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Commissioning of a Test Rig for Turbines and Compressors

TESI DI LAUREA MAGISTRALE IN
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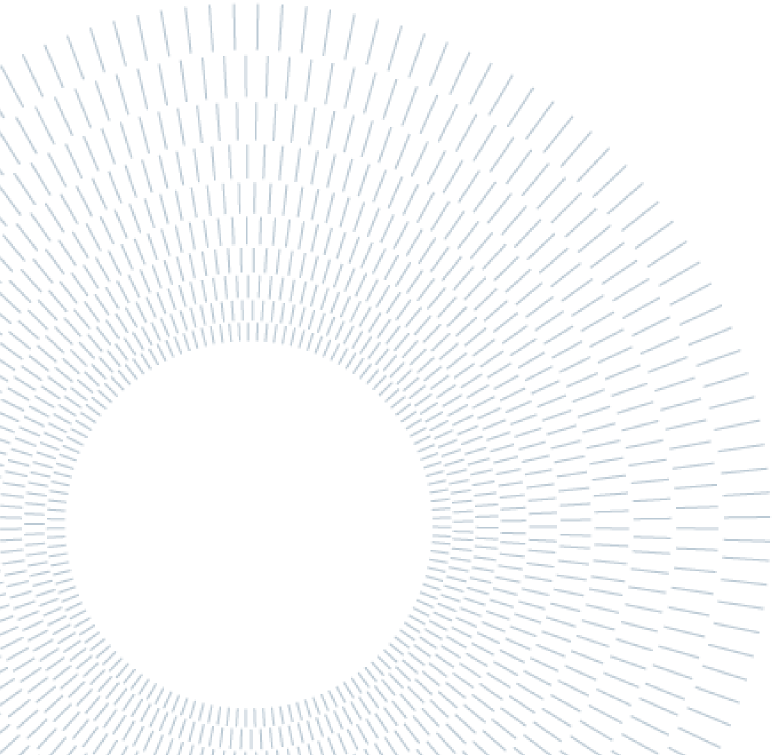


Abstract

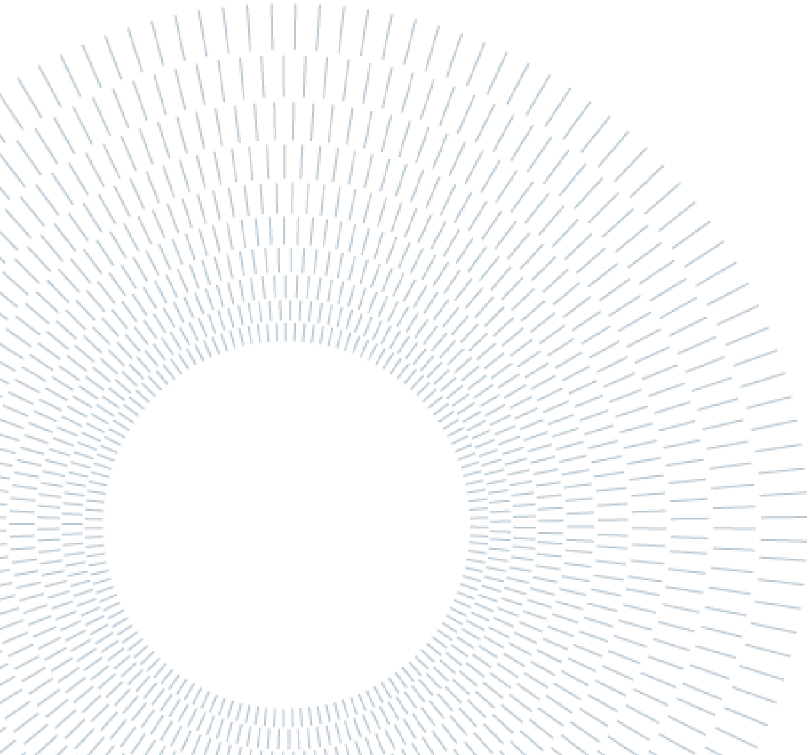
Project commissioning is the process of assuring that all systems and components of a plant are designed, installed, tested, operated, and maintained according to the operational requirements. This work describes the commissioning of the Closed Loop High Speed Test Rig situated in Laboratorio di Fluidodinamica della Macchine at Politecnico di Milano. The commissioning was necessary due to both an obsolescence of some components of the plant and also the different needs brought by a contract with Baker Hughes for the testing of a new model of steam turbine which makes use of nozzle guide vanes. A bypass was mounted of the plant in order to guarantee the correct amount of air flow rate into the turbine. The previously utilized rotor casing and rotor were dismounted. A new venturi tube was placed in between compressor and turbine to calculate the flow rate. It was then studied an approach to reduce the uncertainty by calibration using a sonic venturi. A monitoring hardware and software update has been performed to be consistent with the modifications applied to the plant. New transducers have been calibrated and then wired to the plant in order to account for the measurement of the new components. A confronted calibration has been done by PoliMi and Baker Hughes to get a referrable correction on the values gathered in LFM.

Key-words: Commissioning, Closed loop test rig, turbine, compressor, venturi, transducer.

Abstract in italiano

Il commissioning è un processo che consiste nel verificare che tutti i sistemi e componenti di un impianto siano progettati, installati, testati, messi in funzione e manutenuti secondo le necessità di funzionamento. Il seguente lavoro descrive il commissioning del Banco prova ad anello chiuso ad alta velocità sito nel Laboratorio di Fluidodinamica delle Macchine presso il Politecnico di Milano. Il commissioning si è reso necessario sia per la presenza di alcuni componenti obsoleti o fuori norma, sia per le condizioni imposte da un nuovo contratto con Baker Hughes per testare una turbina a vapore che utilizza la tecnologia NGV. È stato montato un condotto di bypass necessario ad evitare l'insorgere di problemi di pompaggio e a portare alla turbina la corretta portata di aria. È stato smontato il rotore testato in precedenza e anche la sua cassa. Un nuovo tubo di venturi è stato montato tra turbina e compressore per calcolare correttamente la portata di aria in ingresso alla turbina. Per diminuire l'incertezza del calcolo il tubo di venturi è stato tarato tramite l'utilizzo di un tubo di venturi sonico. È stato fatto un aggiornamento hardware e software al sistema di monitoraggio dell'impianto in conseguenza delle modifiche apportate. Alcuni nuovi trasduttori sono stati prima tarati e poi installati nell'impianto in modo da tenere sotto controllo i nuovi componenti. È stata svolta una taratura confrontata tra Baker Hughes ed il Politecnico di Milano in modo da ottenere delle correzioni sui valori misurati in LFM.

Parole chiave: Commissioning, Banco prova ad anello chiuso, turbina, compressore, tubo di venturi, trasduttore.



Contents

Introduction

Project commissioning is the process of assuring that all systems and components of a plant are designed, installed, tested, operated, and maintained according to the operational requirements.

A well-done commissioning of a test bench is crucial for the reliability, in all the fields, of the experimentations performed, such as safety, accuracy and comfort, together with a coherent life duration of all the mechanical and electronic parts involved in the testing. The aim of this work is the description of all the steps of the commissioning of the High Speed Test Bench in the Laboratory of Fluid-dynamics of Machine at Politecnico di Milano, in order to set up the plant to perform experimentations on a newly developed turbine designed by Baker Hughes.

The commissioning was necessary since the plant at different levels wasn't shaped to correctly perform the tests. There has been the need to build a bypass section after the compressor, to substitute the old classical venturi with a new one and to calibrate the new one, to change the bearings of the rotor, to change the installed shaft of the turbine gearbox and to substitute the rotor itself, together with his casing.

To monitor the new plant it has been necessary to modify and upgrade both the monitoring hardware and software, working also on the plant layout electronic-wise and transducers-wise. It was also performed a new calibration of all the devices used, and a venturi calibration program was written.

Some steps required a programming effort, whilst others were more of a handiwork.

1 High Speed Test Rig Layout before Start of Commissioning

This chapter contains a general overview of the turbine-compressor test bench as it was before commissioning started. It is also done brief illustration of the components present in the high speed test bench of the Department of Energy. It is an industrial plant designed in the 1980s for the experimental analysis of compressors and turbines of different nature, in order to determine the general operating characteristics or to conduct fluid dynamics researches in the three-dimensional field. It is important to note that, by exploiting the properties of the similarity, within certain ranges, it is possible to predict the operating characteristics of machines far larger than the ones mounted on the bench. To facilitate the understanding and explanation of how the TC bench is structured, the system has been dissected in several circuits and described one by one, although there are several interactions between each component. The first part to be described is the main air circuit, continuing with the oil circuit, the water circuit, and the air supply circuit. Moreover, a mention regarding the measurement tools has been given. The whole plant has been conceived to be used in the most versatile way in order to perform tests in different configurations without having to make substantial changes to the various parts of the plant. The plant can be seen from a top view in Figure 1.

Figure 1 - High Speed Test Bench at Politecnico di Milano

1.1. Main air circuit

The main air circuit coincides with the proper test circuit, where the turbomachines to be tested, are mounted. An important characteristic is that the working fluid (air in the studied case) is not renewed, which means that the plant is a closed loop system. Figure 2 shows a diagram of the main air circuit where the main elements are the centrifugal compressor and the axial turbine. The pipes colored in blue (representing the route taken by the working fluid) extend from the centrifugal compressor's volute to an air-water heat exchanger that cools the working fluid (S2). Actually, these pipes can be seen in Figure 1. Later on there is a venturimeter for flow measurement, two flow rate regulator valves, an air filter up to the turbine. The turbine discharge is connected to the compressor input thus forming a ring circuit. In the circuit there are also two bypass passages: the first at the compressor discharge, the other one at the turbine input. The closure or opening of these connections is achieved by means of flange valves with which it is possible to interrupt the flow of the fluid, making possible different flow configurations. At the first branch, if the by-pass derivation (VF1) is closed, the air flows completely into the bypass passage characterized by an electrically controlled throttle valve (EV1) and the presence of a second air-to-water heat exchanger (S1). In this configuration the pressure losses grow up considerably due to the sharp direction change and the heat exchanger component (S1). Therefore, in order not to overload the compressor, the VF1 valve is always open and the main disadvantage is not to fully exploit the S1 exchanger and the flow control throttle valve EV1. There are also 2 flanged plugs (TAF1, TAF2) in the system, which if opened can connect the test bench with lab wind tunnels.

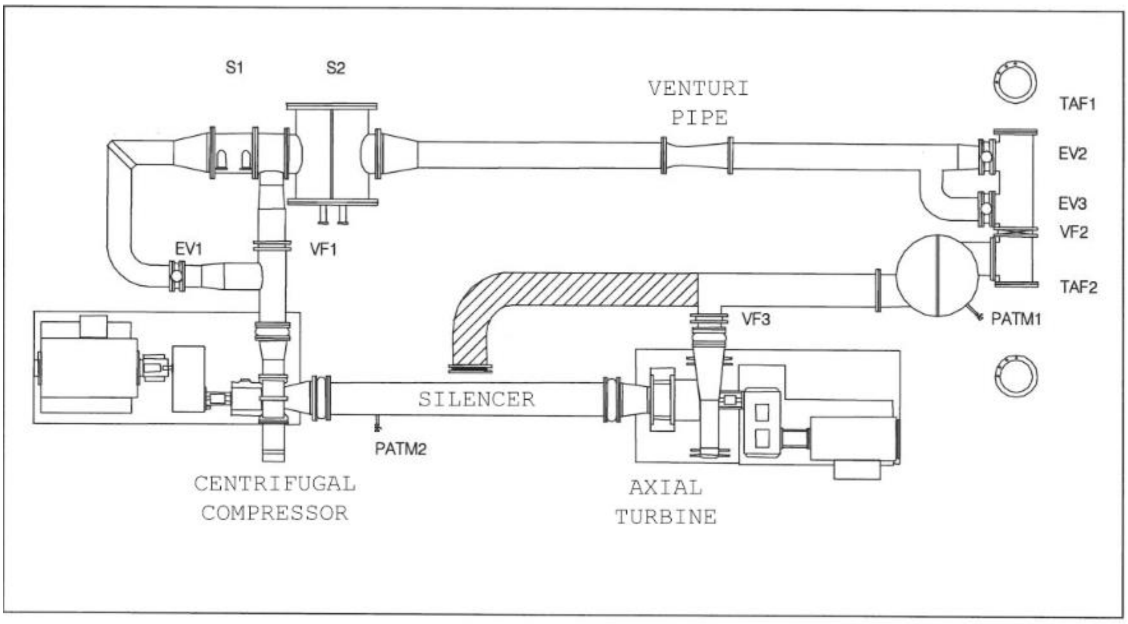


Figure 2 - Schematic Representation of the Test Bench

All the possible configurations achievable by the system are:

1. Centrifugal compressor and Axial turbine
2. Centrifugal compressor only
3. Axial turbine powered by the compressed air of the tank
4. Centrifugal compressor that powers the laboratory's galleries

Inside the plant two pressure taps are present (PATM1 and PATM2), linking the plant with the atmosphere. PATM1 is upstream the turbine, while PATM2 is downstream the turbine.

The opening of one or the other allows to have two different operating conditions:

1. Under pressurized turbine: if PATM1 is open
2. Pressurized turbine: if PATM2 is open

The best condition (from a plant issue point of view) is the second one. In fact, during previous tests, there was a strong oil leak through the seals between the turbine bearings and the air circuit. Oil drainage from the bearings is at ambient pressure and when the downstream air of the turbine is below ambient pressure, the oil tends to cross the seals and go into the main air circuit instead of going into its duct of discharge.

The silencer, Fig. 2.14, connects the two turbomachines and it has been specifically designed to damp the noise produced by the compressor that propagates upstream toward the turbine. This component, that has been placed to substitute a previously present duct, avoids noise propagation towards the turbine section thus allowing to get there correct acoustic measurements. This was the aim of the RECORD project, which focused on core noise reduction in propulsion systems for air transport.

Figure 3 - Silencer

1.2. Thermodynamic Cycle

The plant works with a non-renewed fluid, so it's actually a closed loop working with only one kind of fluid (air) always in gaseous state. The fluid can be assumed as perfect gas with constant specific heat. No chemical reactions in the circuit neither electromagnetic phenomena are present, so it is possible to identify two thermodynamic variables and fully determine the state of the system. The intensive quantities considered are the temperature and the pressure. In fact, since it is an experimental work, values in the different points of the cycle are not determined by theoretical notions, but they are measured appropriately. Figure 4 shows a simplified system diagram, while Figure 5 shows the thermodynamic cycle in the entropy-enthalpy plane.

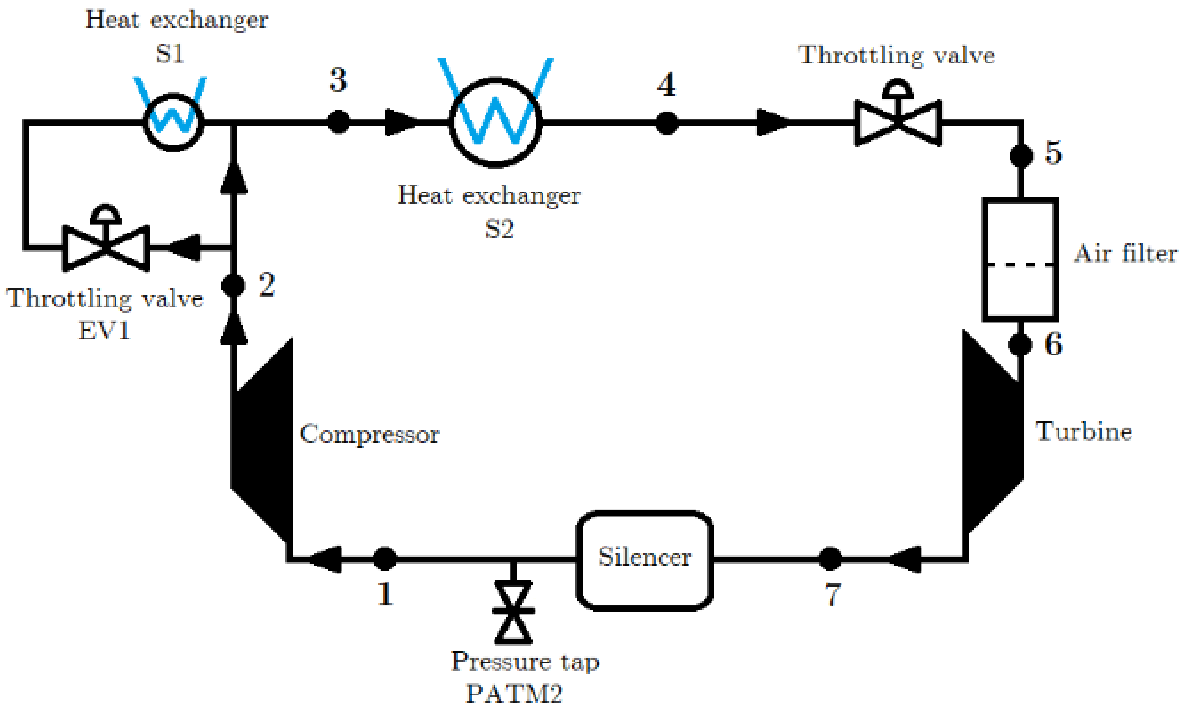


Figure 4 - Simplified HS test bench scheme

Point 1 is representative of the inlet of compressor, here air is at atmospheric pressure, thanks to the pressure tap PATM2. The outlet of compressor (point 2) can be defined if it's known the isentropic efficiency of the compressor. EV1 among point 2 and point 3 works as a throttling valve keeping constant the enthalpy level and forcing the pressure to go down. As air passes through the heat-exchanger (transformation 3 to 4), pressure losses occur again. EV2 and EV3 behave in the same way as EV1, increasing the entropy level at constant enthalpy, lowering static pressure. A further pressure reduction is caused by the filter block, resulting to the point 6 pressure; this is the value at which the expansion process starts. The transformation among point 6 and point 1 is not isentropic, since an ideal machine does not exist. The parameters imposed in the test are the static pressure at point 1 and the static temperature at point 4; this actually allows, once the control valves position is defined, to set the desired operating conditions acting only on the turbine and compressor rotational speed (the two degrees of freedom).

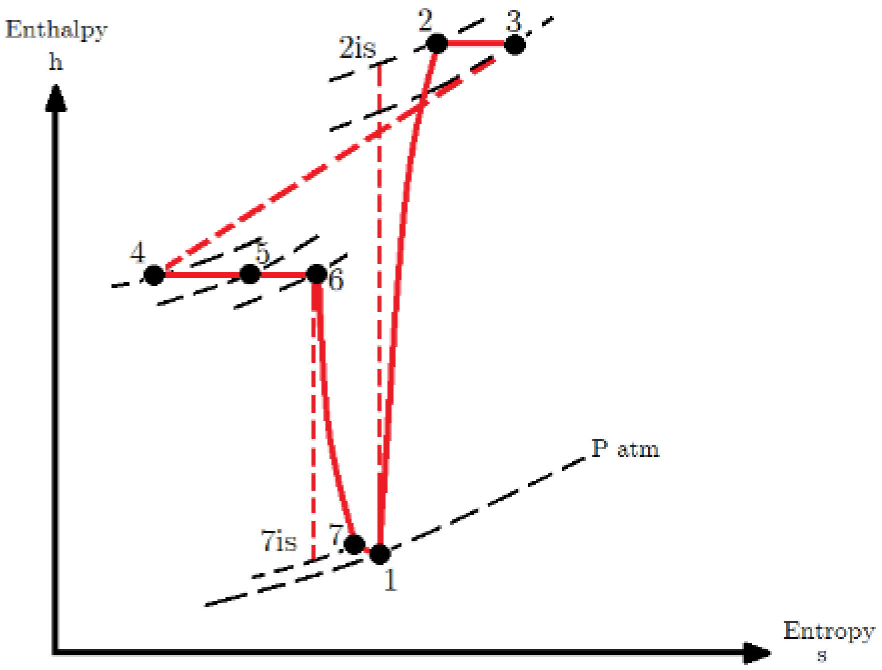


Figure 5 - Enthalpy vs Entropy Diagram of the Cycle

1.3. Centrifugal Compressor

This is a single-stage machine enclosed in a carbon steel case, the impeller is open and the direction of rotation, if viewed from the front, is clockwise. Figure 6 shows a picture of this machine in order to understand the dimensions. The impeller is the fundamental element to convert the mechanical energy supplied by the shaft into fluid-dynamic energy. The blades configuration is backwards with respect to the direction of rotation, an arrangement that imposes a negative geometric angle at output. The impeller is made of sixteen blades in an aluminum alloy, exhibiting an external diameter of 400 mm closed in a carbon steel volute. This component is mounted in cantilever fashion on a forged (alloy) steel shaft.

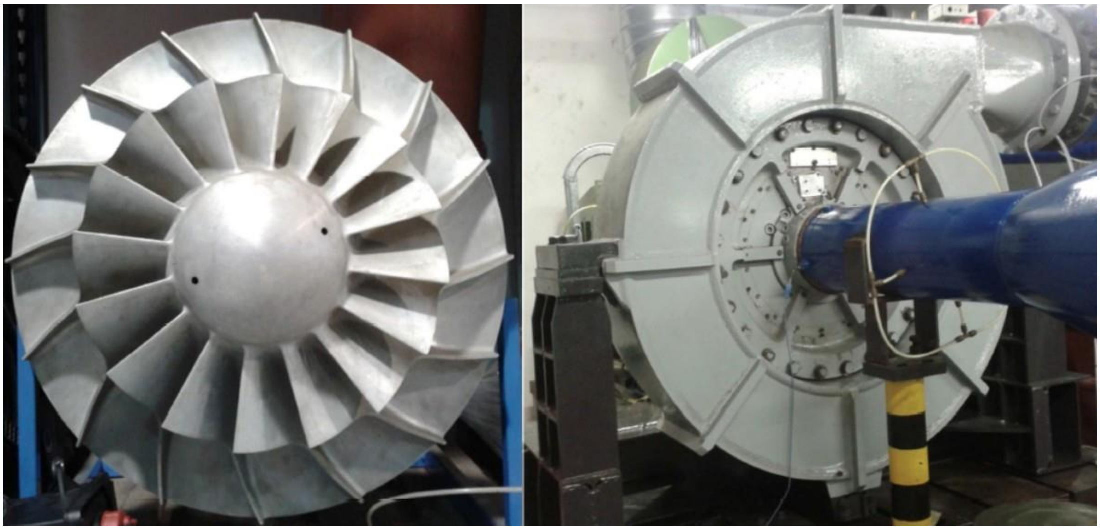


Figure 6 - Centrifugal Compressor, the impeller (left) and the whole turbomachine (right)

The machine is designed to reach up to a speed of 20000 rpm. The nominal speed of the machine is 18000 rpm, and as the permissible overspeed for a turbomachine is equal to 120% of the nominal speed, so the compressor can run up to a speed of about 21500 rpm. The vaned diffuser allows to direct well the flow and convert part of the kinetic energy into static pressure. The distance between the trailing edge of the impeller blades and the leading edge of the diffuser is 20 mm. The compressor is driven by an electric motor with a rated power of 800 kW and a 600 V DC. The rotational speed is manually controlled by a potentiometer. Among the electric engine and the compressor there is a four parallel axes gearbox with helical teeth gears of nominal power 1200 kW: this component allows us to vary the maximum impeller speed, depending on which shaft delivers the torque. Two possibilities are available: the first is to connect the slow output shaft with a maximum speed of 20000 rpm. The second is to connect the fast shaft (that can reach up to a 45,000 rpm rotation speed). The torque available from the electric engine is measured by a torquemeter between the gearbox and the impeller shaft. The features of the compressor are summarized in Table 1.

Impeller		**Vaned Diffuser**	
External Diameter [mm]	400	External Diameter [mm]	764
Blade Height [mm]	17.4	Blade Height [mm]	16.3
Geometric Angle at output [°]	-24	Number of Blades	19
Rotational Speed [rpm]	20000		
Number of Blades	16		

Table 1 - Centrifugal Compressor Features

1.4. Axial Turbine

The downstream section of the plant was equipped with a reaction axial turbine, shown in Figure 7, (reaction degree equal to 0.3) simulating a modern high pressure stage. The stage was characterized by 22 blades for the stator part and 25 blades for the rotor section. The blades' height was 50 mm, constant for all the stage. They were designed in order to get the best efficiency possible, trying to keep the angle of incidence constant across the entire blade height. The rotor rotates nominally at a speed of 12.000 rpm and its outer diameter coincides with the stator's one and it is 400 mm.

Figure 7 - The axial turbine

The shaft (a forged alloy steel) consisted of three parts: two ends (where there are tilting pad bearings) and a center where the turbine stages are present. The shaft was designed in order to make simpler the replacing of the central part, making possible the testing of different kinds of blades. The turbine is connected to a current generator, providing a braking torque and generating electrical power (to the electrical grid). Moreover, this generator can also act as a motor and rotate the turbine blades. Its nominal power is 400 kW and the supply voltage is 600 V. The gear reducer that connects the turbine and electric motor is 4 parallel axes reducer with helical wheels, with a nominal power of 1200 kW. Again, as in a centrifugal machine, it has two output speeds. The blade profiles and the main geometrical features of the machine are reported in **Error! Reference source not found.**.

TURBINE STAGE GEOMETRY						
	VANE			**ROTOR**		
Mean Diameter	350 mm			350 mm		
Span	50 mm			50 mm		
Blade Number	22			25		
Solidity (mid)	1.20			1.25		
Aspect ratio	0.83			0.91		
Blade turning	Hub	Mid	Tip	Hub	Mid	tip
	72.5	75.2	77.5	124.6	115.3	93.2

Table 2 - Turbine stage geometry

1.5. Oil Circuit

The oil circuit (colored in brown in Figure 1) is actually composed of a main circuit and two secondary ones. The first allows the lubrication of plain bearing and the speed multipliers gears, while the other ones are identical and guarantee the lubrication of the rolling bearings of the torsiometers.

1.5.1. Main Oil Circuit

This circuit is of great importance for the plant; in addition to lubricating, it has the function of subtracting heat from the components on which it operates. In the bench there are: 8 bearings in the two gearboxes (each containing 4 shafts); moreover, there are 2 additional radial and radial-axial thrust bearings both for the turbine and the compressor. The total number of the bearings is 20. Two tanks are present and their total capacity is about 4000 liters. They can be seen in Figure 8.

Figure 8 - Picture of the oil tank and the pumping system

Inside the tanks, there are resistances rising the oil temperature until 50◦C in the starting phase of the plant, in order to guarantee the optimal viscosity and chemical characteristics. During steady state operation, the oil temperature is controlled by means of water-oil heat exchangers. The oil circuit is the most critical for what concerns the plant survival. If the oil is missing in the bearing feed circuit, even for only a couple of seconds, there will be considerable damage to the components of the system. Very dangerous would be also the simultaneous shutdown of turbomachines and the oil circuit, because the inertia of the system implies a complete stop delay proportional to the speed of rotation of the machines. If the system does not feed oil to the bearings, they increase their temperature due to the lack of heat removal, resulting in metal-to-metal friction between the bearing and the rotor. The oil is collected in a tank, and then extracted through a primary oil pump (POP) and sent to heat exchangers. In addition to the POP, there are two other pumps parallel to the primary: the auxiliary oil pump (AOP) and the emergency oil pump (EOP). AOP is a safety pump that guarantees the correct working of the plant in case of POP breakdown and it has POP-like characteristics. They are both centrifugal pumps powered by three-phase asynchronous electric motors with a power of 4 kW at a rotation speed of 2900 rpm. Each pump generates prevalence equal to 35 m with a flow rate of 330 l/min. The emergency pump is switched on only if the electrical current in the plant is missing and therefore it allows the correct stop of the plant. The EOP is powered by a turbine driven by the service air at 7 bar pressure and has similar characteristics to the previous ones. The activation of the turbine takes place through the opening of a solenoid valve located before the turbine. Downstream of this valve there is a pressure switch that, if it identifies a pressure lower than 5 bar in the circuit of the compressed air it prevents the running of the plant. In order to maintain optimum viscous and chemical properties of the oil, it must work in a range among 30°C and 50°C and this is achieved, during the start of operation, by means of

15 kW resistors placed inside the collecting tank. The oil, before reaching the bearings, must be filtered in order to stop the solid particles, and so, to prevent them from reaching the bearings. Bearings are one of the most critical parts of the plant and the tolerances on the impurities imposed by the manufacturers are quite restrictive. The maximum particle size is equal to 25 µm. The filter is made up of a cylindrical structure inside which there are cylindrical metal meshes through which oil is forced to pass removing the impurities. The oil collected is divided and distributed partially towards the compressor area and partly to the turbine area. There are two shut-off valves at these branches to adjust or exclude the power supply. The oil then reaches the plain bearings and the various gears and is then conveyed through two large diameter pipes directly into the tank. The exhaust operation of the oil takes place tanks to gravity, meaning that the return pipes are slightly inclined downward to help oil drainage from the bearings.

A fume extraction system is installed over the tank, performing the following functions:

- expulsion of oil vapors, formed during the run of the test bench
- help the extraction of the air present in oil
- Depressurize the tank thus facilitating the ejection of oil from the bearings after it has been used

Main Oil Characteristics	
Kinematic viscosity at 40 $^\circ C[mm^2/s]$	46
Kinematic viscosity at 100 $^\circ C[mm^2/s]$	7.2
Flash point $[^\circ C]$	230
Density $[kg/m^3]$	860
Pour point $[^\circ C]$	-42

Table 3 - Main Circuit Oil Features

1.5.2. Oil Secondary Circuit

These are two circuits and they are actually specular. Their goal is lubricating the torquemeters rolling bearings. They are open type circuit, the oil is contained in a vessel, from which it is withdrawn through a suction system. It is mixed with air to form an emulsion by means of a nebulizer and sent to the torsiometer through plastic pipes. All the circuit is powered through the service air and there is a pressure reducer for precise flow regulation.

Torsiometer Oil	
Kinematic viscosity at 40 $^\circ C[mm^2/s]$	10.3
Kinematic viscosity at 100 $^\circ C[mm^2/s]$	2.6
Flash point $[^\circ C]$	166
Density $[kg/m^3]$	854
Pour point $[^\circ C]$	-15

Table 4 - Torsiometer Oil Features

1.6. Water circuit

The aim of the water circuit is basically one: cooling. The fluids to be cooled are the air out of the compressor and the oil before sending it to the lubricating organs. Since the temperature of the two fluids to be controlled is quite similar, water from the same circuit is used. The circuit is pressurized by a centrifugal pump powered by a 7.5 kW three-phase asynchronous electric motor with a flow rate of 1300 l/min and a prevalence of 30 m. The heat exchangers are the components needed to be reached by water and through a return line it is brought into the evaporative tower where the absorbed heat is dissipated. This circuit is not to be considered as critical as the oil circuit, because an abrupt abortion of the water circulation causes a rise in the temperature of the primary fluid, which results in a change of the test conditions, but not in a damage of the system. Sending oil at a temperature higher than the nominal one to the bearings is less severe than not sending them totally and it has been estimated that with a lack of water it is still possible to stop the test bench without damaging any component.

1.6.1. Heat Exchangers

The primary heat exchanger consists of a cylinder inside which there are several brass tubes inside which the process air flows. It is basically a cross-flow exchanger. Inside these tubes several aluminum finned profiles are placed to increase the heat transfer surface and consequently the effectiveness of convective exchange. Outside the tubes, the cooling water flows. The second heat exchanger, downstream of the primary, has lower efficiency with significantly higher losses. The two oil-water heat exchangers are countercurrent type, with a nominal power of about 78 kW. The primary and the secondary heat exchangers have the task of cooling the main circuit air until 30/60 ◦C, while the oil-water heat exchangers must cool the oil to 40 ◦C. To maintain the calibration temperature, the heat exchangers rely on special TIC valves, which vary the input water flow into the heat exchangers in order to exchange more or less heat. Valve handling is possible thanks to compressed air supply. The two heat exchangers can be seen in Figure 9.

Figure 9 - The two heat exchangers

1.6.2. Cooling Tower

The heat received from the water is dissipated by means of a forced draught cooling tower placed outside the building. This latter one receives hot water in its upper part where it is distributed uniformly through several nebulizers. Its nominal exchange power is 900 kW.

1.7. Air Secondary Circuit

Some components of the plant, like the auxiliary turbine that feed the EOP and others require pressurized air. The required air is taken from an external tank at the pressure of about 7 bar by means of the secondary air circuit. At each connection there is a self-cleaning air filter, to prevent any impurities go to the components and a pressure reducer that brings the air pressure at an optimal level for the component operating conditions. It’s important that the OEP is supplied with a pressurized air for the reasons already explained. The pressure level required is higher than 5 bar, if this condition does not exist anymore the plant is automatically stopped.

1.8. Control and Measurement Tools

The plant exploited is actually very complex, there different systems are provided.

- Electromechanical panel
- HS test bench software
- The operator

All these systems are linked to a series of measuring instruments, collecting the main thermodynamics and mechanical quantities. Here a brief list of these tools:

- Thermostats
- Thermocouples
- Pressure switches
- Manometers
- Pressure transducers
- Torque meters
- Proximity sensors

The physical quantities measured are temperature, pressure, vibration, speed and torque. The measurement tool changes according to the object on which the measurement is taken, to the degree of precision of the measurement and also from what use we want to make from the measurement. Figure 10 shows a diagram of the main measurements and control components and their various connections.

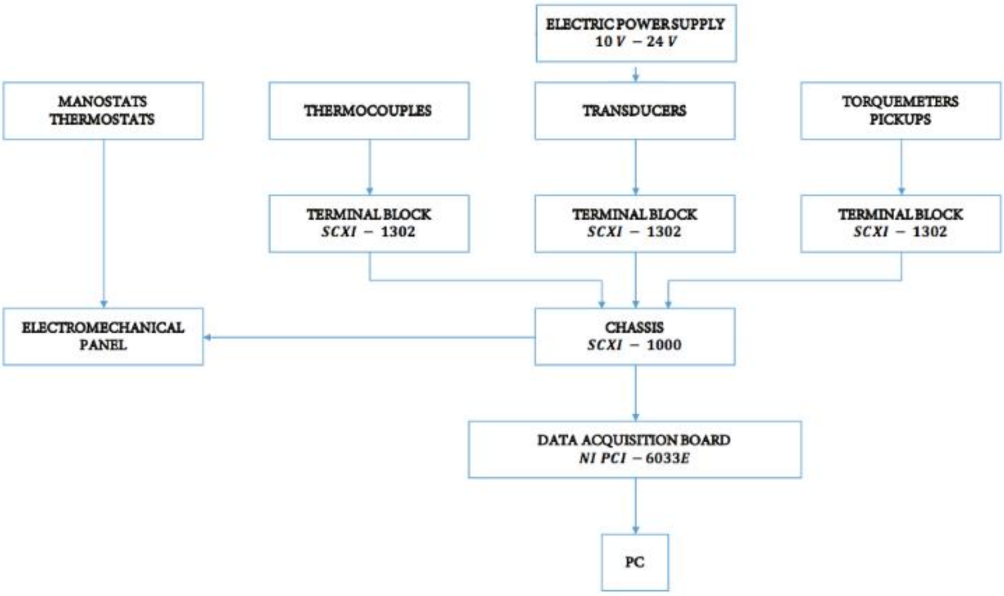


Figure 10 - Block scheme of data acquisition

1.8.1. Electromechanical Panel and Control Pulpit

All the operations involving the electromechanical devices in the test bench are allowed by the electromechanical panel. The panel is placed inside a closet having on the external side a series of light bulbs and push-button switches, which enable the various components of the system to be switched on or off. The system is powered with a 380V voltage from the electrical cable tray, which is then converted to 48V or 24V for the operation of the various relays, while on another line the tension remains equal to the original value of 380V for power supply (pumps, resistors, etc.). Signals from the pressure switches and thermostats of the system arrive at the panel; these signals activate the various relays that close or open the various electromechanical circuits according on the logic present. Connected to the electromechanical control panel there is also the control pulpit: it allows the operator to switch on the turbine and the compressor engines and also if the turbine acts as a generator or as a motor. On the control pulpit there are two potentiometers needed to regulate the speed of rotation of the two machines and the analog indicators showing us the value of the current circulating in the armature of each motor and the speed of rotation of the shafts. This pulpit can be seen is Figure 11.

Figure 11 - Control Pulpit of the Test Bench

1.8.2. Temperature Measurements

Temperature is one of the fundamental physical quantities for the analysis and the main tools exploited are the thermocouples. Thermocouples are used because they are reliable, accurate and relative cheap and they are used in different parts of the plant: for measurements of the air in the main circuit, of the oil and of the water. The physical principle of thermocouples is very simple: when two ends of different metals are connected and the two joints are brought to different temperatures, an electromotive force in the circuit is generated proportional to the temperature difference among the ends.

All voltages from thermocouples are instantly processed and displayed on monitors for each acquisition cycle in order to have an overview of the thermodynamic states of the parts. Type T (copper-constantan) thermocouples are used in the system (as in Figure 12). The only weak point in thermocouples is their extreme delicacy but they are suited for measurements in the -200 to +400 ◦C range.

Figure 12 - Type T (copper-constantan) thermocouple

1.8.3. Pressure Measurement

Pressure is also a fundamental physical quantity, both for the control of the plant and for the experimental phase. Most of the sensors in the TC test bench are piezoresistive Kulite transducers (example in Figure 13), which measure the change of the electrical resistance value due to the application of a mechanical tension. In fact, these sensors exploit the piezo resistivity characteristic of the silicon, that actually varies its electrical resistance if a mechanical stress is applied on the material. From a constructive point of view, the membrane in silicon is inserted into an electrical circuit able to detect a resistance variation. In particular, one side of the membrane is placed in contact with the fluid whose pressure must be measured, while the other side is in contact with the atmosphere. A relative pressure is measured. A continuous voltage in input must be provided for the correct operation and a system that processes the output signal to obtain the physical data. In the system it is also necessary to measure a difference of pressure between two distinct sections: in the venturimeter the pressure delta between the inlet and the throat is measured for the calculation of the flow rate. Moreover, in the oil critical points, some manometers are installed in order to have an instantaneous value of pressure. In addition, there's also a differential manometer on the oil filter in order to detect dirt's presence possible causing a stoppage of oil circulation.

Figure 13 - Picture of a Kulite type pressure transducer

1.8.4. Vibrations Measurements

A Bentley Nevada is the vibration measurement tool, it consists of proximity sensors, signal conditioners and analogic monitors. On each machine one axial sensor and six radial sensors are present, in particular two sensors phased at 90° for each bearing in order to obtain the mean position of the shaft's center with respect to the bearing.

The proximity sensors are based on the eddy current, a coil is powered by an alternate electric current producing an electromagnetic field and consequently an eddy current on the shaft's surface. The intensity of the current is function of the distance among sensor and shaft, so a measure can be estimated.

1.8.5. Rotational Speed Measurements

The number of revolutions of the compressor and the turbine is measured by mean of a sensor able to detect a magnetic field variation caused by a key installed on the external circumference of the shaft under examination. The reference signal ideally passes from 10 V to 0 V. By counting the generated peaks of this square wave over a given time interval, it is possible to determine the angular velocity rotating the rotor.

$$\omega = \frac{2\pi}{60} * \frac{n}{T}$$

where T is the sampling period and n is the number of measured peaks in the period. The sampling frequency has to be high due to the very small time among one peak and another.

1.8.6. Torque Measurements

The measurement of the torque is fundamental, because if both the torque and the rotational speed are known the power at the shaft is simply measured by means of the formula

$$P = T * \omega$$

The torque T is measured by means of two components, the Torquemeter and the Signal elaboration and conditioning module. The torquemeter works by exploiting the principle of variation of the inductive field generated between two coils, whose relative position varies according to the deformation of the shaft's surface. This relative rotation is the function of the torque transmitted for the elasticity of the material of the shaft. An appropriate electrical circuit elaborates all the data and send the evaluated torque value to the acquisition board.

1.8.7. Data Acquisition System

The data acquisition system displays to the operator the main thermodynamic quantities in order to control and check if the correct operation of the test bench is reached. It works in parallel with the electromechanical panel, to ensure a greater security.

The acquisition system consists of:

- The various electrical supplies of the pressure transducers. They are AC-DC converters delivering as an output a given voltage value; this value is chosen based on the maximum limits imposed by the transducers. There are 2 power supplies: one with 10 V output and one 24 V.
- The data acquisition board with its components have the task of measuring output quantities from instruments in the form of continuous mV variations and transmitting them to the software, converted into a digital bit signal. The acquisition board is a National Instruments PCI-6033E and it is connected to a signal conditioning platform, the SCXI, through which the information is sent to the data acquisition board.
- A PC, with a LabVIEW software and the Measurement & Automation Explorer tool for controlling the various channels.

In order to have a video reading and to save the various values during the plant runs, a LabVIEW program was implemented in the computer. This program has as main file MainBANCOTC.vi containing the various subroutines:

- impianto.vi allows to choose which operating configuration must be implemented for the correct condition of the plant; moreover, it imposes the limit value of the turbine’s rotational speed.
- zeri.vi able to define the zeros of the pressure transducers.
- monitoraggio.vi that displays all the quantities needed to be shown for the monitoring of the plant; moreover, it displays possible alerts and contains the controllers to start and stop acquisition. The program automatically creates a historical file in which all values of the acquired quantities are stored in a 2-second interval.

Features of NI PCI-6033E	
Number of channels	64
Resolutions *bit*	16
Maximum acquisition frequency [*kHz*]	100
Input Range [*V*]	±0.1 → ±10
Digital *I/O*	8

Table 5 - Data acquisition system features

2 General Layout Modifications

As presented above, the study described consists in modifying the High Speed Test Rig in order to grant correct and accurate tests on a Baker Hughes's turbine. In this page it is presented in general every modification needed.

2.1. Bypass

The first layout change of the plant is the build up of a bypass duct right after the compressor, since the Baker Hughes's turbine has half the flow rate of the previously tested turbine. In this way the right amount of flow rate flows through the turbine and we do not have risks of surge, which consists in violent air flow oscillating in the axial direction of a compressor, which indicates that the axial component of fluid velocity varies periodically and may even become negative. The bypass duct has been designed by Professor Alessandro Mora and Professor Paolo Gaetani, and has been mounted by Cati S.p.a. It consists in a T duct downstream of the compressor which goes by one side to the main heat exchangers and then to the turbine, and by the other side goes to a secondary heat exchanger which cools down the air to a temperature value close to the inlet of the compressor. Downstream of the bypass heat exchanger is mounted a venturi nozzle to calculate the flow rate of air. Then two valves are mounted in parallel in order to have greater control and accuracy on the air flowing through the bypass, and then there in another long duct that brings the bypassed air behind the pressure tap (PATM2) in order to put also the pressure level as the same of the inlet of compressor.

2.2. Classical Venturi

Another important layout change is the classical venturi placed in between compressor and turbine in order to calculate the air flow rate passing through the turbine, crucial to understand the performances of the turbomachine. The previously installed venturi tube was not following any standards, so it was necessary, in order to have the precision Baker Hughes required, to design and install a normed venturi tube. Also, to increase more the accuracy of the value of the flow rate, the classical venturi is calibrated through a sonic venturi placed downstream the classical one.

2.3. Gearbox and Torsiometers

Due to the characteristics of the turbine, it is necessary to mount a different shaft among the four of the gearbox. Until now it was mounted the 20000 rpm shaft, for the Baker Hughes's turbine it is necessary to mount the 11000 rpm one. Also, since the tests are going to be developed on the turbine only, the compressor's torsiometer is going to be removed since it was causing rotordynamic instabilities, probably due to some damaged bearing. The turbine's torsiometer instead is going to be substituted by a new one with higher accuracy.

2.4. Rotor

The last big element to be changed is of course the rotor. All the work that has been done is to perform tests on the new turbine, so the rotor must change. First of all, we dismounted the previous casing of the rotor together with the rotor, so that it is possible to mount the new casing and the new rotor.

3 Detailed Layout Modifications

In this chapter are going to be analysed all the procedure needed to perform the layout modifications, together with the procedure that came consequentially of the modifications.

3.1. Rotor Dismount

The rotor dismount was performed in different phases, since the casing of the turbine is made of different parts and has a unique dismounting pattern to be followed. First of all, it is necessary to remove the part that link the turbine casing with the silencer, which is a duct placed in order to study the flow downstream of the turbine, and that's done by unscrewing all the bolts which are on the silencer side and on the rotor casing side, and removing the part, visible in Figure 14, thanks to a crane present in the LFM.

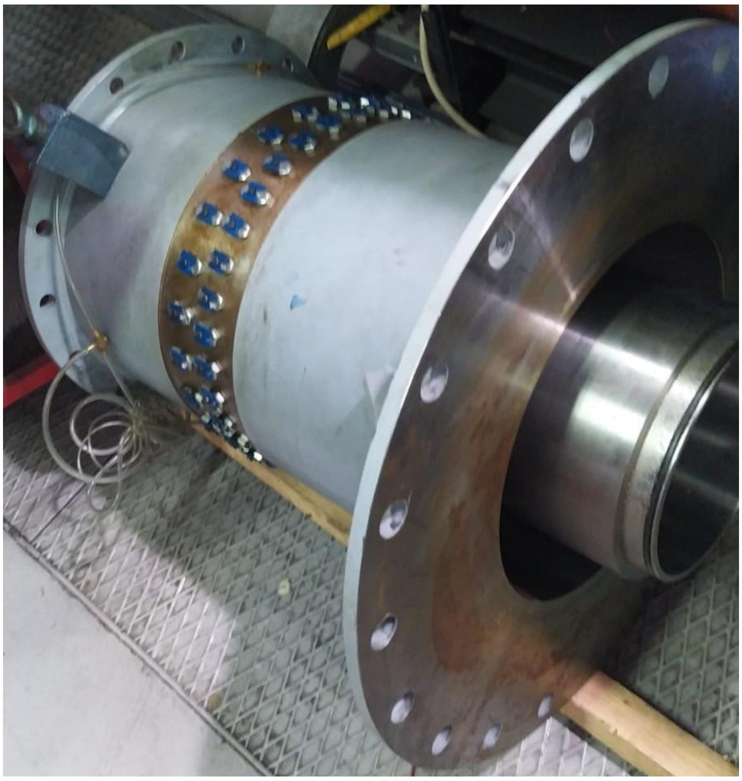


Figure 14 - Duct for downstream flow analysis

Then there is the casing itself which is designed to have two pieces, which are made of a top part and a bottom part, half a cylinder each, shown in Figure 15.

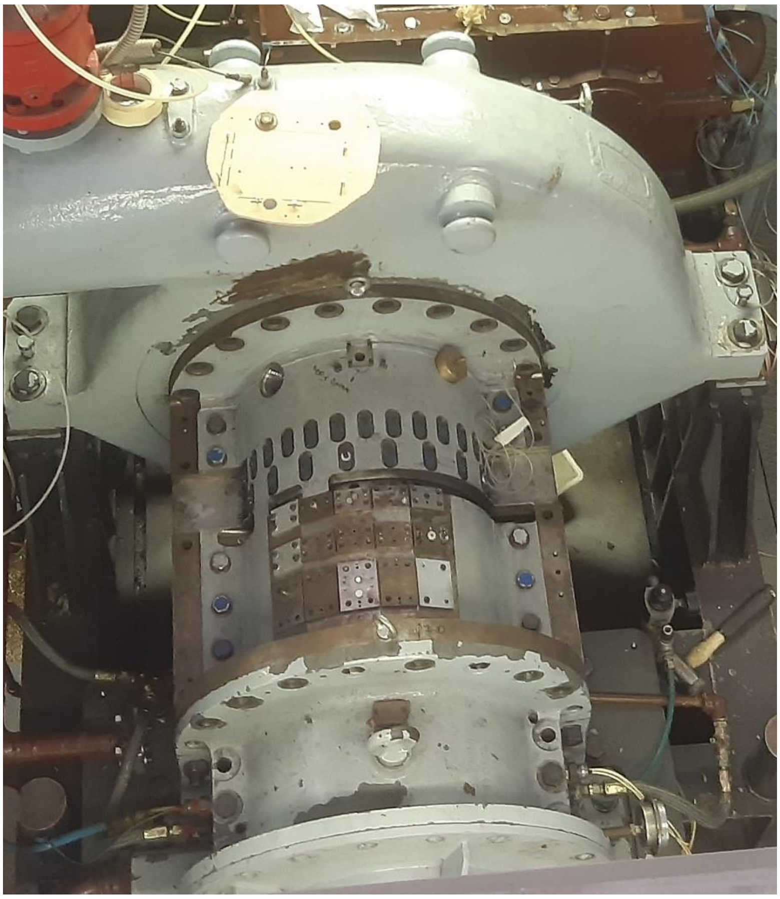


Figure 15 - The two parts of the rotor's casing

The first part to be dismounted is the one on the silencer’s side, in which is placed one of the two bearings, since at the other side there’s the volute, which is a full piece, and so it is not possible to let the rotor exit from that side. After the dismount of the top part of the first piece, together with all the transducers inserted in that section, like the vibrations sensors, it is possible to also dismount the top part of the other piece, in order to have the whole rotor uncovered. Now it is necessary to remove the stator, which has an upper and lower part which are bonded together, so first the junctions are removed and then the two parts can be removed. Also, between the rotor and the volute there is a honeycomb, which in a section used to break the swirl coming from the volute and let the fluid enter as axial as possible. Stator and honeycomb are shown in Figure 16.

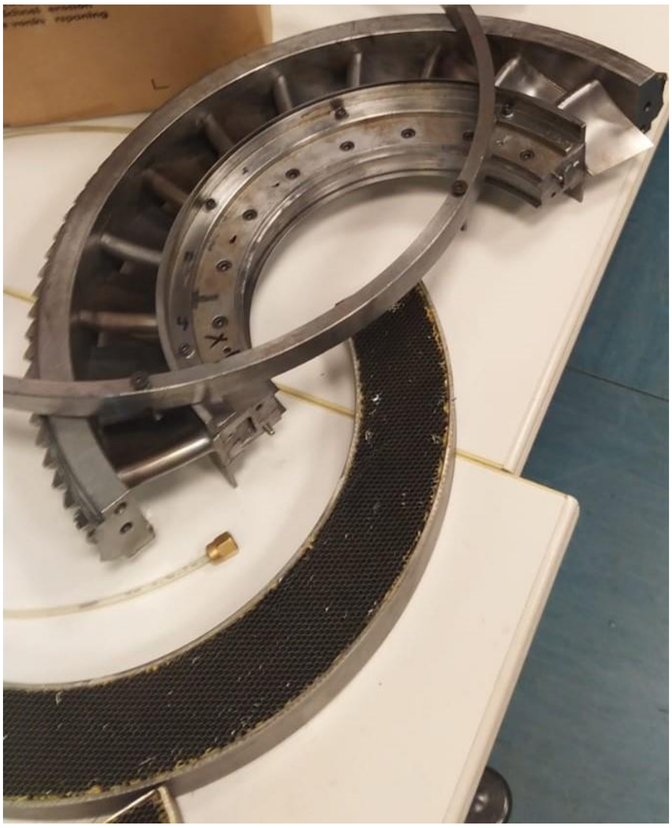


Figure 16 - Stator and Honeycomb

Then in is necessary to remove the bottom part of the silencer side piece, in order to let the rotor slide out of the volute horizontally without obstacles, since if the bearing is at its place the rotor won’t move, as slightly visible from Figure 7. To do so it has been implemented a scaffolding which is designed to keep up the rotor without the need of bearings. It is composed by two long bars that are screwed in the threaded holes of the bolts that links the volute and the rotor casing; the two bars have themselves two threaded holes each in order to screw in them two little bars of regulable length, so that they are able to support the long bars by unloading the rotor’s weight on the bottom part of the casing as shown in Figure 17. In the end, there are two traverses that can slide inside the two long bars, to which the rotor is fixed thanks to ropes and hoists.

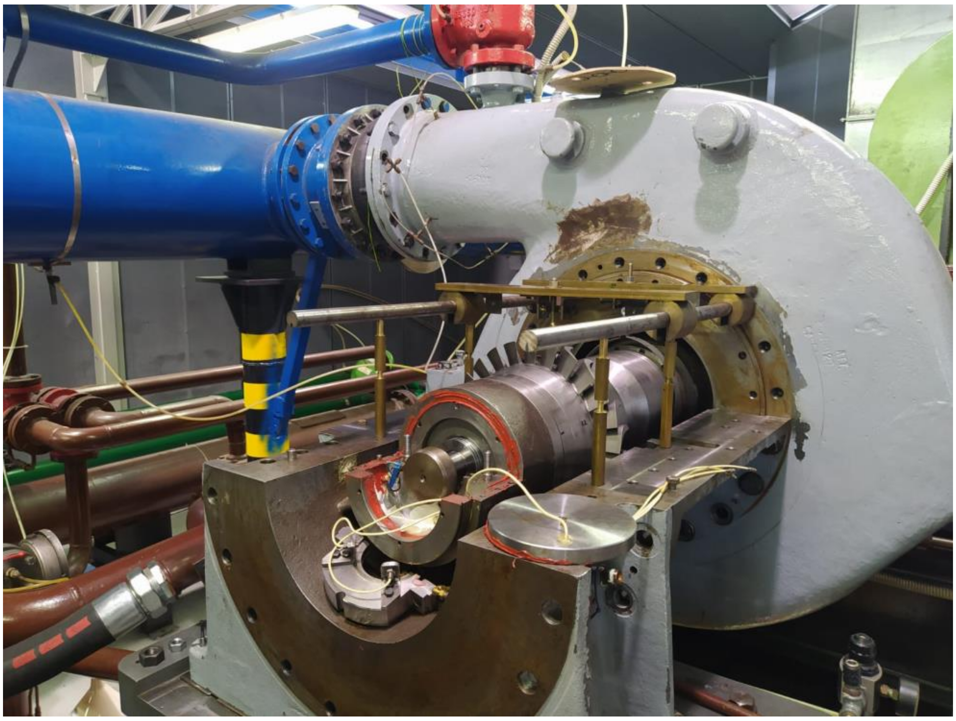


Figure 17 - The scaffolding

Once the rotor is completely hung by the scaffolding, it is possible to remove the bearing and the bottom part that was housing the bearing. In this way is then possible to take out the rotor, visible in Figure 18 from the volute by making it move horizontally, and then putting it away thanks to the crane.

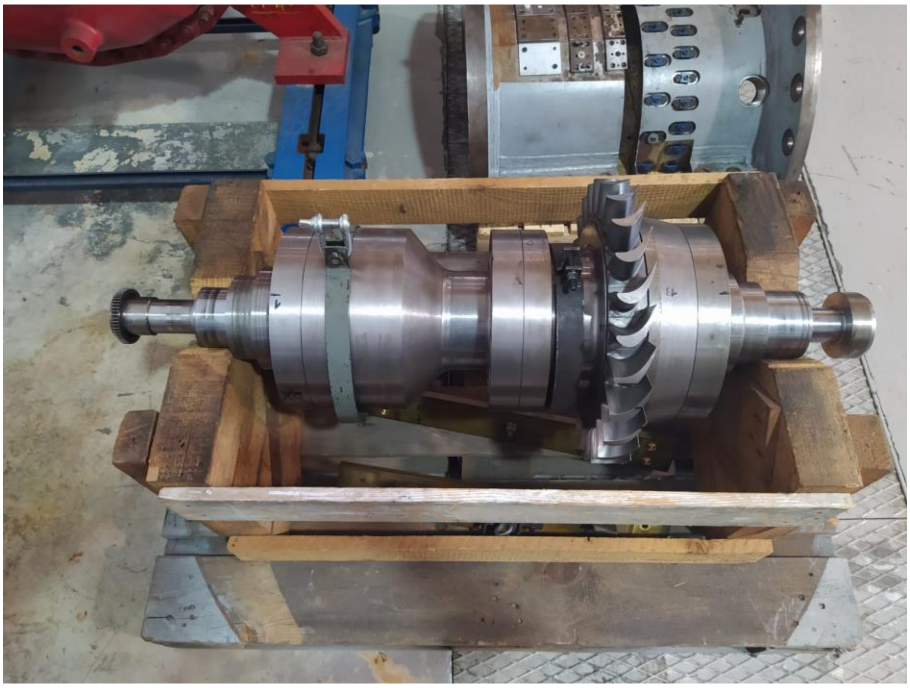


Figure 18 - Dismounted rotor

Now it is just possible to remove the bottom part of the casing. The whole old casing will not be utilized anymore, since a new one is ready to be mounted together with the rotor.

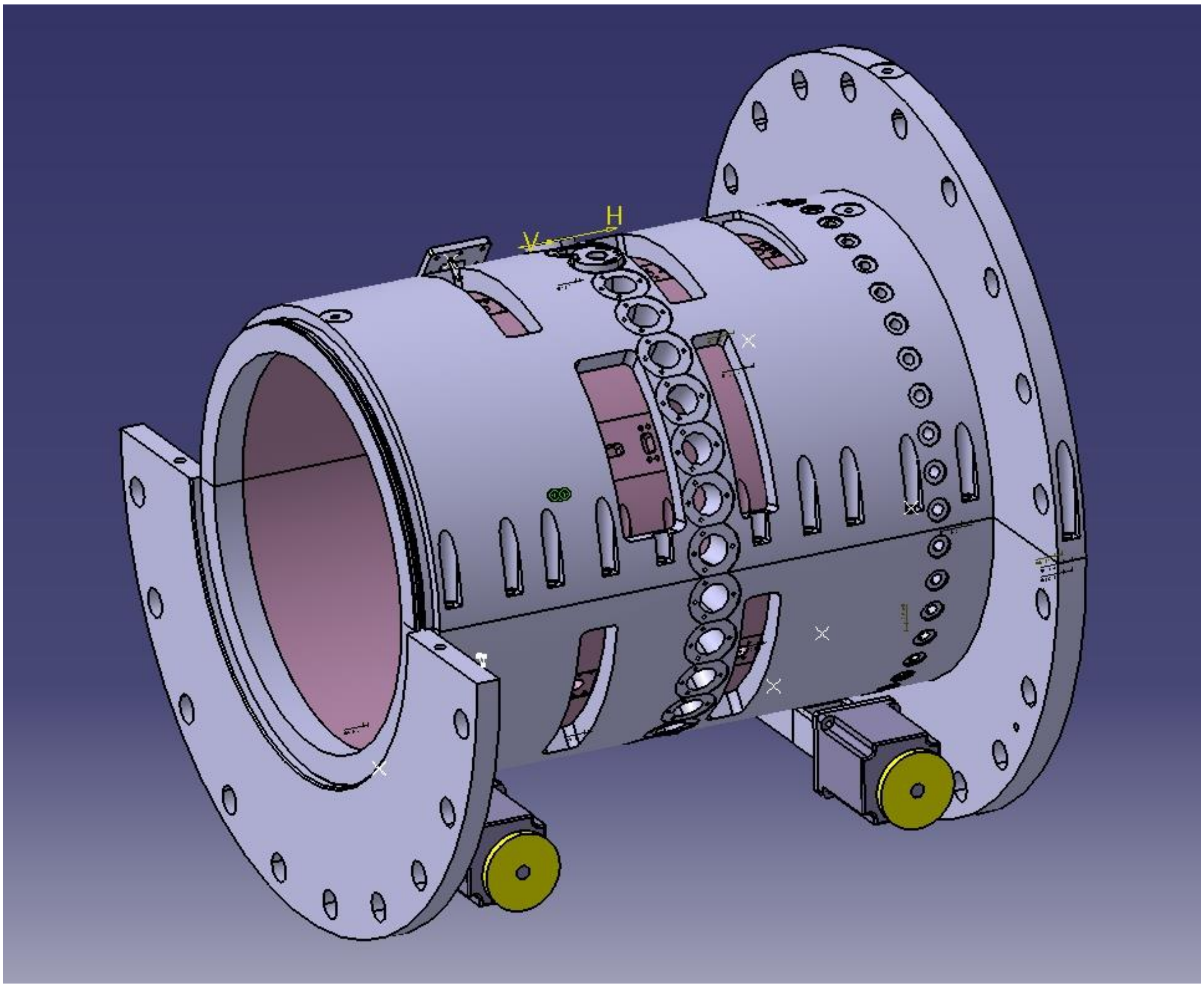


Figure 19 – New rotor casing

In Figure 19 it is possible to notice underneath the casing two motors. These motors are the ones controlling the nozzle guide vanes which are going to be used to increase the turbine efficiency.

Once the dismount was completed, the bearings were analyzed and some black stain was found on the pads of both the bearings. To be sure that the spot wasn't dirt or else they were cleaned up with alcohol. Looking at the pads through the microscope it was found that there was a groove in the pads, due to the removal of the white metal layer through the friction between bearing and rotor, visible in Figure 20. The only possible decision was to buy a new pair of bearings for the rotor.

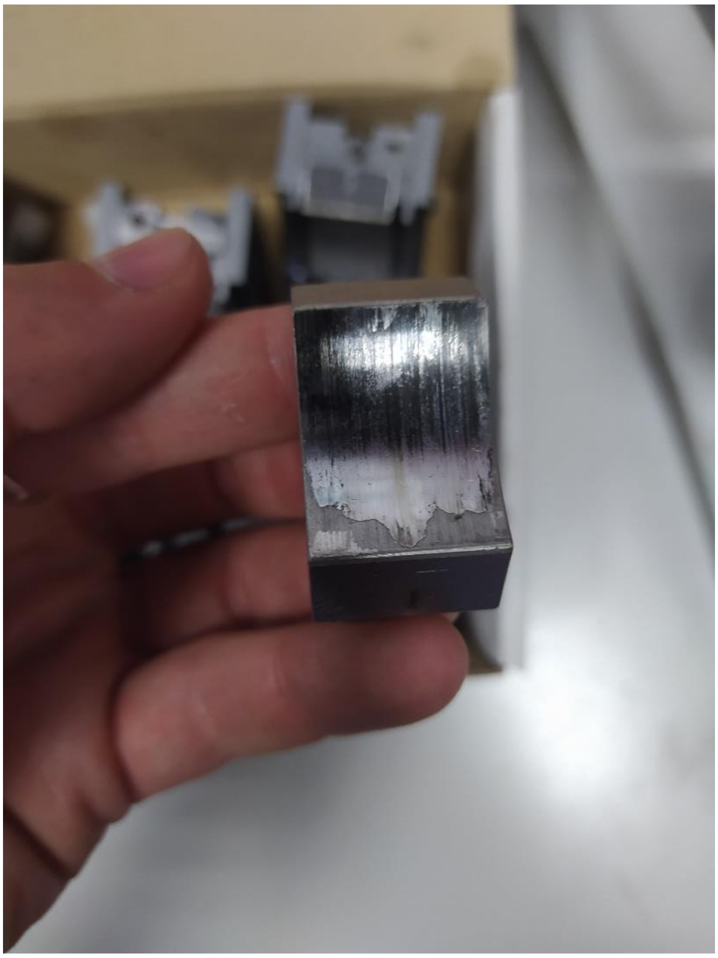


Figure 20 - A damaged pad of the axial bearing

3.2. Bypass Mounting

The mounting of the bypass has surely been the heaviest job amongst all, in fact it has been done by professional welders and assemblers. The first step was to remove the previous bypass, which, as described in the first chapter, was meant to increase the heat exchanged by the compressed flow to the water circuit. We said that the secondary heat exchanger wasn't performing good, so it was necessary to change it, since now it is not anymore in parallel with the main one, but it is working all alone on the cooling of the flow that is going to enter again in the compressor.

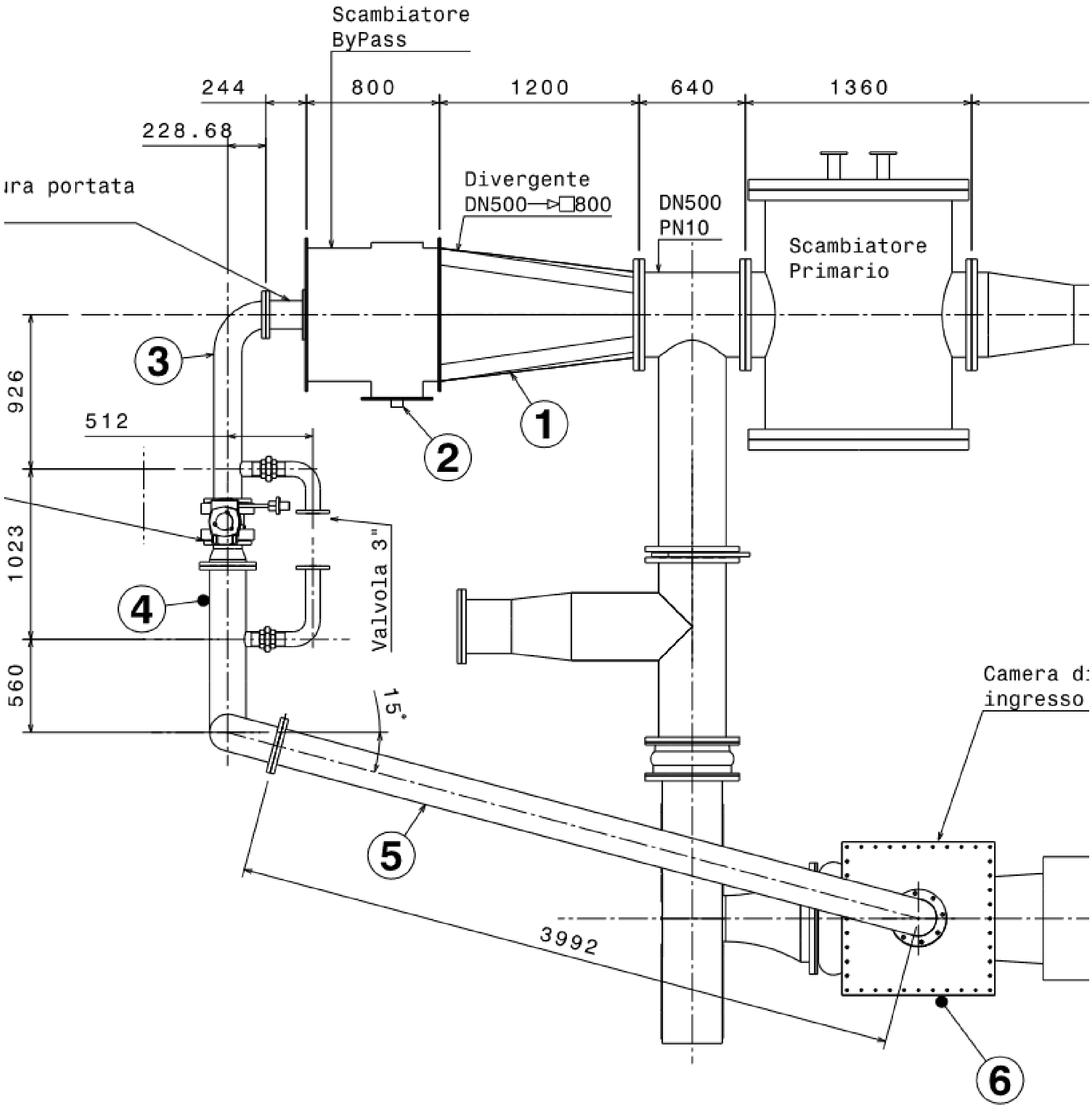


Figure 21 - Drawing of the Bypass

The T duct that started the bypass (in Figure 21 theDN500 PN10) was made on the measure of a special flange of the previous heat exchanger, and so in the assembly stage, since the flange was special, which means that it doesn’t follow any standard, there was no matching between the diameters of the T duct and the one of the convergent part, visible in Figure 22. Many solutions were discussed to fix the issue, and in the end, since it was found a flange of the right diameter, that flange was welded on the convergent duct.

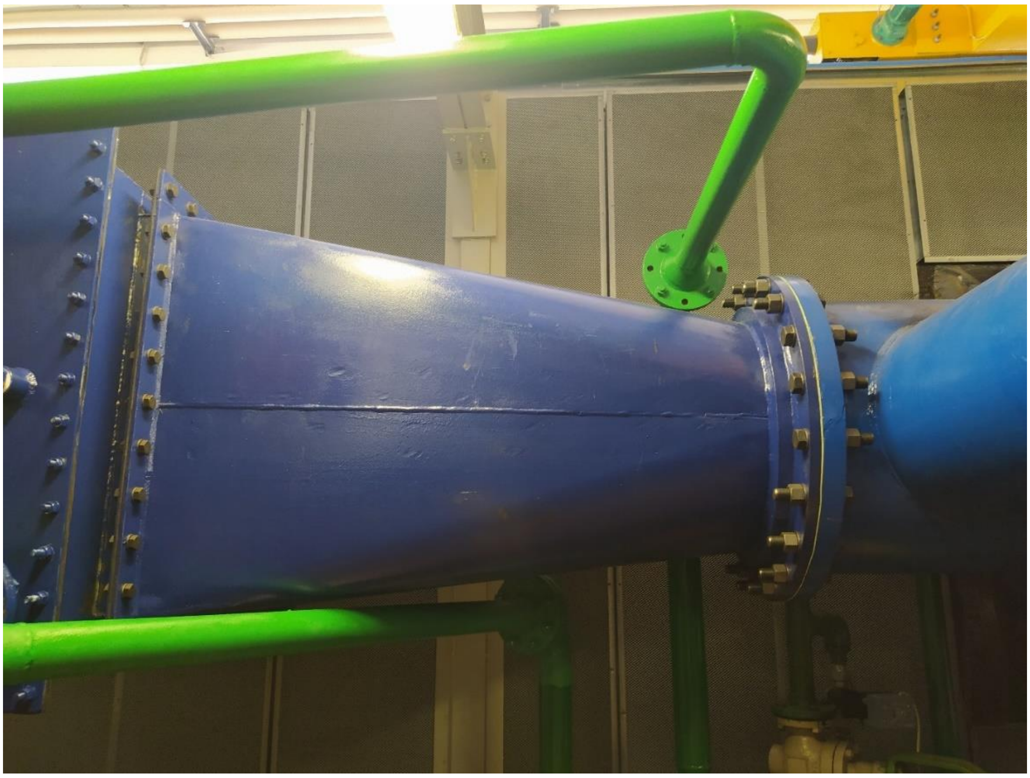


Figure 22 - Convergent duct

After, while mounting the heat exchanger's casing and putting in the heat exchanger itself, it was found out that the two plugs for the hydraulic test weren't fitting in the case due, as visible in Figure 23 to a too small tolerance came out of the manufacturing procedure which made impossible to put all the heat exchanger into its casing.

Figure 23 - Heat Exchanger in its casing

Again, multiple solutions where proposed, and in the end it was decided to make two holes into the casing plug and to use some sealing elements to inhibit flow of air. The mounting of the valves was then performed, together with the one of the venturi nozzle, being the two necessary to calculate and control the mass flow rate of air flowing into the bypass, so to keep constant and correct the amount of air passing through the turbine.

Figure 24 - Valves in parallel

For what concerns the long pipe from the valves to the upstream of the rotor, it was done as the last thing, in order to have more elasticity (given by welding more slices of pipe) in the collimation of the long duct and the circumference on the top of the tank which has the task to mix the flow coming from the turbine with the one coming from the bypass.

Figure 25 - Mixing tank

To complete the bypass assembly the operators continued the work by starting from the other end of the bypass, which is the compressor. Between compressor and silencer there is a duct, visible in the right of Figure 25, which has a tap to make the pressure atmospheric upstream of the compressor. By adding the mixing tank to the plant it was necessary to shorten that duct. Also, that duct in mounted to the compressor by inserting a portion of its length into the compressor casing, which causes the need to move all the other parts in the turbine direction. This procedure was very delicate since a lot of attention has to be paid in order to not lose the co-axiality of the whole train of ducts between turbine and compressor, since all the ducts are bolted together and the axiality is so spread from one piece to the other. Also, since the air coming to the tank had to flow through a pipe with lots of bends, there was a risk of having a swirled flow entering in the mixing tank, and during the mixing to increase its swirl motion. For this reason, professor Gaetani decided to weld into the convergent duct (the one from the mixing tank to the compressor) a set of four metal plates perpendicular to the circumference and spaced of 90° one from the other, in order to break the swirl. Once all this in set up, it is possible to conclude the bypass assembly with the welding of the long pipe coming from downstream of the valves to the mixing tank.

3.3. Writing of the software for calibration of classical venturi

As previously described, the High Speed Test Bench has been also modified by removing the previously existing venturi tube, shown in Figure 26, which wasn't in accordance with ISO 5167-4:2003(E), with a newly designed venturi tube which met the standards.

Figure 26 - Non standard classical venturi tube

The aim of this venturi tube is to calculate the mass flow rate flowing to the turbine, which is a critical parameter to study the turbomachine. The flow rate calculation is carried out through the formula

$$q_{m,c} = \frac{C}{\sqrt{1-\beta^4}}\varepsilon\frac{\pi}{4}d^2\sqrt{2\Delta p\,\rho_1}$$

where

- C is a coefficient, defined for an incompressible fluid flow, which relates the actual flowrate to the theoretical flowrate through a device, and is given by the formula for incompressible fluids
- β is the ratio of the diameter of the throat of the primary device to the internal diameter of the measuring pipe upstream of the primary device
- ε is a coefficient used to take into account the compressibility of the fluid
- d is the diameter of throat of primary device under working conditions
- Δp is the difference between the (static) pressures measured at the wall pressure tappings, one of which is on the upstream side and the other of which is in the throat, inserted in a straight pipe through which flow occurs, when any difference in height between the upstream and downstream tappings has been taken into account
- ρ_1 is the density of the fluid at the upstream tapping plane

The need of a calibration comes from the fact that the relative uncertainty for classical venturi tubes of the discharge coefficient can at best reach 0.7%, which is not an acceptable value to BH. So, the idea is to put a critical flow venturi nozzle downstream of the classical venturi, since following the ISO 9300:2005(E) it is obtained a relative uncertainty of the discharge coefficient for cylindrical throat venturi nozzles (which is the chosen layout for the case) of 0.3%. In this way we can first refer to the values obtained by the critical flow venturi nozzle and apply them to the classical venturi ones, in order to find a correlation to be used during the testing of the BH's turbine's setup.

The goal is to calculate the actual flow rate passing through the critical flow venturi nozzle and dividing it by the ideal flow rate passing through the classical venturi, in order to have as result the discharge coefficient of the classical venturi with the relative uncertainty of the critical flow venturi nozzle. At first, we have to gather some values out of the transducers placed on the test bench. In particular are keys for this calibration the upstream pressure, the upstream temperature and the difference of pressure between upstream and throat of both the classical venturi and the critical flow venturi nozzle.

At first, we calculate the flowrate in the sonic venturi nozzle which is given by the following formula

$$q_{m,s} = \frac{A_{nt} C_{d'} C_* p_0}{\sqrt{\frac{R}{M} T_0}}$$

Where:

- A_nt is the section at the nozzle throat
- p_0 in the upstream pressure
- T_0 in the upstream temperature
- R is the universal gas constant
- M is the molar mass
- C* is the Critical flow function for one-dimensional flow of a real gas which is

$$C_* = \sqrt{\gamma}\,(1 + \frac{\gamma - 1}{2})^{\frac{1+\gamma}{2(1-\gamma)}}$$

- Cd′ is the coefficient of discharge which, for a cylindrical throat venturi nozzle is

  $$C_{d'} = 0.9976 - 0.1388\, {Re_{nt}}^{-0.2}$$

  The Reynolds number at the throat is calculated as

  $$Re = \frac{\rho_{nt} V_{nt} D_{nt}}{\mu}$$

  The density is

  $$\rho_{nt} = \rho_0 (\frac{p_0 - \Delta p}{p_0})^{\frac{1}{\gamma}}$$

  The velocity is

  $$V_{nt} = \sqrt{\gamma\, R\, T_{nt}}$$

  since working in sonic conditions means to have a Mach number equal to 1. In the velocity formula the static temperature at the nozzle throat is calculated as

  $$T_{nt} = T_0 \, \frac{2}{\gamma + 1}$$

  The dynamic viscosity of air is calculated through the following relationship

  $$\mu = 0.001(-9 * 10^{-9} T^2[°C] + 4 * 10^{-5} T[°C] + 0.0168)$$

Once gathered the value of the actual flow rate for the sonic venturi nozzle we calculate the ideal one passing through the classical venturi, so the real value without accounting for the discharge coefficient, of the classical venturi as follows

$$q_{mi,c} = \frac{1}{\sqrt{1-\beta^4}} \varepsilon \frac{\pi}{4} d^2 \sqrt{2\Delta p\, \rho_1}$$

The density upstream is calculated through the perfect gas law, so as

$$\rho_m = \frac{p_m}{R_{gas} T_m}$$

whilst the compressibility factor is calculated as

$$\varepsilon = \sqrt{\left(\frac{\gamma r_d^{2/\gamma}}{\gamma - 1}\right)\left(\frac{1-\beta^4}{1-\beta^4 r_d^{2/\gamma}}\right)\left(\frac{1-r_d^{\gamma-1/\gamma}}{1-r_d}\right)}$$

where γ is the ratio of specific heat capacities and r_d is the pressure ratio $\tau = \frac{p_2}{p_1}$.

So now we can calculate the discharge coefficient of the classical venturi nozzle as

$$C = \frac{q_{ms}}{q_{mi,c}}$$

The procedure is to calculate the value of the discharge coefficient for different working points and then find a rule which links the discharge coefficient with the variables of the plant, since the sonic nozzle is not present during the testing of the BH's system since it would constraint the tests.

There are two other possible approaches to the problem

- to find a correction of the discharge coefficient's values present in the ISO 5167-4:2003(E) with the value found through calibration.
- to wrap up all the constant values used to calculate the mass flow rate and calibrate the classical venturi to the new constant.

Looking for a correction system would mean simply doing a further step which doesn't ease the job. In fact, it is already gathered the correct distribution of the discharge coefficient with the uncertainty required and it is already extracted a reliable law to be applied to the discharge coefficient during the tests. This further step would just mean to have another reliable law which doesn't add accuracy or ease to the trials. The decision was not to follow this approach.

Wrapping up all the constants would carry to a loss of clarity of the values of the variables, since first of all it would not be immediate to compare the discharge coefficient values found with the ones present in the standards, due to the loss of independency from geometry of the duct (β) and compressibility of the fluid (ε). The dependency to those two parameters would complicate the analysis. It was chosen not to follow this approach either.

3.4. Classical Venturi Calibration preparation

What has been practically done during the author's work has been the mounting of the classical venturi by the same operators which assembled the bypass and the buildup of pressure systems in order to gather an averaged value of pressure from the throat of the classical venturi. In fact, at the throat are present, as the standards say, four threaded holes which are the pressure tappings from which air comes to the transducers.

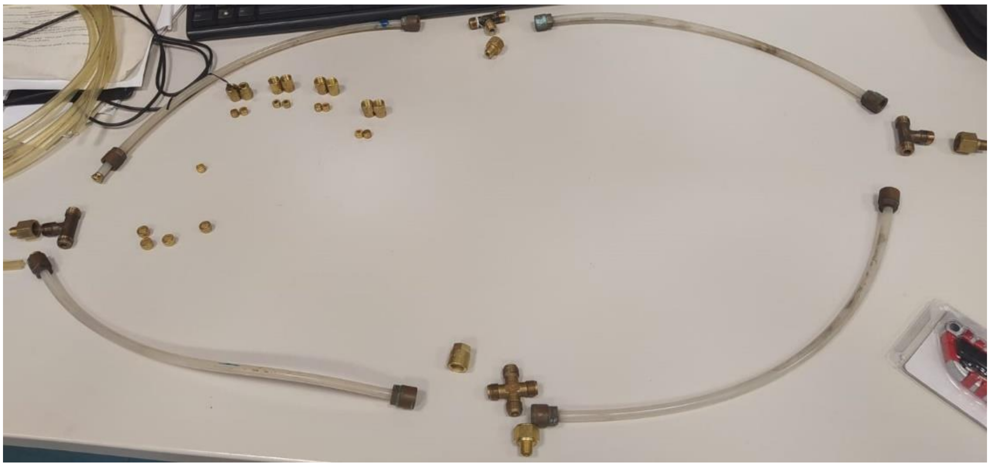


Figure 27 - Elements that compose the pressurized system

The system was built by taking a previously utilized set for pressure tappings from the old classical venturi. But since the previous set was composed by metal distancers that had the task to create more length between one pressure tapping and the other, together with the fact that the diameter of the two venturi tubes was different, it was necessary to cut the plastic tubes linking the tappings. The holes where threaded with $^1/_8$ of gas reference system, whilst the tubes had a dimension, which corresponds to $^1/_4$ of gas. So it was necessary to first insert four $^1/_8$ elements into the threaded holes into the venturi, then to find four others adapters $^1/_4 \rightarrow {}^1/_8$ to put into the elements into the tube, then to find three T joints plus a cross joint of $^1/_4$ and then to link the elements with the plastic tubes, utilizing ogives to keep the sealing between tube and copper elements, as shown in Figure 27. The sealing in between threaded copper elements is granted thanks to coils of nylon over the threads. This assembly is done in order to have an averaged pressure value of the whole circular section of the throat. The result can be seen in Figure 28.

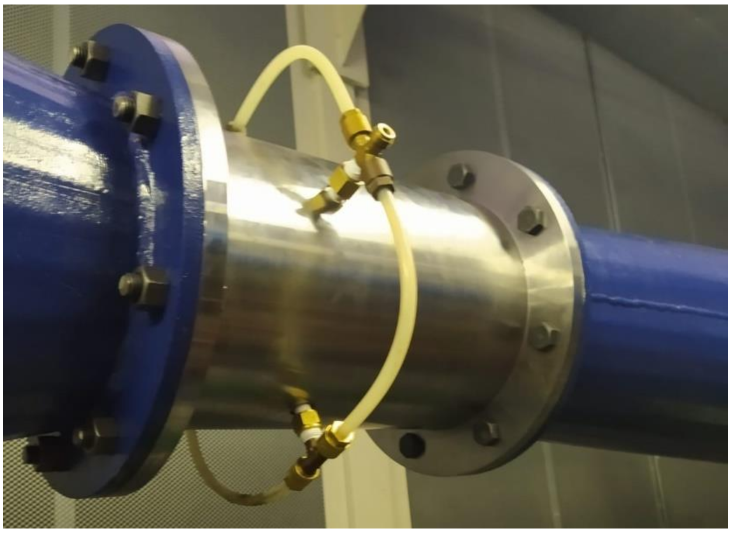


Figure 28 - Throat of classical venturi

Another thing that has been considered is the fact that there is a pressure switch which controls the oil pressure in order to inhibit that the turbomachine works if the oil is not in the bearings. During the test calibration of the classical venturi the turbine is not going to be connected, but the air will just flow to ambient after passing through the sonic venturi. That would mean an inhibition from the pressure switch to the activation of the electric motor that runs the compressor, making it impossible to calibrate the classical venturi. So, what has been done was to link the tube going to the pressure switch with the air secondary circuit, in order to shoot pressurized air during the classical venturi calibration.

3.5. Transducers Layout Modifications

The High Speed Test Rig has the aim, as the name says, of making tests on turbine and compressors. To gather results from the test it is necessary to acquire and register values of different units of measurement along the plant and on the turbomachines. To do so, transducers have to be utilized. It was already reported in the first chapter what was the transducers layout, in this one it’s going to be described what has been changed.

The first thing that has to be accounted for is the fact that an entirely new venturi nozzle is present in the plant, and it is the one that calculates the flow rate in the bypass. Thus, as seen for the classical venturi, it is necessary to have the values of pressure and temperature upstream of the venturi nozzle together with the difference of pressure between upstream and throat. Also, when calibrating the classical venturi, it is necessary to gather the same values also for the sonic venturi. So an idea was to first calibrate the classical venturi, which is a procedure where it is not needed to have control on the venturi nozzle, and then, once the sonic venturi is taken away, to move the transducers used for the sonic to the venturi nozzle. In theory that could have worked, but there is a difference between the sonic and the nozzle about the accuracy needed to calculate the Δp from upstream to throat. For the nozzle one the accuracy is very important, whilst for the sonic one we just want to go to Mach equal to 1 and then the mass flow rate doesn't change anymore. So the decision was to utilize the same transducer for the calculation of the upstream pressure, whilst a different one for sonic and for nozzle for the Δp. In particular the Schaevitz 33246 is the transducer that will be used for the upstream pressure of both the venturi, whilst the Δp of the sonic venturi is going to be calculate through a Shaevitz 36839, and the Δp of the venturi nozzle is going to be calculated through a more precise Rosemount. Since the plant is big and there's plenty of cable, to avoid having too much mess it is important to keep the transducers all in a casing. So, we had to design a way to insert the Schaevitz 33246 and the Rosemount into it, since these two are going to stay for the whole time of testing the Baker Hughes's turbine, whilst the Shaevitz 36839 is just staying temporary for the calibration of the classical venturi. The Rosemount had two bolts incorporated, so we drilled two holes on one side of the transducers casing and we locked it in there, whilst for the Schaevitz 33246 we had to utilize a wire clip and making it pass two holes already drilled in the rear part of the casing, as visible in Figure 29.

Figure 29 - Pressure transducers' casing

The job does not end here, since now all the transducers have to be linked to the chassis, which is far from the transducers casing. In order to keep cables managed inside the plant's floor there are some tunnels, going to the chassis, in which all the transducers' cables are hidden. So it was necessary to open the tunnels in some points, to let the cables enter and then make it follow the route all the way to the chassis.

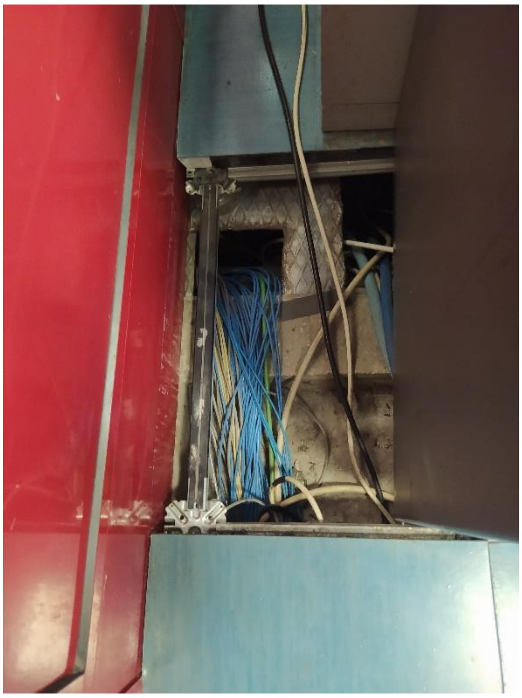


Figure 30 - Underfloor cables

To avoid making mistakes with which cable is linked to which transducer, there were put some labels with written on which variable was bringing to the chassis.

Figure 31 - Sahevitz 36839 with its label

3.6. Design of venturi tubes for oil flow rate calculation

Baker Hughes asked Politecnico di Milano a very high precision analysis on their turbine. That's the reason for the previously mentioned classical venturi calibration, but that's not the only arrangement required. Indeed, there is another tiny variable that Baker Hughes wanted to be taken into account: the amount of power stolen from the turbine by the two bearings. This quantity is usually extracted form curves given from the manufacturer of the bearings that correlate the amount of oil taken by the bearings with the amount of power they are subtracting from the shaft. So, what is needed is a way to calculate the flow rate of oil entering the bearings in order to know how much net power would actually extract the turbine. The idea was to design and manufacture two classical venturi tubes, one for each of the two turbine bearings, and calculate the flow rate of oil going to both the axial and radial bearings. The design of the venturi tubes was not a common one, since the throat dimensions were very small. Indeed, for the axial bearing, which is the one in which it is flowing the greater amount of oil, the throat diameter was supposed to be 6 mm, whilst for the radial one the diameter was 2 mm long. The manufacturing technique had to be additive manufacturing in order to meet the dimensional requests. The classical venturi tubes were designed according to the standard ISO 5167-4:2003(E). Since the tappings were very little it was chosen to build over the throat and upstream series of four tappings a closed chamber with just an output hole of bigger dimension, which was placed in between two pressure tappings, in order to have a more averaged value, as visible in Figure 32. It was done this way for both upstream and throat pressure tappings.

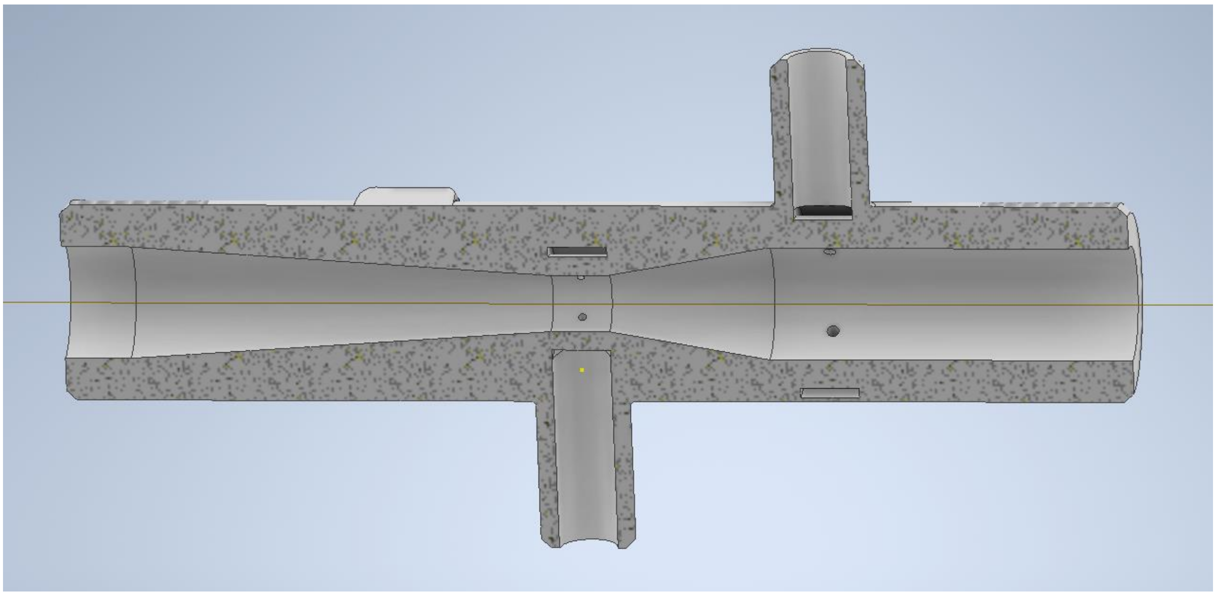


Figure 32 - Section of the venturi for oil flow rate measurement

Also, the tube was designed to have for a given length a hexagonal section, in order to be able to screw it easily with a wrench, that can be seen in Figure 33.

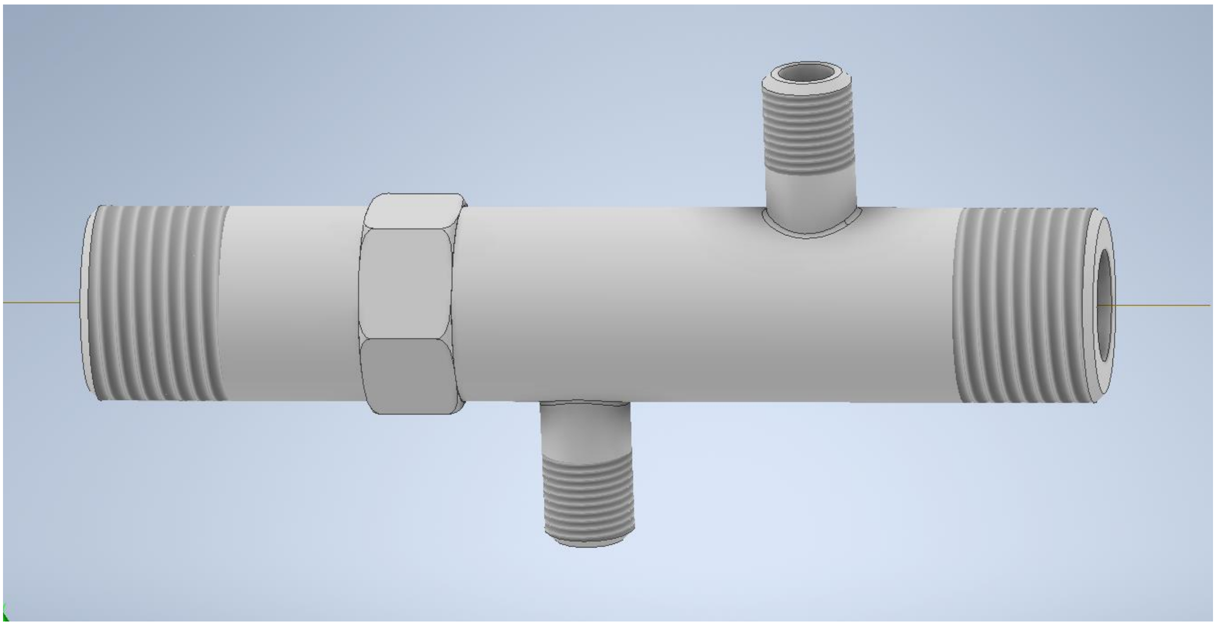


Figure 33 - Venturi for oil flow rate measurement

The issue is that the throat diameters are so small that the ISO 5167-4:2003(E) does not assure anymore a correct value of the discharge coefficient, so it is necessary a calibration.

The calibration procedure proposed to Baker Hughes was the following:

1) Turning on the bearing oil pumps and letting the system run until a condition similar to the test's one is reached.
2) Opening of the venturi tube's feeding valve.
3) Venturi tube's feeding for a known time and measure of the amount of oil flowed through the tube in that given time
4) Closing of the venturi tube's feeding valve

The test had to be repeated for three flow rate values, representative of the turbine's working points.

The design was completed, the quote was asked, but in the end Baker Hughes opted for a different solution, which is the utilization of Coriolis flow meters, since those are calibrated and certified, which is an important warranty of reliability. The certification for the previously presented system is very complex to be performed in a calibration center.

3.7. Transducers calibration

It is crucial for the correct functioning of the test bench to have the transducers working correctly, in order to gather the actual values of temperature, pressure, etc. The pressure transducers for example have the output of the measured pressure in Volt and have a linear behavior between volts and pressures. So, it is necessary to find the angular coefficient and the intercept to write the line that describe the correct correlation between volts and pressures.

3.7.1. Pressure side

The procedure consists in having first of all a decent amount of pressurized air, then to link this pressurized tank to the Mensor, a reservoir filled with pressurized air and able to manage the inlet and outlet of the pressurized air, up to 4 bar. Then with some tubes it is possible to link the Mensor with all the transducers that have to be calibrated. It is indeed possible to calibrate more than a transducer at a time, what is important, in order not to break them, is that their full-scale is comparable. For example, if the Shaevitz 36839, which has a full-scale of 2 bar, is calibrated together with a Rosemount, which has a full-scale of 30 mbar, the options are two:

1) The calibration is performed from 0 to 30 mbar, so there is no damage to any transducer, but whilst the Rosemount is correctly calibrated, the Shaevitz is surely not.
2) The calibration is performed from 0 to 2 bar, the Shaevitz is correctly calibrated while the Rosemount's membrane is damaged.

To link together more than a transducer to the Mensor it is necessary to build a complex connection of tubes, with the help of T joints. Also, it has to be connected to the pressure system a primary device, which is a high precision pressure transducer with both digital and electronic output, to make possible the correlation between volt exiting the transducers and the actual pressure value that they are measuring. Crucial for the calibration is checking if there are pressure losses inside the connection, since it would invalidate all the procedure. As seen in a previous chapter the sealings are usually obtained by means of ogives and teflon coils around threaded junctions. If the Mensor's pressure indicator is stable once all the junctions are made, then it is possible to start the calibration procedure, if not, it is needed to understand where the leakage is happening in order to fix the pressurized air loss. The first thing to do is to fill up a little water reservoir, in which the junctions are immerged. If some bubbles of air come to the water surface, there's clearly a leakage. To fix the issue it is possible to screw more tightly the threaded junction, or to use some more teflon. Sometimes it happens to have some damaged plastic tubes which have to be cut and replaced.

3.7.2. Electronic side

Once that all the pressurized air system is stable, it is necessary to set up the electronic system. First of all, the transducers have an electronic output from which are gathered the volts measured. This electronic output is acquired by inserting the electric cable of the transducer into a module of the National Instruments which is built to calculate the pressure, and by inserting the module into a National Instruments' Chassis, which is then linked to the PC. Now that the PC in acquiring the signal the next step is to open the LabVIEW program *taratura.vi* and select from the list of transducers the ones connected pressure-wise at the Mensor and electronic-wise at the PC. Then the pressure inside the Mensor is brought to 0 bar and the zeros are calculated, in order to take into account the value in volt of the transducers while they are in unloading. After the zeros calculation, it is possible to start building the line in the Pressure VS Volt diagram. A set of points is taken from 0 to the full-scale and from the full-scale to 0. Two possible way exist: looking at the number on the primary's display and manually inserting it into the LabVIEW program or wire the primary to the PC and let it acquire automatically the values. After that, the LabVIEW program automatically calculates the angular coefficient and the intercept, for each of the calibrated transducers. These two values are saved and utilized for all the calculations needed.

3.8. Referability calibration between PoliMi and BH

Since the commissioning was done with the aim of testing the Baker Hughes's turbine, it was important for the client to have a referrable of the results found in the LFM about the calibration of the classical venturi, which is crucial, since the power extracted by the turbine has to be referred to the mass flow rate of air. To assure that, two things were done.

3.8.1. Temperature transducers

For the measurements of temperature, two thermocouples, the ones measuring the upstream temperature of classical venturi and sonic venturi, were sent to the Nuovo Pignone facility in Florence to be calibrated, together with their whole wiring and National Instruments' module. One thermocouple was uninstalled from the plant, with complex unwiring through the tunnels previously mentioned. The other one was taken from the laboratory's inventory, and a proper wire was cut on the measure on the first thermocouple's one. Then the thermocouple was cabled on the wire with scissors and screwdriver.

3.8.2. Pressure transducers

For the measurement of pressure, two Baker Hughes operators came to the Laboratorio di Fluidinamica delle Macchine to calibrate the transducers measuring the upstream pressure and Δp of both classical venturi and sonic venturi. They brought their own instrumentation and took their measures of angular coefficient and intercept in the interval of interest of the pressure for the test that are going to be one on the turbine. This was done also by Polimi, in order to extract a law that correlates the behavior of the different measure systems.

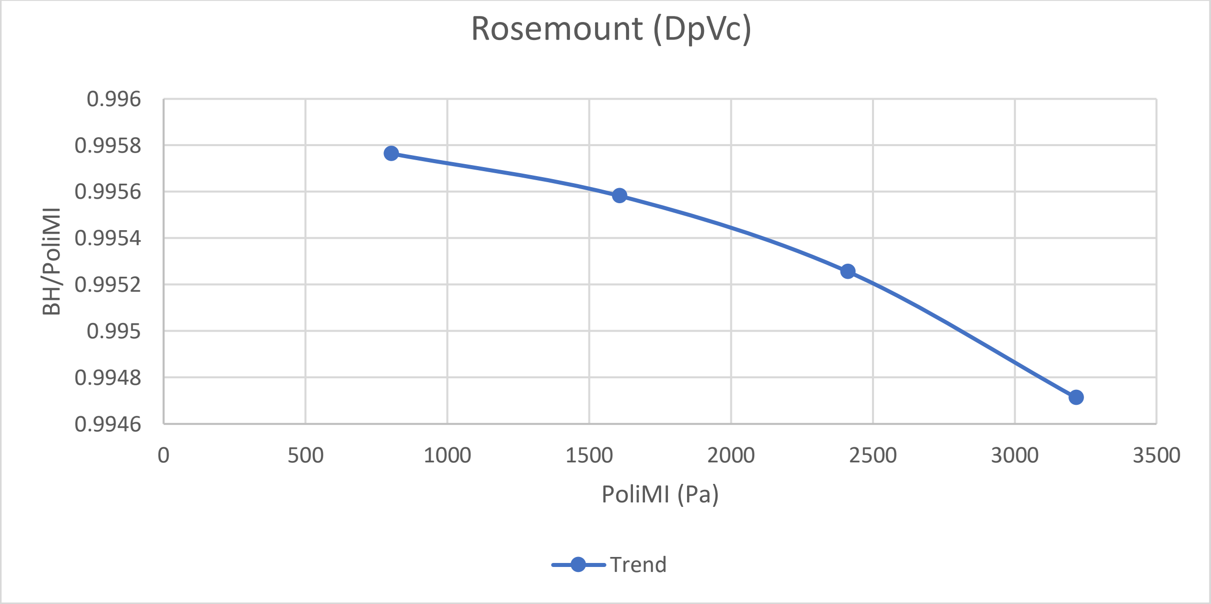


Figure 34 - PoliMi vs BH calibration of the Rosemount

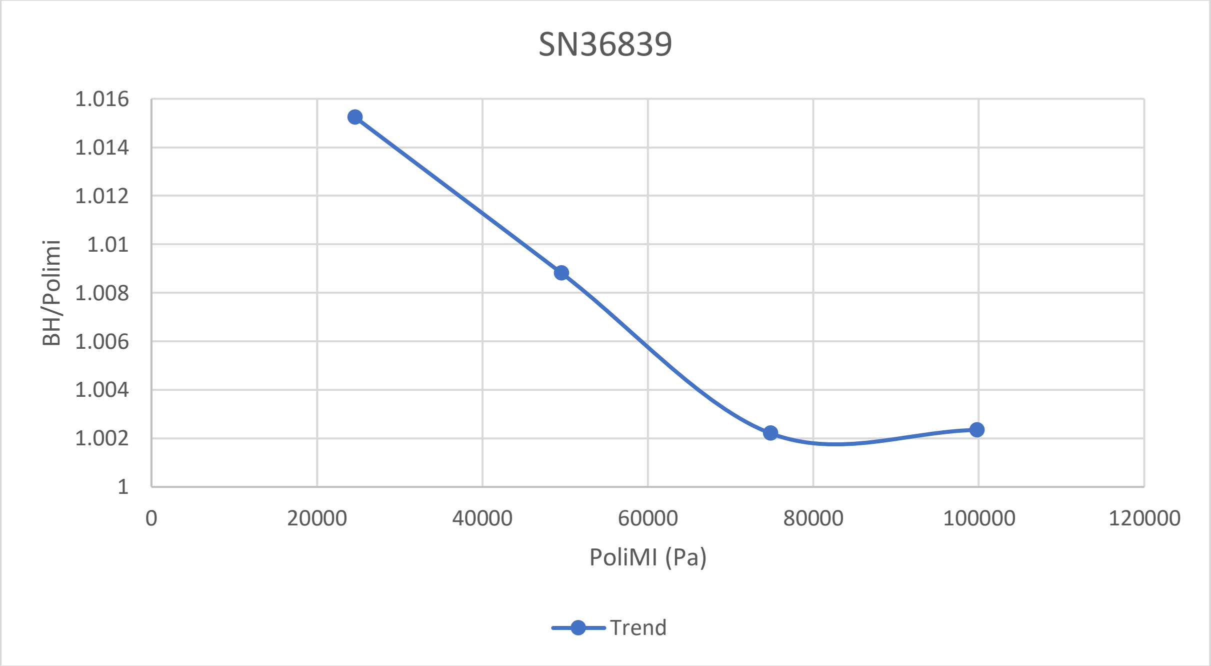


Figure 35 - PoliMi vs BH calibration of the SN36839

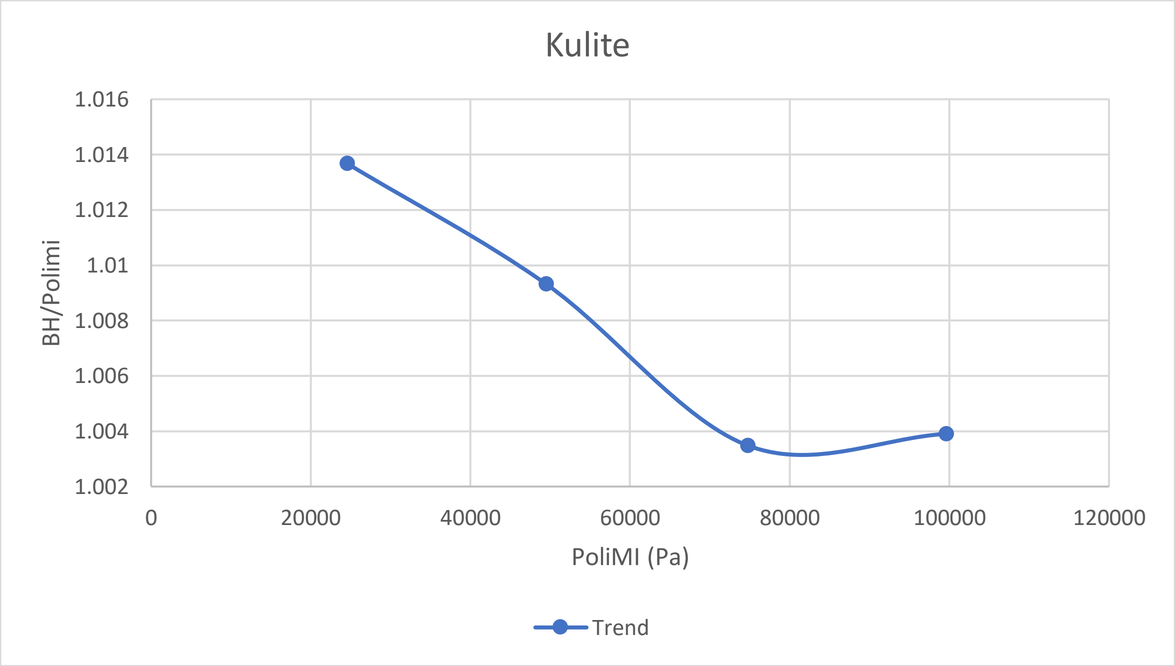


Figure 36 - PoliMi vs BH calibration of the Kulite

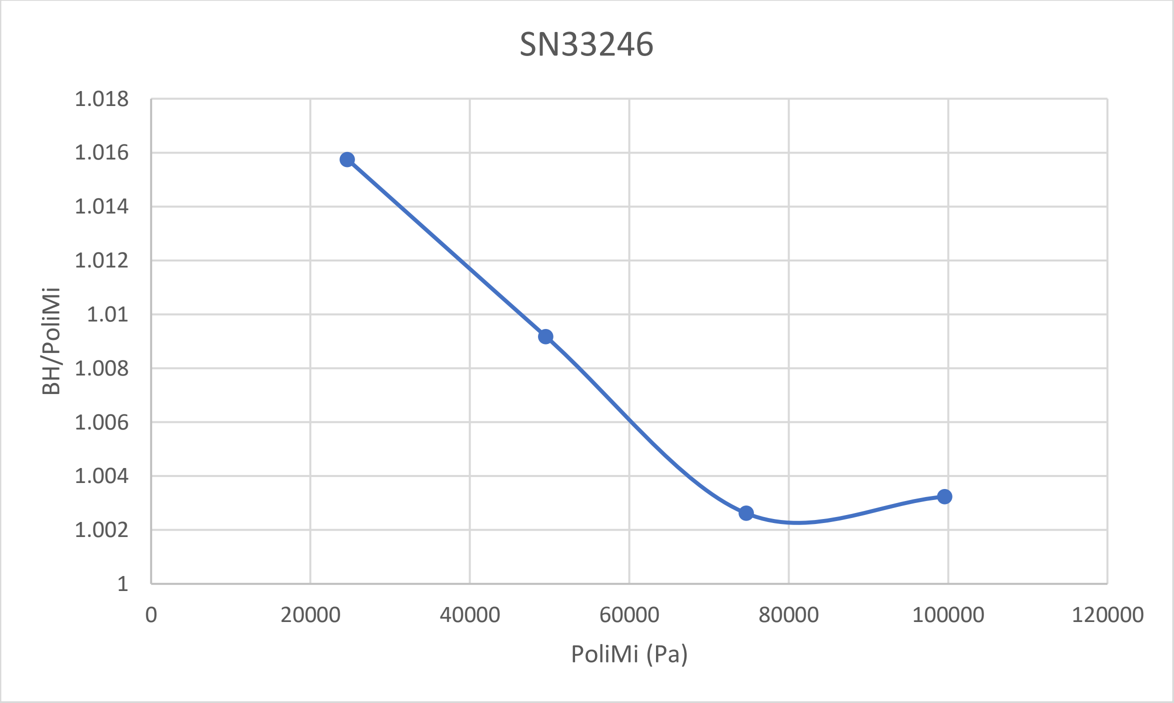


Figure 37 - PoliMi vs BH calibration of the SN33246

In the diagrams above are shown the results of the coordinated calibrations of PoliMi and Baker Hughes. It can be noticed that the Rosemount's PoliMi measure, with respect to the Baker Hughes's one, is slightly off the linear behavior, but of a quite constant value since the ratio between the Baker Hughes's values and PoliMi ones is stable around 0.9955 for the interval of interest. For the other three transducers instead the correlation is not constant, but with the more is the pressure to be measured, the more linear is the behavior. That's true until 0.7 bar, after that value the difference in measurement between PoliMi and Baker Hughes starts increasing again. Through interpolation it is possible to create a relationship to be used to correct the output value of PoliMi transducers into the one that would be calculate by the instrumentations of Baker Hughes, an example is presented in Figure 38.

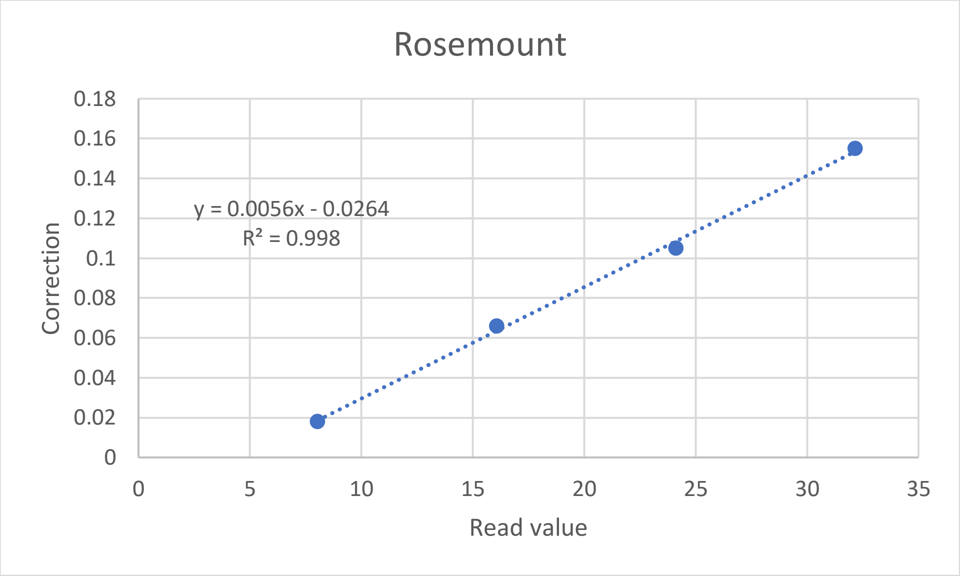
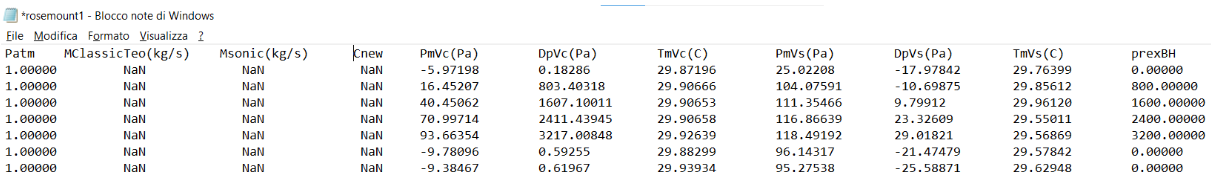


Figure 38 - Equation to correct PoliMi values into BH's ones for the Rosemount

3.9. Update of plant's monitoring software

The plant has a highly complex electronic system with about fifty variables measured each acquisition cycle. To manage all the measured values in a useful way it is necessary to have a program which acquires and elaborates all the data, and then gives as output the values of interest. This program is written in LabVIEW and has the aim to control the whole plant. In fact, the first condition to be matched while testing is that the plant is working safely and correctly. Meaning that the oil has to be put in pressure into the bearings, the rpm of the turbomachines has to be under control, that no leakage of oil or air in happening, that the temperature of all the plant's components is controlled. Hence it is possible to look at how much flow rate is flowing, how much power the turbine is extracting etc.

To keep the program up to date after the plant modifications it has been necessary to make some changes. First of all, with respect to the previous plant, three more signals have to be acquired, those of the Rosemount acquiring the Δp between upstream and throat for the venturi nozzle, of the Shaevitz 33246 acquiring the upstream pressure of the venturi nozzle and of the thermocouple acquiring the venturi nozzle's upstream temperature. Also, one signals is not going to be acquired anymore, which is the compressor's torsiometer. This change in the signals acquired has to be taken in account at different levels into the program. It is necessary to add the global variables in which are stored these three values, together with their angular coefficient and intercept coming from the pressure transducers calibration, and the thermocouple constant coming from the thermocouple calibration, in order to convert the volt coming from the signal to a readable pressure or temperature value. This has been done for all the three configurations: axial turbine only, centrifugal compressor only, whole closed loop cycle. Also, since the program is written to be able to record, writing on a text file cycle by cycle, the acquired values, it is necessary to modify the heading of the columns so that they match the written values.

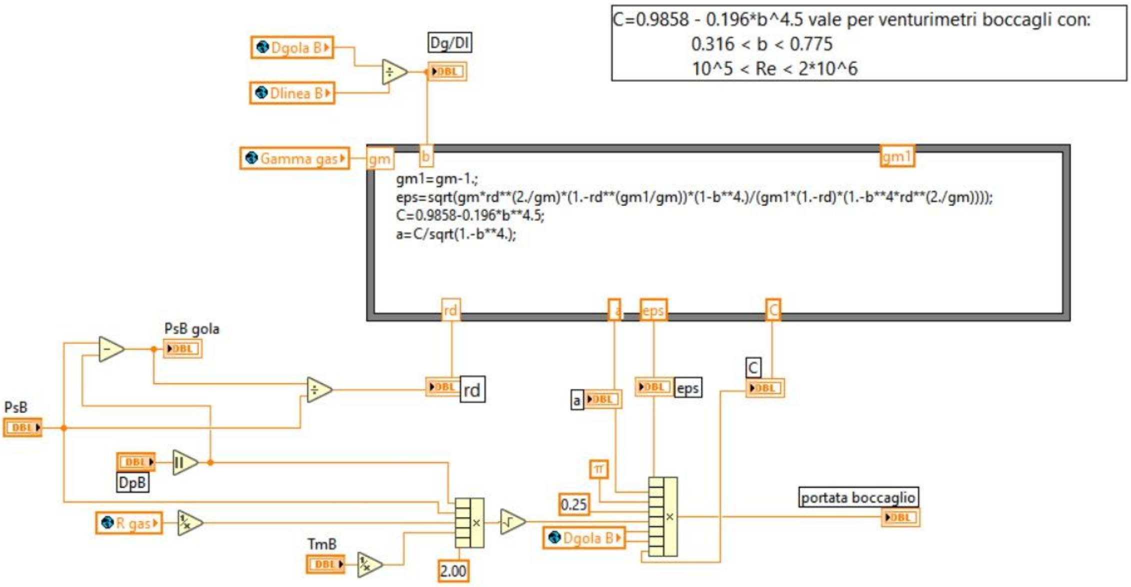


```
Patm     MClassicTeo(kg/s)    Msonic(kg/s)       Cnew     PmVc(Pa)        DpVc(Pa)        TmVc(C)         PmVs(Pa)        DpVs(Pa)        TmVs(C)         prexBH
1.00000          NaN               NaN            NaN     -5.97198        0.18286         29.87196        25.02208        -17.97842       29.76399        0.00000
1.00000          NaN               NaN            NaN     16.45207        803.40318       29.90666        104.07591       -10.69875       29.85612        800.00000
1.00000          NaN               NaN            NaN     40.45062        1607.10011      29.90653        111.35466       9.79912         29.96120        1600.00000
1.00000          NaN               NaN            NaN     70.99714        2411.43945      29.90658        116.86639       23.32609        29.55011        2400.00000
1.00000          NaN               NaN            NaN     93.66354        3217.00848      29.92639        118.49192       29.01821        29.56869        3200.00000
1.00000          NaN               NaN            NaN     -9.78096        0.59255         29.88299        96.14317        -21.47479       29.57842        0.00000
1.00000          NaN               NaN            NaN     -9.38467        0.61967         29.93934        95.27538        -25.58871       29.62948        0.00000
```

Figure 39 - Recorded values from PoliMi vs BH Rosemount calibration

In Figure 39 we can see an example of what is mentioned above: all the values acquired has to be written above their corresponding measured quantity.

Then it was necessary to add a part of the code which calculates through these signals the air flow rate passing through the venturi nozzle, crucial to avoid the surge issue, visible in Figure 40.


C=0.9858 - 0.196*b^4.5 vale per venturimetri boccagli con:
0.316 < b < 0.775
10^5 < Re < 2*10^6

```
gm1=gm-1.;
eps=sqrt(gm*rd**(2./gm)*(1.-rd**(gm1/gm))*(1-b**4.)/(gm1*(1.-rd)*(1.-b**4*rd**(2./gm))));
C=0.9858-0.196*b**4.5;
a=C/sqrt(1.-b**4.);
```

Figure 40 - LabVIEW program for venturi nozzle flow rate calculation

3.10. Update of plant's monitoring hardware

In parallel with the software modifications, there was also the need of some hardware ones. In the LFM there were two National Instruments chassis available, of different generations. The oldest one was able to be connected to the desktop computer used for the plant's monitoring, the one connected to the SCXI, whilst the newer one couldn't. The modules utilized for the classical venturi calibration were fitting only in the newer chassis. For this reason it was decided to perform the calibration of the classical venturi on a laptop able to read the new generation chassis and so the calibration modules. Then another problem came out: there was no room for the venturi nozzle's transducers on the SCXI. Also, there's the fact that the PC cannot read at the same time SCXI and CompactDAQ, which is the chassis, so the signals' acquisition would have been very slow. By keeping separated the two acquisition systems and wiring the two through LAN a good acquisition frequency is obtained.

These situations combined brought to the decision of putting all the signals related to classical venturi and venturi nozzle to be read on the laptop, writing on it another monitoring LabVIEW program, this one accounting for these two components and for the turbine's new torsiometer. This decision came from the fact that both cables wiring the sonic venturi transducers to the National Instrument modules were going to be the same, once completed the classical venturi calibration, of the venturi nozzle. So it was quite consequential to do on the laptop first the classical venturi calibration and then the classical venturi and venturi nozzle monitoring.

4 Conclusions

In this chapter are presented the following tasks to be done after this thesis work.

4.1. Classical Venturi Calibration

The pressure cabling has to be done for the four upstream tappings of the classical venturi and they have to be brought, in parallel with the pressure level from the throat, to the transducers casing and correctly joined to the transducers. Then the venturi nozzle has to be mounted downstream of the classical one and distanced by a duct of a length given by the standards. All the pressure cabling has to be done also for the venturi nozzle. Then the calibration can start.

4.2. FRAPP and Five Holes Pressure Probe calibration

The Frapp (Fast Response Aerodynamic Pressure Probe) and the Five Holes Pressure Probe are two aerodynamic probes used to analyze respectively flow field unsteadiness and steady flow field. As all probes, they have to be calibrated in order to acquire actual values.

4.3. Installation of a humidity transducer

Humidity affects in an unneglectable way the measurements, so in order to have correct values of air dynamic viscosity for example it is going to be installed a transducer able to keep updated the humidity value inside the plant. In particular, the most important value affected by humidity is the gas constant.

4.4. Actual tests of BH’s Turbine

Nowadays the power stations are not producing over a stable working point for long periods, but more and more the effort is to use renewable energy. So the steam turbines are utilized to compensate the lack of renewable energy, which is highly variable, and so a very flexible control consistent with the efficiency has to be applied. The tests on the Baker Hughes’ turbine have the aim to try a new first stage geometry of a steam turbine characterized by the presence of nozzle guide vanes, which is a regulating stator able to substitute the lamination valve previously utilized. This is all about a heavy increase in the efficiency of the stage and so of the turbomachine.

The tests on the turbine are going to be performed in different ranges. Are going to be done about 50 tests with pressure ratios in the range 1.15-1.9 together with a variation in the opening of the nozzle guide vanes from 0 to 25 degrees. Also, are going to be performed global measurements through pressure rakes, temperature rakes and torque meter. Stationary flux measurements are going to be performed at the inlet of both stator and rotor with the Five Holes Pressure Probe, non-stationary flux measurements are going to be performed at the outlet of both stator and rotor through the Fast Response Aerodynamic Pressure Probe.

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