LAPPEENRANTA UNIVERSITY OF TECHNOLOGY
LUT School of Energy Systems
Energy Technology

Eetu Rantala

VALIDATION OF COMPUTATIONAL FLUID DYNAMICS SIMULATIONS FOR CENTRIFUGAL COMPRESSOR PERFORMANCE

Examiners: Professor D.Sc. (Tech.) Jari Backman
D.Sc. (Tech.) Mihail Lopatin

ABSTRACT

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Validation of Computational Fluid Dynamics Simulations for Centrifugal Compressor Performance

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91 pages, 44 figures and 5 tables

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Keywords: centrifugal compressor, validation, computational fluid dynamics, simulation lifecycle management

Compressed air production accounts for about 10 % of the overall electricity consumption in the industrial sector in developed countries. Because the largest share of the compressed air lifecycle costs comes from energy cost, large savings can be achieved with improvement of compressor efficiency. Computational fluid dynamics is a powerful tool when investigating and developing the performance of the compressor. To find the credibility of the numerical results provided by CFD, the simulation results have to be compared with experimental test results. The management of the simulation and test data can become a bottleneck in the validation process. Simulation lifecycle management provides tools for making the comparison more seamless.

The objective of the thesis was to find methods for more efficient simulation and experimental data comparison, and to ensure the reliability of the simulations. Compressor performance standards were studied, and feasibility of simulation lifecycle management in simulation validation process was examined. Fundamentals of validation were studied to find the best practices for validation.

Parameters chosen for validation experiment were efficiency and pressure ratio. Due to the measurement problems with the compressor used in the experiment, reliable validation for efficiency was not possible to perform. For pressure ratio comparison, decent correspondence between the test results and simulation results was found, but due to the inadequate uncertainty estimation, results cannot be completely stated as validated. For future work, improvement proposals for more efficient validation were listed.

TIIVISTELMÄ

Lappeenrannan teknillinen yliopisto
LUT School of Energy Systems
Energiatekniikan koulutusohjelma

Eetu Rantala

Laskennallisen virtausdynamiikan simulointien validointi keskipakokompressorin suorituskyvylle

Diplomityö

2018

91 sivua, 44 kuvaa ja 5 taulukkoa

Tarkastajat: Professori TkT Jari Backman
TkT Mihail Lopatin

Hakusanat: keskipakokompressori, validointi, laskennallinen virtausdynamiikka, simuloinnin elinkaaren hallinta

Paineilman tuotannon osuus on noin 10 % kehittyneiden maiden teollisuuden sähkönkulutuksesta. Koska suurin osa paineilman elinkaarikustannuksesta tulee energiakustannuksesta, voidaan saavuttaa suuria säästöjä kompressorin hyötysuhdetta parantamalla. Laskennallinen virtausdynamiikka (CFD) on tehokas työkalu kompressorin suorituskyvyn tutkimisessa ja kehittämisessä. Jotta CFD:n antamia numeerisia tuloksia voidaan pitää uskottavina, täytyy simulointituloksia verrata kokeellisiin testituloksiin. Simulointi- ja testidatan hallinnasta voi tulla pullonkaula validointiprosessissa. Simuloinnin elinkaaren hallinta tarjoaa työkaluja saumattomampaan vertailuun.

Työn tavoitteena oli löytää menetelmiä tehokkaampaan simulointi- ja testidatan vertailuun, ja varmistaa simulointien luotettavuus. Standardeja kompressorin suorituskyvyn mittaamiseen tarkasteltiin, ja simuloinnin elinkaaren hallinnan käyttökelpoisuus simulointien validointiprosessissa todettiin. Validoinnin perusteet tutkittiin parhaiden validointitapojen löytämiseksi.

Validointikokeeseen parametreiksi valittiin hyötysuhde ja painesuhde. Kokeessa käytetyn kompressorin mittausongelmista johtuen, luotettavaa hyötysuhteen validointia ei voitu suorittaa. Painesuhteen vertailussa löydettiin kohtuullinen vastaavuus testitulosten ja simulointitulosten välillä, mutta puutteellisen epävarmuuden arvioinnin takia tuloksia ei voida pitää täysin validoituina. Tulevaisuutta varten listattiin kehitysehdotuksia tehokkaampaan validointiin.

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Vantaa 27.2.2018

Eetu Rantala

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SYMBOLS AND ABBREVIATIONS

Roman alphabet

A	Area	[m^2]
b	Passage depth	[m]
C	Absolute velocity	[m/s]
C_D	Discharge coefficient	[-]
C_{pr}	Static pressure recovery coefficient	[-]
c	Speed of sound	[m/s]
c_p	Specific heat capacity at constant pressure	[J/kgK]
c_v	Specific heat capacity at constant volume	[J/kgK]
h	Specific enthalpy	[J/kg]
i	Incidence angle	[°]
K	Total pressure loss coefficient	[-]
N	Rotational speed	[1/s]
n	Polytropic exponent	[-]
P	Power	[W]
p	Pressure	[Pa, bar]
q_m	Mass flow	[kg/s]
q_v	Volume flow	[m^3/s]
R	Specific gas constant	[J/kgK]
r	Radius	[m]
T	Temperature	[K, °C]
U	Blade speed	[m/s]
u	Uncertainty	
W	Relative velocity	[m/s]
W_x	Total shaft power per unit mass of fluid	[J/kg]

Greek alphabet

α	Flow angle	[°]
β	Relative flow angle	[°]
γ	Specific heat capacity ratio	[-]
δ	Error	
η	Efficiency	[-]
ρ	Density	[kg/m^3]
τ	Torque, relative uncertainty	[Nm, -]
ω	Angular velocity	[rad/s]

Subscripts

0	total state

1	inlet, impeller inlet
2	outlet, discharge, impeller outlet
3	diffuser inlet
4	diffuser outlet
b	blade
c	choke
D	experimental
des	design
f	flow
h	hub
m	meridional component
meas	measured
N	normal
num	numerical
p	polytropic
ref	reference
S	simulation
s	isentropic
t	tip
val	validation
θ	tangential component

Abbreviations

AIAA	American Institute of Aeronautics and Astronautics
ASME	The American Society of Mechanical Engineers
B-DMU	Behavioural-Digital Mock-Up
BOM	Bill of Materials
CAD	Computer Aided Design
CAGI	United States Compressed Air and Gas Institute
CEN	European Committee of Standardization
CENELEC	European Committee for Electrotechnical Standardization
CFD	Computational Fluid Dynamics
DMU	Digital Mock-Up
ETSI	European Telecommunications Standards Institute
FAD	Free Air Delivery
FEM	Finite Element Method
IGV	Inlet Guide Vanes
ISO	Organization for Standardization
PDE	Partial Differential Equations
PLC	Programmable Logic Controller
PLM	Product Lifecycle Management

PNEUROP	European Association of Manufacturers of Compressors, Vacuum Pumps, Pneumatic Tools and Air & Condensate Treatment Equipment
S&A	Simulation and Analysis
SCS	Society of Computer Simulations
SLM	Simulation Lifecycle Management
SME	Small to Medium Enterprises
SRQ	System Response Quantities
V&V	Verification and Validation
VDI	Verein Deutscher Ingenieure (Association of German Engineers)

1 INTRODUCTION

Compressors are a part of our everyday life. They can be found at homes and workplaces, and in any means of transportation we use. Applications where compressors are needed include e.g. refrigeration, engines, chemical processes, gas transmission and manufacturing. Basically, everywhere where moving or compressing gas is required.

In developed countries, compressed air production accounts for about 10 % of the overall electricity consumption in the industrial sector. Adding all the components linked to compressed air (portable tools, air pumps, pneumatic heating, ventilation, air conditioning, personal uses, etc.), the share of electricity consumption is about 20 % of the industrial electricity needs. (Cipollone 2015, 2)

According to BP Energy Outlook (2017) the demand of energy in the industry will keep on growing 1,2 % annually. As shown in Figure 1, the largest share of the compressed air lifecycle costs comes from energy cost. Noticing the share of electricity consumption in the industry, even with small improvements in compressor efficiency, large energy savings and emission reductions can be achieved.

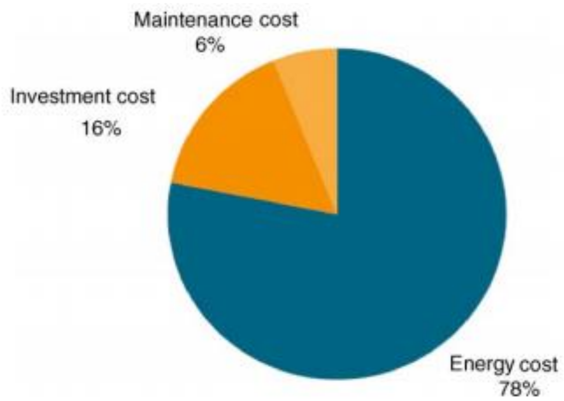

Figure 1. Life cycle costs of compressed air (Saidur et al. 2010, 1137)

Computational methods, and especially computational fluid dynamics (CFD), is a powerful tool when investigating and developing the performance of the compressor. CFD simulation is based on solving the Navier-Stokes equations. CFD enables the calculation of complex flow tasks, and it has become a vital part of the design process in many areas of industry. 3D CFD has become increasingly important in the design and analysis or troubleshooting of compressors. It also reduces the need for physical prototyping and thus saves costs. To take the full advantage of CFD, expertise is required due to the large range of possibilities it provides.

To find the credibility of the numerical results provided by CFD, the simulation results have to be compared with experimental test results. The management of the simulation and test data can become a bottleneck in the validation process. Simulation lifecycle management provides tools for making the comparison more seamless.

1.1 Background

The company this thesis is made for, produces centrifugal compressors. Product portfolio includes several models with different performance parameters. At the moment, the design process works, but improvements have to be considered. Clear simulation workflow is not established. Also, the availability of the simulation results is limited, because of the limited access to the data. Proper validations of CFD simulations has not yet been conducted due to the lack of resources and applicable experimental data. Processing the data is mostly made by hand, which makes it relatively time consuming.

1.2 Research problem and objective

Research problem of this thesis is that how to make sure that the results from CFD simulations are reliable with respect to the experimental results and vice versa. Finding methods and tools for improving the simulation workflow from design to validation is in the key role. Performance aspects of the centrifugal compressor are studied, and standards for the performance measurement are surveyed. The objective is to find methods for more efficient simulation and experimental data comparison, and to ensure the reliability of the simulations.

1.3 Outline of the thesis

Chapter 2 discusses about fundamentals of centrifugal compressors, emphasising on performance aspects. It is also aimed to be training material for current and new employees, and to give new ideas for research and development of the product.

Chapter 3 concentrates on performance standards for centrifugal compressors. The main topics of the standards are summarized. Differences between the standards are also discussed.

Chapter 4 presents simulation lifecycle management. The purpose and role of SLM are defined, and the advantages for enterprises are discussed. Several vendors for SLM tools are presented.

Chapter 5 discusses about the validation in CFD, and why it is important in performing the simulations. Guidelines for validations are given. Fundamentals of estimating validation error and uncertainty are presented.

Chapter 6 is a case study, which is made in order to find the correspondence between experimental and simulation results for efficiency and pressure ratio for the first stage of the compressor. Calculation of experimental uncertainty and validation error is included in the study.

2 CENTRIFUGAL COMPRESSOR

This chapter describes the basic theory of the centrifugal compressor performance. The key factors, characteristics and limitations are discussed. Also, several control systems are described.

Figure 2 presents the categorisation of compressors by operating principle. Centrifugal compressor is classified into dynamic compressor category. It is characterised by continuous flow and the change of the flow direction from axial to radial. Because of the change of the flow direction, it is also known as radial compressor. Another term used is turbo compressor, not to be confused with turbine-compressor combination.

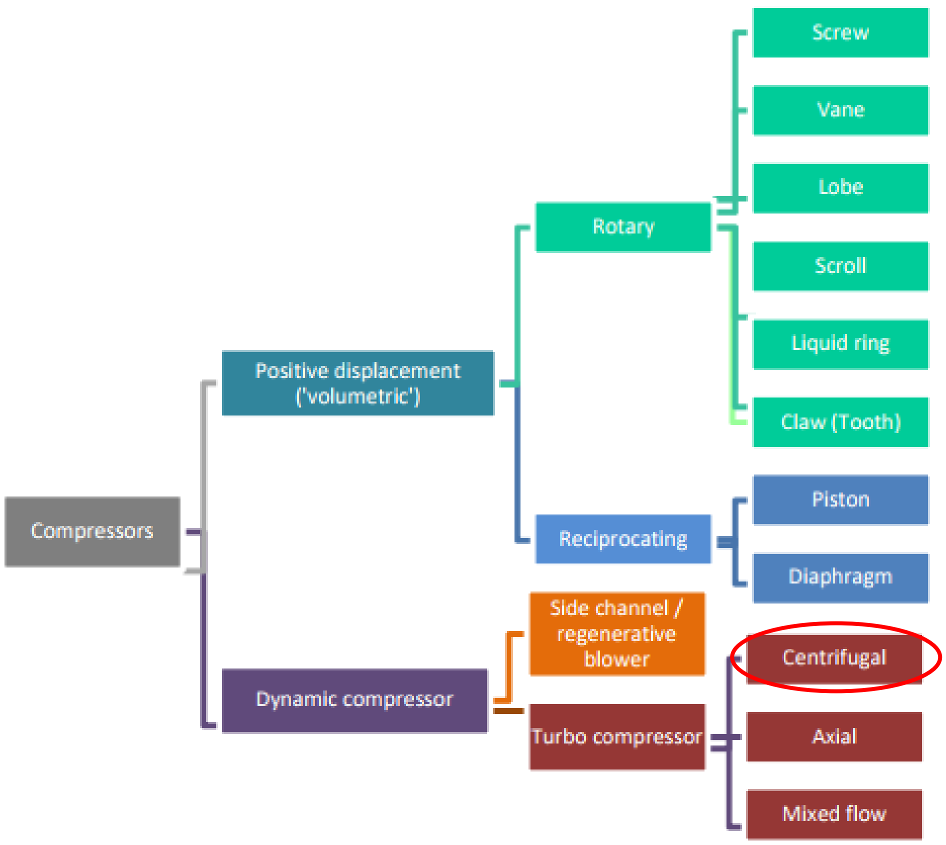

Figure 2. Categorisation of compressors by operating principle (van Elburg & van den Boorn 2017, 28)

2.1 Main components and operating principle

Structure of the centrifugal compressor is fairly simple. It has no reciprocating parts, seals are usually non-contact, and it can be manufactured to run completely oil-free. Compressor can be driven by a variety of drives and forms of energy. A common way is to use electrical motor, direct driven or geared. Main components of the centrifugal compressor are presented in Figure 3. The components are numbered with respect to the direction of the flow.

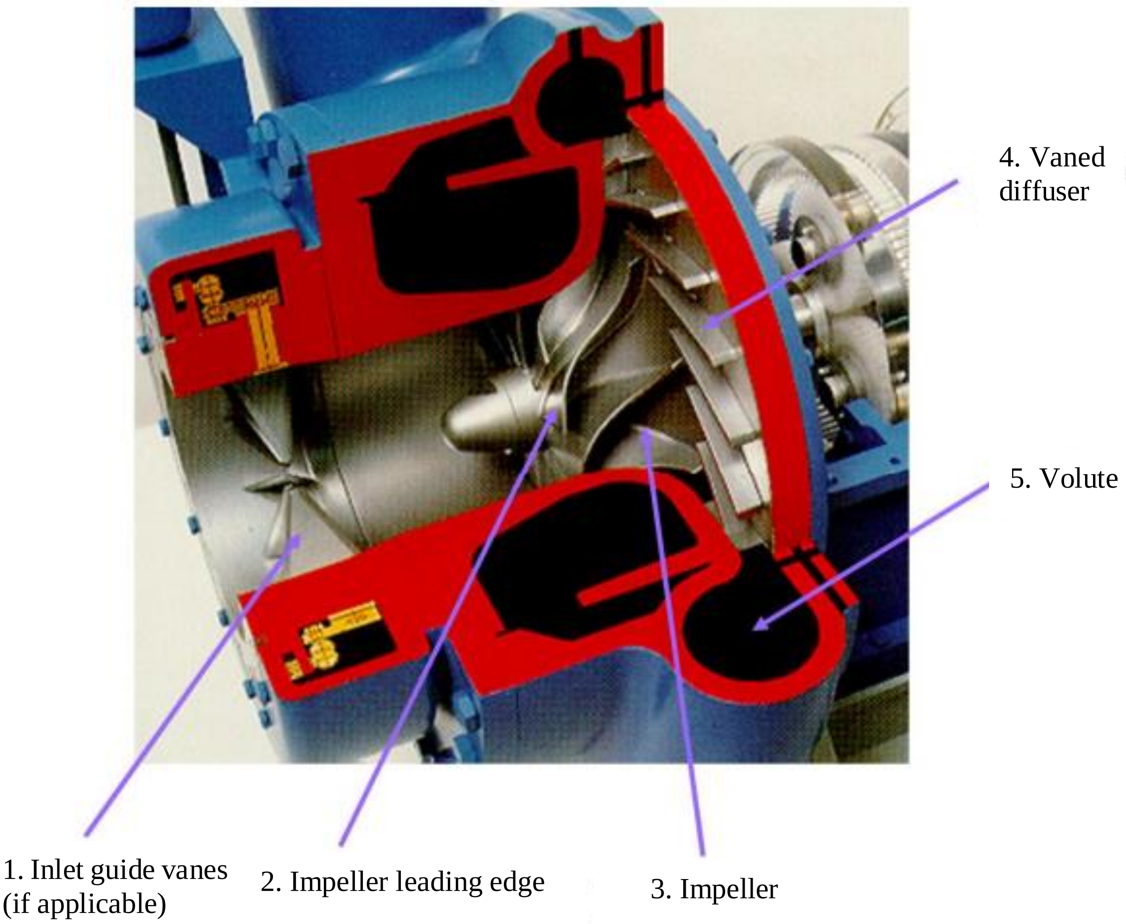

Figure 3. Main components of centrifugal compressor. Adapted from (Larjola et al. 2017, 34)

The generally used method to present a centrifugal compressor is the cross-sectional view. An example of this is illustrated in Figure 4. Black-boxed numbers refer to the state:

0. Inlet duct
1'. Inlet of the inlet guide vanes
1. Impeller inlet
2. Impeller outlet
2'. Diffuser inlet
3. Volute inlet

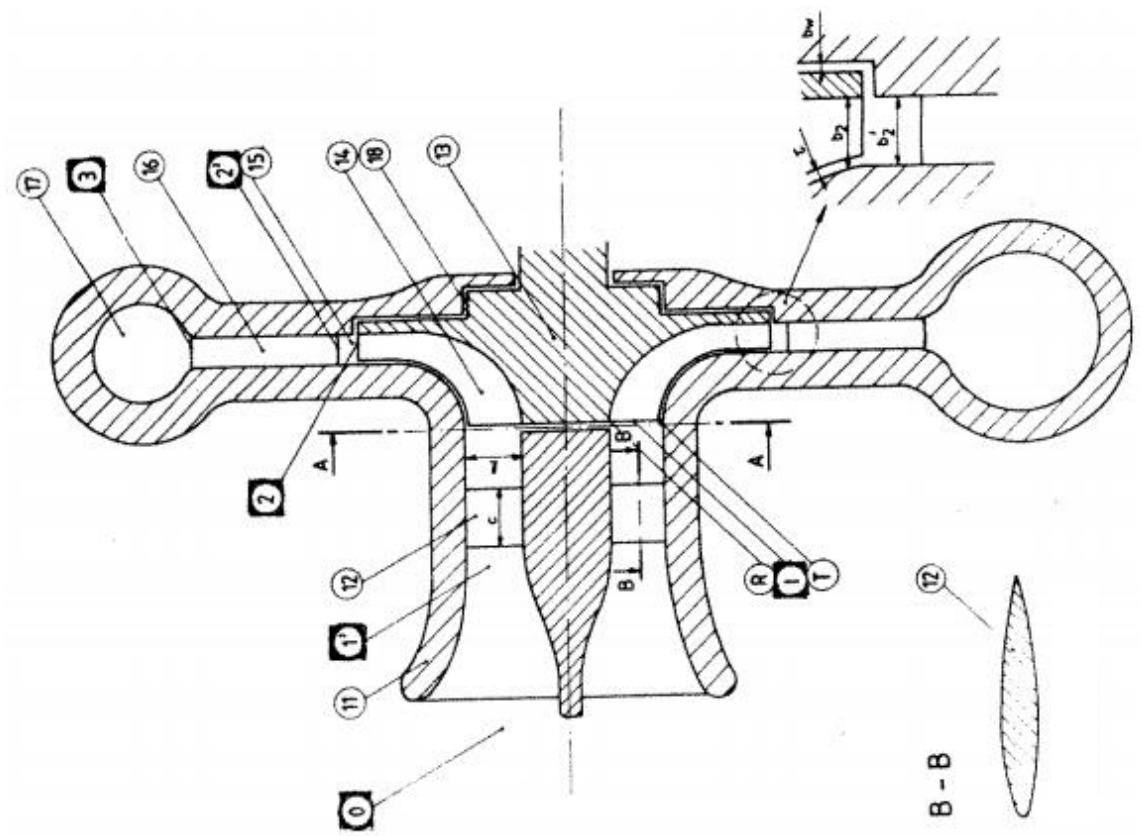

Figure 4. Cross-sectional view of a centrifugal compressor (Larjola 1987, 8)

The gas flows in an axial direction to the impeller through an inlet duct or an inlet bellmouth, unless the pre-rotation is applied. Impeller creates a low pressure region at the inlet face of the impeller, which induces the gas to flow through the inlet duct and enter the impeller. The gas enters the impeller with an incidence angle, and the blades must be bent in a preferred direction to control the relative flow. The gas is drawn through the impeller and the relative flow is decelerated. Due to the deceleration, the pressure is increased, according to Bernoulli’s equation.

Impeller changes the flow direction from axial to radial, and this causes very complex forces, including centrifugal and Coriolis forces, on the flow field. Near the impeller discharge, the flow leaving the impeller is almost according to the blade exit angle. Here the tangential component of absolute velocity is very large and comprised of the strong effect of wheel rotation, usually much larger than the radial or through flow component. In case of a channel diffuser, to control the incidence, the angle of the blades must be appropriately set. The kinetic energy leaving the impeller is usually equal to 30-40 % of the total work input. Therefore, to maximize the static pressure rise, the efficient diffusion of the flow is extremely important. The flow leaving the diffuser is collected in a volute, or taken through a return bend and entered into the subsequent stage. (Japikse & Baines 1994, 1-7)

2.2 Performance

The basis for all turbomachinery applications is Euler turbomachinery equation. It results directly from the energy and momentum equations applied to a blade row. Figure 5 illustrates the rotor, and the symbols used in the equations.

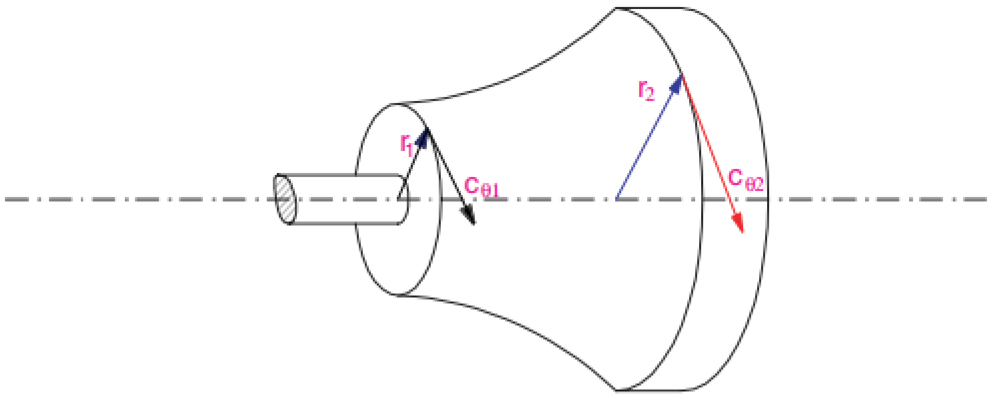

Figure 5. Swirl in a turbomachinery rotor (Hanlon 2001, 3.17)

Considering first Newton's Second Law of Motion applied to a blade row and assuming the mass flow constant, it can be obtained that

$$\tau = q_m(r_2 C_{\theta 2} - r_1 C_{\theta 1}) \tag{1}$$

τ	Torque	[Nm]
q_m	Mass flow	[kg/s]
r	Radius	[m]
C_θ	Tangential component of absolute velocity	[m/s]

In the rotor, the work transferred between the fluid and the shaft per unit mass flow rate is defined as

$$W_x = \frac{\tau\omega}{q_m} = \omega(r_2 C_{\theta 2} - r_1 C_{\theta 1}) \tag{2}$$

W_x	Total shaft power per unit mass of fluid	[J/kg]
ω	Angular velocity	[rad/s]

The first law of thermodynamics states that the work done per unit mass flow is equal to the change in total enthalpy of an adiabatic process. Taking this into account and substituting $\omega r = U$ in the equation (2), between any two points 1 and 2 in a machine it can be stated that

$$\Delta h_{0,1-2} = h_{02} - h_{01} = U_2 C_{\theta 2} - U_1 C_{\theta 1} \tag{3}$$

h	Specific enthalpy	[J/kg]
U	Blade speed	[m/s]

Equation (3) is known as the Euler turbomachinery equation. It applies to compressors, turbines and pumps. The equation can be used with the ideal velocity triangles to establish the ideal enthalpy change, or to the actual velocity triangles to find the actual enthalpy change. (Japikse & Baines 1994, 2-1)

2.2.1 **Impeller velocity triangles**

Impeller velocity triangles are an illustrative method to explain the performance of the compressor. In Figure 6 the velocity triangle in impeller inlet is presented. With velocity triangles, the numbers in subscripts refer to the state of the flow: 1 - impeller inlet, 2 - impeller outlet, 3 - diffuser inlet and 4 - diffuser outlet.

The flow is conveyed to the eye of the impeller with a meridional component of velocity C_{m1}. The compressor wheel moves with a peripheral velocity $U = 2\pi r_1 N$. (Japikse & Baines 1994, 4-1)

According to the fundamental principle of vector addition, the vector triangle is defined as

$$\vec{W} + \vec{U} = \vec{C} \tag{4}$$

W	Relative velocity	[m/s]

If there is no pre-rotation used in the inlet flow ($C_{\theta 1} = 0$), the relative flow angle β_1 is set by the inlet meridional velocity and the local wheel speed. However, in many cases the angle will be greater or less than the blade angle, and hence the blading will be depending on the angle of incidence.

$$i_1 = \beta_{1\mathrm{b}} - \beta_1 \tag{5}$$

i	Incidence angle	[°]
β_{1b}	Inlet blade angle	[°]
β_1	Relative flow angle	[°]

Meridional velocity is defined by continuity in terms of the flow area of the eye of the compressor

$$C_{\mathrm{m}1} = \frac{q_m}{\rho_1 A_{\mathrm{f}1}} = \frac{q_m}{\rho_1 C_D \pi (r_{1\mathrm{t}}^2 - r_{1\mathrm{h}}^2)} \tag{6}$$

C_{m1}	Meridional velocity	[m/s]
ρ	Density of the fluid	[kg/m^3]
A_{f1}	Inlet flow area	[m^2]
C_D	Discharge coefficient	[-]
r_{1t}	Tip radius	[m]
r_{1h}	Hub radius	[m]

Discharge coefficient takes the inlet boundary layer blockage into account. Absolute and relative velocities in the impeller inlet are defined by following equations

$$C_1 = \sqrt{C_{\mathrm{m}1}^2 + C_{\theta 1}^2} \tag{7}$$

$$W_1 = \sqrt{(U_1 - C_{\theta 1})^2 + C_{\mathrm{m}1}^2} \tag{8}$$

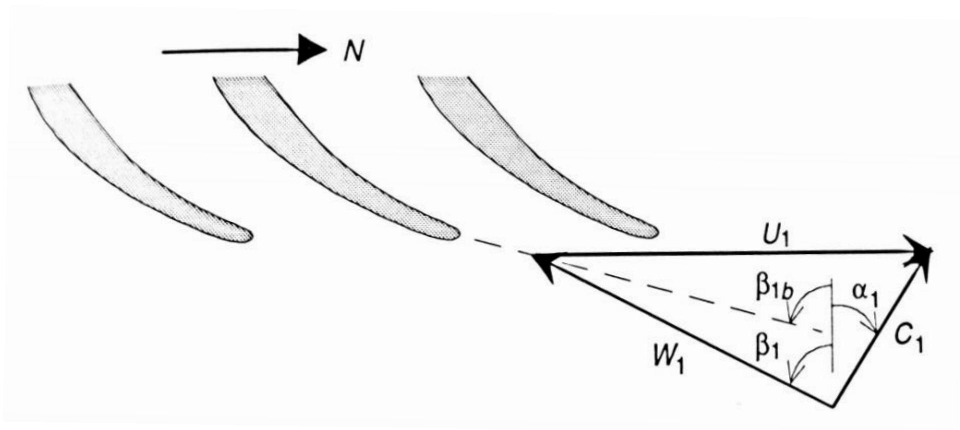

Figure 6. Impeller inlet velocity triangle (Japikse & Baines 1994, 4-3)

Impeller exit triangle is presented in Figure 7. As in impeller inlet, the meridional velocity component C_{m2} is subject to the conservation of mass relationship. Absolute flow angle is a result of the relative flow angle and the wheel speed. Traditionally high-speed impellers were designed with radial blades at exit (β_{2b} = 0°). However, the modern impeller blades are typically backswept by 30-40° at exit. (Japikse & Baines 1994, 4-3)

In reality, the exit flow angle does not exactly follow the blading, but differs by a significant amount. In terms of tangential velocity, this difference is known as slip velocity and it is defined as

$$C_{\mathrm{slip}} = C_{\theta 2\infty} - C_{\theta 2} \tag{9}$$

The term $C_{\theta 2\infty}$ presents the tangential component of absolute velocity which would exist if the flow precisely followed the blades.

The swirl velocity is written by using the velocity triangle relations and the slip velocity as

$$C_{\theta 2} = U_2 + C_{\mathrm{m2}} \tan \beta_{\mathrm{2b}} - C_{\mathrm{slip}} \tag{10}$$

The blade angle in equation (10) is usually negative. The meridional velocity C_{m2} is defined similarly as in the inlet equation (6) except the flow area in this case is defined as

$$A_{\mathrm{f2}} = C_D 2\pi r_2 b_2 \tag{11}$$

b	Passage depth	[m]

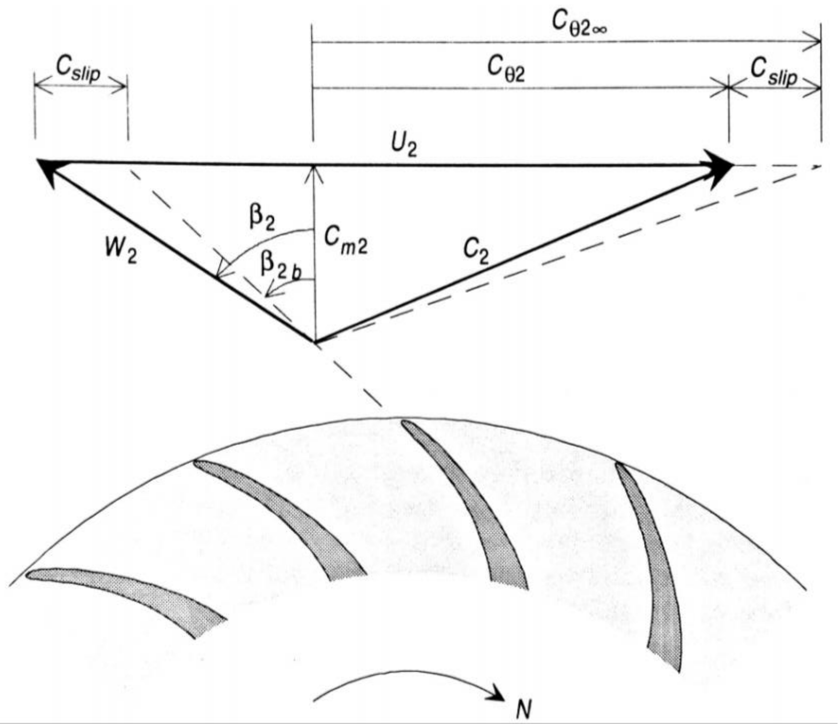

Figure 7. Impeller exit velocity triangle (Japikse & Baines 1994, 4-4)

2.2.2 Diffuser performance

As mentioned in chapter 2.1, effective diffusion is a key factor for achieving a proper pressure rise in the compressor. Performance of the diffuser can be measured with the coefficient of static pressure recovery. The term pressure recovery in diffusers always refers to static pressure, never to total pressure. The coefficient of static pressure recovery for a simple diffuser is defined as

$$C_{pr} = \frac{p_4 - p_2}{p_{02} - p_2} \tag{12}$$

And the total pressure loss coefficient is defined as

$$K = \frac{p_{02} - p_{04}}{p_{02} - p_2} \tag{13}$$

Typically, the values of pressure recovery range from 30-40 % up to peaks of 80-90 % in industrial diffusers. However, to achieve such high values is rare and generally requires highly uniform inlet conditions and some stabilizing benefits as inlet swirl. Usually with high performance diffusers, 60-70 % pressure recovery can be achieved. (Japikse & Baines 1994, 2-26)

Two main diffuser types in centrifugal compressors are vaneless and vaned diffuser. Vaneless diffuser, in practice, is a space between the impeller tip and the beginning of a channel or cascade diffuser, or a space which runs from the impeller discharge to a volute inlet. They are commonly used in automotive turbochargers and in process and refrigeration compressors.

In Figure 8 the velocity components in the vaneless diffuser are presented. For isentropic flow, the velocity components are defined with following equations

$$r_3 C_{\theta 3} = \text{constant} = K' \tag{14}$$

$$C_{\mathrm{m}3} = \frac{q_m}{\rho_3 A_{\mathrm{f}}} = \text{constant} = K'' \tag{15}$$

$$\tan \alpha_3 = \frac{C_{\theta 3}}{C_{\mathrm{m}3}} = \frac{K'}{r_3} \frac{2\pi r_3 b_3 C_D \rho_3}{K''} = K \rho_3 b_3 \tag{16}$$

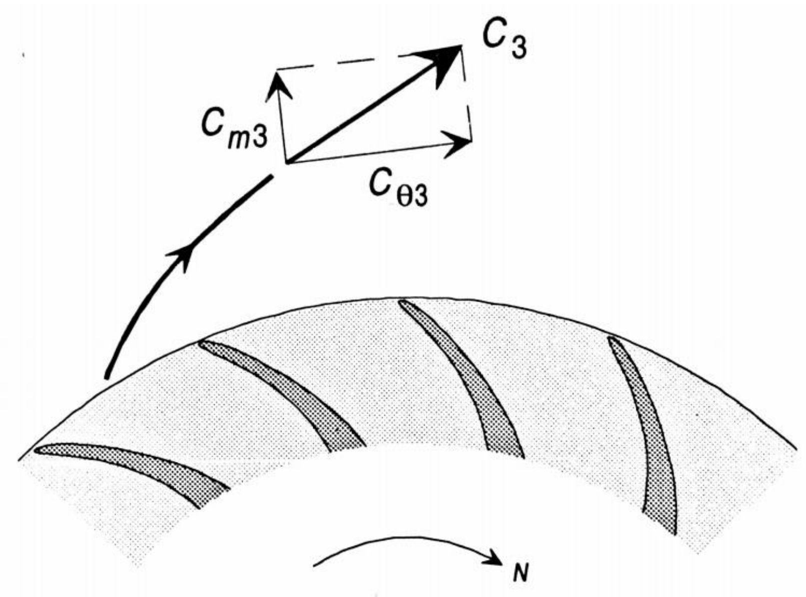

Figure 8. Vaneless diffuser flow (Japikse & Baines 1994, 4-17)

The equation (14) relates the angular momentum in the vaneless diffuser to the impeller exit angular momentum according to the conservation law. In reality, about 5-15 % of this momentum is lost as the flow proceeds through the vaneless diffuser. When the density variation is known, the meridional component of velocity can be calculated from the conservation of mass by equation (15). The flow angle (equation (16)) is then defined by the velocities. It is clearly shown that the flow angle depends on the change in density and passage width. The flow angle control with passage width change can be done by employing the diffuser “pinch” or reduced depth, which is illustrated in Figure 9. (Japikse & Baines 1994, 4-17)

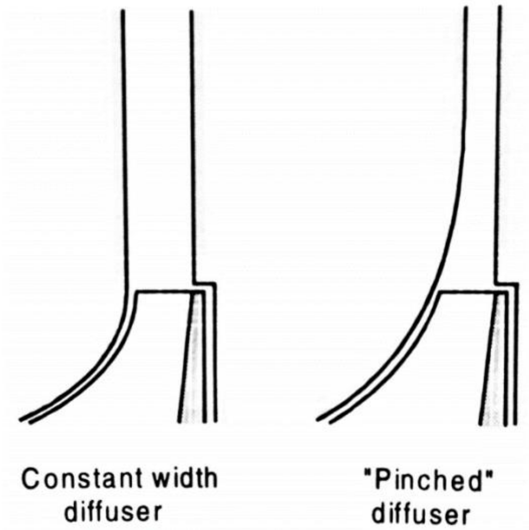

Figure 9. Diffuser “pinch” (Japikse & Baines 1994, 4-17)

Vaned diffuser differs from vaneless that the flow leaving the impeller is considerably directed with guide vanes to open the stream tubes and thus achieving better diffusion. An example of a vaned diffuser is presented in Figure 10. Vaned diffusers usually have a smaller stable operation range compared to vaneless diffusers, due to the possibility of stalling at vanes, especially at moderate incidence levels. The two main types of vaned diffusers are the airfoil and the wedge or channel type. (Japikse & Baines 1994, 4-19)

Figure 10. Vaned diffuser (Jaatinen et al. 2011, 99)

The channel diffuser forces the oncoming flow from an essentially angular motion to a basically linear motion along the path of the diffuser. The location close to the point of minimum geometric area, called throat, is vulnerable to choking. The airfoil diffuser performance differs from a channel or vaneless diffuser, but has some of the principles of each. The basic flow similar to the essentially log spiral flow of the vaneless diffuser, tends to remain the same, but it is forced to depart from it by the lift force created by the airfoil. Thus, the flow expands more rapidly than in the case of the vaneless diffuser. Airfoil diffuser may or may not have an aerodynamic throat. Generally, the highest recovery is achieved with channel diffuser, then with the airfoil and last with the vaneless diffuser. The stable operating range is reverse of this, vaneless giving the largest range and channel smallest. (Japikse & Baines 1994, 4-20)

2.2.3 **Performance map**

The performance characteristic of the centrifugal compressor is visually presented with the performance map. Figure 11 shows an example of a typical centrifugal compressor map. Inlet volume flow is plotted along the x-axis and discharge pressure or pressure ratio along the y-axis. The group of red lines represent different compressor rotational speeds and they define the relationship of pressure ratio and volume flow. Surge limit on the left side of the map defines the lowest volume flow on each rotational speed, before the compressor goes to the damaging surge condition. Stonewall or choke limit on the right side of the map shows the maximum flow on each rotational speed. The slope of the curve depends on the number of stages. Increasing the number of stages, the slope becomes steeper. Also, the configuration of the impeller has an effect on the slope (Hanlon 2001, 4.21). Yellow elliptical curves represent efficiency and because of their shape, they are also known as efficiency contours. The best efficiency is in the centre of the contours. At the design point the rotational speed is 100 % of the nominal rotational speed, and there should be optimal efficiency and a decent safe margin from surge and choke lines.

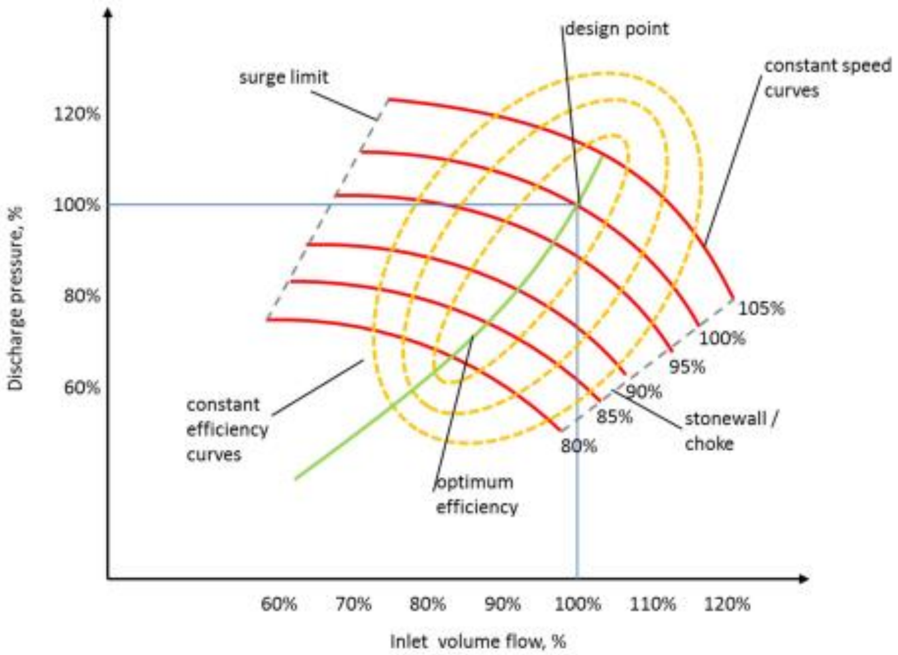

Figure 11. Typical centrifugal compressor map (van Elburg & van den Boorn 2017, 156)

Main internal loss-generating mechanisms of a centrifugal compressor are friction and shock. The effect of these losses on the performance curve shown in Figure 12. The ideal pressure vs. flow curve would be a straight line with pressure being inversely proportional to flow rate. However, due the increasing internal losses, the slope of the actual performance curve gets steeper when approaching to the choke limit.

In addition to the internal losses, there are external losses in the compressor stage that must be taken into account when modelling the system. The most common external losses are friction in bearings and seals, and windage caused by the leakage of the gas from the annulus onto the faces of rotor and impeller disks. (Japikse & Baines 1994, 2-21)

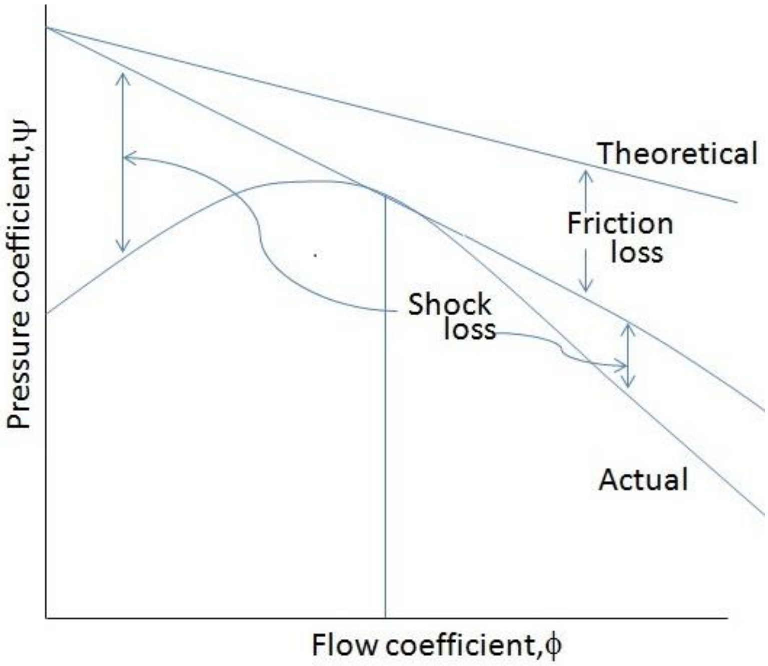

Figure 12. Comparison of theoretical and actual performance curve

2.3 Efficiency

Application of the term efficiency is very wide in turbomachinery. For all kinds of compressors, efficiency is simply defined as

$$\eta = \frac{\text{work into ideal compressor}}{\text{work into actual compressor}} \tag{17}$$

In this definition, the pressure rise is equal in both ideal and actual compressor. Evaluating efficiency is not straightforward as there are several different ways of presenting compressor efficiency and they reveal different information. In the thermodynamic sense, the ideal compressor is reversible. Depending on the duty it may be isothermal, when the temperature is constant, or adiabatic, when there is no heat flow to the gas. Adiabatic ideal assumption can be used when the gas is used directly for propulsion or to be heated or burned, for example. The adiabatic application is more common of these two. In an ideal adiabatic and reversible compressor, the entropy flowing through the compressor is constant. Such compressors are referred isentropic. (Cumpsty 1989, 34)

2.3.1 Isentropic efficiency

Isentropic efficiency is defined as

$$\eta_{\mathrm{s}} = \frac{P_{\mathrm{s}}}{P_{\mathrm{actual}}} \tag{18}$$

P_{s}	Isentropic power	[W]
P_{actual}	Actual power	[W]

The subscript “s” denotes that entropy remains constant in the compression. The requirement for adiabatic compression is that the work input is equal to the rise in total enthalpy. By applying this to the equation (18) the efficiency can be written

$$\eta_s = \frac{h_{02s} - h_{01}}{h_{02} - h_{01}} \tag{19}$$

$h_{02\mathrm{s}}$	Isentropic total enthalpy	[J/kg]

Equation (19) is valid without restriction to the nature of the gas. However, a great simplification can be made if the gas is assumed to be ideal gas. In this case $h = c_p T$ where the specific heat capacity is either constant or a function of temperature alone. Usually for the gases at low pressures in relation to their critical pressure, the ideal gas assumption is good but in many engineering applications of compressors there are exceptions from this simplification. Nevertheless, most air compressor operate at acceptable conditions for ideal gas assumption and compressibility factor can be neglected. (Cumpsty 1989, 34)

For an ideal gas, the isentropic temperature ratio in terms of pressure ratio is defined as

$$\frac{T_{02\mathrm{s}}}{T_{01}} = \left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} \tag{20}$$

$T_{02\mathrm{s}}$	Isentropic total outlet temperature	[K]
T_{01}	Total inlet temperature	[K]
p_{02}	Total discharge pressure	[Pa]
p_{01}	Total inlet pressure	[Pa]
γ	Specific heat capacity ratio	[-]

By using equation (20) the simple expression for isentropic efficiency can be written

$$\eta_s = \frac{\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1} \tag{21}$$

Isentropic efficiency is usually expressed between the total states, as total-to-total efficiency. Another isentropic efficiency form is total-to-static efficiency, where the discharge pressure used in the equation is in static state. (Japikse & Baines 1994, 2-17)

With moderate pressure ratios, specific heat capacity ratio can be assumed near constant. If the humidity of gas and effect of the temperature change is not neglected, it is useful to substitute the exponent as

$$\frac{\gamma - 1}{\gamma} = \frac{\frac{c_p}{c_v} - 1}{\frac{c_p}{c_v}} = \frac{c_p - c_v}{c_p} = \frac{R}{c_p} \tag{22}$$

c_p	Specific heat capacity at constant pressure	[J/kgK]
c_v	Specific heat capacity at constant volume	[J/kgK]
R	Specific gas constant	[J/kgK]

The isentropic efficiency assumes that heat does not exchange to the environment. In reality, the gas is usually cooled down to certain temperatures. The heat can be recovered, but it does not influence in the isentropic efficiency itself. Multistage compression with intercooling, and oil-injection affect the efficiency of the compression process. Isentropic efficiency is usually 70-85 % for most compressor designs, depending on the type, application range, size and task. (van Elburg & van den Boorn 2014, 54)

ISO 1217 annex H provides a simplified method for isentropic efficiency calculation. Isentropic efficiency is defined as a ratio of isentropic power and electric input power of package. Isentropic power is therefore defined as

$$P_s = q_{v1} p_1 \frac{\gamma}{\gamma - 1}\left(\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{23}$$

q_v	Volume flow	[m^3/s]

In Figure 13, different compression processes in *T*, *s*-diagram are presented. In the diagram, the final temperature of each compression type can be seen. The constant pressure lines have a slope proportional to the temperature. As temperature and entropy increases, the pressure lines diverge from each other. This means that the minimum temperature rise to produce a given pressure rise, increases either initial temperature or entropy.

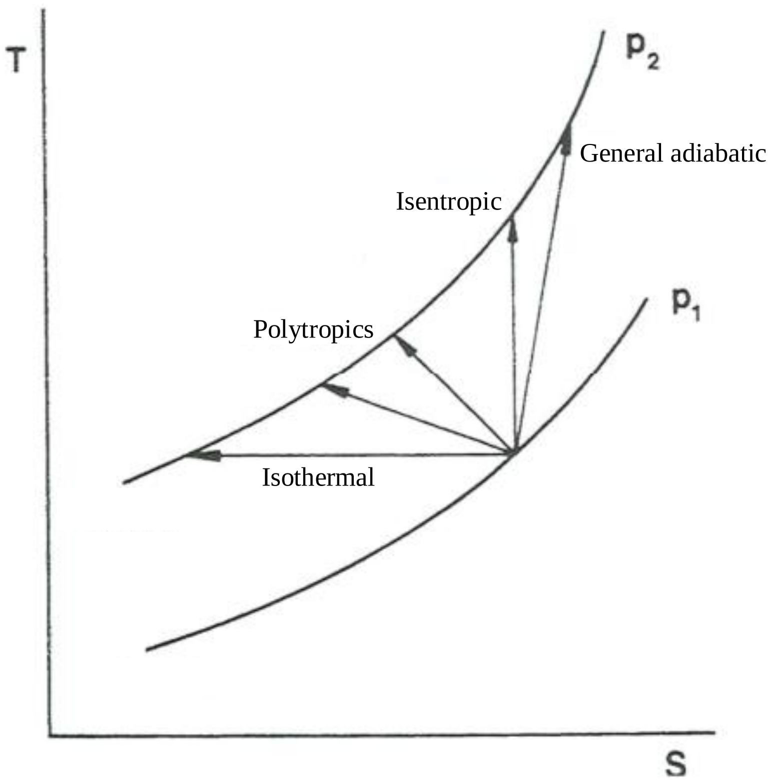

Figure 13. Compression processes in *T*,*s*-diagram. Adapted from (Larjola et al. 2017, 11)

For a multistage compressor, this means that the power requirement is greater for the last stages to produce the same pressure rise as the first stages because the temperature is higher. Also, the losses in early stages increase the power requirement in the later stages. Therefore, as the overall pressure ratio is increased, the isentropic efficiency of aerodynamically identical compressors gets lower.

2.3.2 Polytropic efficiency

Possible confusion caused by pressure ratio dependence can be avoided by using polytropic efficiency. The polytropic efficiency neglects the effect of higher pressure ratio so that the aerodynamically identical compressors with different pressure ratios have the same polytropic efficiency although the isentropic efficiency is different. (Cumpsty 1989, 38)

For an adiabatic and reversible process, the enthalpy rise is defined as

$$\mathrm{d}h_\mathrm{s} = \frac{\mathrm{d}p}{\rho} \tag{24}$$

In a real compression process the enthalpy increases more than $\mathrm{d}h_\mathrm{s}$ and it can be written

$$\mathrm{d}h = \frac{1}{\eta_\mathrm{p}}\mathrm{d}h_\mathrm{s} = \frac{1}{\eta_\mathrm{p}}\frac{\mathrm{d}p}{\rho} \tag{25}$$

If η_p is assumed constant over a finite change in pressure, for a perfect gas can be shown that

$$\frac{T_{02}}{T_{01}} = \left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\eta_\mathrm{p}\gamma}} \tag{26}$$

By using equation (26) the polytropic efficiency can be defined as

$$\eta_\mathrm{p} = \frac{\gamma-1}{\gamma}\cdot\frac{\ln\left(\frac{p_{02}}{p_{01}}\right)}{\ln\left(\frac{T_{02}}{T_{01}}\right)} \tag{27}$$

In a real multistage compressor, the polytropic efficiency for one stage is equal to the polytropic efficiency of the whole compressor. Correspondence between isentropic and polytropic efficiencies can be calculated by substituting the temperature ratio for a given polytropic efficiency to the isentropic efficiency equation (21)

$$\eta_s = \frac{\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\eta_\mathrm{p}\gamma}} - 1} \tag{28}$$

Relation between isentropic and polytropic efficiencies is presented in Figure 14. Increasing of the pressure ratio and decreasing of the efficiency increases the difference between the polytropic and isentropic efficiencies. The efficiencies are equal when $\pi = 1$ or $\eta_\mathrm{p} = 1$. (Cumpsty 1989, 38-39)

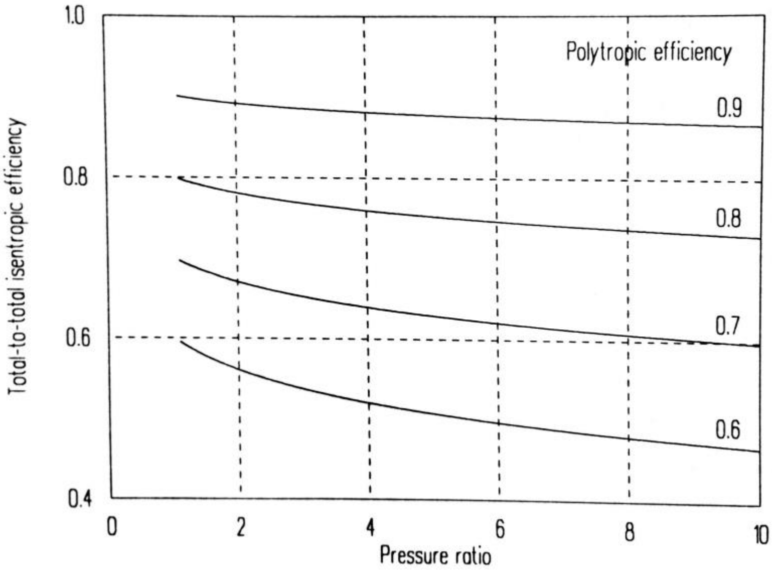

Figure 14. Relation between total-to-total isentropic and polytropic efficiencies (Japikse & Baines 1994, 2-20)

2.3.3 Isothermal efficiency

In an isothermal change of state, the temperature of the gas is assumed to be constant, which means that all the heat produced in the compression process is removed to ambient constantly. It can be treated as a special case of polytropic compression. In the temperature ratio equation (20), it means that $\gamma = 1$.

Compression in real compressors cannot get even close to the isothermal compression. Compressors with oil or liquid cooling are closest to approximate this process. Isothermal efficiency is defined as a ratio of isothermal power and actual power input

$$\eta_{\text{isothermal}} = \frac{P_{\text{isothermal}}}{P_{\text{actual}}} = \frac{q_m R T_1 \ln\left(\frac{p_2}{p_1}\right)}{P_{\text{actual}}} \tag{29}$$

Van Elburg & van den Boorn (2014) listed the reasons why isentropic efficiency is considered more suitable than isothermal efficiency:

- Simplified comparison between different operating pressures/pressure ratios
- Isentropic efficiency is widely accepted in other (i.e. non-industrial air) technical fields like energy technology

- Isentropic efficiency is less sensitive regarding deviation of measurement conditions (operating point) and gas properties when comparing to specific power requirement
- Compressors without internal or interstage cooling are physically not able to compress isothermally
- Due to additional losses, even compressors with internal cooling are not able to reach even isentropic compression

2.4 Intercooling

Almost in all the multistage compressors, the heat is removed from the gas between the stages with intercooling. Intercooling results the compression to more closely approximate the isothermal compression. Therefore, the efficiency is increased and the power requirement is decreased. The cooling is also required to maintain the material temperatures below the limitations. In Figure 15 is illustrated the effect of intercooling on compression process. It is shown that the enthalpy rise of two-stage compression is lower compared to the single-stage compression with same discharge pressure, and thus power is saved. The amount of stages has to be optimised because of the increased costs and pressure losses with additions of stages.

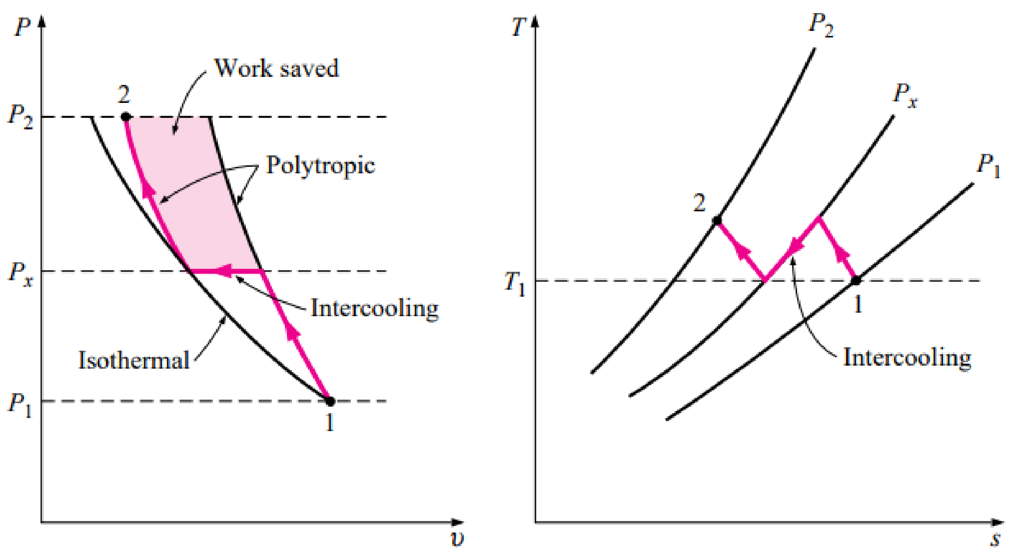

Figure 15. Multistage compression and intercooling in *p*,*V*- and *T*,*s*-diagrams (Cengel et al. 2016, 310)

Figure 16 presents the compression process in three stages with intercooling. The gas first enters the compressor through the inlet pipe. After compression the gas enters the first intercooler (3), where heat is removed. Cooled gas enters the second stage and after that, heat is removed again in the second intercooler (5). Finally, the gas is compressed to the desired discharge pressure in the third stage, from where it exits to the aftercooler.

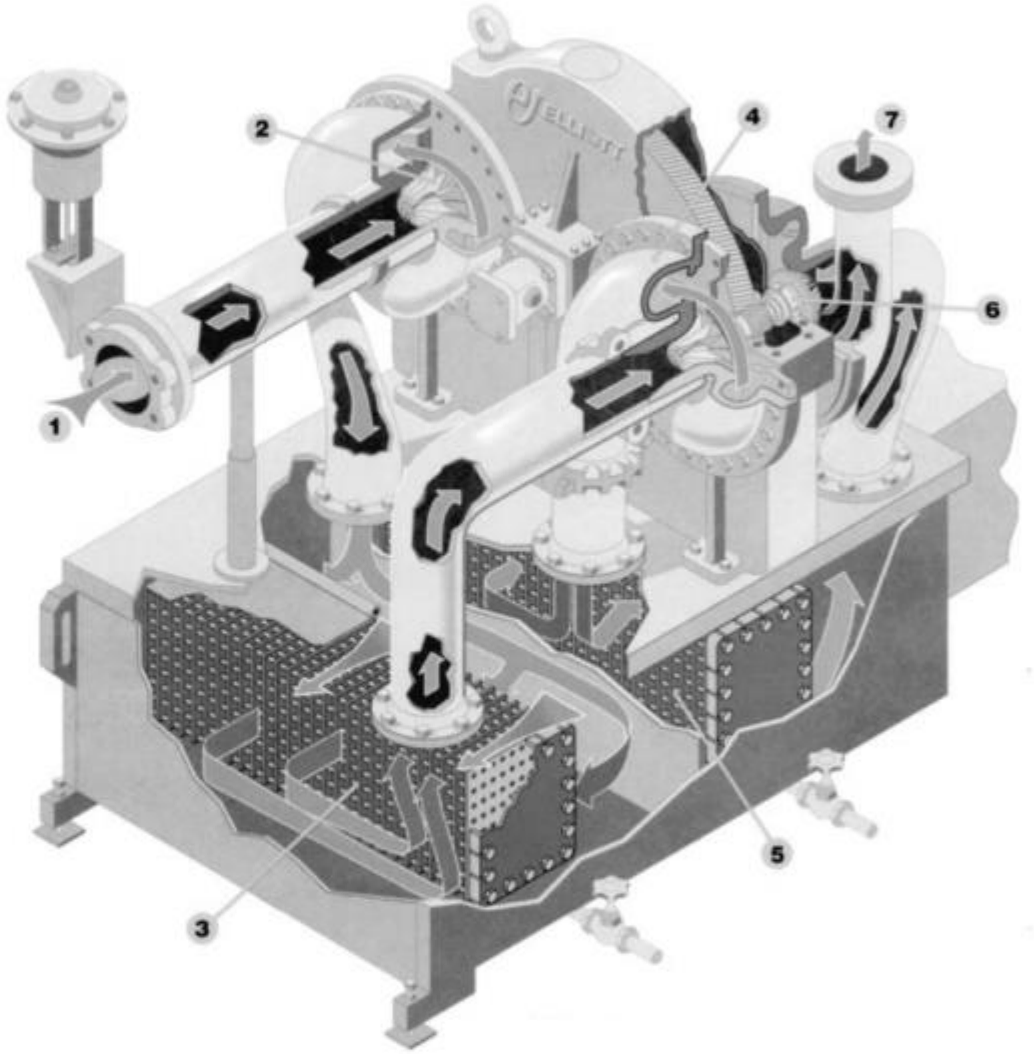

Figure 16. Intercooling in multistage centrifugal compressor (Hanlon 2001, 4.4)

The intercooler does not necessarily have to be a traditional water-air heat exchanger. Kang (1986) examined the intercooling by spraying water to the air. Cooling water being in direct contact with air is more effective for heat transfer because of the increased heat transfer area and decreased heat resistance. With this method a proper water separation after the intercooler has to be taken into consideration.

2.5 Influence of the inlet conditions

Centrifugal compressor is a mass flow device, so the inlet conditions influence to the performance of the compressor. Power requirement for a rated volume flow at rated pressure is determined by the mass of the air. Consequently, the performance of a single compressor may vary significantly depending on the geographical location or ambient temperature. The environmental parameters influencing the performance are inlet temperature, inlet pressure, relative humidity and cooling water temperature. The total influence of any of these conditions depends on the actual performance curve and aerodynamic characteristics of the compressor.

Influence of the inlet temperature on pressure, mass flow and power, is presented in Figure 17. Increasing the inlet temperature reduces the density of air. This reduces the mass flow and power requirement of the compressor. (van Elburg & van den Boorn 2017, 154) (CAGI 2015a, 1)

The change in gas density has the effect on the available turndown of the compressor. This refers to the flow range where efficient regulation by throttling or inlet guide vanes regulation is possible. With lower inlet temperature, the higher turndown range is available. (CAGI 2015a, 1)

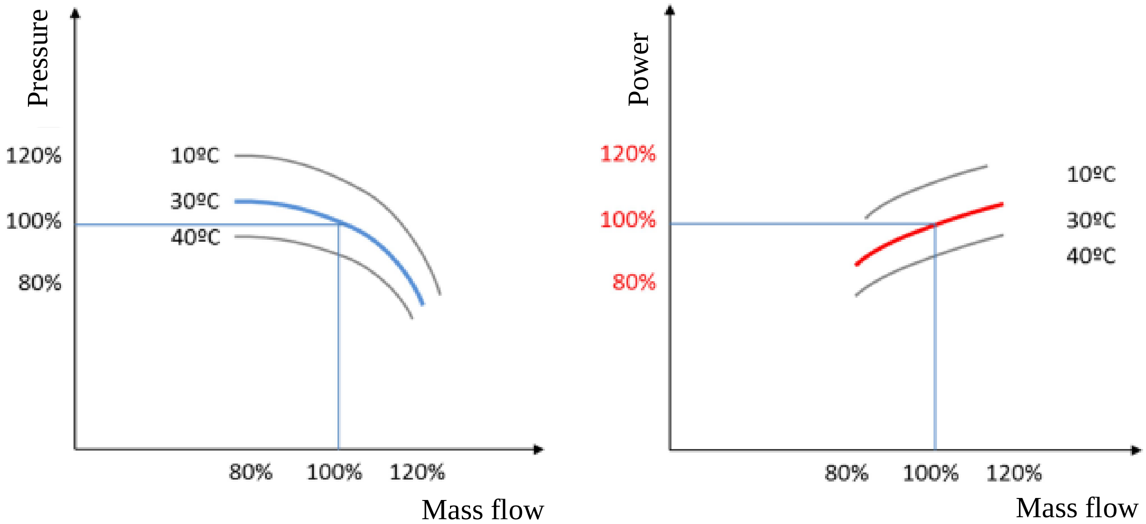

Figure 17. Effect of the inlet temperature on pressure, mass flow and power. Adapted from (van Elburg & van den Boorn 2017, 155)

In Figure 18 can be shown that the lower inlet pressure results in decreased mass flow and power requirement due to the reduction in density of air. Reason for decreased inlet pressure can be fouled or poorly sized inlet filters, or change of ambient pressure. Also, the available turndown is smaller with lower inlet pressure. (CAGI 2015a, 3)

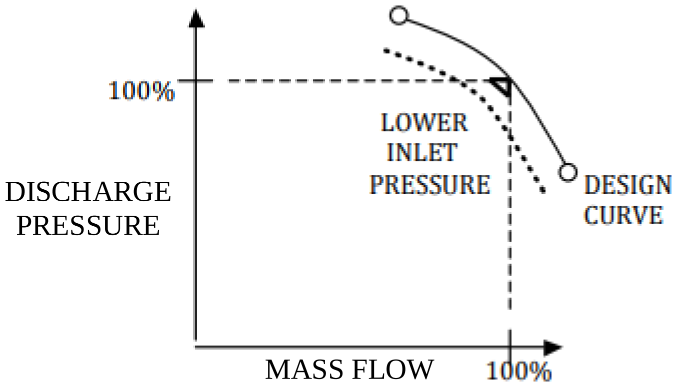

Figure 18. Effect of the inlet pressure on pressure and mass flow. Adapted from (CAGI 2015a, 4)

Increased relative humidity reduces mass flow and power requirement. Humid air is resulted of adding water vapor to the air. It reduces the density of the air due to the molar mass of water being less than the molar mass of air. Influence of the relative humidity on pressure and mass flow is presented in Figure 19. (CAGI 2015a, 4)

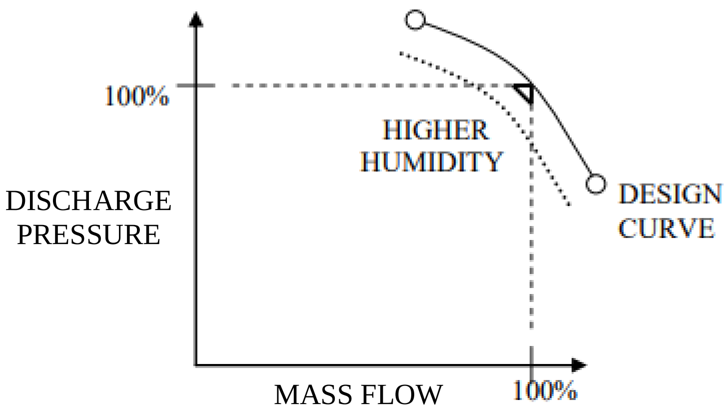

Figure 19. Effect of the relative humidity on pressure and mass flow. Adapted from (CAGI 2015a, 5)

In a multistage compressor, the cooling water temperature affects to the inlet temperature of the second, third and subsequent compressor stages where the intercooling applies. As shown in Figure 20, colder cooling water increases mass flow and power requirement, and warmer water opposite. (CAGI 2015a, 5)

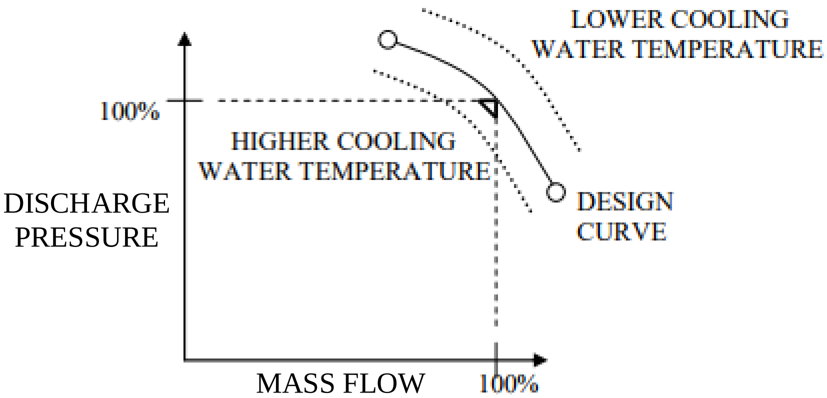

Figure 20. Effect of the cooling water temperature on pressure and mass flow. Adapted from (CAGI 2015a, 6)

Compilation of the influences of each inlet parameter on compressor performance is presented in Table 1. Arrow up represents increasing of the parameter and arrow down decreasing.

Table 1. Influence of inlet parameters on compressor performance

Parameter change		Mass flow	Discharge pressure	Power requirement
Inlet temperature	↑	↓	↓	↓
Inlet pressure	↓	↓	↓	↓
Relative humidity	↑	↓	↓	↓
Cooling water temperature	↓	↑	↑	↑

For the performance of the compressor to be comparable in varying conditions, the measured values are converted to reference values. If the inlet conditions of the compressor map are known, the performance of the compressor in other conditions can be compared to the original compressor map. Reynolds number effect neglected, mass flow, rotational speed and power conversion to the reference conditions can be executed with equations (30), (31) and (32), respectively. (Jaatinen et al. 2011, 101)

$$q_{m,\mathrm{ref}} = q_m \frac{p_{01,\mathrm{ref}}}{p_{01}} \sqrt{\frac{T_{01}}{T_{01,\mathrm{ref}}} \frac{R}{R_{\mathrm{ref}}}} \tag{30}$$

$$N_{\mathrm{ref}} = N \sqrt{\frac{T_{01,\mathrm{ref}}}{T_{01}} \frac{R_{\mathrm{ref}}}{R}} \tag{31}$$

N	Rotational speed	[1/s]

$$P_{\mathrm{ref}} = P\frac{q_{m,\mathrm{ref}}}{q_m}\frac{T_{01}}{T_{01,\mathrm{ref}}}\frac{R}{R_{\mathrm{ref}}} \tag{32}$$

Free air delivery (FAD) is a term used in many instances to state the performance. It is defined as "delivered flow converted back to the inlet thermodynamic condition”. FAD can be referred to standard condition or suction condition, but usually to inlet pressure 1 $\mathrm{bar_a}$ and inlet temperature 20°C. (van Elburg & van den Boorn 2017, 32) depending on the standard used

Ignoring the humidity, FAD is defined as

$$q_{v,\mathrm{FAD}} = q_{v,\mathrm{N}}\frac{T_{\mathrm{FAD}}}{T_{\mathrm{N}}}\frac{p_{\mathrm{N}}}{p_{\mathrm{FAD}}} \tag{33}$$

$q_{\mathrm{v,N}}$	Normal volume rate of flow	[m^3/s]
T_{FAD}	Standard inlet temperature	[K]
T_{N}	Normal reference temperature (0ºC = 273,15 K)	[K]
p_{FAD}	Standard inlet pressure	[Pa, bar]
p_{N}	Normal reference pressure (1,013 $\mathrm{bar_a}$)	[Pa, bar]

2.6 Limitations

Even though the centrifugal compressor has a large operating range, it has its limitations. As presented in Figure 11, surge and choke lines set the limits for the compressor performance. These are discussed in this chapter. Moreover, increasing the rotational speed, at some point the compressor faces the maximum rotational speed where the stresses and vibrations cross the allowable limits, and damage the machine.

2.6.1 Surge

Surge refers to the state of instability which occurs at low flow rate values. It involves the whole compressed air system, not only the compressor itself. The flow separates from the blades due to the inversion of the direction of velocity in the boundary layer in proximity to a solid wall. Separation is linked to the presence of adverse pressure gradients in respect to the main direction of motion. Flow separation in the compressor blade can be compared to the condition in an airplane wing, where the angle of attack exceeds the limiting value and the flow separates from the wing causing the loss of lift.

Stall describes the condition where the stage pressure ratio does not vary in a stable manner with the flow rate due to the low flow rate. In a stage, it is a result of the separation phenomena in one or more of its components. Stall is visually presented in Figure 21. (Hanlon 2001, 3.27)

Figure 21. Stall cell in a centrifugal impeller passage (Yoon et al. 2013, 7)

During the surge, intense and rapid flow and pressure fluctuations take place throughout the system. The oscillating flow is accompanied by strong noises, depending on the geometry and nature of the installation. The noise can vary from low-frequency booming sound to squeal. Backflow causes the same gas to back up and recompress until the next backflow. This causes the temperature at the inlet to rise relatively fast. Each pass through the compressor adds additional heat of compression. Even though the thrust bearing takes most the injurious effects of the action, continuous surge can severely damage the machines involved. (Brown 2005, 221)

For low pressure compressors, the surge usually starts in the diffuser and for higher pressure compressors, the starting point usually moves into the impeller. An experienced listener can identify the beginning of the surge. Experimental tests are run to measure the pressure pulsation at low flow rates and thus identify the values of the stable operation. (Brown 2005, 221) (Hanlon 2001, 3.27)

Surge can be avoided by either increasing speed, decreasing discharge pressure and/or increasing the flow by application of anti-surge valves. Hanlon (2001, 3.82) describes the simple method for surge protection with the application of by-pass valve. Necessary information for anti-surge control can be collected with two differential pressure transducers measuring pressure drops in the compressor. Thus, the anti-surge set-point for the pressure loss coefficient can be determined. By-pass valve regulates the system from not going below the established set-point.

Yoon et al. (2013) examined that it is possible to control the surge with active magnetic bearings. Controlling the impeller tip clearance with axial actuation by active magnetic bearings has the effect on the characteristic curve, and thus the location of the surge point is changed. At 16290 rpm the stable flow range of the controlled system was extended by 21,3 % with no loss in the maximum compressor efficiency.

2.6.2 **Choke**

Choke line sets the limit for the maximum flow rate. Chocking occurs when the increased flow rate in coincidence with a port, reaches the sonic speed in the impeller or vaned diffuser. With low rotational speed, choking usually takes place in the throat of the diffuser and with high rotational speed in the throat of the impeller leading edge (Larjola 1987, 48). Increasing the flow rate through the compressor after choking is not possible. As seen in the compressor map, there is a sudden drop in the discharge pressure, which leads to the reduced mass flow and input power. Continued choking can cause a surge-like operation with damaging vibrations. In some engineering purposes, choking is also known as “stonewall”.

Choking mainly depends on the geometry and operating conditions of the compressor, but also the thermodynamic properties of the fluid have an effect on chocking. Especially with fluids of high molecular weight, choking particularly limits the performance range. However, compressors rarely operate in conditions close to the choke limit. Vibration monitors attached to the system can recognize choking and shut down the compressor. (Hanlon 2001, 3.27) (van Elburg & van den Boorn 2017, 32)

For the impeller, the mass flow rate with choking is defined as

$$q_{m,\mathrm{c}} = A_1 \rho_{01} c_{01} \left(\frac{2}{\gamma+1}\right)^{\frac{1}{n-1}+\frac{1}{2}} \left(1 + \frac{\gamma-1}{2}\frac{U_1^2}{c_{01}^2}\right)^{\frac{1}{n-1}+\frac{1}{2}} \tag{34}$$

A_1	Through-flow area	[m^2]
c_{01}	Speed of sound in the inlet of impeller	[m/s]
n	Polytropic exponent	[-]

The most critical location for choking is the inducer inlet, because the further the flow enters, the lower is the velocity, and the higher is the enthalpy. (Dick 2015, 528)

2.7 Control methods

The flow rate and pressure ratio can be controlled with several different methods. These methods can also be combined. Methods to control the compressor are throttling inlet or discharge, regulating the rotational speed, changing the pre-rotation of the inlet flow with inlet guide vanes (IGV) or adjusting the angle of the diffuser vanes.

According to McMillan (1983) the conventional control methods listed from the most energy efficient to the most energy consuming are

1. Speed control
2. Inlet guide vanes
3. Inlet throttling
4. Discharge throttling

When the same control methods are listed by the control range, the order changes a little. From the largest range towards small flow rate to the smallest range towards small flow rate, the list is arranged

1. Inlet guide vanes
2. Speed control
3. Inlet throttling
4. Discharge throttling

The energy efficiency of speed control is a result from maintaining the operating point at a high efficiency range in a relatively large flow range. IGV is both economical and efficient method, and it is almost without exception used when the speed regulation is not possible.

2.7.1 Speed control

Speed control is the most efficient method to maximize the performance flexibility of the compressor. The effect of the speed control can be seen in the compressor map Figure 11. The pressure-flow curve shape changes with speed due to higher losses at higher speed.

The minimum volume flow rate is typically 60-70 % of the maximum volume flow rate for higher pressure applications or 20-50 % for low pressure applications, depending on discharge pressure. The application of variable speed drives is more expensive and less efficient when driven at a single speed only compared to fixed speed designs. (van Elburg & van den Boorn 2017, 157)

2.7.2 **Throttle valve**

Capacity of the compressor can be regulated with throttle valve at the inlet or outlet. Surge line sets the minimum flow limit which can be achieved with throttling the inlet. Throttle valves require about 8-9 % more power near full closure, reducing efficiency by the same amount, compared to variable inlet guide vanes (van Elburg & van den Boorn 2017, 157). Throttling is a simple but not an efficient method to control the flow, and it should be only used for fine adjustment purposes. (Larjola et al. 2017, 42)

2.7.3 **Variable inlet guide vanes**

Variable inlet guide vanes are an efficient flow control method for a centrifugal compressor. IGV produces pre-rotation to the gas flowing into the compressor reducing the axial velocity component of the absolute velocity. The axial component controls the capacity to the impeller. Pre-rotation changes the incidence angle of the air approaching the inducer section of the impeller and due to this pre-rotation parallel to impeller rotational direction, less power in compression is required. In other words, the tangential component of absolute velocity $C_{\theta 1}$ is introduced in Euler turbomachinery equation (3), which changes the work input in the compressor.

Controlling the flow with IGV is most effective on single-stage compressors, but they can also be used in multistage compressors in front of the first impeller. For complex compressor arrangements, this method is not practical. The guide vanes are located immediately in front of the impeller, directly in the flow path. The shanks of the vanes are connected to an external linkage through the inlet housing. The linkage is connected to a power operator which controls the position of the vanes. Figure 22 presents the installation of the inlet guide vanes. In the right-hand side of the figure, the vanes are fully closed. (Brown 2005, 258)

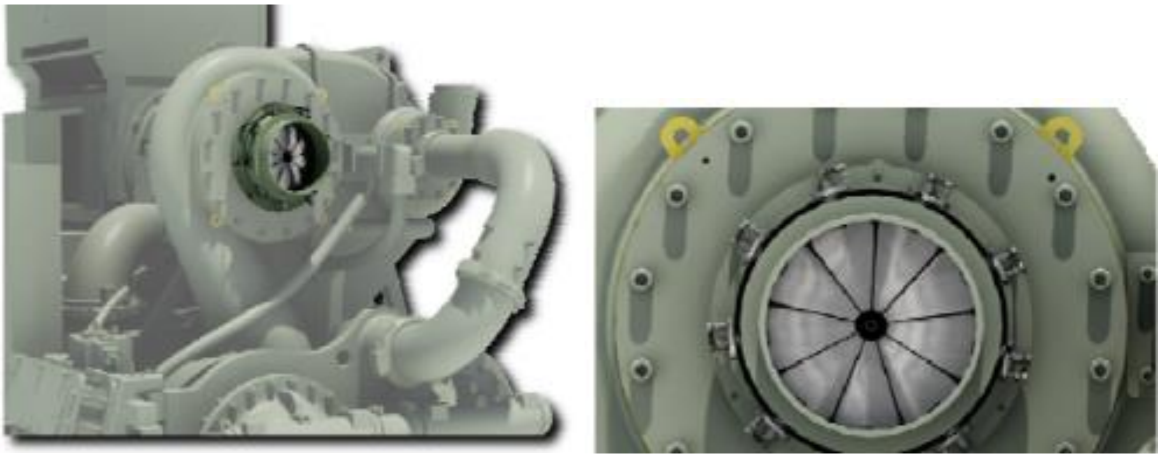

Figure 22. Inlet guide vanes in the inlet of a compressor (CAGI 2015, 1)

As presented in Figure 23, with IGV the volume flow can be decreased to 50-70 % of the design volume flow. By turning the vanes in the opposite direction, it is also possible to increase the capacity and pressure to a certain degree, but it may impair performance.

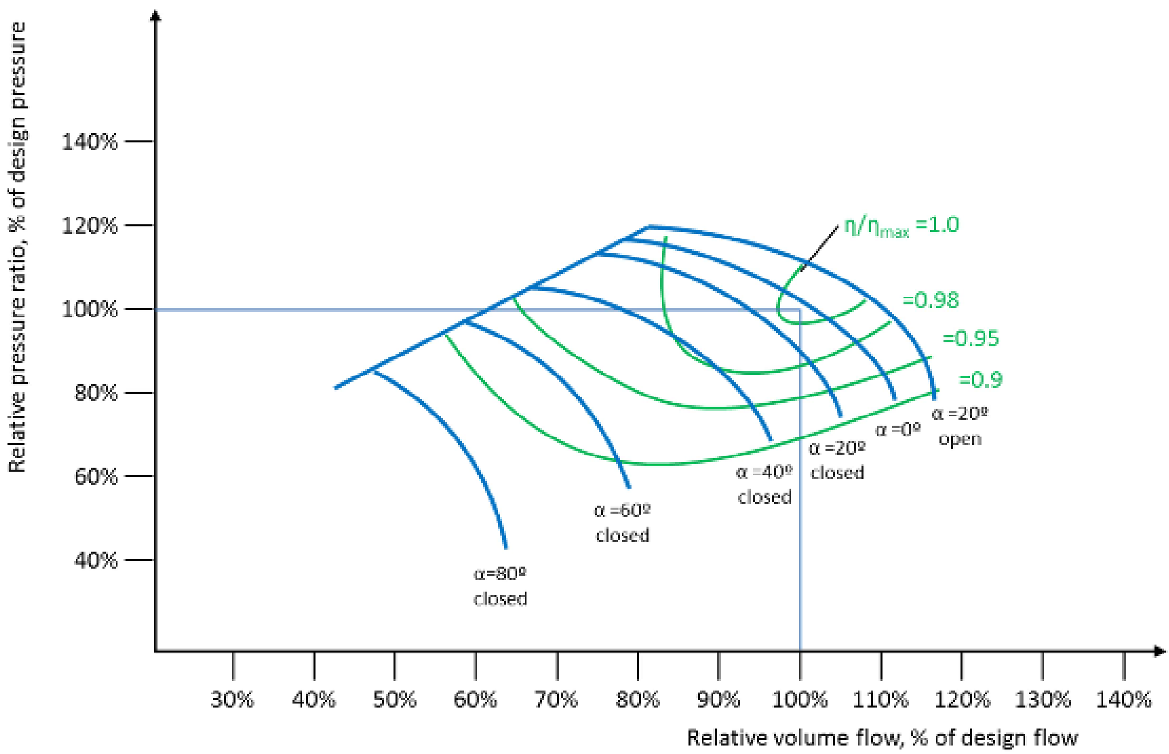

Figure 23. Inlet guide vanes control (van Elburg & van den Boorn 2017, 158)

2.7.4 Variable diffuser guide vanes

The flow in the diffuser can be controlled with variable diffuser guide vanes. By changing the angle of the diffuser guide vanes, the volume flow can be decreased down to 30 % of the design volume flow with pressure remaining constant. According to Smith et al. (1987) by a suitable adjustment of the diffuser, in certain conditions the compressor can operate in a stable operating point, even though the impeller lead edge had already stalled. Due to the complexity and increased cost, the usage of the variable diffuser guide vanes is mainly limited to single-stage compressors. Variable diffuser guide vanes do not have a significant influence on impeller performance, so to achieve wider operating range, use of other technologies (e.g. IGV) combined with variable diffuser guide vanes, is needed (Jiao et al. 2009, 1069). The influence of the diffuser guide vanes angle to compressor operating range is presented in Figure 24. (Atlas Copco 2015, 56)

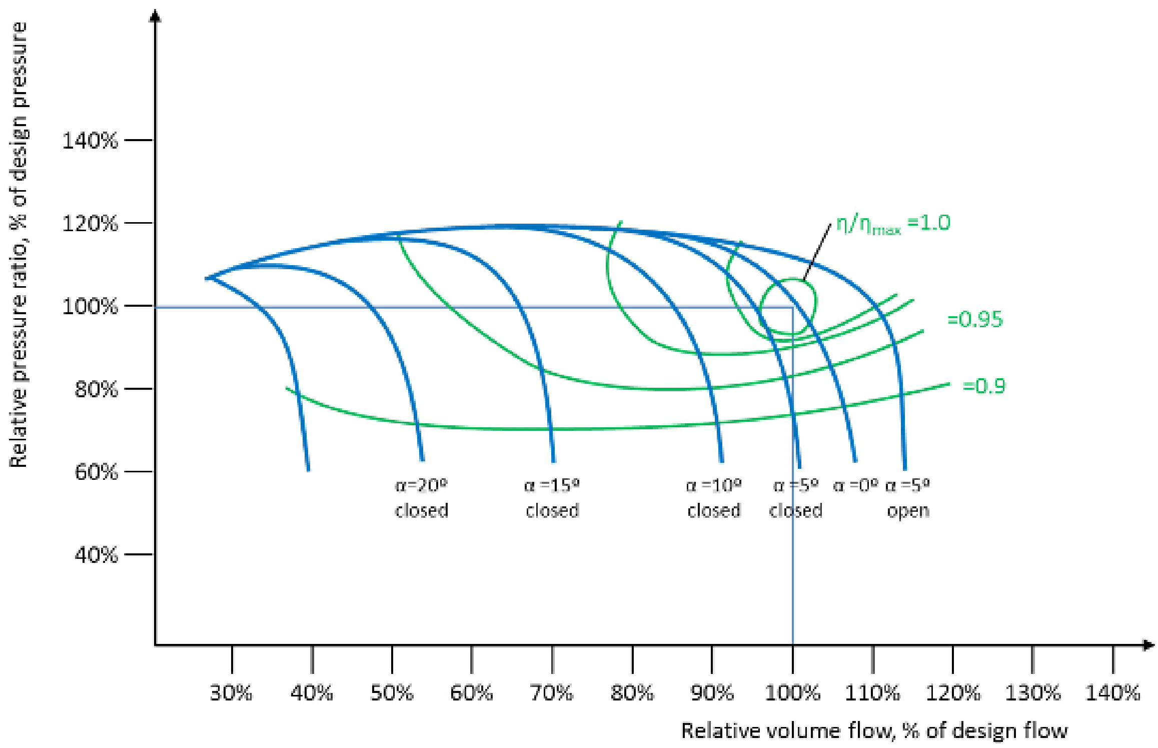

Figure 24. Variable diffuser guide vanes control (van Elburg & van den Boorn 2017, 159)

3 COMPRESSOR PERFORMANCE STANDARDS

There are different regulations that apply in the compressed air sector. Some of the regulations are requirements defined by legislation, but they also could be optional regulations or recommendations. The standards are either national or international. Regulations in standards can sometimes become binding through legislation, or when quoted in a commercial agreement. These binding regulations usually regard safety for people and property. Optional standards are made to give recommendations in several areas of work, for example quality, measuring, manufacturing etc.

International standardization benefits manufacturers, intermediate parties and final customers. It makes the comparability of performance statements on equal terms possible as well as increases the interchangeability of products and systems between different parties. The performance statements may contain operational, environmental and safety topics. (Atlas Copco 2015, 135)

Tests may be performed in the manufacturer’s shop or in the field. Standards help the customer to understand the review and acceptance of the test plan for a particular machine. Detailed review is mandatory, because of the options available under the standards, exceptions taken by manufacturers and the complexity of properly simulating design point operating conditions. (Van Laningham 1981, 169)

Standards establish a basis for testing, but the methods and procedures of the test must be agreed upon by the manufacturer and the user. Rarely the tests are run in strict conformance to the code and some deviation from the standard is acceptable. Reasons for these shortcuts are for example to expedite the test, hold down the costs or to adjust the installation or facilities. The user must understand the requirements of the standard and where these deviations appear. (Matthews 1981, 165)

The preparation and maintaining of the standards is usually made by standardization organizations. These organizations are either on national, supranational (European) or international levels but equally focusing on the specific industrial sector. International Organization for Standardization (ISO) is an independent, non-governmental international organization with a membership of 162 national standards bodies. For a draft to become published as an International Standard, at least 75 % of the member bodies must give an approval for the document. The standards produced by ISO can be converted into national standards. CEN (European Committee of Standardization) standards are made for use by 30 national members and in case of harmonized standards the conversion into national standards may be mandatory.

Standards may also be produced by trade associations. In the compressed air industry, there are associations such as PNEUROP (European Association of Manufacturers of Compressors, Vacuum Pumps, Pneumatic Tools and Air & Condensate Treatment Equipment) and its counterpart CAGI (United States Compressed Air and Gas Institute). These additional documents are produced while awaiting an international standard to be published. (Atlas Copco 2015, 135)

This chapter describes the two most important performance test standards: ISO 5389 for turbocompressors and ISO 1217 for displacement compressors. Also, BL 300 performance test standard for low pressure air compressor packages is presented. It was introduced by CAGI to a need for acceptance tests for both positive displacement and centrifugal low pressure technologies.

3.1 ISO 5389

ISO 5389 “Turbocompressors - Performance test code” is an international standard, which applies to all types of turbocompressors. The standard defines turbocompressors as machines with continuous flow in inlet, compression and discharge. The gas is moved and compressed in impellers and decelerated in fixed vaned or vaneless stators with increase in pressure. The standard excludes fans, high-vacuum pumps and jet-type compressors with moving drive components. ISO 5389 is based on American ASME PTC 10 “Performance Test Code on Compressors and Exhausters” and German VDI 2045-1 and VDI 2045-2.

The standard intends to give provisions for preparation, procedure, evaluation and assessment of performance tests for turbocompressors. Performance test code is the base for an acceptance test. The order conditions and guarantees are specified in the contract and the purpose of the acceptance tests is to fulfil these agreed values. (ISO 5389, 1)

3.1.1 **Guarantees**

To guarantee the properties and characteristics of the compressor by the acceptance test, there must be a contractual agreement between the customer and the manufacturer. These properties are verified by means of the values measured in the acceptance test. The test values are then converted to the guarantee conditions. (ISO 5389, 6)

In the contract, there are defined conditions for the guarantee. These preconditions can include following conditions (ISO 5389, 7):

- Inlet pressure and inlet temperature
- For intercooled compressor, recooling temperatures and pressure drops between the relevant compressor sections
- Physical properties of the gas or vapour and its composition
- Coolant, its mass flow and inlet temperature
- Operating conditions of the driving machine
- Inlet and outlet state referred to the inlet and outlet flow area of the compressor
- Speed

Under the preconditions, the following values can be guaranteed (ISO 5389, 7-8):

- Actual inlet volume flow
- Discharge pressure
- Power for inlet volume flows, for example the electrical power of the drive motor
- Efficiency related to a suitable reference process
- Power of auxiliary machinery
- Operating range limits

Also, some additional guarantees can be specified if they are of significance for operation. These can include for example part-load efficiencies, temperature of the compressed gas or cooling efficiencies. The measured and converted test results are compared to values guaranteed with the allowance limits of measuring uncertainties. In case of series production, each individual compressor doesn’t have to go through the acceptance test process. A few randomly selected compressors from the series successfully tested shall be deemed to suffice. (ISO 5389, 8)

3.1.2 **Measurements**

ISO 5389 recommends using the measuring methods and measuring instruments inclusive of the rules if applicable. Also upon agreement, other measuring methods regarding testing and fitting may be used. The measuring points and equipment for measurements shall be incorporated into the compressor during design and its installation into the subsequent system. These measuring points include pressure, temperature, flow, power and speed. Especially shall be taken into account at all points of the flow measurement that the adequate lengths of straight pipe are available and suitable flanged joints for installation of the orifices and nozzles as specified in ISO 5167-1 “Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full”. Guarantees should be referred to the measuring points. (ISO 5389, 8)

For acceptance tests the following measuring instruments shall be used:

a) Measuring instruments calibrated by comparison with measuring instruments as specified in c)
b) Measuring instruments for which a calibration or test certificate issued by an accredited authority is available
c) Other tried and proven measuring instruments of known accuracy, the use of which has been agreed between the parties to the contract

The check for measuring instruments shall be done before installation and/or before and after the test for condition and dimensional accuracy. The results of the check shall be stored. When using measuring instruments with transducers of any type and digital evaluation is possible, the calibration shall be done and a record kept of calibration. The measuring systems shall be possible to check by suitable means. The same procedure applies also to the use of data acquisition systems and electronic data processing. (ISO 5389, 9)

3.1.3 **ISO 18740 - Simplified acceptance test**

In July 2016, ISO 18740 “Turbocompressors - Performance test code - Simplified acceptance test”, was published. Currently it complements the primary standard ISO 5389 for standard packages, but will ultimately become an annex of ISO 5389 until enough experience has been gained from its use in practical conditions. The simplified test code applies to any fixed speed, liquid cooled, packaged centrifugal air compressor driven by an electric motor. The compressors inside the scope are designed to operate with atmospheric air from their surroundings and the performance data usually relates to a normal ambient air inlet pressure. At a later date, it’s planned that variable speed types will be included in the standard.

For electrically driven packaged air compressors of standard types sold against performance data published in the manufacturer’s sales documentation the acceptance tests are defined and described. Based on specified test conditions, the performance statement can be given when the key measured variables are maintained within the test limitations. (ISO 18740, 1)

3.2 ISO 1217

ISO 1217 “Displacement compressors - Acceptance tests” is an international standard which defines methods for acceptance tests regarding volume rate of flow and power requirements of displacement compressors. The standard defines the operating and testing conditions for a full performance test. Normative annex E concentrates on any electrically driven compressor with variable speed drive (e. g. variable frequency drive, direct current drive and switched reluctance) which contains a displacement compressor of any type driven by an electric motor. In 2016, annex H describing the calculation of isentropic efficiency was published. It is available as an additional document ISO 1217:2009/Amd.1:2016 “Calculation of isentropic efficiency and relationship with specific energy"

The standard provides instructions for a full performance test which includes the measurement of volume flow rate and power requirement. Instructions are also given for the correction of measured values to specified conditions and means of comparing the corrected values with the guarantee conditions. The methods for determining the value of the tolerances for measurement of flow, power and specific power is specified. (ISO 1217, 1)

The comparison with the guarantee or specified performance shall include:

- Comparison of the corrected power consumption (specific power consumption, fuel consumption or efficiency) with the guaranteed power consumption
- Comparison of the corrected volume flow rate with the guaranteed volume flow rate at the specified pressure rise or pressure ratio

When presenting the comparison, there should be a conclusion included whether or not the performance of the compressor meets the specification. Several factors shall be taken into account in making the comparison, for example uncertainty of measurement, errors due to the properties of the gas used or errors due to the inaccuracy in correction of the test results. (ISO 1217, 21)

3.3 BL 300

BL 300-2016 "Performance Test Code for Electric Driven Low Pressure Air Compressor Packages" was put together by CAGI and the PNEUROP PN2 Low Pressure Working Group. A new standard was needed for the comparison of different low pressure air compressor package technologies. The existing standards for positive displacement compressors and dynamic compressors, ISO 1217 and ISO 5389, respectively, are not providing clear and concise methods for comparing different technologies. The purpose of BL 300 is to provide simplified wire-to-air performance methods for measuring true package performance of low pressure air compressors. Wire-to-air is a term to describe the total energy needed to produce the required flow and pressure for any particular application (Balberg 2013). The comparison is important when verifying the performance of a compressor package measured at any facility with varying inlet conditions which usually are different from guarantee conditions. (BL 300, 1)

Low pressure is defined with the following limits in the standard (BL 300, 4):

- 0,5 bar $\leq p_1 \leq$ 1,1 bar
- 0,1 bar $\leq p_2 - p_1 \leq$ 2,5 bar
- $1,1 \leq p_2/p_1 \leq 3,5$

Because of the limitations of the method, for example no cooling during compression, extrapolation to higher pressures (5-7 bar_a) is not sought in the standard (van Elburg & van den Boorn 2017, 45).

3.3.1 Guarantees

Similar preconditions as in the above-mentioned conventional performance test standards shall be agreed. Because the compressor is tested as a whole package, also preconditions for the ancillary machines can be specified. At least the following preconditions shall be specified for testing to be possible (BL 300, 4):

- Air inlet pressure
- Air inlet temperature
- Air inlet humidity
- Coolant inlet temperature
- Coolant flow
- Supply voltage
- Supply frequency
- Electromagnetic emissions
- Noise level outside the package (e.g. by law)

Within the defined preconditions, the following values are to be guaranteed (BL 300, 11):

- Inlet volume flow rate
- The discharge pressure at the outlet of the package.
- The total Specific Energy of the package for the delivered flow at the guaranteed discharge pressure

3.4 Comparison of the performance test standards

The full base test standards ISO 5389 and ISO 1217 address many different types of compressor product variations. It turns them lengthy and complex to follow precisely. The annexes focusing on specific types of compressors describe simplified and rigorous methods, yet providing extremely accurate results. ISO 1217 and BL 300 and their annexes provide true wire-to-air methods to calculate the specific power from power inputs. ISO 18740 now demonstrates this kind of a method for constant speed compressors. (CAGI 2012)

ISO 5389 does not include wire-to-air methods. The standard does not cover losses across the compressor package or other powers than the shaft power. It has three different classes of conversion of the test results. The class chosen for converting the results depends on the tolerance of the ratio of volume flow ratios (ISO 5389, 28). The standard allows for many deviations and the instructions for choosing the processes to measure flow is unclear. Also, the conversion from test conditions to site conditions is not clear and subject to interpretation. (Balberg 2013)

3.4.1 Standard inlet conditions

Ideally the conditions of the performance test should be identical to the end application. In reality, this is not always possible. Therefore, the results obtained from the real inlet conditions tests are converted to be equivalent to standard inlet conditions or agreed conditions. Also, for the test results to be comparable, they have to be linked to standard inlet conditions. The standard inlet conditions for each standard is presented in Table 2. ISO 18740 differs from other standards by providing two options for inlet conditions to choose.

Table 2. Standard inlet conditions

	ISO 1217	BL 300	ISO 5389	ISO 18740	
				Option A	Option B
Pressure	100 kPa	101,325 kPa	101,325 kPa	100 kPa	100 kPa
Temperature	20 °C	20 °C	0 °C	20 °C	35 °C
Relative humidity	0 %	0 %	(Not indicated)	0 %	60 %
Coolant	20 °C	20 °C	(To be indicated)	20 °C	30 °C

3.4.2 Tolerances

To guarantee the performance of the compressor, the test conditions shall be as close as reasonably possible to the specified conditions of guarantee. Maximum deviations from these values are presented in Table 3.

Table 3. Maximum deviations from specified values during an acceptance test

	ISO 1217	BL 300	ISO 18740
Inlet pressure	± 10 %	± 10 %	± 5 %
Discharge pressure	± 2 %	± 1 %	-
Overall pressure ratio	-	-	± 2 %
Inlet temperature	-	± 10 K	± 8,5 K
External coolant quantity	± 10 %	± 10 %	± 5 %
Inlet temp. of external air coolant	± 10 K	-	± 8,5 K
Inlet temp. of external liquid coolant	± 5 K	± 15 K	
Liquid injection temperature	± 5 K	-	-
Speed	± 4 %	± 3 %	± 0,5 %

In ISO 1217 for multi-stage compressors with intercooling, the difference between gas inlet temperature and external coolant temperature shall not exceed to ± 2 K in the case of liquid and ± 4 K for air (ISO 1217, 39). In ISO 18740 allowable inlet temperature deviations of external coolants are same for both air and liquid coolants. The speed is specified as rotational speed for ISO 18740 and BL 300 and shaft speed for ISO 1217. In ISO 18740 the maximum deviations from specified values are more stringent than other standards because of the fixed speed.

For ISO 1217 and BL 300 the acceptable values for volume flow rate, specific power requirement and unloaded power requirement are defined in four different scales of volume flow rate. The compressor tested shall be considered acceptable when the test results remain within the allowances presented in Table 4.

Table 4. ISO 1217 and BL 300 maximum deviations permissible at test

Volume flow rate at specified conditions [$m^3/s \times 10^{-3}$]	Volume flow rate [%]	Specific power requirement [%]	Unloaded power requirement [%]
$0 < q_v \leq 8{,}3$	± 7	± 8	± 10
$8{,}3 < q_v \leq 25$	± 6	± 7	± 10
$25 < q_v \leq 250$	± 5	± 6	± 10
$q_v > 250$	± 4	± 5	± 10

Because of fixed speed, ISO 18740 tolerances for performance acceptance are not for different volume flow rate scales. The deviations allowed for acceptance are (ISO 18740, 5):

- Volume flow rate: ± 4 %
- Specific power requirement: ± 5 %
- Unloaded power requirement: ± 10 %

In ISO 5389 tolerances for acceptance are treated differently. It provides an extensive method for verifying guaranteed performances. In section 4.5 it is stated: "In case of an acceptance test, the test results measured and converted to the guarantee conditions shall be assessed against the values guaranteed (see Clause 8), making allowance for the limits of measuring uncertainties (see 6.4). Any manufacturing tolerances for the guarantee shall be deemed to constitute a component of the contract of supply and not of this International Standard." (ISO 5389, 8)

3.5 Additional standards

In addition to the performance test standards, there are other important standards concerning compressed air industry. These additional standards describe secondary performance parameters, which influence indirectly in the primary performance parameters described earlier. They also can be indispensable for the application of the performance test standards. These secondary parameters are for example gas quality, control capabilities and environmental issues. A few of them are presented in the next paragraphs.

3.5.1 ISO 5167

For different measurements methods, the performance test standards often refer to ISO 5167. It consists of five parts, under the general title “Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full”:

- Part 1: General principles and requirements
- Part 2: Orifice plates
- Part 3: ISA 1932 nozzles, long radius nozzles and Venturi nozzles
- Part 4: Classical Venturi tubes
- Part 5: Cone meters

ISO 5167-1 defines the general principles for methods of measurements and computation of the flow rate of fluid flowing in a conduit by means of pressure differential devices (orifice plates, nozzles and Venturi tubes) when they are inserted into a circular cross-section conduit running full. Also, the general requirements for methods of measurement, installation and determination of the uncertainty of the measurement of flow rate are specified. The standard applies only to single-phase subsonic flow. The measurement of pulsating flow is also out of scope. (ISO 5167-1, 1)

Detailed specifications for the devices used for measurements are specified in other parts of the standard. At the moment, “Part 6: Wedge meters” is under development.

3.5.2 **ISO 8573**

Quality standard ISO 8573 defines the allowable amounts of residual particles, water and oil for compressed air. The first part of the standard, ISO 8573-1 “Compressed air - Part 1: Contaminants and purity classes”, defines the purity classes of compressed air. The maximum number of particles, water and oil per cubic meter is specified for each class, independent of the location in the compressed air system at which the air is specified or measured. The standard is divided into nine parts. Parts 2-9 provide methods for testing different contaminations. Classes and values for all contaminations is presented in Table 5. Class zero doesn’t automatically mean zero contamination but the maximum level must be agreed by the user and the manufacturer. (Parker Hannifin 2010, 3)

Table 5. Compressed air purity classes (Parker Hannifin 2010, 3)

ISO8573-1:2010 CLASS	Solid Particulate				Water		Oil
	Maximum number of particles per m³			Mass Concentration mg/m³	Vapour Pressure Dewpoint	Liquid g/m³	Total Oil (aerosol liquid and vapour)
	0.1 - 0.5 micron	0.5 - 1 micron	1 - 5 micron				mg/m³
0	As specified by the equipment user or supplier and more stringent than Class 1						
1	≤ 20,000	≤ 400	≤ 10	-	≤ -70°C	-	0.01
2	≤ 400,000	≤ 6,000	≤ 100	-	≤ -40°C	-	0.1
3	-	≤ 90,000	≤ 1,000	-	≤ -20°C	-	1
4	-	-	≤ 10,000	-	≤ +3°C	-	5
5	-	-	≤ 100,000	-	≤ +7°C	-	-
6	-	-	-	≤ 5	≤ +10°C	-	-
7	-	-	-	5 - 10	-	≤ 0.5	-
8	-	-	-	-	-	0.5 - 5	-
9	-	-	-	-	-	5 - 10	-
X	-	-	-	> 10	-	> 10	> 10

3.5.3 **EN 1012-1**

For compressor safety regulations, a harmonised standard EN 1012-1 “Compressors and vacuum pumps - Safety requirements - Part 1: Air compressors” is used. Harmonised standards are developed by a recognized European Standards Organisation: CEN, CENELEC or ETSI, and they can be used to demonstrate that products, services or processes are complying with relevant EU legislation (European Commission 2017). EN 1012-1 is under Directive 2006/42/EC for Machinery.

The standard is applicable to compressors and compressor units designed to compress air, nitrogen or inert gases with an operating pressure greater than 0,5 bar. Parts 2 and 3 of the standard are for vacuum pumps and process compressors, respectively. The topics discussed in the standard are all significant hazards, hazardous situations and events relevant to the design, installation, operation, maintenance, dismantling and disposal of compressors and compressor units. (EN 1012-1, 4-5)

4 SIMULATION LIFECYCLE MANAGEMENT

Utilisation of computational methods with multidisciplinary models and simulation is considered as an important factor for progress in research and fast development of advance products. Nevertheless, lack of software tools for management and integration of the modelling and simulation data from multiple sources has been a bottleneck in the use of computational methods in research and industry.

The pressure in today’s market forces companies to innovate and bring new products at a rapid pace to the customer. This creates challenges in product development processes. The integration of functionalities from various disciplines becomes an important topic with increasing complexity of products. This integration is the source of innovation. At the same time, the development costs and delivery time to the market are being decreased. During the design process, digital tools are considered indispensable. They are also becoming more and more common within the whole product lifecycle. This leads to more complex systems, while the number of physical prototypes is decreased, and the amount and diversity of data is increased. The large amount of data is shared across the different teams, sometimes even worldwide. Often the teams are quite isolated from each other, because the tools are generally much specialised. (Sibois & Muhammad 2015, 6)

Because of the increased need for and importance of simulation, the information from simulation and analysis (S&A) is more and more recognised as valuable intellectual property, which has to be captured, shared and leveraged throughout the product lifecycle. In the new approaches, the product-related S&A is transformed into a visible and accessible component of the product development process. The purpose is to share the information across the full product lifecycle and enterprise, not just keeping it as a domain for specialists. Simulation across the various domains and disciplines is presented in Figure 25. (CIMdata 2011, 1)

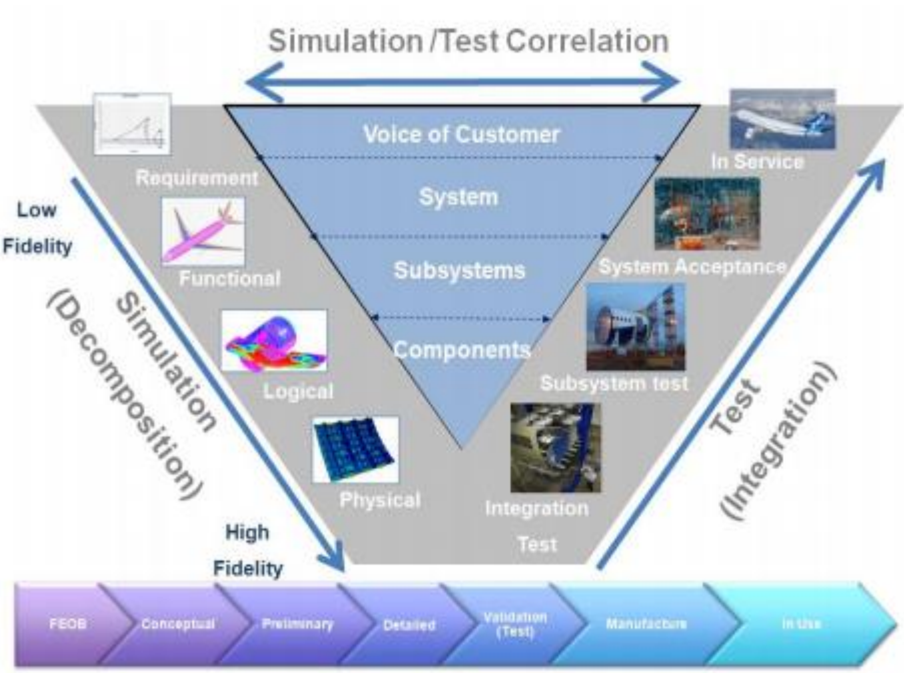

Figure 25. V-chart of the lifecycle of simulation (CIMdata 2011, 2)

The current trend in the industry is simulation-based product development, where the number of physical prototypes is reduced and computational analyses and simulations are increased. The design process is targeted around simulations and computational analyses. The consequence of this is the requirement of collecting data from various fields in various formats. This leads to a large amount of results data, when the efficient data management becomes important. (Sibois & Muhammad 2015, 6)

The evolution of the utilisation of simulation in the product process is presented in Figure 26. In the first phase, simulations are used to solve specific problems in a detail of a product, but the development of the product is driven by other factors. The second phase increases the requirements compared to the first phase by introducing the modelling and simulation of the whole product or large sub-systems of the product. The design of the product is still driven by traditional design methodologies. In the third phase, the modelling and simulation methods may remain the same compared to the second phase, but the traditional development approach is overtaken by simulation-based approach. In this case, at first the coarse model of the product is simulated and then the information obtained from the simulations is used in the design. This approach often requires major changes in the process. The fourth phase is the utilisation of the simulation-based approach to the management of the whole product lifecycle. In this approach, also the influence of the design decisions on business and environmental issues is taken into account.

This development trend sets high requirements on computational systems and data management. On the other hand, the connection between optimisation methods and simulations enables the efficient use of the computational resources, which due to the shortened time-to-market in product development may be the key to success in the markets. It also leads to more efficient use of resources and better understanding of the product lifecycle topics. (Kortelainen & Miettinen 2015, 13-14)

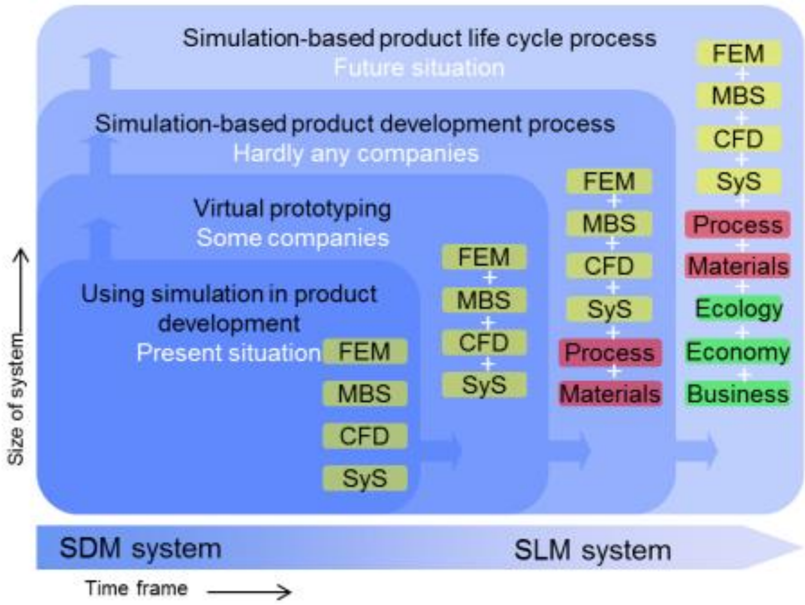

Figure 26. Evolution of application of simulation in product process (Sibois & Muhammad 2015, 8)

4.1 Simulation-based design process

Increasing complexity of the products makes physical prototyping more and more costly and time consuming. Simulations decrease the need of physical testing by virtually testing the designed product. Finding and correcting the design flaws in early stages has a significant influence on cost-effectiveness compared to correcting the flaws in later phases. This is one of the biggest added values of simulation-based design process.

In a simulation-based design process, simulations are the central part of the development of the product. It aims to verify the virtual prototype in order to avoid multiple iterations during the validation phase. This type of design process has numerous challenges, but the value of it is the reduction of the risk of errors and shortened product time-to-market. (Sibois & Muhammad 2015, 11)

The four phases of design process are illustrated in Figure 27. Requirements are used as input for design concept. In the design concept phase, the different design concepts are evaluated and one of the concepts is selected for further development. In the design product phase, detailed level simulations are performed for a 3D model of the system. The fourth phase consists of the verification of the virtual model and the model validation by way of a physical prototype. (Sibois & Muhammad 2015, 11-12)

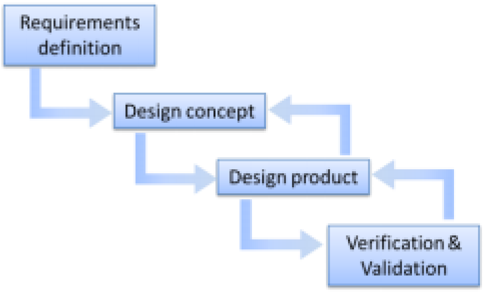

Figure 27. Product design process (Sibois & Muhammad 2015, 11)

4.2 Turbomachinery design process

Figure 28 presents a diagram of a simulation-based design process for turbomachinery. In this example, the preliminary design is made with CFturbo software. CFturbo is an interactive design software for turbomachinery components. It includes the design of impeller, vaneless and vaned diffuser and volute. The design process of turbomachinery is complex and cannot be calculated straightforward by a closed mathematical model. Turbomachinery design is an iterative process and to create a smooth workflow, CFturbo has interfaces to CAD, FEM and CFD systems. The design loop can also be automated using optimization software. (Kreuzfeld & Müller 2011, 2)

Starting point of the design is to define the design point parameters: flow rate, pressure ratio and rotational speed. Based on this data, a rough model of the turbomachine is created. The mesh is then generated for the coarse model. The flow part is solved with a CFD solver, and based on the results obtained from CFD calculations, the changes are made for the original design. The iteration loop is continued as long as the results from the simulations are satisfactory. After that, a prototype is built and tested. If the prototype corresponds to the simulations, it can enter to the production phase. It is possible to start manufacturing the product without the prototype phase, but this depends on the case.

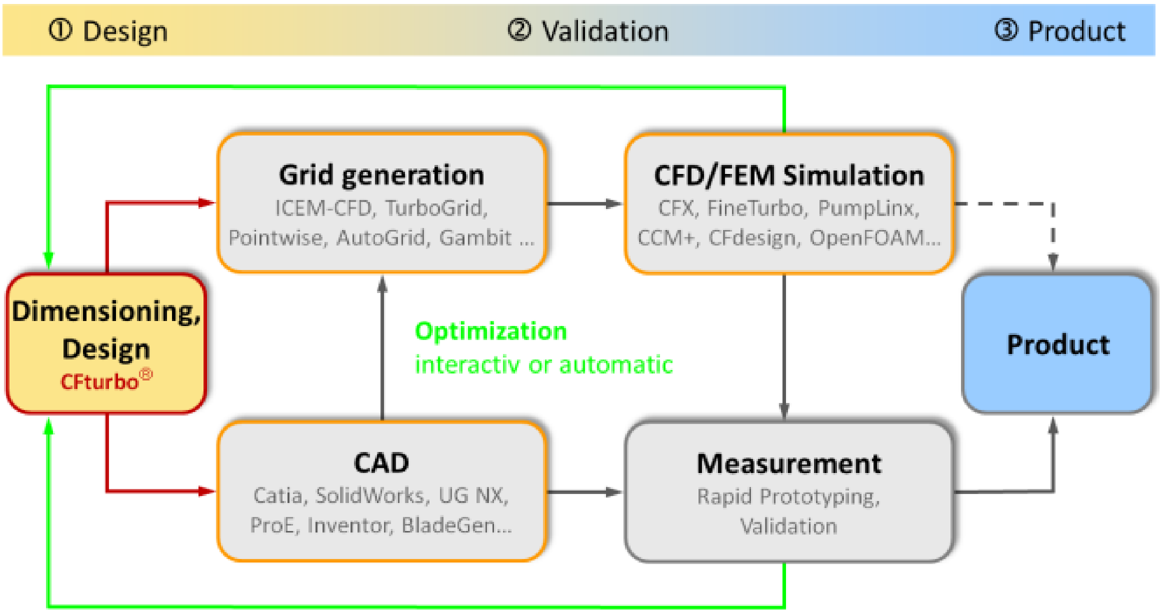

Figure 28. Turbomachinery design process (Kreuzfeld & Müller 2011, 2)

4.3 Definition of simulation lifecycle management

Simulation lifecycle management (SLM) is a complementary part of product lifecycle management (PLM) which associates behavioural simulation data and processes with the digital mock-up (DMU). Essentially it offers behavioural-digital mock-up (B-DMU). This provides a single source for all design and S&A information and processes. The purpose of SLM is to transform simulation from a specialty operation to an enterprise product development enabler that consists of many segments of the product lifecycle. (CIMdata 2011, 5)

The components of SLM are presented in Figure 29. These four foundational areas SLM should provide technology are:

- Simulation and test data management
- Simulation and test process management
- Decision support
- Enterprise collaboration

Figure 29. Components of SLM (CIMdata 2011, 5)

4.3.1 Simulation and test data management

Growing size of collections has made tracking and managing data more and more difficult. Crompton (2010) stated that engineers spend 30 % of their time looking for data, verifying data accuracy, and formatting data. When working on their personal space, the data can get lost when they leave. For this reason, simulation data management is a critical component of simulation analytics software.

Simulation and test data management tools make the data used and generated by simulations searchable, traceable, and associated with enterprise business data and practices. It manages the data used as part of simulation including geometry, simulation representations (e.g. meshes and models), input and output parameters, test conditions and options, supporting references and tools, and the results of the simulations executed. SLM is able to connect simulation information and processes to the product design information and structures. Information gained from simulations can be linked to specific components and versions of a product and its bill of materials (BOM). The simulation information can also be associated with product requirements so the validation of the requirements is possible to be tracked.

SLM has an architecture made for management and analysing of simulation and test data seamless. This means managing and minimizing file transfers over the network so the users have access to simulation data regardless of its size or geographic location. Data inside SLM is automatically revisioned, dynamically attributed and access controlled. Users can find the simulation information they need by making queries for simulation inputs and results and the status of simulation processes. (CIMdata 2011, 5)

SLM provides the same data management platform for both virtual and physical testing personnel to work anywhere in the world. It has a uniform data model for handling both the simulation and test data in a consistent manner in the product development environment. The purpose of SLM is not to displace the special applications dedicated to test data collection and preparation, but rather to connect them to company's primary design and engineering information systems, usually PLM. Thus, decision makers can access to the information during the product development process for making better and faster decisions. (CIMdata 2011, 6)

Future Market Insights (2016) has forecast that the simulation and test data management market will grow at 12,5 % compound annual growth rate during the forecast period 2016-2026. Demand for protection by data management software against loss of data generated is one of the main drivers in the market.

4.3.2 Simulation and test process management

Besides simulation data, SLM manages the simulation processes - what will be done, when, by whom, and where the results will be delivered for both use and to archive. SLM can manage the execution of integrated simulations and supporting tools. It can support applications from multiple disciplines, e.g. CFD, kinematics, cost, and mathematical models, allowing the full scope of simulation to be addressed. Assignment of simulations to the appropriate computing systems and the behaviour can be done by SLM. A range of flexible simulation processes provided include:

- Ad-hoc: Dynamic process that involves a high degree of user interaction
- Best Practice: A process with documented methods driven by analyst interaction
- Guided Practice: Domain-focused template methods to guide the user through a best practice; may be a combination of interactive and automated steps
- Fully Automated: A fully automated template driven process; complete with automatic in-process data management

Capture of the complete range of simulation processes allows organizations to distribute and develop simulation knowledge for maximum reuse and efficiency. Different simulation templates can be created by simulation experts and then used by other engineers. (CIMdata 2011, 6)

4.3.3 Decision support

Fundamentally, the simulations are performed to validate decision making based on functional, logical and physical requirements. SLM enables the design decisions by being capable of capturing and presenting simulation information and results. Organizations can quickly access and visibly associate simulation information with its design definition. Thus, simulation results can be interpreted to make collaborative design decisions. SLM also makes it possible to explore multiple design options with the application of multi-run design exploration methods. The information received can be used to collaborate with colleagues, partners or customers. (CIMdata 2011, 6)

4.3.4 Enterprise collaboration

SLM enables a secure simulation collaboration within an enterprise and across the value chain. All simulation activities are managed by the SLM system, and user roles and responsibilities can be defined to give each employee access to the data needed. They can only perform actions for which they are authorized. Companies can capture the best practices and make them usable throughout the product lifecycle. Anyone in the organization can then gain value from them.

The role of SLM in the system lifecycle is presented in Figure 30. SLM enables simulations to be a part of the product lifecycle and it focuses mainly on the virtual side of it. Simulation information and processes can be connected to requirements, parts, BOM and other elements of PLM. Verification and validation become more important. The evolution of the design and selection of certain designs is easier to see. The bridge between design and engineering is better provided with this exposure of simulation to the enterprise PLM. (CIMdata 2011, 7)

Figure 30. Product and simulation data management in the system lifecycle (Kortelainen et al. 2015, 82)

4.4 SLM tools

Tools for test data management are typically used for post-processing raw test data in a certain department or a test type. Focus on these tools has traditionally been in device or hardware acquisition and maintenance, and raw data collection and preparation and correction. This approach is efficient for the particular needs of the experimental team, but advantage for broader enterprise consumption of test data outside of the test lab remains limited. The increased value of both testing and simulation can be realized when the data and methods are maintained in a common platform. (CIMdata 2011, 4)

It has been noticed that software vendors have invariably targeted their SLM software development to large enterprises which has resulted in centralised applications. These are mainly beyond the means of small to medium enterprises (SMEs). This makes knowledge management, reusing existing expertise, and collaboration with other SMEs and larger enterprises difficult. Inversely, this also affects to the large enterprises that have SMEs in their supply chain. Another problem is the communication with suppliers and original equipment manufacturer (OEM) customers. Having similar tools on either side helps to make the workflow seamless and efficient. (Aziz et al. 2005, 261)

Aziz et al. (2005) have drawn up three main requirements for product and process information management for the needs of SMEs.

- Enabling project managers and all knowledge workers to have access to the applications needed to create and manage knowledge within their domain, according to the agreed nomenclature and ontological representation
- Information created has to be in a form that can be queried, reused and transformed into new representations through the use of rules and agents
- Enabling the real-time collaboration between SMEs and larger partners, by facilitating the fast and costless construction of virtual enterprises

4.4.1 Vendors

There are several commercial vendors for SLM tools. These tools are usually integrated in the PLM system as a module. Leading vendors for SLM tools are:

- Dassault Systèmes - SIMULIA
- Siemens PLM Software - Teamcenter for Simulation
- ANSYS - EKM (Engineering Knowledge Manager)
- MSC Software - SimManager
- Altair - HyperWorks

CPDA (2009) performed an analysis of these tools. The analysis included 60 criteria in four main categories: workgroup process support, simulation data management, PLM integration framework and utilities. The results of the analysis are presented in Figure 31. Siemens Teamcenter for Simulation achieved the highest ranking in the review. However, it has to be noticed that the survey was performed in 2009, and the programs have gone through changes ever since. Newer reviews were not found at the time of writing.

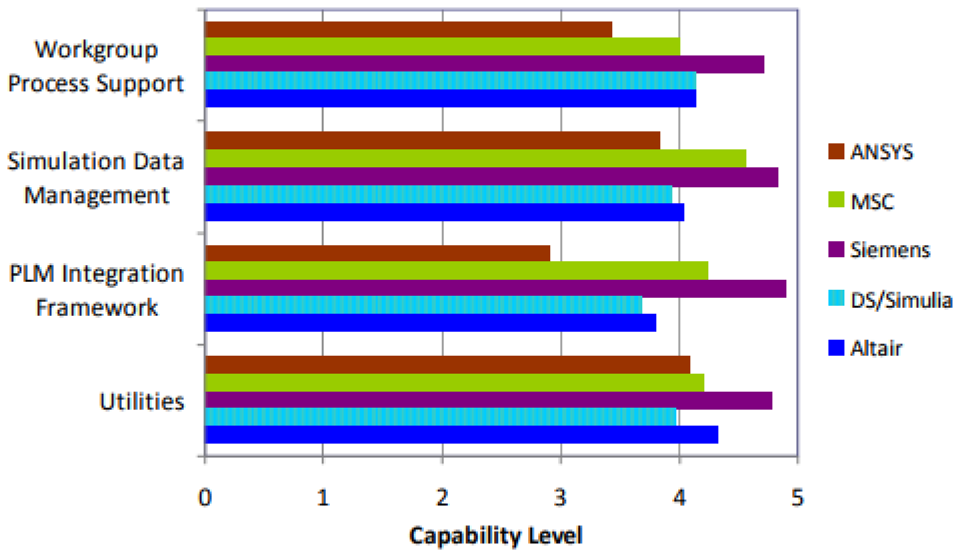

Figure 31. Scorecard of the SLM vendors (CPDA 2009, 3)

Sibois et al. (2015) made three case studies about commercial SLM solutions utilisation in industry. Tools chosen for studies were SIMULIA by Dassault Systèmes and EKM by ANSYS. First case study was a simulation process with CAD software applications implemented in the ANSYS Workbench and ANSYS EKM. Second case study consisted of large simulation files management in SIMULIA environment. Third case focused on Abaqus data storing and extraction in ANSYS EKM. Conclusion of the case studies was that the SLM tools can be used to implement an improved product development and lifecycle data management process. However, the studies revealed that practical implementation of SLM tools is yet far from seamless. Proprietary design and simulation tools integration is a major challenge and the implementation of processes requires expertise. Also, the price of SLM system and dependence on the vendors is a significant bottleneck, especially for SMEs.

5 VALIDATION OF CFD SIMULATIONS

As presented in Figure 25 and Figure 27, the last phase of the simulation lifecycle and simulation-based design process is verification and validation, where the virtual simulations performed earlier are being validated through physical testing. In this chapter, the fundamentals and importance of the validation is discussed. Also, factors causing errors in the validation are briefly examined. This thesis only concentrates on validation. Deeper examination of verification is excluded.

A critical issue concerning computational simulations is how should confidence in modelling and simulation be critically assessed. Verification and validation (V&V) of simulations are the means for building and quantifying this confidence. Simplified, verification is the assessment of the accuracy of the solution to a computational model by comparison with known solutions. It is primarily a mathematics issue and not in contact with the real world. Validation is the assessment of the accuracy of a computational simulation by comparison with experimental data. Validation creates a relationship between computation and the real world and it is mostly a physics issue. (Oberkampf & Trucano 2002, 211)

5.1 Fundamentals of CFD validation

There are several definitions for validation in literature. Society of Computer Simulations (SCS) defines model validation as: "Substantiation that a computerized model within its domain of applicability possesses a satisfactory range of accuracy consistent with the intended application of the model". Figure 32 presents the phases of modelling and simulation, which has two types of models: a conceptual model and a computerized model.

The conceptual model includes all information, mathematical modelling data and mathematical equations, that describe the physical system or process of interest. In case of CFD, it mainly consists of partial differential equations (PDEs) for conservation of mass, momentum, energy, and also the auxiliary equations, such as turbulence models and chemical reaction models, and all of the initial and boundary conditions of the PDEs.

The computerized model is an operational computer program that implements a conceptual model. In modern terminology the term computerized model refers to the computer model or code. (Oberkampf & Trucano 2002, 213)

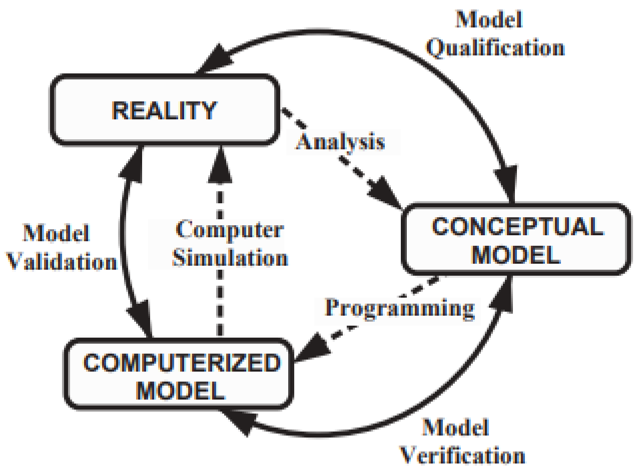

Figure 32. Role of verification and validation in modelling and simulation (Oberkampf & Trucano 2002, 213)

American Institute of Aeronautics and Astronautics (AIAA) defines validation as: "The process of determining the degree to which a model is an accurate representation of the real world from the perspective of the intended uses of the model". While being fundamentally similar to SCS's definition, it provides clearer and more intuitively understandable point of view for validation. Accuracy of the model is measured in relation to experimental data. However, it has to be taken into account that the model has to be verified before the validation can be considered reliable. (Oberkampf & Trucano 2002, 215)

Typically, the validation procedure in CFD, as well as other fields, is executed by graphically comparing the computational results and experimental data. Declaration of "validated" is usually given when the computational results "generally agree" with experimental data. According to Oberkampf & Trucano (2002), comparison of computational results and experimental data on a graph, is only incrementally better than a qualitative comparison. With a graphical comparison, quantification of the numerical error or quantification of computational uncertainties due to missing initial conditions, boundary conditions, or modelling parameters may remain unnoticed. Also, it does not clearly show the variation over the range of the independent variable, e.g. space or time, or the parameter of interest, e.g. Reynolds number or a geometric parameter. It is suggested that validation quantification should be considered as the evaluation of a metric, or a variety of appropriate metrics, which would quantify the errors and uncertainties between the computational results and experimental data. This topic is discussed in section 5.4. (Oberkampf & Trucano 2002, 216)

The fundamental validation strategy includes several factors: identification and quantification of the error and uncertainty in the conceptual and computational models, quantification of the numerical error in the computational solution, estimation of the experimental uncertainty, and finally, comparison between the computational results and the experimental data. The data from experimental measurements is not assumed to be more accurate than the computational results but most faithful reflection of reality for the purposes of validation. Figure 33 presents the validation process and comparison of the simulations with experimental data from various sources. (Oberkampf & Trucano 2002, 218)

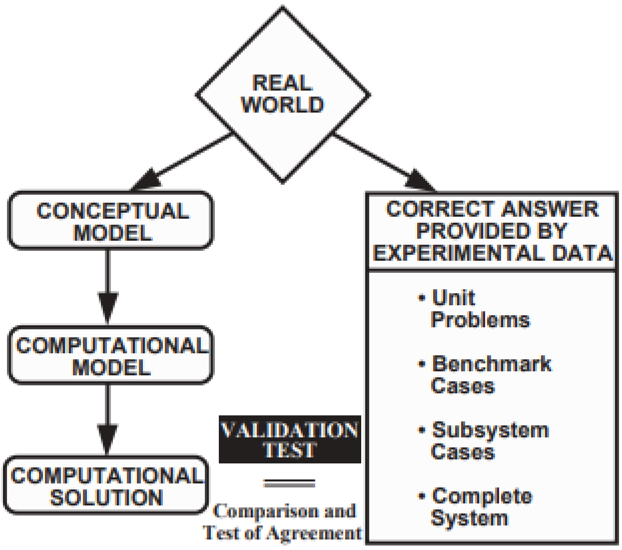

Figure 33. Validation process (Oberkampf & Trucano 2002, 218)

Oberkampf & Trucano (2002) divide validation experiments in three categories:

- Purpose of improving the fundamental understanding of some physical process
- Constructing or improving mathematical models of fairly well-understood flows
- Determining or improving the reliability, performance, or safety of components, subsystems, or complete systems

Validation of compressor performance falls into the last category. These experiments are often called “tests” of engineered components and systems in the industries.

5.2 Guidelines for validation experiments

Several philosophical guidelines for conducting validation experiments are presented by Oberkampf & Trucano (2002). The guidelines were developed in a joint computational and experimental program, and they can be applied over the whole range of fluid dynamics. The most important guidelines for the company are discussed in this section.

Guideline 1 proposes: “A validation experiment should be jointly designed by experimentalists, model developers, code developers, and code users working closely together throughout the program, from inception to documentation, with complete candour about the strengths and weaknesses of each approach.”

Guideline 1 demands for intensive cooperation between the CFD and experimental teams. However, it is easier said than done. Different backgrounds between theoretical and experimental personnel may cause cultural hurdles that are difficult to overcome. Also, competition for funding or recognition may affect to the openness and teamwork.

Guideline 2 suggests: “A validation experiment should be designed to capture the essential physics of interest, including all relevant physical modelling data and initial and boundary conditions required by the code.”

Experimental team has to understand the assumptions and requirements of the model so that the test can match the simulation. In case of compressors, it means that the boundary conditions (inlet conditions, rotational speed etc.) defined in the model must be achievable. On the other hand, the computational team must understand the experimental facilities and their limits, and thus define physically realizable boundary conditions. Referring to guideline 1, proper communication between the teams is necessary to carry out an appropriate validation experiment.

Guideline 3 instructs: “A validation experiment should strive to emphasize the inherent synergism between computational and experimental approaches.”

Synergism refers to the activity, either CFD or experiment, which creates improvements in the capability, understanding, or accuracy of the other approach. This benefit is considered as one of the primary values of validation experiments. The experimental team can use CFD simulations to improve the design and instrumentation of the experiment. For example, the locations of the material temperature measurements can be defined on the basis of the information of the hottest and coldest spots predicted by simulations. Again, the teams must share their knowledge with each other to achieve the benefits.

In guideline 4 it is said that: “Although the experimental design should be developed cooperatively, independence must be maintained in obtaining both the computational and experimental results.”

The reason for this guideline is that it is so common to calibrate the CFD codes to the experimental results, that many people do not see the difference of calibrating versus validating. The independence between the teams should be kept by giving careful attention to procedural details. In practice, the experimental team should not give the reduced and analysed test results initially to the CFD team, but complete details of the physical modelling parameters and the initial and boundary conditions of the experiment, exactly as it was conducted. The errors and uncertainties must be quantified by the simulation team, and then the results be presented for comparison with experimental data. Typically, there will be agreement on some measured quantities and disagreement on other quantities. After the discussion, some checks can be made that may improve the agreement or not. However, these discussions and iterations are beneficial to the both teams, referring to guideline 3. It is recommended that management should not be involved in this procedure, so the unnecessary comparison of the teams does not poison the teamwork. (Oberkampf & Trucano 2002, 247-250)

It is understandable that following these guidelines requires effort and probably change of attitudes, but it will be rewarding in the long run. Although it may initially cause extra costs and work, improvements in test accuracy and teamwork will be significant.

5.3 Uncertainty and error

CFD simulations always contain uncertainty and error which makes the simulation results to differ from their true or exact values. Here are presented the fundamentals of uncertainty and error determination. Deeper examination is excluded from this thesis.

AIAA (1998) has given definitions for both terms. Uncertainty is defined as: "A potential deficiency in any phase or activity of the modeling process that is due to the lack of knowledge." Error is defined as: “A recognizable deficiency in any phase or activity of modeling and simulation that is not due to lack of knowledge.” The key difference between the definitions is “lack of knowledge”.

“Lack of knowledge” primarily refers to the lack of knowledge of the physical processes involved in building the model. The definition of uncertainty states that the deficiencies may or may not exist. Uncertainty can be determined with sensitivity and uncertainty analyses. For example, there is a lot of lack of knowledge in turbulence modelling, so uncertainty can be used to describe the deficiencies in turbulence models. To determine the level of uncertainty and its effect to the analysis, one can run a number of simulations with a variety of turbulence models and examine the effect of the model to the results.

The definition for error signifies that the deficiency is identifiable upon examination. Errors can be divided to acknowledged or unacknowledged errors. For acknowledged errors (e.g. round-off error and discretization error) there are procedures for identifying them and possibility of removing them. Otherwise they can remain in the code with their error estimated and listed. Unacknowledged errors (e.g. computer programming errors or usage errors) have no set procedures for finding them and may remain within the code or simulation. Errors can also be classified as local and global errors. Local error refers to a mesh point or cell, and global to over the entire flow path. Local errors can be transported, advected and diffused throughout the mesh. (NPARC Alliance 2008)

In turbomachinery simulations, there are several reasons why the simulation results usually do not match exactly the test results. Like any analytical tool, CFD approximates the real world, and the mathematical models are not perfect. As discussed above, especially turbulence models are problematic, but also other factors like surface roughness effects, heat transfer convection coefficients, gas properties and transitions between laminar and turbulent flow may cause inaccuracies. Due to the practical limit on computer resources, the mesh cell sizes and time steps in simulations are not always as small as would be optimum. Usually the simulations are run steady-state, even though the turbomachinery flow field is fundamentally transient. Models do not always take into account the changes in shape because of heat and rotation. Boundary conditions applied to the model have a significant influence on simulation results. For compressors, surge is influenced by the entire downstream piping system, which is not included in the model. Inlet conditions are usually assumed uniform, although they are not exactly in real world. Heat transfer through walls is typically neglected and real values can be difficult to set correctly if considered. (Sorokes et al. 2016, 10)

5.4 Validation metrics

As discussed in section 5.1, quantitative methods for comparing computational and experimental results are needed, instead of traditional qualitative graphical comparisons. Figure 34 presents the role of the validation metric in the validation process. Validation metric refers to the mathematical procedure that operates on the computational and experimental system response quantities (SRQs). The SRQ is a physically measurable quantity, or a quantity that is based on, or inferred from, measurements. It can involve derivatives, integrals, or more complex data processing of computed or measured quantities. (Oberkampf & Barone 2006, 9)

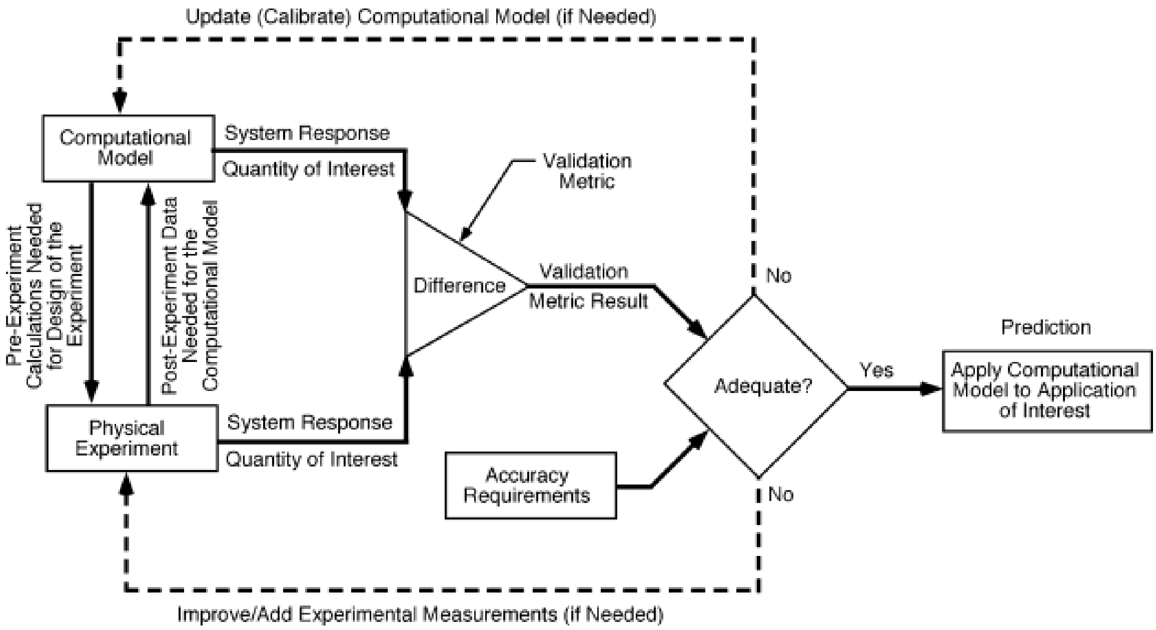

Figure 34. Model validation, calibration and prediction (Oberkampf & Barone 2006, 8)

According to Oberkampf & Barone (2006), a validation metric should include a numerical error estimation in the SRQ of interest resulting from the computational simulation. The numerical error can also be excluded, if it is estimated to be small by some reasonable means. A metric should quantitively evaluate predictive accuracy of the SRQ of interest. It should include modelling assumptions, physics approximations, and previously obtained physical parameters embodied in the computational model. Validation metric should also include an estimate of the error resulting from postprocessing of the experimental data, such as the construction of a regression function. Also, an estimate of the measurement errors in the experimental data should be included in the metric. Any indications of the level of adequacy in agreement between computational and experimental results should be excluded. This means that for example value judgments, such as “good” or “excellent” should be separated from the metric. (Oberkampf & Barone 2006, 11-12)

ASME has published a standard establishing detailed procedures for V&V of CFD simulations. The standard is called “ASME V&V 20: Standard for Verification and Validation in Computational Fluid Dynamics and Heat Transfer”. The objective of the standard is the specification of a verification and validation approach that quantifies the degree of accuracy inferred from the comparison of solution and data for a specified variable at a specified validation point (ASME 2009, 1).

As seen in Figure 35, comparison error *E* is a difference between the result of a simulation *S* and a result of an experiment *D* at a particular validation point. The parameters used in this example are temperature and Reynolds number.

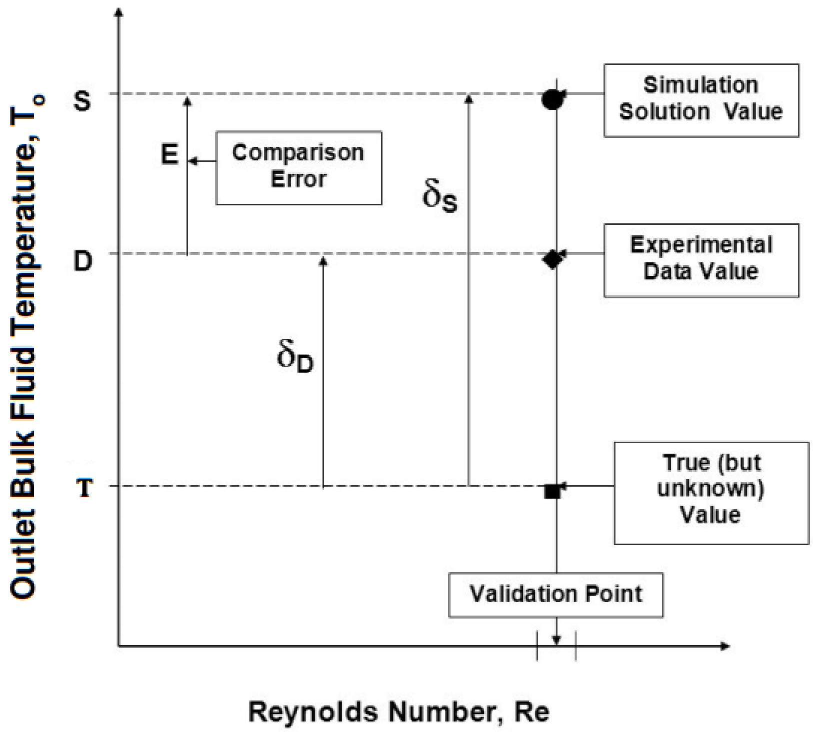

Figure 35. Validation comparison (ASME 2009, 3)

The validation comparison error E is defined as

$$E = S - D = \delta_S - \delta_D \tag{35}$$

δ_S	Simulation error
δ_D	Experimental error

The validation comparison error is a combination of all of the errors in simulation and experimental results, and its sign and magnitude are known once the validation comparison is made. (ASME 2009, 3)

The sources of error in validation process are presented in Figure 36. Simulation error can be decomposed in three categories:

- Modelling error δ_{model} that is due to assumptions and approximations
- Numerical error δ_{num} that is due to the numerical solution of the equations
- Input error δ_{input} that is due to errors in simulation input parameters

The objective of a validation exercise is the estimation of the modelling error within an uncertainty range. Modelling error can be written as

$$\delta_{model} = E - (\delta_{num} + \delta_{input} - \delta_D) \tag{36}$$

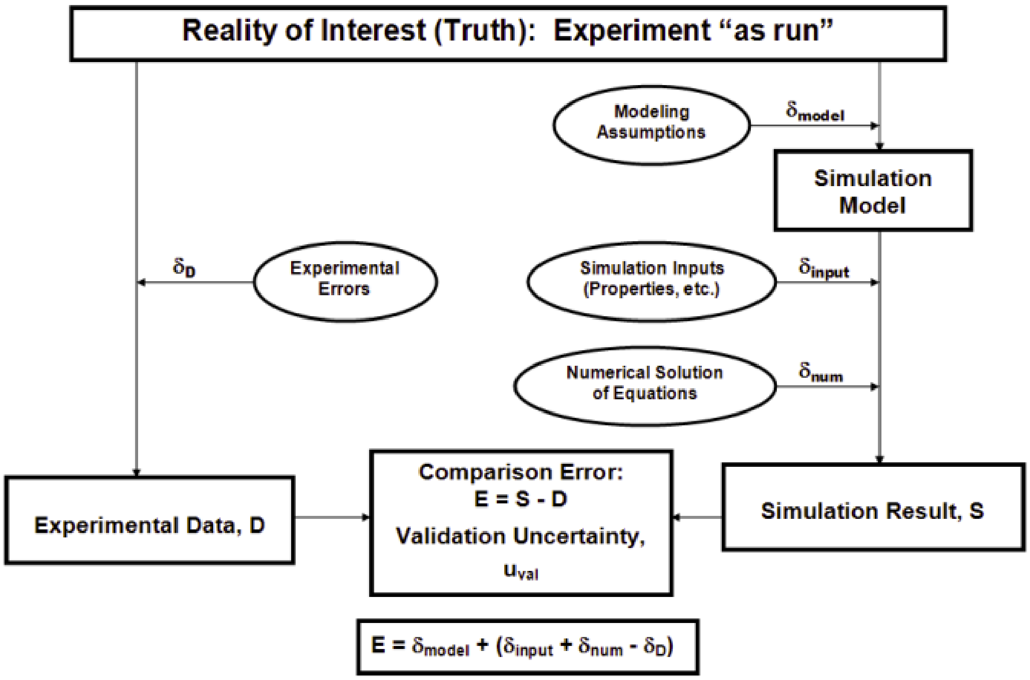

Figure 36. Validation process with sources of error (ASME 2009, 4)

Validation standard uncertainty u_{val} is defined as an estimate of the standard deviation of the parent population of the combination of the errors (δ_{input} + δ_{num} - δ_D). Modelling error can therefore be presented as

$$\delta_{\text{model}} = E \pm u_{\text{val}} \tag{37}$$

The estimation of u_{val} is at the core of the validation process. Thus, E and u_{val} are the validation metrics. Assuming that the errors are effectively independent, u_{val} can be defined as

$$u_{\text{val}} = \sqrt{u_{\text{num}}^2 + u_{\text{input}}^2 + u_{\text{D}}^2} \tag{38}$$

Estimation for numerical uncertainty u_{num} can be made by code verification and solution verification. Code verification establishes that the code accurately solves the conceptual model included in the code. Solution verification estimates the numerical accuracy of a particular calculation.

Input uncertainty u_{input} can be estimated by two methods:

- Sensitivity coefficient (local) method that requires estimates of simulation solution sensitivity coefficients
- Monte Carlo (sampling, global) method that makes direct use of the input parameter standard uncertainties as standard deviations in assumed parent population error distributions

Experimental uncertainty u_D can be estimated by using well-accepted experimental uncertainty analysis techniques presented in ISO test performance standards, or ASME PTC 19.1 “Test Uncertainty” standard, for example. Detailed techniques for estimating the validation uncertainties are presented in ASME V&V 20 standard. (ASME 2009, 4-5)

6 CASE STUDY: VALIDATION OF EFFICIENCY AND PRESSURE RATIO

A case study was made to validate efficiency and pressure ratio for a centrifugal compressor. The validation consisted of performing the test, running the simulation and comparison between the results gained from them.

6.1 Test procedure

The test was performed with a 3-stage centrifugal compressor. Intercooling was applied between the stages and aftercooling after the last stage. Efficiency and pressure ratio were calculated for the first stage of the compressor. For the performance calculations, static pressure, temperature, flow rate and relative humidity of the air were measured before the stage. After the stage, outlet static pressure and temperature were measured. Outlet flow rate measurement was not possible to execute. Electric signals from the measurement devices were transformed and collected with a data acquisition system. The compressor was operated with programmable logic controller (PLC) system. A piping and instrumentation diagram of the test measurements is presented in Figure 37.

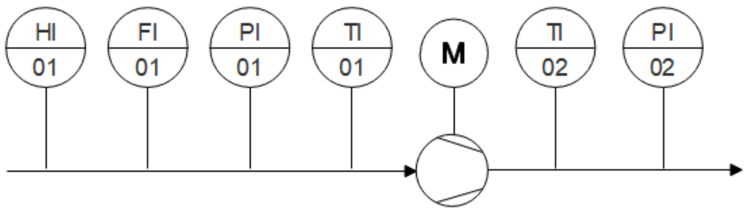

Figure 37. P&ID of the test measurement instrumentation

Before the test, the compressor was warmed up, so the measurement values were stabilised. The compressor was run with constant speed and the data was collected in four different operating points. At each point, one sample per second for one minute was logged. The test conditions were maintained in an acceptable range.

6.2 Boundary conditions

Boundary conditions in the CFD simulations were set according to the test conditions. The simulation was made with NUMECA FINE/Turbo software. Parameters measured during the test and given for defining the boundary conditions were total inlet temperature, total inlet pressure and outlet mass flow.

Other parameters for boundary conditions were:

- Turbulence model: Spalart-Allmaras with extended wall function
- Number of cells in the mesh: $5 \cdot 10^6$
    - Impeller: $2{,}5 \cdot 10^6$ with periodical boundary conditions
    - Diffuser: $1{,}8 \cdot 10^6$ with periodical boundary conditions
    - Volute: $0{,}7 \cdot 10^6$
- No heat exchange between the surroundings and fluid
- No leakages in the stage
- Fluid model: Air real gas
- Impeller-diffuser interface: Conservative coupling by pitchwise row
- Diffuser-volute interface: Full non-matching mixing plane

6.3 Results

Data used in calculations was averaged from the test data in Excel with TRIMMEAN function. It excludes the percentage of data points from the top and bottom tails of a data set and calculates the average from the rest of the points. Percentage used was 30 %, which means that in the 60-point data set, 9 highest and 9 lowest values were excluded in the averaging. The simulation results were provided in a .xml-file. The result file presents integrated values in the inlet, outlet and interface of the impeller and diffuser. The values needed in the comparison were chosen from the file.

6.3.1 Efficiency

Isentropic total-to-total and total-to-static efficiencies were calculated according to equation (21) based on the averaged measurement data. When analysing the test results and comparing them to the simulation results, it was shown that the efficiency based on the test data was more than 10 percentage points higher than the efficiency predicted by the simulation.

Single values were compared between each other, and the problem was tracked down to the outlet temperature, which showed much lower value than the simulation, causing the higher efficiency.

To compare the efficiencies to the existing correlations, specific speed was calculated. Specific speed is defined as

$$N_s = \frac{\omega\sqrt{q_{v1}}}{\Delta h_s^{0,75}} \tag{39}$$

Comparing the results to the several N_s-η_s correlations, the efficiency predicted by the simulations seemed to be more credible than the efficiency based on the test data.

A rough simulation of volute outlet temperature profile was made. The temperature profile is presented in Figure 38. According to the simulation, the temperature gradient inside the pipe is high, causing a relatively big temperature difference between centre and wall of the pipe. The thermometer was located near the wall, so most likely the average temperature inside the pipe was higher than the measured value. The pipe was not properly insulated, which increased the heat exchange from the fluid to the surroundings.

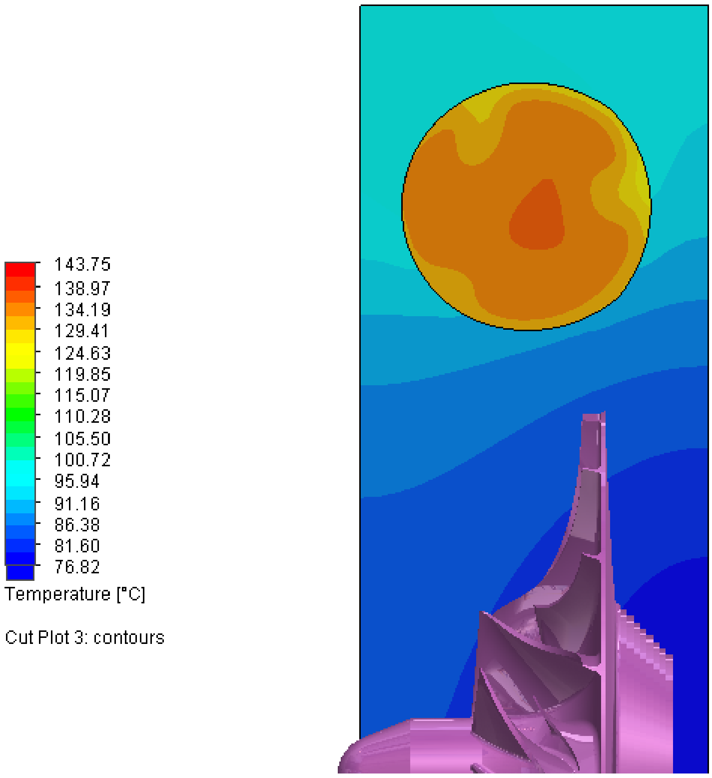

Figure 38. Simulation of volute outlet temperature profile

The test was not possible to repeat with corrections due to the lack of time, and for this reason, a proper comparison of efficiencies was not possible to perform. The upside of the test was that the influence of the location of the thermometer in the pipe can now be tested. Also, the importance of insulation was noticed.

6.3.2 **Pressure ratio**

In spite of the unsuccessful efficiency comparison, pressure ratio was possible to compare. The influence of the outlet temperature to the pressure ratio is negligible, so a reliable validation can be made. The comparison between the test results and simulation results was made for both static and total pressure ratio.

The relative uncertainty of test results for the pressure ratio was calculated according to ISO 5389

$$\tau_{\pi} = \frac{1}{X_{\mathrm{N}}^2}\sqrt{(\ln \pi)^2(4 \cdot \tau_{\mathrm{N}}^2 + \tau_{\mathrm{T1}}^2 + \tau_{\mathrm{R}}^2 + \tau_{\mathrm{Z1}}^2) + \tau_{\mathrm{p1}}^2 + \tau_{\mathrm{p2}}^2} \tag{40}$$

τ	Relative uncertainty	[-]
X_{N}	Ratio of reduced speeds of rotation	[-]
π	Pressure ratio	[-]

Relative uncertainties for rotational speed, gas constant and compressibility were neglected in the calculation. Measuring uncertainties provided by the manufacturers were for inlet temperature measurement ±0,2 °C and for inlet and outlet pressure transmitters ±0,1 % of the span. Based on this information and the measurement values, relative uncertainty of test results for the pressure ratio was ±0,7 %. According to ASME V&V 20, also numerical and input error should be estimated for validation standard uncertainty, but they were excluded from this thesis due to the lack of time.

For the results presentation, normalized mass flow and normalized rotational speed was calculated in each measuring point. Normalized values are defined as a ratio of measured value and design value.

$$q_{m,\mathrm{N}} = \frac{q_{m,\mathrm{meas}}}{q_{m,\mathrm{des}}} \qquad N_{\mathrm{N}} = \frac{N_{\mathrm{meas}}}{N_{\mathrm{des}}} \tag{41}$$

Comparison for static pressure ratio is presented in Figure 39. Bars for experimental uncertainty are added in the experimental results.

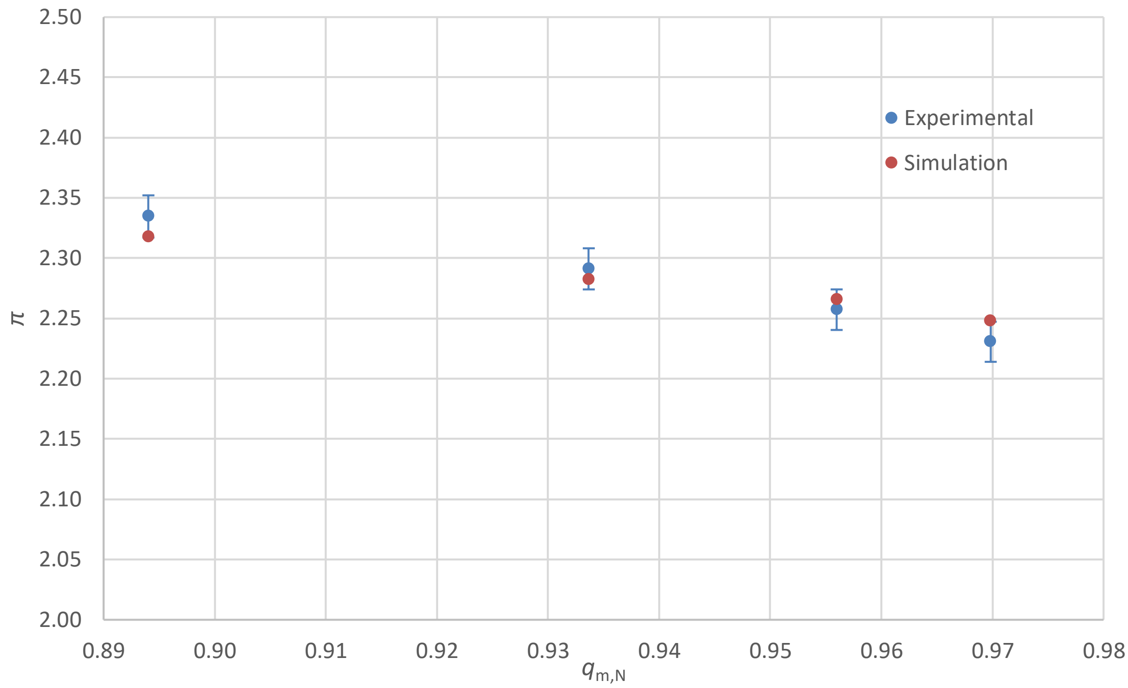

Figure 39. Static pressure ratio (N_N = 0,91)

Validation error and uncertainty for static pressure ratio is presented in Figure 40. Error was calculated according to equation (35), added with validation uncertainty (equation (37)).

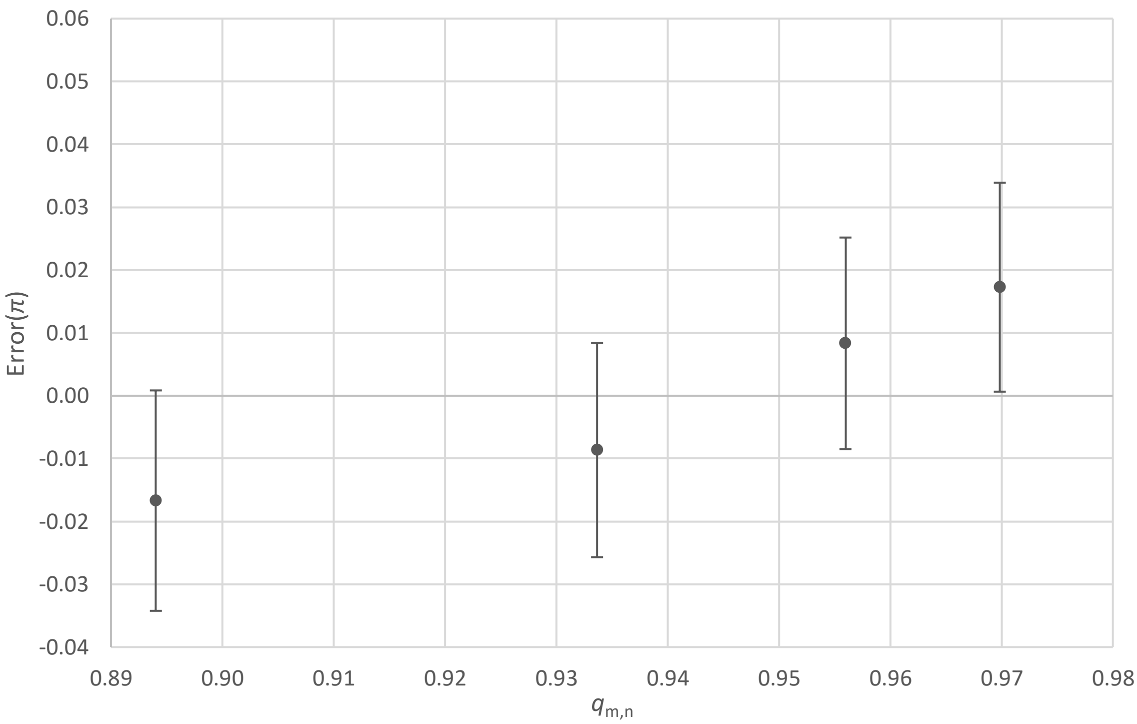

Figure 40. Validation error and uncertainty for static pressure ratio

Similar comparison and validation error calculation was performed for total pressure ratio. They are presented in Figure 41 and Figure 42, respectively.

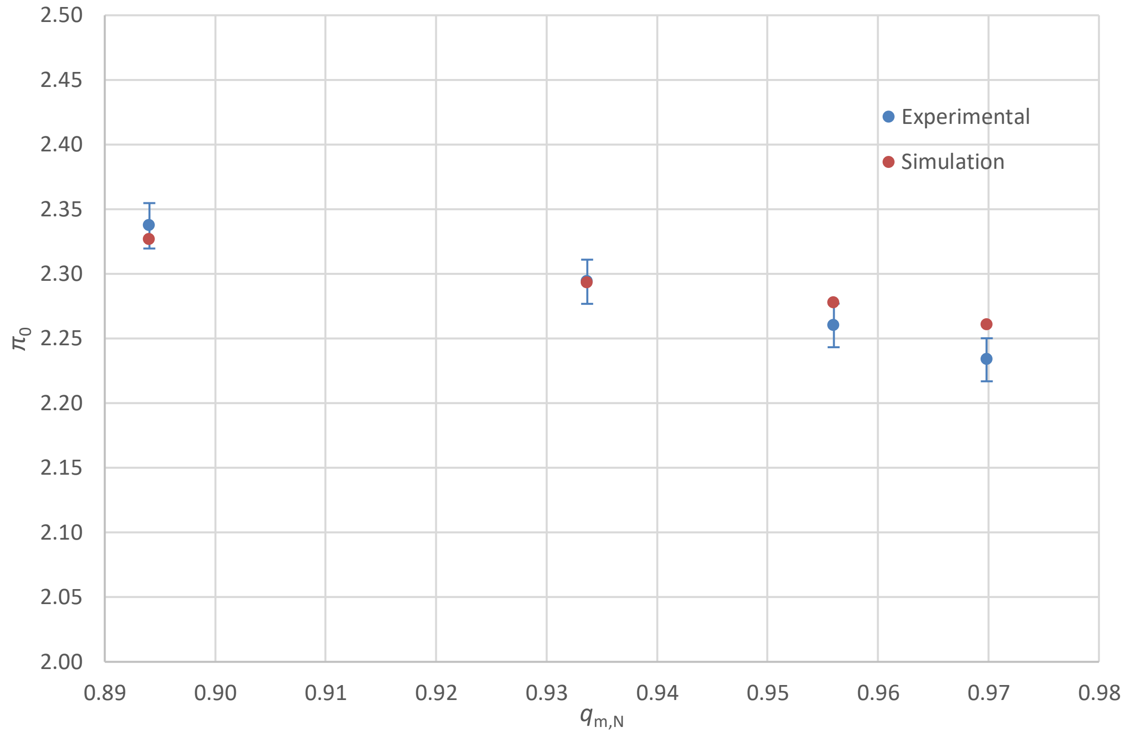

Figure 41. Total pressure ratio (N_N = 0,91)

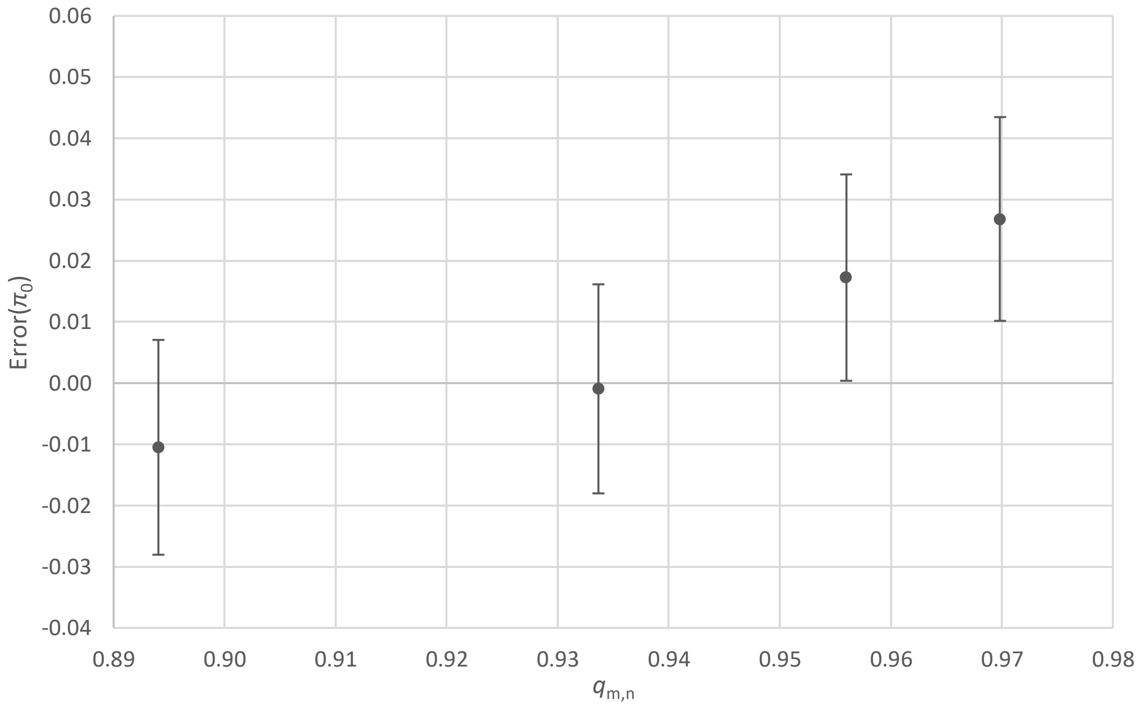

Figure 42. Validation error and uncertainty for total pressure ratio

It can be observed in the figures that the best correspondence between the test and simulation occurs at moderate pressure ratios. With lower pressure at $q_{m,N}$ = 0,97, the validation error is highest, pressure ratios not being even inside the uncertainty range. Pressure ratio prediction for lower pressures in the simulation is more difficult than higher pressures, because of the higher probability for flow separation and losses, for example.

Both static and total pressure ratio comparisons show similar behaviour. Because of the lack of outlet flow measurement, leakages in the stage were neglected in the calculation. Thus, the real value for the outlet total pressure is a bit smaller.

Even though the validation error is relatively small, the results still cannot be completely stated as validated, because of the inadequate uncertainty estimation. Especially the reasons for difference in pressure ratio at higher mass flows must be investigated. More complete test data is required for detailed analysis of the stage elements, including total and static pressures and temperatures at the outlet of the impeller and diffuser. Also, the amount and effect of leakage shall be examined.

Possible changes in validation error with CFD can be achieved by, e.g. changing the turbulence model or increasing the solver precision from single to double. An option recommended in the literature for turbomachinery is the SST k-ω turbulence model (CFD Online 2015). Also increasing the mesh size may have an effect to the to the correspondence to a certain limit.

Effect of the turbulence model change for static pressure ratio was tested for $q_{m,N}$ = 0,97 and $q_{m,N}$ = 0,956 test points. The turbulence models tested, in addition to Spalart-Allmaras, were Baldwin-Lomax, k-epsilon and SST. Pressure ratio with each turbulence model is presented in Figure 43 and Figure 44.

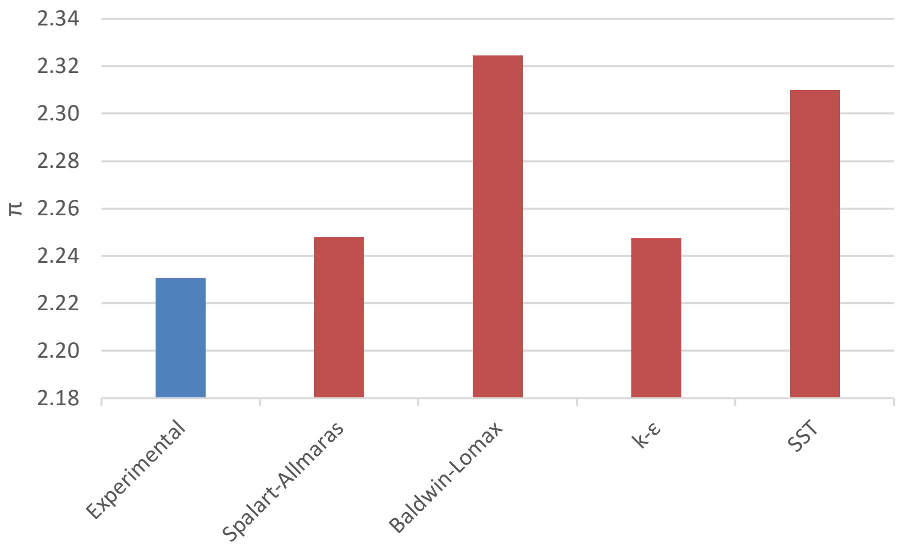

Figure 43. Static pressure ratio with different turbulence models ($q_{m,N}$ = 0,97)

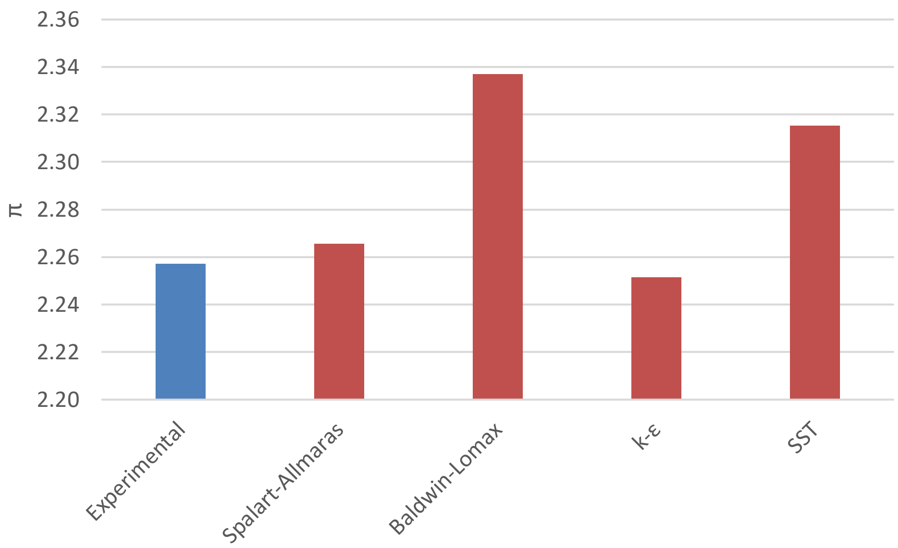

Figure 44. Static pressure ratio with different turbulence models ($q_{m,N}$ = 0,956)

According to the figures, the best correspondence between the experimental and simulation values is achieved with Spalart-Allmaras and k-epsilon turbulence models. Surprisingly, SST differed much from the experimental value, despite the recommendations. Baldwin-Lomax had the fastest convergence, but the correspondence to the experimental value was the worst. The convergence time with Spalart-Allmaras was smaller than with k-epsilon, so there is no need for changing the turbulence model.

7 CONCLUSIONS

Objective of this thesis was to validate CFD simulations for the compressor performance, and to make the validation process more efficient. Due to the practical limits and setbacks, some topics that were planned to be investigated, were left for the future work. Even though the objective of the thesis was not completely achieved, a lot of essential information for improving the validation process was gained. Based on this information, several improvement proposals are listed.

7.1 Suggested improvements

Complexity of ISO 5389 standard makes it difficult to follow precisely. For the performance testing in the future, it is advised to check up on the feasibility of the simplified version of it, ISO 18740. Also, when entering the American market, implementation of PTC 10 performance test code has to be considered.

Information from simulations and tests is an important factor to be acknowledged when implementing the new PLM system. The first step in efficient management of intellectual property provided by simulations is the management of simulation data. Seamless sharing of knowledge decreases the amount of useless work and breakdowns in communication in the design process.

In the validation process and other R&D-related processes, the cooperation and communication are important between the simulation and experimental teams, as stated in the guidelines for validation. It has been noticed that the cooperation between product development and test laboratory staff in the company is very limited at the moment. The teams are too isolated from each other. Especially due to the different locations of the teams, much effort should be addressed to this issue.

For making the proper validation possible, the uncertainties for numerical and input error should be defined and total validation uncertainty calculated. Knowledge of the uncertainty in all parameters brings reliability in validations and possibility of removing some errors after identifying them.

Calculation of efficiency was mainly made by hand. To make the calculation more efficient and error-free, the equations and correlations used in the calculation process could be implemented in the laboratory PLC system. During the test, the efficiency and influence of the different parameter change to it would be instantly on view.

The importance of the insulation and location of the thermometer in temperature measuring was acknowledged after the analysis of the test results. The temperature of air, especially inside the outlet pipe varies a lot, which is why the optimal location for the thermometer have to be examined to get reliable measurement results in the future.

When performing the simulations with CFD, there are many parameters to be chosen by the user. Some parameters have stronger effect on the simulation results than others. The influence of the different parameter change in validation error is an important topic to be investigated.

8 SUMMARY

Even though the structure of the centrifugal compressor is fairly simple, understanding the performance characteristics is rather complicated. Centrifugal compressor can be equipped with variety of auxiliaries, each of them differently influencing in the performance. When discussing about the compressor efficiency, it is essential to understand the differences between the different definitions of efficiency. It is also important to be aware of the limitations of performance.

Performance test standards enable comparability of performance statements for different compressors. They define the test procedure and tolerances for acceptance. Therefore, it is important to understand the aspects of different performance standards when comparing the performance characteristics with each other. There are also several additional standards which are referred to in the performance test standards.

Simulation lifecycle management strives to solve the bottleneck problems in management and integration of the modelling and simulation data from multiple sources. Turbomachinery design is an iterative process, where the efficient data management and clear workflow is required. SLM consists of simulation and test data management, simulation and test process management, decision support and enterprise collaboration. There are several commercial vendors for SLM tools, but these tools have traditionally been targeted for large enterprises.

To find the confidence of the numerical results provided by CFD, the simulation results have to be validated. Validation is the assessment of the accuracy of a computational simulation by comparison with experimental data. For a proper validation, quantitative methods for comparing computational and experimental results are needed, instead of traditional qualitative graphical comparisons. Intensive cooperation between simulation and experimental teams is in the key role for a good validation experiment.

A case study was made to validate efficiency and pressure ratio for a centrifugal compressor. Due to the problems in outlet temperature measurement in this compressor, reliable validation for efficiency was not possible to perform. For pressure ratio comparison, decent correspondence between the test and simulation was found, but due to the inadequate uncertainty estimation, results cannot be completely stated as validated. For future work, improvement proposals for more efficient validation were listed.

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