Department of Precision and Microsystems Engineering

A Validation System for the Compressor Side of an Electric Assisted Turbocharger

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Challenge the future

Summary

This study focuses on investigating an alternative validation method to validate the compressor side of turbochargers. Nowadays, the performance of turbochargers is tested using hot gas stands. These hot gas stands are expensive. The application of an externally coupled e-motor to the turbocharger eliminates the need for a hot gas burner - significantly reducing the cost. However, the validation range with electric motors is limited. Therefore, it is investigated if the circulation of the compressor outlet flow provides additional shaft power by the turbine. Thereby it would be possible to generate a bigger range of compressor performance maps in terms of power and rotational speed with a limited electric motor.

In the first part of the study the test setup is designed. The original 12kW electric motor of the e-turbo is not available yet. This led to an alternative electric motor-turbocharger combination. After trying various options, an electric motor with a relatively high power output but not enough speed to reach the lower speed line of the compressor map was chosen. It was decided to test the principle of range extension with the turbine in the lower rotation speed range. It is possible to match the speed of the electric motor and the compressor using gears. This has not been applied due to complexity and unforeseen risks at high speeds. The test set-up will be used in two different configurations. In one configuration, the compressor is driven by only the electric motor without turbine housing. In the other configuration, the outlet flow from the compressor will be sent to the turbine. The two compressor maps of those two configurations will be compared. For both configuration the input of the electric motor is kept constant in order to validate if by adding turbine to the test setup a shift can be measured in operating points. To generate such compressor maps, the inlet and outlet pressure and the temperature around the compressor must be measured. And also the flow rate and the rotational speed of the turbocharger must be measured. These parameters will be measured using the guidelines of the SAE standards for the validation of turbochargers.

In the second part, a performance expectation model is set up in Matlab. The model is built from the existing compressor map, turbine map and electric motor characteristics. Those are used to calculate the determine which operating point can be reached with only the electric motor and also with the electric motor and the turbine. For every operating point in the compressor map the power and torque are calculated which is needed to reach that operating point. Then the compressor powers and torques are compared with the available electric motor and turbine power. When the available electric motor and turbine values are higher as the compressor values, an operating point is considered in reach. The outcome of the model is a compressor map in which is indicated which operation points can be reached with only the electric motor and with the combination of the electric motor and the turbine.

The result of the model shows that the outlet flow in both configurations always remains above 1 bar and 0 degrees Celsius. This means that no freezing gas is expected to be generated. This could be harmful for the turbocharger. Furthermore, the model shows that for a selected operating point it is possible, depending on the mass flow control, to push the operating point to a higher rotational speed or mass flow value. In addition, the gear ratio was still built into the model. With the current electric motor, it was possible to validate a larger area in terms of power. Because this is now limited by the rotational speed, it is possible to validate all operation points within the power area of the electric motor with the right gear ratio. So potentially, with an electric motor with the right specifications and the right gear ratio, the full compressor map can be validated.

The aim of the research was to use an alternative method to validate the compressor of the turbocharger whereby validation range could be extended by means of energy recovery with the turbine. This study shows a test setup design and a performance prediction model that shows that it is possible to eliminate the use of a hot gas burner for compressor wheel validation and use an electric motor together with the turbine to extend the validation range in a compressor map. However, within the time frame of this study, no physical test was performed due to the unavailability of different electric motors. Nevertheless, this study shows the potential for an alternative way of compressor wheel validation for turbochargers and contributes to the development of the electric assisted turbocharger.

A Validation System for the Compressor Side of an Electric Assisted Turbocharger

A study about the validation range of the Compressor map powering the compressor with the electric motor and the Turbine

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Preface

This report covers a research into the application of a validation technique for turbocharger compressor wheels in the turbo system industry. It has been written for the purpose of obtaining a Master's degree in Mechanical Engineering at the Department of High-Tech Engineering, TU Delft. This research was carried out at the company Van Der Lee Turbo Systems, located in Zaandam in the Netherlands.

The thesis consists out of two parts. In the first part the design of the new validation setup is discussed with the associated design considerations. In the second part a performance prediction for the validation principle and test setup is set up. A general discussion and conclusion can be found at the end of the report. The evaluation of the physical design and measurements has not been carried out due to delivery problems of the suitable electric motor.

I would like to thank my supervisors Chris Davies and Jo Spronck for their support during my thesis. The weekly support during meetings gave me new insights and tips during the project. I would also like to thank Ron van Ostayen, thanks to his input the academic goal of the project became clear. In addition, Ron provided valuable feedback during the beginning of the project on a 3 to 4 week basis. I would also like to thank the other colleagues at Van Der Lee Turbo Systems for their support and guidance, and Jos van Driel for his support during the measurement set-up and the delivery of the measuring equipment.

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Nomenclature

A_{pipe}	Cross area of the pipe	$[m^2]$
C_p	Heat capacity at a certain condition	$[J/(kg \cdot K)]$
C_{pa}	Heat capacity at the compressor side of the turbocharger	$[J/(kg \cdot K)]$
C_{pe}	Heat capacity at the turbine side of the turbocharger	$[J/(kg \cdot K)]$
$CPSR$	Constant power speed ratio	$[-]$
g	Gravity constant	$[m/s^2]$
GR	Gear ratio	$[-]$
$h_{comp,AC}$	Actual enthalpy rise over the compressor	$[J/kg]$
$h_{comp,IS}$	Isentropic enthalpy rise over the compressor	$[J/kg]$
h_m	Pressure loss of components in flow	$[Pa]$
K	Loss coefficient	$[-]$
L	Length between operating points in the compressor map	
$L_{new\ OP}$	Length between the new operating point and the lower operating point in the compressor map	
M	Mach Number	$[-]$
$\dot{m}$	Mass flow	$[kg/s]$
$\dot{m}_{comp}$	Mass flow through the compressor	$[kg/s]$
$\dot{m}_{corr,c}$	Corrected mass flow	$[kg/s]$
$\dot{m}_{turb}$	Mass flow through the turbine	$[kg/s]$
N_{corr}	Corrected Rotational speed	$[rev/min\ or\ rpm]$
$N_{comp,OP}$	Rotational speed of the compressor at an operating point	$[rev/min\ or\ rpm]$
$N_{comp,high\ lim}$	Highest closest rotational speed in the compressor map	$[rev/min\ or\ rpm]$
$N_{comp,low\ lim}$	Lowest closest rotational speed in the compressor map	$[rev/min\ or\ rpm]$
$N_{desired}$	Desired rotational speed for the new operating line	$[rev/min\ or\ rpm]$
N_{emotor}	Rotational speed of the electric motor	$[rev/min\ or\ rpm]$
$N_{emotor,CPSR}$	Rotational speed of the electric motor at the CPSR point	$[rev/min\ or\ rpm]$
$N_{emotor,max}$	Maximum rotational speed of the electric motor	$[rev/min\ or\ rpm]$
$N_{gearbox}$	Rotational speed gearbox output	$[rev/min\ or\ rpm]$
N_{Turbo}	Rotational speed of the turbocharger shaft	$[rev/min\ or\ rpm]$
P_{comp}	Power of the compressor	$[W]$
$P_{Comp,OP}$	Power of the compressor at an operating point	$[W]$
$P_{emotor,max}$	Maximum power of the compressor	$[W]$
$P_{emotor,OP}$	Power of the electric motor at an operating point	$[W]$
P_{turb}	Power of the turbine	$[W]$
$P_{turb,OP}$	Power of the turbine at an operating point	$[W]$
p_1	Pressure at location 1	$[Pa]$
p_2	Pressure at location 2	$[Pa]$
p_{amb}	Ambient pressure	$[Pa]$
$p_{comp,out}$	Pressure at the outlet of the compressor	$[Pa]$
$p_{comp,in}$	Pressure at the inlet of the compressor	$[Pa]$
Pr	Prandtl number	$[-]$
p_{ref}	Reference pressure of 100 000 Pa	$[Pa]$
$p_{static,out}$	Static pressure	$[Pa]$
$p_{static,out}$	Static pressure at the compressor outlet	$[Pa]$
$p_{total,in}$	Total pressure	$[Pa]$
$p_{total,in}$	Total pressure at the compressor inlet	$[Pa]$
$p_{turb,out}$	Pressure at the turbine outlet	$[Pa]$
$\dot{Q}$	Volume flow of the air	$[m^3/s]$
$\dot{Q}_{point\ 1}$	Volume flow of the air at point 1	$[m^3/s]$
$\dot{Q}_{point\ 2}$	Volume flow of the air at point 2	$[m^3/s]$
R	Specific gas constant	$[J/(kg \cdot K)]$
rec	Recovery factor for temperature	$[-]$
T_{amb}	Ambient temperature	$[K]$
T_{aw}	Adiabatic wall temperature	$[K]$
ΔT_{comp}	Temperature difference over the compressor	$[K]$
$T_{comp,in}$	Temperature at the inlet of the compressor	$[K]$
$T_{comp,out}$	Temperature at the outlet of the compressor	$[K]$
T_{prob}	Temperature measured by the probe	$[K]$
T_{ref}	Reference temperature of 298 K	$[K]$
T_{static}	Static temperature	$[K]$

T_{total}	Total temperature	$[K]$
$T_{turb,out}$	Temperature of the outlet of the turbine	$[K]$
ΔT_{turb}	Temperature difference over the turbine	$[K]$
$\mathrm{T}_{Comp,OP}$	Torque of the compressor at an operating point	$[Nm]$
T_{comp}	Torque of the compressor	$[Nm]$
T_{emotor}	Torque of the electric motor	$[Nm]$
$\mathrm{T}_{emotor,max}$	Maximum torque of the electric motor	$[Nm]$
$\mathrm{T}_{emotor,OP}$	Torque of the electric motor at an operating point	$[Nm]$
$\mathrm{T}_{\mathrm{gearbox}}$	Torque of the output of the gearbox	$[Nm]$
T_{turb}	Torque of the turbine	$[Nm]$
$\mathrm{T}_{turb,OP}$	Torque of the turbine at an operating point	$[Nm]$
V_1	Air velocity at location 1	$[m/s]$
V_2	Air velocity at location2	$[m/s]$
γ	Heat capacity ratio of air at a certain condition at turbine side	$[-]$
$\eta_{comp,IS}$	Isentropic efficiency of the compressor of the turbocharger	$[-]$
η_{turb}	Efficiency of the turbine	$[-]$
κ	Heat capacity ratio of air at a certain condition	$[-]$
Π_{comp}	Pressure ratio over the compressor	$[-]$
$\Pi_{point\ 1}$	Pressure ratio of point 1	$[-]$
$\Pi_{point\ 2}$	Pressure ratio of point 2	$[-]$
Π_{turb}	Expansion ratio over the turbine	$[-]$
ρ	Air density	$[kg/m^3]$

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1

Introduction

This chapter introduces the reader step-by-step to the topic of turbochargers and turbocharger validation techniques by explaining the basic principles. The validation of the compressor side of the turbocharger is introduced as the main research theme of this thesis. The relevance of this study is pointed out and also the outline of this thesis is sketched.

1.1 Introduction

With the growing focus on climate change and emissions of combustion engines, engine manufacturers desire to improve the efficiency of their engines. Nowadays, combustion engines are still providing the main source of mechanical energy in the automotive, aviation, marine or transport industry in general [1]. An engine has the purpose to convert thermal energy (fuel and air) into mechanical energy in order to drive all kinds of means of transports. In most cases the engine is designed as an air pump should produce as much power as possible. The more air an engine can pump the more power it could generate. However, pumping more air does not mean the engine will produce more power. The power is produced by the combustion of fuel with air. Roughly can be said, the more fuel can be combusted the more power is produced. Therefore enough air should be present in engine cylinders. The air inlet of an engine can be increased by making use of for example a turbocharger [2]. A typical turbocharged engine system is visualized in figure 1.1.

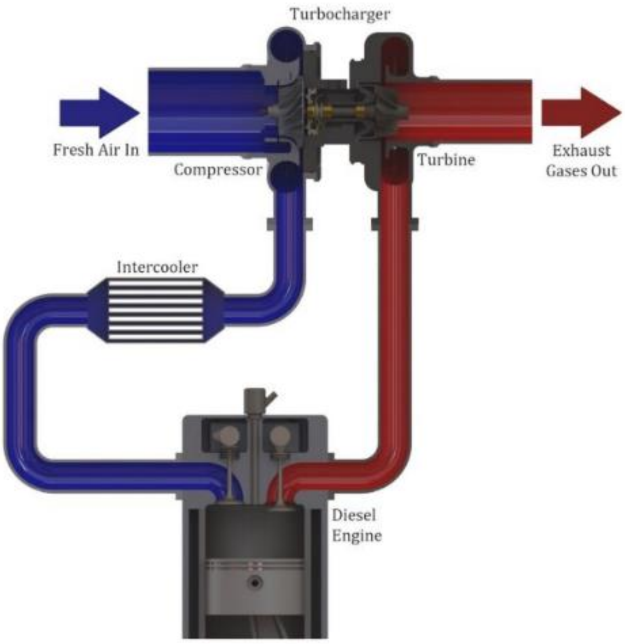


FIGURE 1.1 – OVERVIEW OF A TURBOCHARGER-ENGINE SYSTEM WHERE THE AIR GUIDED FROM THE COMPRESSOR TO THE INTERCOOLER, THEN THE ENGINE AND AS LAST THE TURBINE SIDE OF THE TURBOCHARGER [2]

A turbocharger is an add on device for combustion engines that forces more air into the engine. The turbocharger extracts energy from the hot exhaust gas with a turbine wheel and uses this to compress air with a compressor wheel that will be fed into the engine. The turbine wheel and compressor wheel are mounted on the same shaft, so they spin at the same rotational speed. The advantage of turbochargers is that it does not consume power from the engine to increase the power output of the engine because they are mechanically disconnected. On the other hand, the disadvantage is that it takes some time to reach the desired rotational speed of the rotating assembly where the compressor wheel generates the desired boost pressure that will increase engine power. This phenomena is called turbo lag. The reason that turbo lag occurs is because of the inertia of the shaft and wheels and the amount of power that is available in the exhaust gas is for a small time not enough to power the compressor to the desired operating range [3].

Turbo lag can be decreased by making use of an electric-assisted in the form of an electric motor. These turbochargers are called electric-assisted turbochargers, or in short e-turbo. For this turbochargers an electric motor is mechanically linked on the same shaft as the compressor and turbine wheel. The electric motor can assist the turbine wheel to accelerate faster and also convert overspeed into electric energy. At the moment, Van Der Lee Turbo Systems is developing an electric-assisted turbocharger where the electric motor is placed on the compressor side of the turbocharger. The idea of this e-turbo is that every component could be replaced for a component with different specifications [4].

Selecting the appropriate turbocharger for an engine involves many considerations. One of the most important considerations is selecting the right compressor and turbine wheel pair. A tool that is used for the selection of these wheels are the performance maps of these wheels. The performance maps are a visualisation of the performance data. In this research will be focussed on a new validation technique for the compressor side of the turbocharger. A typical compressor map of a Garrett compressor wheel is shown in figure 2.2.

Compressor performance tests can be measured on different type of test setups. The procedure at every performance setup is more the less the same. The turbine side is loaded with air, the rotational speed will be kept constant and the mass flow will be varied. The test setup that are mostly used in industry to determine the performance of a turbocharger are the hot gas stand. In this test stand the engine flow is replaced by a burner or heater to create a high turbine inlet temperature. The hot gas stand is currently used at Van Der Lee as test setup to validate the performance of turbochargers. An example of such a test facility is developed by Young and Penz (1990). In this stand, compressed hot gas between 150°C and 812°C can be fed into the turbine. By making use of a compressors and a burner a wide range of inlet conditions can be simulated [5]. Naundorf et al. (2001) developed the hot gas stand that is being sold by Kratzer Automation which has a turbine inlet temperature range from 150°C to 1200°C and pressure from 0 to 8 bar [6]. The hot gas measured performance is mainly important to extract the turbocharger aerodynamic performance [7]. To give an impression on how a hot gas stand looks like in figure 1.2 the test stand of Shaaban is shown.

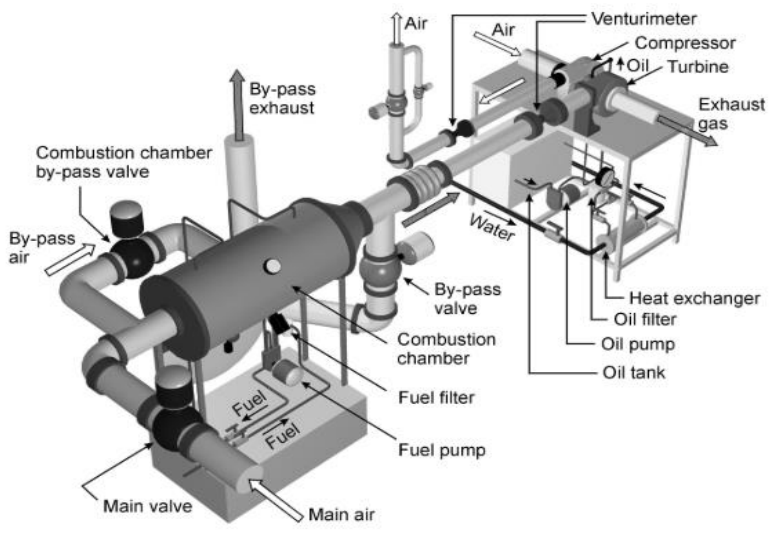


FIGURE 1.2 – EXAMPLE OF A HOT GAS STAND FROM THE RESEARCH OF SHAABAN [4]. SUCH A PERFORMANCE TEST IS MAINLY USED IN INDUSTRY FOR PERFORMANCE TESTING.

Another type of test setup is the test setup of R. Kirk et al. (2008) who developed a test setup where the turbocharger turbine is powered with cold compressed air. Driving the turbine with cold air can lead to an turbine outlet that is below 0 °C. When the outlet air of the turbine drops 0 °C, snow and ice are generated. The generation of snow and ice is called freezing gas. Freezing gas can cause stalling or damage to the turbine wheel and must therefore be avoided. Therefore, the turbine outlet temperature must always be above 0 °C [7]. Due to the possibility of creating freezing gas, this type of test setups are not widely used in the industry. But for low rotational speeds cold air powered test setups are used for research purposes. In the test setup of this study, the turbine is also driven by cold air. Because an electric motor is used to drive the turbocharger instead of an hot gas stand, the generation of freezing gas must be prevented in this study as well.

The last type of test setup pointed out for this research is the electric motor driven test setup. In the research of Andersen et al. (2009) a test stand has been developed specially designed for compressor validation in the limit regions of the compressor performance map. By driving the compressor wheel into surge and choke regions of the compressor wheel, these type of operating points can be analysed [8]. Further performance tests are often not carried out by electric motor powered test setups because of limited rotational speed and power available to operate at the desired operating point. In this study therefore a test setup will be designed and investigated where the limited validation range of an electric motor driven test setup is extended.

In this study the principle of the electric-assisted turbocharger will be used to investigate a new validation technique for compressor performance. As mentioned, generating a compressor map by powering the compressor with an electric motor is now new. However, to generate the full performance map a big electric motor is needed in terms of power and rotational speed. Therefore, in this research, a method is defined where an electric motor and re-circulation of the compressor outlet flow will provide shaft power to produce a full-range compressor map. The additional shaft power is used to produce a full-range compressor map. The turbine will recover some energy out of the exhaust in order to drive the compressor to higher operating points in terms of power and/or shaft speed. Thereby it would be possible to generate a compressor map with a smaller electric motor. Or the other way around that with the available electric motor a bigger range of compressor maps could be measured.

The above led to the research goal to design a turbocharger performance test lay out to measure compressor maps where this new concept of an externally coupled e-motor to the turbocharger eliminates the need for a hot gas burner - significantly reducing the cost. This test lay out will be used to proof the strategy of extending map range in terms of power and speed with circulation of flow through the turbine. In addition, this study will lay a base for new bigger developments such as a better e-turbo design or a sequential axial and radial compressor stage.

1.2. Thesis Outline

To explain what has been researched and done during this study, the thesis is divided into 4 chapters. The first chapter is the introduction in which the subject is introduced paragraph by paragraph and the purpose of the study is explained.

In the second chapter, the design of the test setup is explained. This chapter explains what needs to be measured to create a compressor map and how this is done. Furthermore, different design choices are explained and test strategy discussed.

In the third chapter, a model was made of the designed system to predict its performance. This model has been used during the design process to arrive at the correct composition of the test set-up. In this chapter, a performance expectation is calculated based on known test data of the different components of the test setup. The result is a compressor map with an area that can be validated with the electric motor and an area that can be validated with the turbine and electric motor.

The fourth and final chapter draws conclusions from the previous chapters. In addition, these conclusions will be critically examined in the discussion and a reflection on the work done will be made. As last some recommendations will be given to give direction to future research.

2

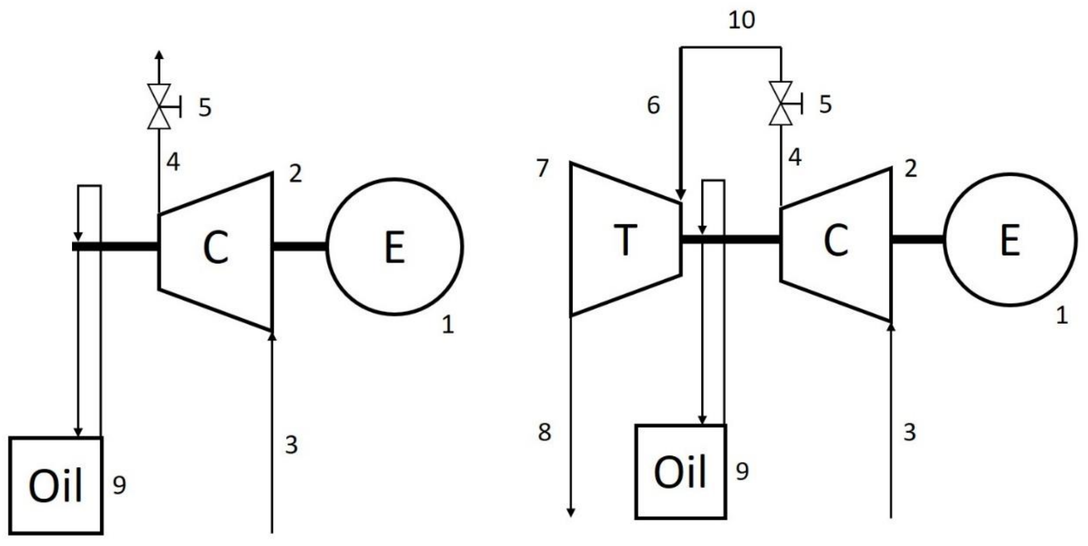
Experimental Performance Test Setup

In this chapter, the design of a new electric assisted turbocharger test setup will be presented. Each parameter to be measured is discussed individually, and the relevant design considerations with their fulfilling function are pointed out. Also the test strategy is presented. The design of the test setup is mainly focused to proof the estimated performance that will be discussed in the next chapter.

2.1 Introduction into the Experimental Setup

As mentioned in the introduction, in this work an electric assisted turbocharger test setup is being designed to measure compressor performance of the compressor side of the turbocharger. With the proposed concept of the performance test setup, the aim of this study is to prove that this strategy allows compressor map measurement extending by combining the use of the externally coupled electric motor with the circulation of flow through the turbine. The design of the test setup is based on the design of the electric-assisted turbocharger that is being developed at Van Der Lee Turbo Systems. With the test lay out, two performance tests will be conducted. The difference between these two performance test is that in the second performance test under the same input conditions the compressor outlet flow will be directed to the turbine inlet. This way, the turbine will be recovering some energy that is aimed to be used to drive the compressor operating point to a higher operating point in terms of power and/or rotational speed.

In figure 2.1, a schematic is shown of the test setup in the two different configurations. The physical difference between the two configurations is the connection between the compressor outlet and the turbine inlet. The turbine wheel, compressor wheel and electric motor will be mounted on the same shaft. In the inlet and outlet section of the compressor wheel the pressure and temperature will be measured to determine the compressor wheel performance. In table 2.1 all the general components of the test lay out are named by the corresponding number in figure 2.1.



A. *B.*

*FIGURE 2.1 – TWO CONFIGURATIONS OF THE ELECTRIC ASSISTED TURBOCHARGER TEST SETUP. **A** SCHEMATIC REPRESENTATION OF THE CONFIGURATION WITH ONLY THE ELECTRIC MOTOR. **B** SCHEMATIC REPRESENTATION OF THE CONFIGURATION WITH ELECTRIC MOTOR AND TURBINE.*

Number	Test Setup Components
1	Electric motor
2	Turbocharger compressor
3	Inlet section compressor
4	Outlet section compressor
5	Butterfly valve
6	Turbine inlet section
7	Turbocharger turbine
8	Outlet turbine section
9	Oil lubrication system
10	Connection compressor outlet and turbine inlet

TABLE 2.1 – COMPONENT LIST OF THE SCHEMATIC REPRESENTATION OF FIGURE 2.1

2.2 Compressor Performance Map

With the proposed performance test lay out, the compressor performance is aimed to be measured. The performance of turbocharger compressor wheel is given in datasets that can be visualized in so-called compressor maps. Every Compressor wheel has its own compressor map and this way it can be matched to the needs of the application. The compressor performance consists out of four performance parameters. The parameters are the pressure ratio, isentropic efficiency, corrected mass flow, and corrected compressor rotational speed. Some turbocharger manufacturers use instead of the mass flow the volume flow. A typical arbitrary compressor performance map is shown in figure 2.2 [9]. The global test procedure to generate a compressor map on a hot gas stand is to keep the rotational speed constant and vary the mass flow through the turbine. On the conventional gas stand the turbine is powered independently from the compressor. So the air flow through the turbine is controlled to control the rotational speed. On every speed line around in most cases around 6 to 8 operating points are measured between the limits of operating at that speed line [10].

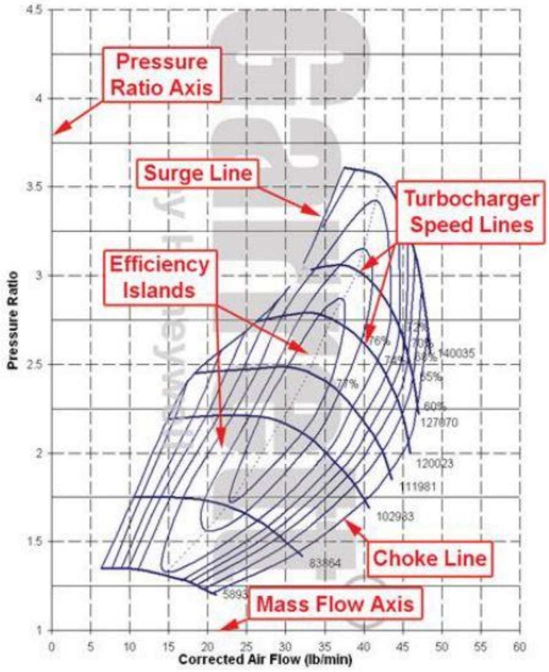


FIGURE 2.2 – EXAMPLE COMPRESSOR PERFORMANCE MAP OF A GARRET COMPRESSOR WHEEL WITH ASSOCIATED DESCRIPTIONS OF THE MAIN COMPONENTS [9]

On the X-axis of the compressor maps the corrected mass flow is displayed. The corrected mass flow is the measured mass flow through the compressor that is standardized using the inlet and a reference pressure and temperature for the compressor. The reference temperature is set by the SAE standard and is 298 K and same holds for the reference pressure which is 100 kPa. The inlet temperature should be the total temperature of the inlet flow. The total temperature is the static temperature and the dynamic temperature. In section 2.6.2 will be explained how this temperature can be measured. Correcting the performance parameters to the same reference temperature and pressure allows different compressor maps to be compared. The expression for the corrected mass flow is given in equation 2.1.

$$\dot{m}_{corr,c} = \dot{m}\,\frac{p_{ref}}{p_{comp,in}}\sqrt{\frac{T_{comp,in}}{T_{ref}}} \tag{2.1}$$

On the Y-axis the static to total pressure over the compressor is presented. This pressure ratio means that the compressor static outlet pressure is divided by the compressor total inlet pressure. The static pressure is the pressure measured perpendicular to the wall of the air flow. The total pressure consists of the static pressure and the dynamic pressure. The dynamic pressure is the kinetic energy of the air. The dynamic pressure depends on the air speed. The equation for the pressure ratio is given in equation 2.2.

$$\Pi_{Comp} = \frac{p_{static,out}}{p_{total,in}} \qquad 2.2$$

Like mass flow, also every measured rotational speed is corrected. This done in a similar way as for the mass flow and is given in equation 2.3.

$$N_{corr} = N_{Turbo}\sqrt{\frac{T_{ref}}{T_{comp,in}}} \qquad 2.3$$

The set of operating points will be measured between the limits of the compressor map which are called surge and choke. The surge line is the line most left of the compressor map. This is the lower bound for mass flow at a certain compression ratio. The surge limit is the area where the air flow becomes unstationary through the compressor which results in a high pressure oscillation across the compressor. This fluctuation can cause strong vibrations which could damage the turbocharger or lead to stall of the compressor wheel [1]. The lower limit of the compressor map is the choke line. At choke the flow reaches the velocity of sound at the impeller eye. This results in that no gas can be delivered through that section of the impeller [1, 9]. However some manufacturers define the choke as the line where the compressor efficiency drops below an unacceptable value [1, 9]. On the limits of the compressor maps in most cases a safety margin is taken. This margin differs for every manufacturer or test facility and is considered as confidential.

In some compressor maps also the efficiency islands are plotted. Each contour of an island presents a constant efficiency value. In most cases these are only provided as calculated data. The efficiency is determined by dividing the isentropic enthalpy rise across the compressor based on compression ratio by the actual enthalpy rise. This equation can be seen in equation 2.4. For this study the determining the efficiency of every point.

$$\eta_{comp,IS} = \frac{\Delta h_{comp,IS}}{\Delta h_{comp,AC}} = \frac{C_p T_{comp,in}\left(\Pi_{comp}{}^{\frac{\kappa-1}{\kappa}} - 1\right)}{C_p\,(T_{comp,out} - T_{comp,in})} \qquad 2.4$$

2.3 Electric-assisted turbocharger design

The base of the test lay out is the design of the electric-assisted turbocharger (e-turbo) that is being developed at Van Der Lee Turbo Systems. In figure 2.3, the mock-up is shown of the first version of the design where on the inlet of the compressor a donut-shaped nozzle is placed [4]. This is a non-functional mock-up that is based on the MHI (Mitsubishi) TF06 turbocharger and a 12 kW electric motor. In this concept of the e-turbo, the electric motor is connected to the compressor side of the turbocharger. The shaft of the electric motor will be connected to the compressor wheel nut to the turbocharger shaft.

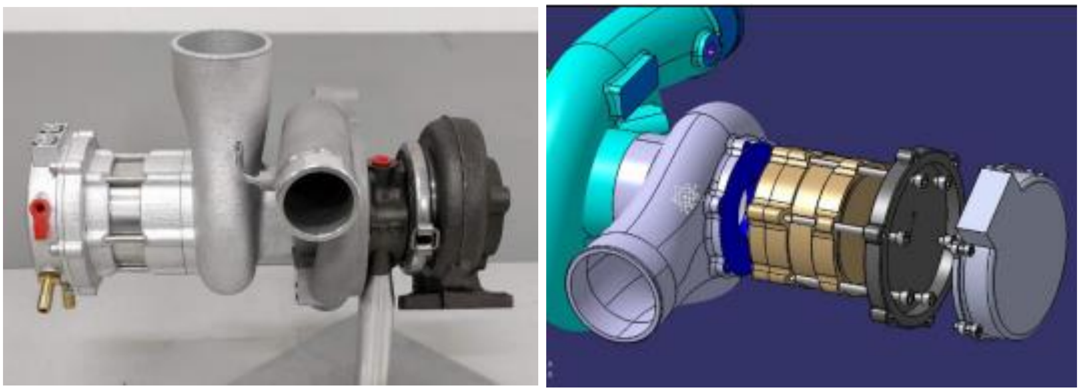


FIGURE 2.3 – FIGURES OF THE FIRST MOCK-UP OF THE ELECTRIC ASSISTED TURBOCHARGER. IN THIS DESIGN, A DONUT-SHAPED ATTACHMENT IS DESIGNED FOR THE COMPRESSOR ON WHICH THE ELECTRIC MOTOR CAN BE ASSEMBLED [4].

Recent research resulted in a change in the design of the electric-assisted turbocharger which is shown in figure 2.4. The compressor housing is changed such that the donut-shaped inlet is incorporated in the compressor housing. In the results of that research also the connection was redesigned in order to solve the issue of reaching the eigenfrequency of the rotating assembly (electric motor shaft, compressor wheel and turbine wheel) at around 168 000 RPM. The expectation is that in this study the rotational speeds of 168 000 RPM will not be reached. However by decoupling the electric motor and the turbocharger rotating assembly with a spline shaft, the eigenfrequency of the total rotating assembly will shifted to higher rotational speed [4]. At the start of the research, it was planned to proof the testing concept with the components of the original electric-assisted turbocharger design. However the electric-assisted turbocharger design is not yet finished and being realized. For the test setup another turbocharger and electric motor will be incorporated in the test setup. Therefore the design recommendations of the e-turbo will be taken into account for the test lay out.

FIGURE 2.4– FIGURES OF THE SECOND MOCK-UP OF THE ELECTRIC ASSISTED TURBOCHARGER. IN THIS DESIGN, A COMPRESSOR HOUSING IS DESIGNED WHERE THE DONUT SHAPED ATTACHMENT IS INCORPORATED IN THE COMPRESSOR HOUSING [4].

The first recommendation is that for the electric motor-turbocharger connection a donut-shaped inlet should be used that is based on the geometry on figure 2.4. During this study it was chosen to not design a new compressor housing with an incorporated donut inlet because of the focus lies on proof the way of testing. So a donut-shaped inlet similar to figure 2.3 will be used but then with the (scaled) geometry of the incorporated donut compressor housing. This way, an existing compressor housing can be used and the inlet could be 3D printed to reduce costs. The other recommendation is the shaft coupling between the electric motor and the turbocharger. The research about connecting the turbocharger and electric motor with a rigid connection showed that the problem of reaching the eigenfrequency during operation can be avoided using a spline shaft [4]. This spline shaft will on one end be connected to the compressor nut and on the other end on the electric motor spline output. This will result in a shaft that has on one end an outer spline that will be inserted in the compressor nut. On the other end the spline will have an inner spline for the electric motor.

2.4 Selection Turbocharger and Electric Motor

To measure a point in a compressor map and the range extension, a turbocharger (compressor and turbine) and electric motor should be selected. To reach a point in a compressor map three requirements should be met by the turbine and electric motor. The requirements for every operating point are listed below. The validation range depends on the combination of compressor of the turbocharger and the electric motor.

- $T_{available} \geq T_{Operating\ point}$
- $P_{available} \geq P_{Operating\ point}$
- $N_{available} = N_{Operating\ point}$

The main power source of the turbocharger test setup is the electric motor. Therefore the electric motor should be chosen properly. The choice of the electro motor is mainly depended on what motors are available and what the validation range is with the turbocharger options. The plug-and-play electric assisted turbocharger that Van Der Lee is developing makes it possible to adapt the design to different electric motors and compressor and turbine wheels. Therefore different combinations of compressor wheels and electric motors are plugged in the model that is described in the next chapter. In an iterative way, a combination was determined that was suitable for demonstrating range extension in the compressor map. The final choice of combination depended mainly on which electric motor and turbocharger were available. The minimum requirement for electric motors was that the lower speed line of the respective compressor map should be reached. This requirements are listed below.

- The electric motor should be able to reach an operating point of the bottom speed line.
- $T_{Electric\ motor} \geq T_{Lowest\ operating\ point}$
- $P_{Electric\ motor} \geq P_{Lowest\ operating\ point}$
- $N_{Electric\ motor} = N_{Lowest\ operating\ point}$

The first intension was to realize the test setup using the design of the original electric assisted turbocharger described in the previous section. This would mean an electric motor of 12kW and a 71mm compressor wheel would have been used. But during the project it appeared that the motor will not be available during the project time. Therefore alternatives were investigated. The considered options for the electric motor are listed in table 2.2. The different combinations that are considered are listed in table 2.3. An important note is that apart from the EM65 the other motors have a maximum rotational speed which is way lower. This will result in an alternative strategy in the prediction calculation. The prediction of the different combinations that will be presented in the next chapter and in the appendix.

MOTOR NAME	EM65	EM150	X520 3D
TYPE OF ELECTRIC MOTOR	Induction motor	Induction motor	DC brushless motor
PEAK MOTOR SPEED [RPM]	175000	26000	30000
PEAK POWER [KW]	12	20000	5280
PEAK TORQUE [NM]	0.7	7.35	9.55
CPSR (CONSTANT POWER SPEED RATIO)	1.1	1.1	-
VOLTAGE [V]	48	48	23

Table 2.2 – Electric motor options with their corresponding specifications. These are the 3 electric motors that are included as options. Other options were not worth mentioning.

	Electric motor	Compressor wheel diameter	Turbine wheel diameter
1	EM65	71 mm	63 mm
2	EM150	78 mm	74 mm
3		72 mm	70 mm
4	EM150	170 mm	-
5	AquaStar T20 X520 3D	78 mm	74 mm

TABLE 2.3 – ALL ELECTRIC MOTOR, COMPRESSOR AND TURBINE OPTIONS THAT HAVE BEEN CONSIDERED. THESE ARE THE COMBINATIONS THAT HAVE BEEN TRIED IN THE MODEL. THE COMPRESSOR MAP RESULTS OF THESE COMBINATIONS ARE SHOWN IN THE NEXT CHAPTER OR IN THE APPENDIX.

The combination selected is the EM150 with a 78 mm compressor wheel. This option does not meet the requirement for the lowest compressor speed line but for a long time this seemed to be the only motor option available. It was therefore chosen to demonstrate the principle of range extension in the area below the lowest compressor speed line. Since the lowest operating point cannot be reached with the electric motor, a gear transmission was considered. This is also included in the forecast model. In the end it was decided not to go for a gearbox. By using gears to achieve the desired rotation speeds, complexity is added to the test set-up. Because of the high rotational speeds, it was decided not to add a mechanical transmission, thus reducing the risk of mechanical failure.

2.5 Test setup Design

With the test lay out the general goal is to measure a part of the compressor performance map in order to investigate the validation range. For the design choices the test rig will be discussed in the second configuration. That is the configuration where the turbine is used to recover power. A top view of the total design can be seen in figure 2.5.

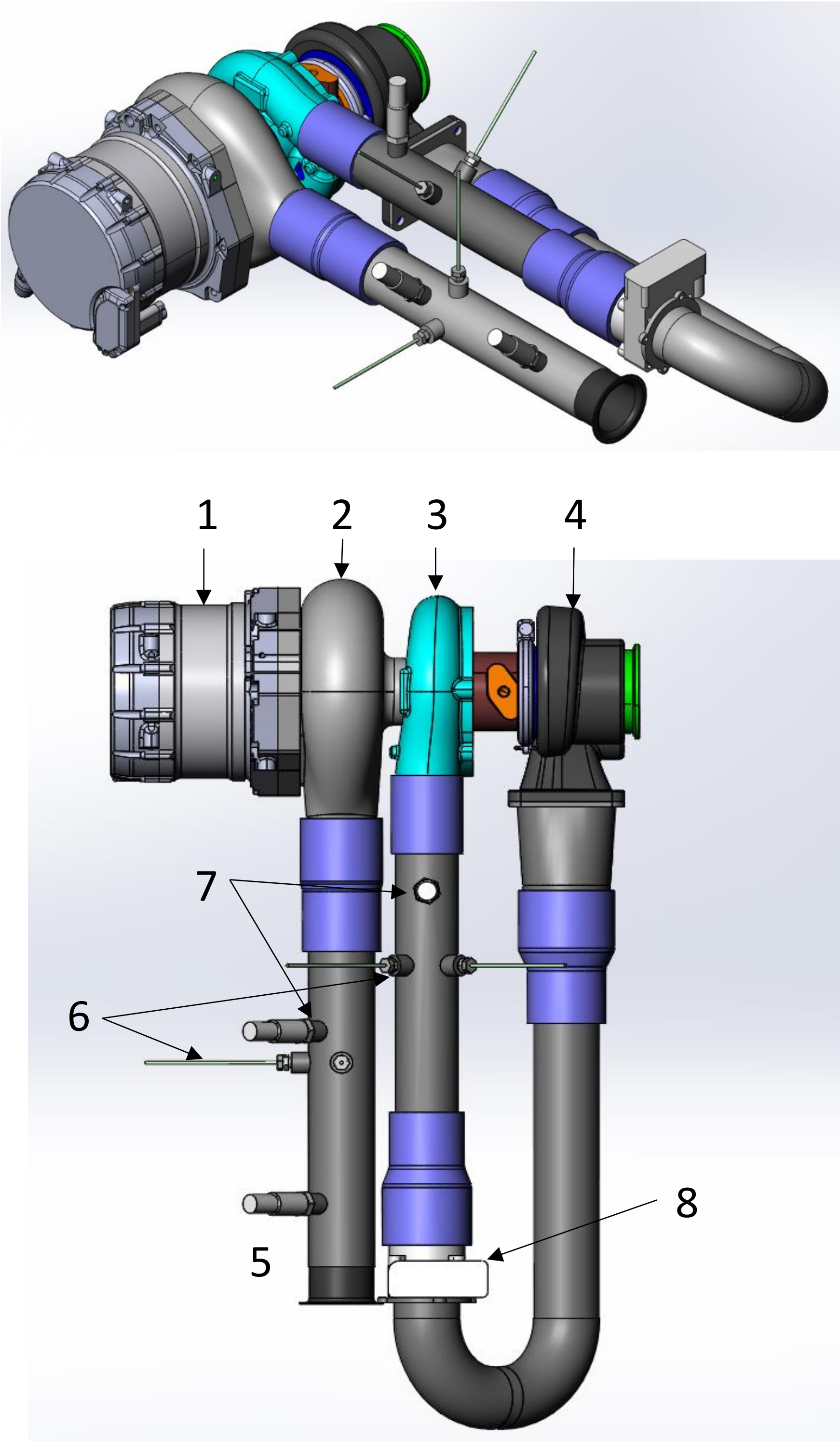


FIGURE 2.5 – 1 ELECTRIC MOTOR, 2 COMPRESSOR INLET COVER, 3 COMPRESSOR, 4 TURBINE, 5 FLOW MEASUREMENT SECTION, 6 TEMPERATURE MEASUREMENT SECTIONS, 7 PRESSURE MEASUREMENT SECTIONS, 8 VALVE SECTION.

The base of the test setup is the turbocharger. In figure 2.5 the two parts of the turbocharger are indicated with two numbers because it consists of the compressor and the turbine. Those are the number 3 and 4. The number 3 is the compressor housing of a compressor wheel with the diameter of 78 mm. The turbine that is indicated with the number 4 is a 72 mm turbine wheel with his corresponding housing. This turbine and compressor are a matched set of wheels for the application of an aircraft diesel engine. The turbine is connected with a V-band. This makes it easy to switch to the other configuration of the test setup by remove the turbine housing. At the inlet of the compressor a 3D printed inlet is designed. The inlet is a donut-shaped inlet that is based on a preliminary research that has been done about the electric assisted turbocharger [4]. The design is taken from this research. In this research the recommendation has been done that a width increase would benefit the flow. Therefore the donut-shaped is scaled to increase the width and keep the flow conditions similar to the flow analysis in the work of Herrema [4]. The inlet diameter and the outlet diameter of the inlet donut are kept the same. The inlet donut is the connection between the electric motor and the turbocharger. The electric motor shaft is connected to the turbocharger shaft with a spline shaft. The spline shaft is in this set up made of steel. For the real turbocharger it will be made of an engineered plastic to reduce the moment of inertia of the rotating assembly. The plastic shaft is not yet used in the test setup because of the fatigue torque of the plastic shaft is not yet tested in the application conditions. Therefore is chosen for the save option for now. The spline shaft connection is chosen to decouple the eigenfrequencies of the electric motor and the turbocharger shaft [4]. A section view of the turbocharger can be seen in figure 2.6. The spline shaft is the yellow coupling that is connected to the compressor wheel via the compressor nut in purple. In the rotational speed range of the test rig the eigenfrequencies are not critical. The total design is considered as a good first prototype for the actual electric assisted turbocharger.

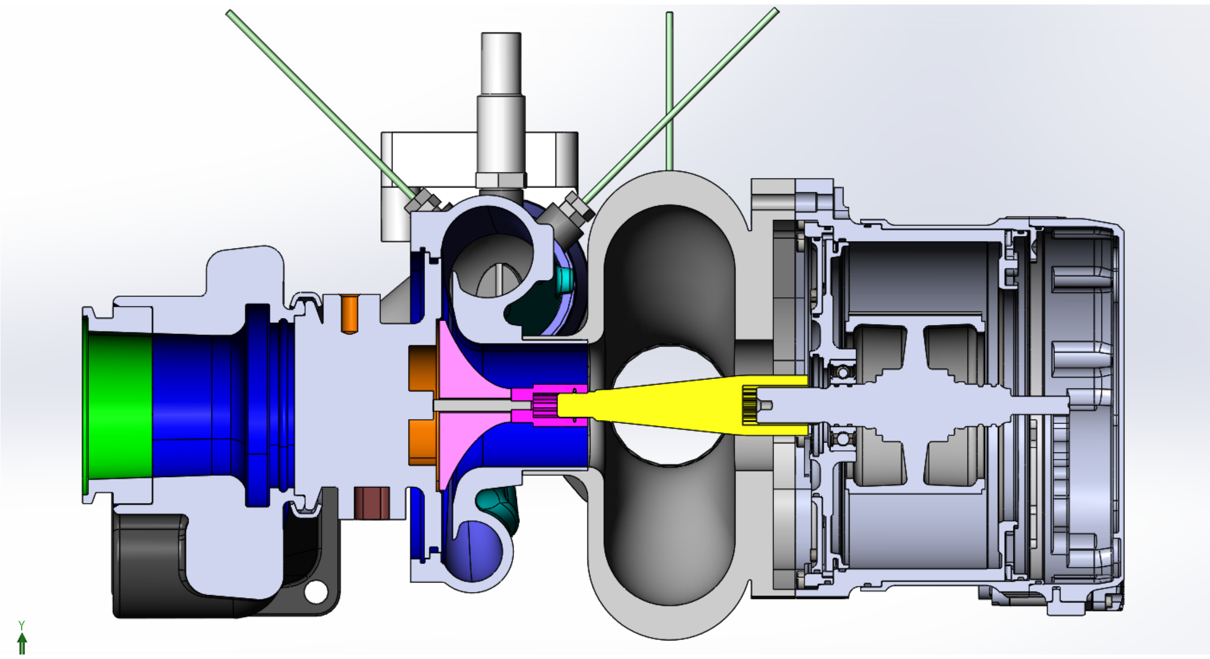
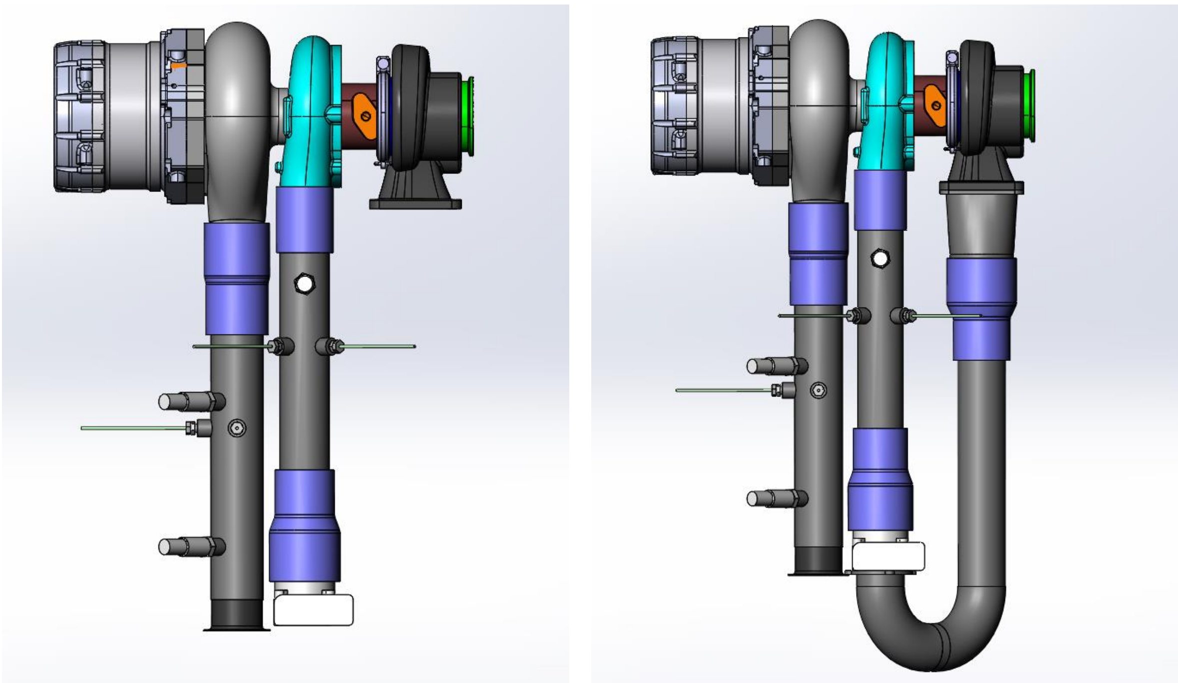


FIGURE 2.6 – CROSS SECTION OF THE TURBOCHARGER WITH ELECTRIC MOTOR TO SHOW THE SPLINE CONNECTION. THE SPINE SHAFT IS SHOWN IN YELLOW AND THE COMPRESSOR SPLINE NUT IS SHOWN IN DARK PINK. THE TURBINE WHEEL IS NOT SHOWN IN THIS CROSS SECTION. THE DESIGN IS BASED ON FINDINGS FROM PRELIMINARY RESEARCH. THIS IS THE CROSS SECTION WITH THE EM150 MOTOR

In the design of the original electric assisted turbocharger a 12 kW electric motor incorporated. This electric motor is not available due during the time of this study. An alternative was not easy available. Therefore it is chosen to select the EM150 which is an 20kW motor that can reach a speed up to 26 000 RPM. This is a 48 V electric motor. The power supply and electronics were said to be arranged by the motor manufacturer. The electric motor will be connected via bolts to the turbocharger. Due to the motor size, the inlet section of the compressor is scaled up. Herrema's study states that a larger donut to send the air into the compressor improves the intake conditions [4]. The inlet flow will have less turbulent behaviour.

At the inlet section of the compressor a pipe is mounted with the same diameter as the compressor inlet. The same diameter is chosen so the flow at the inlet section can be considered the same as at the inlet of the compressor. In this sections the measurements will be taken to determine the inlet flow conditions. In the next sections the design choices of the measurement sections such as the sensor choice, positions, and configurations will be discussed in more details. In figure 2.5, the number 5 is the position of the flow sensor. The number 6 indicate where the flow temperatures are measured and at the positions of number 7 the flow pressures are measured.

At the outlet section of the compressor between the turbine, at location 8 a butterfly valve is placed. This butterfly valve will be an electric throttle valve. This valve is placed after the compressor to put backpressure on the compressor and control the mass flow [11, 12, 13, 14]. The pipe section that is connected downstream the valve can be removed to change in configuration. In figure 2.7 two top views are shown of the two configurations. On the left side the first configuration is shown where only the electric motor will be powering the compressor and on the right side the second configuration is shown where both the electric motor and the turbine will power the compressor. In the figure on the left (the first configuration) also the turbine housing will be removed.



A. ***B.***

*FIGURE 2.7 – **A** VALIDATION CONFIGURATION WITH ONLY THE ELECTRIC MOTOR. IN THE CAD-FILE, THE TURBOCHARGER WAS A SINGLE PART. THEREFORE THE TURBINE HOUSING IS NOT REMOVED IN THE CAD-SOFTWARE. IN THIS CONFIGURATION HOWEVER THE TURBINE HOUSING WILL AND CAN BE REMOVED FROM THE TURBOCHARGER.. **B** VALIDATION CONFIGURATION WITH ELECTRIC MOTOR AND TURBINE. IT IS EXPECTED THAT WITH THE SECOND CONFIGURATION (B) A LARGER RANGE CAN BE MEASURED IN THE COMPRESSOR MAP THAN WITH THE FIRST CONFIGURATION (A). THE EXPECTED DIFFERENCE BETWEEN THE TWO COMPRESSOR MAPS CAN BE SEEN LATER IN E.G. FIGURE 3.9.*

2.6 Parameter Measurement Sections and Instrumentation

To measure the desired parameters to generate (a part of) a compressor map, sensors should be placed such that the performance data can be measured. In this section, for both the test lay outs, the design of the measurement section is sketched and the design considerations are discussed. In this section only the final design solution is presented, for other options, a brief argumentation as to why they have not been chosen

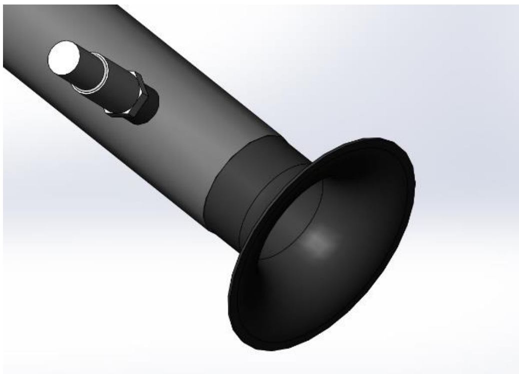
To compute these performance maps, at the compressor inlet the static and total pressure and the total temperature should be determined. Also the air flow velocity should be determined in the compressor. The same holds for the compressor outlet where the static pressure and the temperature should be measured. Next to these flow conditions, also the rotational speed of the turbocharger shaft should be measured. For the choice of the position and the configurations to measure these parameters, the SAE J1826 turbocharger gas stand test code and the performance mapping guidelines of a third party are taken into account as advice [10, 13, 15]. These are the prescriptions of how a turbocharger performance test should be carried out.

For all the parameters the desired accuracy of the measurement is determined. The measurements will be converted to the values that are used to generate the compressor map. The measured performance map will be compared to the predicted result. For the volume flow or flow velocity an accuracy of 3% would be accurate enough to make aa representative comparison. The temperature would like to be measured around 1 degree. The pressure is desired to be measured with an accuracy of 1000 kPa. However these accuracy values are guidelines. The choice of the sensor depends also on what is available. It is not the goal to measure a compressor map that is similar to the map that it measured on a hot gas stand. The repeatability of the measurement is more important, to compare relative measurements with each other.

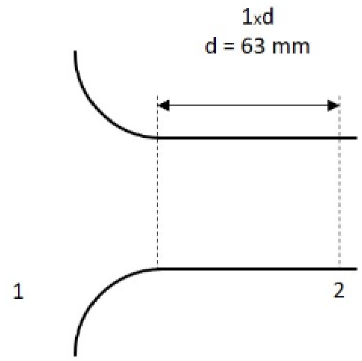
2.6.1 Flow Measurement

The first flow parameter that is discussed is the flow velocity. With the flow velocity, also the volume flow or the mass flow can be determined. In turbocharger test stands the flow velocity is in most cases measured by making use of differential pressure. In such a configuration, the pressure is measured then before and after for example a V-cone, orifice plate or a Venturi tube [2, 9]. For the proposed system it is chosen to measure the flow based on differential pressure over the inlet trumpet [11]. The other option that was considered to measure the flow velocity or volume flow were the mass air hot film sensors (also known as MAF sensor) or V-cone or orifice plate. The MAF sensors are used in vehicles. However these sensors were not recommended because in practice the accuracy of those sensors is estimated on 10%. Chosen is to measure the pressure difference over the inlet trumpet by taking the ambient pressure as reference pressure. This means that there is no additional component in the pipe that would cause a large head loss. Also in price this option is considered more appropriate.

In figure 2.8A a close up is shown of the design of the flow measurement section. As mentioned the Bernoulli equation is used to determine the flow velocity over at a location in the pipe. In figure 2.8B a schematic is shown of the design that indicates location 1 and 2. It is said that the Bernoulli equation may be used along a flow line under laminar conditions. However is found that in practice for Mach number lower as 0.2 the Bernoulli equation is still used [16]. The Bernoulli equation can be found in equation 2.5. Location 1 in figure 2.6B is the ambient conditions. These conditions are stated in table 3.1. Besides the ambient pressure also the ambient temperature is used in order to determine the density of the air. The location 2 is the location in the pipe where flow velocity will be determined. This location is located 1 diameter from the inlet. At this location a wall tap is made where the static pressure will be determined using a pressure transducer. The wall tap will be 1 mm hole where a weld bung is welded to mount the sensor on the pipe. With the static pressure at location 1 and 2 and the density of the air the flow velocity can be determined with the rewritten Bernoulli equation 2.6. The flow velocity at location 1 in front of the inlet is assumed 0 m/s. With equation 2.7 and 2.8, the volume and the mass flow thought the compressor can be determined. This is done using the cross area of the pipe and the density of the air.



A.



B.

*FIGURE 2.8 – **A** ZOOMED IN ISOMETRIC VIEW OF THE FLOW MEASURING SECTION. **B** SCHEMATIC OF THE FLOW MEASURING SECTION*

$$p_1 + \frac{1}{2}\rho V_1 = p_2 + \frac{1}{2}\rho V_2 \qquad 2.5$$

$$V_2 = \sqrt{\frac{2}{\rho}(p_1 - p_2)} \qquad 2.6$$

$$\dot{Q} = V_2 A_{pipe} \qquad 2.7$$

$$\dot{m} = \rho \dot{Q} \qquad 2.8$$

To use this method of determining the flow rate, it is important to keep the inlet head losses as low as possible. By using an inlet trumpet, the entrance loss can be minimised. Besides this, an estimation of the pressure loss can be made with the help of figure 2.9 in White's book.This figure shows the loss coefficient for a ratio between the diameter and the trumpet radius. This can be included in the Bernoulli equation as in equation 2.10. The trumpet would be a plastic insert into the pipe. The diameter of the pipe 55 mm and the radius is 30 mm. According to lowest line of figure 2.9, this is equivalent to a K-coefficient smaller than approximately 0.02. Which is will result in a small loss that is acceptable.

$$h_m = \frac{V_2^2}{2g} K \qquad 2.9$$

$$p_1 + \frac{1}{2}\rho V_1 = p_2 + \frac{1}{2}\rho V_2 + h_m \qquad 2.10$$

Combining those two equations and make use of the ABC formula to determine a expression for the flow velocity at location 2 the expression would be equal to equation 2.11. This is the outcome that will result in a positive flow velocity into the compressor.

$$V_2 = -\frac{g\rho - \sqrt{(4K^2p_1 - 4K^2p_2 + g^2\rho^2)}}{2K} \quad 2.11$$

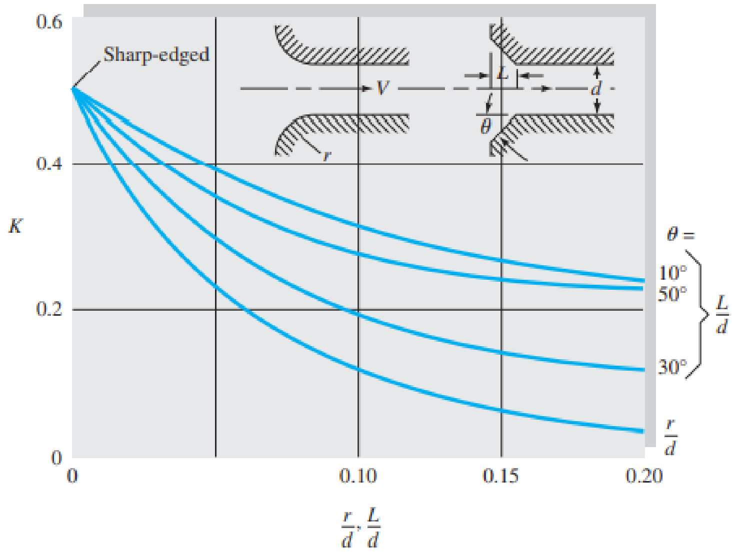


FIGURE 2.9 – LOSS COEFFICIENT K VERSUS THE RATIO BETWEEN TRUMPET RADIUS AND PIPE DIAMETER [16]. FOR FLOW MEASUREMENT, THE CONFIGURATION SHOWN ON THE LEFT-HAND SIDE OF THIS FIGURE (ROUND TRUMPET) WAS USED. THE LOWEST LINE IN THE FIGURE CORRESPONDS TO THIS CONFIGURATION. THIS FIGURE IS USED TO ESTIMATE THE LOSS COEFFICIENT K TO CORRECT FOR THE MEASURED FLOW VELOCITY.

2.6.2 Temperature Measurement

Another parameter that needs to be measured is the total temperature at the inlet and outlet of the compressor. The temperature sensor that is commonly used during turbocharger performance mapping are thermocouples. The other option that is commonly used are thermistors (so-called Pt100 sensors). For the purpose of this test setup thermocouples of the type K are suitable for the application. The main reason for the choice of thermocouples is at they are more robust and price effective than thermistors. This thermocouple has a temperature range up to 1200 °C and it is stated to have an accuracy value around ±1.5°C. Two thermocouples are used so an average can be taken to have a more reliable measurement. Those two sensors are placed between 2 to 3 diameters from the in- and outlet of the compressor housing as described in the SAE-norm [10, 13, 15].

Since the temperature is not measured by means of total temperature probes, the measured temperature does neither represent the static nor the total temperature. The temperature that is measured is known as the stagnation temperature. In Literature, two ways have been found to convert the measured temperature to the total temperature. Both methods will be presented because there is a difference in between the methods but why one is better as the other one is not found. Therefore will this be tried with the test data.

The first method places the thermocouple flush to the pipe wall. Then the total temperature can be calculated with the relation in equation 2.12 [16].

$$T_{aw} = \frac{T_{total}\left(1 + rec\frac{(\kappa-1)}{2}M^2\right)}{1 + \frac{\kappa-1}{2}M^2} \quad 2.12$$

The temperature recovery factor in this relation is defined by equation 2.13. For the temperature interval from ambient to 220 °C the recovery factor is assumed to be constant. The recovery factor is approximated using the Prandtl number for turbulent flow conditions. For air at ambient conditions the Prandtl number is around 0.7 that leads to an recovery factor of around 0.89.

$$rec = \frac{T_{prob} - T_{static}}{T_{total} - T_{static}} = \sqrt[3]{Pr} \qquad 2.13$$

In the other method the thermocouple is placed such that it sticks in the flow. In the SAE J1826 1/3 of the pipe diameter is recommended. Using the same temperature recovery factor and the flow velocity the measured temperature with the thermocouple can be converted to the static temperature [18].

$$T_{static} = T_{aw} - rec\frac{V_2^2}{2\, c_p}\ Where\ c_p = \frac{\kappa R}{\kappa - 1} \qquad 2.14$$

With this static temperature the total temperature can be calculated based on a relationship with the Mach number which is given in equation 2.15.

$$\frac{T_{total}}{T_{static}} = 1 + \frac{\kappa - 1}{2} M^2 \qquad 2.15$$

2.6.3 Pressure Measurement

The last flow parameter that should be measured at the in- and outlet is the total pressure. The total pressure is measured using piezoresistive pressure sensor that measures the static pressure. For measuring the static pressure different type of pressure transducers can be used. In consultation with Jos van Driel, a pressure sensor was selected that met the application based on the pressure range, accuracy and readout possibilities. To measure the total pressure of an air flow also total pressure probes are available. Chosen is not to use such total pressure probe because this probe sticks in to the flow which can lead to an undesired disturbance. With equation 2.16 the static pressure can be converted to the total pressure in the pipe by making use of the Mach number. The static pressure is measured with pressure transducers that is mounted on a 1 mm hole with wall mount. At the inlet the pressure is measured at the 1.5 diameters from the inlet of the compressor housing. The pressure transducer that is used to measure the pressure is an gauge pressure sensor that has a pressure range up to 10 bar and an accuracy of 1% The outlet static pressure is measured in a similar way at 1.5 diameters from the compressor outlet and will be measured with the same sensor as at the inlet [10, 13, 15].

$$\frac{p_{total}}{p_{static}} = \left(1 + \frac{\kappa - 1}{2} M^2\right)^{\frac{\kappa}{\kappa - 1}} \qquad 2.16$$

2.6.4 Rotational Speed Measurement

The measurement of the rotational speed of the turbocharger shaft is made by a speed sensor. This speed sensor is mounted in the housing of the compressor of the turbocharger. The rotational speed sensor is measured with a Picoturn sensor. The Picoturn sensor is a sensor that works based on eddy currents Every time a blade of the wheel passes by, the sensor will give a pulse. It is given that with this Picoturn PT2G-SM5.5 sensor, a maximum speed of 400000 RPM can be measured. A data acquisition system must ensure that pulses are counted. The sensor is positioned at an angle of 45 degrees to the compressor wheel. The distance between the blade and the sensor must be about 1 mm. To reduce costs the NI-USB 6211 is used. This is a data acquisition system from National Instruments that was available from TU Delft. With a program written in Labview, the measured signal is converted into the rotation speed of the compressor wheel. In this program the sampling frequency can be adjusted according to what the maximum measured rotation speed will be. The maximum rotation speed that can be measured is estimated at 40,000 rpm. Each rotation of the compressor wheel will consist of 12 pulses through the 12 compressor blades, so the measured frequency will be 12 times higher than the rotation frequency. In addition, the rule of thumb is that the sampling frequency must be 2.5 times the maximum frequency. This would mean a sampling frequency of 20,000 Hz should be enough. In the end, in consultation with Jos van Driel and with the help of small experiments it was determined that with a sampling frequency of 100,000 Hz every rotation speed can be measured well within the experiments. The Labview script and the test measurements can be found in appendix.

2.7 Test Strategy

As no alternative electric motor options were available, the EM150 was chosen in the test setup design. This electric motor can reach a maximum speed of 26,000 rpm while the lowest speed line of the 78 mm compressor wheel is around 50,000 rpm. In order to demonstrate the principle of increasing the validation range with the turbine, the following method was devised as a test strategy.

The aim is to demonstrate, within the range of the electric motor, that with the turbine side of the turbocharger a larger range can be tested in the compressor map. This is done by selecting a relatively low RPM operation point. In the first configuration (figure 2.7A), this operation point will only be controlled by the electric motor. When the point is determined in the compressor map and the input that is given to the electric motor is known, the test setup is converted to the second configuration. In the second configuration (figure 2.7B), the same input is given to the electric motor to reach the same operation point. However, the turbine will now recover energy so that the operation point will shift. It is expected that the original determined operating point will either move to a higher mass flow or to a higher speed. Where the new operating point will actually shift to, will depend on the position of the electronic valve.

2.8 Summary Design

The design of the test setup will consist of an 20 kW electric motor, a 78 mm compressor wheel and a 74 mm turbine wheel. At the inlet of the compressor a pipe will be connected with a diameter of 60 mm. At the inlet of that pipe the flow will be measured using differential pressure between the pipe and the ambient conditions using Bernoulli's equation. At 2.75 diameter from the inlet of the compressor the temperature is measured using two thermocouples. At 2.25 diameters form the compressor inlet the inlet pressure is measured. The outlet of the compressor is connected to a 60 mm pipe as well. At the same distances and configurations the pressure and temperature are measured form the compressor outlet. In the pipe a valve is placed to control the mass flow though the compressor. The turbine housing and pipe section from the valve to the turbine can be removed to change to the other configuration.

3

Test Setup analysis for Performance Prediction

In this Chapter, an analysis is made of the test setup in order to get a better understanding to predict the test setups performance. Therefore a model is set up. With a matched compressor and turbine wheel the operation points that can be reached with the potential test setup are estimated.

3.1 Model Introduction

To better understand how the designed test setup from the previous chapter works, a model is set up in Matlab. The test setup will consist of the electric motor, the compressor and the turbine. The building blocks of the model are the performance characteristics of the compressor, turbine and electric motor. This means that the performance maps of the turbine and compressor from table 2.3 are the input for this model. The characteristic values of the electric motor are also used as input. Each of the three components has its own sub-model which gives as output the torque and power level of a set of operation points. The torque and power values of the compressor will be compared with those of the electric motor and turbine. This will result in a compressor map where two validation ranges will be visualized. One of the validation range can be reached by using only electric motor power. The other operating range can be reached by also recovered compressor power using the turbine. For the operating points that are within reach of the electric motor, the following conditions should hold for every operating point.

$$P_{emotor,OP} \geq P_{Comp,OP} \qquad 3.1$$

and

$$\mathrm{T}_{emotor,OP} \geq \mathrm{T}_{Compr,OP} \qquad 3.2$$

For operating range that can be reached with electric motor and turbine the following conditions should hold. The only difference with the conditions above is that now the sum of power and torque of the electric motor and turbine should be bigger than the compressor power and torque.

$$P_{emotor,OP} + P_{turb,OP} \geq P_{Comp,OP} \qquad 3.3$$

And

$$\mathrm{T}_{emotor,OP} + \mathrm{T}_{turb,OP} \geq \mathrm{T}_{Compressor,OP} \qquad 3.4$$

For these conditions, the efficiencies of the compressor and turbine at every operating point are taken into account. The electric motor efficiency is assumed to be 100%. Later this could be changed to get a result closer to real conditions. The option of implementing a gear is also included in the model. Although this will not be used in the test setup, it will provide insight into what the maximum validation range will be with the given motor. The efficiency used for the gears is 95%. In addition, it is assumed that no energy will be lost in the airflow in the pipe.

As mentioned, different combinations of the turbocharger wheels and electric motors were introduced into the model. To explain the model, the combination is used which is intended for testing. This was the EM150 as electric motor, the 78mm compressor wheel and the 74mm turbine wheel. Other outputs or (intermediate) results of the model with other combinations of wheels and motors can be found in the appendix.

3.2 Performance Model – Compressor Part

For the compressor part of the model, the performance map of the compressor wheel is used as input. In figure 3.1 the compressor map of the 78 mm compressor wheel is shown. As mentioned in the previous chapter, each operating point in the map has four parameters. To calculate the power that is needed to operate at a desired compressor operating point, first the temperature difference is calculated over the compressor. This temperature difference is determined using the equation 3.5 which makes use of the pressure ratio, efficiency and the specific heat coefficient. To calculate the temperature difference over the compressor the condition of the inlet air should be known. The ambient conditions of the air are shown in table 3.1. It is assumed that the compressor will compress air at ambient conditions at 294 Kelvin.

$$\Delta T_{comp} = \frac{1}{\eta_{comp,IS}} T_{amb} (\Pi_{comp}^{\frac{\kappa-1}{\kappa}} - 1) \quad 3.5$$

With the temperature difference and mass flow at every operating point the needed power can be calculated for each point. The mass flow of every operating point is presented in the performance map data.

$$P_{comp} = C_{pa} \Delta T_{comp} \dot{m}_{comp} \quad 3.6$$

For every operating point also the torque that is needed to drive the compressor can be calculated. This can be done based on the compressor power of equation 3.6 and the rotational speed.

$$\mathrm{T}_{comp} = \frac{P_{comp}}{\frac{2\pi}{60} N_{turbo}} \quad 3.7$$

Also the compressor outlet air conditions are calculated. These are used in the turbine model as input. The pressure is calculated using the pressure ratio of the compressor and the outlet temperature is calculated by adding the corresponding temperature rise.

$$T_{Comp,out} = T_{amb} + \Delta T_{comp} \quad 3.8$$

$$p_{comp,out} = p_{amb} \Pi_{comp} \quad 3.9$$

Ambient condition parameter	Value	Description
$\boldsymbol{p_{amb}}$	$101325\ \mathrm{Pa}$	Pressure
$\boldsymbol{T_{amb}}$	$294.15\ \mathrm{K}$	Temperature
$\boldsymbol{\rho_{amb}}$	$1.225\ \mathrm{kg/m^3}$	Density
$\boldsymbol{\mu_{amb}}$	$1.82 \cdot 10^5 \mathrm{Pa\ s}$	Dynamic viscosity
$\boldsymbol{\nu_{amb}}$	$1.516\ \mathrm{m^2 s}$	Kinematic viscosity

TABLE 3.1 ASSUMED INLET AIR CONDITIONS (AMBIENT CONDITIONS) THAT ARE USED IN THE PREDICTION MODEL

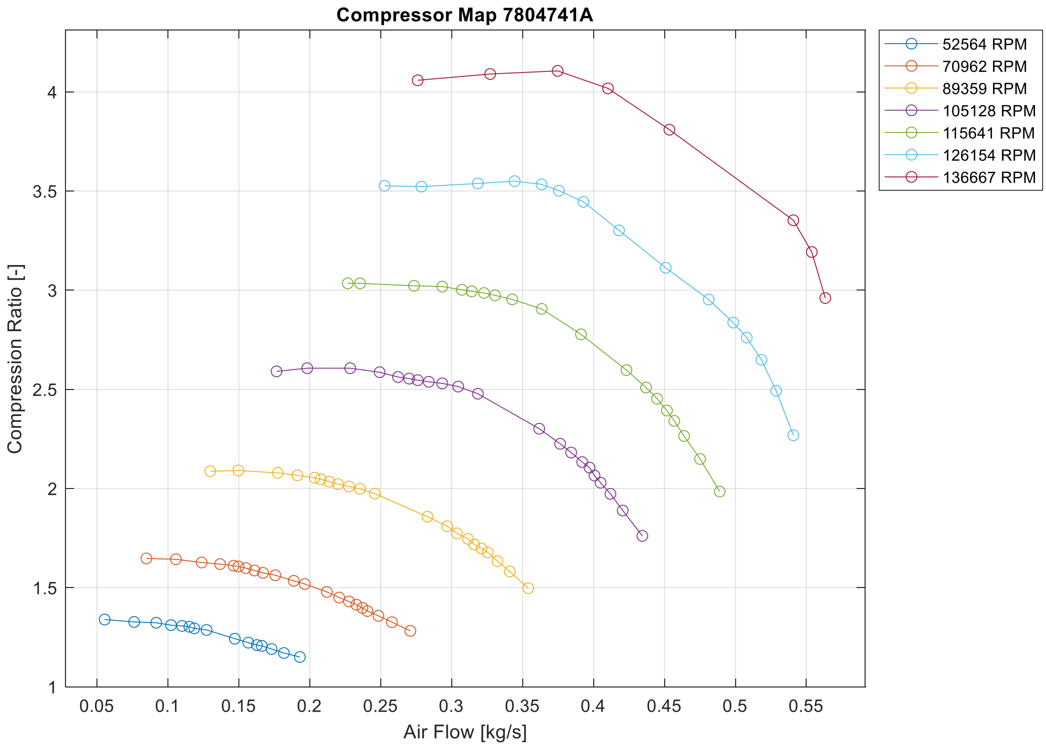


FIGURE 3.1 – COMPRESSOR MAP OF THE 78 MM COMPRESSOR WHEEL. THIS COMPRESSOR MAP IS THE INPUT FOR THE PREDICTION MODEL AND IS IN THE END USED IN THE TEST SETUP DESIGN.

3.2.1 Compressor map extrapolation

When the model was made, it was known that some electric motors could not operate at the rotation speed of the lower speed line of the performance map. For example the EM150 is only capable of running at a rotational speed of 26,000 RPM where most compressor performance maps start at 50,000 RPM. Therefore, a Matlab function is written that could plot an operation line in the compressor map based on the existing performance data. The input for this function is the desired rotation speed for the new operating line and all compressor map data. First is checked if the new line is below the lowest speed line, between all existing speed lines or above the highest speed line. Based on this a factor is calculated how far the new speed line is from the nearest existing speed line in the map. If the desired speed line is below the lowest speed line in the available compressor data, an extrapolation is made between the lowest speed line and the zero point. The zero point is at no pressure ratio and no mass flow. The new speed line is placed linearly between these two points. If a new speed line lies within the range of the existing compressor data, the same thing happens as in the case just described. Only the new line is linearly interpolated between the existing speed lines. The closest upper and the closest lower speed line are selected. In a similar way the new speed line is located between these two speed lines as is done in the lower limit case. In the last case where a line is higher than the available maximum speed line then the highest two speed lines are used to determine where this speed line will be located. The shift of the speed line in all three cases can be characterised by the equation 3.10. The shift is always viewed from the speed line below it.

$$Shift = \frac{N_{desired} - N_{comp,low\ lim}}{(N_{comp,high\ lim} - N_{comp,low\ lim})} \qquad 3.10$$

To determine the new line, the number of data points of the two nearest lines was first considered. For the method chosen, it is important that these two speed lines have the same number of data points. If this is not the case, the speed line with the least number of data points will be chosen as reference. This means that the new speed line will have the same number of data points as this line. Every new operating point will be calculated by scaling a triangle with the factor of equation 3.10. The length of the hypotenuse which is the length of the line between every operating point is calculated using equation 3.11. This length is determined using a simple Pythagorean theorem.

$$L = (\Pi_{point\ 2} - \Pi_{point\ 1})^2 + (\dot{Q}_{point\ 2} - \dot{Q}_{point\ 1})^2 \qquad 3.11$$

The position of the new operating point is determined by scaling the hypotenuse.

$$L_{new\ OP} = shift * L \qquad 3.12$$

With this scaled triangle the shift in mass flow and pressure ratio is known relative to the lower reference speed line. This determines the new position in pressure ratio and mass flow of one operating point on the speed line. For every point on the new rotational speed line this is done in order to plot an extra speed line. This has been repeated for every operational point on the new speed line. For the efficiency of the new operational point a worst case is taken into account. Therefore the new operating points have the same efficiency values as the relative lower speed line. In figure 3.2 is a visualisation to show how this works in the compressor map.

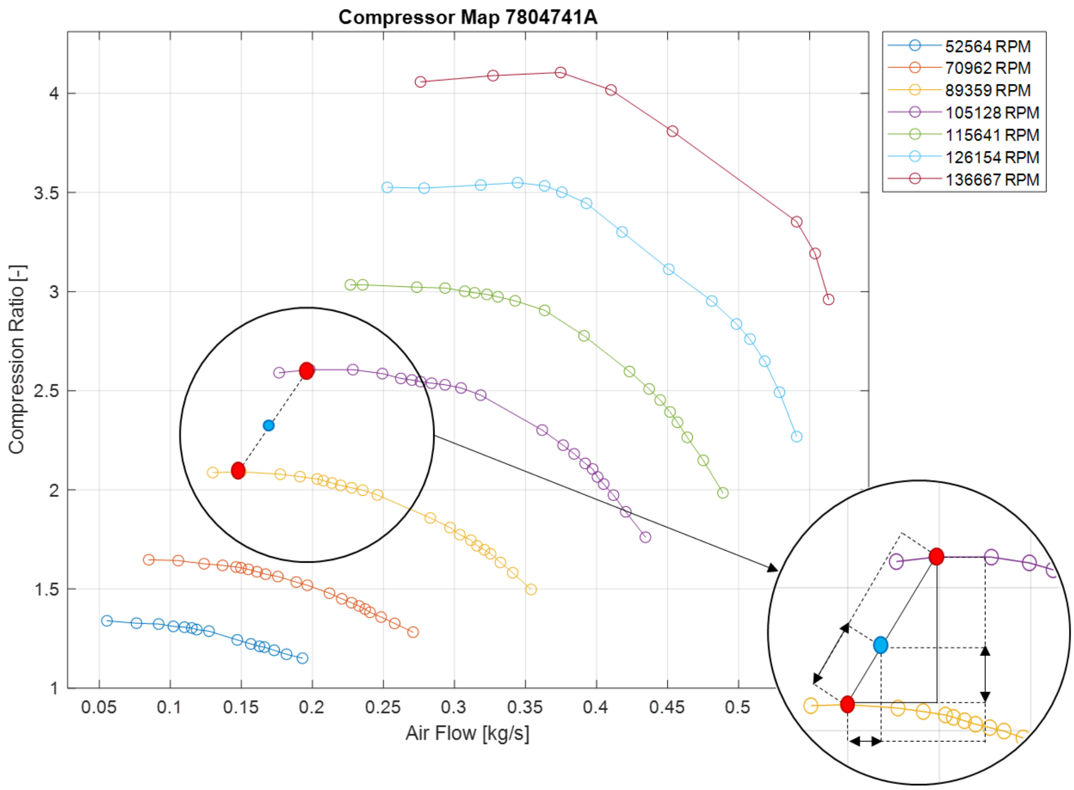


FIGURE 3.2 – COMPRESSOR MAP WITH THE VISUALISATION HOW A NEW OPERATING POINT IS POSITIONED. FOR THE NEW OPERATING POINT A SHIFT IS CALCULATED BETWEEN THE TWO NEAREST KNOWN OPERATING SPEEDS. THEN THE POINT IS SHIFTED ALONG THE SLOPE OF A LINE BETWEEN THESE POINTS. THIS IS DONE FOR A NUMBER OF POINTS TO CREATE A NEW OPERATING LINE

For the compressor map of the 78 mm that will be used with the EM150 electric motor is chosen to extrapolate operation lines at the bottom end of the compressor map. In figure 3.3 is shown what the result is of the new compressor map with 8 extra operating lines in the low rotational speed range. Important to notice is that some of the operating points at lower rotational speeds will not be a feasible operating point. Especially at the lower mass flows on the extrapolated bottom speed lines, some points will be located in the choke region. It is said that for the choke and surge area in the lower region the choke and surge line follow more the less the trend of the known data points [19]. A rough indication is given for those lines with the red dotted lines in figure 3.3. However the expected choke points have not been removed because this criteria was hard to implement [19].

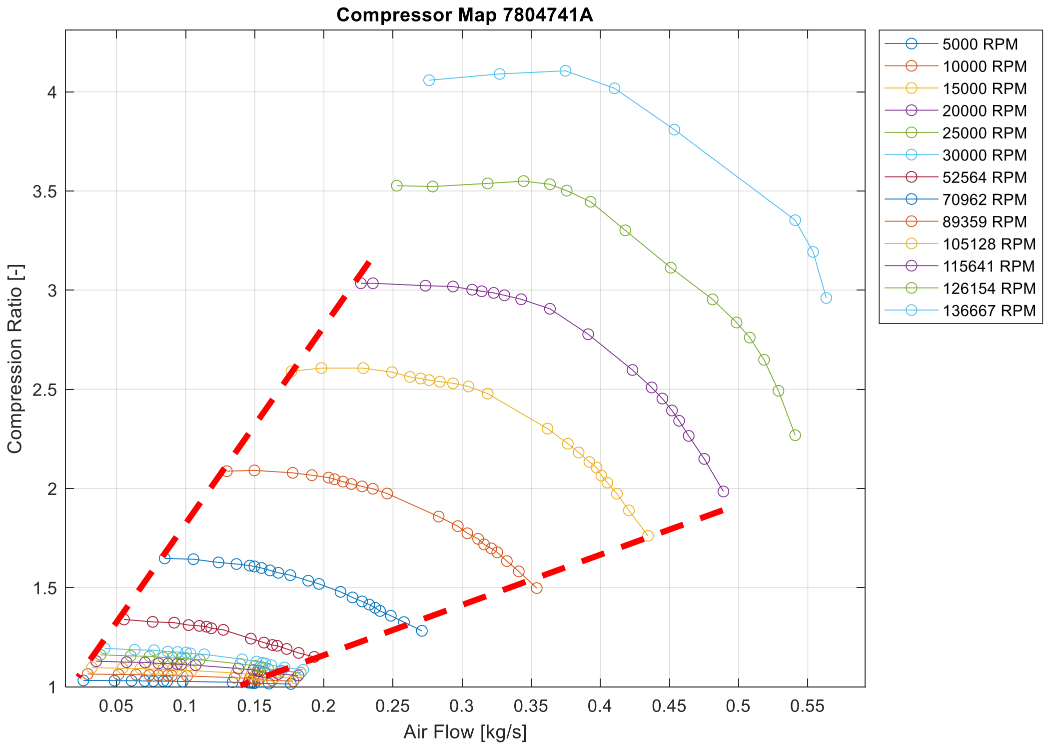


FIGURE 3.3 – COMPRESSOR MAP WITH EXTRA OPERATING LINES THAT ARE EXTRAPOLATED ON THE LOWER REGION OF THE COMPRESSOR MAP WITH THE APPROACH EXPLAINED IN FIGURE 3.2. THIS RESULTS IN A FEW OPERATING POINT THAT WILL BE PLACED IN THE REGION OF COMPRESSOR CHOKE. THE LINE OF THE SURGE AND THE CHOKE IN THE LOWER REGION CAN BE ESTIMATED BY CONSIDERING THE PREVIOUS CHOKE OR SURGE POINTS. THESE ARE INDICATED WITH THE RED DOTTED LINES

3.3 Performance Model - Electric motor

To plot the performance characteristics of the electric motor, three properties of the electric motor are used. The three properties used are: the maximum rotational speed, the maximum power and the constant power speed ratio (CPSR). Because it was initially assumed that induction motors would be used, the CPSR is used. With the CPSR value the rotation speed can be calculated where the electric motor changes from constant maximum torque to maximum power. This is a turning point in the electric motor characteristic curve. To determine this point, first the maximum torque of the electric motor is calculated using the maximum power and rotation speed as in equation 3.13.

$$T_{emotor,max} = \frac{P_{emotor,max}}{\frac{2\pi}{60} N_{emotor,max}} \qquad 3.13$$

The rotational speed where the electric motor switches to constant power can be calculated with equation 3.14.

$$CPSR = \frac{N_{emotor,max}}{N_{emotor,CPSR}} \qquad 3.14$$

With these two calculated parameters a characteristic for the electric motor can be generated. For the 20 kW electric motor this characteristic can be seen in figure XX. On the left Y-axis the power is displayed. For these type of induction motor the power rises linear till the CPSR rotational speed. On the right Y-axis, the toque is displayed. The torque is constant till the CPSR rotational speed and then drops to zero.

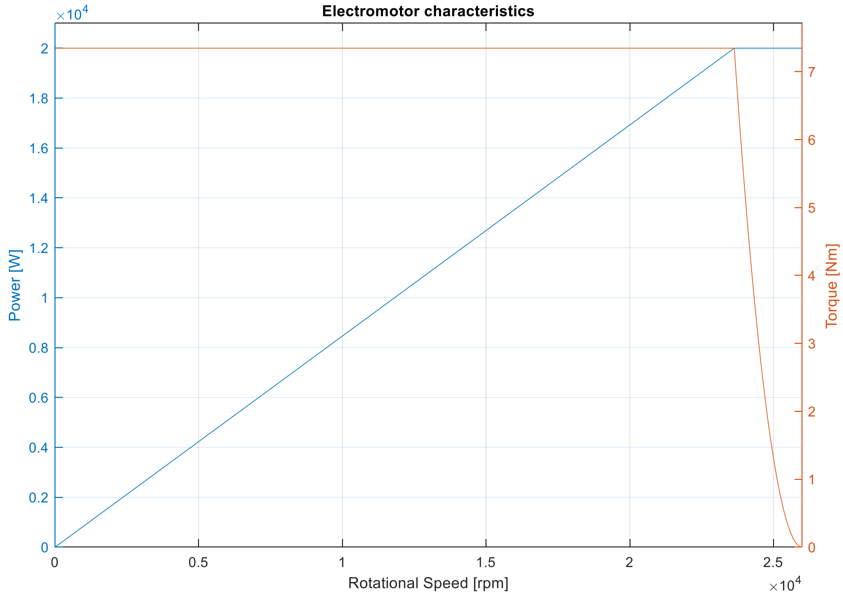


FIGURE 3.4 – CHARACTERISTIC OF THE ELECTRIC MOTOR EM150. THE BLUE LINE IS THE POWER LINE THAT SHOWS THE POWER AT A CERTAIN RPM. THE RED LINE IS THE TORQUE LINE THAT SHOWS THE TORQUE AT A CERTAIN RPM. THE POINT WHERE THESE TWO LINES CROSS IS INDICATED BY THE CPSR VALUE AND IS CONSIDERED THE OPTIMAL PERFORMANCE POINT OF THE ELECTRIC MOTOR

If there was a gearbox between the electric motor and the turbocharger, the output of the motor to the turbocharger would change. The output shaft of the gearbox will have the same power at a different rotational speed and torque value. The gearbox output rotational speed will be the electric motor rotational speed multiply with the gear ratio. For the output torque of the gearbox, the torque output drops with the gear ratio.

$$N_{gearbox} = GR * N_{emotor} \quad 3.15$$

$$\mathrm{T}_{gearbox} = \frac{1}{GR}\mathrm{T}_{emotor} \quad 3.16$$

In the two figures below the difference in electric motor characteristics is plotted with and without gearbox. The red lines and X-axis on the top of the two figures show the characteristics for the gearbox output. The black line is the original electric motor characteristic. The selection of the gear ratio will depend on the validation range that can obtained. In the figures 3.5 and 3.6 a gear ratio of 4.44 is selected to show the change in output. The strange number chosen as the gear ratio comes from the fact that the optimum performance point of the electric motor with the gearbox is placed on a compressor operating line of choice. This equation can be seen in equation 3.17.

$$GR = \frac{N_{comp,OP}}{N_{emotor,CPSR}} \quad 3.17$$

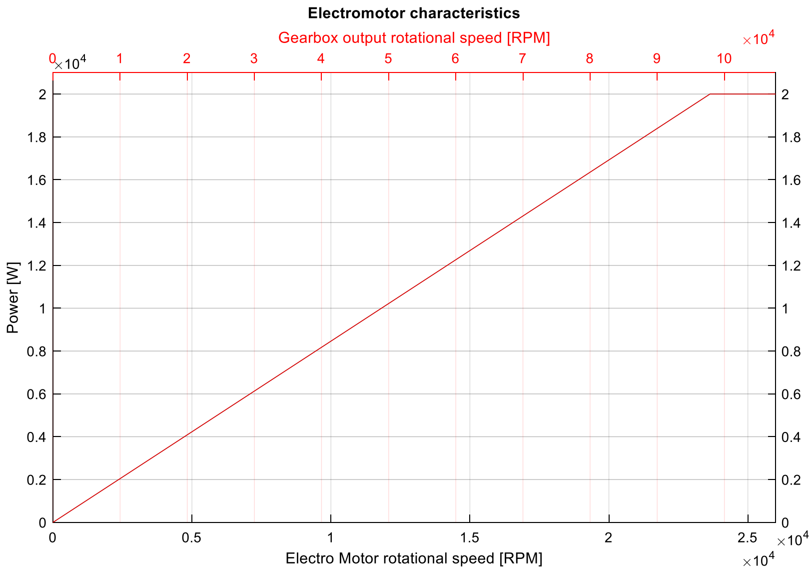


FIGURE 3.5 – POWER OUTPUT FROM THE GEARBOX COMPARED TO THE ELECTRIC MOTOR OUTPUT. THE UPPER AXIS CORRESPONDS TO THE RED LINE WHICH IS THE GEARBOX OUTPUT. IT CAN BE SEEN THAT WITH A GEARBOX, THE SAME POWER IS OBTAINED AT THE ROTATIONAL SPEED WITH THE FACTOR OF THE GEAR RATIO

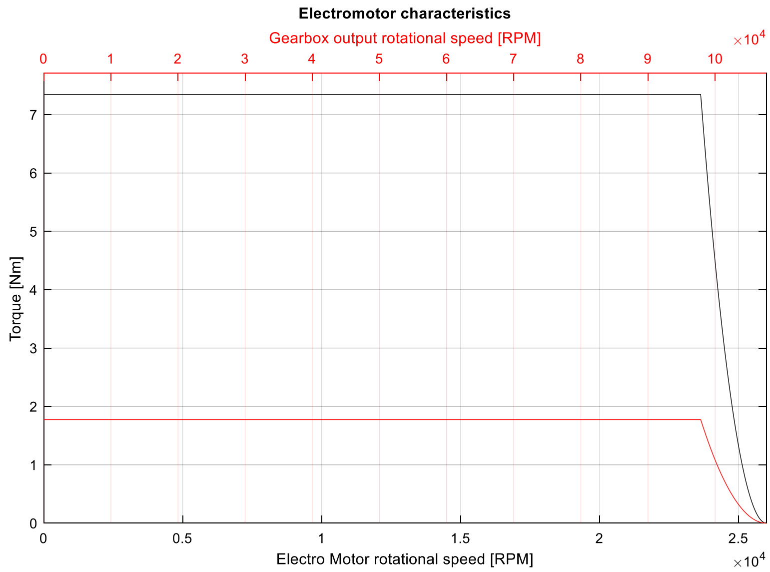


FIGURE 3.6 – TORQUE OUTPUT FROM THE GEARBOX COMPARED TO THE ELECTRIC MOTOR. THE TORQUE DROPS WITH FACTOR OF THE GEAR RATIO. THE RED LINE CORRESPONDS WITH UPPER AXIS WHICH IS THE OUTPUT OF THE GEAR RATIO.

3.4 Performance Model – Turbine Part

Like for the compressor model, for the turbine model the input is the available turbine performance data. For the provided turbine maps the rotational speed and mass flow parameter had to be converted to the noncorrected values. To calculate the turbine power and torque at the right compressor operating points, mass flow and rotational speed at those points should be the same for the compressor and the turbine. In a similar way what has been done for the compressor map, has been done for the turbine map. A Matlab function has been written that gives the operating line at a desired rotational speed from the turbine performance map. How the operating line is created is exactly the same as with the compressor map. The original turbine map has the axes reversed compared to the compressor map. In the model, the units of the axes of the turbine map and compressor map are kept the same. Figure 3.7 shows the turbine map of the 74 mm turbine wheel. For every operating point in the compressor map such a line is plotted.

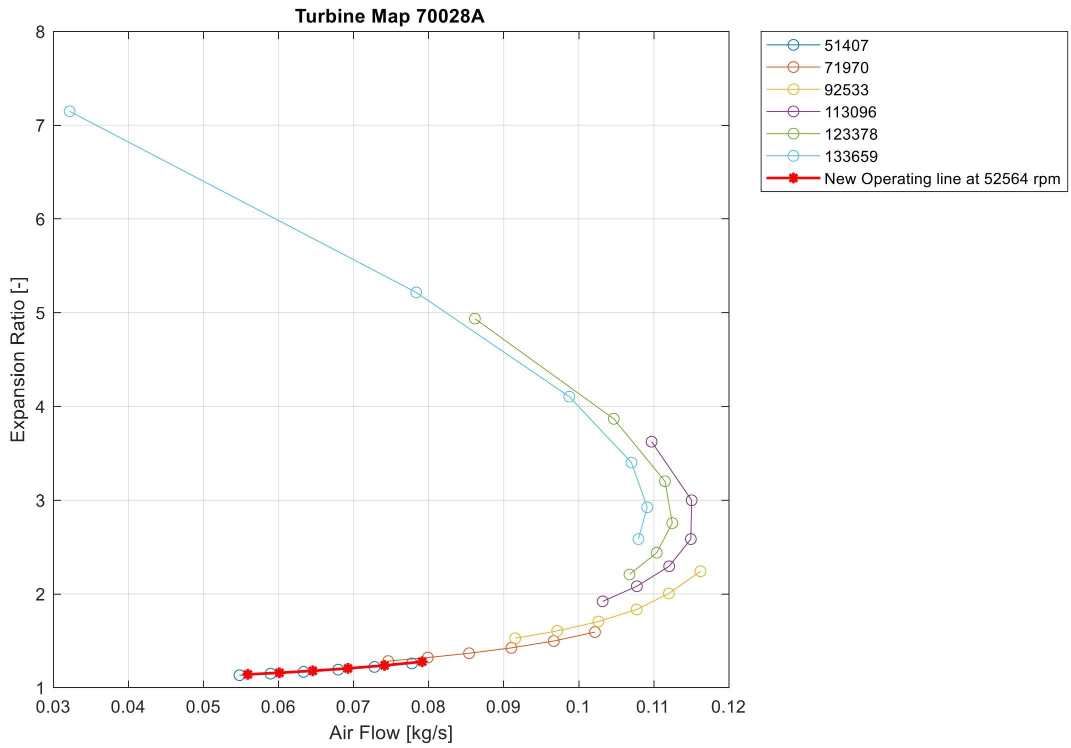


FIGURE 3.7 – TURBINE MAP OF THE 74 MM TURBINE WHEEL. THE RED LINE SHOWS THE OPERATING LINE OF THE LOWEST COMPRESSOR SPEED LINE. THIS LINE IS PLOTTED USING THE SAME PRINCIPLE AS EXPLAINED IN FIGURE 3.2 BUT FOR THE TURBINE MAP

So the output of the turbine interpolation is a dataset of the new operating line which consists of the rotational speed, mass flow, pressure ratio and efficiency. By plugging in the compressor rotational speeds into the function. Those points should still being matched in terms of mass flow with the compressor operating points. Therefore every time the operating point is selected that is closest in terms of mass flow to the compressor operating point. Together with the compressor outlet air conditions, it is used to calculate the power that can be recovered by the turbine and also the torque it generates. The temperature difference over the turbine is calculated as follows.

$$\Delta T_{turb} = \eta_{turb} T_{comp,out} \left(1 - \left(\frac{1}{\Pi_{turb}}\right)^{\frac{\gamma-1}{\gamma}}\right) \quad 3.18$$

This temperature difference is used together with the mass flow to calculate the power over the turbine. This is done with the same equation as the compressor with a different constant for the specific heat coefficient.

$$P_{turb} = \dot{m}_{turb} C_{pe} \Delta T_{turb} \qquad 3.19$$

Also the torque in the turbine model is calculated in a similar way as in the compressor model.

$$\mathrm{T}_{turb} = \frac{P_{turb}}{\frac{2\pi}{60} N_{Turbo}} \qquad 3.20$$

As last the outlet conditions of the turbine are calculated. The outlet temperature of the turbine cannot be lower as 0 °C and the outlet pressure should be higher as 1 bar (atmospheric pressure). This condition must be monitored to prevent the generation of freezing gas.

$$T_{turb,out} = T_{comp,out} + \Delta T_{turb} \qquad 3.21$$

$$p_{turb,out} = \Pi_{turb} p_{comp,out} \qquad 3.22$$

3.5 Model Results

The model drawn up in this study is a result in itself. Before this study, there was no model that did a performance prediction of a similar system to validate turbochargers with an electric motor and a turbine. As mentioned at the beginning of this chapter in this model, the compressor power and torque can be compared with the available power and torque of the electric motor and the turbine at a certain rotation speed. In all figures, the red operation points are the points that can be reached with only the electric motor. The blue operation points indicate that these points can also be reached using the turbine. In the end, the model drawn up is a result in itself. Before this study, there was no model that did a performance prediction of a similar system to validate turbochargers with an electric motor and a turbine.

3.5.1 Results original test setup design

First, the results of the chosen test setup are displayed. This is the 78 mm compressor wheel with the EM150 and the 74 mm turbine wheel. Figures 3.8 and 3.9 show avalidation range. These two figures are zoomed in on the lower part of the compressor map as the higher speed lines are not relevant. In figures 3.8 and 3.9 operating points are indicated in red at the lower speed lines. The red operating points are operating points that could only be validated using the electric motor. It can be seen that no additional points are indicated in blue. This means the turbine does not add any range in this case. Nevertheless, it was decided to test with this setup because this is the only hardware available and the principle could still be proven with this hardware.

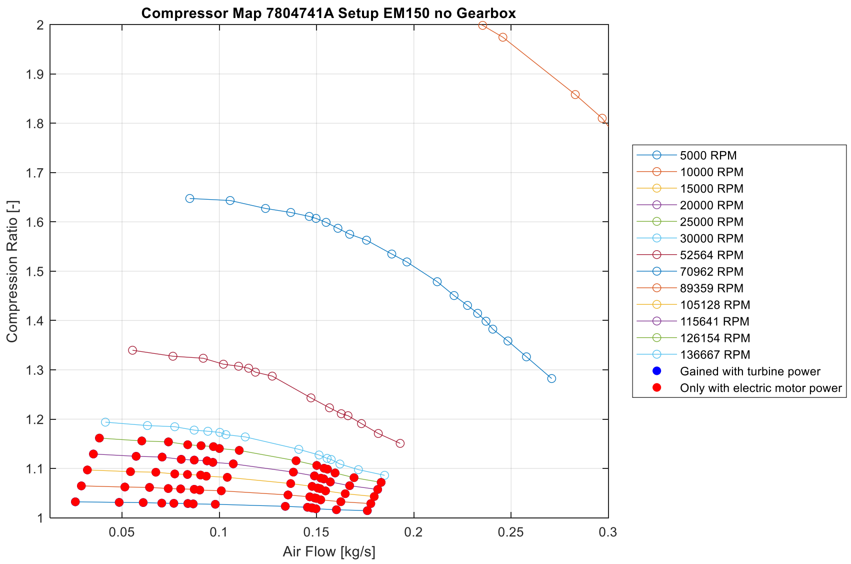


FIGURE 3.8 – COMPRESSOR MAP WITH THE REACHABLE OPERATION POINTS IN TERMS OF ELECTRIC MOTOR AND TURBINE POWER. THE RED MARKED DOTS ARE THE OPERATING POINTS THAT CAN BE REACHED WITH THE ELECTRIC MOTOR. THERE ARE NO BLUE OPERATING POINTS, WHICH MEANS THAT NO OPERATING POINTS CAN BE SPECIFICALLY REACHED USING THE TURBINE WITH THE TECHNIQUE USED IN THE MODEL.

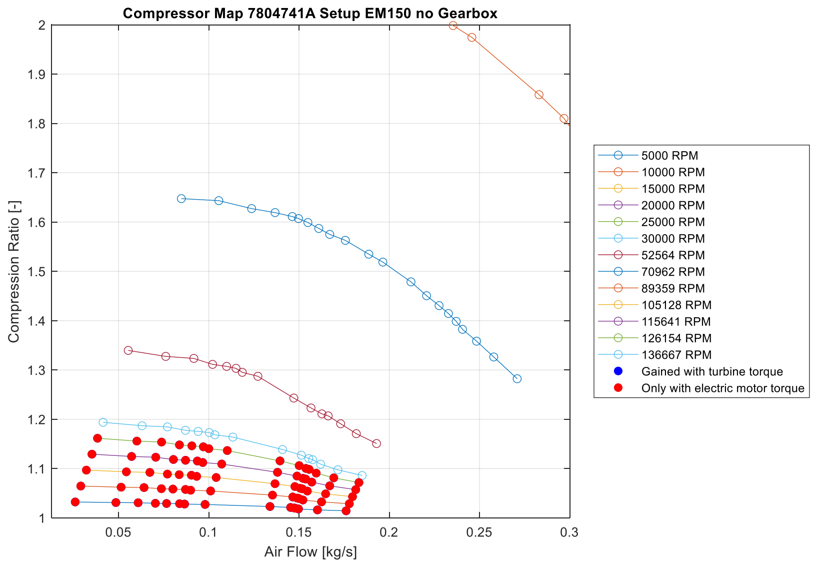


FIGURE 3.9– COMPRESSOR MAP WITH THE REACHABLE OPERATION POINTS IN TERMS OF ELECTRIC MOTOR AND TURBINE TORQUE. FOR THE TORQUE IN THIS CASE THE SAME APPLIES AS FOR THE POWER AS IN FIGURE 3.7.

Figures 3.8 and 3.9 only show operating points that can be validated with the electric motor. It was expected that also with this configuration in the lower area the turbine will increase the range and therefore also operating points would be indicated in blue. Therefore, a modification was made to still demonstrate the principle of range extension. The model set up to this point made an estimate of how much range could be validated based on the power and torque needed for that operating point. The model has been modified such that a single operating point can be selected. For this operating point a shift has been determined that represents the change in configurations of the test setup as mentioned in the test strategy in section 2.7. In figure 3.10 one point is selected which is marked with a red star. For this point it is known, as figure 3.8 and 3.9 show, that there is enough electric motor power available. The available turbine power for this point has been calculated. The required compressor power and the available turbine power are added together. This power is used to plot the red line. To check the accuracy of this line, the lines of equal power are also shown. The red line shows how the selected operation point may shift when turbine power is used. For this line the constant efficiency is used which is equal to the efficiency of the selected operation point. By this assumption it is expected that this line deviates slightly from the equal power lines at lower mass flow values. The expectation is that the valve in the system can determine where on this line the new operation point will lie by regulating the mass flow. For every point in the electric motors range this performance estimation can be done.

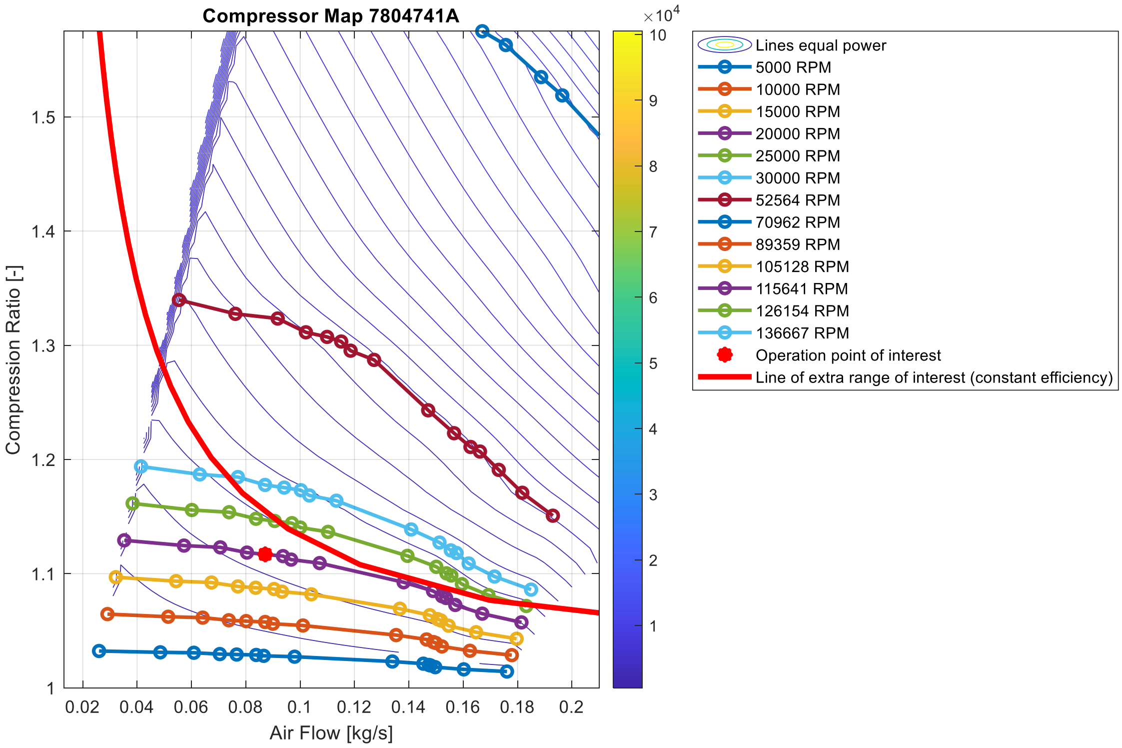


FIGURE 3.9 – COMPRESSOR MAP WHERE 1 RANDOM POINT IS SELECTED. AT THIS POINT, THE AVAILABLE TURBINE POWER IS ADDED TO THE COMPRESSOR POWER NEEDED TO DRIVE THIS OPERATION POINT. THE RED LINE INDICATES HOW FAR THE OPERATION POINT CAN BE SHIFTED WHEN TURBINE POWER CAN ALSO BE USED. THE OPERATION POINT WILL MOVE TO A POINT ON THIS RED LINE.

3.5.2 Results Test Setup Design EM150 With Gearbox

Although the use of a gearbox was not chosen, it is still included in the model. The 20 kW that the electric motor can deliver can potentially reach a larger part of the compressor map. In Figure 3.10, lines of equal power are plotted in the compressor map. The line at 20 kW is shown in red. With a gear, it should be possible to reach all operation points below the 20 kW line.

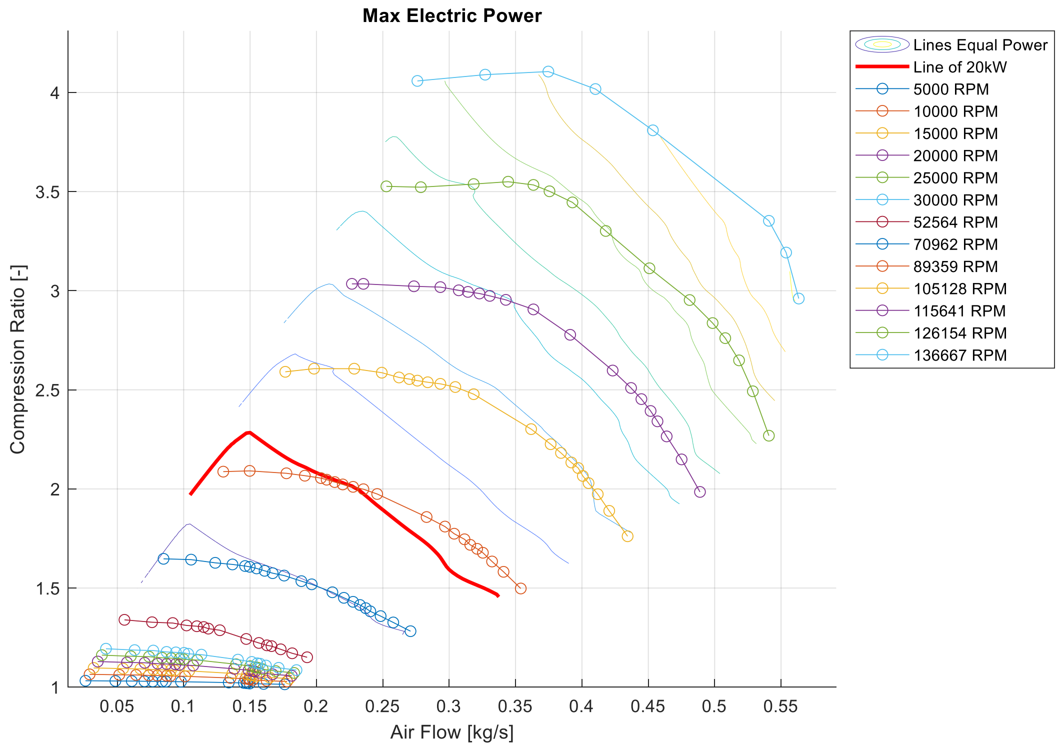


FIGURE 3.10 – 78MM COMPRESSOR MAP WITH THE 20 KW POWER LINE INDICATED IN RED. THE OTHER POWERLINES SHOW FOR EVERY 5KW A NEW POWER EQUIVALENT LINE. THIS LINES SHOW THE POTENTIAL RANGE OF THE ELECTRIC MOTOR. WITH A GEARBOX IT SHOULD BE POSSIBLE TO REACH ALL THE OPERATING POINTS BELOW THE MAXIMUM POWER LINE.

Figure 3.11 shows the validation range for different gears. In figure 3.11 the operating points are indicated in red if for the electric motor both torque and power are higher as the required compressor torque and power. The operating points are indicated in blue if for both the electric motor and turbine torque are higher as the required compressor power and torque. The optimal performance point of the electric motor is at the speed of the CPSR. At this speed the electric motor torque and power are at their maximum. Each gear is chosen such that the optimum performance point lies on the speed line. For a 2.22 gear, the N_{CPSR} is at 52564 RPM, for 3 at N_{CPSR} it is at 70962 RPM and so on. For a gear ratio of 4.45, it can be seen that the validation range is the largest. For larger gear ratios, the range decreases again. For a gear ratio of 4.45, it can be seen that approximately all the operation points below the red 20 kW line in Figure 3.10 are reached.

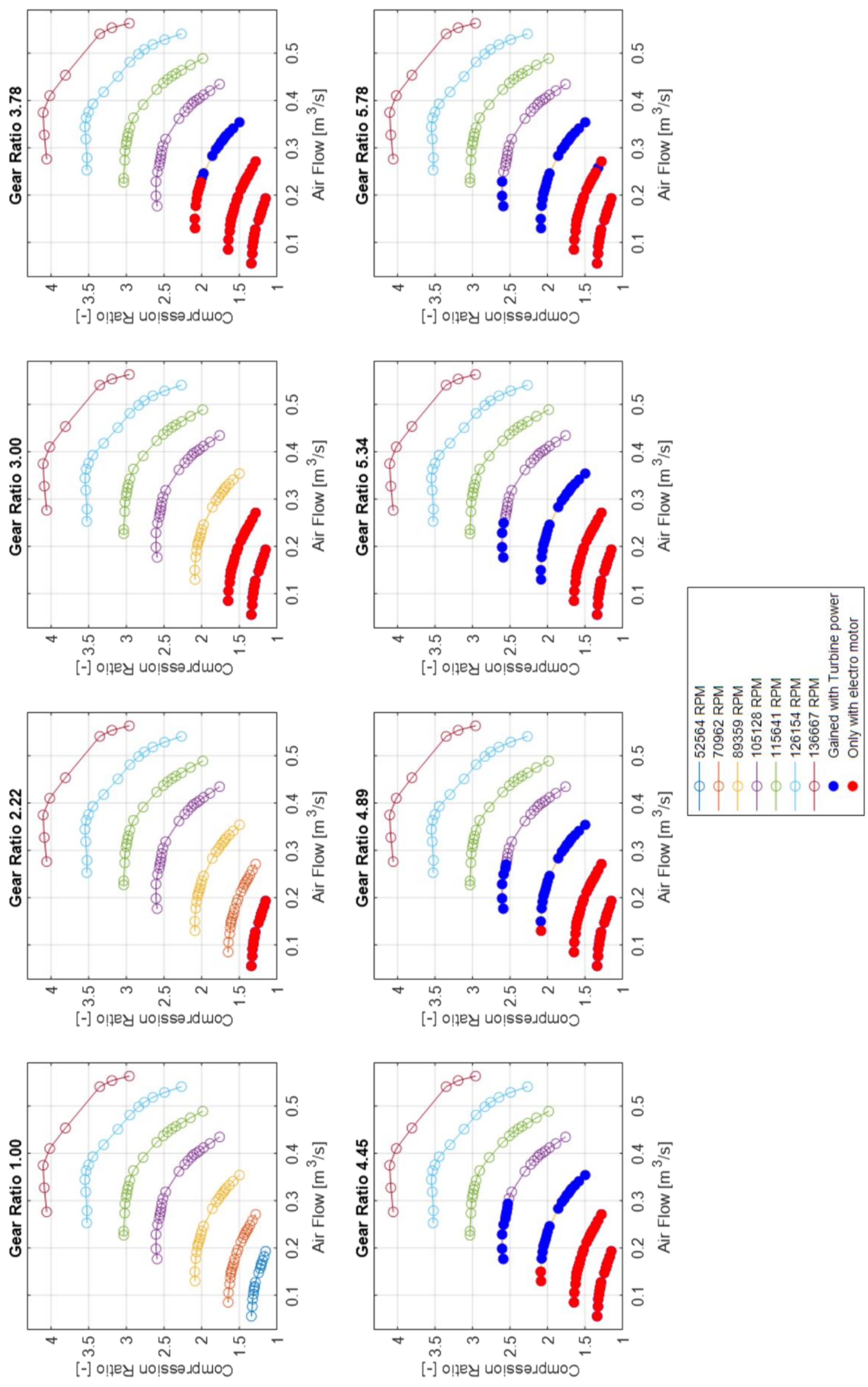


Figure 3.11 – Validation range for different gear ratios for the 78 mm compressor wheel and EM150 electric motor. Every gear ratio shifts the optimal operating point of the electric motor to a higher speed line. It can be seen that till the gear ratio of 4.45 the validation range increases. At higher gear ratios the validation range decreases again.

3.5.3 Results Test Setup Design with the EM65

The original plan was to use the EM65 for the test set-up. This is a 12 kW motor that can deliver a speed up to 175000 RPM. Since alternatives were not available, the EM150 was chosen. Nevertheless the EM65 has been used as input in the model to show that the original E-turbo concept can also be used as a test facility. Figure 3.13 and 3.14 show the compressor map when the EM65 electric motor was used. With the EM65 no gear is needed to get to the first speed line of the original compressor data. For each operating point within the validation range the outlet temperature was above 0 degrees Celsius and above 1 bar so no freezing gas will be created.

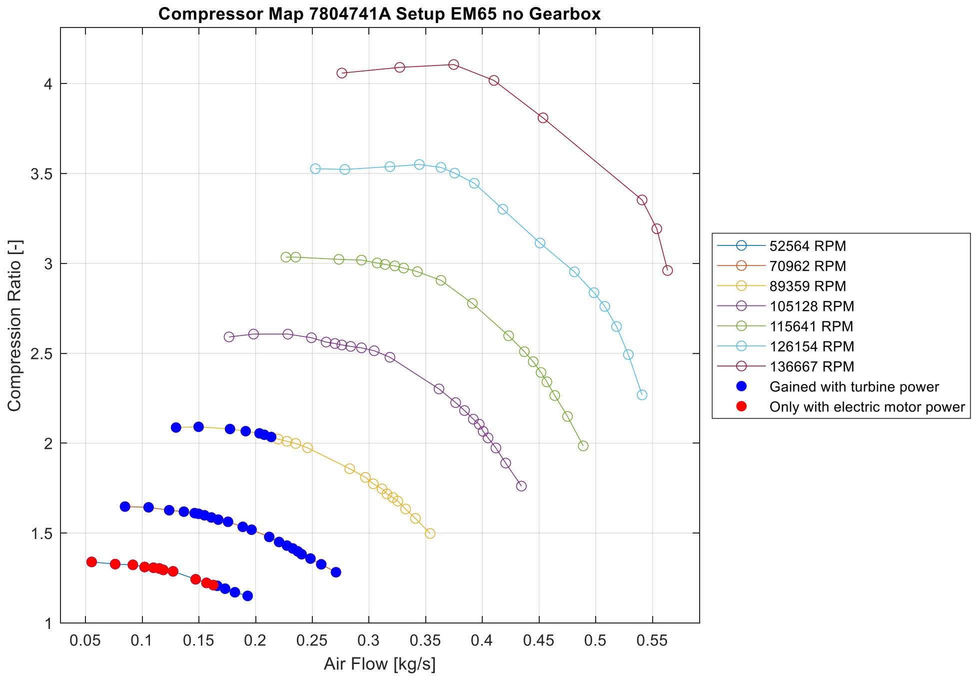


Figure 3.12 – 78 mm Compressor map with the EM65 electric motor for the power of the operating points. With the electric motor a selection of points on the lowest speed line can be validated according the power criteria. With the turbine operating points can be reached up to the third speed line

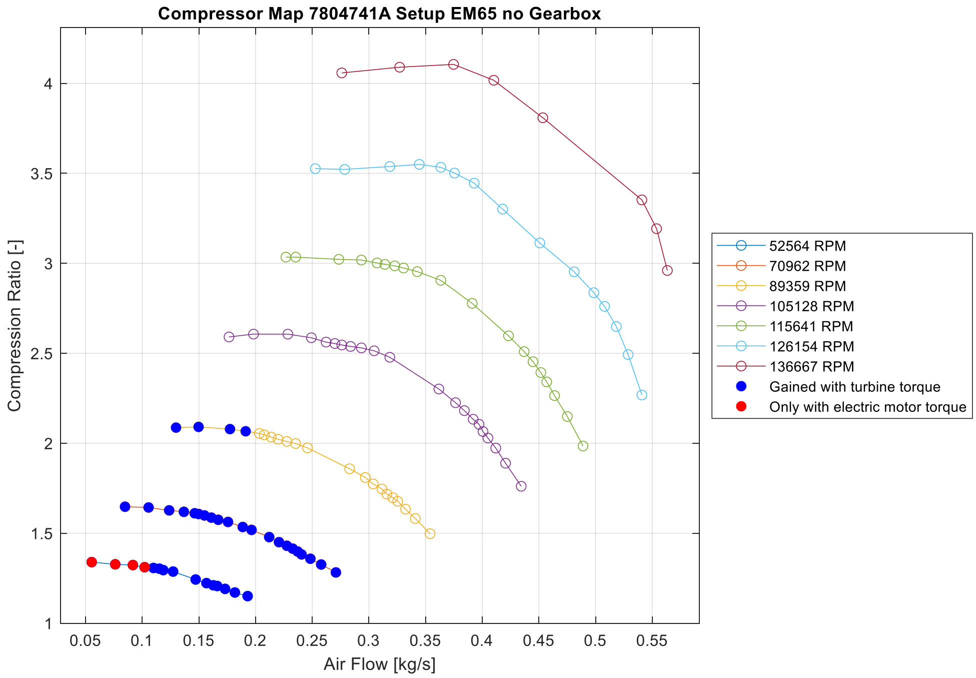


Figure 3.13 – 78 mm Compressor map with the EM65 electric motor for the torque of the operating points. Compared to the power in figure 3.12, it can be seen that on the lowest speed line less points can be validated with the electric motor. That means that this figure is leading for the validation range.

Comparing figures 3.12 and 3.13, it can be said that the torque is the critical factor that determines how large your validation range is. Figure 3.13 shows that with turbine power it should be possible to reach operating points up to 2 speed lines higher. However, this does not show how for the same engine input the validation range is increased by using the turbine. Therefore, the same figure as Figure 3.9 has been made for this combination. In figure 3.14 the 4th operating point of the original map is selected. The red line shows where the operating point will shift to when turbine power is used.

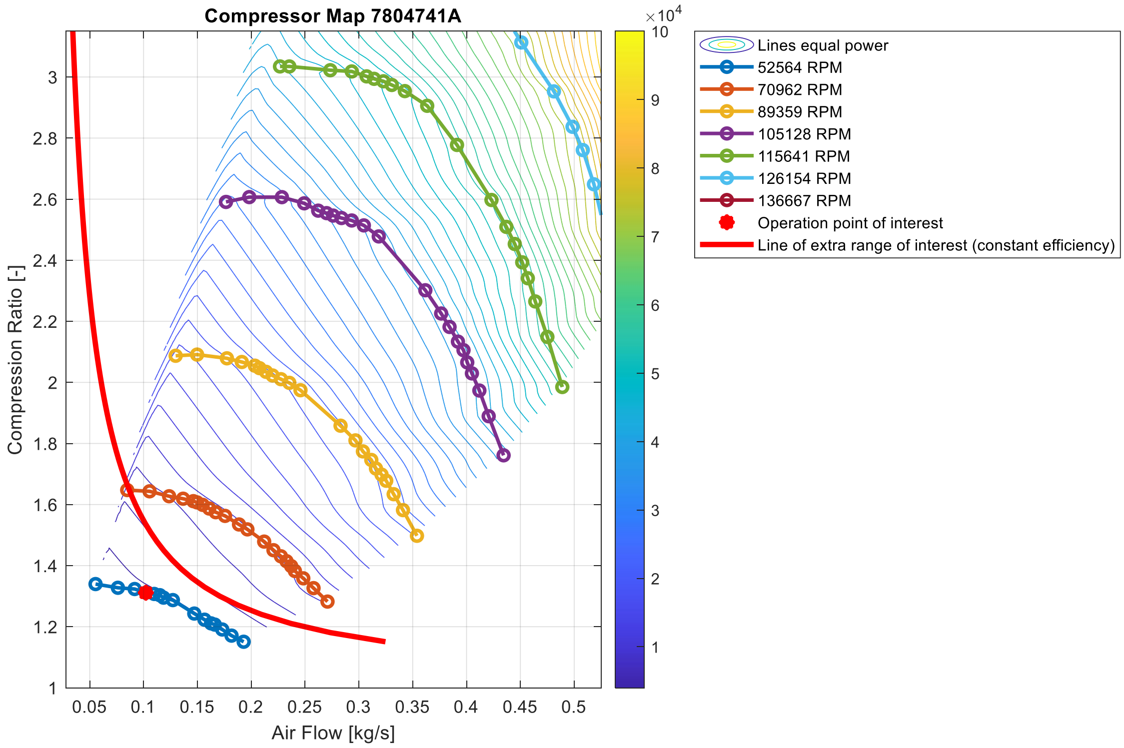


Figure 3.14 – 78 mm Compressor map with a 4th operating point selected. The red line represents the line where the operating point can shift to by connecting the turbine.

4

Reflection, Conclusions and Recommendations

This Chapter reflects on the approach, process and execution of the graduation project. Conclusions have been drawn from the results of the prediction model and the design of the test setup. These conclusions were discussed in the discussion. Finally, a reflection was made on the research done and some recommendations were made for follow-up research.

4.1 Conclusions

In this study, a test setup was designed in which a compressor map was generated from a turbocharger. With this test setup it was intended to show that it is possible to measure a larger area of the compressor map by using the turbine of the turbocharger and the electric motor together. Within the time span of the research different electric motors were not available. As a result, the test setup was not built and no physical test took place. From the design process of the test setup and the results of the performance prediction model some relevant conclusions can be drawn.

The performance prediction model created in this study shows two different figures as output. One figure shows the maximum validation range of the electric motor and the maximum validation range of the turbine and electric motor. In the other figure the shift of the operating point is shown when the test configuration is changed from electric motor to turbine and electric motor.

Figure 3.8 shows the shift of a random operating point for the designed test setup. For every operating point this shift differs but the trend of the curve is the same. Figure 3.8 shows that within the speed range of the 20 kW electric motor by keeping the system mass flow constant with use of the turbine, the operating point can be brought to a higher rotational speed. Since the shift is indicated by a higher power line to which the operating point can shift, it also seems possible to shift the operating point to a higher mass flow at the original speed line, but also higher speeds at less mass flow.

To investigate the full potential of the 20 kW electric motor, a gear ratio was implemented in the performance prediction model that led to new insights. With this addition to the prediction model, an motor could potentially be selected with a corresponding gear ratio for which the full compressor map could be validated. In case of the designed test setup with a gear ratio of 4.45 the most operation points can be achieved with the electric motor. In addition, the most operation points can also be added using the turbine. At this gear ratio, the optimal operating point of the electric motor is at the highest line that can be achieved.

With the original e-turbo electric motor, it is possible to reach map operating points without gears in the compressor map. The range of the e-turbo electric motor is not limited by rotational speed. When comparing the validation ranges in terms of torque and power of the e-turbo test setup, it can be seen that the validation range based on torque is smaller. The torque is therefore the critical parameter for determining the validation range. Figure 3.13 shows the validation range for the test setup with the original e-turbo electric motor based on torque. In figure 3.13 can be seen that the lower speed line can be reached with the electric motor. With the turbine, more power and torque becomes available. Thereby it is possible to reach all the points of the next speed line and some of the points of the speed line that follows.

For each feasible design combination of the electric motor and turbocharger, it was checked whether the outlet temperature and pressure were above 0 degrees Celsius and 1 bar. Creating freezing gas can lead to stall and damage to the turbine wheel and should therefore be avoided. The performance prediction in Matlab showed that freezing gas will not be created for any of the investigated test setup combinations.

Although no electric motor was available for testing, investigating the validation range of different test setup combinations provided insight into what electric motor is needed. For the turbocharger that was chosen in the design, a 10 kW electric motor was needed that can deliver a speed up to about 70000 RPM. With this motor the lower speed line can be reached. Such an electric motor would also be suitable for validating operating points of a reasonably wide range of turbocharger compressor wheels. Unfortunately electric motors with these specifications are not widely available although there are a few companies who reported to be able to supply them.

The aim was to design a turbocharger performance test layout and to use it to perform measurements in a compressor map. The test layout would be driven by an electric motor and the turbine of the turbocharger. This test layout would be used to prove a strategy of extending the validation range of the electric motor by using the turbine. The research so far indicates that it is possible to measure a larger area of the compressor map with the named strategy. Although this strategy has not been tested. The performance expectation shows that it is possible to eliminate the use of a hot gas burner for compressor wheel validation. Using the turbine and electric motor, it is predicted that operation points can be increased by up to 2 speed lines.

4.2 Discussion

The aim was to validate the calculated performance expectation with the designed test set-up. In the end, the test stand was not built, therefore the physical validation of both the design and the prediction was not done. The design of the test rig was based on test rigs of hot gas burners and the SAE standards. It was expected that measuring parameters such as pressure and temperature would not pose many unforeseen challenges. The coupling between the electric motor and the turbocharger was a component designed on the basis of previous research. The coupling in the test setup would be used as the first prototype for the electric assisted turbocharger but was expected to have some unforeseen challenges.

Within the time frame of the research no electric motor was available with the required specifications, therefore no testing was carried out during this study. At first, the electric assisted turbocharger with its electric motor was intended to be the basis for this project. Due to a third party, this development project was delayed. This had direct consequences on this study. Therefore, alternatives were looked into. It soon became apparent that an electric motor with the requirements of 10 kW and a rotational speed over 70 000 RPM is not readily available. The market for electric motors for e-turbo's (high speed-high power) is still at a relatively early stage of development so there is relatively little freely available. By selecting another turbocharger another electric motor could be used. A turbocharger with a larger compressor wheel requires more power at lower revs and vice versa. A smaller compressor wheel runs at higher speeds but also requires less power to be driven.

A challenge of the test setup itself was how the whole system would work in total to generate a compressor map as a result. The electronic valve is placed after the compressor outlet to avoid influencing the airflow around the compressor but to be able to set a restriction to control the mass flow. In the second configuration the valve is placed between the compressor and the turbine. When there is a big pressure drop over the valve, less energy can be recovered from the airflow with the turbine. Therefore the extra range could be less than expected. It has not been investigated whether this will result in problems such as the creation of freezing gas.

The test setup design with a lower maximum rotational speed electric resulted in that the test strategy was modified. The strategy was designed in such a way that it could be done with an electric motor that did not meet the desired rotational speed. This shifted the operational area of the test setup to the lower area of the compressor map. This led to speed lines being extrapolated in the compressor map of the 78 mm compressor wheel in the lower area. The lower area of the compressor map is often avoided by turbocharger manufacturers. Because it is easier to operate in surge or choke areas and the efficiency in this area will go down. In the model, the efficiency of the points in the lower compressor map area is assumed to be equal to the points of the lower original speed line. This assumption could be a bit optimistic.

In order to match different operating points of the compressor and the turbine, interpolation and extrapolation were done in both maps. The method devised for this purpose seems representative [19], but there are more elaborate but especially more complicated methods to do this. As it was not the intention to focus on modelling in this study, the method used was chosen. The model was drawn up with the intention of being a feasibility calculation and a means of gaining a better understanding of the validation ranges.

It is difficult to say how accurate the written model is. Different constants and input conditions are assumed based on the conditions in which the test will take place. These can easily be adjusted once the test setup and test conditions are known. However, these assumptions may cause a difference in for example calculated power in the current result compared to what will actually be measured later on. Furthermore, the approach of how the inter- and extrapolations are made, can cause a deviation from reality. Finally, the model is a highly simplified form in which many losses have been omitted.

Examples of omitted losses are the pressure drop over the pipe sections including the 180 degree turn and also the influence of the butterfly valve is kept zero. The model created calculates, based on existing performance data, which part of a compressor map can be validated and how much range can be additionally validated with the turbine. However, the model is not a model in which the hardware and its mechanics are entered with certain input conditions in, for example, a program similar to COMSOL or Ansys. Such models require a lot of computational power and quickly become very complex. Within the company, there is a project that deals with a sequential axial compressor stage for a radial compressor. These are projects in themselves. Therefore, it was assumed that with the current model, a good enough representation of the test setup could be given.

4.3 Recommendations

The biggest recommendation for further research is to wait for the developments around the electric assisted turbochargers. When an electric motor for the e-turbo can be delivered, a prototype of the test setup can be built. Then the design of the test stand and the results of the performance expectation can be validated.

Perhaps it is also possible to adapt the test set-up to compressor wheels of a different size. As mentioned in the section above, the specifications that an electric motor must meet are related to the size of the compressor wheel. By looking at a wider range of combinations than what is available in current company projects, perhaps a different combination can be found. That test setup would then focus on a different range of compressor wheels. This does not matter for demonstrating the principle of increasing the validation range using turbine power.

4.4 Reflection

Looking back on the study, I learned a lot, both technical skills and project management skills. As the research progressed, it became clear that it was not obvious that a required electric motor was available. This caused continuous changes in approach depending on the news of that week or period. I have learned to be flexible and to adapt the research in such a way that even without an actual test, a valuable result was achieved. When putting together this study, the intention was to place the focus of the study on physical measurement. It was therefore a bit of a switch to try and tell the full story. Due to the continuous changes, my own initiative sometimes lagged behind. At times, I had to wait for updates in order to build on them because continually building on incorrect information is very inefficient. Ultimately, it has been a project in which I have learned both soft and hard skills. Something I might not have expected at the beginning but in the end the result is a report that will help the new development of validating compressor wheels and electric assisted turbochargers.

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Appendices

The appendices contain additional information on the project for ones that are interested or proceeding with the project. The appendices start at the next page, having the following content:

A. Model Result of Different Combinations
B. Labview with test measurements
C. Test setup design with the EM65
D. Bill of materials of the test setup
E. Matlab script of the performance prediction

A. Model Result of Different Combinations

Several options were considered for the design of the test rig. This section of the appendix presents the results of the performance expectation. Based on these figures in combination with what was available, the combination in the study was chosen. By combination the results can be found in the sections below. The options of the EM150 electric motors with the 78 mm compressor and 74 turbine are shown in the report as well as the option of the same wheels with the EM65 electric motor.

Combination 1 – EM65 – Compressor 71 mm – Turbine 63 mm

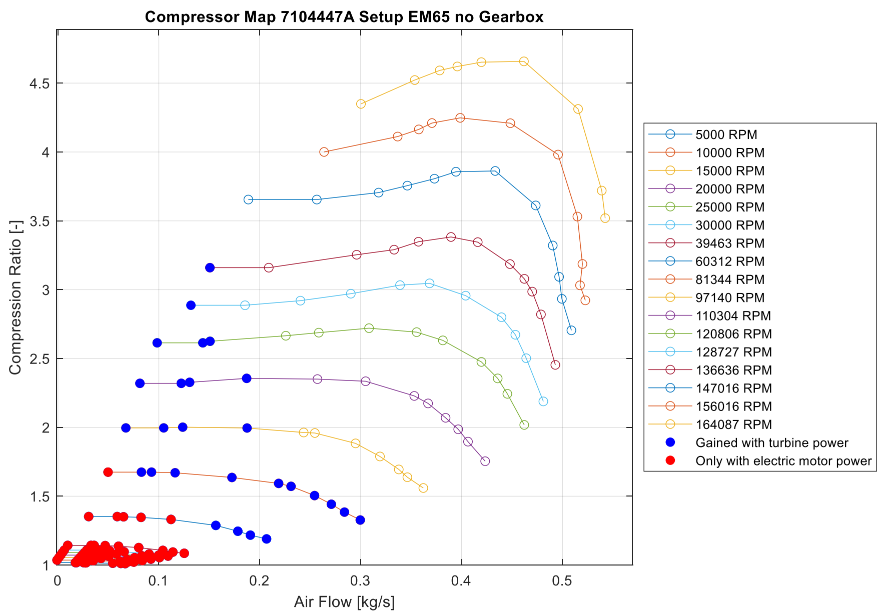

Figure B.1 – 71 mm Compressor map with the EM65 electric motor for the power of the operating points

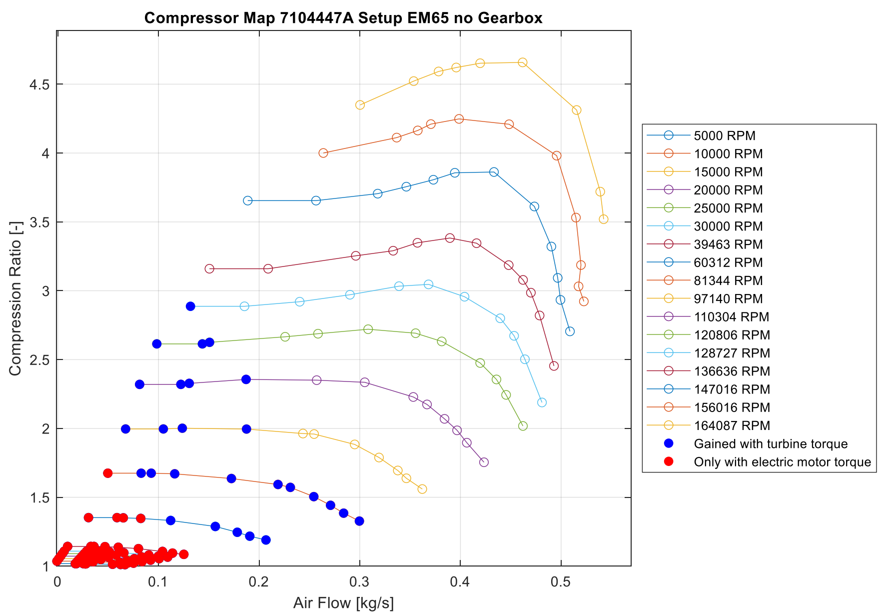

Figure B.2 – 71 mm Compressor map with the EM65 electric motor for the torque of the operating points

Combination 2 – EM65 – Compressor 78 mm – Turbine 74 mm

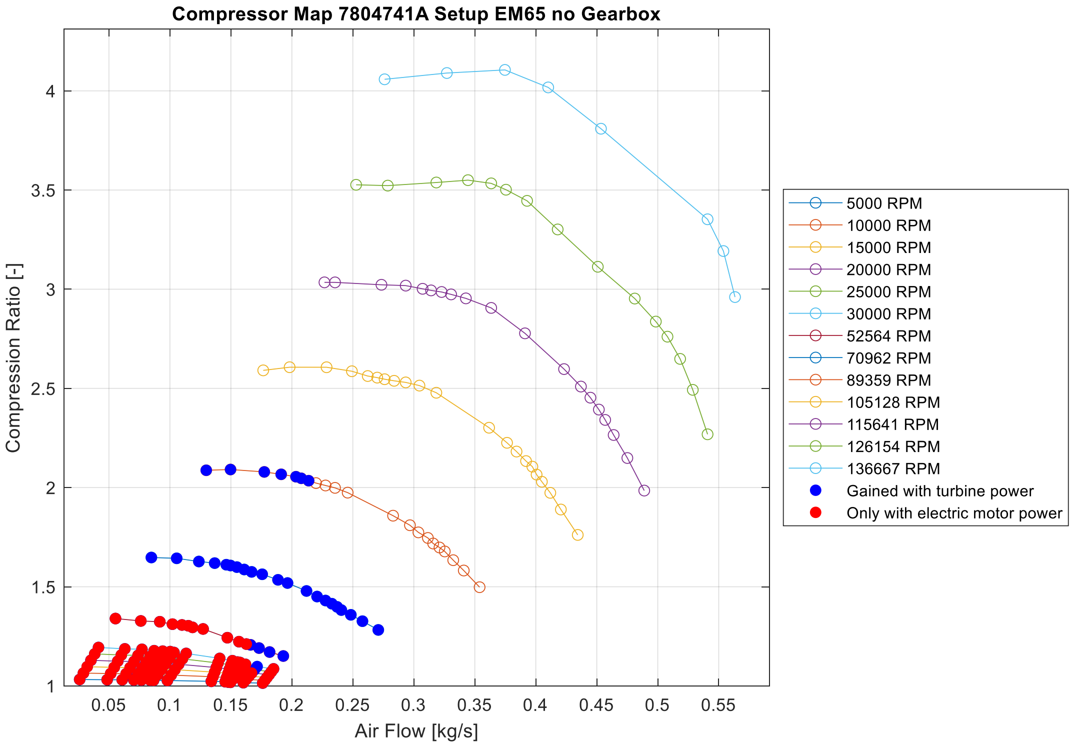

Figure B.3 – 78 mm Compressor map with the EM65 electric motor for the power of the operating points

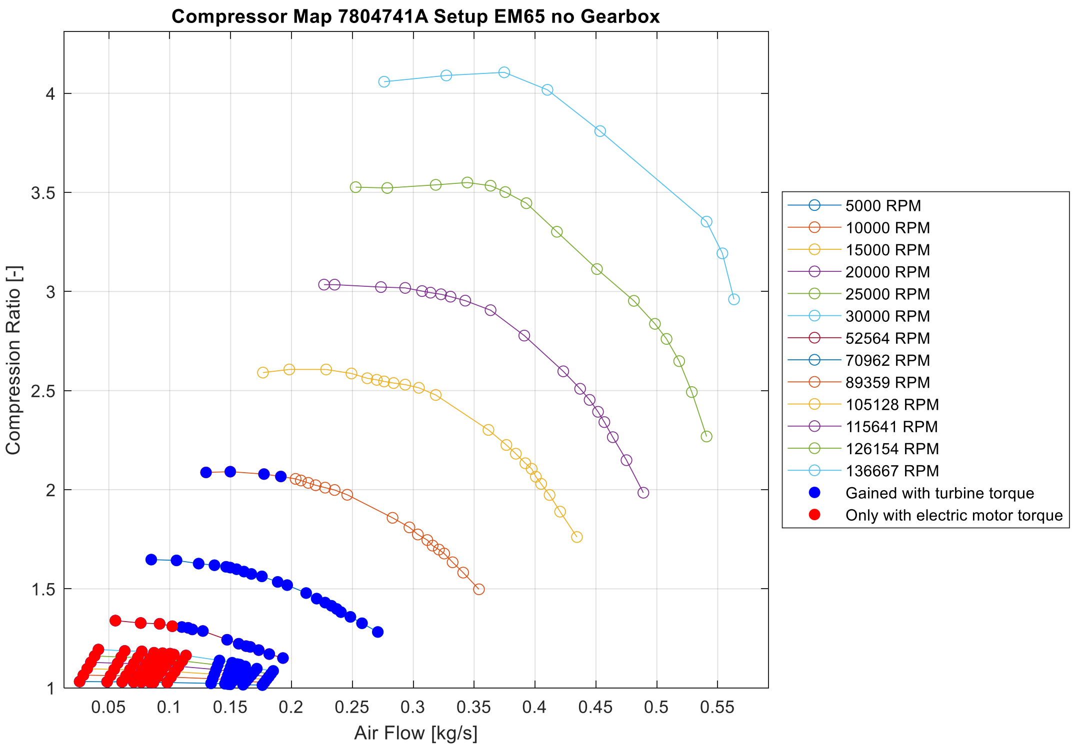

Figure B.4 – 78 mm Compressor map with the EM65 electric motor for the torque of the operating points

Combination 3 – EM150 – Compressor 72 mm – Turbine 70 mm

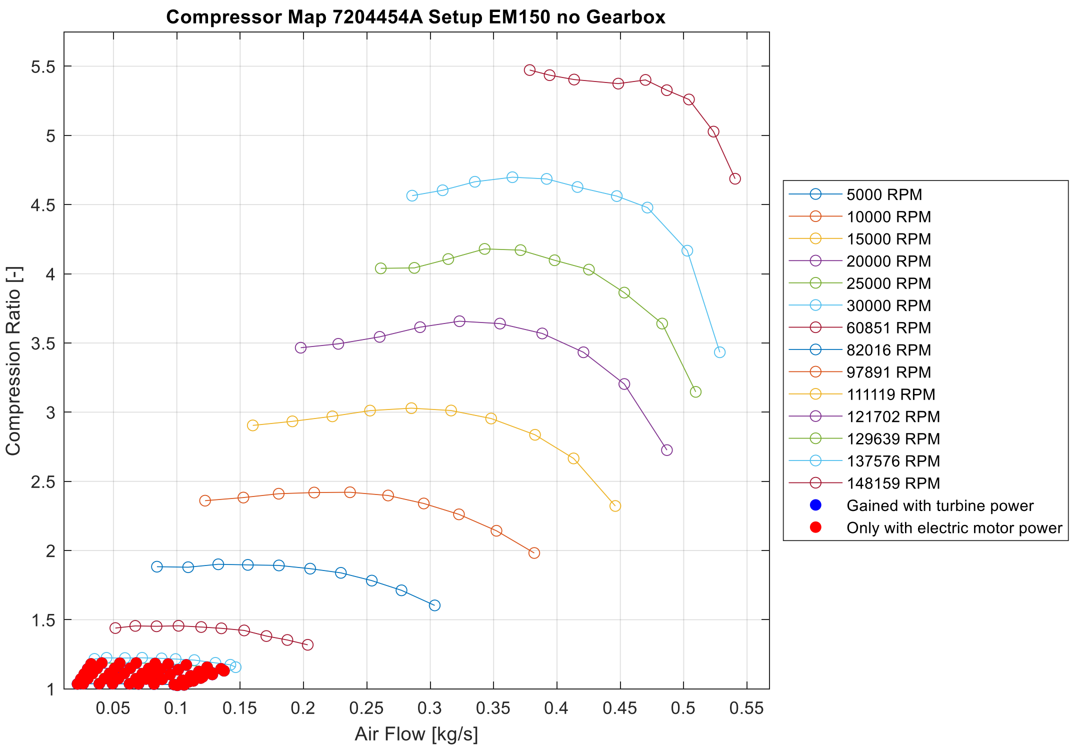

Figure B.5 – 72 mm Compressor map with the EM150 electric motor for the power and the torque of the operating points. For this combination the two maps are the same.

Combination 4 – EM65 – Compressor 72 mm – Turbine 70 mm

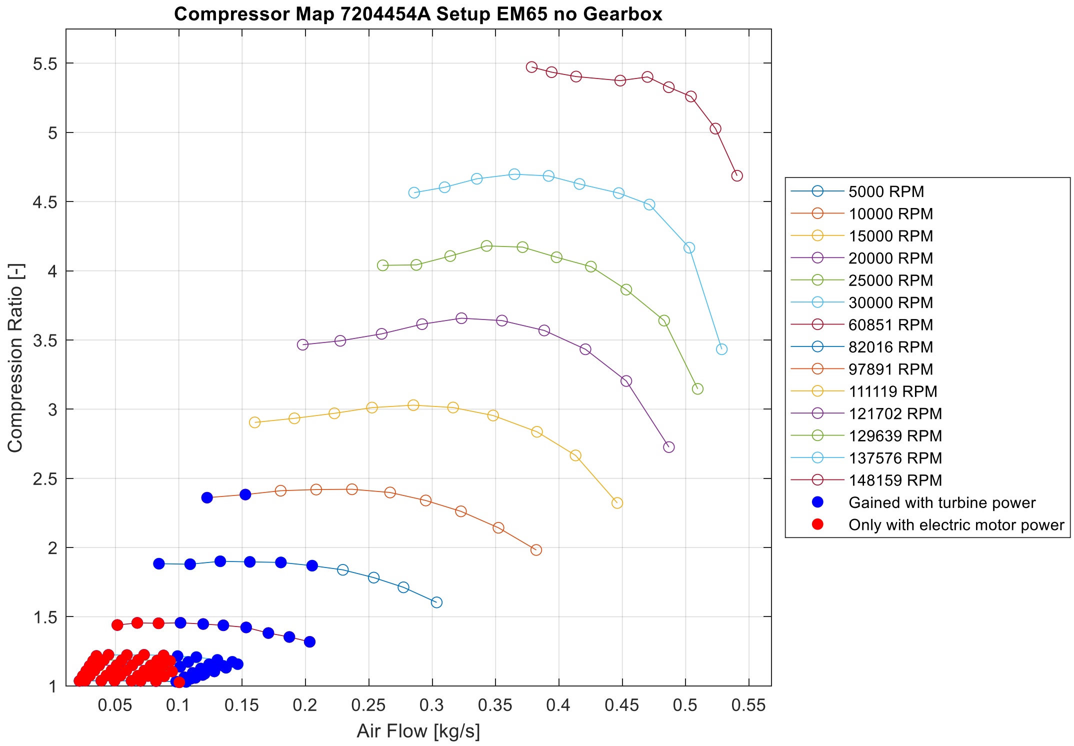

Figure B.6 – 72 mm Compressor map with the EM65 electric motor for the power of the operating points

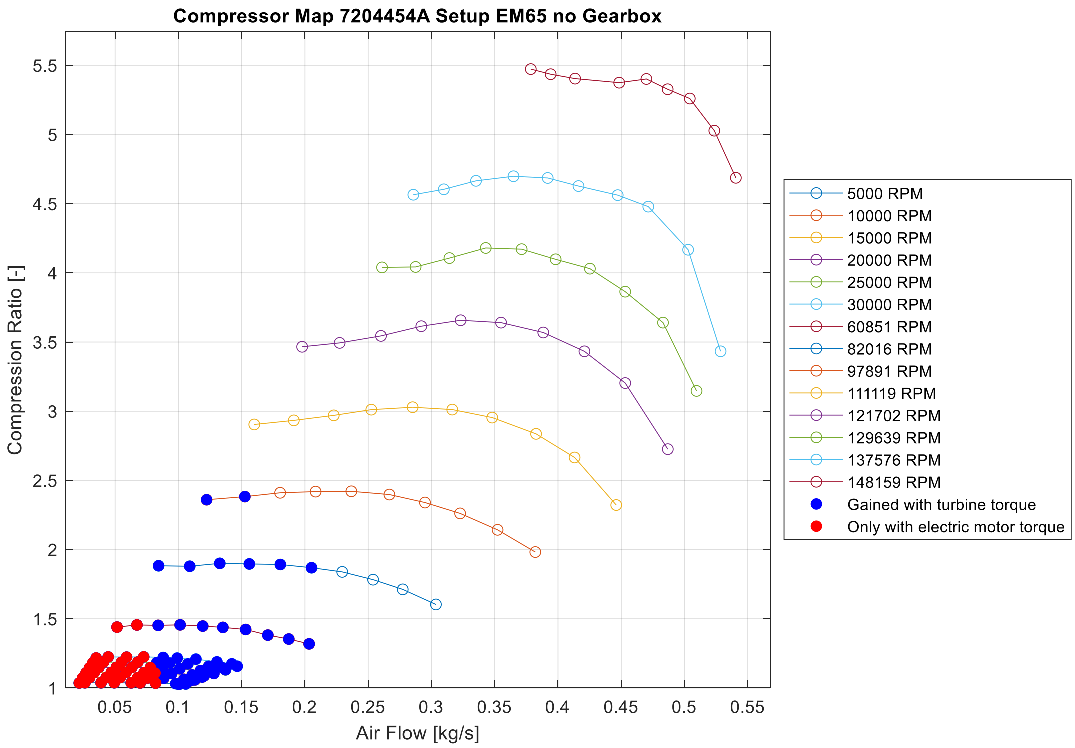

Figure B.7 – 72 mm Compressor map with the EM65 electric motor for the Torque of the operating points

Combination 5 – AquaStar T20 X520 3D – Compressor 78 mm – Turbine 74 mm

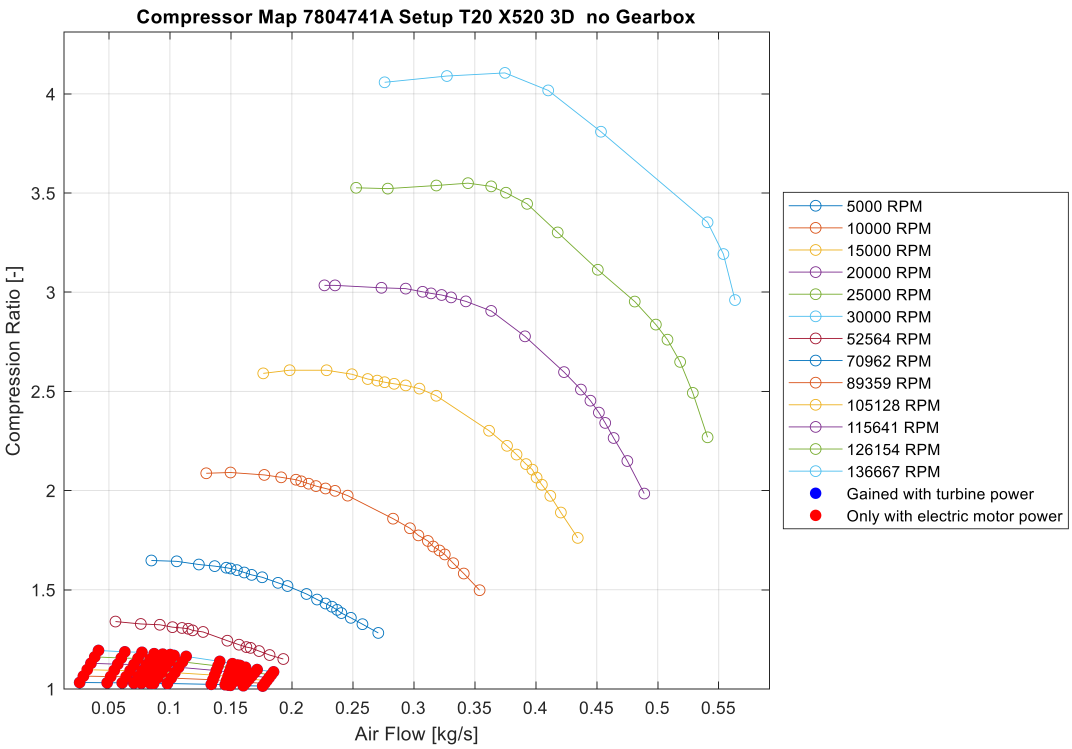

Figure B.8 – 78 mm Compressor map with the Aquastar T20 X520 3D electric motor for the torque of the operating points

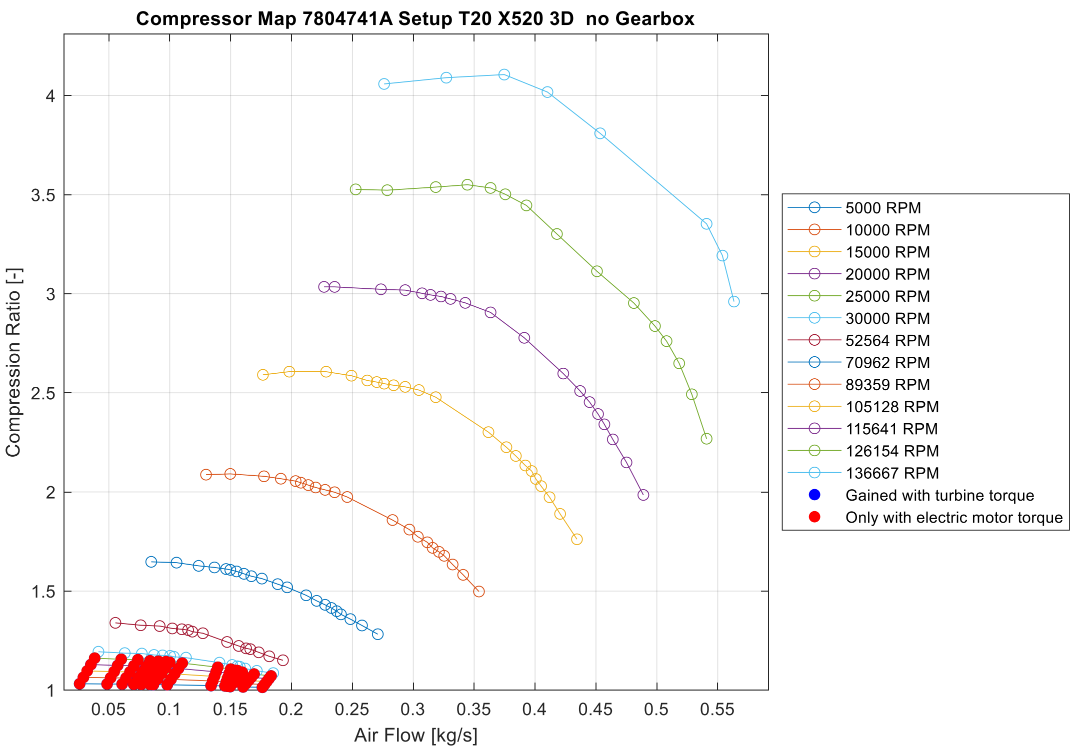

Figure B.9 – 78 mm Compressor map with the Aquastar T20 X520 3D electric motor for the torque of the operating points

Combination 6 – EM65 – Compressor 63 mm -Turbine 55 mm

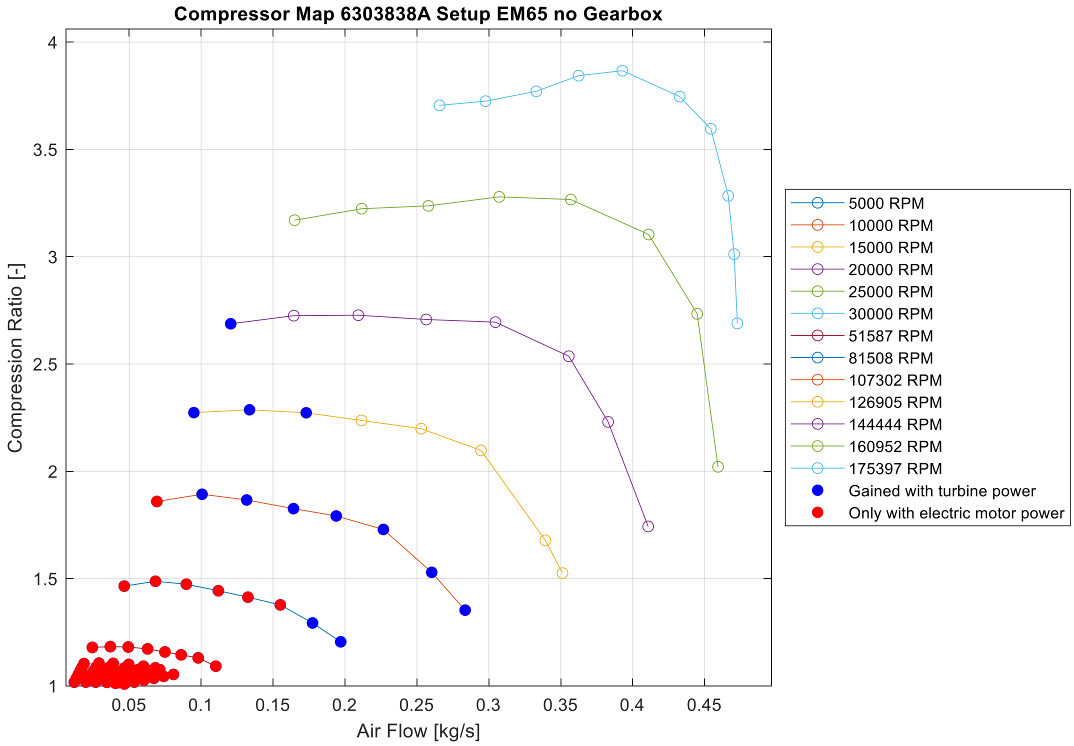

Figure B.10 – 63 mm Compressor map with the EM65 electric motor for the torque of the operating points

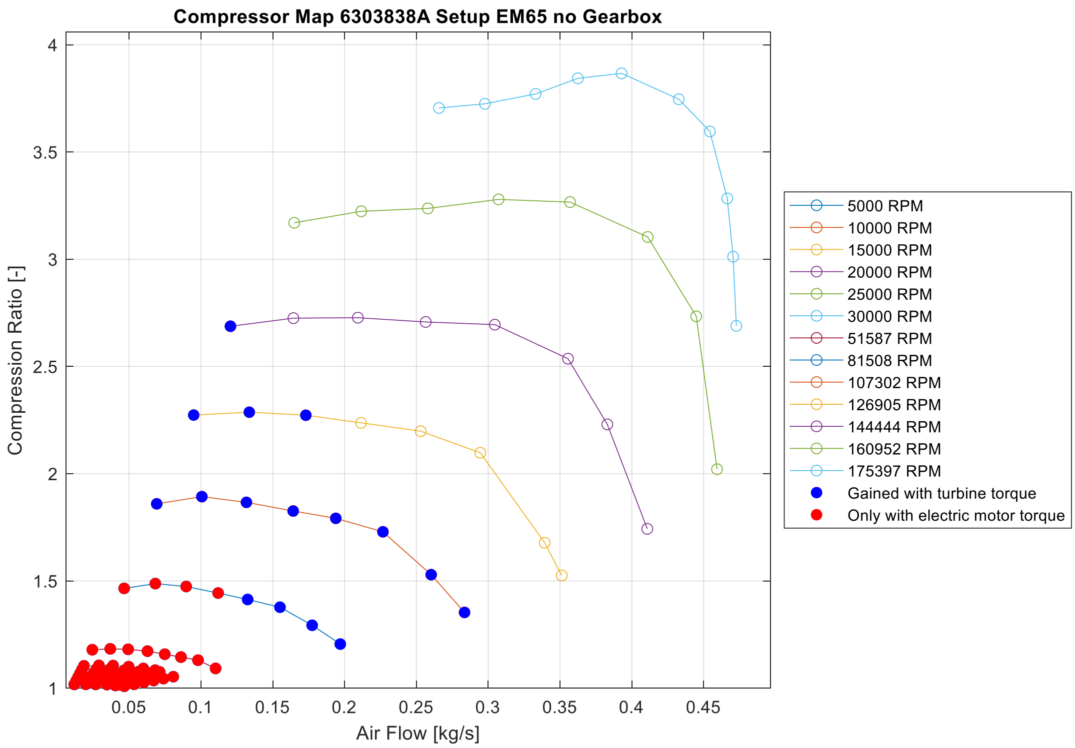

Figure B.11 – 63 mm Compressor map with the EM65 electric motor for the torque of the operating points

Combination 7 – EM150 – Compressor 170 mm

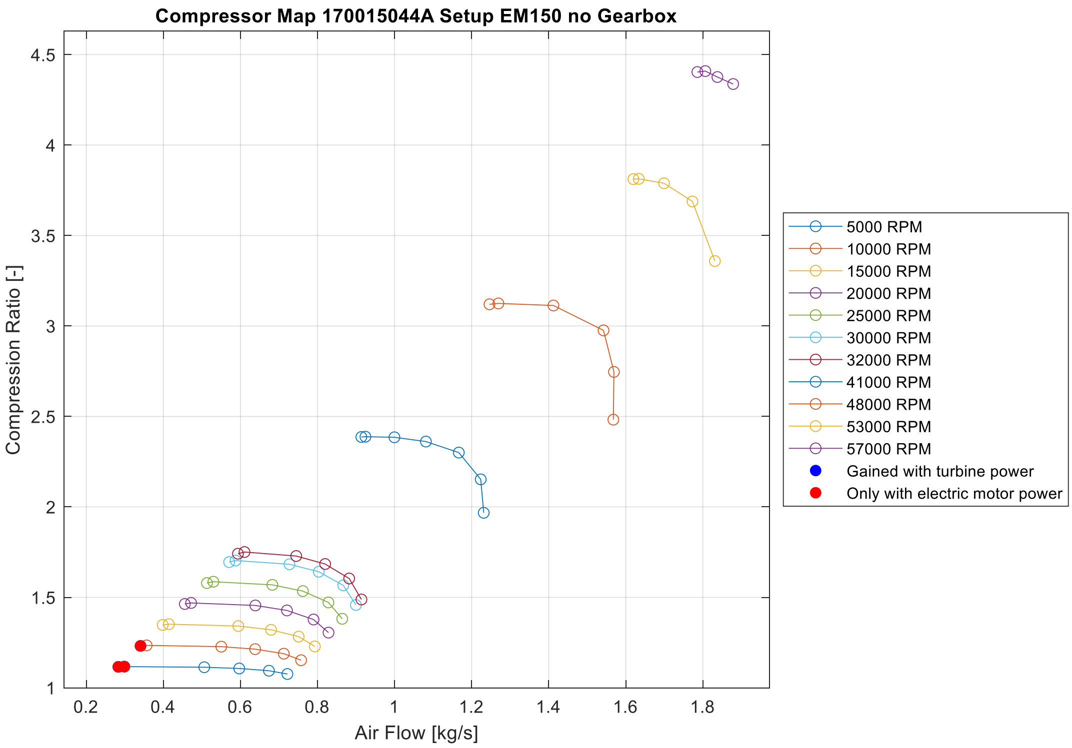

Figure B.12 – 170 mm Compressor map with the EM150 electric motor for the power of the operating points. Only the lowest extrapolated operating point within reach which is considered too small .

B. LABVIEW with Measurement Test

In the test set-up, several sensors are placed whose data are to be read out and processed into an operating point in a compressor map. For the data acquisition, a programme was written together with Jos van Driel. In this section of the appendix, the LABVIEW interface is discussed and the program is briefly explained.

LABVIEW is used to read out and process the measurement data. Jos van Driel made a program in LABVIEW that is used to convert all sensor data into a txt-file or excel-file. In a number of cases, the read-out values are converted directly to the value used to generate a compressor map. Figure C.1 shows the interface with which the measurements would be made. The first charts show the absolute measured values of the rotational speed, pressure and temperature. The two smaller charts show the pressure ratio and the calculated flow. These charts are added as an indication that can be used during validation. In the top right corner is shown which parameter is being saved in which column in the file. The interface also contains a number of input fields. In the upper left corner you can see the samples per channel and the rate. This is the sample frequency and the number of samples that can be input for the rotational speed calculation. With the current settings, a speed up to 100 000 rpm can be measured. These values could go down a bit when measuring in the lower rotational speed range. In LABVIEW a buffer is created for the rotational speed, in which firstly it is checked whether the previous measured value is a positive speed. The frequency of the signal is calculated with a separate frequency detection block. This result is used together with the number of blades to calculate the rotational speed. Further input fields are for the input of the ambient air conditions and the reference temperature for the thermocouples. The temperature measurements are averaged per 50 measurements.

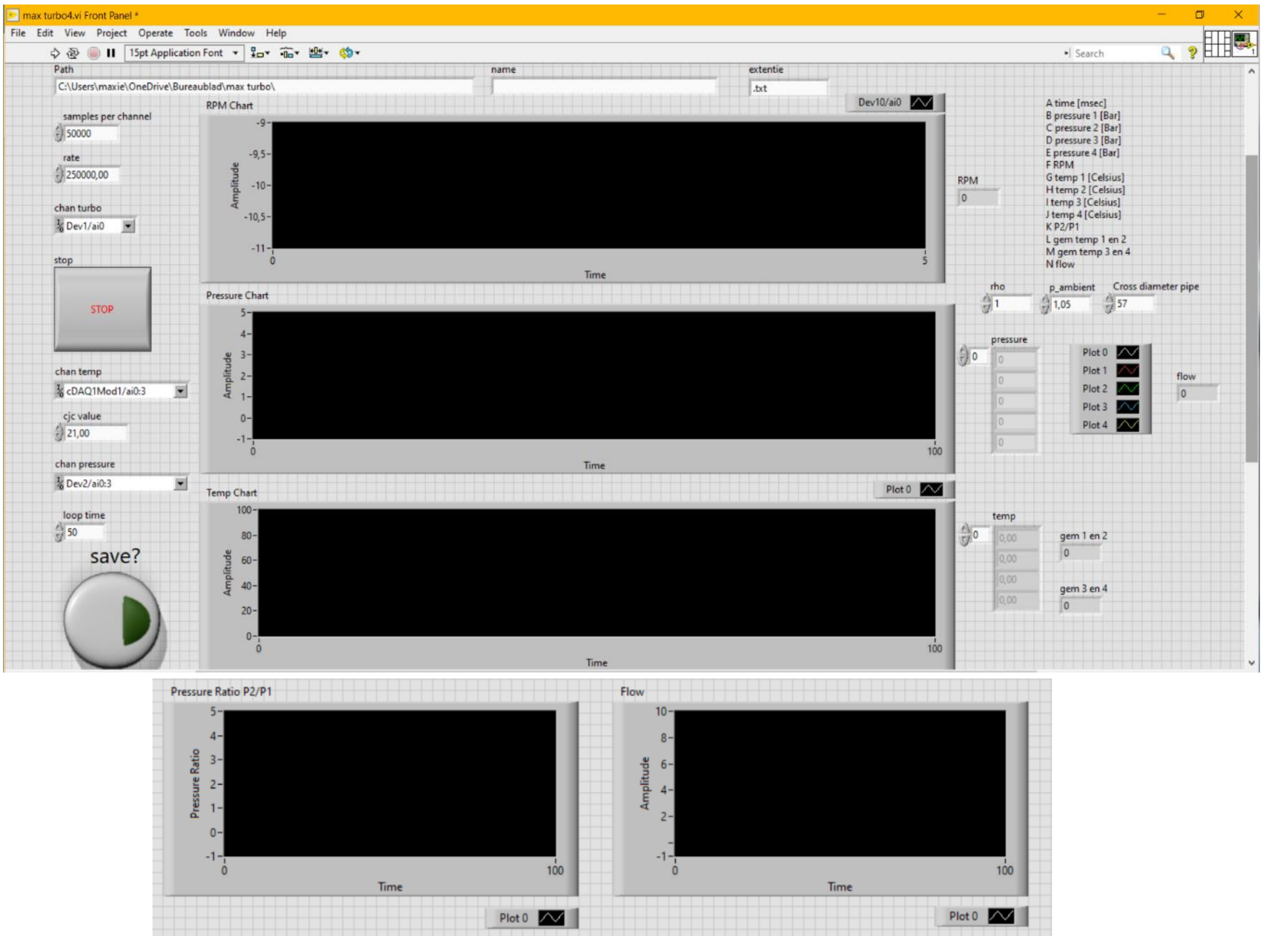

Figure C.1 – Interface of the LABVIEW program

C. Test Setup design with the EM65

This section of the appendix shows the figures of the test setup with the electric motor of the electric assisted turbocharger. In this design the inlet donut of the compressor is scaled to fit the smaller 12 kW electric motor on the test stand. The connection between the electric motor and the compressor is also made smaller. The electric motor of the e-turbo is a lower-power motor, allowing for a thinner spline connection. This electric motor is smaller and lighter and its maximum operating speed is 175000 rpm. Therefore, this motor is more suitable for the test set-up. Besides the above changes there are no changes to the test stand.

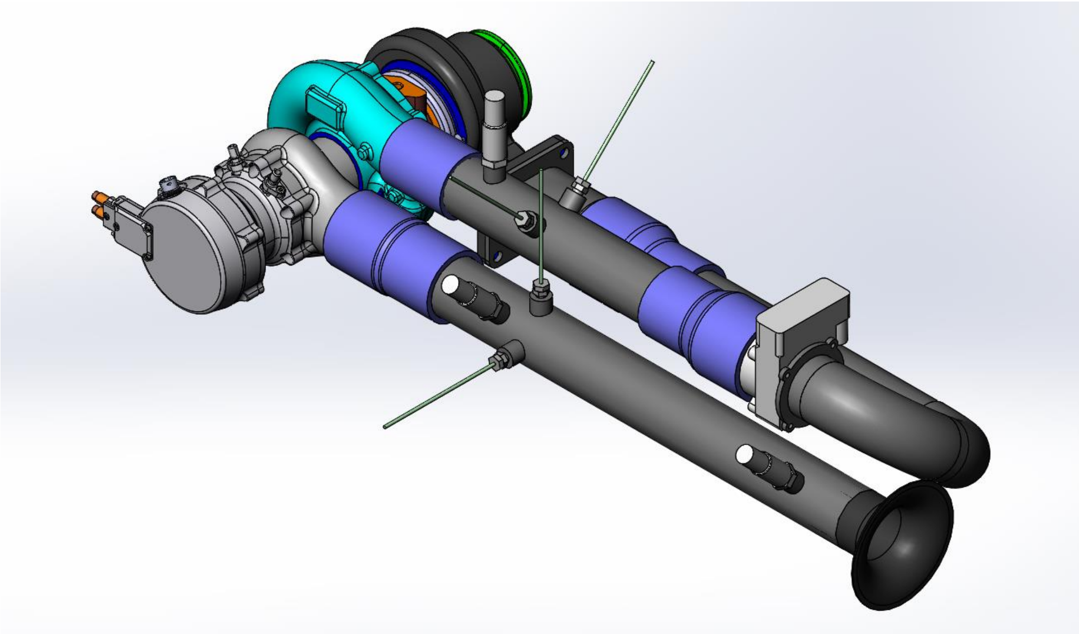

Figure D.1 – Isometric view of the test setup design with the EM65 electric motor

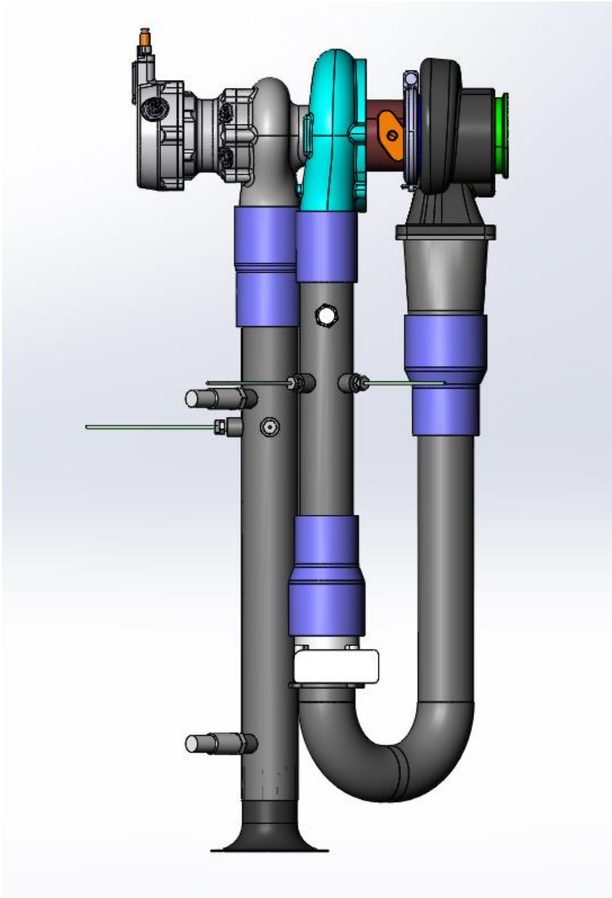

Figure D.2 – Top view of the test setup design with the EM65 electric motor

D. Bill of materials of the Test setup

#	Part	Amount	Supplier
	Turbocharger		
1	EM150 electromotor (20kW, 26,000 RPM)	1	Empel
2	Spline shaft	1	
3	Spline nut	1	
4	Compressor Donut inlet housing	1	3D hubs
5	Compressor housing	1	VDLEE
6	Compressor wheel 78 mm	1	VDLEE
7	Cartridge Assy	1	VDLEE
8	Turbine wheel 72 mm	1	VDLEE
9	Turbine Housing	1	VDLEE
10	V band	1	VDLEE
11	Snap ring	1	VDLEE
	Compressor Inlet		
12	Pipe diameter 60 mm length 360 mm	1	RVSuitlaatdelen
13	Inlet trumpet	1	
14	Silicone pipe connector Diameter 63.5 to 60 mm	1	RVSuitlaatdelen
15	Thermocouple K-type	2	TU Delft (Loan)
16	Thermocouple boss	2	
17	Thermocouple compression fitting	2	RS online
18	Pressure transducer	2	TU Delft (Loan)
19	pressure transducer boss	2	
	Compressor Outlet		
20	Pipe diameter 57 mm length 342 mm		RVSuitlaatdelen
21	Silicone pipe connector Diameter 55 mm		RVSuitlaatdelen
22	Thermocouple K-type	2	TU Delft (Loan)
23	Thermocouple boss	2	
24	Thermocouple compression fitting	2	RS online
25	Pressure transducer	1	TU Delft (Loan)
26	pressure transducer boss	1	
27	Silicone pipe connector Diameter 70 to 57 mm		RVSuitlaatdelen
28	Electic Thorttle body diameter 60 mm	1	Winparts
29	Pipe Throttle body + flange diameter 60 mm	1	
	Turbine Inlet		
30	Pipe diameter 60 mm elbow 90 degree	2	RVSuitlaatdelen
31	Pipe diameter 60 mm length	1	RVSuitlaatdelen
32	Pipe diameter 60 mm length	1	RVSuitlaatdelen
33	Silicone pipe connector Diameter 60 to 76.1 mm	1	RVSuitlaatdelen
34	RVS connection part oval to round	1	RVSuitlaatdelen
35	T4 flange	1	

E. MATLAB script of the Prediction Model

This section of the appendix shows the scripts of the prediction model. When the MATLAB script is run, 7 figures are plotted. The figures that are plotted are listed below. Original compressor map.

1. Compressor map for the power validation range
2. Compressor map for the torque validation range
3. Original turbine map
4. Electric motor characteristic power line
5. Electric motor characteristic torque line
6. Zoomed in Compressor map for single operating point prediction

Complete model

```
%%
% Author: M. N. Ligthart
% Date: 01/09/2021
clc
clearvars
close all

%% Initialize variables
InitializeVariables;

%% Compressor Map Data
CompressorMap = cell2mat(struct2cell(load('CMS02Compressormap7804741A.mat')));

CompressorN = CompressorMap(:,1);
CompressorNset = unique(CompressorMap(:,1));

CompressorQdot = CompressorMap(:,2);

CompressorPi = CompressorMap(:,3);
Compressoreta = CompressorMap(:,4);

ExtraRPM = [5000 10000 15000 20000 25000 30000];
for i=length(ExtraRPM):-1:1
      [CompressorInterpRPM, CompressorInterpQdot, CompressorInterpPi, CompressorInterpeta] =
CompressorMapInterpolation(ExtraRPM(i),CompressorN, CompressorNset, CompressorQdot,
CompressorPi, Compressoreta);
      CompressorN = [CompressorInterpRPM; CompressorN];
      CompressorNset = unique(CompressorN);
      CompressorQdot = [CompressorInterpQdot; CompressorQdot];
      CompressorPi = [CompressorInterpPi; CompressorPi];
      Compressoreta = [CompressorInterpeta; Compressoreta];
end

%% Turbine Map Data
TurbineTref = 373.15;      % Turbine reference temperature in [K]
Turbinepref =  10000;   % Turbine reference pressure in [Pa]

TurbineMap = cell2mat(struct2cell(load('TM046Turbinemap74035A.mat')));

TurbineN = round(TurbineMap(:,1)*(TurbineTref)^0.5);
TurbineNset = unique(TurbineN);
Turbinemdot = TurbineMap(:,3)*(Turbinepref/100/((TurbineTref)^(0.5)));
TurbinePi = TurbineMap(:,2);
Turbineeta = TurbineMap(:,4);

%% E-motor
% Values Empel motor
EmotorPowerMax = 20000;                                             % Power in [W]
EmotorNMax = 26000;                                                 % Motor Speed in [RPM]
EmotorTorqueMax = EmotorPowerMax/(2*pi*EmotorNMax/60);              % Torque in [Nm]
EmotorCPSR = 1.1;                                                   % Constant power speed
range
EmotorNCPSR = EmotorNMax/EmotorCPSR;                                % CPSR =
omega_max/omega_base

GearRatio = 1;
```

```
%% E-motor Characteristics and POWER and TRORQUE Calculation
EmotorN = linspace(0, EmotorNMax, 1000+1)';
EmotorTorque = zeros(length(EmotorN),1);
EmotorPower = zeros(length(EmotorN),1);

for j = 1:1:length(EmotorN)
    if EmotorN(j) <= EmotorNCPSR
        EmotorPower(j) = (EmotorPowerMax/(2*pi*EmotorNCPSR/60))*(2*pi*EmotorN(j)/60);
        EmotorTorque(j) = EmotorTorqueMax;
    else
        EmotorPower(j) = EmotorPowerMax;
        EmotorTorque(j) = (EmotorTorqueMax/((2*pi*EmotorNCPSR/60)-
(2*pi*EmotorNMax/60))^2)*((2*pi*EmotorN(j)/60)-(2*pi*EmotorNMax/60))^2;
    end
end

GearboxOuputN = (GearRatio)*EmotorN;
GearboxOuputT =  (1/GearRatio)*EmotorTorque;

EmotorPowerMaxSet = interp1(EmotorN, EmotorPower, CompressorNset);
EmotorPowerMaxSetGearbox = interp1(GearboxOuputN, EmotorPower, CompressorNset);          %
eta_mech_gear

EmotorTorqueSet = interp1(EmotorN, EmotorTorque,  CompressorNset);
EmotorTorqueSetgearbox = interp1(GearboxOuputN, GearboxOuputT,  CompressorNset);

%% Compressor POWER and TORQUE Calculation
Compressormdot = CompressorQdot*rhoambient;
CompressordeltaT = (1./Compressoreta)* Tambient.* (CompressorPi.^((kappa-1)/kappa)-1);  %
Compressor temperature rise [C / K]
CompressorPower = Compressormdot.*Cpa.*CompressordeltaT;                                %
Compressor Power [W]

CompressorTorque = CompressorPower./(2*pi*CompressorN/60);                              %
Compressor Torque [Nm]

CompressorTOut = Tambient+CompressordeltaT;                                             %
Compressor Outlet Temperature [K]
CompressorpOut = pambient*CompressorPi;                                                 %
Compressor Outlet Pressure [Pa]

AlphaCompressor = CompressorTorque./(Icompressor+Iturbine);
t = (2*pi*CompressorN/60)./AlphaCompressor;

for k = 1:1:length(CompressorN)
    for j = 1:1:length(CompressorNset)
        if CompressorN(k) == CompressorNset(j)
            EmotorCompressorPowerSet(k) = CompressorPower(k) < EmotorPowerMaxSet(j);
            EmotorCompressorPowerSetGearbox(k) = CompressorPower(k) <
EmotorPowerMaxSetGearbox(j);
            EmotorPowerCompressor(k, :) = EmotorPowerMaxSet(j);
            EmotorPowerCompressorGearbox(k, :) = EmotorPowerMaxSetGearbox(j);
            EmotorTorqueCompressor(k,:) = EmotorTorqueSet(j);
            EmotorTorqueCompressorGearbox(k,:) = EmotorTorqueSetgearbox(j);
        end
    end
end

%% Turbine Map Interpolation
RPMsInterp = CompressorNset;
RPMset = ismember(CompressorN,RPMsInterp);
Indices1 = find(RPMset);

for k = 1:1:length(RPMsInterp)
[TurbineInterpRPM, TurbineInterpmdot, TurbineInterpPi, TurbineInterpeta] =
TurbineMapInterpolation(RPMsInterp(k),TurbineN, TurbineNset, Turbinemdot, TurbinePi,
Turbineeta);
RPMset = ismember(CompressorN,RPMsInterp(k));
Indices = find(RPMset);
TurbinemdotHULP = Compressormdot(Indices);
IndicesN = [IndicesN;Indices];
    for j = 1:1:length(TurbinemdotHULP)
        difference = TurbinemdotHULP(j)-TurbineInterpmdot;
        [minumum, indexmin] = min(abs(difference));
        TurbinemdotCorHulp(j, 1) = TurbineInterpmdot(indexmin);
        TurbinePiCorHulp(j, 1) = TurbineInterpPi(indexmin);
        TurbineetaCorHulp(j, 1) = TurbineInterpeta(indexmin);
```

```
    end
TurbineInterpmdotset = [TurbineInterpmdotset; TurbinemdotCorHulp];
TurbineInterpPiset = [TurbineInterpPiset; TurbinePiCorHulp];
TurbineInterpetaset = [TurbineInterpetaset; TurbineetaCorHulp];

TurbinemdotHULP = [];
TurbinemdotCorHulp = [];
TurbinePiCorHulp = [];
TurbineetaCorHulp = [];
end

CompressorTOutset = CompressorTOut(IndicesN);
CompressorpOutset = CompressorpOut(IndicesN);

TurbineInterpdeltaTset = TurbineInterpetaset.* CompressorTOutset .* (1-
(1./TurbineInterpPiset).^((gamma-1)/gamma));
TurbineInterpTOut = CompressorTOutset +TurbineInterpdeltaTset;
TurbineInterppOut = CompressorpOutset .*TurbineInterpPiset;
TurbineInterpPower = TurbineInterpmdotset.*Cpe.*TurbineInterpdeltaTset;
TurbineInterpTorque = TurbineInterpPower./(2*pi*CompressorN(IndicesN)/60);                %
Turbine Torque [Nm]
AlphaTurbineInterp = TurbineInterpTorque./(Icompressor+Iturbine);
t_turbine = (2*pi*CompressorN(IndicesN)/60)./AlphaTurbineInterp;

%% Combining of electric motor and turbine POWER and TORQUE
CombinedPower = EmotorPowerCompressor;
CombinedPowerGearbox = EmotorPowerCompressorGearbox;

CombinedTorque = EmotorTorqueCompressor;
CombinedTorqueGearbox = EmotorTorqueCompressorGearbox;

for ii = 1:1:length(CombinedPower)
    for jj = 1:1:length(Indices1)
        if ii == Indices1(jj)
        CombinedPower(ii) = CombinedPower(ii) + TurbineInterpPower(jj);
        CombinedPowerGearbox(ii) = CombinedPowerGearbox(ii) + TurbineInterpPower(jj);
        CombinedTorque(ii) = CombinedTorque(ii) + TurbineInterpTorque(jj);
        CombinedTorqueGearbox(ii) = CombinedTorqueGearbox(ii) + TurbineInterpTorque(jj);
        end
    end
end

CombinedPowerSet = CompressorPower < CombinedPower;
CombinedPowerSetGearbox = CompressorPower < CombinedPowerGearbox;

CombinedTorqueSet = abs(CompressorTorque) < abs(CombinedTorque);
EmotorTorqueReach = CompressorTorque < EmotorTorqueCompressor;

CombinedTorqueSetGearbox = abs(CompressorTorque) < abs(CombinedTorqueGearbox);
EmotorTorqueReachGearbox = CompressorTorque < EmotorTorqueCompressorGearbox;
BothConditions = logical(CombinedPowerSetGearbox);

%%
RecoveryFactorPower = TurbineInterpPower./CompressorPower*100;

%% Power & Efficiency lines
Compressormdotrange = linspace(0, max(Compressormdot), 200);
CompressorPirange = linspace(1, max(CompressorPi), 200);
[XCompressormdot, YCompressorPirange] = meshgrid(Compressormdotrange, CompressorPirange);
ZPowerCompressor = griddata(Compressormdot, CompressorPi, CompressorPower, XCompressormdot,
YCompressorPirange, 'natural');
ZToutCompressor = griddata(Compressormdot, CompressorPi, CompressorTOut, XCompressormdot,
YCompressorPirange);
ZTorqueCompressor = griddata(Compressormdot, CompressorPi, CompressorTorque, XCompressormdot,
YCompressorPirange);

%% Compressor Map & Turbine Map & Electro motor Charateristic
%% Plot Raw Compressor Map
figure('Name','Compressor Map')
set(gcf, 'Position', [100 100 900 600])
for j = 1:1:length(CompressorNset)
    RPMset = ismember(CompressorN,CompressorNset(j,:));
    Indices = find(RPMset);
    plot(Compressormdot(Indices), CompressorPi(Indices), '-o', 'DisplayName', sprintf('%4.f
RPM', CompressorNset(j)))
    hold on
end
```

```
title('Compressor Map 7804741A');
xlabel('Air Flow [kg/s]')
ylabel('Compression Ratio [-]')
xlim([0.5*min(Compressormdot) 1.05*max(Compressormdot)])
ylim([1 1.05*max(CompressorPi)])
grid on
legend('location', 'northeastoutside')

%% Plot Compressor map POWER
figure('Name','Power of Compressor Map')
set(gcf, 'Position', [100 100 950 600])
for j = 1:1:length(CompressorNset)
    RPMset = ismember(CompressorN,CompressorNset(j,:));
    Indices = find(RPMset);
    plot(Compressormdot(Indices), CompressorPi(Indices), '-o', 'DisplayName', sprintf('%4.f RPM', CompressorNset(j)))
    hold on
end
plot(Compressormdot(CombinedPowerSet), CompressorPi(CombinedPowerSet), 'b.', 'MarkerSize', 20, 'DisplayName', 'Gained with turbine power')
plot(Compressormdot(EmotorCompressorPowerSet), CompressorPi(EmotorCompressorPowerSet), 'r.', 'MarkerSize', 20, 'DisplayName', 'Only with electric motor power')
title('Compressor Map 7804741A Setup EM150 no Gearbox');
xlabel('Air Flow [kg/s]')
ylabel('Compression Ratio [-]')
xlim([0.5*min(Compressormdot) 1.05*max(Compressormdot)]) %
ylim([1 1.05*max(CompressorPi)])%
grid on
legend('location', 'eastoutside')

%% Plot Compressor map TORQUE
figure('Name','Torque of Compressor Map')
set(gcf, 'Position', [100 100 900 600])
for j = 1:1:length(CompressorNset)
    RPMset = ismember(CompressorN,CompressorNset(j,:));
    Indices = find(RPMset);
    plot(Compressormdot(Indices), CompressorPi(Indices),'-o', 'DisplayName', sprintf('%4.f RPM', CompressorNset(j)))
    hold on
end
plot(Compressormdot(CombinedTorqueSet), CompressorPi(CombinedTorqueSet), 'b.', 'MarkerSize', 20, 'DisplayName', 'Gained with turbine torque')
plot(Compressormdot(EmotorTorqueReach), CompressorPi(EmotorTorqueReach), 'r.', 'MarkerSize', 20, 'DisplayName', 'Only with electric motor torque')
title('Compressor Map 7804741A Setup EM150 no Gearbox');
xlabel('Air Flow [kg/s]')
ylabel('Compression Ratio [-]')
xlim([0.5*min(Compressormdot) 1.05*max(Compressormdot)]) %
ylim([1 1.05*max(CompressorPi)])%
grid on
legend('location', 'eastoutside')

%% Plot Turbine map
figure('Name','Turbine Map')
set(gcf, 'Position', [100 100 900 600])
for j = 1:1:length(TurbineNset)
    RPMset = ismember(TurbineN,TurbineNset(j,:));
    Indices = find(RPMset);
    plot(Turbinemdot(Indices), TurbinePi(Indices), '-o', 'DisplayName', string(TurbineNset(j)))
    hold on
end
title('Turbine Map 74035A.');
xlabel('Air Flow [kg/s]')
ylabel('Expansion Ratio [-]')
xlim([0.5*min(Turbinemdot) 1.05*max(Turbinemdot)])
ylim([1 1.05*max(TurbinePi)])
grid on
legend('location', 'northeastoutside')

%% Plot Electric Characteristics
%% POWER Plot electric motor
figure('Name','Electromotor characteristics Power')
set(gcf, 'Position', [100 100 900 600])
line(EmotorN,EmotorPower,'Color','k','DisplayName', 'electromotor operation line');
ax(1)=gca;
set(ax(1),'Position',[0.12 0.12 0.75 0.70])
```

```
set(ax(1),'XColor','k','YColor','k');
set(ax(1),'XLim',[min(EmotorN) max(EmotorN)]);
ylim([0, 1.05*max(EmotorPower)])
grid on
ax(2)=axes('Position',get(ax(1),'Position'),...
   'XAxisLocation','top',...
   'YAxisLocation','right',...
   'Color','none',...
   'XColor','r','YColor','k');
set(ax,'box','off')
line(GearboxOuputN,EmotorPower,'Color','r','Parent',ax(2), 'DisplayName', 'gearbox operation
line','LineStyle','-');
set(ax(2),'XLim',[min(GearboxOuputN) max(GearboxOuputN)]);
ylim([0, 1.05*max(EmotorPower)])
grid on
title('Electromotor characteristics')
set(get(ax(1),'xlabel'),'string','Electro Motor rotational speed [RPM]')
set(get(ax(2),'xlabel'),'string','Gearbox output rotational speed [RPM]')
set(get(ax(1),'ylabel'),'string','Power [W]')

%% TORQUE Plot electric motor
figure('Name','Electromotor characteristics Torque')
set(gcf, 'Position', [100 100 900 600])
hl1=line(EmotorN,EmotorTorque,'Color','k','DisplayName', 'electromotor operation line');
ax(1)=gca;
set(ax(1),'Position',[0.12 0.12 0.75 0.70])
set(ax(1),'XColor','k','YColor','k');
set(ax(1),'XLim',[min(EmotorN) max(EmotorN)]);
ylim([0, 1.05*max(EmotorTorque)])
grid on
ax(2)=axes('Position',get(ax(1),'Position'),...
   'XAxisLocation','top',...
   'YAxisLocation','right',...
   'Color','none',...
   'XColor','r','YColor','k');
set(ax,'box','off')
hl2=line(GearboxOuputN,GearboxOuputT,'Color','r','Parent',ax(2), 'DisplayName', 'gearbox
operation line');
set(ax(2),'XLim',[min(GearboxOuputN) max(GearboxOuputN)]);
ylim([0, 1.05*max(EmotorTorque)])
grid on
title('Electromotor characteristics')
set(get(ax(1),'xlabel'),'string','Electro Motor rotational speed [RPM]')
set(get(ax(2),'xlabel'),'string','Gearbox output rotational speed [RPM]')
set(get(ax(1),'ylabel'),'string','Torque [Nm]')

%% ALTERNATIV SINGLE OPERATING POINT PREDICTION
SelectionPoint = 50;                                                    % Selection of
Operating point
SinglePointRPM = CompressorN(SelectionPoint);
SinglePointCompressorPower = CompressorPower(SelectionPoint);
SinglePointPi = CompressorPi(SelectionPoint);
SinglePointEta = Compressoreta(SelectionPoint);
SinglePointCompressormdot = Compressormdot(SelectionPoint);
SinglePointTurbinePower = TurbineInterpPower(SelectionPoint);
SinglePointTurbineDeltaT = TurbineInterpdeltaTset(SelectionPoint);
SinglePointTurbinemdot = TurbineInterpmdotset(SelectionPoint);
SinglePointExtraPi = linspace(min(CompressorPi), max(CompressorPi));
SinglePointExtraDeltaT =  (1./SinglePointEta)*Tambient.* (SinglePointExtraPi.^((kappa-
1)/kappa)-1);
SinglePointExtramdot =
(SinglePointTurbinePower+SinglePointCompressorPower)./(Cpe*SinglePointExtraDeltaT);

figure('Name','Compressor Map Zoomed in')
set(gcf, 'Position', [100 100 900 600])
contour(XCompressormdot, YCompressorPirange, ZPowerCompressor,'LevelStep', 500, 'DisplayName',
'Lines equal power')
hold on
for j = 1:1:length(CompressorNset(1:13))
    RPMset = ismember(CompressorN,CompressorNset(j,:));
    Indices = find(RPMset);
    plot(Compressormdot(Indices), CompressorPi(Indices), '-o', 'DisplayName', sprintf('%4.f
RPM', CompressorNset(j)), 'LineWidth', 2)
    hold on
end
plot(Compressormdot(SelectionPoint), CompressorPi(SelectionPoint), '*', 'Color',
'r','LineWidth', 7, 'DisplayName', 'Operation point of interest')
```

```
plot(SinglePointExtramdot, SinglePointExtraPi, '-', 'Color', 'r','LineWidth', 3,
'DisplayName', 'Line of extra range of interest (constant efficiency)')
colorbar
title('Compressor Map 7804741A');
xlabel('Air Flow [kg/s]')
ylabel('Compression Ratio [-]')
xlim([0.5*min(Compressormdot) 1.05*0.2])
ylim([1 1.05*1.5])
grid on
legend('location', 'northeastoutside')
```

Compressor interpolation function

```
function [CompressorInterpRPM, CompressorInterpQdot, CompressorInterpPi, CompressorInterpeta]
= CompressorMapInterpolation(RPMinterp,CompressorN, CompressorNset, CompressorQdot,
CompressorPi, Compressoreta)
%% function [CompressorInterpRPM, CompressorInterpQdot, CompressorInterpPi,
CompressorInterpeta] = CompressorMapInterpolation(RPMinterp,CompressorN, CompressorNset,
CompressorQdot, CompressorPi, Compressoreta)
% This function interpolates a new rpm isoline in the known compressor map
% Function input: RPM of interest, all data arrays of the compressor map
(RPMinterp,CompressorN, CompressorNset, CompressorQdot, CompressorPi, Compressoreta)
% Function Output: the new map line (CompressorInterpRPM, CompressorInterpQdot,
CompressorInterpPi, CompressorInterpeta)
%%
if RPMinterp < min(CompressorNset)
    CompressorRPMinterpValues = [CompressorNset(1), CompressorNset(2)]';
    InterpShift = (RPMinterp-0)/(CompressorRPMinterpValues(1)-0);
elseif RPMinterp > max(CompressorNset)
    CompressorRPMinterpValues = [CompressorNset(end-1), CompressorNset(end)]';
    InterpShift = (RPMinterp-CompressorRPMinterpValues(1))/(CompressorRPMinterpValues(2)-
CompressorRPMinterpValues(1));
else
    CompressorRPMinterpValues = [CompressorNset(find(CompressorNset <= RPMinterp, 1, 'last'
)), CompressorNset(find(CompressorNset >= RPMinterp, 1, 'first'))]';
    InterpShift = (RPMinterp-CompressorRPMinterpValues(1))/(CompressorRPMinterpValues(2)-
CompressorRPMinterpValues(1));
end

CompressorRPMIndices =
[ismember(CompressorN,CompressorRPMinterpValues(1)),ismember(CompressorN,CompressorRPMinterpVa
lues(2))];

CompressorRPMinterpSet =
[ones(sum(CompressorRPMIndices(:,1)~=0),1)*CompressorRPMinterpValues(1),
ones(sum(CompressorRPMIndices(:,1)~=0),1)*CompressorRPMinterpValues(2)];
CompressorInterpRPM = RPMinterp*ones(length(CompressorRPMinterpSet),1);

RPMlineSelect = 1;

if RPMinterp == CompressorRPMinterpValues(1)
   RPMset = ismember(CompressorN,RPMinterp);
   Indices = find(RPMset);
   CompressorInterpQdot = CompressorQdot(Indices);
   CompressorInterpPi = CompressorPi(Indices);
   CompressorInterpeta =  Compressoreta(Indices);

elseif RPMinterp == CompressorRPMinterpValues(2)
   RPMset = ismember(CompressorN,RPMinterp);
   Indices = find(RPMset);
   CompressorInterpQdot = CompressorQdot(Indices);
   CompressorInterpPi = CompressorPi(Indices);
   CompressorInterpeta =  Compressoreta(Indices);
   RPMlineSelect = 2;

else
    if length(find(CompressorRPMIndices(:,1))) < length(find(CompressorRPMIndices(:,2)))
        CompressorRPMIndices([find(CompressorRPMIndices(:,1), 1, 'last'
)+length(find(CompressorRPMIndices(:,1)))+1:find(CompressorRPMIndices(:,2), 1, 'last' )],2) =
0;
        Indices = [find(CompressorRPMIndices(:,1)), find(CompressorRPMIndices(:,2))];
        RPMlineSelect = 2;
    elseif length(find(CompressorRPMIndices(:,1))) > length(find(CompressorRPMIndices(:,2)))
```

```
        CompressorRPMIndices([find(CompressorRPMIndices(:,1), 1
)+length(find(CompressorRPMIndices(:,2))):find(CompressorRPMIndices(:,1), 1, 'last' )],1) = 0;
        Indices = [find(CompressorRPMIndices(:,1)), find(CompressorRPMIndices(:,2))];
        RPMlineSelect = 2;
    else
        Indices = [find(CompressorRPMIndices(:,1)), find(CompressorRPMIndices(:,2))];
        RPMlineSelect = 2;
    end

    if RPMinterp < min(CompressorNset)
        CompressorPiMinimum =
1*ones(length(find(CompressorN==CompressorRPMinterpValues(1))),1);

        CompressorQdotSet = CompressorQdot(Indices);
        CompressorPiSet = CompressorPi(Indices);

        a = (CompressorPiSet(:, 2)-CompressorPiSet(:, 1))./(CompressorQdotSet(:, 2)-
CompressorQdotSet(:, 1));
        b = CompressorPiSet(:, 1) - a.*CompressorQdotSet(:, 1);
        CompressorQdotMinimum = (CompressorPiMinimum - b)./a;

        CompressorQdotSet = [CompressorQdotMinimum, CompressorQdotSet(:, 1)];
        CompressorPiSet = [CompressorPiMinimum, CompressorPiSet(:, 1)];

        RPMlineSelect = 1;

    else
        CompressorQdotSet = CompressorQdot(Indices);
        CompressorPiSet = CompressorPi(Indices);

    end
        Length = ((CompressorQdotSet(:, 2)-CompressorQdotSet(:, 1)).^2 + (CompressorPiSet(:,
2)-CompressorPiSet(:, 1)).^2).^(0.5);
        ScaledLength = InterpShift*Length;

        CompressorInterpQdot = ((ScaledLength./Length).*(CompressorQdotSet(:, 2)-
CompressorQdotSet(:, 1))+CompressorQdotSet(:, 1));
        CompressorInterpPi = ((ScaledLength./Length).*(CompressorPiSet(:, 2)-
CompressorPiSet(:, 1))+CompressorPiSet(:, 1));
        CompressorInterpeta = Compressoreta(Indices(:,RPMlineSelect));
end

end
```

Turbine interpolation function

```
function [TurbineInterpRPM, TurbineInterpmdot, TurbineInterpPi, TurbineInterpeta] =
TurbineMapInterpolation(RPMinterp, TurbineN, TurbineNset, Turbinemdot, TurbinePi, Turbineeta)
%% function [TurbineInterpRPM, TurbineInterpmdot, TurbineInterpPi, TurbineInterpeta] =
TurbineMapInterpolation(RPMinterp,TurbineN, TurbineNset, Turbinemdot, TurbinePi, Turbineeta)
% This function interpolates a new rpm isoline in the known turbine map
% Function input: RPM of interest, all data arrays of the turbine map [TurbineInterpRPM,
TurbineInterpmdot, TurbineInterpPi, TurbineInterpeta]
% Function Output: the new map line (TurbineInterpRPM, TurbineInterpmdot, TurbineInterpPi,
TurbineInterpeta)(RPMinterp, TurbineN, TurbineNset, Turbinemdot, TurbinePi, Turbineeta)
%%
if RPMinterp < min(TurbineNset)
    TurbineRPMinterpValues = [TurbineNset(1), TurbineNset(2)]';
    InterpShift = (RPMinterp-0)/(TurbineRPMinterpValues(1)-0);
elseif RPMinterp > max(TurbineNset)
    TurbineRPMinterpValues = [TurbineNset(end-1), TurbineNset(end)]';
    InterpShift = (RPMinterp-TurbineRPMinterpValues(1))/(TurbineRPMinterpValues(2)-
TurbineRPMinterpValues(1));
else
    TurbineRPMinterpValues = [TurbineNset(find(TurbineNset <= RPMinterp, 1, 'last' )),
TurbineNset(find(TurbineNset >= RPMinterp, 1, 'first'))]';
    InterpShift = (RPMinterp-TurbineRPMinterpValues(1))/(TurbineRPMinterpValues(2)-
TurbineRPMinterpValues(1));
end

TurbineRPMIndices =
[ismember(TurbineN,TurbineRPMinterpValues(1)),ismember(TurbineN,TurbineRPMinterpValues(2))];

TurbineRPMinterpSet = [ones(sum(TurbineRPMIndices(:,1)~=0),1)*TurbineRPMinterpValues(1),
ones(sum(TurbineRPMIndices(:,1)~=0),1)*TurbineRPMinterpValues(2)];
TurbineInterpRPM = RPMinterp*ones(length(TurbineRPMinterpSet),1);
```

```
RPMlineSelect = 1;

if RPMinterp == TurbineRPMinterpValues(1)
    RPMset = ismember(TurbineN,RPMinterp);
    Indices = find(RPMset);
    TurbineInterpmdot = Turbinemdot(Indices);
    TurbineInterpPi = TurbinePi(Indices);
    TurbineInterpeta =  Turbineeta(Indices);
elseif RPMinterp == TurbineRPMinterpValues(2)
    RPMset = ismember(TurbineN,RPMinterp);
    Indices = find(RPMset);
    TurbineInterpmdot = Turbinemdot(Indices);
    TurbineInterpPi = TurbinePi(Indices);
    TurbineInterpeta =  Turbineeta(Indices);
else
    if length(find(TurbineRPMIndices(:,1))) < length(find(TurbineRPMIndices(:,2)))
            TurbineRPMIndices([find(TurbineRPMIndices(:,1), 1, 'last'
)+length(find(TurbineRPMIndices(:,1)))+1:find(TurbineRPMIndices(:,2), 1, 'last' )],2) = 0;
            Indices = [find(TurbineRPMIndices(:,1)), find(TurbineRPMIndices(:,2))];
    elseif length(find(TurbineRPMIndices(:,1))) > length(find(TurbineRPMIndices(:,2)))
            TurbineRPMIndices([find(TurbineRPMIndices(:,1), 1
)+length(find(TurbineRPMIndices(:,2))):find(TurbineRPMIndices(:,1), 1, 'last' )],1) = 0;
            Indices = [find(TurbineRPMIndices(:,1)), find(TurbineRPMIndices(:,2))];
            RPMlineSelect = 2;
    else
            Indices = [find(TurbineRPMIndices(:,1)), find(TurbineRPMIndices(:,2))];
            RPMlineSelect = 2;
    end

    if RPMinterp < min(TurbineNset)
        TurbinePiMinimum = 1*ones(length(find(TurbineN==TurbineRPMinterpValues(1))),1);

       TurbinemdotSet = Turbinemdot(Indices);
        TurbinePiSet = TurbinePi(Indices);

        a = (TurbinePiSet(:, 2)-TurbinePiSet(:, 1))./(TurbinemdotSet(:, 2)-TurbinemdotSet(:,
1));
        b = TurbinePiSet(:, 1) - a.*TurbinemdotSet(:, 1);
        TurbinemdotMinimum = (TurbinePiMinimum - b)./a;

        TurbinemdotSet = [TurbinemdotMinimum, TurbinemdotSet(:, 1)];
        TurbinePiSet = [TurbinePiMinimum, TurbinePiSet(:, 1)];

        RPMlineSelect = 1;

    else
        TurbinemdotSet = Turbinemdot(Indices);
        TurbinePiSet = TurbinePi(Indices);

    end
        Length = ((TurbinemdotSet(:, 2)-TurbinemdotSet(:, 1)).^2 + (TurbinePiSet(:, 2)-
TurbinePiSet(:, 1)).^2).^(0.5);
        ScaledLength = InterpShift*Length;

        TurbineInterpmdot = ((ScaledLength./Length).*(TurbinemdotSet(:, 2)-TurbinemdotSet(:,
1))+TurbinemdotSet(:, 1));
        TurbineInterpPi = ((ScaledLength./Length).*(TurbinePiSet(:, 2)-TurbinePiSet(:,
1))+TurbinePiSet(:, 1));
        TurbineInterpeta = Turbineeta(Indices(:,RPMlineSelect));

end
end
```