LAPPEENRANTA UNIVERSITY OF TECHNOLOGY
FACULTY OF TECHNOLOGY
ENERGY TECHNOLOGY

MASTER’S THESIS

# DESIGN OF A TEST STAND FOR A CENTRIFUGAL COMPRESSOR

Supervisors and examiners: Professor, D. Sc. Jari Backman
D. Sc. Ahti Jaatinen-Värri

Lappeenranta, 2013

Mikhail Chernov

# ABSTRACT

Lappeenranta University of Technology
Faculty of Technology
Degree Programme in Energy Technology

Mikhail Chernov

**Design of a test stand for a centrifugal compressor**

Master's Thesis
2013
59 pages, 34 figures, 2 tables, and 6 appendixes.

Examiners: Professor, D. Sc. Jari Backman
D. Sc. Ahti Jaatinen-Värri

Keywords: Test stand, centrifugal compressor, flow measurements.

The goal of the Master's thesis is to design a test stand for a centrifugal compressor. Different theoretical aspects of flow parameters measurements and test rigs built for the similar purposes in other research units are described in the theoretical part of the work. The process of components selection and the description of chosen components are given in the second part of the thesis. Besides measuring and control stages, the designed test stand has a closed-loop piping, an aftercooler and a surge tank. Overview and layout of the test rig is presented in the last chapter of the work.

## ACKNOWLEDGMENTS

First of all, I am forever grateful to my parents, Victor and Alevtina, who supported me during my staying in Finland. I really appreciate your help, without you it would be impossible.

I would like to thank my supervisors, Professor Jari Backman and D. Sc. Ahti Jaatinen-Värri, for their huge help in this work and in writing of this Master’s Thesis. I am also very grateful to all teaching staff of LUT Energy department, especially to Professor Esa Vakkilainen, the head of the Degree Programme in Energy Technology.

I want to say “Thank you” to all friends of mine who made my life in Lappeenranta bright and cheerful, especially to Daria, Sergey and Polina, Eugenia, Nikita, Andrei, Roman. I will never forget the time I have spent with you.

# TABLE OF CONTENT

**Latin letters**

| Symbol | Meaning | Unit |
|---|---|---|
| A | area | $m^2$ |
| C | discharge coefficient | – |
| $c_p$ | specific heat capacity | kJ/(kg·K) |
| d | diameter | m |
| g | local acceleration due to gravity | $m/s^2$ |
| h | elevation of a point above a reference plane | m |
| k | isentropic exponent | – |
| l | length | m |
| M | molar mass | kg/mol |
| m | mass | kg |
| p | pressure | Pa |
| $q_m$ | mass flow | kg/s |
| $q_v$ | volumetric flow | $m^3/s$ |
| R | specific gas constant | J/(kg·K) |
| $R_{mol}$ | universal gas constant | J/(K·mol) |
| T | temperature | K |
| V | volume | $m^3$ |
| v | velocity | m/s |
| Z | compressibility factor | – |

**Greek letters**

| Symbol | Meaning | Unit |
|---|---|---|
| $\eta$ | efficiency | – |
| $\pi$ | pressure ratio | – |
| $\rho$ | density | $kg/m^3$ |

**Subscripts**

| Index | Meaning |
|---|---|
| 1 | inlet |
| 3 | outlet |

| | |
|---|---|
| amb | ambient |
| d | dynamic |
| m | mass |
| s | isentropic |
| tot | total |
| v | volumetric |

# 1 INTRODUCTION

## 1.1 Objectives of the thesis

The objective of this Master's thesis work is to design a test stand for a centrifugal compressor. The compressor will be tested in order to study its overall performance and flow fields in the diffuser.

The test stand should consist of a centrifugal compressor driven by an electric motor, a close-looped piping with a reservoir large enough to reduce wave effects, flow straightener, control valves, a heat exchanger, and a number of pressure, temperature and mass flow sensors. Data from the sensors are read and converted from analogue format into digital format by a data acquisition system.

# 2 THEORETICAL BACKGROUND AND STANDARDS

Test stands for centrifugal compressors must meet the requirements that are specified in the International Standard ISO 5389: Turbocompressors – performance test code. This standard describes the requirements for measuring methods and measuring equipment; preparation and execution of the test, evaluation and measuring uncertainty of test results. (1)

## 2.1 Measuring methods and measuring equipment

Measuring methods for physical parameters (pressure, temperature, mass flow, humidity, etc.) and the equipment that should be used for this purpose are described in the standard (1). This equipment should be included into a test stand during the design stage. All the points where measurements take a place should be located at a straight pipe with an appropriate length. (1)

Acceptance tests should be performed with measuring instruments that have a test or calibration certificate issued by an accredited authority or with measuring instruments of a known accuracy. (1)

### 2.1.1 Pressure

There are two different approaches in pressure measurements: absolute pressure measurement and gauge pressure measurement. A gauge pressure is the difference between the atmospheric pressure and current absolute pressure. In some cases gauge pressure is more convenient way of pressure measurements. Absolute pressure might be calculated by summarizing of atmospheric and gauge pressures.

A pressure measurement system might be divided into the following typical elements: a pressure tap or a probe, a connection tube, and a device that treats the signals. (2)

The static pressure, $p$, near a wall is measured with a help of holes in the wall. These holes should be done with carefulness. Any burr or other surface damages are unacceptable. The holes should be done with the smallest possible diameter, but large enough to avoid a blockage. (1) Typical values for pressure taps in clean gases are the following (2):

$d \approx 0.3 – 0.6$ mm,

L ≈ 0.5 – 3 mm.

If a pipe is straight and long enough, the flow inside is parallel to the pipe axis and the static pressure in a perpendicular cross-section might be supposed to be constant. Then, sufficient measurement may be done with a sampling of pressure by means of a hole drilled in the pipe wall. (1)

If an average velocity, $v$, is obtained from flow measurements, an average dynamic pressure, $p_d$, may be calculated from this, static pressure $p$, average total pressure $p_{tot}$, (1). Average velocity is calculated according to Eq. 1:

$$v = -\frac{c_p \cdot p \cdot A}{q_m \cdot R \cdot Z} + \sqrt{{\frac{c_p \cdot p \cdot A}{q_m \cdot R \cdot Z}}^2 + 2 \cdot c_p \cdot T_{tot}} \qquad (1)$$

Where $v$ – velocity, m/s

$c_p$ – specific heat capacity, kJ/(kg·K)

$p$ – pressure, Pa

$A$ – cross-sectional area, $m^2$

$q_m$ – mass flow, kg/s

$R$ – specific gas constant, J/(kg·K)

$Z$ – compressibility factor

$T_{tot}$ – total temperature, K.

Total pressure consists of ambient, static and dynamic pressures, according to Eq. 2:

$$p_{tot} = p_{amb} + p + p_d \qquad (2)$$

Where: $p_{tot}$ – total pressure, Pa

$p_{amb}$ – ambient pressure, Pa

$p$ – static pressure, Pa

$p_d$ – dynamic pressure, Pa

The ratio of total to static pressure is defined according to Eq. 3 (1):

$$\frac{p_{tot}}{p} = \frac{p + p_d}{p} = \frac{T_{tot}}{T}^{\frac{k}{k-1}} \quad (3)$$

Where $T_{tot}$ - total temperature, K

$T$ – static temperature, K

$k$ – isentropic exponent.

According to the standard (1), the dynamic and total pressure can be calculated with these equations with sufficient accuracy.

The length of the connection tubes depends on the application. For pressure fluctuations measurements sensor should be placed as close to the flow as possible. Steady-state measurements allow longer pipes, but material selection should be considered carefully. (2)

Pressure transducers with electrical output detect the pressure by a sensing element. It might be a plate or tube with some surface area for the pressure to act upon. As a result of this action the surface deflects and this deflection is transformed into an electrical output signal. When the other side of the surface is under the influence of another pressure the transducer will measure differential pressure. If the pressure on the other side is ambient pressure then the transducer will measure gauge pressure. If there is a vacuum on the other side of the surface, absolute pressure is measured (Fig. 1). (3)

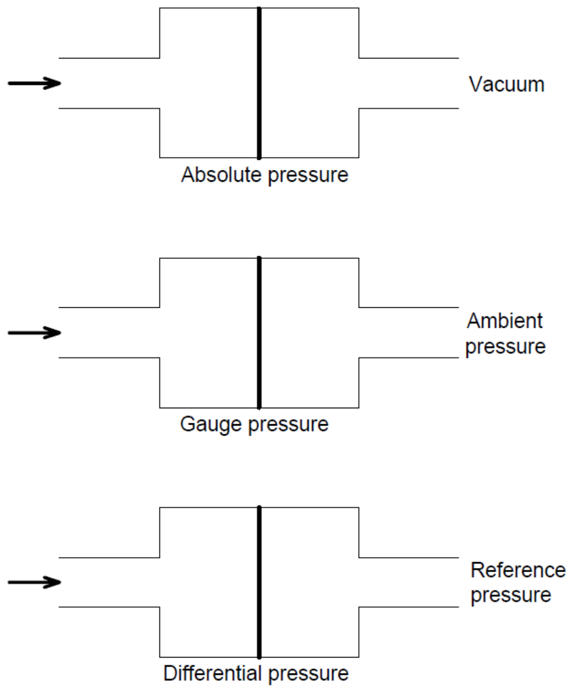



Figure 1. Pressure measurements types.

Inductive pressure transducers have a mechanical part that is moved by the measured pressure. This movement changes the inductance of a coil in the transducer. Single-coil transducers have a problem with temperature compensation. Because of that fact a more common type of inductive pressure transducers uses two coils. These transducers must be reactively and resistivity balanced at null, their volumetric displacement tends to be large. Magnetic objects and fields, as well as mechanical friction and consequent wear might cause errors. Advantages of this type of pressure transducers are possible high output, possibility to respond to both static and dynamic measurements, possible high signal/noise ratio. (3)

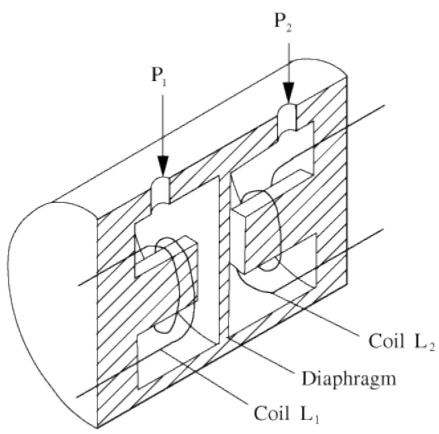



Figure 2. Inductive pressure transducer with two coils. (4)

Capacitive pressure transducers rely on capacitance change which is caused by a diaphragm movement under the pressure. These transducers have high frequency responses and small sizes. They allow the measurement of static and dynamic pressures and can be operated at high temperatures. Also these sensors are inexpensive and have a low shock response. Disadvantages of capacitive sensors are the high impedance output that must be balanced, movement of connecting cables that will cause signal distortion, sensitivity to temperature variations, complex receiving and conditioning circuitry. (3) The range of measured pressures is from high vacuums to 70 MPa. Capacitance pressure transducers can measure very low differential pressures (2-3 Pa). These features lead to widespread of this type of pressure sensors, especially as secondary standards in low-differential and low-absolute pressure measurements. (5)

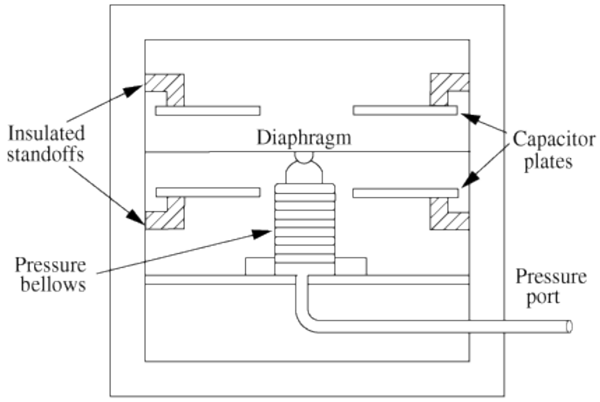



Figure 3. Capacitive pressure transducer. (4)

Potentiometric pressure transducers are based on a resistance element that is contacted by a slider. This slider is moved by a pressure change that leads to a change of the resistance. The contact between the slider and the resistive element is a weak point of this transducer. Despite that the potentiometer sensors are widely used. Its advantages are listed below (3):

- High output.
- Ac or dc excitation.
- Inexpensive.
- Signal conditioning is not necessary.
- Wide output functions range.

The disadvantages of potentiometric transducers are the following ones (3):

- High mechanical friction.

- Finite resolution.
- High noise levels.
- Sensitive to vibration.
- The accuracy depends upon the force required because of friction.
- Low frequency response due to mechanical contact.
- Large size.

These factors permit them to be used in low power applications. They can measure pressures in the range from 35 kPa to 70 MPa. (5)

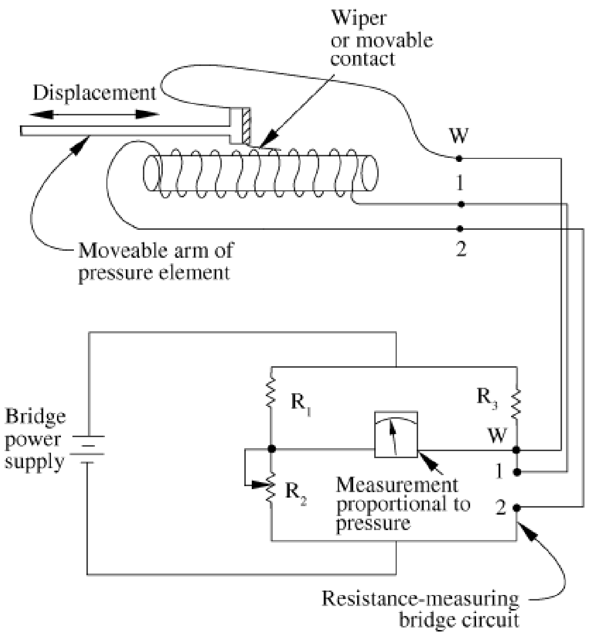



Figure 4. Potentiometric pressure transducer. (4)

Strain gauge pressure transducers utilize the effect of a resistance change due to mechanical strain that is caused by a pressure change. A diaphragm or a tube is used as the pressure sensing element. Some designs also use a beam or armature as a secondary sensing element. Strain gauges might be made from different materials including metal and metal alloys, semiconductor materials, and thin-film materials. A silicon diaphragm has suitable mechanical properties since it is elastic and does not have hysteresis effects. The mass of the silicon diaphragm is very low, that gives a fast response and low sensitivity to accelerations caused by shock and vibrations. (3) Strain gauge pressure transducers are available for pressure ranges from 0.7 kPa to 1400 MPa. (5)

Piezoelectric pressure transducers have a quartz crystal inside. The crystal is mechanically stressed by a diaphragm. Piezoelectric sensors might be divided into the next groups, according to measured parameter: electrostatic, piezoresistive, or resonant. Pressure applied to a crystal causes its elastic deformation. This deformation results in an electric charge that can be measured. These pressure transducers are not able to measure static pressures. Electrostatic pressure transducers are compact and rugged. Since quartz is used as a generating crystal, these sensors are usually inexpensive. They provide high speed responses (up to 100 kHz) and can detect pressures from 0.7 kPa to 70 MPa. Electrostatic pressure transducers are widely used for rapid changing dynamic pressure measurements in rocket engines, motors, engines, compressors. (5)

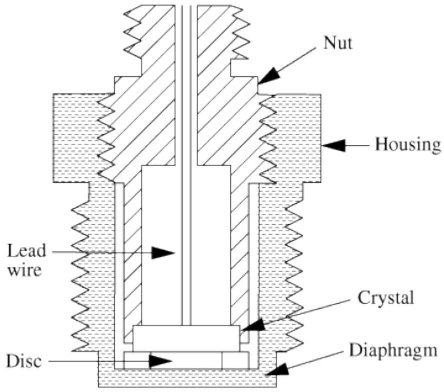



Figure 5. Piezoelectric pressure sensor. (4)

Piezoresistive pressure transducers are similar to a strain gauge, but here silicon resistors are bonded onto a diaphragm. The diaphragm itself is made of silicon and during the manufacturing process the resistors are diffused into the silicon. These sensors are sensitive to temperature changes and can detect pressures from 21 kPa to 100 MPa. (5)

Resonant piezoelectric pressure transducers are based on measurement of the variation in resonant frequency of quartz crystal under the pressure. These sensors can measure absolute (0-100 kPa to 0-60 MPa) or differential pressures (0-40 kPa to 0-275 kPa). (5)

### 2.1.2 Temperature

There are different types of temperature measuring devices that are based on various physical principles and properties.

The relation between total and static temperatures is given in Eq. 4:

$$T_{tot} = T + \frac{v^2}{2c_p} \tag{4}$$

Where $T_{tot}$ – total temperature, K

$T$ – static temperature, K

$v$ – velocity, m/s

$c_p$ – average specific heat capacity, kJ/(kg·K).

Conventional temperature sensors have specific feature to gravitate to their characteristic temperature when they are exposed to a flow, even if they are correctly installed.

Thermocouple is a device that consists of two wires that are made of different metals. These wires are joined or welded together at one end. When the temperature difference exists between the sensing and reference junctions of a thermocouple it causes a change in electromotive force between the other ends. This change is measured and then converted into a temperature, since the voltage is known for given metals and temperature (6). The overall relation between the voltage and temperatures is called the Seebeck effect (7). Basic thermocouple circuit is shown in the Fig. 6.

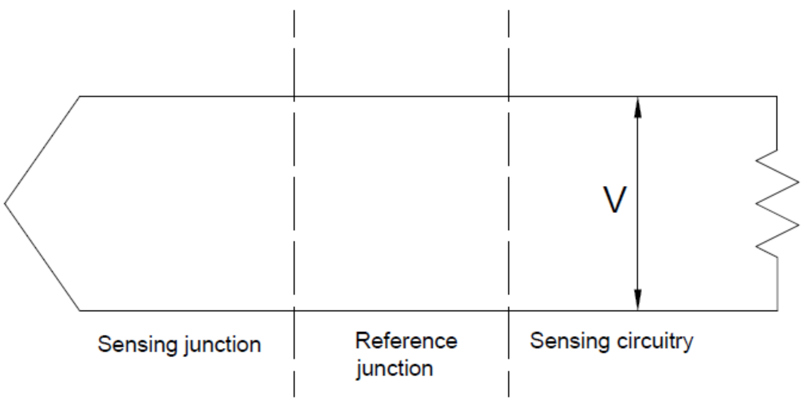



Figure 6. Basic thermocouple circuit. (3)

Resistive temperature meters are based on the feature of materials to change their electrical resistance when its temperature changes. Resistance temperature detectors (RTDs) are made of metals and their resistance change is linear (6). Copper is one of the most linear metals and an inexpensive material for RTD, but its low resistivity and tendency to oxidize at moderate temperatures are disadvantages of this metal. Low cost and high temperature coefficient of nickel lead to its wide usage as an RTD element in the temperature range from -100 to 300 °C. But its R/T relation is neither well known nor as reproducible as that of platinum, which provides high accuracy in the temperature range from -260 to 1500 °C. Platinum is chemically inert resistant to contamination and oxidization, but it is more expensive than other resistance temperature sensors (3).

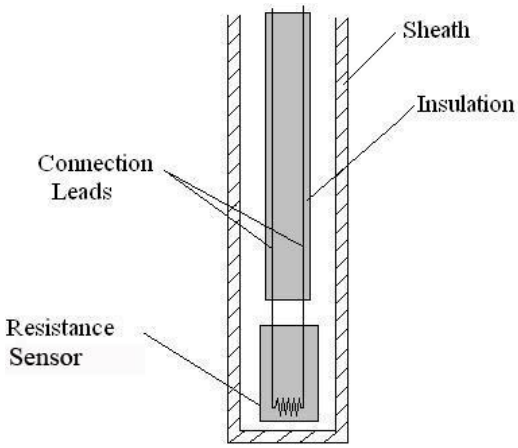



Figure 7. Resistance temperature detector. (8)

Thermistors rely on resistance change in a semiconductor which is nonlinear and varies rapidly with temperature (6). The materials that are used in thermistors are sintered mixtures of sulphides, selenides, and metal oxides. Typically these sensors have high resistivity and high negative temperature coefficient. Thermistors are relatively inexpensive and small, but they have a narrow temperature range, so there is a need to use several sensors to cover a wide temperature span (3).

Infrared temperature measurement devices measure thermal radiation emitted by a material. It is a noncontacting device (6).

Bimetallic temperature meters are based on the different rate of thermal expansion of various metals. Two metal plates are joined together. When a temperature increases one side will expand more. It leads to bending that is converted into a temperature. This device does not require a power supply, but its accuracy is lower in comparison with thermocouples or resistance temperature devices (6).

### 2.1.3 Flow measurement

Mass flow measurements with nozzles and orifices are defined by the standard ISO 5167-1 (9). Volume flow measurements using calibrated gas meters are also permitted by the standard (1) if the gas flows smoothly thru the meter, without any pulsations or disruption.

Turbine flow meter has a turbine wheel that is turned by the moving fluid. Rotational speed of the rotor is proportional to the flow rate. There is a coil inside the transducers. Each turbine blade induces a voltage inside the coil during every rotation. Output signal of the meter is an AC voltage. Therefore, the frequency of this voltage is proportional to the flow rate. Turbine flow meters are widely used in many industries due to their performance characteristics and wide operational range. Measured flow could be from 20 l/h to 600 l/s with pressures up to 500 bars in the temperature range from 33 K to 500 °C (3).

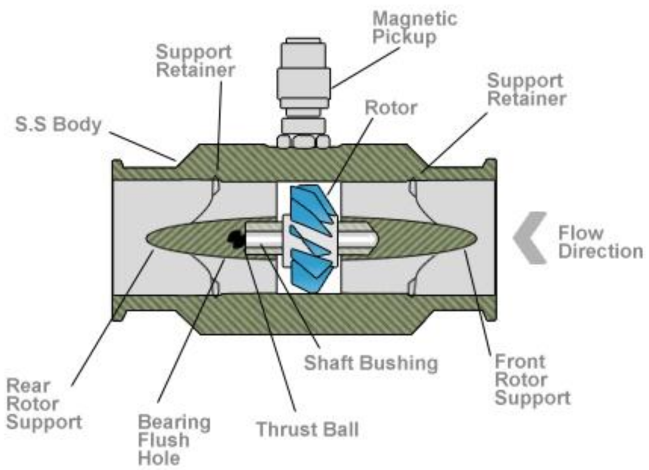



Figure 8. Turbine flow meter. (10)

The weakest point of turbine meters is the bearing that has to support the rotor. Size of the hub, the clearance between blade tip and pipe wall, the shape, size and spacing of the blades are also parameters that affect the performance of turbine flow meter. Since hydrodynamic theory cannot accurately predict the performance of the meter, individual calibration of each turbine meter is necessary. Advantages of this type of flow meters are listed below (11):

- Excellent short-term repeatability and accuracy of large flow meters if they are periodically recalibrated.
- Digital output signal is practically linear in wide range of flow rates.
- Low level of head loss

- Compact design
- They do not block the flow in a case of failure.

Disadvantages of turbine flow meters (11):

- They are more expensive in comparison with other types of flow meters.
- A turbine meter requires periodical recalibration due to wear and fouling of surfaces.
- Flow disturbances might affect the flow meter performance.

Variable-area meter, or rotameter, uses a float that is located in a tapered section of tubing to measure the stream flowing through. The float responses to the flow rate, its density and viscosity. The position of the float may be monitored optically in glass tubing and magnetically in opaque tubes. Accuracy of this sensor is 1% for flows up to 300 l/s. Allowable temperatures are up to 500 °C and pressures up to 170 bars. Rotameters create low pressure drops, but require vertical installation (3).

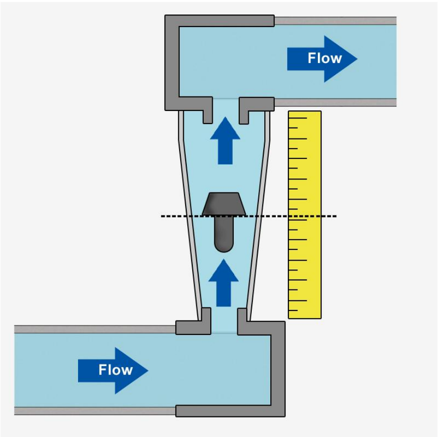



Figure 9. Rotameter. (12)

Main element of hot-wire anemometers is an electrically heated element that is placed in air or gas flow. The higher the velocity of the stream the more it tends to cool the wire. There are two different types of these transducers: the constant-current type and constant-temperature type. In the first case a fixed current flow thru the wire and the resistance change is measured. The second type of hot-wire anemometers uses a feedback system to keep the constant temperature and resistance of the heated element by adjustment of current that flows through the wire. The applied voltage is measured in this case. This system has a better response to rapid changes in air flow, but the constant-current system is simpler and cheaper (11).

Laser anemometer is a non-intrusive measuring device, so it does not disturb the flow. This device directly measures the absolute velocity of the flow, independently of other thermodynamic characteristics of the flow. There are two most used laser anemometer types: Laser Doppler anemometer (LDA) and Laser two focus (L2F).

Laser Doppler anemometer relies on the Doppler principle: when light is reflected on a moving particle its frequency changes. This change depends on the particle velocity and direction and therefore determines (with certain precautions) the flow velocity. Hence, since the LDA measures the velocity of the particles which is not the same for all the particles, the result of LDA measurement is a histogram of the velocity. Then, the flow velocity might be determined as the point with the highest probability (2).

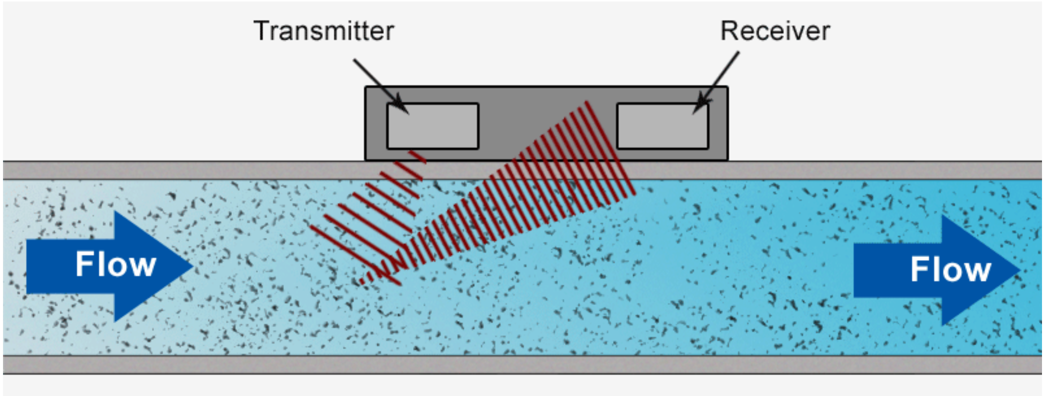



Figure 10. Doppler anemometer. (12)

The L2F device also detects the light that is reflected from the flow particles. There are two focal points that are formed by two laser beams. The distance between these points is known, so, by measurement of the time of particle flight it is possible to calculate its velocity. However, these two measurement points should be rotated to find the average direction of the particles traverse. Thus, the result of L2F measurement is a three-dimensional histogram (axis are measurement direction, number of pulses and the time between the pulses) and the flow velocity is the maximum point (2).

According to researchers' estimations, L2F system is better for measurements of large velocities and low turbulence intensity flows in small nozzles. LDA is a preferable choice for low velocity flows and high turbulence intensity (2).

Flow measurements using nozzles, orifices, Venturi meters, Pitot tubes, or, in other words differential pressure flow meters are pressure-based measurements. They rely on Bernoulli's equation (13):

$$p_1 + \frac{1}{2}\rho v_1^2 + \rho g h_1 = p_2 + \frac{1}{2}\rho v_2^2 + \rho g h_2 \tag{5}$$

Where: $p_{1,2}$ – fluid pressure at the points 1 and 2, respectively, Pa

$\rho$ - density of the fluid, kg/m$^3$

$v_{1,2}$ – fluid velocity at the points 1 and 2, respectively, m/s

$g$ – local acceleration due to gravity, m/s$^2$

$h_{1,2}$ - elevation of the points 1 and 2 above a reference plane, m.

According to this equation, increasing of fluid velocity leads to lowering of fluid pressure if there are no leakages. In other words, kinetic energy of fluid flow increases due to pressure drop by a corresponding amount, in accordance with the energy conservation law (11). Pressure difference between the point before a restriction and in its narrowest point is measured by differential pressure transmitters. Then main flow parameters might be calculated with the Eq. 6-8 (14):

$$v = C \ \frac{dp}{\rho}^{\ 0.5} \tag{6}$$

$$q_v = C * A \ \frac{dp}{\rho}^{\ 0.5} \tag{7}$$

$$q_m = C * A \ dp * \rho^{\ 0.5} \tag{8}$$

Where: $v$ – fluid velocity, m/s

$C$ - discharge coefficient of the element

$dp$ - pressure difference, Pa

$\rho$ - fluid density, kg/m$^3$

$q_v$ – fluid volumetric flow, m$^3$/s

$A$ - cross-sectional area of the pipe's opening, m$^2$

$q_m$ – fluid mass flow, kg/s.

Discharge coefficient, according to its definition, is the ratio between true and theoretical flow rates (15):

$$C = \frac{true\ flow\ rate}{theoretical\ flow\ rate}$$

The discharge coefficient might be presented in four different ways that are described in (15). It corrects the theoretical equation for the influence of velocity profile (Reynolds number) and geometry of the meter (15).

Summarizing all above mentioned facts, pros and cons of differential pressure mass flow meters could be defined. Advantages of these meters are listed below (11):

- Simplicity of construction: the lack of moving or rotating parts inside the pipe and placement of all measuring sensors outside of the pipeline lead to easy service and reliability.
- Versatility: these meters can be used with various types of fluids.
- Experience: differential pressure meters were used for a long time. This fact leads to abundance of information and experience in usage of these devices in many various ways.

As any other measuring device, differential pressure flow meters have its disadvantages (11):
Accuracy is not high when the meter is not properly calibrated (for instance, for the sake of cheapness). This type of measuring device should be calibrated for known flow conditions.
Differential pressure is not linear with flow rate. It leads to lower range in comparison with linear output devices.

## 2.2 The old test stand at LUT

There is an old test stand for centrifugal compressors in Laboratory of Fluid Dynamics at Lappeenranta University of Technology. It was designed for studying flow field and efficiency of vaned and vaneless diffusers and the effect of tip clearance in centrifugal compressors. Different high-speed, variable speed compressors were used with a number of various diffusers in research activities. Some of the results are presented in papers (16), (17), (18), (19), (20).

The test rig layout is presented in Fig. 11. The compressor is driven by high-speed electric motor and equipped with magnetic bearings. The test rig is open-looped: the compressor intakes air from and discharges into the atmosphere through a silencer (18). There are mass flow nozzle, flow straightener and throttling valve before the compressor stage. The mass flow nozzle is an ISA 1932 and it is made according to DIN 1952 standard (20). Measurements of temperature and pressure before and after the compressor were used in the overall performance measurements at different rotational speeds (20).

The flow fields were measured with three Kiel probes and a three-hole cobra probe. The placement of these probes and the position of static pressure measurements are presented in Fig. 12 (20).

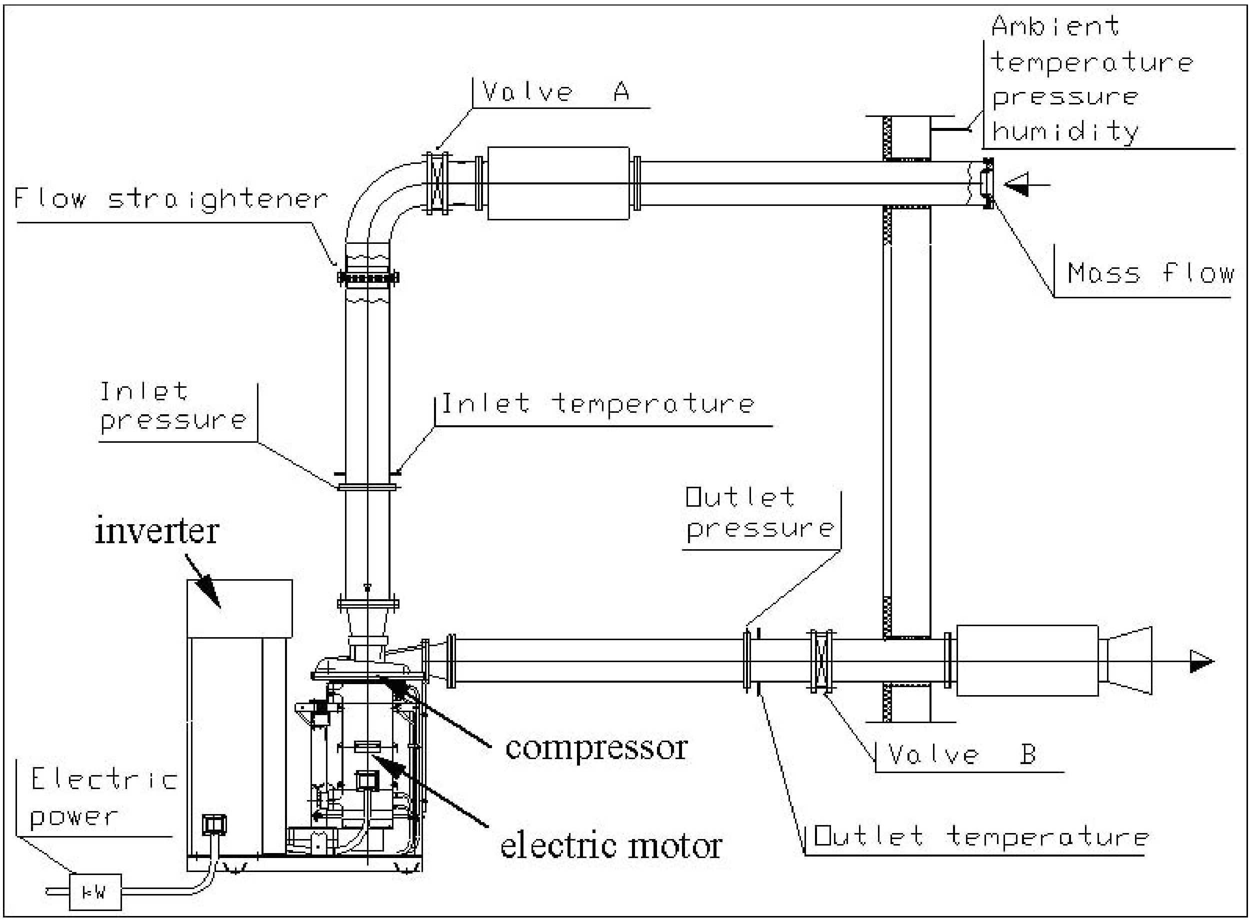



Figure 11. Previous test rig layout (20).

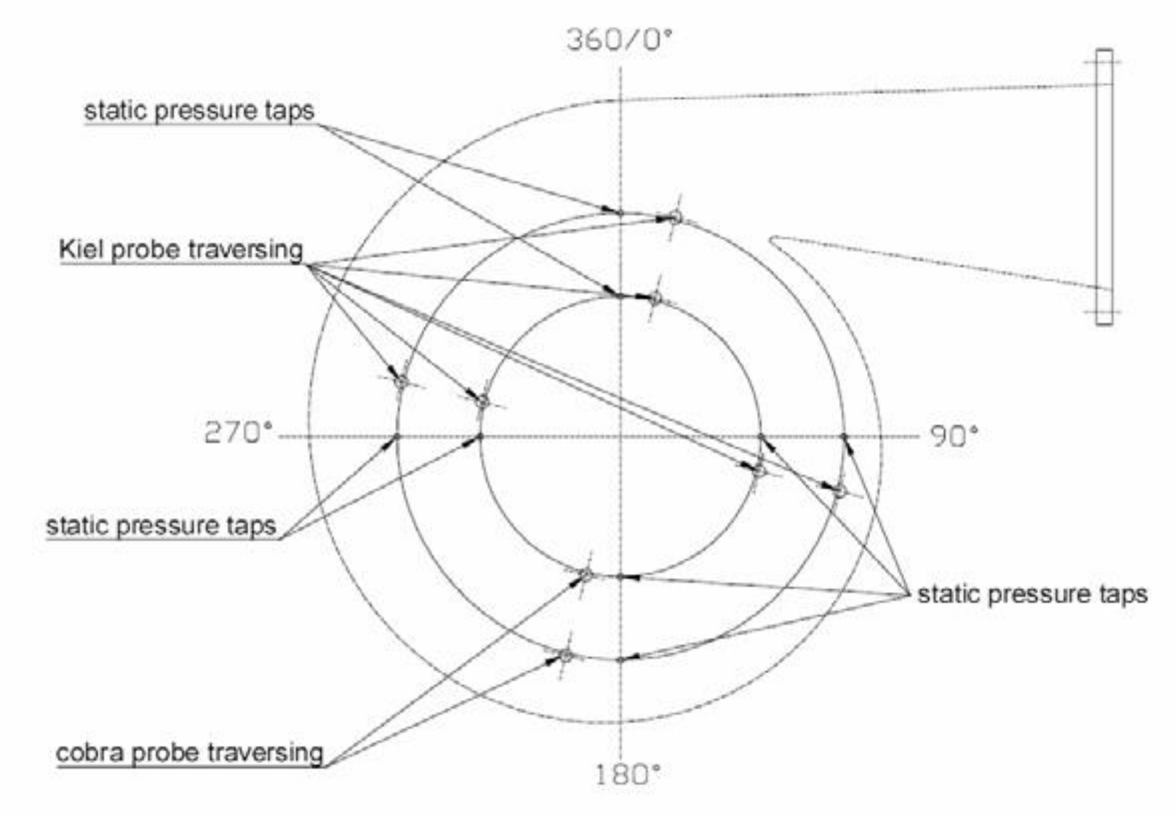



Figure 12. Probes and static pressure tap positions (20).

## 2.3 Test stands in other laboratories

A number of universities and research institutes have various test stands for centrifugal compressors. They are built for different tasks and purposes. Short description and main features of these test rigs are presented in the brief overview below.

### 2.3.1 RWTH Aachen University

Institute of Jet Propulsion and Turbomachinery in RWTH Aachen University has several test stands for different types of compressors. The first one is called RADIVER. The compressor stage consists of semi-open impeller with backward curved blades and wedge vane diffuser with variable geometry (22).

Figure 13. The impeller and diffuser of RADIVER compressor (22)

Total pressure ratio of the compressor stage is 4.07, mass flow rate is 2.47 kg/s. Rotational speed of the impeller is 35200 $min^{-1}$ and circumferential velocity at the impeller outlet is 498 m/s. The impeller has 15 backward curved blades. Its outer diameter is 270 mm. Number of diffuser vanes is 23. The compressor is driven by a 500 kW DC motor Inlet temperature and pressure may be varied (22).

This test stand is used for studying next areas (22):

- Guide vane for control of compressor
- Early detection of pumping and instability
- Flow processes at the choke boundary
- Losses in individual stages and components
- Impeller – diffuser interaction
- Influence of diffuser flow non-stationarity on the diffuser boundary layers
- Steady and unsteady flow field in the impeller outlet and diffuser area

Measuring devices of the test rig (22):

Stationary:

- Holes in the walls in housing over the impeller and diffuser for pressure measurement
- Holes in the diffuser vanes for pressure profile measurement
- Pitot tubes in the impeller outlet
- Three-hole cobra probe at the exit of diffuser blades

- Three-hole cylinder probe in the cylinder outlet
- High-resolution spatial temperature probe in the diffuser outlet

Transient:

- Semiconductor pressure sensor in the housing of the impeller inlet and diffuser
- Semiconductor pressure sensor in the diffuser for pressure profile measuring
- Laser-2-focus anemometry in the impeller and diffuser outlet area

Schematic drawing of the test rig is presented in the Fig. 14. The positions in the illustration are the followings: 1 – centrifugal compressor stage, 2 – inlet section with honeycomb, 3 – settling chamber with filters, 4 – DC motor, 5 – planetary gearbox, 6 – pressure-side throttle valve, 7 – engine valve, 8 – heat exchanger, 9 – valve, 10 – mass flow nozzle (DIN 1952), 11 – suction throttle valve.

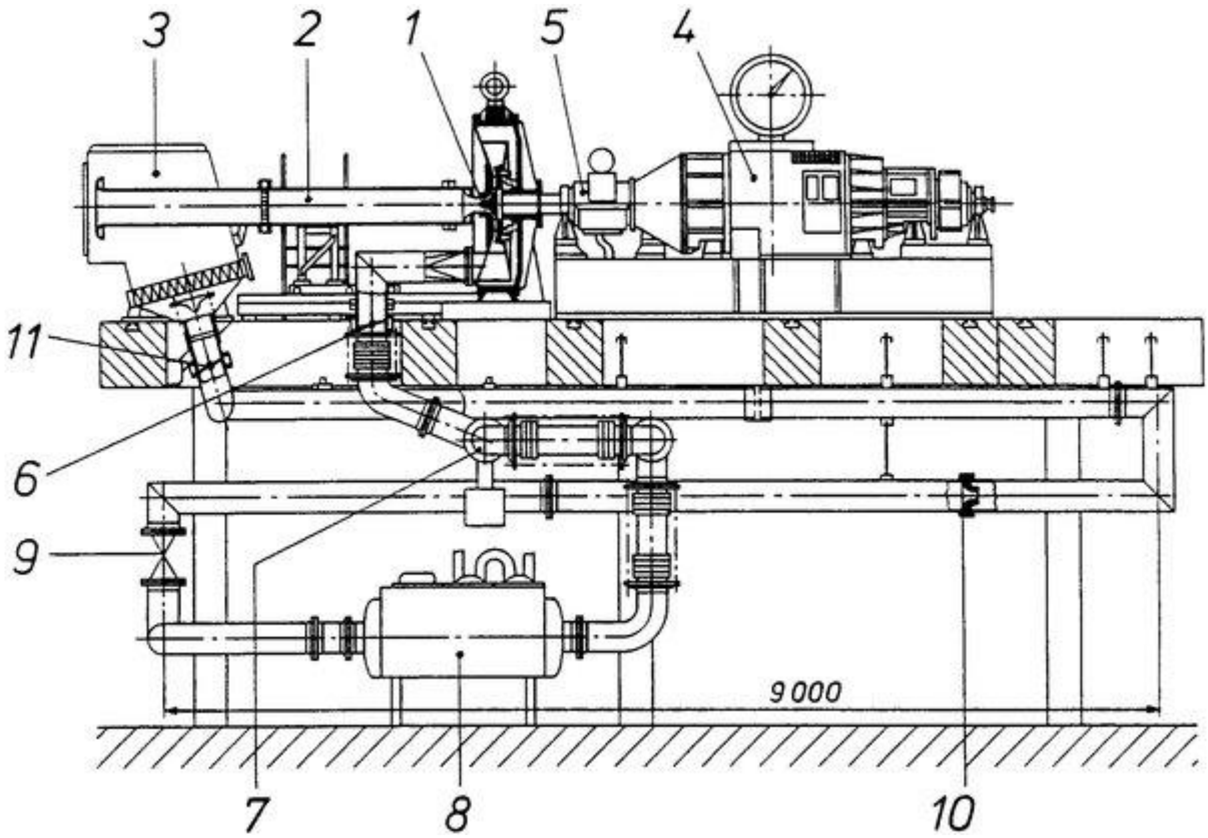



Figure 14. Schematic drawing of the test rig RADIVER (22)

The next test stand in the Institute of Jet Propulsion and Turbomachinery in RWTH Aachen University is called transonic centrifugal compressor. It consists of highly-loaded single-stage compressor with total pressure ratio 6. Rotational speed is 31400 min$^{-1}$. The outer diameter of the impeller is 356 mm. It has 13 main and 13 pitch blades. Asynchronous electric motor is used in this test rig (23).

Next areas are studied in this test facility (23):

- Transonic flow conditions in highly loaded centrifugal compressors
- Acoustics of centrifugal compressors
- Aerodynamics
- Stability and structural studies

For these purposes the next measuring devices are used (23):

- Pressure  and temperature sensors in inlet and outlet plenum
- Magnetic flow measurement in different stages
- Torque sensors in the drive train for torque and rotational speed measurements

3D models of the test rig and the compressor's impeller are presented in the Fig. 15.

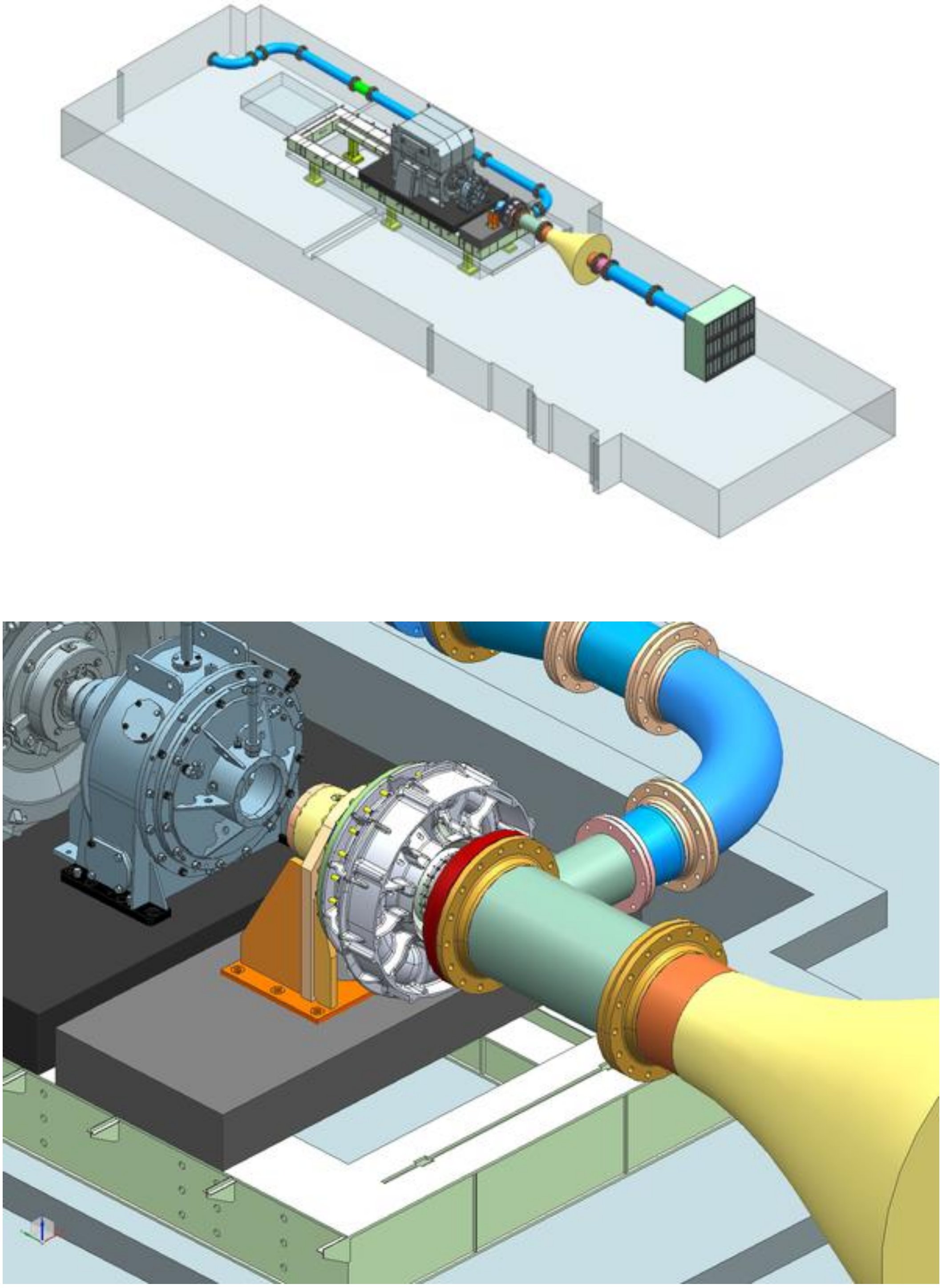


Figure 15. Impeller and 3D models of the test stand (23)

The third test stand in the Institute of Jet Propulsion and Turbomachinery in RWTH Aachen University is used for testing aero-engine centrifugal compressor. Diffusers with new geometry are studied as well as influence of other parameters is considered. Total pressure ratio of the compressor stage is 3 and the rotational speed is 19400 $min^{-1}$. Inlet section of the compressor has adjustable guide vane. The diameter of the impeller with 23 backward curved blades is 400 mm and it is driven by asynchronous electric motor. Closed air circuit is used in this test stand.

Compressor inlet pressure might be adjusted within the range 0.4 – 3 bar. Inlet temperature is varied by heat exchanger. Different areas are studied including the next topics (24):

- Effectiveness of new geometries
- Detailed studies of the flow inside pipes and outlet guide vane diffuser
- Variations of Reynolds number
- Variations of rotor gap and rotor positions
- Variations of bleed air

Flow and engine conditions are measured with the following devices (24):
Stationary measurements:

- Holes in the walls in housing over the impeller and diffuser for pressure measurement
- Five-hole probe in the compressor inlet
- Pitot tubes in the area between impeller and diffuser
- Three-hole cobra probe at the exit of diffuser blades
- Pressure and temperature measurements in the settling tank
- 360° adjustable probe rake in the compressor outlet for pressure and temperature measurement
- Torque meter for measurements of torque and rotational speed

Transient measurements:

- Semiconductor pressure sensor in the housing of the impeller inlet and diffuser
- Hot-wire anemometry in the compressor inlet
- Rotor clearance measurements by capacitive sensors
- Particle Image Velocimetry in the diffuser

Sectional image of the compressor stage is presented in the Fig. 16, where 1 – torque meter, 2 – magnetic bearing, 3 – exit plenum, 4 – deswirler blades, 5 – pipe diffuser, 6 – impeller, 7 – adjustable inlet guide vanes, 8 – spinner, 9 – inlet section (24).

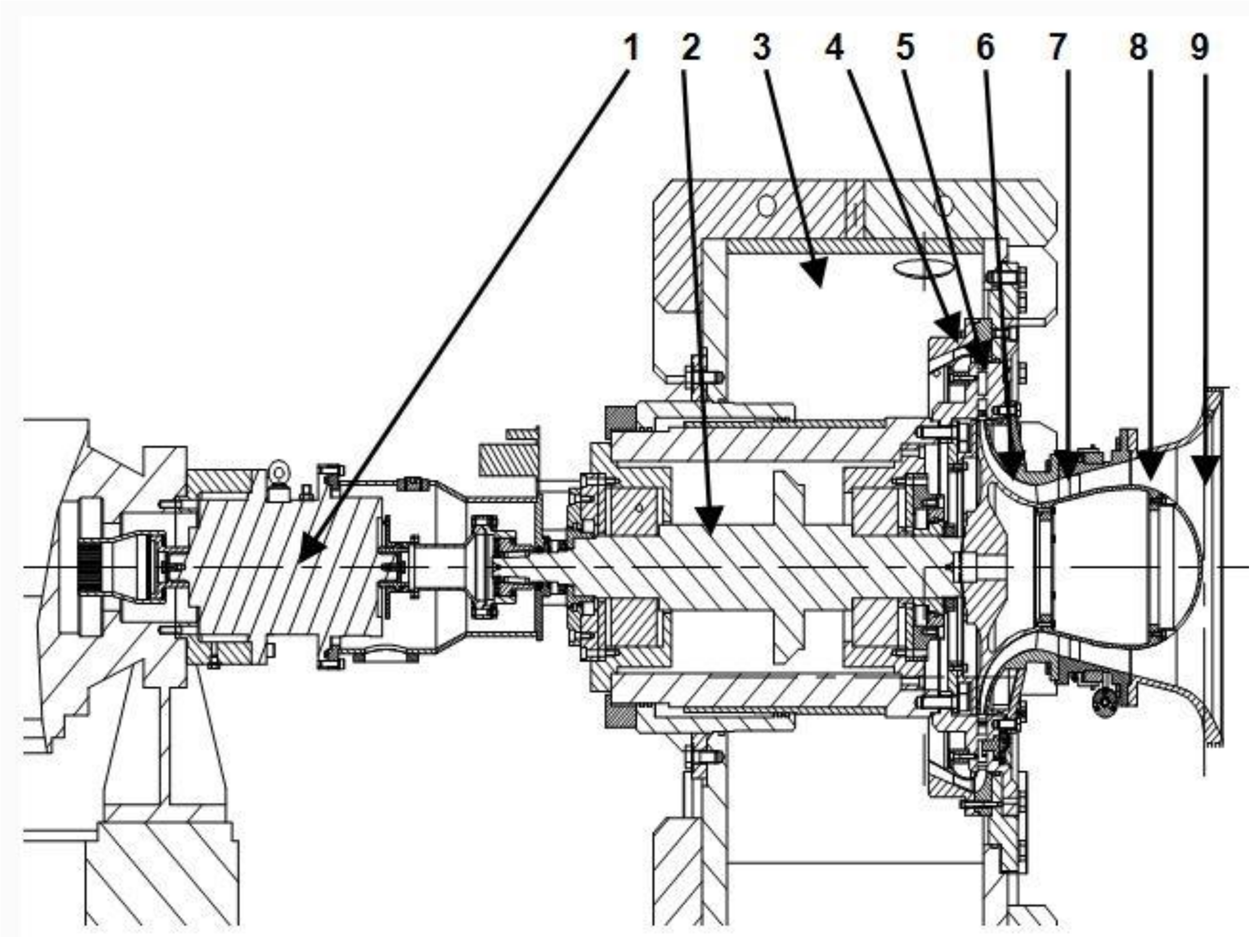



Figure 16. Sectional image of the compressor stage (24)

Overall image of the test rig is shown in the Fig. 17. The positions in the image are the next: 1 – electric motor, 2 – transmission, 3 – centrifugal compressor stage and discharge plenum, 4 – inlet section, 5 – surge tank, 6 – throttle, 7 – intercooler, 8 – mass flow aperture (24).

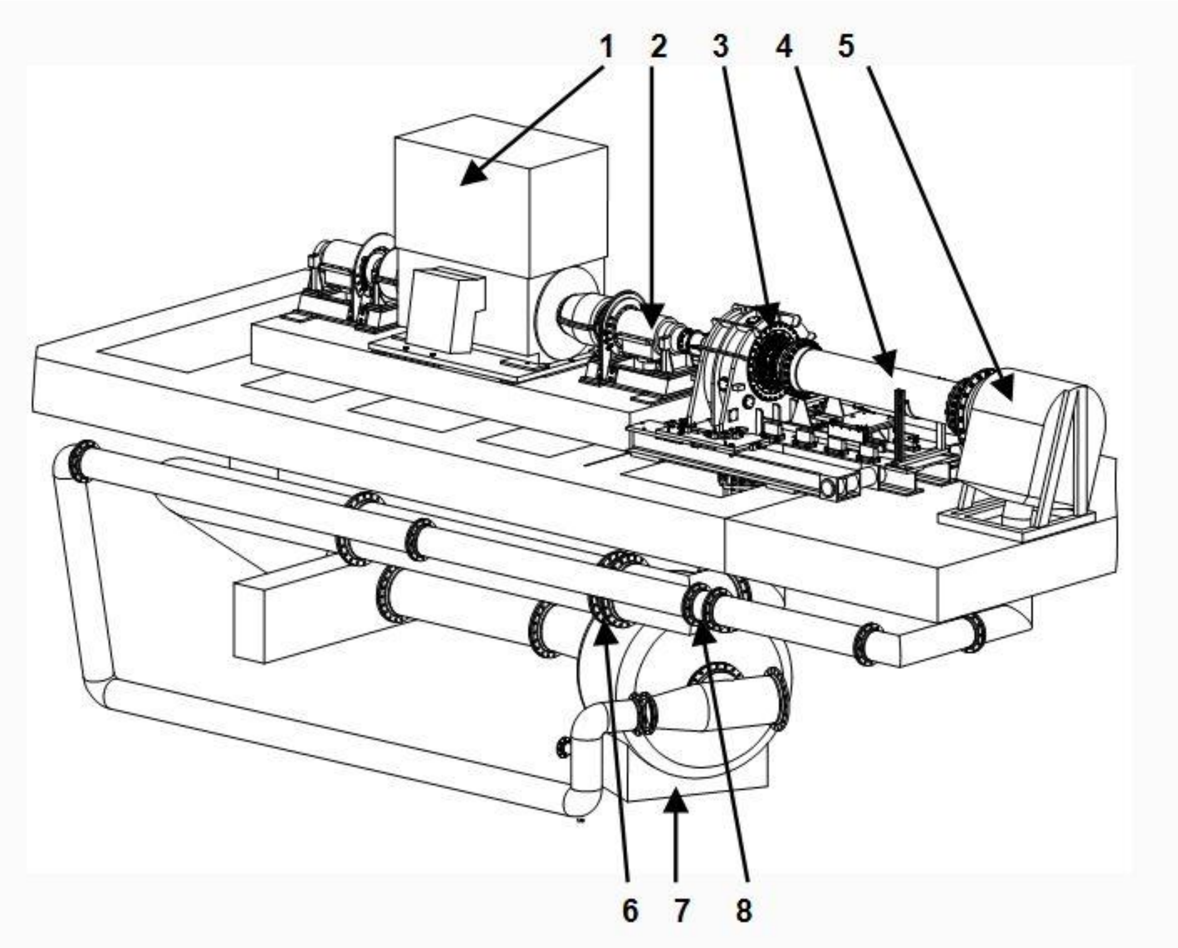



Figure 17. Overall view of the test stand for engine centrifugal compressor (24)

### 2.3.2 Swiss Federal Institute of Technology, Zurich

Laboratory for Energy Conversion in Swiss Federal Institute of Technology Zurich performs an experimental work with centrifugal compressors. During previous research activities the focus of the work was a measurement of aerodynamic quantities. Optical Laser Doppler Anemometry system and Fast Response Aerodynamic Probes were used to study the flow structure within the compressor. Acquired data were used in design of the diffuser and impeller. Further studies were aimed at resolving flow structure within the tip gap while the mass flow rate was varied. Fast pressure measurement was used for this purpose (25).

Current research work is intended for studying forced response or flutter of impeller blades. A lot of experimental researches have been performed in this field. As a result, two sources of forced response excitation can be recognized: upstream excitation from flow distortion and downstream excitation from diffuser potential flow field. During these experiments the test rig was modified and signal transmission from the rotating impeller (fast measurements of strain and pressure on the blade surfaces and impeller disk temperature) was enabled.

Interchangeable distortion screens are used for measuring of dynamic response characteristics of impeller blades when resonance occurs. Strain and pressure on the impeller disk are obtained and then with known inlet pressure and mass flow damping quantities can be derived (25).

### 2.3.3 German Aerospace Centre DLR

Institute of Propulsion Technology in German Aerospace Centre carries out researches with highly loaded radial impellers. Their studies showed that unfavourable flow conditions resulting from flow separation and non-uniform and unsteady flow leads to a very high flow angle difference in the vane diffuser (26).

A new impeller has been developed to achieve improved diffuser flow. Conventional measurement methods as well as newly developed “3-component Doppler Laser-2-focus” measurement technology for the impeller and “PIV” measurement technology for the diffuser were used during the experiments. These tests were accompanied by steady and unsteady 3D calculations with TRACE method developed in the Institute that helped to bring a physical understanding of the flow processes in highly loaded centrifugal compressors (26). The test rig is shown in the Fig. 18. Position number 1 is pressure and temperature probes, 2 – Charge-coupled device camera, 3 – particles probe, 4 – outlet housing, 5 – laser window, 6 – laser swivel arm, 7 – transmission, 8 – coupling, 9 – motor.

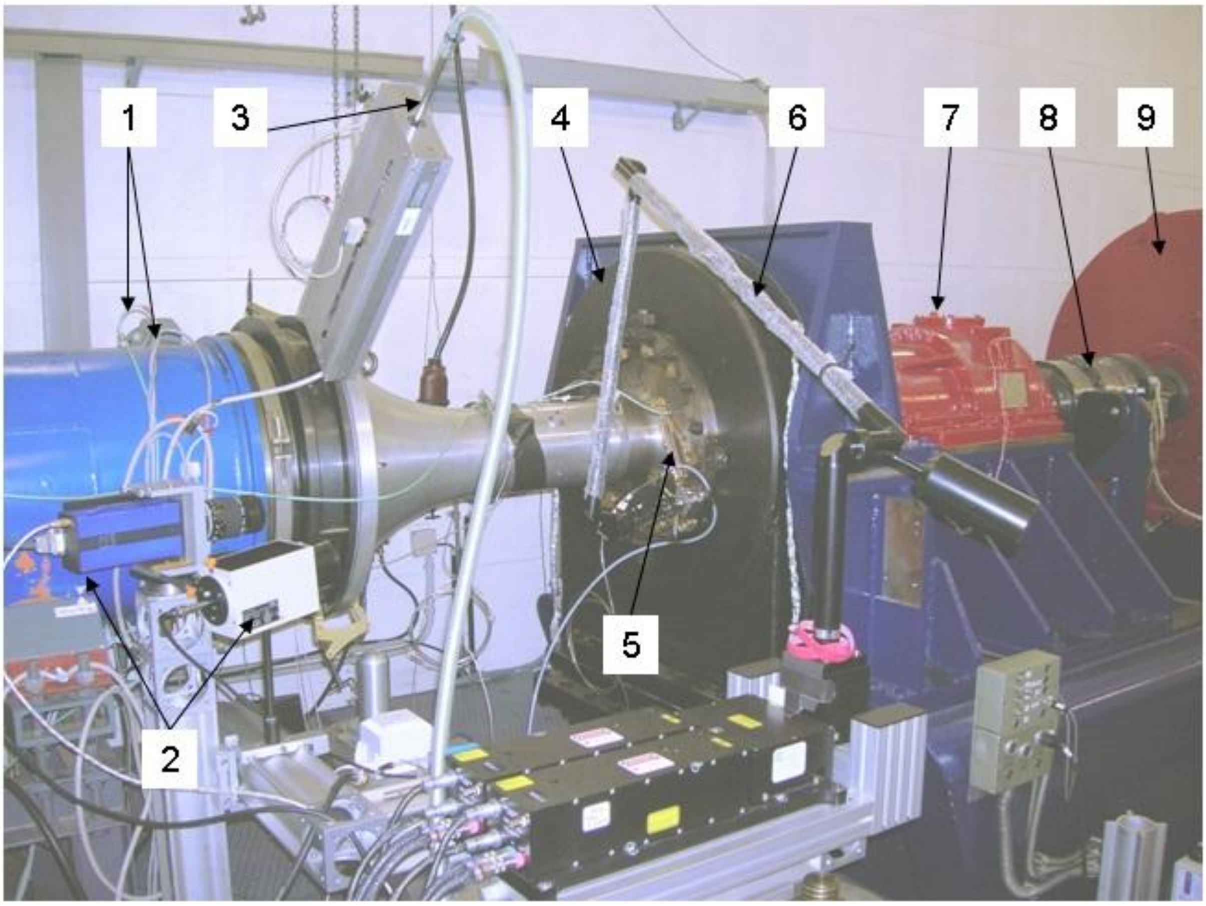



Figure 18. Test stand for centrifugal compressors in Institute of Propulsion Technology (26)

# 3 DESCRIPTION OF THE NEW TEST STAND

## 3.1 Centrifugal Compressor

The compressor that will be tested in the designed test stand is a centrifugal type. Nominal pressure ratio of the compressor is 2.5; the rotational speed is 27660 $min^{-1}$. Nominal mass flow is 1.8 kg/s and gas power is 190 kW. The impeller has 18 blades and its outer diameter is 270 mm.

Air flow conditions in different compressor's stages are presented in the Fig. 19.

| Results | Mass flow | Speed | Pressure ratio (p5/pt0) | Gas power | Number of blades | Specific speed |
|---|---|---|---|---|---|---|
| | 1.800 kg/s | 461 rps | 2.50 | 190.67 kW | 18 | 0.700 |

| | STATE | p kPa | T K | h kJ/kg | v m3/kg | cn m/s | d1t mm | d1h mm |
|---|---|---|---|---|---|---|---|---|
| Compressor inlet | 00 | 100.30 | 288.0 | 14.89 | 0.82 | | | |
| Impeller inlet | 1 | 91.22 | 280.6 | 7.42 | 0.88 | 122.2 | 134.9 | 40.5 |
| | 01 | 99.96 | 288.0 | 14.89 | | | | |

| | STATE | p kPa | T K | h kJ/kg | v m3/kg | c m/s | d mm | b mm | η | Cpr | p/pt0 | N2 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Impeller outlet | 2 | 192.05 | 353.4 | 80.75 | 0.53 | 283.1 | 270.9 | 12.16 | 0.557 | | 1.91 | 18 |
| | 02 | 279.27 | 392.8 | 120.82 | | | | | 0.928 | | 2.78 | |
| Diffuser outlet | 3 | 241.07 | 382.9 | 110.76 | 0.46 | 141.8 | 541.9 | 10.33 | 0.777 | 0.562 | 2.40 | |
| | 03 | 263.85 | 392.8 | 120.82 | | | | | 0.869 | | 2.63 | |
| Volute outlet | 4 | 243.85 | 386.7 | 114.55 | 0.46 | 111.9 | 96.5 | | 0.789 | 0.122 | 2.43 | |
| | 04 | 257.62 | 392.8 | 120.82 | | | | | 0.844 | | 2.57 | |
| Compressor outlet | 5 | 250.74 | 392.5 | 120.43 | 0.45 | 27.6 | 193.1 | | 0.817 | 0.500 | 2.50 | |
| | 05 | 251.58 | 392.8 | 120.82 | | | | | 0.820 | | 2.51 | |

| | Triangle | c | cu | cr | w | u | α | β | γ | ι | Mach |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Detail | 1t | 122.2 | 0.0 | 122.2 | 230.6 | 195.6 | 0.0 | 58.0 | 54.0 | 4.0 | 0.69 |
| | 2 | 283.1 | 264.0 | 102.1 | 164.7 | 392.8 | 68.9 | 51.7 | 40.0 | -11.7 | 0.75 |
| | 3 | 141.8 | 132.0 | 51.9 | | | 68.6 | | | | |

Figure 19. Air flow conditions inside the compressor.

Fig. 20 and 21 show the compressor flow maps, where design point (mass flow 1.8 kg/s, pressure ratio 2.5) is marked as a red cross.

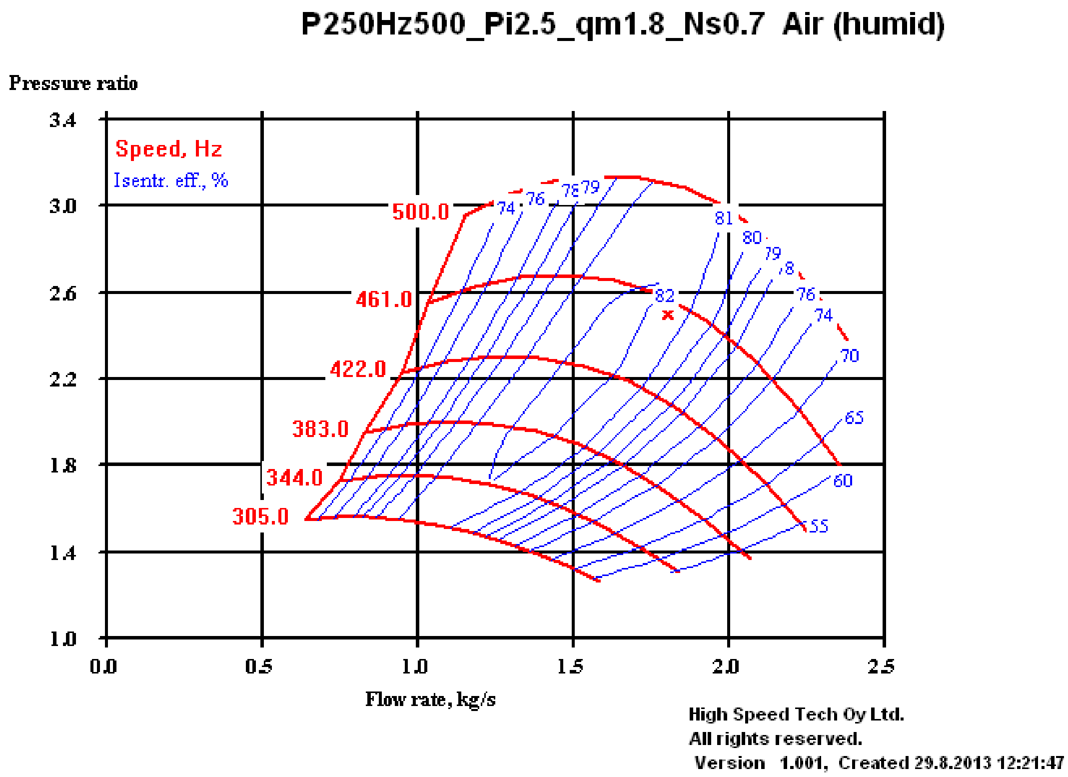



Figure 20. Compressor flow map.

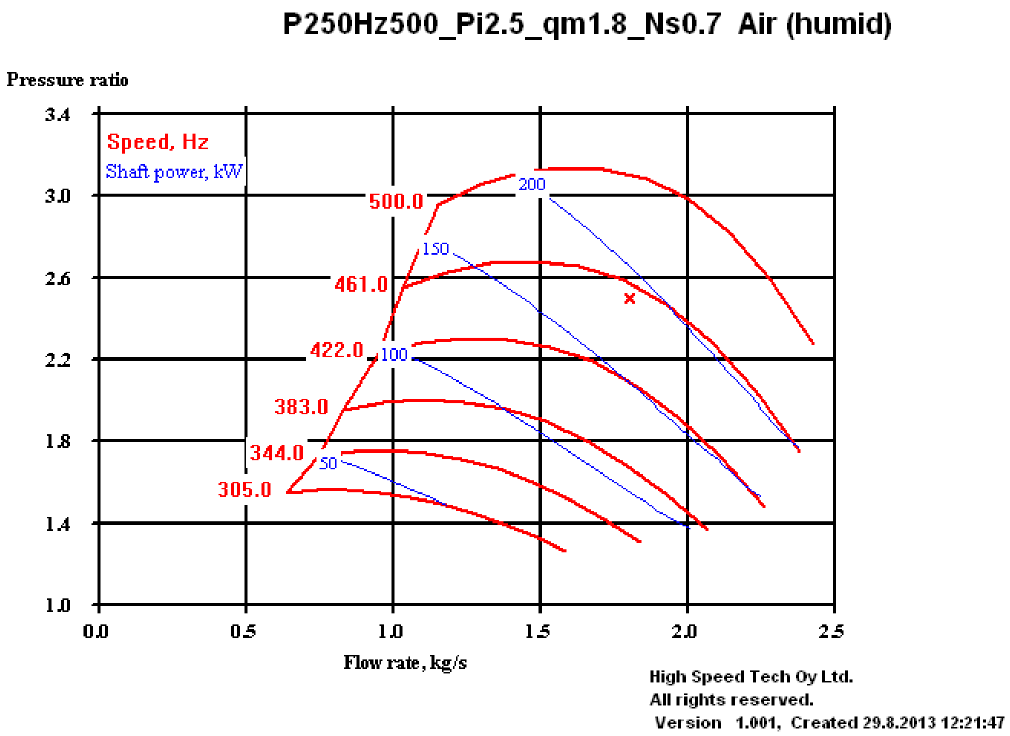



Figure 21. Compressor flow map.

The impeller of the compressor is shown in the Fig. 22.

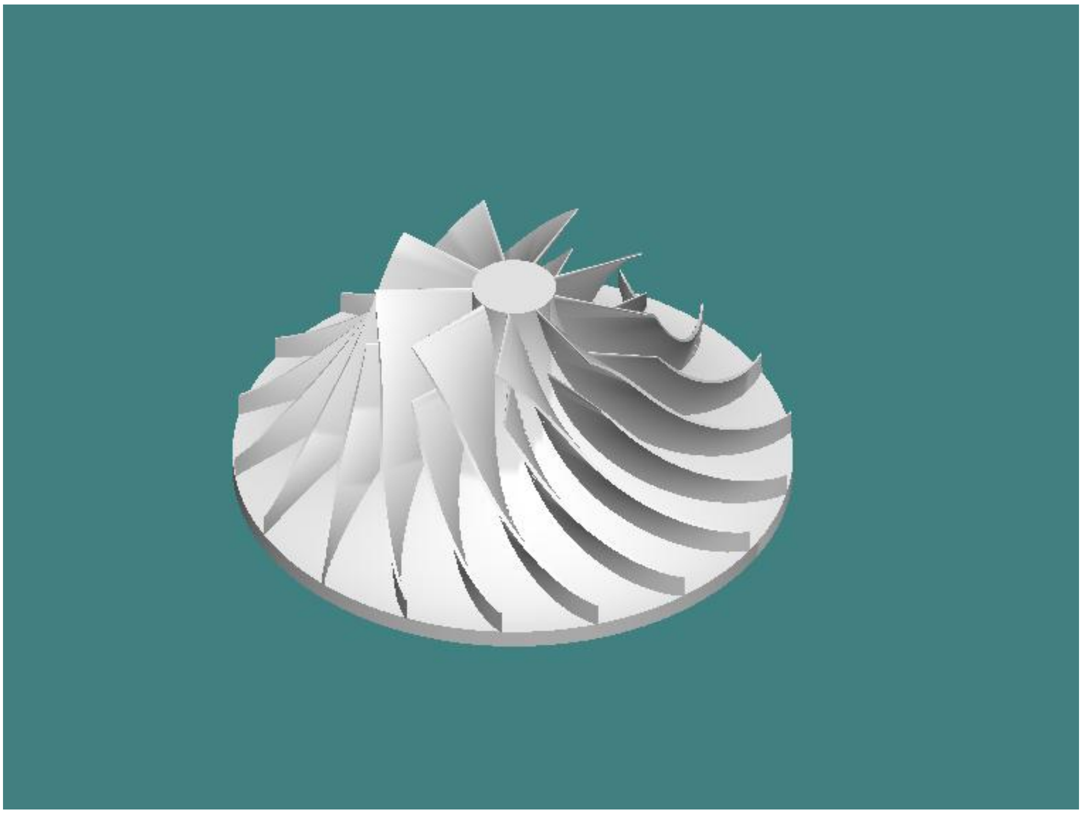


Figure 22. The impeller of the compressor.

## 3.2 Test Rig

The designed test stand is dedicated to study the performance of the centrifugal compressor that was described in the chapter 3.1. The following issues and topics will be investigated:

The test rig should be a closed loop stand. It means that there must be a heat exchanger between the compressor and surge tank to cool the circulating air. The cooling fluid in this heat exchanger is water from the University's water pipe. The cooling water has the following parameters: pressure is 6 bar, temperature is 10 °C, and mass flow is 4 kg/s. Also, there must be a surge tank in the loop to diminish waves and pulsations in the outlet flow and to create an appropriate inlet flow conditions.

In order to conduct these studies, the test rig should be equipped with a number of sensors and control devices. Pressure and temperature sensors in the compressor diffuser might be varied from one experiment to another, but the test stand must measure the following parameters:

- Conditions of the flow before the compressor inlet duct.
- Flow parameters after the compressor outlet duct.
- Air conditions inside the surge tank

- Ambient conditions in the laboratory

Moreover, the test stand should have a number of control valves to regulate air flow, and to reduce a pressure created by the compressor there should be a pressure reduction valve between the heat exchanger and the surge tank. To enable a negative gauge pressure in the inlet duct, there should be a control valve between the surge tank and inlet sensors. To regulate a water flow into the heat exchanger and, consequently, the outlet air temperature there should be a control valve in the inlet water pipe. Finally, to regulate the amount of air inside the test stand there should be an on/off valve in the surge tank.

Based on the above mentioned requirements, it is possible to establish a principal scheme of the designed test stand. The scheme is presented in the Fig. 23.

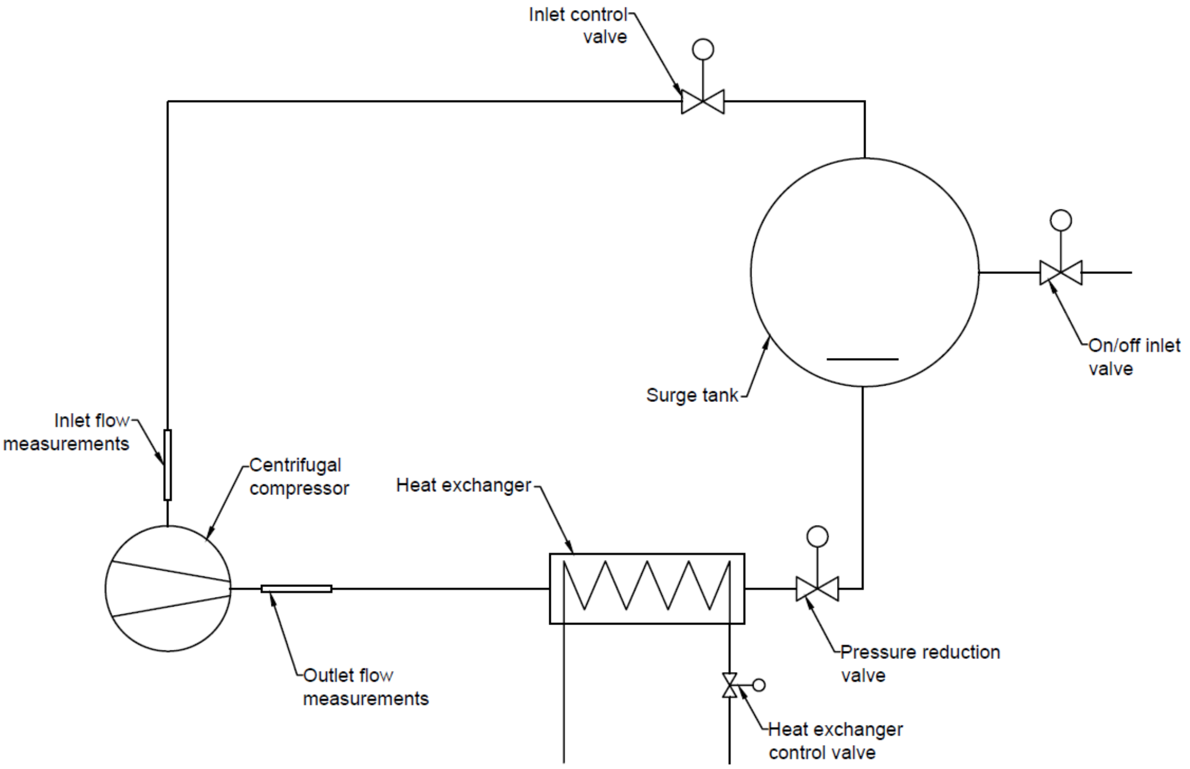



Figure 23. Test stand scheme.

Design process of this test stand is described in the chapters below.

# 4 DESIGNING OF THE NEW TEST STAND

## 4.1 Placing of Measuring and Control Devices

### 4.1.1 Pressure sensors

Pressure is one of the crucial parameters of air flow. The whole compressor performance is defined by the pressure ratio it produces. For this purpose pressure sensors are placed in the next locations:

- The inlet pressure is measured in the inlet of the compressor. To avoid distortions of the flow, this sensor is installed after a flow straightener in 2 m distance, which more than minimal length required by the standard (1) (2 diameters of the pipe).
- Outlet pressure is measured after the compressor outlet duct. This sensor is placed in 2 m distance after the outlet pipe. Consequently, the flow is stabilized and the pressure measurement is correct.
- Pressure inside the surge tank is measured through the holes in the reservoir's walls.

These sensors must have very good accuracy and must not affect the flow. This goal is achieved by drilling of 4 holes in a pipe and 2 holes in the surge tank. These openings are connected by 1 hose. The pressure sensors measure pressure inside the hose, thus an average values for selected cross-sectional areas are obtained.

For the designed test stand Druck UNIK 5000 pressure transmitters were chosen. Its specifications are listed below (27):

- Pressure range: 0 – 10 bar absolute.
- Temperature range: -55 °C to 125 °C.
- ±0.04% full scale best straight line accuracy.
- ±0.1% full scale stability.
- 4 to 20 mA output signal.

Figure 24. Druck UNIK 5000 pressure transmitters. (27)

### 4.1.2 Temperature sensors

Temperatures are also used in calculation of the compressor and heat exchanger efficiencies. For this purposes temperature sensors are placed in the next locations:

- Inlet temperature is measured before the compressor inlet. There is a flow straightener after the last pipe bend to stabilize the flow.
- Outlet temperature is measured in the outlet pipe. These sensors are placed in 2 m distance after the outlet of the compressor and measure the temperature before the heat exchanger.
- Temperature after the heat exchanger is measured in the pipe between the heat exchanger and pressure reduction valve.
- The temperature inside the surge tank is measured by the sensors on its walls.
- Cooling water inlet and outlet temperatures are measured by the sensors installed before and after the heat exchanger, respectively.

There are 4 sensors in first three locations that are placed with 90° interval in cross-sectional area. These values are summarized and average temperature is used in post processing. Measurements in the surge tank and water pipes are conducted with 4 and 2 thermocouples located with 90° and 180° interval, respectively.

Different types of thermocouples were studied before the final selection. Comparison of these thermocouples is presented in the Tab. 1.

Table 1. Comparison of different thermocouples

| Type | Temperature range, °C | Accuracy, whichever is greater | |
|---|---|---|---|
| | | Standard | Special |
| B | 0 to 1700 | 0.5 °C over 800 °C | - |
| C | 0 to 2320 | 4.5 °C to 425 °C, 1.0% to 2320 °C | - |
| E | -200 to 900 | 1.7 °C or 0.5 % above 0 °C, 1.7 °C or 1.0 % below 0 °C | 1.0 °C or 0.4 % |
| J | 0 to 750 | 2.2 °C or 0.75 % | 1.1 °C or 0.4 % |
| K | -200 to 1250 | 2.2 °C or 0.75 % above 0 °C, 2.2 °C or 2.0 % below 0°C | 1.1 °C or 0.4 % |
| N | -270 to 1300 | 2.2 °C or 0.75 % above 0 °C, 2.2 °C or 2.0 % below 0°C | 1.1 °C or 0.4 % |
| R | 0 to 1450 | 1.5 °C or 0.25 % | 0.6 °C or 0.1 % |
| S | 0 to 1450 | 1.5 °C or 0.25 % | 0.6 °C or 0.1 % |
| T | -200 to 350 | 1.0 °C or 0.75 % above 0 °C, 1.0 °C or 1.5 % below 0 °C | 0.5 °C or 0.4 % |

On the basis of this table thermocouples Type T were selected for designed test rig. These sensors have suitable temperature range and best accuracy in comparison with other thermocouples. T Type thermocouple consists of copper and constantan, both are non-magnetic metals.

### 4.1.3 Mass flow sensors

Mass flow rate is another important parameter of air flow. In the designed test stand pressure-based flow meters were chosen. As it was shown in the chapter 2.1.3, this type of meters relies on Bernoulli's equation (chapter 2.1.3, Eq. 5).

According to this equation, increasing of velocity leads to decreasing of pressure. Velocity is increased due to smaller cross-sectional area. In cone meters flow is accelerated by cone inside the tube. Since this cone also acts as a conditioning device, upstream requirements for pipe length are lower than in conventional differential pressure flow meters (from 0 to 3 pipe diameters according to (28)). Pressure difference is measured by differential pressure transmitter and then converted into analogue output signal. For the designed test stand a McCrometer V-cone flow meter is selected. Its picture is presented below:

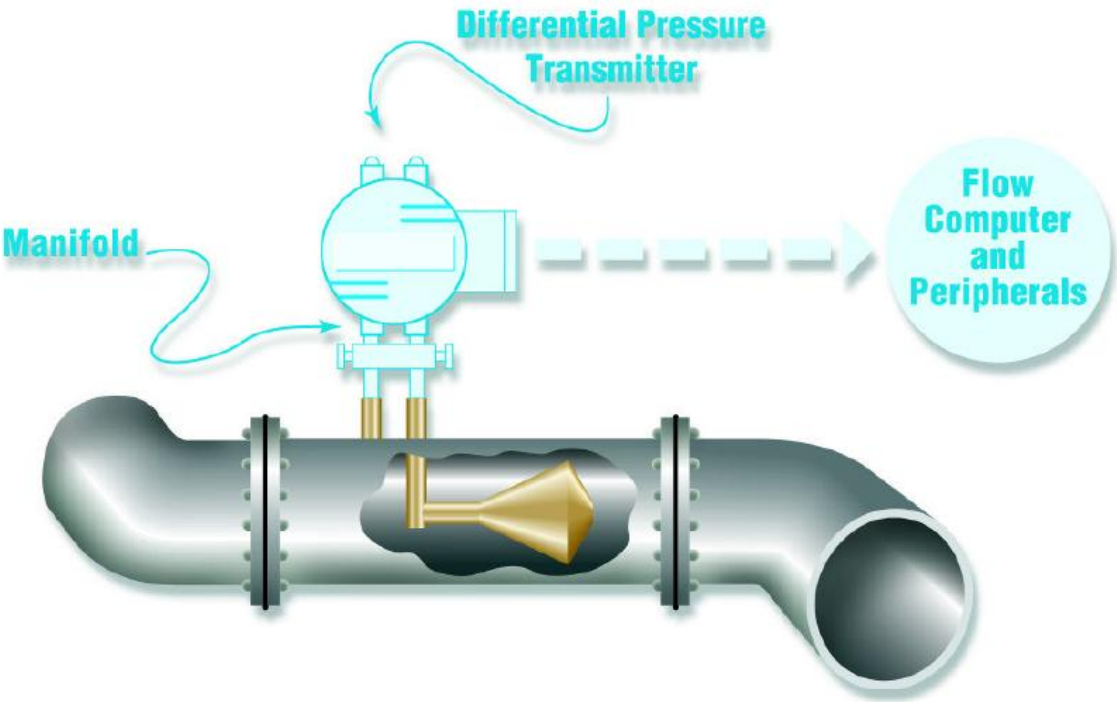



Figure 25. V-cone mass flow meter. (28)

The accuracy of the flow meter is ±0.5% of actual flow and repeatability is ±0.1%. Two flow meters are placed in the following locations:

- After the surge tank to measure inlet mass flow
- After the pressure and temperature sensors to the outlet pipe, to measure outlet mass flow.

Therefore, leakages in the compressor can be observed.

### 4.1.4 Valves.

As it was mentioned in the chapter 3.2, there are 4 different control valves in the designed test rig. Each of the valves has its own purpose and, therefore, designs. The purposes and the valve designs are described below.

The valve after the heat exchanger is a pressure reduction valve. The purpose of this valve is to reduce the pressure that is created by the centrifugal compressor and to create a back-pressure if it is required. The pressure after the valve in most cases should be equal to the atmosphere pressure, the maximum pressure before the valve is 300 kPa. For this purpose a butterfly valve is the most suitable one. A ball valve is also appropriate in this case, but its price is significantly higher.

A regulating mechanism in a butterfly valve is a disc. This disc rotates because of a rod that passes through the disc and connects it to an actuator. When the disc is turned along the pipe axis the valve is fully opened, but there is still a small pressure drop. When the disc is turned perpendicular to a flow the valve is fully closed. The actuator that turns the rod should be an electrical device governed by a 4…20 mA input signal, so the valve might be controlled by a data acquisition system.

Valves produced by different companies were studied, and a valve produced by a Metso company was chosen. The valve is called Neles High Performance Triple Eccentric Disc Valve. The feature of this valve is a flow balancing trim that reduces dynamic torque, noise and vibration levels, optimizes flow and improves control. The principle of the flow balancing trim in comparison with conventional butterfly valve is illustrated in the Fig. 26. Detailed description of this valve is given in (29).

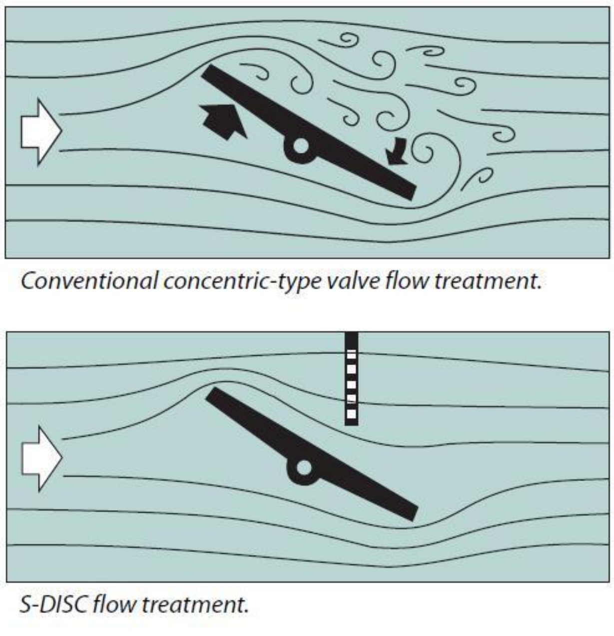



Figure 26. Conventional and S-DISC butterfly valves. (29)

According to Metso company recommendations a suitable electric actuator for this valve is Auma SGC 10.1. Overall view of the selected valve with the actuator (29) and its performance calculated by the Metso Company are presented in the appendix I.

The second valve is installed after the surge tank before the inlet flow meter. The purpose of this valve is to create a negative gauge pressure in the inlet manifold if it is necessary for a conducted experiment. In this case a butterfly valve is not suitable since it does not have enough capacity and full bore ball valve should be used instead. A metal sphere with a hole inside is used in this valve instead of metal disc. When the port is in line with pipe axis the flow will occur and the valve is fully opened. The ball might be rotated due to a rod that passes through the valve and connects it to the actuator. When the hole in the sphere is perpendicular to the flow direction, the valve is fully closed. A full bore valve has a sphere whose diameter is larger than the pipe diameter. It allows having a hole with the same size as the pipeline. Therefore, the flow friction is lower.

For the designed test rig the ball valve produced by Metso Company was selected. The model of the valve is Neles Flanged Full Bore MBV Ball Valve, Series M1 for PN Ratings. Description of this valve with technical specification is given in (30). The actuator for this valve is also electric with 4…20 mA input signal, but made by Remote Control company. Overall view of the valve with the actuator (30) and the performance of the valve provided by Metso Company and presented in the appendix II.

The valve for the heat exchanger control is a segment valve. The operational principle of this valve is close to the ball-type valve, but a rotating element of the valve that cut-off the flow is a segment of a sphere instead of a full ball. This selection was made due to different properties of the flowing fluid, which is water in this case. The model of the valve is Neles RA Series V-Port Segment Valve. Its technical specification and description are given in (31). An actuator for this valve is also electric with 4…20 mA signal made by Remote Control company. Overall view of this valve with the actuator (31) and its performance for different flow rates of inlet water are given in appendix III. Its temperature and pressure do not vary. The calculation is provided by Metso Company.

The purpose of the last valve is to regulate the amount of air inside the test stand loop before and after an experiment. Since it is a simple on/off valve, a butterfly valve might be used in this case.

This valve is similar to the pressure reduction valve after the heat exchanger except the flow balancing trim which is unnecessary in this case. Technical description and specification of this valve might be obtained from (32). Actuator for this valve is similar to described above. Fig. 27 shows an overall view of the selected valve with the actuator.

Figure 27. Overall view of the on/off intake valve. (32)

Since this valve is an on/off one a performance calculation is not required in this case.

## 4.2 The heat exchanger.

Centrifugal compressor significantly increases air pressure. As a result of this increment air temperature also increases. Since a closed-loop scheme is utilized in the designed test stand the need to cool circulating air occurs. For this purpose a heat exchanger has to be incorporated into the test rig.

To design a heat exchanger compressed air maximum outlet temperature should be defined. It might be obtained from the Eq. 9 for compressor isentropic efficiency:

$$\eta_s = \frac{T_{3s} - T_1}{T_3 - T_1} \quad (9)$$

Where: $\eta_s$ - compressor isentropic efficiency

$T_{3s}$ - theoretical outlet temperature, K

$T_1$ - inlet temperature, K

$T_3$ - outlet temperature, K

From the Eq. 9 outlet temperature is defined:

$$T_3 = \frac{T_{3s} - T_1}{\eta_s} + T_1 \tag{10}$$

Theoretical outlet temperature might be calculated from the Eq. 11:

$$T_{3s} = T_1 * \pi^{\frac{1.4-1}{1.4}} \tag{11}$$

Where: $\pi$ – compressor pressure ratio.

Compressor isentropic efficiency and pressure ratio are obtained from the compressor map. Therefore, compressor theoretical outlet temperature:

$$T_{3s} = 288 * 3^{\frac{1.4-1}{1.4}} = 394.2\ K$$

Compressor outlet temperature:

$$T_3 = \frac{394.2 - 288}{0.75} + 288 = 430\ K$$

Consequently, compressor maximum outlet temperature is ≈160 °C. The temperature after the heat exchanger should be approximately 30 °C.

Heat exchanger design process should consider the following topics (33):

- Fouling. It is inevitable process in heat exchangers. Despite that, the heat exchanger must perform required heat duty. If possible, the self-cleaning design should be realized, so the fouling layer will grow until a certain level and then stop. In the designed test stand a fouling issue is not a large problem since the fluids (compressed air and water) are relatively clean.
- Environment of use. The heat exchanger must be designed to withstand possible damages from surrounding environment during its operation, transportation, installation and potential misuse. The designed heat exchanger will be used in the Laboratory of Fluid Dynamic where the operational conditions are relatively good and any dangerous situations that could damage the heat exchanger are highly unlikely.

- Maintenance. It means that those parts of the heat exchanger that tend to suffer from corrosion, erosion, fouling or other forms of damage should be easily cleaned or removed. Maintainability also means that surrounding space should be suitable for above mentioned procedures.
- Flexibility. Heat exchanger should be suitable for the different mass flow rates that are possible for the compressor.
- Limitations of mass and dimensions. These parameters might be restricted due to different reasons. The designed heat exchanger is limited in its length at 3 m.
- Minimizing the cost. The cost of the heat exchanger always should be minimized. Often the total heat transfer area is the main cost-related parameter. Usually it means that the fluid velocities should be maximized in order to minimize the heat transfer area.

Flow arrangement is very important topic in the design of a heat exchanger for efficient usage of the heat transfer area. The simplest single-pass arrangements are counter-flow and parallel-flow. Their temperature profiles are shown in Fig. 28.

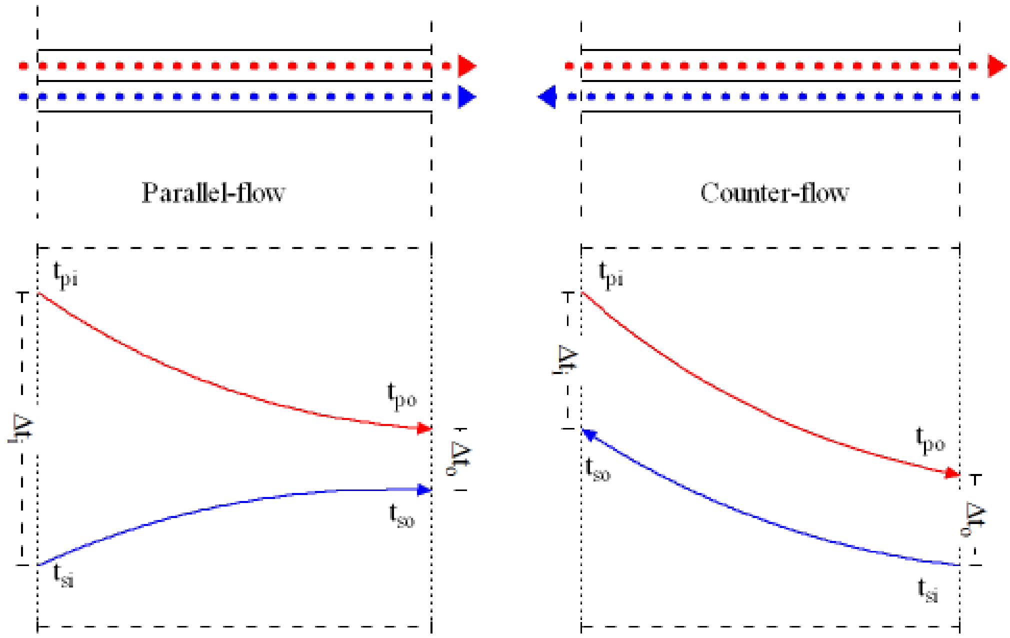



Figure 28. Parallel-flow and counter-flow arrangements. (33)

These temperature profiles show that the counter-flow arrangement enables more heat transfer with moderate heat transfer rates. Parallel-flow arrangement has the very large temperature difference at the inlet stage and small temperature difference at the outlet stage. It might create high thermal stresses in heat transfer areas. The main advantage of the parallel-flow arrangement is a more uniform temperature distribution and a lower maximum temperature of the heat transfer surface. In some cases these advantages might be very important (33).

There are also different types of physical construction of heat exchanger (33).

The first and the simplest type is tubular double-pipe heat exchanger. It consists of two concentric pipes and appropriate end fittings. If necessary, the inner tube might have longitudinal fins on its surface to increase the heat transfer area which is relatively small compared to other heat exchanger types. Double-pipe heat exchangers are used in the applications where a small heat transfer area is needed. Also, the following advantages of double-pipe heat exchangers are mentioned in (33):

- Flexibility in building and installing.
- Quick designing and assembling from standard components.
- Easy accomplishing of counter-flow arrangement.
- Suitable for high-pressure fluids without excessive metal thickness.
- Easy maintenance and disassembling for cleaning.
- Well-known simple calculation methods with accurate results are available.

Shell-and-tube is the most popular type of heat exchangers due to its flexibility in sizes, pressure and temperature ranges, construction materials, and designs (33). Fig. 29 shows a schematic image of shell-and-tube heat exchanger with its main parts.

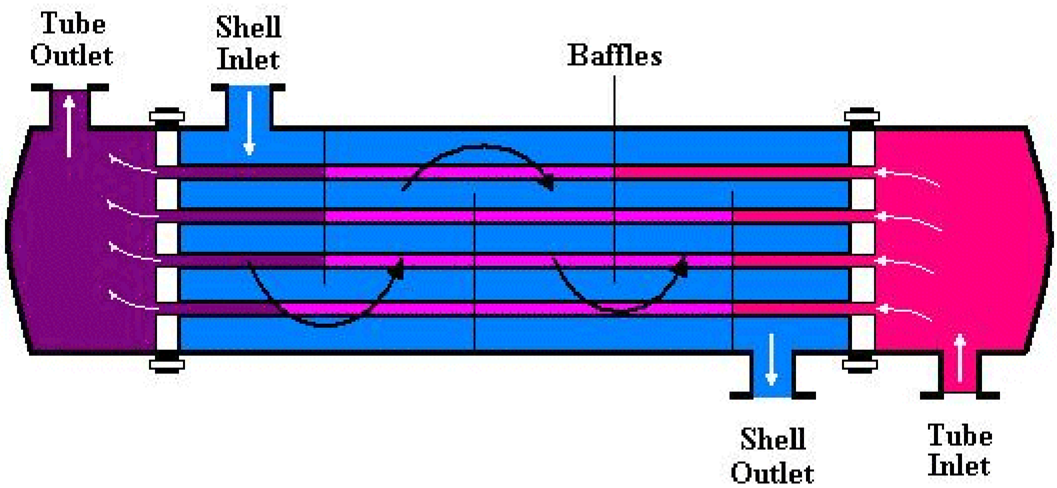



Figure 29. Shell-and-tube heat exchanger scheme. (35)

Tube side fluid (hotter one in the example) enters the heat exchanger through tube inlet section at one end and then moves into the tubes. At the other end of the heat exchanger, the fluid exits the tubes and then the heat exchanger through the tube outlet section. Cold fluid enters the shell side of the heat exchanger through shell inlet section and then flows in cross-parallel direction in respect to the tubes direction guided by baffle plates, until shell outlet section. Overall flow direction of cold and hot fluids in the given example is counter-flow, but parallel-flow is also possible in shell-and-tube heat exchangers.

Different designs of shell, tube bundle, front- and rear-end heads, and flow directions are possible. Their description and classification is provided by Tubular Exchanger Manufacturer's Association (TEMA). A three-letter abbreviation defines the most common types of shell-and-tube heat exchangers. Fig. 30 represents this classification.

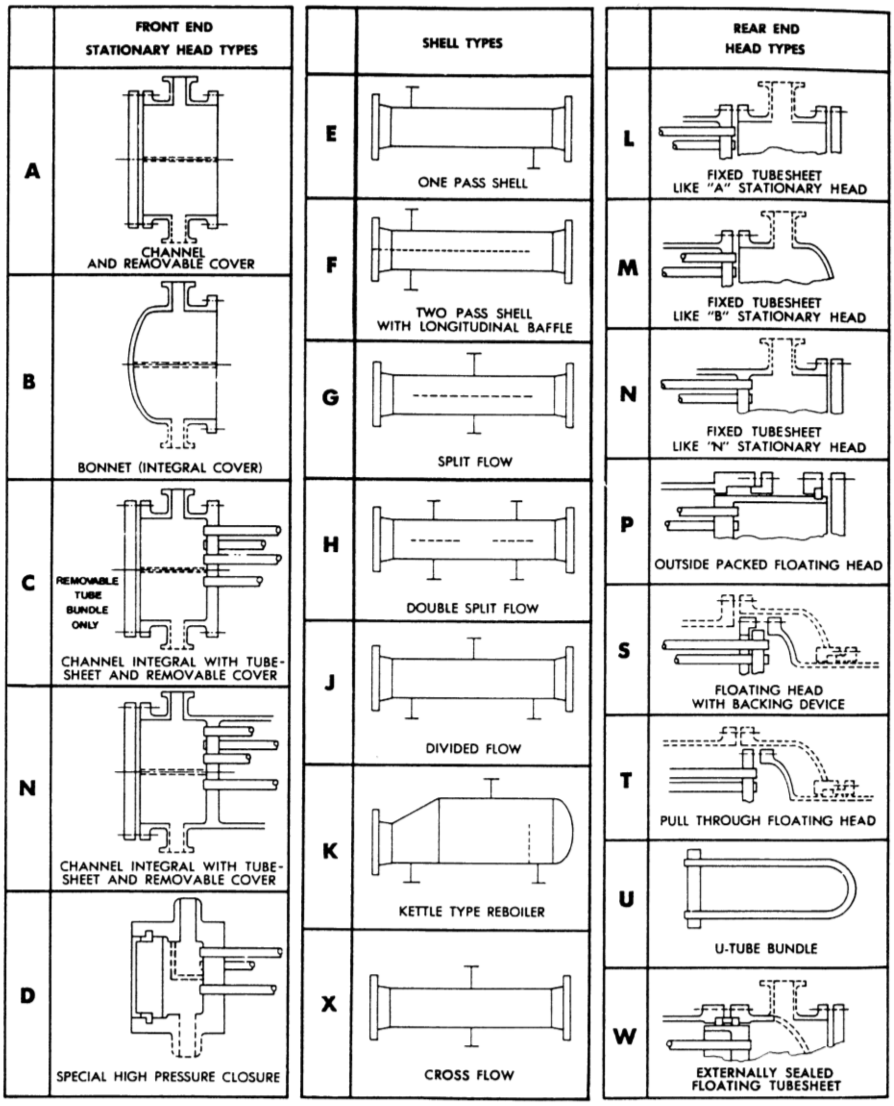



Figure 30. TEMA shell-and-tube heat exchangers classification. (36)

The first letter defines the design of front end head. The second letter describes the shell type and the third letter specifies the rear end head. According to this classification, the heat exchanger in Fig. 29 is a BEM type heat exchanger. In general, the E type shell construction is the most popular and the simplest while at the same time it provides the most effective use of heat transfer area (33).

Other design parameters that affect a tube-and-shell heat exchanger performance and efficiency are the tubes arrangement and baffles configuration. The factors that affect the selection of these parameters are the following ones: expected fouling rate, fluid velocities and flow regimes, fluid properties, permitted pressure drops, and other factors. Detailed description of different tubes and baffles arrangements is given in (33).

Plate heat exchangers are usually smaller, lighter and cheaper than shell-and-tube heat exchangers, but their maximum operational pressures and temperatures are also lower. Typical fluid pressure for this type of heat exchangers is 0.1 to 1.0 MPa and typical temperature is up to 150 °C (33). The restricting detail is gasket that controls fluid flows between the plates. Usual material for this gasket is hard rubber. The plates are made of stainless steel or aluminium or titanium since carbon steel is unsuitable due to corrosion. A typical plate heat exchanger is presented in Fig. 31.

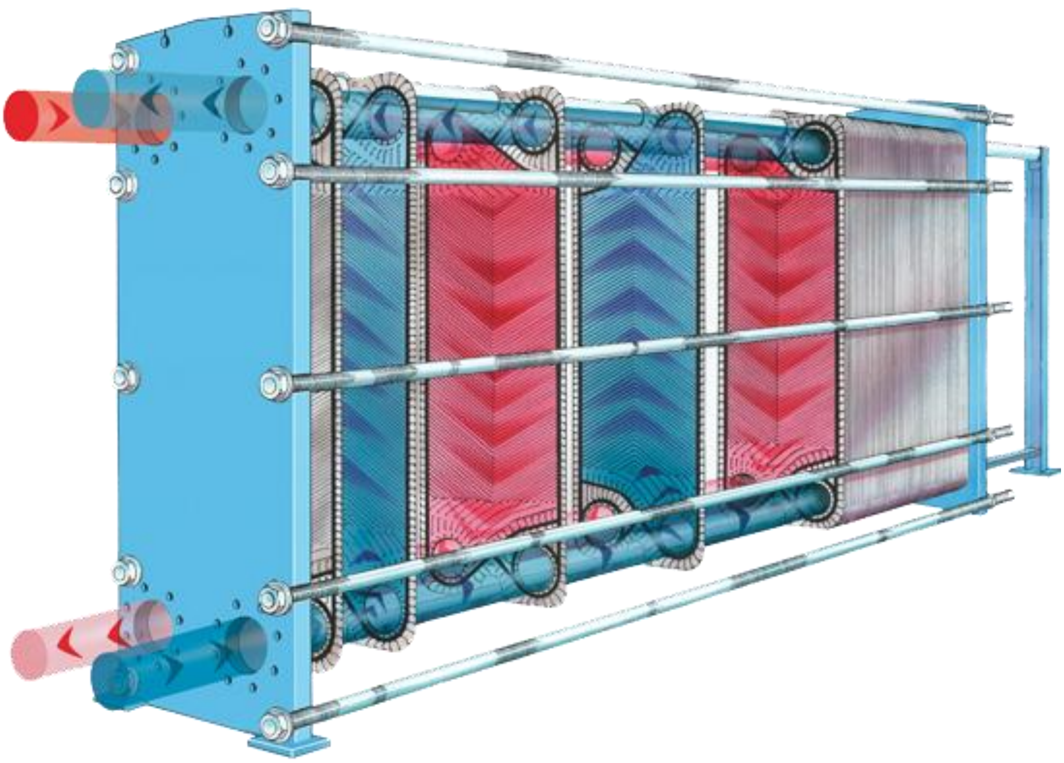


Figure 31. Plate heat exchanger. (37)

Selection of appropriate heat exchanger type is described in the next way in (33): “Generally the goal should be the cheapest overall solution that achieves the required thermal performance within described limits (i.e. maximum allowable pressure drop, size, weight, dimensions, etc.)”. Taking into account everything that was mentioned above the compact tube-and-shell heat exchanger type was selected. It is produced by a company GEA Renzmann & Grünewald GmbH. Shell side fluid is cooling water and tube side fluid is air. TEMA specification sheet and the heat exchanger drawings are given in the appendix I and II, respectively. 3D model of the after cooler is presented in the Fig. 32.

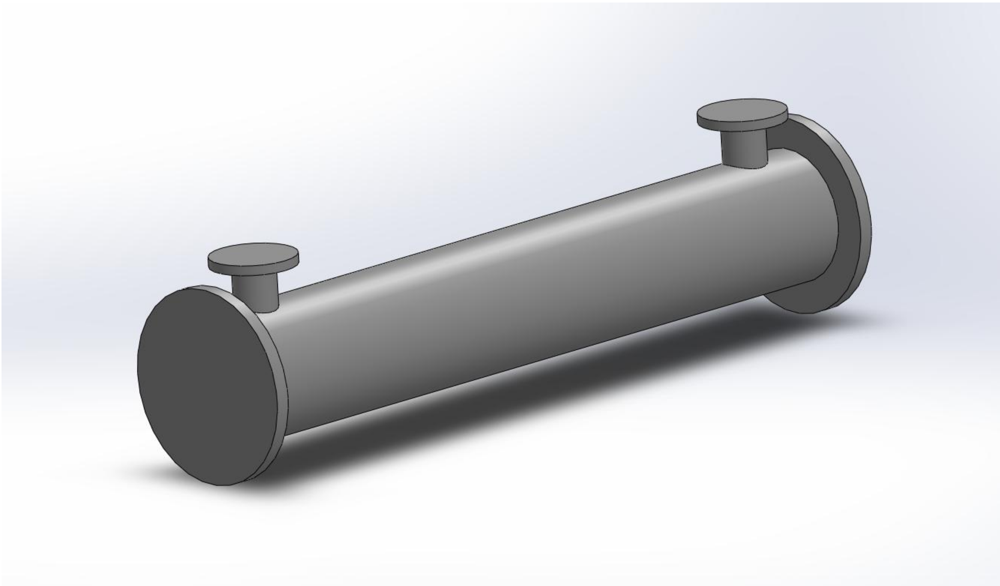


Figure 32. 3D model of the chosen heat exchanger.

## 4.3 Piping

All the components of the test stand are connected together with pipes. The diameter of these pipes should be selected according to expected mass flow, so the velocity of air inside the pipes will be appropriate. Wall thickness must be large enough to withstand all possible pressures inside the pipes. Since the maximum possible gauge pressure inside the piping is 4 bar, the tubes with pressure rating PN10 were selected. For better work of pressure reduction valve a nominal pipe diameter DN200 was chosen. Outer diameter of this pipe is 219 mm, internal diameter is 213.5 mm. Thus, the velocity in the inlet duct of the compressor in this case might be obtained from the Eq. 12 for mass flow rate calculation:

$$q_m = \rho * v * A \tag{12}$$

Where $q_m$ – mass flow rate, kg/s

$\rho$ – fluid density, kg/m$^3$

$v$ – fluid velocity, m/s

$A$ – pipe cross-sectional area, m$^2$

Then, the velocity of the flow is defined:

$$v = \frac{q_m}{\rho * A} = \frac{q_m * 4}{\rho * \pi * D^2} \tag{13}$$

$$v = \frac{1.8 * 4}{1.225 * \pi * 0.213^2} = 41.3\ m/s$$

To define the velocity in the outlet duct of the compressor the density of the compressed air should be defined. The equation of state of ideal gas:

$$p * V = \frac{m}{M} * R_{mol} * T \tag{14}$$

Where $p$ – gas pressure, Pa

$V$ – gas volume, m$^3$

$m$ – gas mass, kg

$M$ – molar mass, kg/mol,

$R_{mol}$ – universal gas constant, J/(K·mol)

$T$ – gas temperature, K

Dividing left and right parts of the Eq. 14 by $V$ yields:

$$p = \frac{m}{V * M} * R * T$$

Notice, that $\frac{m}{V}$ is, by definition, equal to gas density. Thus,

$$p = \frac{R * T}{\rho * M} \tag{15}$$

And air density might be calculated according to the Eq. 15:

$$\rho = \frac{p * M}{R * T} = \frac{250000 * 0.02896}{8.31 * 391} = 2.23\ kg/m^3$$

Therefore, the velocity in the compressor outlet duct might be defined according to Eq. 13:

$$v = \frac{q_m * 4}{\rho * \pi * D^2} = \frac{1.8 * 4}{2.23 * \pi * 0.213^2} = 22.7\ m/s$$

The calculated values of the velocities are within recommended range (up to 25 m/s (33)). In other words, the pipe size DN200 is suitable for the designed test stand.

## 4.4 The surge tank

As it was mentioned in the chapter 3.2, the closed-loop compressor test stand must be equipped with a surge tank. The purpose of this tank is to minimize wave and pulsation processes and to create appropriate inlet flow conditions. The tank should have 3 connections: inlet pipe, outlet pipe and auxiliary pipe with on/off valve to the atmosphere.

The volume of the tank might be roughly estimated as a volume of all the air tubes in the test stand. Estimated length of the connecting pipes is 10 m plus ≈4 m of mass flow meters, heat exchanger, control valves; internal diameter is approximately 215 mm. Thus, the volume of the pipes might be calculated as a volume of large cylinder:

$$V = \frac{\pi * d^2}{4} * l = \frac{\pi * 0.215^2}{4} * 14 = 0.508\ m^3$$

Therefore, the volume of the tank should be approximately 0.6 m$^3$. From technological point of view the cylindrical shape of the tank is preferable. The volume of the cylinder is defined according to the Eq. 16:

$$V = \frac{\pi * d^2}{4} * l \qquad (16)$$

Thus, to define dimensions of the tank with known volume, one of the dimensions should be established. Assuming that the diameter of the tank is 0.7 m, the length of the tank can be defined from the Eq. 16:

$$l = \frac{4 * V}{\pi * d^2} = \frac{4 * 0.6}{\pi * 0.7^2} = 1.56\, m$$

So, the required surge tank has a cylindrical shape with the following dimensions: diameter is 0.7 m and the length is 1.6 m. The most suitable way of installing this tank in the laboratory is the vertical installation. It means that the inlet and outlet pipes are located at the top and at the bottom of the tank and auxiliary pipe with on/off valve is located at the side of the cylinder. Schematic drawing of the tank is presented in the Fig. 33 and detailed drawings are given in Appendix VI.

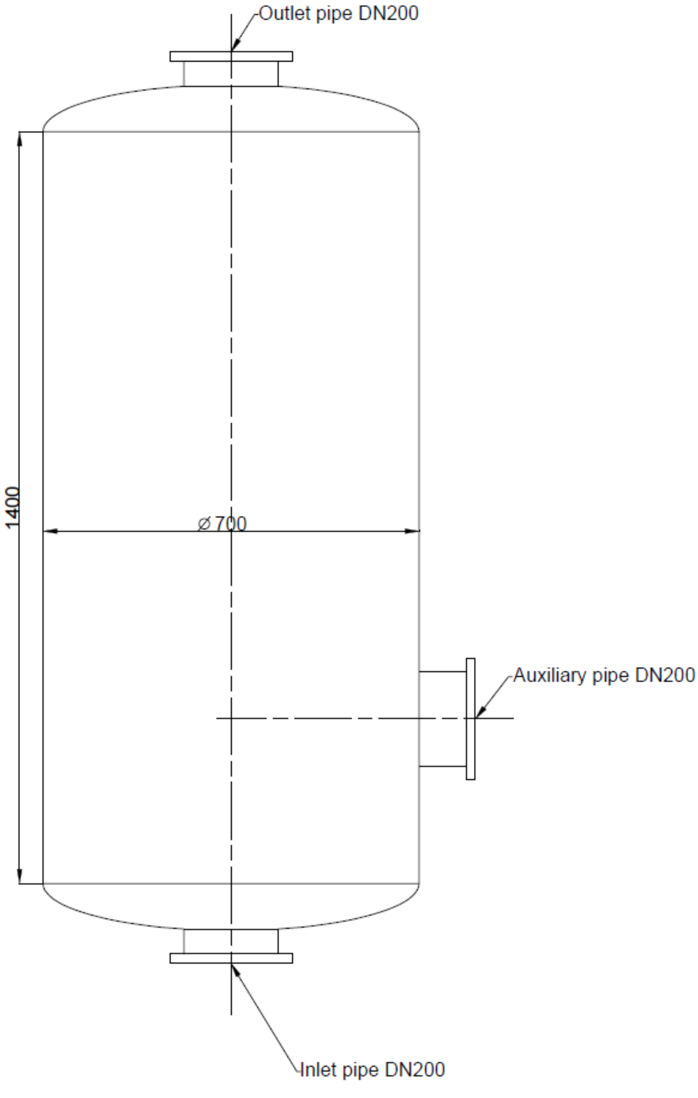



Figure 33. Schematic drawing of the surge tank.

## 4.5 Data acquisition system

Data acquisition is a process of measuring physical values and converting them into digital form that might be processed by a computer. Typically, a data acquisition system consists of the following elements:

- Sensing devices that transform physical values into electrical analogue signals (mA or mV)
- Signal conditioning devices that process the signals. The purpose of these devices is to convert input signal into appropriate output signal that might be easily read by data acquisition device. Amplification, filtering, excitations, linearization are typical treatment that are applied to the signals of the sensors.
- Analogue-to-digital converter that transforms input conditioned analogue signal into output digital signal.

To select suitable data acquisition system it is necessary to determine the number and type of input signals from sensors and required output signals to control devices. Analysis of input signals is presented below:

- Mass flow sensor has a 4…20 mA output signal and there are two mass flow sensors in the test rig.
- Temperature sensor (thermocouple) has an output signal in several millivolts. This signal requires several conditioning to be read by data acquisition system afterwards. There are 4 thermocouples in the compressor inlet pipe, 4 thermocouples in the compressor outlet pipe, 4 thermocouples after the heat exchanger, 2 thermocouples in the surge tank, and 4 thermocouples in the water pipe: 2 before and 2 after the heat exchanger. The total number of thermocouples in the designed test stand is 18.
- Pressure sensor has a 4…20 mA output signal. The test stand is equipped with 3 pressure sensors.
- Ambient sensor (that measures pressure, temperature and humidity in the laboratory) has 3 output signals (one for each measured parameter) 4…20 mA.
- Humidity sensor in the surge tank also has a 4…20 mA output signal.
- Actuators for the valves have a 4…20 mA feedback signal. There are 4 valves in the test rig.
- Power measuring device generates 4…20 mA output signal.
- Rotational speed measurements yields 4…20 mA output signal.

Data acquisition system also acts as a governing device for control valves. It means that there should be 4 output signals from DAQ system to electric actuators. The type of these signals is 4…20 mA.

Data acquisition system of the National Instruments Company was selected for the designed test stand. The type of the system is a NI Compact DAQ system. It consists of a desktop chassis and different modules that are installed into this chassis. Each module might acquire or generate specific type of signal with several channels. Tab. 2 summarizes the number and types of above mentioned signals in the designed test stand.

Table 2. Signals in the designed test stand.

| Type of signal | Number of channels | Possible DAQ system module |
|---|---|---|
| Output 4…20 mA | 15 | NI 9208 |
| Output thermocouples mV signal | 18 | NI 9214 |
| Input 4…20 mA | 4 | NI 9265 |

Chassis for the designed test stand is NI cDAQ-9188 chassis. It allows using up to 8 different National Instruments C Series modules. The chassis is connected to a desktop computer via Ethernet cable.

Output 4…20 mA signals are acquired by NI 9208 16-Channel current input module. It has 16 current inputs (±21.5 mA) with 500 S/s sample rate and 24-bit resolution. The module has built-in 50/60 Hz noise rejection.

NI 9214 High-Accuracy Thermocouple Module reads the signals from thermocouples. This module might acquire signals of J, K, T, E, N, B, R and S-types thermocouples. Multiple cold-junction compensation is built-in into this module. Voltage measurements range is ±78.125 mV.

Output signals are created by NI 9265 Analogue Output Module. It might generate signals from 0 to 20 mA output range with 16-bit resolution. It has 4 analogue outputs 100 kS/s each.

# 5. OVERVIEW OF THE DESIGNED TEST STAND

The chapters 4.1 – 4.5 describe the main components and parts of the designed test stand. Summarizing above mentioned information, one can conclude that the test stand looks in the next way: it is a closed-loop stand with the compressor installed in a vertical position. After the compressor outlet duct compressed air passes the measuring stage located at 2 m distance where the flow pressure, temperature and mass flow are measured. There is an air/water heat exchanger after the measuring stage where cold water from a tap cools the compressed air. The temperature of the cooled air is measured after the heat exchanger and then the air pressure is reduced by the pressure reduction valve. The air with reduced pressure goes to the surge tank. This tank has inlet pipe, outlet pipe, and auxiliary pipe with on/off valve for regulating the amount of air inside the test stand. The inlet pipe consists of the valve, flow straightener and inlet air measuring stage. The valve is installed after the surge tank and its task is to regulate air mass flow and pressure.

A 3D model of the designed stand is presented in the Fig. 34. The positions on the figure are the following ones:

1 Inlet pressure and temperature measurements,

2 Flow straightener,

3 Inlet mass flow meter,

4 Inlet control valve,

5 Surge tank,

6 Auxiliary valve,

7 Outlet pressure and temperature measurements,

8 Outlet mass flow meter,

9 Heat exchanger,

10 Temperature measurements after the heat exchanger,

11 Pressure reducing valve.

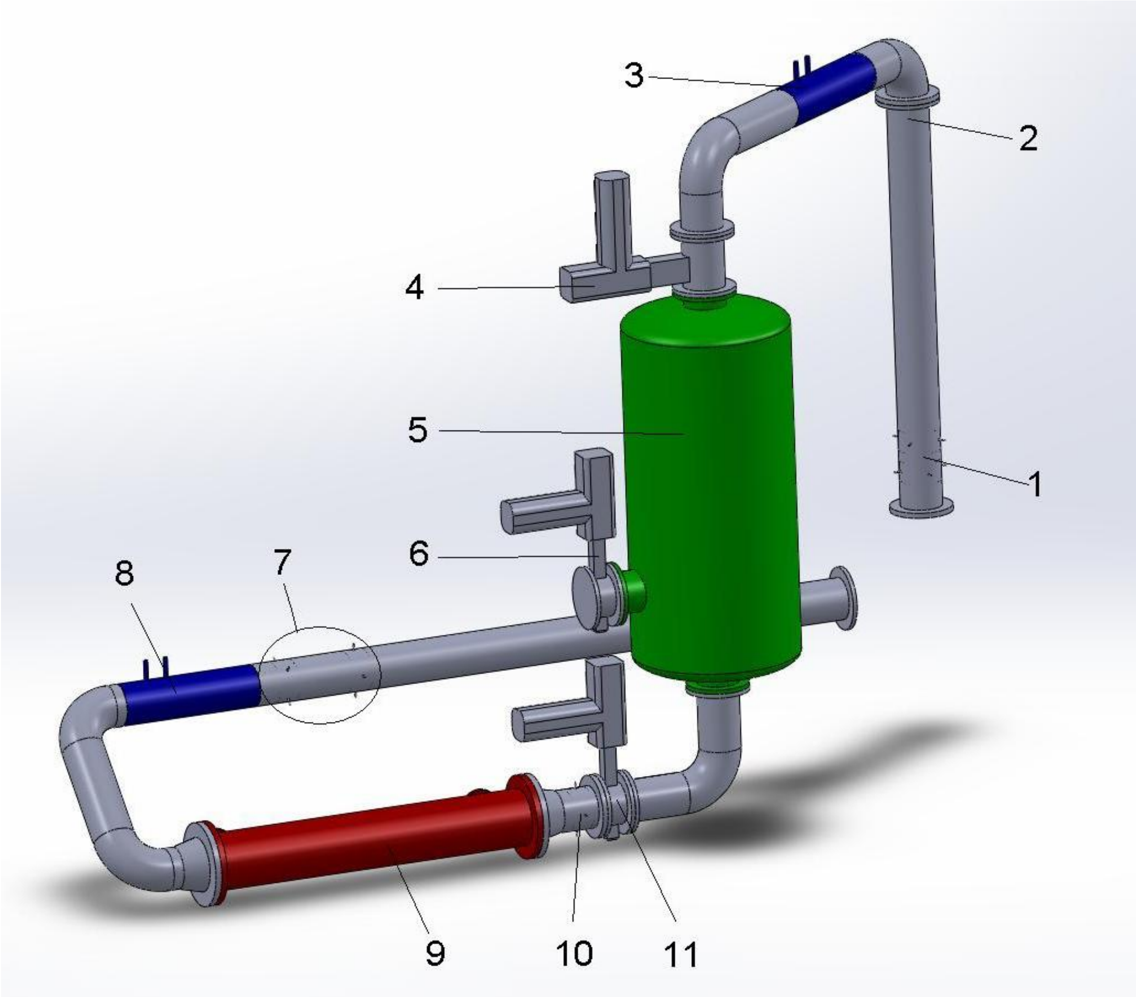



Figure 34. Layout of the designed test stand.

The designed test stand will be equipped with strain gauge pressure transducers that have suitable pressure and operating temperature ranges and high accuracy. The temperature will be measured by T type thermocouples. This type of thermocouples combines the most suitable measurement range for the expected temperatures with the best accuracy, in comparison with other thermocouples. Mass flow rate measurements will be conducted with V-cone differential pressure transmitter. This flow meter requires very short straight inlet pipe since its cone acts as a conditioning device.

## 5.1 Measuring uncertainties

Since measurement errors are unavoidable in any physical experiments it is necessary to determine an accuracy of the test stand. Determination of the measuring uncertainties depends upon the method of measurement. The obtained results of a direct measurement correspond directly to the final value, while indirect measurements require mathematical or physical processing. In (2) the following principal errors are specified:

1) Before the experiment:
    a) Simulation uncertainties of the physical phenomena
    b) Representation uncertainty
2) Measurement instrument problems:
    a) Error in determination of reference value
    b) An instrument might influence the measuring parameter
    c) If the sensor consume energy and it is furnished by the sensor itself, the output signal might be influenced
    d) The imperfectness (friction, clearances, ageing, etc.) of the instrument
    e) Hysteresis, zero-drift, reference value errors
    f) Noise in the instrument and in the measured values
    g) Errors in analogue to digital conversion.
3) Human error sources:
    a) The observer is imperfect
    b) Accuracy of the manipulations is not sufficient.
    c) Interpretation of the obtained results is not objective.

From other point of view, these errors might be classified into the next categories (2):

- Large errors.
- Systematic errors.
- Stochastic errors.

Large errors are easy to detect since they are usually far apart from other obtained results.

There are many reasons for systematic errors and these kinds of errors are usually difficult to find. They have the same sign and absolute value and usually are repetitive. Furthermore, there is no theory to treat them (2).

Stochastic errors might also appear due to different sources. Some systematic errors might also occur as a stochastic and vice versa: if a stochastic error occurs constantly and with the same sign is might be considered as a systematic error.

Analogue to digital conversion error depends on the resolution of the converter and the measuring interval. For instance, the NI 9214 High-Accuracy Thermocouple Module has a 24 bits resolution. It means that the number of possible values in this case is $2^{24}$=16777216. Measurement range of this module is ±78.125 mV. Thus, 1 bit corresponds to $9.31*10^{-6}$ mV and the quantification error is $\pm 4.65*10^{-6}$ mV, which is relatively small in comparison with other errors in the measuring circuitry. This high accuracy is achieved because of high resolution of the measuring module. Another module with 12 bits resolution would have $2^{12}$=4096 values and 0.038 mV quantification error, which is significantly greater and might affect the final result.

# 6 CONCLUSIONS

During this thesis work the test stand for centrifugal compressors was designed. Before the actual design several test rigs in different research laboratories were studied (their description might be found in the chapter 2.3), as well as a theoretical background and standards for measuring methods and equipment (chapter 2.1).

The stand has a closed-loop scheme with a surge tank. The air flow is controlled by control valves that are installed before and after the compressor. Selection process and description of the chosen valves are given in the chapter 4.1.4. Compressed air is cooled in the air/water tube-and-shell heat exchanger. Design and selection of the heat exchanger is described in the chapter 4.2.

Air flow parameters are measured in different locations: mass flow meters are installed before and after the compressor; air pressure is measured in the same places plus there is a pressure transducer in the surge tank. Temperature of the air is measured in the next locations: before the compressor, after the compressor, after the heat exchanger and in the surge tank. Also there are temperature measurements of the cooling water before and after the heat exchanger. Ambient conditions in the laboratory (temperature, pressure, and humidity) are measured in order to have reference values.

# REFERENCES

1. *Turbocompressors - Performance test code.* s.l. : ISO 5389:2005.

2. **Fransson, Torsten H.** *Measuring Techniques in Thermal Engineering: An Introduction in the Form of Lecture Notes.* Stockholm, Sweden : s.n., 1994.

3. **Hordeski, Michael.** *Transducers for automation.* New York : Van Nostrand Reinhold Company, 1987. ISBN 0-442-23700-6.

4. Pressure Sensors. *Controls Wiki.* [Online] [Cited: 10 08, 2013.] https://controls.engin.umich.edu/wiki/index.php/PressureSensors.

5. Force-related measurements. [Online] [Cited: 08 27, 2013.] http://www.omega.com/literature/transactions/volume3/trantocvol3.html.

6. Introduction to temperature measurement. [Online] Omega. [Cited: 08 26, 2013.] http://www.omega.com/prodinfo/temperaturemeasurement.html.

7. **Doebelin, Ernest.** *Measurement Systems. Application and Design. Fourth Edition.* s.l. : McGraw-Hill, Inc, 1990. ISBN 0-07-017338-9.

8. Temperature Sensors. *Controls Wiki.* [Online] [Cited: 10 08, 2013.] https://controls.engin.umich.edu/wiki/index.php/TemperatureSensors.

9. *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full -.* s.l. : ISO 5167:2003.

10. Velocity meters. *Automation Wiki.* [Online] [Cited: 10 08, 2013.] http://automationwiki.com/index.php?title=Velocity_Meters.

11. **Hayward, Alan.** *Flowmeters. A basic guide and source-book for users.* London : The Macmillan Press Ltd, 1979. ISBN 0-333-21920-1.

12. Types of Gas Mass Flow Meters. [Online] Alicat. [Cited: 10 08, 2013.] http://www.alicat.com/technical-information/types-of-gas-mass-flow-meters/.

13. Bernoulli's Equation. [Online] [Cited: 08 26, 2013.] http://hyperphysics.phy-astr.gsu.edu/hbase/pber.html.

14. Differential Pressure Flowmeters. *Flow & Level Measurement .* [Online] Omega. [Cited: 08 27, 2013.] http://www.omega.com/literature/transactions/volume4/T9904-07-DIFF.html.

15. **Miller, Richard.** *Flow measurement engineering handbook.* s.l. : McGraw-Hill Book Company, 1983. ISBN 0-07-042045-9.

16. *Effect of Vaneless Diffuser Width on the Overall Performance of a Centrifugal Compressor.* **Jaatinen, A.; Grönman, A.; Turunen-Saaresti, T.; Röyttä, P.** 665-673, s.l. : Proc. Inst. Mech. Eng., A J. Power Energy, 2011, Vol. 225.

17. *Experimental study of the effect of the tip clearance to the diffuser flow field and stage performance of a centrifugal compressor.* **Jaatinen A.; Turunen-Saaresti T.; Grönman A.; Röyttä P.; Backman J.** Copenhagen, Denmark : ASME Turbo Expo, June, 11-15, 2012. paper GT2012-68445.

18. *Experimental study of centrifugal compressor vaneless diffuser width.* **Jaatinen A.; Röyttä P.; Turunen-Saaresti T.; Grönman A.** s.l. : Journal of Mechanical Science and Technology, 2013, Vol. 27 (4). 1011-1020.

19. *Experimental study of vaned diffusers in centrifugal compressor.* **Jaatinen A.; Turunen-Saaresti T.; Grönman A.;Backman J.** Glasgow, UK : ASME Turbo Expo, June 14-18, 2010. paper GT2010-22883.

20. *Experimental Study of Centrifugal Compressor Tip Clearance and Vaneless Diffuser Flow Fields.* **Jaatinen A.; Turunen-Saaresti T.; Röyttä P.; Grönman A.; Backman J.** s.l. : Proc. Inst. Mech. Eng., A J. Power Energy, in press.

21. *DIN 1952:1971 Durchflußmessung mit genormten Düsen, Blenden und Venturidüsen.* s.l. : Deutsches Institut für Normung e.V.

22. RADIVER. *Institute of Jet Propulsion and Turbomachinery.* [Online] RWTH Aachen University. [Cited: 08 22, 2013.] http://www.ist.rwth-aachen.de/typo3/index.php?id=953&L=1.

23. Transsonic Centrifugal Compressor. *Institute of Jet Propulsion and Turbomachinery.* [Online] RWTH Aachen University. [Cited: 08 22, 2013.] http://www.ist.rwth-aachen.de/typo3/index.php?id=1048&L=1.

24. Aero-Engine centrifugal Compressor. *Institute of Jet propulsion and Turbomachinery.* [Online] RWTH Aachen University. [Cited: 08 22, 2013.] http://www.ist.rwth-aachen.de/typo3/index.php?id=954&L=1.

25. Radial Compressors. *Laboratory for Energy Conversion.* [Online] Swiss Federal Institute of Technology Zurich. [Cited: 08 22, 2013.] http://www.lec.ethz.ch/research/power_energy_turbomachinery/radial_compressors.

26. Radialverdichter. *Institute of Propulsion Technology.* [Online] German Aerospace Centre. [Cited: 08 22, 2013.] http://www.dlr.de/at/en/desktopdefault.aspx/tabid-1529/2164_read-4786/.

27. Druck UNIK 5000 Pressure Sensing Platform. *GE Measurement & Control.* [Online] 2011. [Cited: 09 26, 2013.] http://www.ge-mcs.com/download/pressure-level/920-483I-LR.pdf.

28. *McCrometer V-cone.* USA : s.n., 2005.

29. Neles High Performance Triple Eccentric Disc Valve With Flow Balancing Trim. *Metso valves.* [Online] [Cited: 07 10, 2013.] www.metso.com/valves.

30. Neles Flanged Full Bore MBV Ball Valve. *Metso Valves.* [Online] [Cited: 08 26, 2013.] www.metso.com/valves.

31. Neles RA Series V-port Segment Valve. *Metso Valves.* [Online] [Cited: 07 16, 2013.] www.metso.com/valves.

32. Neles High Performance Triple Eccentric Disc Valve, Metal Seated Neldisc Series L12. *Metso Valves.* [Online] [Cited: 07 16, 2013.] www.metso.com/valves.

33. **Saari, Jussi.** *Heat Exchanger Thermal Design Guide.* Lappeenranta : Lappeenranta University of Technology, 2010. ISBN 978-952-214-968-8.

34. Arithmetic and Logarithmic Mean Temperature Difference. *The Engineering ToolBox.* [Online] [Cited: 09 04, 2013.] http://www.engineeringtoolbox.com/arithmetic-logarithmic-mean-temperature-d_436.html.

35. Specifying a liquid-liquid heat exchanger. *Cheresources.* [Online] [Cited: 09 05, 2013.] http://www.cheresources.com/content/articles/heat-transfer/specifying-a-liquid-liquid-heat-exchanger.

36. Heat exchanger nomenclature. *Standards of The Tubular Exchanger Manufacturers Association.* [Online] [Cited: 09 05, 2013.] http://standards.tema.org/images/HeatExchangerNomenclature.pdf.

37. Plate heat exchangers. *Heat exchangers.* [Online] [Cited: 09 09, 2013.] http://www.tranter.com/Pages/products/plate-heat-exchangers/description-benefits.aspx.

38. Suggested Compressed Air Flows in Standard Pipe Sizes. *myChemE.* [Online] [Cited: 10 07, 2013.] http://www.mycheme.com/suggested-compressed-air-flows-in-standard-pipe-sizes/.

# Pressure reduction valve and its performance

PROCESS DATA

| | | | | | |
|---|---|---|---|---|---|
| Pipe size inlet / outlet | mm | **250 / 250** | Wall thickness | Schedule | **40** |
| Valve duty | **Control** | | | Fluid nature | **Gas** |
| Description | **Air /** | | | | |
| Critical pressure | | | | | |
| Molecular weight | **28.969** | | Ratio of specific heats | **1.4** | |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Flow rate | kg/s | **1.8** | **2.5** | **1.6** | **1.6** |
| Upstream temperature | degC | **80** | **80** | **80** | **80** |
| Upstream pressure | kPaG | **10** | **180** | **250** | **300** |
| Differential pressure | kPa | **10** | **180** | **250** | **300** |
| Downstream pressure | kPaG | **0** | **0** | **0** | **0** |
| Compressibility | | **1** | **0.999** | **0.999** | **0.999** |

CALCULATED PERFORMANCE

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Capacity | FpCv | **761.81** | **234.18** | **120.01** | **105.06** |
| Percent of full travel | % | **65.8** | **30.8** | **17.4** | **15.4** |
| Opening in degrees | deg | **53.6** | **26.7** | **16.4** | **14.9** |
| Sound pressure level | dBA [IEC] | **63** | **89** | **87** | **87** |
| Flow velocity (outlet) | Mach | **0.15** | **0.21** | **0.13** | **0.13** |
| Pressure drop ratio (Xt) | | **0.51** | **0.57** | **0.57** | **0.57** |

VALVE SELECTION

| | | | | | |
|---|---|---|---|---|---|
| Nominal size mm | **200** | Maximum capacity FpCv | **1503.18** | Cv | **1610** |
| Valve type | **BUTTERFLY PN 10** | **BALANCED NELDISC, METAL SEATED BUTTERFLY VALVE, RATINGS ANSI 150, PN10-25** | | | |
| Valve serie | **S-L-ANSI150** | | | | |

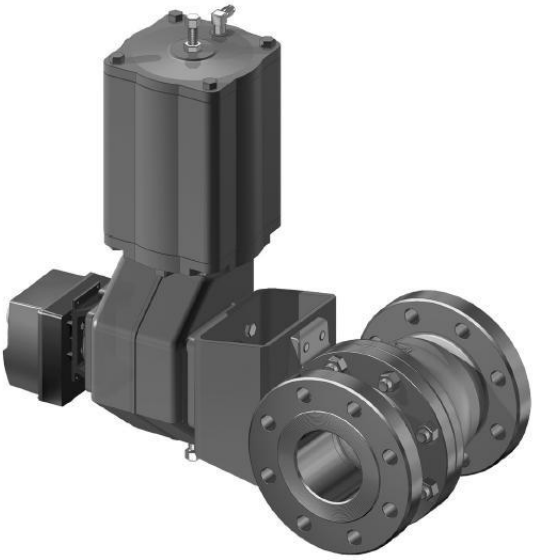
ACTUATOR SIZING RESULTS

Selected actuator **B1C13** **DOUBLE ACTING CYLINDER ACTUATOR**

| | | | | | |
|---|---|---|---|---|---|
| Required open | Nm | **94** | Required close | Nm | **300** |
| Opening load factor | % | **16** | Closing load factor | % | **52** |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Req control to open | Nm | **17** | **35** | **28** | **29** |
| Ctrl open load factor | % | **3** | **7** | **6** | **6** |
| Req control to close | Nm | **7** | **-11** | **-4** | **-5** |
| Ctrl close load factor | % | **1** | **ok** | **ok** | **ok** |

# Inlet manifold valve and its performance



PROCESS DATA

| | | | | | |
|---|---|---|---|---|---|
| Pipe size inlet / outlet | mm | **200 / 200** | Wall thickness | Schedule | **40** |
| Valve duty | **Control** | | | Fluid nature | **Gas** |
| Description | **Air /** | | | | |
| Critical pressure | | | | | |
| Molecular weight | **28.969** | | Ratio of specific heats | **1.4** | |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Flow rate | kg/s | **2.5** | **2.5** | | |
| Upstream temperature | degC | **25** | **25** | | |
| Upstream pressure | barA | **1** | **1** | | |
| Differential pressure | bar | **0.5** | **0.1** | | |
| Downstream pressure | barA | **0.5** | **0.9** | | |
| Compressibility | | **0.999** | **0.999** | | |

CALCULATED PERFORMANCE

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Capacity | FpCv | **595.82** | **1053.93** | | |
| Percent of full travel | % | **71.2** | **83** | | |
| Opening in degrees | deg | **66.6** | **76.2** | | |
| Sound pressure level | dBA [IEC] | **68** | **55** | | |
| Flow velocity (outlet) | Mach | **0.39** | **0.22** | | |
| Pressure drop ratio (Xt) | | **0.62** | **0.39** | | |

VALVE SELECTION

| | | | | | | |
|---|---|---|---|---|---|---|
| Nominal size | mm **200** | Maximum capacity | FpCv | **2330** | Cv | **2330** |
| Valve type | **BALL PN 10** | **FULL BORE BALL VALVE, SEAT SUPPORTED, WITH Q-TRIM FOR NOISE AND CAVITATION ABATEMENT, DIN RATINGS** | | | | |
| Valve serie | **Q-M1** | | | | | |

ACTUATOR SIZING RESULTS

| | | | | | |
|---|---|---|---|---|---|
| Selected actuator | **B1C25** | **DOUBLE ACTING CYLINDER ACTUATOR** | | | |
| Required open | Nm | **1715** | Required close | Nm | **1372** |
| Opening load factor | % | **69** | Closing load factor | % | **55** |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Req control to open | Nm | **270** | **178** | | |
| Ctrl open load factor | % | **10** | **7** | | |
| Req control to close | Nm | **140** | **135** | | |
| Ctrl close load factor | % | **5** | **5** | | |

# Heat exchanger control valve and its performance

PROCESS DATA

| | | | | | |
|---|---|---|---|---|---|
| Pipe size inlet / outlet | mm **80 / 80** | Wall thickness | | Schedule | **40** |
| Valve duty | **Control** | | | Fluid nature | **Water** |
| Description | | | | | |
| Critical pressure | barA **221.2** | | | | |
| Molecular weight | | Ratio of specific heats | | | |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Flow rate | l/s | **0.5** | **2** | **4** | **6** |
| Upstream temperature | degC | **10** | **10** | **10** | **10** |
| Upstream pressure | barG | **6** | **6** | **6** | **6** |
| Differential pressure | bar | **0.5** | **0.4** | **0.3** | **0.2** |
| Downstream pressure | barG | **5.5** | **5.6** | **5.7** | **5.8** |
| Vapour pressure | barA | **0.017** | **0.017** | **0.017** | **0.017** |

CALCULATED PERFORMANCE

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Capacity | FpCv | **2.95** | **13.18** | **30.43** | **55.9** |
| Percent of full travel | % | **13** | **34.5** | **53.8** | **73.9** |
| Opening in degrees | deg | **25.4** | **42.6** | **58** | **74.1** |
| Sound pressure level | dBA [IEC] | **<40** | **<40** | **<40** | **<40** |
| Flow velocity (inlet) | m/s | **0.25** | **1.02** | **2.04** | **3.06** |
| Terminal pressure drop | bar | **6.16** | **5.79** | **4.97** | **4.19** |
| Pressure recovery factor (Fl) | | **0.94** | **0.91** | **0.85** | **0.78** |

VALVE SELECTION

| | | | | | |
|---|---|---|---|---|---|
| Nominal size mm | **50** | Maximum capacity FpCv | **117.44** | Cv | **180** |
| Valve type | **SEGMENT PN 10** | | | | |
| Valve serie | **RA** | **ROTARY CONTROL VALVE, METAL SEATED, SEGMENT / V-PORT** | | | |

ACTUATOR SIZING RESULTS

Selected actuator **B1C6** **DOUBLE ACTING CYLINDER ACTUATOR**

| | | | | | |
|---|---|---|---|---|---|
| Required open | Nm | **6** | Required close | Nm | **6** |
| Opening load factor | % | **9** | Closing load factor | % | **9** |

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Req control to open | Nm | **4** | **4** | **4** | **4** |
| Ctrl open load factor | % | **6** | **6** | **5** | **5** |
| Req control to close | Nm | **4** | **4** | **4** | **4** |
| Ctrl close load factor | % | **6** | **6** | **5** | **5** |

# Heat exchanger specification sheet.

**HEAT EXCHANGER SPECIFICATION SHEET**

**SI Units**

| | | | | | |
|---|---|---|---|---|---|
| | | | Job No. | | |
| Customer | | | Reference No. | | |
| Address | | | Proposal No. | | |
| Plant Location | | | Date | 15.08.2013 | Rev |
| Service of Unit | | | Item No. | | |
| Size | DN 250 x approx. 1640 mm | Type BEM Horz. | Connected In | 1 Parallel | 1 Series |
| Surf/Unit (Gross/Eff) 21,15 / 21,05 m2 | | Shell/Unit 1 | Surf/Shell (Gross/Eff) 21,15 / 21,05 m2 | | |

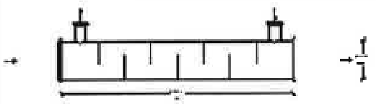
**PERFORMANCE OF ONE UNIT**

| | Unit | Shell Side | | Tube Side | |
|---|---|---|---|---|---|
| Fluid Allocation | | Shell Side | | Tube Side | |
| Fluid Name | | WATER | | Air 7000 Nm³/h | |
| Fluid Quantity, Total | kg/hr | 26831,7 | | 9051,06 | |
| Vapor (In/Out) | | | | 9051,06 | 8951,87 |
| Liquid | | 26831,7 | 26831,7 | | 99,1849 |
| Steam | | | | 181,021 | 81,8364 |
| Water | | 26831,7 | 26831,7 | | 99,1849 |
| Noncondensables | | | | 8870,04 | 8870,04 |
| Temperature (In/Out) | C | 10,00 | 23,00 | 160,00 | 30,00 |
| Specific Gravity | | 1,0003 | 0,9981 | | 0,9962 |
| Viscosity | mN-s/m2 | 1,3051 | 0,9318 | 0,0242 | 0,0188 V/L 0,7971 |
| Molecular Weight, Vapor | | | | | |
| Molecular Weight, Noncondensables | | | | | |
| Specific Heat | kJ/kg-C | 4,1944 | 4,1821 | 1,0476 | 1,0125 V/L 4,180 |
| Thermal Conductivity | W/m-C | 0,5823 | 0,6047 | 0,0359 | 0,0266 V/L 0,615 |
| Latent Heat | kJ/kg | | | 2392,64 | 2426,39 |
| Inlet Pressure | kPa | 400,006 | | 301,334 | |
| Velocity | m/s | 0,60 | | 31,86 | |
| Pressure Drop, Allow/Calc | kPa | | 16,947 | | 10,692 |
| Fouling Resistance (min) | m2-K/W | | | | |

| | | | | | |
|---|---|---|---|---|---|
| Heat Exchanged W | 405707 | | MTD (Corrected) | 49,2 C | |
| Transfer Rate, Service | 392,01 W/m2-K | Clean | 472,99 W/m2-K | Actual | 472,99 W/m2-K |

**CONSTRUCTION OF ONE SHELL**

| | | | Shell Side | Tube Side | Sketch (Bundle/Nozzle Orientation) |
|---|---|---|---|---|---|
| Design/Test Pressure | | kPaG | | | |
| Design Temperature | | C | | | |
| No Passes per Shell | | | 1 | 1 | |
| Corrosion Allowance | | mm | | | |
| Connections | In | mm | 1 @ 80,000 | 1 @ 250,000 | |
| Size & | Out | mm | 1 @ 80,000 | 1 @ 250,000 | |
| Rating | Intermediate | | @ | @ | |

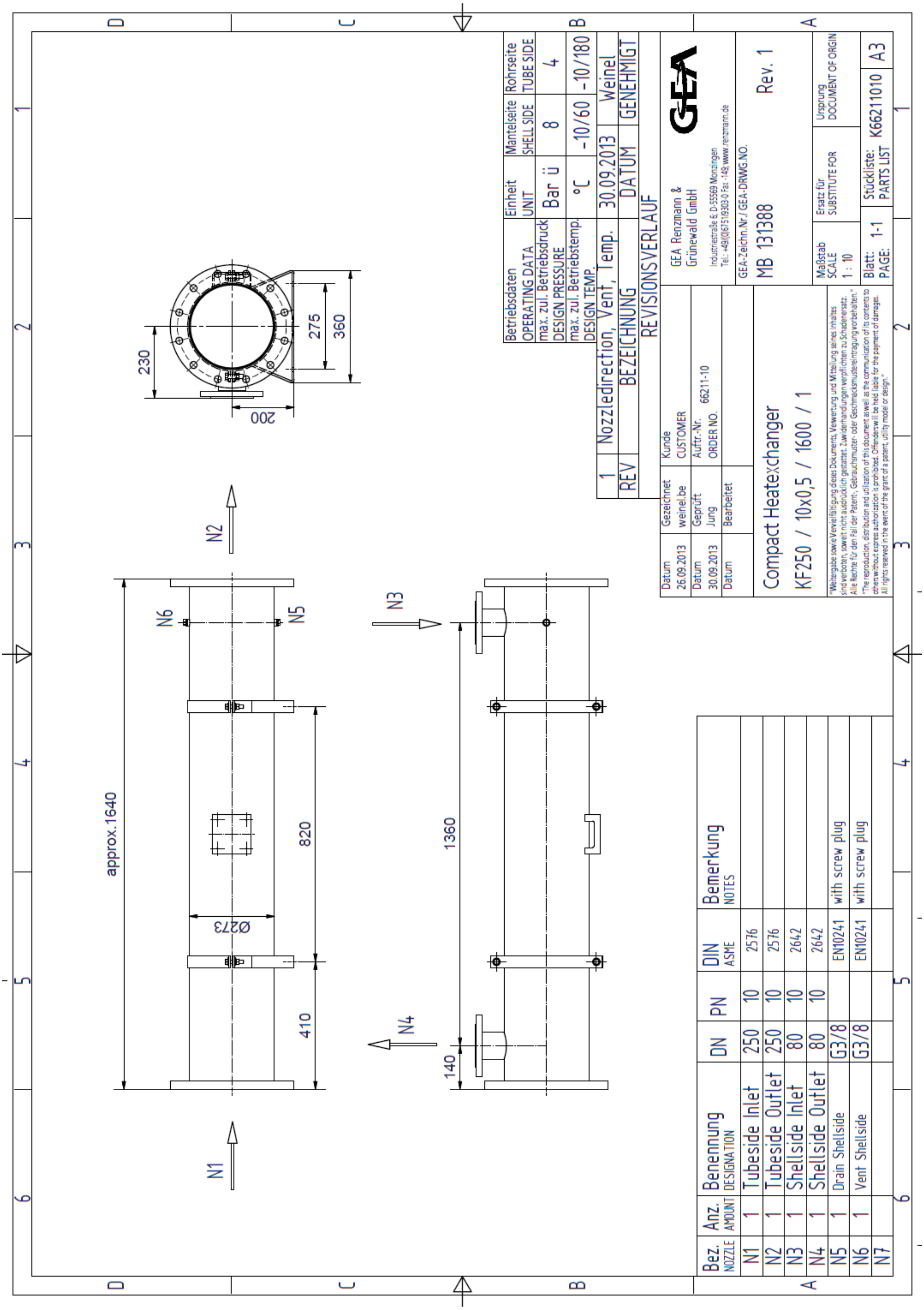


| | | | |
|---|---|---|---|
| Tube No. | OD 10,000 mm | Thk(Avg) 0,500 mm | Length 1,658 m |
| Tube Type | Plain | | |
| Shell | | OD 273,00 mm | |

Remarks:

Reprinted with Permission (v6 SP3)

# Heat exchanger drawings.



Appendix VI

# Surge tank drawings