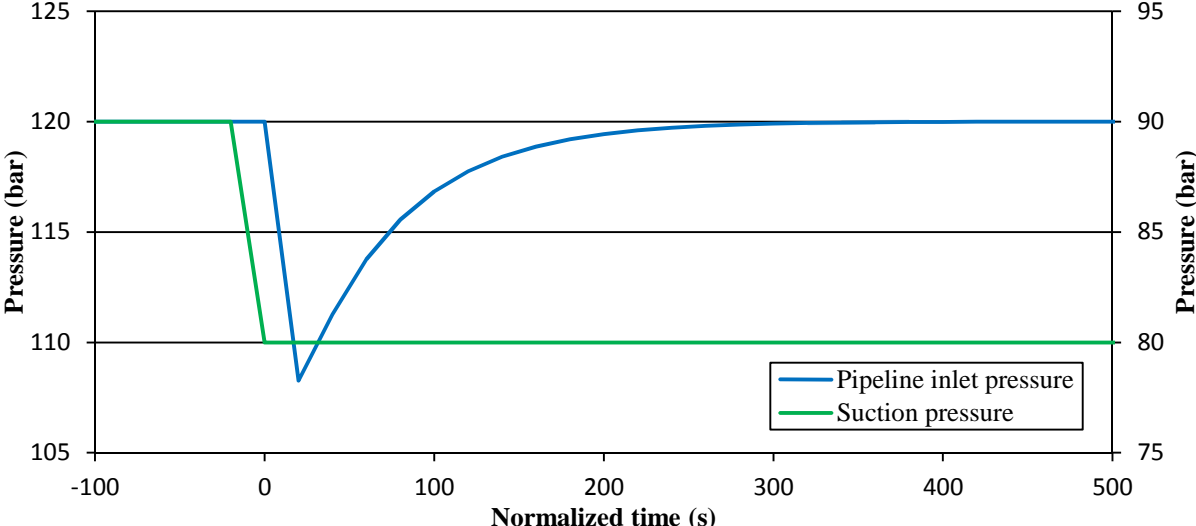


Figure 6.21 Case 3: Pipeline inlet pressure profile

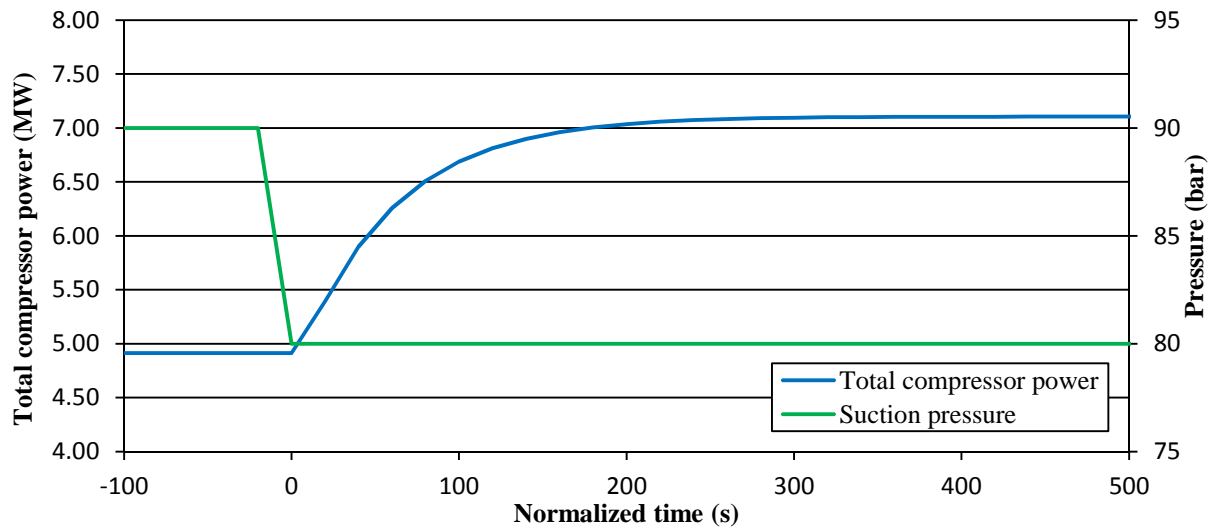


Figure 6.22 Case 3: Total compressor power profile

The Table 6.11 summarizes the steady-state compression performance values.

Table 6.11 Case 3: Steady-state performance values

| | First impeller | | | Second impeller | | | Total compressor power | | |
|---------|----------------|--------|----------|-----------------|--------|----------|------------------------|------------|-----------|
| | \dot{m} | H_p | η_p | \dot{m} | H_p | η_p | HYSYS | Calculated | Deviation |
| | kg/s | m | % | kg/s | m | % | MW | MW | % |
| Initial | 144.74 | 1406.3 | 78.21 | 137.50 | 1332.2 | 76.35 | 4.91 | 5.00 | 1.83 |
| Final | 144.74 | 2033.1 | 79.22 | 137.50 | 2025.3 | 79.13 | 7.10 | 7.23 | 1.83 |

7 Conclusions

The HYSYS dynamic functionalities were explored in order to build a dynamic wet gas compressor model. In particular, it was of interest the use of the Automation feature to implement two different correction methods that accounts for wet gas impact on compression performance, in dynamic-state. This was done through the creation of a VBA script in Microsoft Excel. The HYSYS dynamic model and the VBA script were used to explore the performance of the compressor and the system under different operating conditions including wet gas.

Both developed correction methods, interpolation correction and Wood's correction, proved to be fast enough to take into account changes in GVF. Even sudden changes in GVF are dandled effectively and corrections are made faster than the controller's action.

The interaction between the VBA script and the HYSYS flow sheet is dependent on the defined tolerance ϵ . A high tolerance value produces stability problems on the compressor performance parameters and limits the real time monitoring of the process.

Since the Wood's correction uses the GMF as correction parameter it is more sensible for changes in the fluid conditions than the interpolation method that only uses the GVF. As consequence, the Wood's method could take more time to reach the steady-state.

The total compressor power decreases when the GVF decreases. This is product of the resulting reduction in polytropic head and mass flow rate. The power reduction is greater when water is used to control the GVF at the inlet of the compressor than when condensate is used. This is due to the high molecular weight of the condensate which supposes an increment in the mass flow.

The interpolation correction method reported, in all cases, the highest compressor power due to the polytropic efficiency is corrected and lowered based on typical experimental compressor behavior.

The pipeline inlet pressure signal registered a maximum overshoot of 10%. This value seems acceptable meaning the good tuning of the pressure controller.

The system showed to be more stable (low overshoot in pressure signal) for fluids of high molecular weight.

A direct consequence and major drawback of the Wood's correction method is that in some cases the compressor model will report higher efficiencies at wet gas conditions than a dry gas conditions, which is inconsistent with experimental tests (Brenne, Bjørge et al., 2005; Hundseid, 2008).

8 Recommendations

The following recommendations can be made based on the obtained results:

- Include a complete piping system of the compression system. This will allow having more reliable representation of the compressor and system response in dynamic-state. The piping system is particular for each specific case (Åsgard, Ormen Lang, Gulfaks) and actual data is needed.
- It will be also of interest to include more information of the compressor-motor system: compressor speed-torque curve, motor inertia, friction loss models, etc. This will permit to study the compressor transient rundown behavior. This issue is well addressed by (Tveit, Bakken et al., 2005).
- It is necessary to study further the potential impact of the compressor outlet pressure signal fluctuation. The simulations showed that the two impellers are operating far from the surge area then the magnetic bearings should be able to handle this pressure fluctuation. If deterioration occurs due to vibrations, it could be product of improper compressor design.

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Appendices

Appendix A Impellers characteristic curves

Appendix B Correction methods

Appendix C Interpolation procedure based on experimental data

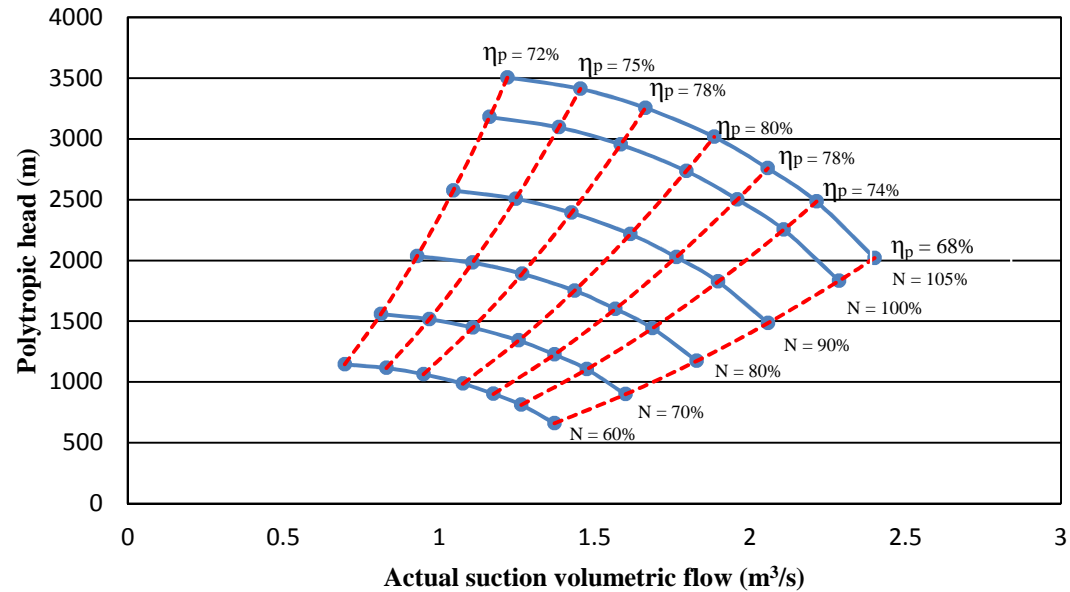
Appendix D VBA Scripts

Appendix E Gas and condensate compositions

Appendix A Impellers characteristic curves

First Impeller

| Characteristic curve | | |
|-----------------------|--------------------|--------------------|
| N = 100% (10 000 rpm) | | |
| Q (m ³ /s) | H _p (m) | η _p (%) |
| 1.163 | 3179.4 | 72.00 |
| 1.386 | 3094.9 | 75.30 |
| 1.585 | 2952.5 | 78.40 |
| 1.795 | 2736.0 | 80.30 |
| 1.960 | 2501.4 | 78.10 |
| 2.109 | 2253.0 | 74.20 |
| 2.286 | 1832.1 | 68.00 |

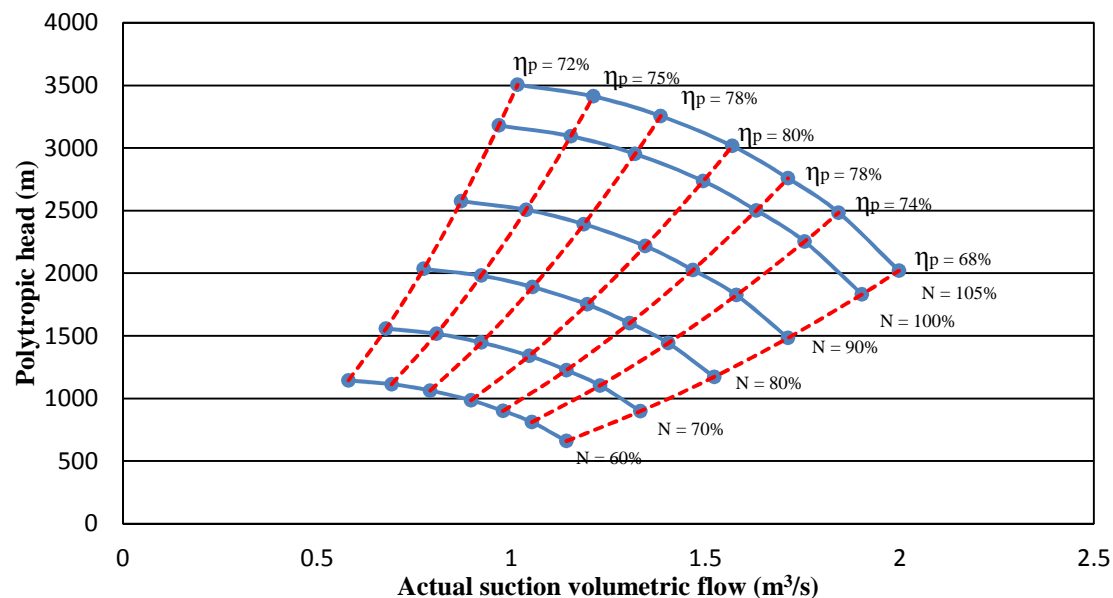


The full compressor map is generated by applying the fan laws:

| N = 60 % | | | N = 70 % | | | N = 80 % | | | N = 90 % | | | N = 105 % | | |
|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|
| Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) |
| 0.698 | 1144.6 | 72.00 | 0.814 | 1557.9 | 72.00 | 0.930 | 2034.8 | 72.00 | 1.047 | 2575.3 | 72.00 | 1.2209 | 3505.3 | 72.00 |
| 0.831 | 1114.1 | 75.30 | 0.970 | 1516.5 | 75.30 | 1.109 | 1980.7 | 75.30 | 1.247 | 2506.8 | 75.30 | 1.4550 | 3412.1 | 75.30 |
| 0.951 | 1062.9 | 78.40 | 1.109 | 1446.7 | 78.40 | 1.268 | 1889.6 | 78.40 | 1.426 | 2391.5 | 78.40 | 1.6638 | 3255.2 | 78.40 |
| 1.077 | 985.0 | 80.30 | 1.257 | 1340.7 | 80.30 | 1.436 | 1751.1 | 80.30 | 1.616 | 2216.2 | 80.30 | 1.8852 | 3016.5 | 80.30 |
| 1.176 | 900.5 | 78.10 | 1.372 | 1225.7 | 78.10 | 1.568 | 1600.9 | 78.10 | 1.764 | 2026.1 | 78.10 | 2.0575 | 2757.8 | 78.10 |
| 1.265 | 811.1 | 74.20 | 1.476 | 1104.0 | 74.20 | 1.687 | 1441.9 | 74.20 | 1.898 | 1824.9 | 74.20 | 2.2141 | 2483.9 | 74.20 |
| 1.372 | 659.5 | 68.00 | 1.600 | 897.7 | 68.00 | 1.829 | 1172.5 | 68.00 | 2.058 | 1484.0 | 68.00 | 2.4007 | 2019.9 | 68.00 |

Second impeller

| Characteristic curve | | |
|-----------------------|--------------------|--------------------|
| N = 100% (10 000 rpm) | | |
| Q (m ³ /s) | H _p (m) | η _p (%) |
| 0.968 | 3179.4 | 72.00 |
| 1.153 | 3094.9 | 75.30 |
| 1.319 | 2952.5 | 78.40 |
| 1.494 | 2736.0 | 80.30 |
| 1.631 | 2501.4 | 78.10 |
| 1.755 | 2253.0 | 74.20 |
| 1.903 | 1832.1 | 68.00 |



The full compressor map is generated by applying the fan laws:

| 60 % | | | 70 % | | | 80 % | | | 90 % | | | 105 % | | |
|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|-----------------------|--------------------|--------------------|
| Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) | Q (m ³ /s) | H _p (m) | η _p (%) |
| 0.581 | 1144.6 | 72.00 | 0.677 | 1557.9 | 72 | 0.774 | 2034.8 | 72 | 0.871 | 2575.3 | 72 | 1.0162 | 3505.3 | 72 |
| 0.692 | 1114.1 | 75.30 | 0.807 | 1516.5 | 75.3 | 0.923 | 1980.7 | 75.3 | 1.038 | 2506.8 | 75.3 | 1.2110 | 3412.1 | 75.3 |
| 0.791 | 1062.9 | 78.40 | 0.923 | 1446.7 | 78.4 | 1.055 | 1889.6 | 78.4 | 1.187 | 2391.5 | 78.4 | 1.3847 | 3255.2 | 78.4 |
| 0.897 | 985.0 | 80.30 | 1.046 | 1340.7 | 80.3 | 1.195 | 1751.1 | 80.3 | 1.345 | 2216.2 | 80.3 | 1.5690 | 3016.5 | 80.3 |
| 0.979 | 900.5 | 78.10 | 1.142 | 1225.7 | 78.1 | 1.305 | 1600.9 | 78.1 | 1.468 | 2026.1 | 78.1 | 1.7125 | 2757.8 | 78.1 |
| 1.053 | 811.1 | 74.20 | 1.229 | 1104.0 | 74.2 | 1.404 | 1441.9 | 74.2 | 1.580 | 1824.9 | 74.2 | 1.8428 | 2483.9 | 74.2 |
| 1.142 | 659.5 | 68.00 | 1.332 | 897.7 | 68 | 1.522 | 1172.5 | 68 | 1.713 | 1484.0 | 68 | 1.9981 | 2019.9 | 68 |

Appendix B Interpolation procedure based on experimental data

Curves from experimental data:

| Point | GVF = 1, N = 100% | | |
|-------|-----------------------|--------------------|--------------------|
| | Q (m ³ /s) | H _p (m) | η _p (%) |
| 1 | 1.158 | 9215.0 | 72.0 |
| 2 | 1.38 | 8970.0 | 75.3 |
| 3 | 1.578 | 8557.5 | 78.4 |
| 4 | 1.788 | 7930.0 | 80.3 |
| 5 | 1.9515 | 7250.0 | 78.1 |
| 6 | 2.1 | 6530.0 | 74.2 |
| 7 | 2.277 | 5310.0 | 68.0 |

| GVF = 0.95, N = 100% | | |
|-----------------------|--------------------|--------------------|
| Q (m ³ /s) | H _p (m) | η _p (%) |
| 0.615 | 7360.0 | 61.8 |
| 0.909 | 7250.0 | 68.4 |
| 1.1925 | 6920.0 | 71.0 |
| 1.362 | 6660.0 | 71.3 |
| 1.545 | 6225.0 | 69.4 |
| 1.848 | 5045.0 | 64.6 |
| 1.9875 | 3825.0 | 59.9 |

A performance ratio is calculated for each point of the characteristic curve. The ratio is defined as the difference from the point at GVF = 0.95 to the point at GVF = 1. As example, the volume flow ratio for the 1st point would be:

$$Q_{ratio,1} = 1 - \frac{Q_{GVF=1,1} - Q_{GVF=0.95,1}}{Q_{GVF=1,1}} = \frac{Q_{GVF=0.95,1}}{Q_{GVF=1,1}}$$

This ratio is defined in the same way for the polytropic head and efficiency. Hence, it is possible to obtain a table with the performance ratios for each point:

| Performance ratios | | |
|--------------------|---------------------------|---------------------------|
| Wet Q_ratio | Wet H _p _ratio | Wet η _p _ratio |
| 0.531 | 0.799 | 0.858 |
| 0.659 | 0.808 | 0.908 |
| 0.756 | 0.809 | 0.906 |
| 0.762 | 0.840 | 0.888 |
| 0.792 | 0.859 | 0.889 |
| 0.880 | 0.773 | 0.871 |
| 0.873 | 0.720 | 0.881 |

Once the interpolation factors are determined, it is possible to multiply them, point by point, to the impeller characteristic curve at GVF = 1 and obtain the corresponding curve at GVF = 0.95. As example, for the first impeller this would be:

| GVF = 1, N = 100% | | | |
|--------------------------|----------------------------|--------------------------|--------------------------|
| Point | Q (m³/s) | H_p (m) | η_p (%) |
| 1 | 1.163 | 3179.4 | 72.00 |
| 2 | 1.386 | 3094.9 | 75.30 |
| 3 | 1.585 | 2952.5 | 78.40 |
| 4 | 1.795 | 2736.0 | 80.30 |
| 5 | 1.960 | 2501.4 | 78.10 |
| 6 | 2.109 | 2253.0 | 74.20 |
| 7 | 2.286 | 1832.1 | 68.00 |



| GVF = 0.95, N = 100% | | |
|-----------------------------|--------------------------|--------------------------|
| Q (m³/s) | H_p (m) | η_p (%) |
| 0.615 | 7360.0 | 61.8 |
| 0.909 | 7250.0 | 68.4 |
| 1.1925 | 6920.0 | 71.0 |
| 1.362 | 6660.0 | 71.3 |
| 1.545 | 6225.0 | 69.4 |
| 1.848 | 5045.0 | 64.6 |
| 1.9875 | 3825.0 | 59.9 |

Appendix C Correction methods

C.1 Interpolation correction method

In this procedure the GVF is used to interpolate the characteristic curve at any given GVF within the range $GVF = 0.95$ and $GVF = 1$. For this, an interpolation factor is defined based on the GVF:

$$IF = 20 \times GVF - 19$$

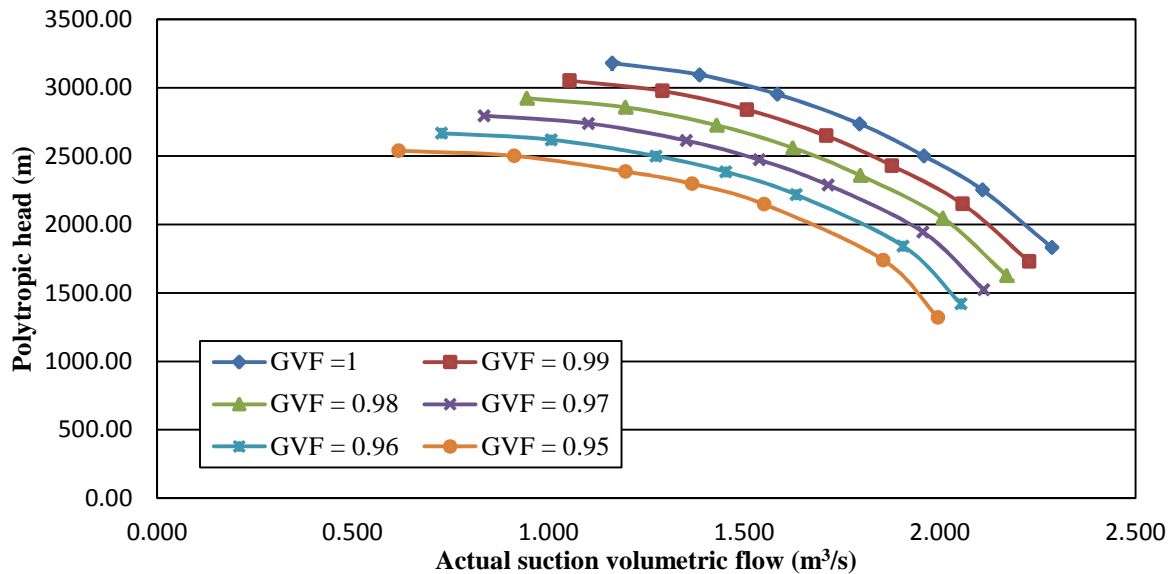
The interpolation factor is then used to interpolate point by point between the curve at $GVF = 0.95$ and the curve at $GVF = 1$:

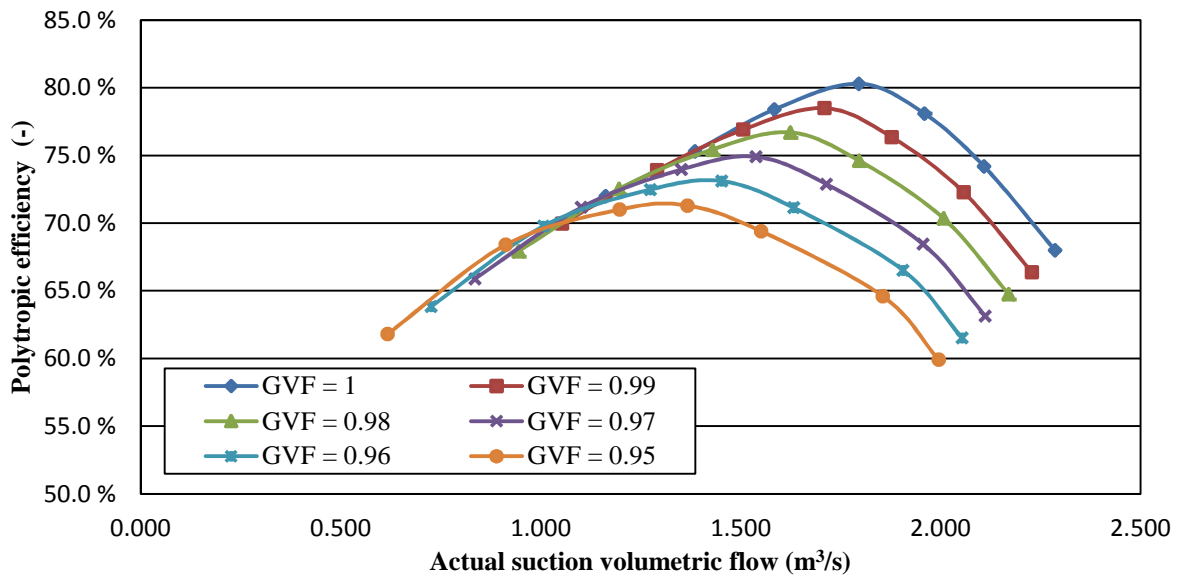
$$Q_{GVF=x} = Q_{GVF=0.95} + IF \cdot (Q_{GVF=1} - Q_{GVF=0.95})$$

$$H_{p_{GVF=x}} = H_{p_{GVF=0.95}} + IF \cdot (H_{p_{GVF=1}} - H_{p_{GVF=0.95}})$$

$$\eta_{GVF=x} = \eta_{GVF=0.95} + IF \cdot (\eta_{GVF=1} - \eta_{GVF=0.95})$$

Hence, as shown in the figures below, it is possible to generate a new impeller characteristic curve for any GVF value.





C.2 Wood's correction method

First, the input parameters are defined:

$$\begin{aligned} \text{GVF} &= 0.97 \\ \rho_g &= 105 \quad \text{kg/m}^3 \\ \rho_l &= 1000 \quad \text{kg/m}^3 \end{aligned}$$

The f and W are calculated by the Eq. 5.5 and 5.6:

$$f = \frac{[0.97 \cdot 105 + (1 - 0.97) \cdot 1000]}{105} = 1.256$$

$$W = \frac{1}{1.256 \cdot \sqrt{1.256 \cdot 0.97}} = 0.7216$$

Then, each point of the characteristic head curve is multiplied by W :

| | GVF = 1, N = 100% | | | | GVF = 0.97, N = 100% | | |
|-------|-----------------------|--------------------|--------------------|---|-----------------------|--------------------|--------------------|
| Point | Q (m ³ /s) | H _p (m) | η _p (%) | | Q (m ³ /s) | H _p (m) | η _p (%) |
| 1 | 1.163 | 3179.4 | 72.0 | → | 1.163 | 2294.14 | 72.0 |
| 2 | 1.386 | 3094.9 | 75.3 | | 1.386 | 2233.15 | 75.3 |
| 3 | 1.585 | 2952.5 | 78.4 | | 1.585 | 2130.45 | 78.4 |
| 4 | 1.795 | 2736.0 | 80.3 | | 1.795 | 1974.23 | 80.3 |
| 5 | 1.960 | 2501.4 | 78.1 | | 1.960 | 1804.94 | 78.1 |
| 6 | 2.109 | 2253.0 | 74.2 | | 2.109 | 1625.69 | 74.2 |
| 7 | 2.286 | 1832.1 | 68.0 | | 2.286 | 1321.96 | 68.0 |

Appendix D VBA Scripts

D.1 Interpolation correction method

```
Sub Simulation()
```

```
' Defines the time vectors
```

```
Dim deltatime() As Long  
Dim deltatimef() As Long  
Dim j As Integer  
Dim i As Integer  
Dim a As Integer  
Dim b As Integer
```

```
' Hides the user dialog box
```

```
UserForm1.Hide
```

```
' Creates an interface between CURRENT HYSYS file and the Script
```

```
Set hyApp = CreateObject("HYSYS.Application")  
Set hyCase = hyApp.ActiveDocument
```

```
' Connects to the Case Flowsheet
```

```
Set hyFlowsheet = hyCase.Flowsheet
```

```
' Connects to Integrator
```

```
Set hySolver = hyCase.Solver  
Set hyIntegrator = hySolver.Integrator
```

```
' Reads the current simulation time value (time_new)
```

```
Set hycr = hyIntegrator.CurrentTime  
time_new = hycr.Value
```

```
Debug.Print ("The current/starting simulation time is = ") & time_new & (" seconds")
```

```
' Activates the simulation
```

```
Let hyIntegrator.IsRunning = True
```

```
time_old = 0
```

```
' The connections with the compressors and spreadsheets is done after the simulation is started and  
outside the While Do Loop
```

```
'===== C100 & SS1
```

```
' Connects to the compressor C100
```

```
Set Comp1 = hyFlowsheet.Operations.Item("C100")
```

```
' Connects to the Characteristic Curves of C100
```

```
Set curvesC1 = Comp1.Curves()  
Set C10 = curvesC1.Item("60%")  
Set C11 = curvesC1.Item("70%")  
Set C12 = curvesC1.Item("80%")
```

```
Set C13 = curvesC1.Item("90%")
Set C14 = curvesC1.Item("100%")
Set C15 = curvesC1.Item("105%")
```

```
' Connects to the Spreadsheet: Curves1stg
```

```
Set hySS1 = hyFlowsheet.Operations.Item("Curves1stg")
```

```
'Connects and reads value of cell F2 (Target IF) of the SS1
```

```
Set IF_SS1 = hySS1.Cell(5, 1)
```

```
IF_SS1_target = IF_SS1.CellValue
```

```
'Connects to cells B5 - B11 of the SS1, Volumetric Flow values:
```

```
Set Q1_SS1 = hySS1.Cell(1, 4)
```

```
Set Q2_SS1 = hySS1.Cell(1, 5)
```

```
Set Q3_SS1 = hySS1.Cell(1, 6)
```

```
Set Q4_SS1 = hySS1.Cell(1, 7)
```

```
Set Q5_SS1 = hySS1.Cell(1, 8)
```

```
Set Q6_SS1 = hySS1.Cell(1, 9)
```

```
Set Q7_SS1 = hySS1.Cell(1, 10)
```

```
'Connects to cells C5 - C11 of the SS1, Polytropic Head values:
```

```
Set Hp1_SS1 = hySS1.Cell(2, 4)
```

```
Set Hp2_SS1 = hySS1.Cell(2, 5)
```

```
Set Hp3_SS1 = hySS1.Cell(2, 6)
```

```
Set Hp4_SS1 = hySS1.Cell(2, 7)
```

```
Set Hp5_SS1 = hySS1.Cell(2, 8)
```

```
Set Hp6_SS1 = hySS1.Cell(2, 9)
```

```
Set Hp7_SS1 = hySS1.Cell(2, 10)
```

```
'Connects to cells D5 - D11 of the SS1, Polytropic Efficiency values:
```

```
Set Etap1_SS1 = hySS1.Cell(3, 4)
```

```
Set Etap2_SS1 = hySS1.Cell(3, 5)
```

```
Set Etap3_SS1 = hySS1.Cell(3, 6)
```

```
Set Etap4_SS1 = hySS1.Cell(3, 7)
```

```
Set Etap5_SS1 = hySS1.Cell(3, 8)
```

```
Set Etap6_SS1 = hySS1.Cell(3, 9)
```

```
Set Etap7_SS1 = hySS1.Cell(3, 10)
```

```
'===== C200 & SS2
```

```
' Connects to the compressor C200
```

```
Set Comp2 = hyFlowsheet.Operations.Item("C200")
```

```
' Connects to the Characteristic Curves of C200
```

```
Set curvesC2 = Comp2.Curves()
```

```
Set C20 = curvesC2.Item("60%")
```

```
Set C21 = curvesC2.Item("70%")
```

```
Set C22 = curvesC2.Item("80%")
```

```
Set C23 = curvesC2.Item("90%")
```

```
Set C24 = curvesC2.Item("100%")
```

```
Set C25 = curvesC2.Item("105%")
```

```
' Connects to the Spreadsheet: Curves2stg
```

```
Set hySS2 = hyFlowsheet.Operations.Item("Curves2stg")
```

'Connects and reads value of cell F2 (Target IF) of the SS2

```
Set IF_SS2 = hySS2.Cell(5, 1)
IF_SS2_target = IF_SS2.CellValue
```

'Connects to cells B5 - B11 of the SS2, Volumetric Flow values:

```
Set Q1_SS2 = hySS2.Cell(1, 4)
Set Q2_SS2 = hySS2.Cell(1, 5)
Set Q3_SS2 = hySS2.Cell(1, 6)
Set Q4_SS2 = hySS2.Cell(1, 7)
Set Q5_SS2 = hySS2.Cell(1, 8)
Set Q6_SS2 = hySS2.Cell(1, 9)
Set Q7_SS2 = hySS2.Cell(1, 10)
```

'Connects to cells C5 - C11 of the SS2, Polytropic Head values:

```
Set Hp1_SS2 = hySS2.Cell(2, 4)
Set Hp2_SS2 = hySS2.Cell(2, 5)
Set Hp3_SS2 = hySS2.Cell(2, 6)
Set Hp4_SS2 = hySS2.Cell(2, 7)
Set Hp5_SS2 = hySS2.Cell(2, 8)
Set Hp6_SS2 = hySS2.Cell(2, 9)
Set Hp7_SS2 = hySS2.Cell(2, 10)
```

'Connects to cells D5 - D11 of the SS2, Polytropic Efficiency values:

```
Set Etap1_SS2 = hySS2.Cell(3, 4)
Set Etap2_SS2 = hySS2.Cell(3, 5)
Set Etap3_SS2 = hySS2.Cell(3, 6)
Set Etap4_SS2 = hySS2.Cell(3, 7)
Set Etap5_SS2 = hySS2.Cell(3, 8)
Set Etap6_SS2 = hySS2.Cell(3, 9)
Set Etap7_SS2 = hySS2.Cell(3, 10)
```

'Re-define time vector

```
j = 0
ReDim deltatime(0 To j) As Long
```

'----Do While loop

'Beginning of correction method iteration

'Variables to count number of changes made in each curve

```
a = 0
b = 0
```

Do While UserForm1.press.Value = True

' Shows the user dialog box

```
UserForm1.Show vbModeless
```

```
deltatime(j) = time_new - time_old
```

```
Set hycr = hyIntegrator.CurrentTime
time_old = hycr.Value
```

' Calculates difference between Interpolation Factor of C100


```

IF_SS1_current = IF_SS1.CellValue
'Debug.Print ("The current Interpolation Factor of C100 is = ") & IF_SS1_current

Delta_IF1 = Abs(IF_SS1_current - IF_SS1_target)
'Debug.Print ("The difference between the Interpolation Factors of C100 is = ") & Delta_IF1

If Delta_IF1 >= 0.01 Then

    'Holds the simulation
    Let hyIntegrator.IsRunning = False

    'Assigns new value for IF target value of C100
    IF_SS1_target = IF_SS1_current

    'Assigns the GVF value from HYSYS spreadsheet to Excel spreadsheet
    Worksheets("First impeller").[D61].Value = hySS1.Cell(1, 0).CellValue

    Debug.Print ("The Interpolation Factor of C100 will be re-calculated")

    'Holds the simulation
    Let hyIntegrator.IsRunning = False

    'Sets new Q values for C100 Characteristic curve, at given GVF
    Let Q1_SS1.CellValue = Worksheets("First impeller").[C76].Value
    Let Q2_SS1.CellValue = Worksheets("First impeller").[C77].Value
    Let Q3_SS1.CellValue = Worksheets("First impeller").[C78].Value
    Let Q4_SS1.CellValue = Worksheets("First impeller").[C79].Value
    Let Q5_SS1.CellValue = Worksheets("First impeller").[C80].Value
    Let Q6_SS1.CellValue = Worksheets("First impeller").[C81].Value
    Let Q7_SS1.CellValue = Worksheets("First impeller").[C82].Value

    'Set new Hp values for C100 Characteristic curve, at given GVF
    Let Hp1_SS1.CellValue = Worksheets("First impeller").[D76].Value
    Let Hp2_SS1.CellValue = Worksheets("First impeller").[D77].Value
    Let Hp3_SS1.CellValue = Worksheets("First impeller").[D78].Value
    Let Hp4_SS1.CellValue = Worksheets("First impeller").[D79].Value
    Let Hp5_SS1.CellValue = Worksheets("First impeller").[D80].Value
    Let Hp6_SS1.CellValue = Worksheets("First impeller").[D81].Value
    Let Hp7_SS1.CellValue = Worksheets("First impeller").[D82].Value

    'Sets new Etab values for C100 Characteristic curve, at given GVF
    Let Etab1_SS1.CellValue = Worksheets("First impeller").[E76].Value
    Let Etab2_SS1.CellValue = Worksheets("First impeller").[E77].Value
    Let Etab3_SS1.CellValue = Worksheets("First impeller").[E78].Value
    Let Etab4_SS1.CellValue = Worksheets("First impeller").[E79].Value
    Let Etab5_SS1.CellValue = Worksheets("First impeller").[E80].Value
    Let Etab6_SS1.CellValue = Worksheets("First impeller").[E81].Value
    Let Etab7_SS1.CellValue = Worksheets("First impeller").[E82].Value

    ' Activates ALL the curves of C100
    C10.ActivateCurve = True
    C11.ActivateCurve = True
    C12.ActivateCurve = True
    C13.ActivateCurve = True
    C14.ActivateCurve = True

```

```
C15.ActivateCurve = True
Comp1.UsingCurves = True
```

```
'Sets the correct value for the Constant C in the anti-surge controller
```

```
'The Excel cell E94 is exported to the cell B24 in the HYSYS SS1
```

```
Let hySS1.Cell(1, 23).CellValue = Worksheets("First impeller").[E94].Value
```

```
'Activates again the curves to avoid problems
```

```
C10.ActivateCurve = True
```

```
C11.ActivateCurve = True
```

```
C12.ActivateCurve = True
```

```
C13.ActivateCurve = True
```

```
C14.ActivateCurve = True
```

```
C15.ActivateCurve = True
```

```
Comp1.UsingCurves = True
```

```
'Activates the simulation
```

```
Let hyIntegrator.IsRunning = True
```

```
a = a + 1
```

```
Else
```

```
'Debug.Print ("The Interpolation Factor of C100 will not change")
```

```
End If
```

```
' Calculates difference between Interpolation Factor of C200
```

```
IF_SS2_current = IF_SS2.CellValue
```

```
'Debug.Print ("The current Interpolation Factor of C200 is = ") & IF_SS2_current
```

```
Delta_IF2 = Abs(IF_SS2_current - IF_SS2_target)
```

```
'Debug.Print ("The difference between the Interpolation Factors of C200 is = ") & Delta_IF2
```

```
If Delta_IF2 >= 0.01 Then
```

```
'Holds the simulation
```

```
Let hyIntegrator.IsRunning = False
```

```
Debug.Print ("The Interpolation Factor of C200 will be re-calculated")
```

```
'Assigns new value for IF target value of C200
```

```
IF_SS2_target = IF_SS2_current
```

```
'Assigns the GVF value from HYSYS spreadsheet to Excel spreadsheet
```

```
Worksheets("Second impeller").[D61].Value = hySS2.Cell(1, 0).CellValue
```

```
'Holds the simulation
```

```
Let hyIntegrator.IsRunning = False
```

```
'Sets new Q values for C200 Characteristic curve, at given GVF
```

```
Let Q1_SS2.CellValue = Worksheets("Second impeller").[C76].Value
```

```
Let Q2_SS2.CellValue = Worksheets("Second impeller").[C77].Value
```

```
Let Q3_SS2.CellValue = Worksheets("Second impeller").[C78].Value
```

```
Let Q4_SS2.CellValue = Worksheets("Second impeller").[C79].Value
Let Q5_SS2.CellValue = Worksheets("Second impeller").[C80].Value
Let Q6_SS2.CellValue = Worksheets("Second impeller").[C81].Value
Let Q7_SS2.CellValue = Worksheets("Second impeller").[C82].Value
```

'Set new Hp values for C200 Characteristic curve, at given GVF

```
Let Hp1_SS2.CellValue = Worksheets("Second impeller").[D76].Value
Let Hp2_SS2.CellValue = Worksheets("Second impeller").[D77].Value
Let Hp3_SS2.CellValue = Worksheets("Second impeller").[D78].Value
Let Hp4_SS2.CellValue = Worksheets("Second impeller").[D79].Value
Let Hp5_SS2.CellValue = Worksheets("Second impeller").[D80].Value
Let Hp6_SS2.CellValue = Worksheets("Second impeller").[D81].Value
Let Hp7_SS2.CellValue = Worksheets("Second impeller").[D82].Value
```

'Sets new E_{tap} values for C200 Characteristic curve, at given GVF

```
Let Etap1_SS2.CellValue = Worksheets("Second impeller").[E76].Value
Let Etap2_SS2.CellValue = Worksheets("Second impeller").[E77].Value
Let Etap3_SS2.CellValue = Worksheets("Second impeller").[E78].Value
Let Etap4_SS2.CellValue = Worksheets("Second impeller").[E79].Value
Let Etap5_SS2.CellValue = Worksheets("Second impeller").[E80].Value
Let Etap6_SS2.CellValue = Worksheets("Second impeller").[E81].Value
Let Etap7_SS2.CellValue = Worksheets("Second impeller").[E82].Value
```

' Activates ALL the curves of C200

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

'Sets the correct value for the Constant C in the anti-surge controller

'The Excel cell E94 is exported to the cell B24 in the HYSYS SS2

```
Let hySS2.Cell(1, 23).CellValue = Worksheets("Second impeller").[E94].Value
```

'Activates again the curves to avoid problems

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

'Activates the simulation

```
Let hyIntegrator.IsRunning = True
```

```
b = b + 1
```

```
Else
```

```
'Debug.Print ("The Interpolation Factor of C200 will not change")
```

```
End If
```

```
'Reads the new simulation time value (time_new)
Set hycr = hyIntegrator.CurrentTime
time_new = hycr.Value
```

```
j = j + 1
ReDim Preserve deltatime(0 To j) As Long
```

```
'Holds the simulation
Let hyIntegrator.IsRunning = False
```

```
'Activates ALL the curves of C100 and C200
```

```
C10.ActivateCurve = True
C11.ActivateCurve = True
C12.ActivateCurve = True
C13.ActivateCurve = True
C14.ActivateCurve = True
C15.ActivateCurve = True
Comp1.UsingCurves = True
```

OPTIONAL

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

```
'Activates the simulation
Let hyIntegrator.IsRunning = True
```

```
DoEvents
Loop
```

```
'Deactivates the simulation
Let hyIntegrator.IsRunning = False
```

```
'Creates a new vector eliminating the first and last value of the deltatime vector
'due to the first value would be equal to the starting simulation time
'and the last would be zero
```

```
ReDim deltatimef(0 To j - 2) As Long
```

```
For i = 1 To j - 1
deltatimef(i - 1) = deltatime(i)
Next
```

```
'Deactivates the simulation
Let hyIntegrator.IsRunning = False
```

```
Debug.Print ("The simulation has finished")
```

```
maxt = WorksheetFunction.Max(deltatimef)
Debug.Print ("Maximum correction time is ") & maxt & (" seconds")
```

```
mint = WorksheetFunction.Min(deltatimef)
Debug.Print ("Minimum correction time is ") & mint & (" seconds")

avet = WorksheetFunction.Average(deltatimef)
Debug.Print ("The average correction time is ") & avet & (" seconds")

Debug.Print ("The Curve C100 was changed ") & a & (" times")
Debug.Print ("The Curve C200 was changed ") & b & (" times")

'Deactivates the simulation
Let hyIntegrator.IsRunning = False

End Sub
```

D.2 Wood's correction method

```
Sub Simulation()
```

```
' Defines the time vectors
```

```
Dim deltatime() As Long
```

```
Dim deltatimef() As Long
```

```
Dim j As Integer
```

```
Dim i As Integer
```

```
Dim a As Integer
```

```
Dim b As Integer
```

```
' Hides the user dialog box
```

```
UserForm1.Hide
```

```
' Creates an interface between CURRENT HYSYS file and the Script
```

```
Set hyApp = CreateObject("HYSYS.Application")
```

```
Set hyCase = hyApp.ActiveDocument
```

```
' Connects to the Case Flowsheet
```

```
Set hyFlowsheet = hyCase.Flowsheet
```

```
' Connects to Integrator
```

```
Set hySolver = hyCase.Solver
```

```
Set hyIntegrator = hySolver.Integrator
```

```
' The connections with the compressors and spreadsheets is done after the simulation is started and  
outside the While Do Loop
```

```
'===== C100 & SS1
```

```
' Connects to the compressor C100
```

```
Set Comp1 = hyFlowsheet.Operations.Item("C100")
```

```
' Connects to the Characteristic Curves of C100
```

```
Set curvesC1 = Comp1.Curves()
```

```
Set C10 = curvesC1.Item("60%")
```

```
Set C11 = curvesC1.Item("70%")
```

```
Set C12 = curvesC1.Item("80%")
```

```
Set C13 = curvesC1.Item("90%")
```

```
Set C14 = curvesC1.Item("100%")
```

```
Set C15 = curvesC1.Item("105%")
```

```
' Connects to the Spreadsheet: Curves1stg
```

```
Set hySS1 = hyFlowsheet.Operations.Item("Curves1stg")
```

```
' Connects and reads value of cell D2 (Target Wood's factor) of the SS1
```

```
Set W_SS1 = hySS1.Cell(3, 1)
```

```
W_SS1_target = W_SS1.CellValue
```

```
' Connects to density values of SS1
```

```
Set rog_1 = hySS1.Cell(3, 0)
```

```
Set rol_1 = hySS1.Cell(5, 0)
```

```
'Connects to cells B5 - B11 of the SS1, Volumetric Flow values:
```

```
Set Q1_SS1 = hySS1.Cell(1, 4)  
Set Q2_SS1 = hySS1.Cell(1, 5)  
Set Q3_SS1 = hySS1.Cell(1, 6)  
Set Q4_SS1 = hySS1.Cell(1, 7)  
Set Q5_SS1 = hySS1.Cell(1, 8)  
Set Q6_SS1 = hySS1.Cell(1, 9)  
Set Q7_SS1 = hySS1.Cell(1, 10)
```

```
'Connects to cells C5 - C11 of the SS1, Polytropic Head values:
```

```
Set Hp1_SS1 = hySS1.Cell(2, 4)  
Set Hp2_SS1 = hySS1.Cell(2, 5)  
Set Hp3_SS1 = hySS1.Cell(2, 6)  
Set Hp4_SS1 = hySS1.Cell(2, 7)  
Set Hp5_SS1 = hySS1.Cell(2, 8)  
Set Hp6_SS1 = hySS1.Cell(2, 9)  
Set Hp7_SS1 = hySS1.Cell(2, 10)
```

```
'Connects to cells D5 - D11 of the SS1, Polytropic Efficiency values:
```

```
Set Etap1_SS1 = hySS1.Cell(3, 4)  
Set Etap2_SS1 = hySS1.Cell(3, 5)  
Set Etap3_SS1 = hySS1.Cell(3, 6)  
Set Etap4_SS1 = hySS1.Cell(3, 7)  
Set Etap5_SS1 = hySS1.Cell(3, 8)  
Set Etap6_SS1 = hySS1.Cell(3, 9)  
Set Etap7_SS1 = hySS1.Cell(3, 10)
```

```
'===== C200 & SS2
```

```
' Connects to the compressor C200
```

```
Set Comp2 = hyFlowsheet.Operations.Item("C200")
```

```
' Connects to the Characteristic Curves of C200
```

```
Set curvesC2 = Comp2.Curves()  
Set C20 = curvesC2.Item("60%")  
Set C21 = curvesC2.Item("70%")  
Set C22 = curvesC2.Item("80%")  
Set C23 = curvesC2.Item("90%")  
Set C24 = curvesC2.Item("100%")  
Set C25 = curvesC2.Item("105%")
```

```
' Connects to the Spreadsheet: Curves2stg
```

```
Set hySS2 = hyFlowsheet.Operations.Item("Curves2stg")
```

```
'Connects to density values of SS2
```

```
Set rog_2 = hySS2.Cell(3, 0)  
Set rol_2 = hySS2.Cell(5, 0)
```

```
'Connects to cells B5 - B11 of the SS2, Volumetric Flow values:
```

```
Set Q1_SS2 = hySS2.Cell(1, 4)  
Set Q2_SS2 = hySS2.Cell(1, 5)  
Set Q3_SS2 = hySS2.Cell(1, 6)  
Set Q4_SS2 = hySS2.Cell(1, 7)
```

```
Set Q5_SS2 = hySS2.Cell(1, 8)
Set Q6_SS2 = hySS2.Cell(1, 9)
Set Q7_SS2 = hySS2.Cell(1, 10)
```

'Connects to cells C5 - C11 of the SS2, Polytropic Head values:

```
Set Hp1_SS2 = hySS2.Cell(2, 4)
Set Hp2_SS2 = hySS2.Cell(2, 5)
Set Hp3_SS2 = hySS2.Cell(2, 6)
Set Hp4_SS2 = hySS2.Cell(2, 7)
Set Hp5_SS2 = hySS2.Cell(2, 8)
Set Hp6_SS2 = hySS2.Cell(2, 9)
Set Hp7_SS2 = hySS2.Cell(2, 10)
```

'Connects to cells D5 - D11 of the SS2, Polytropic Efficiency values:

```
Set Etap1_SS2 = hySS2.Cell(3, 4)
Set Etap2_SS2 = hySS2.Cell(3, 5)
Set Etap3_SS2 = hySS2.Cell(3, 6)
Set Etap4_SS2 = hySS2.Cell(3, 7)
Set Etap5_SS2 = hySS2.Cell(3, 8)
Set Etap6_SS2 = hySS2.Cell(3, 9)
Set Etap7_SS2 = hySS2.Cell(3, 10)
```

'====Sets the fixed values for volumetric flows and efficiencies of the compressors
'====These values are indepent of the GVF

'Sets new Q values for C100

```
Let Q1_SS1.CellValue = Worksheets("First impeller").[C40].Value
Let Q2_SS1.CellValue = Worksheets("First impeller").[C41].Value
Let Q3_SS1.CellValue = Worksheets("First impeller").[C42].Value
Let Q4_SS1.CellValue = Worksheets("First impeller").[C43].Value
Let Q5_SS1.CellValue = Worksheets("First impeller").[C44].Value
Let Q6_SS1.CellValue = Worksheets("First impeller").[C45].Value
Let Q7_SS1.CellValue = Worksheets("First impeller").[C46].Value
```

'Sets new Etap values for C100

```
Let Etap1_SS1.CellValue = Worksheets("First impeller").[E40].Value
Let Etap2_SS1.CellValue = Worksheets("First impeller").[E41].Value
Let Etap3_SS1.CellValue = Worksheets("First impeller").[E42].Value
Let Etap4_SS1.CellValue = Worksheets("First impeller").[E43].Value
Let Etap5_SS1.CellValue = Worksheets("First impeller").[E44].Value
Let Etap6_SS1.CellValue = Worksheets("First impeller").[E45].Value
Let Etap7_SS1.CellValue = Worksheets("First impeller").[E46].Value
```

' Activates ALL the curves of C100

```
C10.ActivateCurve = True
C11.ActivateCurve = True
C12.ActivateCurve = True
C13.ActivateCurve = True
C14.ActivateCurve = True
C15.ActivateCurve = True
```

```
Comp1.UsingCurves = True
```


'Sets new Q values for C200

```
Let Q1_SS2.CellValue = Worksheets("Second impeller").[C40].Value
Let Q2_SS2.CellValue = Worksheets("Second impeller").[C41].Value
Let Q3_SS2.CellValue = Worksheets("Second impeller").[C42].Value
Let Q4_SS2.CellValue = Worksheets("Second impeller").[C43].Value
Let Q5_SS2.CellValue = Worksheets("Second impeller").[C44].Value
Let Q6_SS2.CellValue = Worksheets("Second impeller").[C45].Value
Let Q7_SS2.CellValue = Worksheets("Second impeller").[C46].Value
```

'Sets new Etap values for C200

```
Let Etap1_SS2.CellValue = Worksheets("Second impeller").[E40].Value
Let Etap2_SS2.CellValue = Worksheets("Second impeller").[E41].Value
Let Etap3_SS2.CellValue = Worksheets("Second impeller").[E42].Value
Let Etap4_SS2.CellValue = Worksheets("Second impeller").[E43].Value
Let Etap5_SS2.CellValue = Worksheets("Second impeller").[E44].Value
Let Etap6_SS2.CellValue = Worksheets("Second impeller").[E45].Value
Let Etap7_SS2.CellValue = Worksheets("Second impeller").[E46].Value
```

' Activates ALL the curves of C200

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
```

```
Comp2.UsingCurves = True
```

'Reads the current simulation time value (time_new)

```
Set hycr = hyIntegrator.CurrentTime
time_new = hycr.Value
```

```
Debug.Print ("The current/starting simulation time is = ") & time_new & (" seconds")
```

'Activates the simulation

```
Let hyIntegrator.IsRunning = True
```

```
time_old = 0
```

```
j = 0
```

```
ReDim deltatime(0 To j) As Long
```

'----Do While loop

```
Do While UserForm1.press.Value = True
```

' Shows the user dialog box

```
UserForm1.Show vbModeless
```

```
deltatime(j) = time_new - time_old
```

```
Set hycr = hyIntegrator.CurrentTime
time_old = hycr.Value
```

```
' Calculates difference between Wood's Factor of C100
```

```
W_SS1_current = W_SS1.CellValue
```

```
'Debug.Print ("The current Wood's Factor of C100 is = ") & W_SS1_current
```

```
Delta_W1 = Abs(W_SS1_current - W_SS1_target)
```

```
'Debug.Print ("The difference between the Wood's Factors of C100 is = ") & Delta_W1
```

```
If Delta_W1 >= 0.01 Then
```

```
  'Holds the simulation
```

```
  Let hyIntegrator.IsRunning = False
```

```
  'Assigns new value for W target value of C100
```

```
  W_SS1_target = W_SS1_current
```

```
  'Assigns the GVF and density values from HYSYS spreadsheet to Excel spreadsheet
```

```
  Worksheets("First impeller").[D25].Value = hySS1.Cell(1, 0).CellValue
```

```
  Worksheets("First impeller").[D26].Value = rog_1.CellValue
```

```
  Worksheets("First impeller").[D27].Value = rol_1.CellValue
```

```
  Debug.Print ("The Interpolation Factor of C100 will be re-calculated")
```

```
  'Holds the simulation
```

```
  Let hyIntegrator.IsRunning = False
```

```
  'Set new Hp values for C100
```

```
  Let Hp1_SS1.CellValue = Worksheets("First impeller").[D40].Value
```

```
  Let Hp2_SS1.CellValue = Worksheets("First impeller").[D41].Value
```

```
  Let Hp3_SS1.CellValue = Worksheets("First impeller").[D42].Value
```

```
  Let Hp4_SS1.CellValue = Worksheets("First impeller").[D43].Value
```

```
  Let Hp5_SS1.CellValue = Worksheets("First impeller").[D44].Value
```

```
  Let Hp6_SS1.CellValue = Worksheets("First impeller").[D45].Value
```

```
  Let Hp7_SS1.CellValue = Worksheets("First impeller").[D46].Value
```

```
  ' Activates ALL the curves of C100
```

```
  C10.ActivateCurve = True
```

```
  C11.ActivateCurve = True
```

```
  C12.ActivateCurve = True
```

```
  C13.ActivateCurve = True
```

```
  C14.ActivateCurve = True
```

```
  C15.ActivateCurve = True
```

```
  Comp1.UsingCurves = True
```

```
  'Sets the correct value for the Constant C in the anti-surge controller
```

```
  'The Excel cell E61 is exported to the cell B24 in the HYSYS SS1
```

```
  Let hySS1.Cell(1, 23).CellValue = Worksheets("First impeller").[E61].Value
```

```
  'Activates again the curves to avoid problems
```

```
  C10.ActivateCurve = True
```

```
  C11.ActivateCurve = True
```

```
  C12.ActivateCurve = True
```

```
C13.ActivateCurve = True
C14.ActivateCurve = True
C15.ActivateCurve = True
Comp1.UsingCurves = True
```

```
'Activates the simulation
```

```
Let hyIntegrator.IsRunning = True
```

```
a = a + 1
```

```
Else
```

```
'Debug.Print ("The Wood's Factor of C100 will not change")
```

```
End If
```

```
' Calculates difference between Wood's Factor of C200
```

```
W_SS2_current = W_SS2.CellValue
```

```
'Debug.Print ("The current Wood's Factor of C200 is = ") & W_SS2_current
```

```
Delta_W2 = Abs(W_SS2_current - W_SS2_target)
```

```
'Debug.Print ("The difference between the Wood's Factors of C200 is = ") & Delta_W2
```

```
If Delta_W2 >= 0.01 Then
```

```
'Holds the simulation
```

```
Let hyIntegrator.IsRunning = False
```

```
'Assigns new value for W target value of C200
```

```
W_SS2_target = W_SS2_current
```

```
'Assigns the GVF and density values from HYSYS spreadsheet to Excel spreadsheet
```

```
Worksheets("Second impeller").[D25].Value = hySS2.Cell(1, 0).CellValue
```

```
Worksheets("Second impeller").[D26].Value = rog_2.CellValue
```

```
Worksheets("Second impeller").[D27].Value = rol_2.CellValue
```

```
Debug.Print ("The Interpolation Factor of C200 will be re-calculated")
```

```
'Holds the simulation
```

```
Let hyIntegrator.IsRunning = False
```

```
'Set new Hp values for C200
```

```
Let Hp1_SS2.CellValue = Worksheets("Second impeller").[D40].Value
```

```
Let Hp2_SS2.CellValue = Worksheets("Second impeller").[D41].Value
```

```
Let Hp3_SS2.CellValue = Worksheets("Second impeller").[D42].Value
```

```
Let Hp4_SS2.CellValue = Worksheets("Second impeller").[D43].Value
```

```
Let Hp5_SS2.CellValue = Worksheets("Second impeller").[D44].Value
```

```
Let Hp6_SS2.CellValue = Worksheets("Second impeller").[D45].Value
```

```
Let Hp7_SS2.CellValue = Worksheets("Second impeller").[D46].Value
```

```
' Activates ALL the curves of C100
```

```
C20.ActivateCurve = True
```

```
C21.ActivateCurve = True
```

```
C22.ActivateCurve = True
```

```
C23.ActivateCurve = True
```

```
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

```
'Sets the correct value for the Constant C in the anti-surge controller
```

```
'The Excel cell E61 is exported to the cell B24 in the HYSYS SS2
```

```
Let hySS2.Cell(1, 23).CellValue = Worksheets("Second impeller").[E61].Value
```

```
'Activates again the curves to avoid problems
```

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

```
'Activates the simulation
```

```
Let hyIntegrator.IsRunning = True
```

```
b = b + 1
```

```
Else
```

```
'Debug.Print ("The Interpolation Factor of C200 will not change")
```

```
End If
```

```
'Reads the new simulation time value (time_new)
```

```
Set hycr = hyIntegrator.CurrentTime
time_new = hycr.Value
```

```
j = j + 1
```

```
ReDim Preserve deltatime(0 To j) As Long
```

```
'Holds the simulation
```

```
Let hyIntegrator.IsRunning = False
```

```
'Activates ALL the curves of C100 and C200
```

```
C10.ActivateCurve = True
C11.ActivateCurve = True
C12.ActivateCurve = True
C13.ActivateCurve = True
C14.ActivateCurve = True
C15.ActivateCurve = True
Comp1.UsingCurves = True
```

```
C20.ActivateCurve = True
C21.ActivateCurve = True
C22.ActivateCurve = True
C23.ActivateCurve = True
C24.ActivateCurve = True
C25.ActivateCurve = True
Comp2.UsingCurves = True
```

OPTIONAL

```

'Activates the simulation
Let hyIntegrator.IsRunning = True

DoEvents
Loop

'Deactivates the simulation
Let hyIntegrator.IsRunning = False

'Creates a new vector eliminating the first and last value of the deltatime vector
'due to the first value would be equal to the starting simulation time
'and the last would be zero

ReDim deltatimef(0 To j - 2) As Long

For i = 1 To j - 1
deltatimef(i - 1) = deltatime(i)
Next

'Deactivates the simulation
Let hyIntegrator.IsRunning = False

Debug.Print ("The simulation has finished")

maxt = WorksheetFunction.Max(deltatimef)
Debug.Print ("Maximum correction time is ") & maxt & (" seconds")

mint = WorksheetFunction.Min(deltatimef)
Debug.Print ("Minimum correction time is ") & mint & (" seconds")

avet = WorksheetFunction.Average(deltatimef)
Debug.Print ("The average correction time is ") & avet & (" seconds")

Debug.Print ("The Curve C100 was changed ") & a & (" times")
Debug.Print ("The Curve C200 was changed ") & b & (" times")

'Deactivates the simulation
Let hyIntegrator.IsRunning = False

End Sub

```

Appendix E Gas and condensate compositions

Gas composition:

| Component | Mole Fractions |
|------------------|-----------------------|
| Methane | 0.8200 |
| Ethane | 0.0800 |
| Propane | 0.0400 |
| n-Butane | 0.0100 |
| n-Pentane | 0.0050 |
| n-Hexane | 0.0005 |
| H ₂ O | 0.0000 |
| CO ₂ | 0.0300 |
| Nitrogen | 0.0060 |
| i-Butane | 0.0050 |
| i-Pentane | 0.0035 |

Molecular weight (kg/kmol) 20.34

Condensate composition:

| Component | Mole Fractions |
|------------------|-----------------------|
| n-Butane | 0.25 |
| n-Pentane | 0.15 |
| n-Hexane | 0.2 |
| i-Butane | 0.25 |
| i-Pentane | 0.15 |

Molecular weight (kg/kmol) 67.94