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DEVELOPMENT OF A TURBOCHARGER FOR A FORMULA SAE CAR ENGINE

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ABSTRACT

This thesis explores the development of a turbocharger system for a Formula SAE car, specifically for the Scuderia Tor Vergata's car, aiming to demonstrate the potential for increased power and performance through its implementation. The research encompasses the fundamental concepts of turbocharging, theoretical modeling of engine and turbocharger selection, and subsequent simulation using Simulink software. The study begins by examining turbocharger technology, its principles, advantages, and challenges. Theoretical models are then used to predict performance based on parameters such as compression ratio and turbine size. Simulink simulations compare the engine's behavior with and without the turbocharger, illustrating significant improvements in power output and overall performance. This thesis provides valuable insights into the design and implementation of turbocharger systems for Formula SAE vehicles, showcasing their potential to enhance power and performance, thereby contributing to optimizing vehicle performance on the track.

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INTRODUCTION

In the exciting world of automotive engineering, the Formula Student competition has gained popularity in recent years as a platform for students worldwide to showcase their skills in designing and building high-performance cars. One of the key components in these vehicles is the engine, where performance and efficiency are crucial for achieving optimal track performance.

Within this context, this thesis focuses on the development of a turbocharger specifically designed for a Formula Student car engine, with the main objective of improving both performance and efficiency of this system. The turbocharger is an essential component for increasing the power output of an engine by compressing the air entering the cylinders, allowing for a greater amount of fuel-air mixture and thus generating higher output power.

As the Formula Student competition becomes more competitive, an innovative approach and intelligent design are required to optimize the engine's performance, maximize efficiency, and comply with the regulatory constraints imposed by the competition. Therefore, this thesis will concentrate on the design, development, and optimization of a turbocharger that meets the specific requirements of a Formula Student car, taking into account aspects such as weight limitations, emission regulations, and physical space constraints.

To achieve this goal, an exhaustive study of theoretical principles and design methodologies of turbochargers will be conducted, focusing on key parameters that influence their performance, such as compression ratio, impeller geometry, gas flow, and pressure management. Simulation and computational modeling tools will be utilized to evaluate different configurations and optimize the turbocharger's performance based on the specific requirements of the Formula Student engine.

The ultimate outcome of this thesis will be the proposal of a turbocharger specifically optimized for the Formula Student car engine, enabling a significant improvement in its performance and efficiency compared to conventional solutions. It is anticipated that this work will contribute to the advancement and continuous development of the Formula Student competition by providing an innovative and competitive solution for participating teams.

CHAPTER 1: BACKGROUND

1.1 THE INTERNAL COMBUSTION ENGINE

Internal combustion engines have been an integral part of the automotive industry for over a century. These devices convert the chemical energy contained in fossil fuels into mechanical energy used to propel vehicles, machinery, and various equipment.

On this chapter, we will explore into the basics of combustion, internal combustion engines, and forced induction. These topics are crucial to understanding the operation and efficiency of internal combustion engines, as well as the technologies used to enhance their performance.

1.1.1 Combustion basics

Combustion is a spontaneous reaction that occurs between two substances: a fuel and an oxidizer, which is typically air. In the intake mixture, these substances must be present in appropriate quantities.

The motion of the piston within the cylinder is caused by the pressure energy released during the combustion of the fresh charge. In internal combustion engines with spark ignition, such as our system, combustion is initiated by the spark generated by the spark plug, and its optimization is crucial for engine performance.

Perfect combustion occurs when they are in a stoichiometric ratio, which varies depending on whether gasoline or E-85 is used. The stoichiometric AFR changes from 14.7:1 for gasoline to 8:1 for E-85.

It is important to note that the AFR is a ratio of masses. For example, an AFR of 8:1 means that for every unit mass of fuel, 8 units mass of air are required for complete combustion. However, this implies that, despite the benefits in terms of raising the detonation level, ethanol will result in higher fuel consumption for the vehicle.

While the stoichiometric condition is an ideal calculation assumption based on an ideal fluid a slightly richer mixture than stoichiometric is used to compensate for losses due to actual fuel efflux in the intake ducts.

1.1.2 Engine basics

The Otto cycle is a thermodynamic process that describes the basic operation of spark-ignition internal combustion engines, such as gasoline engines. This cycle consists of four main stages: intake, compression, combustion, and exhaust.

1. Intake: During this phase, the air-fuel mixture is drawn into the cylinder as the piston moves downward and the intake valve opens. The mixture enters the combustion chamber in preparation for the next steps.
2. Compression: As the piston moves upward, it compresses the air-fuel mixture inside the cylinder. This compression increases the pressure and temperature of the mixture. The goal is to achieve a high compression ratio, which leads to more efficient combustion and power output.
3. Combustion: At the top of the compression stroke, a spark plug generates a spark that ignites the compressed air-fuel mixture. The spark ignites the mixture, resulting in a rapid and controlled combustion process. This combustion releases a considerable amount of energy in the form of heat and expanding gases, creating a force that pushes the piston downward.
4. Exhaust: Once the power stroke is complete, the exhaust valve opens, allowing the burned gases to exit the cylinder. The piston moves upward, pushing the remaining exhaust gases out through the open exhaust valve and into the exhaust system.

The important thing to note from the 4-stroke engine is that power is only produced for 1 out of the 4 strokes. Displacement has a large effect of how much power can be produced in that one stroke, because a larger displacement allows more air/fuel mixture into the cylinder while the intake valve is open. Combusting more air/fuel mixture creates a higher pressure, and consequently more power.

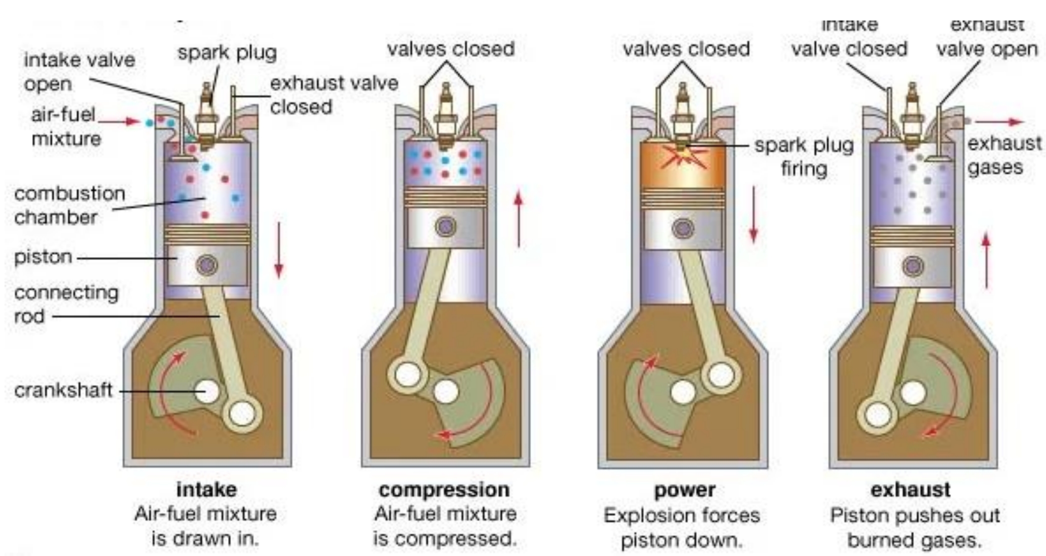

Figure 1: Four Stroke cycle

1.1.3 Forced induction

In a 4-stroke engine, the intake stroke draws in air through the intake manifold, and the fuel is injected into the mixture. In a naturally aspirated engine, atmospheric pressure alone forces the air into the cylinder. However, in forced induction, an additional device is employed to compress the incoming air, increasing its density and allowing more oxygen to be packed into the cylinder. This technique increases power output and improve efficiency. The two main methods of forced induction are turbocharging and supercharging.

Turbocharging uses a turbine-compressor that is powered by the engine's exhaust gases. The exhaust gases flow through a turbine wheel, which drives a compressor wheel that compresses the incoming air. This compressed air is then directed into the intake manifold, resulting in increased power output.

On the other hand, supercharging uses a belt-compressor, or supercharger, which is directly connected to the engine's crankshaft. As the engine spins, the supercharger compresses the air and delivers it to the intake manifold. This method provides an immediate boost in power, as the compressor is not dependent on exhaust gases like a turbocharger.

Forced induction offers several advantages, mainly, it allows engines to produce more power without increasing the engine's displacement. However, it also has some challenges. The increased pressure and temperature in the combustion chamber can cause higher mechanical and thermal stresses on engine components. Additionally, forced induction systems can be complex and require careful tuning and maintenance.

1.1.4 Turbocharge

The operation in which the entire (or a part of) fresh charge is pre-compressed outside the working cylinder in order to increase the mass of air or mixture that an engine can intake per cycle is called forced induction. In a four-stroke engine, as known, the charge is renewed through the piston's downward stroke to expel the exhaust gases and its subsequent upward stroke to draw in the fluid present in the intake manifold. It is considered "forced induction" when the density of the fresh charge in the intake manifold is increased to a value higher than that corresponding to the surrounding environment conditions using a compressor. The main purpose of forced induction is to introduce a greater mass of fresh charge into the cylinder than what is achieved through natural aspiration, in order to obtain higher power output with the same displacement. Turbocharging increases the intake air density. This results in a higher density ρ in the IMEP equation, leading to an increase in IMEP. An increase in IMEP leads to a higher engine torque T, according to the equation $T = IMEP * Vd$.

Finally, greater power P is obtained by multiplying the engine torque T by the angular velocity ω of the engine, as per the equation $P = T * \omega$.

Considering that internal combustion engines typically operate under non-steady-state conditions, while turbomachinery can be designed to accommodate variable flows over time but operate with higher efficiency under steady-state conditions, the latter solution can lead to the following configurations:

- Constant-pressure turbocharging: Wide exhaust ducts dampen flow fluctuations for a steady turbine flow. However, energy from exhaust gas kinetic energy is underutilized, dissipating and degrading in the large manifold volume, producing entropy.

- Pulse turbocharging: Small ducts connect each cylinder to the turbine, transferring kinetic energy as pressure waves. Grouping exhausts from multiple cylinders reduces flow unsteadiness, enabling efficient energy conversion and improving performance and efficiency.

Figure 2: Cutaway of a Turbocharger

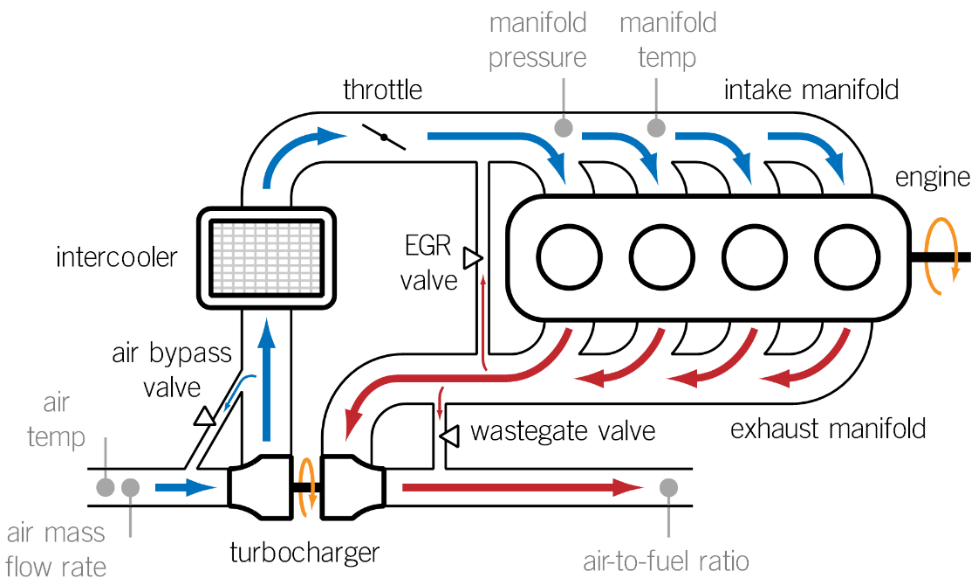

Figure 3: Turbocharged engine scheme

1.2 FORMULA SAE

Formula SAE is an international collegiate engineering competition where students from universities worldwide design, build, and compete with small-scale formula-style race cars. The competition is organized by the Society of Automotive Engineers (SAE).

The goal of Formula SAE is to provide students in the engineering world, experience that simulates the process of designing, manufacturing, and testing a high-performance race car. Participants are required to develop a prototype vehicle that meets specific design criteria and performance specifications.

The competition involves a series of static and dynamic events that evaluate different aspects of the race cars. The static events typically include design evaluation, cost analysis, and a business presentation, where teams present their engineering design choices, manufacturing cost considerations, and marketing strategies.

The dynamic events focus on evaluating the performance and capabilities of the cars. These events include acceleration tests, where the car's ability to accelerate from a standing start is measured. There are also skid pad tests to assess the car's lateral grip and maneuverability, and autocross events that test the car's handling and agility through a tight, timed course.

The points schedule for most Formula SAE events is:

Static Events	
Design event	150
Cost & Manufacturing Analysis Event	100
Presentation Event	75
Dynamic Events	
Acceleration Event	100
Skid pa**d** Event	75
Autocross Event	125
Fuel Economy Event	100
Endurance Event	275
Total Points Possible	**1000**

Table 1: Distribution of points in Formula SAE events

Overall, Formula SAE offers challenging platform for students to apply their engineering knowledge, develop practical skills, and gain a comprehensive understanding of the automotive design and performance aspects. It combines engineering excellence, innovation, teamwork, and a passion for motorsports into an exciting and rewarding experience for participating students.

1.2.1 Formula SAE rules

The vehicles must comply with a specific set of regulations designed primarily to ensure safety and limit performance, while allowing for a wide range of technical solutions. Regarding the engine, the most significant constraints are on the displacement (710 cc) and the requirement to incorporate a 20 mm diameter restrictor in the intake system. Typically, motorcycle-derived engines are used, with no specific requirements regarding the configuration. Additionally, the engine cannot be originally turbocharged or supercharged, but any forced induction must be designed and added during the vehicle's construction phase.

The car must include adequate protection for the driver in the event of a side impact or rollover. Furthermore, a system that absorbs a portion of the energy in case of a frontal impact must be implemented in the front portion of the vehicle. Since adapting the engines to comply with these regulations requires significant effort, understanding the functioning principles of the engines is crucial for effective design.

The only allowed sequence of components for turbocharged or supercharged engine is the following: restrictor, compressor, throttle body, engine.

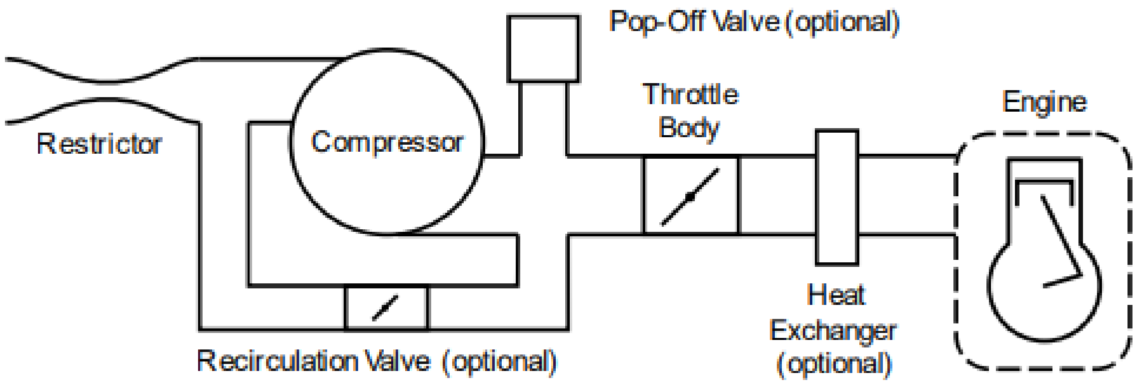

Figure 4: Sketch of engine disposition that follows Formula SAE rules.

About Turbochargers & Superchargers:

- The intake air may be cooled with an intercooler (a charge air cooler).

    a. It must be located downstream of the throttle body.

    b. Only ambient air may be used to remove heat from the intercooler system.

    c. Air to air and water to air intercoolers are permitted.

- The maximum allowable area of the inner diameter of the intake runner system between the restrictor and throttle body is 2825 mm2.

- Plenums must not be located anywhere upstream of the throttle body.

1.2.2 Air Restrictor

The power generated by an engine depends on the torque it produces and the rotational speed of the crankshaft. As the engine spins faster, it requires a greater amount of air for combustion. This is where the air restrictor comes into play. Its purpose is to restrict the airflow to prevent teams from harnessing excessive power by excessively revving the engine. The restrictor is a small device that the air must pass through, with specific dimensions such as 20mm for gasoline engines and 19mm for E-85 engines.

At higher engine speeds, the restrictor becomes a limiting factor, as it cannot allow sufficient airflow for the fuel to combust effectively. Teams have traditionally tackled this restriction by designing converging-diverging nozzles that optimize airflow and minimize losses. However, the restrictor still plays a significant role in limiting engine power output.

In summary, the air restrictor specification is implemented to control engine power by restricting airflow. It prevents teams from exploiting extremely high engine speeds to generate excessive power. While design optimizations like converging-diverging nozzles help mitigate the restriction's effects, the restrictor remains a key element in limiting power output.

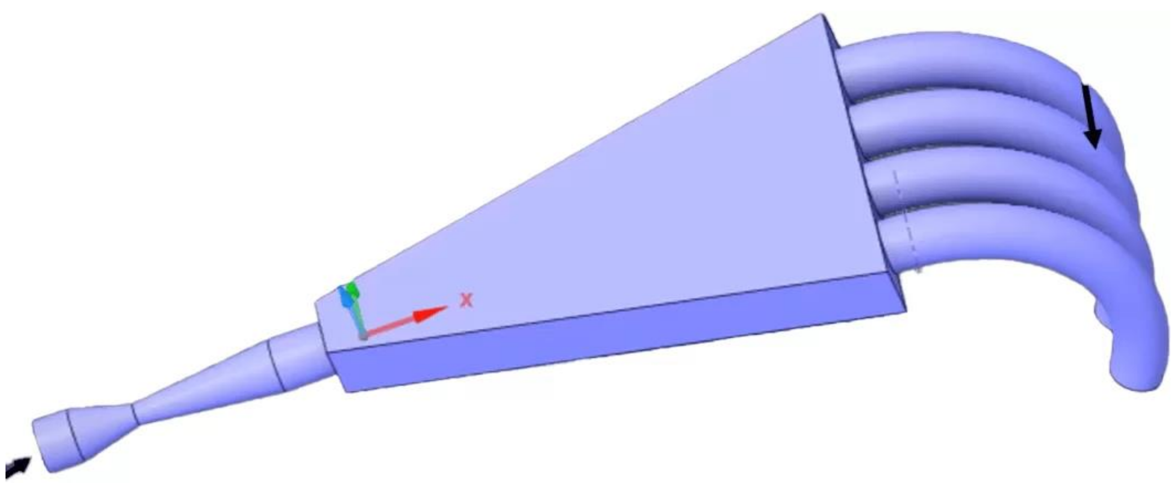

Figure 5: Intake Restrictor that follows Formula SAE rules.

CHAPTER 2: DESING CHOICES.

Scuderia Tor Vergata has decided to uphold its previous choices for both the engine and fuel selection. Extensive evaluations of the engine's performance and durability, along with comprehensive research on fuel options, led the team to conclude that the existing system best aligns with their objectives. On this chapter, we will elaborate on the rationale behind these decisions and why the team opted to retain the same system.

2.1 ENGINE CHOICE

Increasing the power of the single cylinder is great, but if it is not the best fit for the entire car then it should not be used. Table 2 below, uses past information from the 4-cylinder and single cylinder performance characteristics and compares them to the projected performance of a turbocharged single cylinder.

	(Weight)	4 Cylinder	Single Cylinder
Good Reliability	(5)	5	4
Power	(4)	5	1
Low Weight	(5)	1	5
Good Fuel Mileage	(2)	1	5
Uniqueness	(3)	1	3
Low Cost	(3)	5	4
Feasibility	(5)	2	5
Drivability	(4)	3	5
	TOTAL	**92**	**125**

Table 2: Decision matrix of engine choice

Analyzing in detail the entries in the table and the reasons, we have:

• **Good Reliability**: crucial for easily finding the engine and, more importantly, spare parts in case of breakage.

• **Power**: given regulatory constraints, having an engine with high power in its original configuration is preferable.

• **Low weight**: in a racing car, weight is a dominant factor, so the engine should be as light as possible.

• **Good Fuel Mileage**: highly important as cars are put to the test, especially during endurance races.

• **Uniqueness**: during the business plan evaluation, which represents a significant portion of the achievable score in static tests, the uniqueness of the project is assessed.

• **Low cost**: in addition to uniqueness, an essential aspect evaluated in the business plan is the cost-effectiveness of the vehicle, both in terms of production and product management.

• **Feasibility**: for the project to be economical, solutions with a good level of feasibility are necessary.

• **Drivability**: to win an event, excellent performance in dynamic tests is crucial. Having a difficult-to-handle engine does not assist the driver during the race, resulting in a loss of points.

From this decision matrix, it can be deduced that the **single-cylinder** solution is indeed the one that best meets the team's needs.

Among the various single-cylinder engines available on the market, the decision was made to adopt the ***KTM 690 LC4 MY 2014*** originally mounted on the ***DUKE R*** motorcycle.

This choice stemmed from the fact that it best meets the characteristics, as it:

• Has excellent availability of spare parts.

• Is the single cylinder engine with the highest specific power currently on the market.

• Is relatively lightweight (35 kg with fluids compared to the 65 kg of the CBR).

• Is cost-effective and reliable.

• Exhibits good flexibility and smoothness in power delivery.

This last evaluation was made considering the torque curve of the engine as declared by the manufacturer.

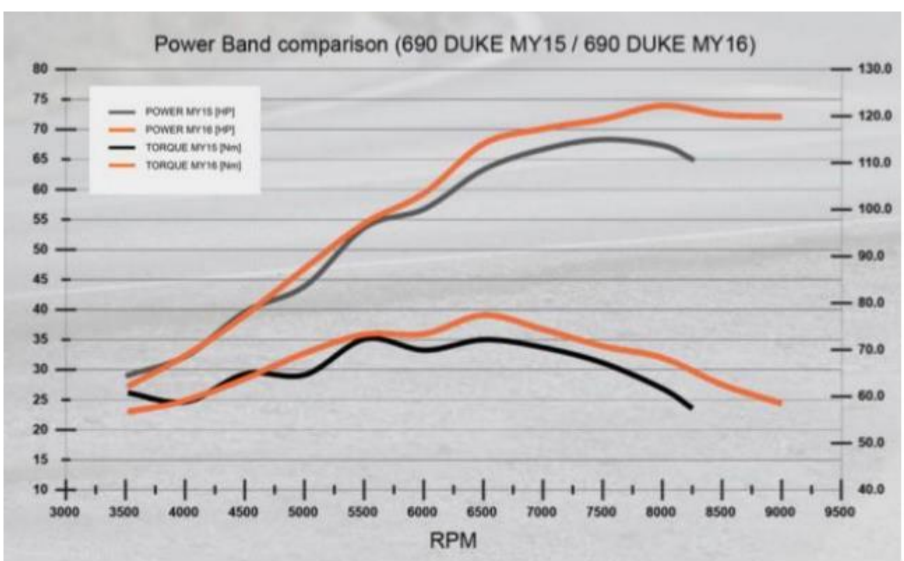

Figure 6: DUKE KTM 690 power and torque map. Comparison between MY 15 and MY 16

2.2 FUEL CHOICE

When preparing a high-performance racing engine, the choice of fuel is a critical decision. The type of fuel used directly affects the calculations and determines the appropriate compressor to be utilized. In the context of Formula SAE, two fuel options are allowed: RON 98 gasoline, which boasts an octane rating of 98, and E-85, an 85% ethanol and 15% gasoline blend with an equivalent octane rating of 110.

To make this choice, a decision matrix has been developed, just like for other components, to facilitate the selection process.

The decision matrix considers various factors such as engine requirements, performance objectives, and environmental considerations. By carefully analyzing these variables alongside the characteristics of each fuel option, we can make informed choices regarding their suitability for the specific application in a racing engine.

Ultimately, the fuel selection holds significance in achieving optimal performance and aligning with the desired goals of the competition.

	(Weight)	**E-85**	**RON 98**
Availability (Test)	(3)	2	5
Availability (Race)	(5)	5	5
Low Cost	(3)	3	1
Delivered Power	(4)	5	2
Consumption	(3)	4	2
	TOTAL	**72**	**57**

Table 3: Decision Matrix of fuel choice

As can be seen, the most influential factors in the decision are the availability during the race and the power that the fuel itself can deliver under the same conditions.

Regarding availability, it has a significant impact since during the competition, refueling can only be done by the organization, thus forcing teams to use the only two types of fuel provided by them. Any other type of fuel cannot be requested or brought by the individual team, as it would result in disqualification.

As for power, due to ethanol having a higher octane rating, it allows for reaching more critical pressure and temperature conditions in the combustion chamber, thereby enabling greater exploitation of the engine's capabilities.

2.2.1 Comparison of gasoline and E-85 properties

The evaluation carried out for fuel selection involved examining the characteristics of both options, which are summarized in the table below:

FUEL

PROPERTY	UNIT	GASOLINE	E-85
Octane number	RON	95	106
	MON		89
	AKI		97,5
Lower Heat Value	LHV [MJ/kg]	43	29,53
Stoichiometric air-to-fuel ratio	A/F	14,5	9,825
Vaporization Heat	Hv	380 ÷ 500	822 ÷ 840
Autoignition Temperature	[ºC]	370	365
Density at 15°C	ρ [kg/l]	0,745	0,783
Vapor Pressure	RVP [kPa]	60	43

Table 4: Properties of fuels RON 98 and E-85

Let's delve into a more detailed analysis of one of the most important factors named before:

- **RON** (Research Octane Number): This is a measure of the fuel's octane rating determined using the "Research Method" (ASTM-D 2700).
- **MON** (Motor Octane Number): This measures the fuel's octane rating using the "Motor Method" (ASTM-D 2699).
- **AKI** (Anti-Knock Index): This is the average value between RON and MON. The MON method is more stringent in terms of testing compared to the RON method.

  It involves preheating the air-fuel mixture before admission to the engine, and the ignition timing is higher and increases with the compression ratio. As a result, the MON octane number values are lower than the RON values.
- **LHV** (Lower Heat Value): Refers to its reduced energy content per unit of mass, resulting in less heat and power output when combusted. This can impact the efficiency and performance of combustion engines, requiring larger quantities of fuel to achieve the desired energy output.

• **Reid Vapor Pressure** (RVP): This measures the vapor pressure of petroleum products at 37.8°C, following the testing methodology defined by ASTM-D-323. It represents the fuel's volatility. A higher RVP indicates greater volatility, meaning the fuel is more likely to transition spontaneously from a liquid to a gaseous state. This factor is crucial during cold starts, where high volatility is required to evaporate the fuel under low-temperature conditions.

2.3 WAYS TO INCREASE POWER OF THE ENGINE

When assessing the engine, there are several key considerations that heavily influence the decision-making process. The foremost factor is reliability, as the engine's performance has historically been the least dependable aspect of the vehicle and must run consistently to successfully complete the competition. Equally important is the pursuit of increased power, which lies at the core of the entire endeavor. Consequently, this factor carries significant weight in the decision matrix.

While the engine's weight is a crucial aspect, it is the power-to-weight ratio that holds paramount importance. Achieving an optimal balance between power output and weight is essential for overall performance. Additionally, feasibility plays a vital role in the decision-making process. It is crucial to assess whether potential options can be effectively implemented, as pursuing an impractical or unfeasible choice would be counterproductive.

To aid in the selection process, a decision matrix has been developed, presenting a comprehensive overview of the various factors considered. This matrix serves as a valuable tool in objectively evaluating and comparing the available options, ensuring a well-informed decision is made for engine selection.

Power Increase Methods

GOALS	(weight)	Forced Induction	Intake	Exhaust	High Compression	Variable Valve Timing	Direct Injection
Good Reliability	(5)	4	4	4	3	2	2
Power Increase	(5)	5	1	1	1	3	2
Low Weight	(3)	3	5	5	5	4	4
Low Cost	(2)	2	3	3	4	1	1
Feasibility	(4)	4	1	1	3	1	1
	TOTAL	**76**	**48**	**48**	**54**	**53**	**38**

Table 5: Decision Matrix for Increasing Power

Analyzed methods are:

- **Forced Induction**: Forced induction, such as superchargers and turbochargers, is commonly used to increase engine power. By compressing the incoming air and forcing it into the combustion chamber at a higher pressure, more oxygen is available for combustion. This allows for a larger amount of fuel to be burned, resulting in increased power output.
- **Intake**: The intake system plays a crucial role in increasing engine power. It aims to optimize the flow of air into the combustion chamber. By improving the intake design, including the air filter, intake manifold, and throttle body, the engine can receive a larger volume of air, facilitating a greater fuel burn and generating more power.
- **Exhaust**: Enhancing the exhaust system can help increase engine power. A well-designed exhaust system minimizes backpressure, allowing the exhaust gases to flow more freely. This enables the engine to expel exhaust gases more efficiently, improving the scavenging effect and reducing power loss. Additionally, performance-oriented exhaust systems may incorporate headers, high-flow catalytic converters, and less restrictive mufflers to further enhance power output.
- **High Compression**: Increasing the compression ratio can result in more engine power. A higher compression ratio allows for a more efficient combustion process, as it squeezes the air-fuel mixture to a greater extent. This leads to better utilization of fuel and increased power output. However, higher compression ratios may require the use of higher-octane fuel to prevent engine knocking.
- **Variable Valve Timing**: Implementing variable valve timing technology can improve engine power. By adjusting the timing of the intake and/or exhaust valves, the engine can optimize the air-fuel mixture's intake and exhaust phases. This maximizes volumetric efficiency, torque, and power output across different engine speeds and loads.
- **Direct Injection**: Direct injection can contribute to increased engine power. By injecting fuel directly into the combustion chamber at high pressure, a finer and more precisely controlled fuel spray pattern can be achieved. This allows for better mixing of fuel and air, leading to improved combustion efficiency and power output.

In conclusion, the decision has been made to develop **forced induction**, which has shown to be the most advantageous option in terms of both performance and feasibility. This choice is supported by the sponsorship campaigns of Formula SAE teams by numerous leading companies in the turbocharging industry.

2.4 COMPRESSOR CHOICE

Once the decision was made to work on a turbocharged single-cylinder engine, we began evaluating potential companies that could assist us in our endeavor.

Honeywell - Garrett emerged as a global leader in the production of turbochargers for both industrial and automotive applications. Moreover, they had been sponsoring numerous Formula SAE teams for years, providing components and materials for the study of these engines.

At the time of contact, Scuderia Tor Vergata was offered two types of turbochargers by the company: the GT06 model and the MGT1238Z model.

Figure 7: Turbine Honeywell-Garret GT06

Figure 8: Turbine Honeywell-Garret MG1238

2.4.1 Calculation of sonic blocking flow rate

Given that a converging-diverging duct must be placed upstream of the compressor, the first step was to calculate the maximum air flow corresponding to the sonic block flow at the 19 mm restriction.

To deal with air, which is a fluid, the conservation of mass flow was considered. This principle states that the mass flow is equal to the product of density, fluid velocity, and the cross-sectional area of the duct.

By applying the conservation of mass flow equation,

$$\boldsymbol{m} = \boldsymbol{\rho} \times \boldsymbol{c} \times \boldsymbol{A} \quad [1]$$

it becomes evident that, for a constant density and a given area, the flow can be increased indefinitely by increasing the velocity.

However, in real fluids, density changes with increasing velocity due to compressibility effects. Therefore, this density variation needs to be considered to determine the mass flow at high fluid velocities.

Using equations related to Mach numbers and speed of sound:

$$\mathbf{c} = \boldsymbol{Ma} \times \boldsymbol{cs} = \boldsymbol{Ma} \times \sqrt{\boldsymbol{\gamma} \times \boldsymbol{R} \times \boldsymbol{T}} \quad [2]$$

and the equation of state for the fluid:

$$\boldsymbol{p} = \boldsymbol{pt} \times \frac{\boldsymbol{T}}{\boldsymbol{Tt}}^{\frac{\gamma}{\gamma-1}} \ [3], \text{ where } \frac{\boldsymbol{T}}{\boldsymbol{Tt}} = \left(\mathbf{1} + \frac{\boldsymbol{\gamma}-\mathbf{1}}{\mathbf{2}} \times \boldsymbol{Ma}^{\mathbf{2}}\right)^{-\mathbf{1}} \quad [4]$$

the relationship between [1] and [3]:

$$\frac{\boldsymbol{p}}{\boldsymbol{\rho}} = \boldsymbol{R} \times \boldsymbol{T} \quad [5].$$

Substituting [4] into [3], [3] into [5], and [5] and [2] into [1], we have developed an equation for mass flow in compressible fluids:

$$\boldsymbol{m} = \frac{\boldsymbol{A} \times \boldsymbol{pt}}{\sqrt{\boldsymbol{Tt}}} \times \sqrt{\frac{\boldsymbol{\gamma}}{\boldsymbol{R}}} \times \boldsymbol{Ma} \times \left(\mathbf{1} + \frac{(\boldsymbol{\gamma}-\mathbf{1})}{\mathbf{2}} \times \boldsymbol{Ma}^{\mathbf{2}}\right)^{-\frac{\boldsymbol{\gamma}+\mathbf{1}}{\mathbf{2}\times(\boldsymbol{\gamma}-\mathbf{1})}} \quad [6]$$

By substituting the values into this equation: Ambient pressure **pt=101325Pa**; temperature **Tt=25º**; gas constant **R=0,286kJ/(kg*K)**; duct area **A=0,001256m2** *(calculated with diameter 19mm)*; specific heat ratio $\boldsymbol{\gamma}$ **=1,4**; Match number **Ma=1** *(sonic choke condition)***:** the calculated **maximum mass flow m** for the given conditions is **0.0703 kg/s**.

NOTE: This deduction is based on certain assumptions and simplifications related to airflow in a duct. The accuracy and applicability of these calculations depend on the specific context and conditions of the engine or system being considered, but in our case, this approximation can be used to calculate the operating point.

2.4.2 Calculation of corrected flow rate

To start with the calculation, an initial determination was made regarding the engine's rotational speed that corresponds to the sonic block flow. It was chosen to assume that this specific flow rate aligns with an engine speed of 8000 rpm. Considering the usage of ethanol E-85 as the fuel, with a specific fuel consumption (**BSFC**) of **0.75 lb/(hp*hr)** and an Air/Fuel Ratio (**AFR**) of **8:1**, the maximum achievable power can be accurately computed:

$$HP = \frac{Wa \times 60}{AFR \times BSFC}$$

With a sonic blocking flow rate of (**Wa=9,2991lb/min kg/s)**, the maximum power that it could be got is **HP=92hp** at **8000rpm.**

Therefore, the next step involved calculating the necessary intake pressure to generate the desired power output.

$$MAP_{req} = \frac{Wa \times R \times (460 + T_m)}{VE \times Vd \times \frac{N}{2}}$$

By substituting the values into this equation: sonic blocking flowrate **Wa = 9,2991lb/min**; intake temperature **Tm=130ºF** *(with intercooler)*; gas constant **R=639.6 in/ºF**; volumetric efficiency. for our engine, which has a 4-valve cylinder head, we assumed a value of **VE=0.95**; engine's rotational speed **N=8000rpm**; engine displacement **Vd=42.11cubic inches:**

The calculated **Manifold Pressure Required** MAP_{req} for the given conditions was determined to be 22,771 psi, which means **1,571bar.**

In order to calculate the pressure ratio Π_c we must calculate the inlet (P_{1C}) and outlet (P_{2C}) pressures of the compressor. The following assumptions have been made:

- A pressure drops from the compressor outlet to the intake duct (including the throttle body) of $\Delta P_{loss} = 2psi$ *(0.14 bar).*
- Ambient pressure is $P_{amb} = 14.7\ psi$ *(1 bar).*
- A pressure loss due to the intake duct of $\Delta P_{intake} = 0.5\ psi$
- A 15% loss due to the 19 mm restrictor, so $\eta_r = 0.85$

$$P_{2C} = MAP_{req} + \Delta P_{loss} = 23.836psi$$

$$P_{1C} = (P_{amb} - \Delta P_{intake}) \times \eta_r = 12,07\ psi$$

Therefore, the compression ratio is calculated as:

$$\Pi_c = \frac{P_{2C}}{P_{1C}} = 1.9748$$

Furthermore, the flow rate $(W_a)_l$ has been calculated for N **= 6500 rpm**, which corresponds to the speed at which we assume to achieve maximum torque.

$$(W_a)_l = \frac{MAP_{req} \times VE \times Vd \times \frac{N}{2}}{R \times (460 + T_m)} = 7,52\,\frac{lb}{min} = 0,05685\,\frac{kg}{s}$$

2.4.3 Calculation of the operating point

The presence of regulatory restrictions causes an amplified workload for the compressor compared to its intended operating conditions. To account for this effect, the company Honeywell has provided the following suggested relationships. These equations will shift the compressor's position on the performance map without altering the physical flow within the engine. The proposed formulas are as follows:

- For Honeywell-Garret GT06:

$$Corrected\ flowrate = W_a \times \sqrt{\frac{T_m}{298}} \Big/ \left(\frac{P_{amb}}{750}\right)$$

- For Honeywell-Garret MGT1238Z:

$$Corrected\ flowrate = W_a \times \sqrt{\frac{T_m}{540}} \Big/ \left(\frac{P_{amb}}{28.4}\right)$$

By substituting the values into this equation: sonic blocking flowrate **Wa= 9,2991lb/min**; intake temperature in Ranking degrees **Tm=589.67º**R; ambient pressure **Pamb=14.7:**

The calculated **Corrected flowrate** for the given conditions was determined to be of

- 9.90 lb/min for **GT06**

- 9.05 lb/min (0.068 kg/s) for **MGT1238Z**.

However, it has been assumed that the change in operating point positions on the compressor maps does not affect any other calculations made.

2.4.4 Final choice

Based on the calculations performed, the operating points have been plotted on each of the compressor maps provided by Honeywell, considering the values of corrected flow rate and compression ratio.

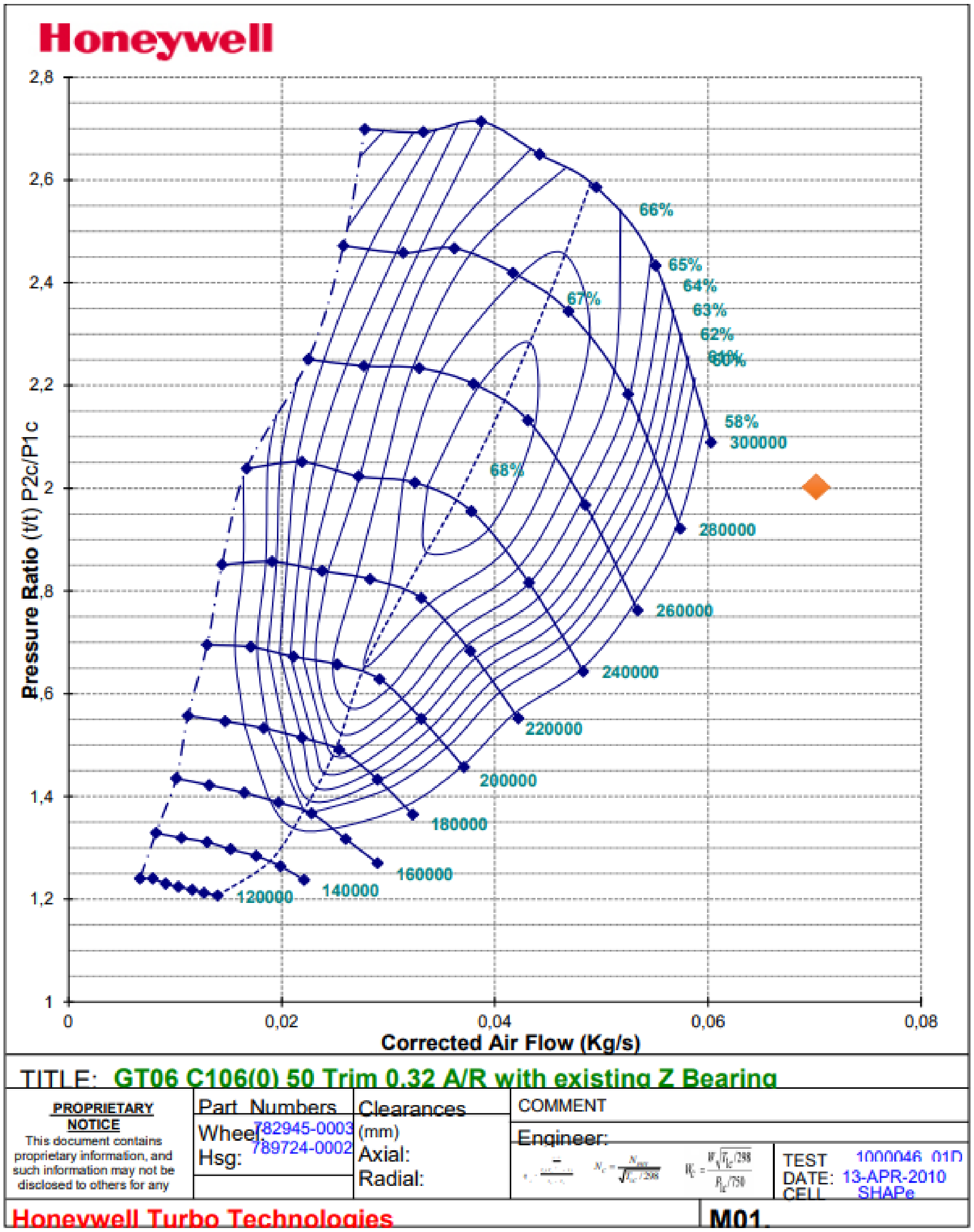

Figure 9: Operating point on GT06 engine map

The engine's operating point falls outside the compressor map, indicating that the compressor is too small for the required flow rate. Using it would lead to premature compressor failure and limit engine performance, so it has been discarded.

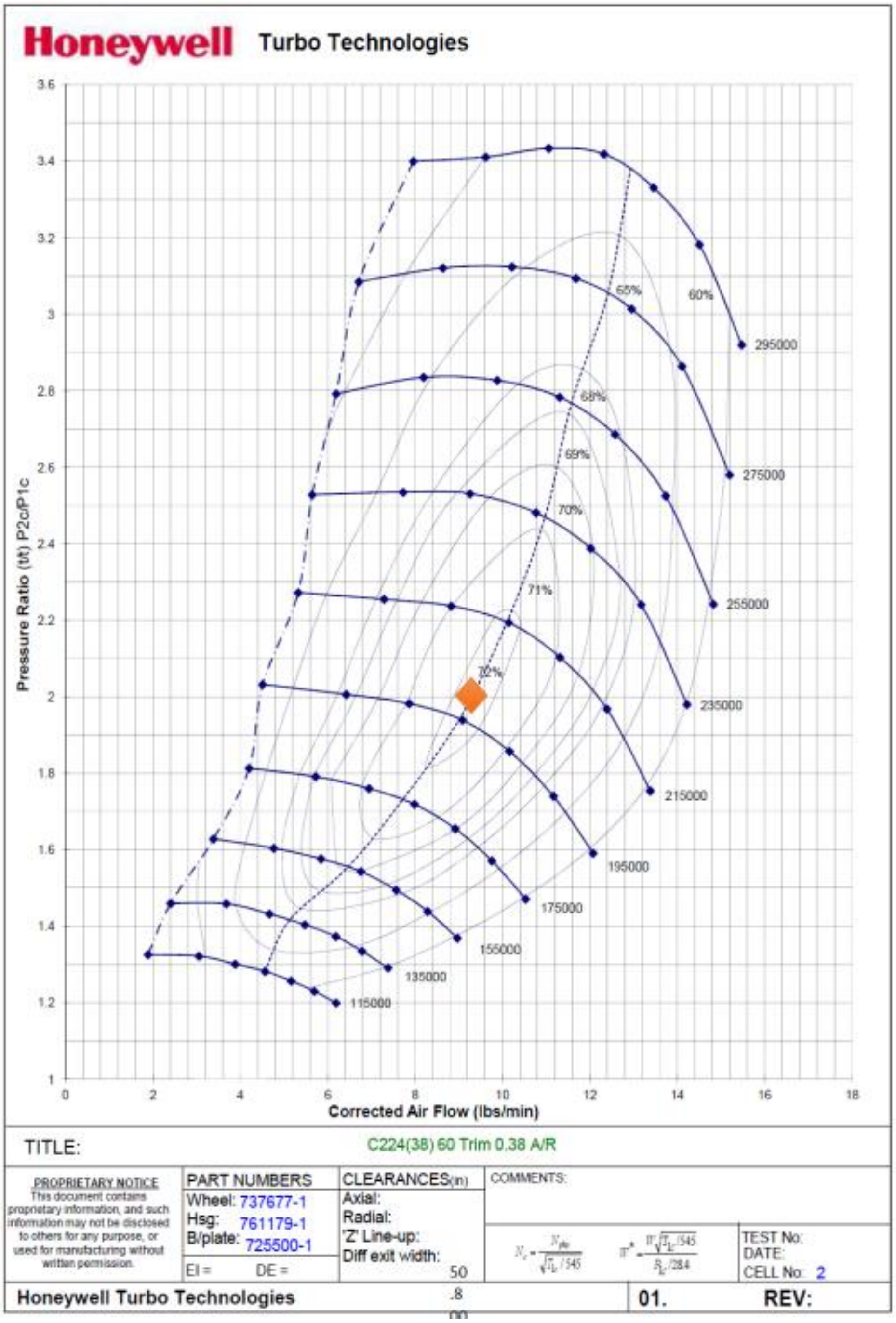

Figure 10: Operating point on MGT1238Z engine map

When representing the same point on the MGT1238Z compressor map, it becomes evident how well it aligns with the selected engine. In fact, the point falls within the region of maximum efficiency, indicating an optimal match between the compressor's capacity and the engine's performance requirements. This alignment is crucial for ensuring efficient and optimal system operation, maximizing both engine performance and fuel consumption. Therefore, it confirms that the **MGT1238Z** compressor is an optimal **choice** for our application.

Figure 11: Turbine Honeywell-Garret MG1238

CHAPTER 3: DESIGN DEVELOPMENT

Following the completion of the design choice phase, an in-depth examination of the selected components was carried out to achieve a thorough comprehension of the engine's operational dynamics. This involved employing Simulink, a software tool developed by MathWorks as part of the MATLAB suite.

3.1 GOALS OF THE PROJECT

The main goal of this project is to enhance the performance of the single-cylinder engine through the implementation of a turbocharging system. Firstly, the objective was to design and install a turbocharger to achieve a significant increase in power output compared to the previous engine configuration. This involved careful selection and integration of a suitable turbocharger unit that could effectively compress the incoming air to maximize combustion efficiency (as we have seen in chapter 2).

Additionally, the project aimed to optimize the engine's intake and exhaust system to accommodate the turbocharger. Also, will be necessary to analyze and select internal engine components capable of withstanding the increased power output generated by the turbocharging system.

3.2 METHODOLOGY

To achieve these objectives, we conducted a study of the engine model. For this investigation we use a Simulink tool, which will be explained in the following sections.

3.2.1 SIMULINK-MATLAB

Simulink, a powerful software tool developed by MathWorks, is worldwide recognized for its capabilities in modeling, simulating, and analyzing dynamic systems. It provides a graphical interface that allows engineers to design and simulate complex systems, such as the turbocharged engine, in a visual and intuitive manner.

In the context of my thesis, Simulink can create detailed models of the engine, including the turbocharger, intake and exhaust systems, combustion process, and other relevant components. Once the model is set up, Simulink allows to simulate the engine's behavior under different operating conditions. This includes analyzing parameters such as air-fuel ratio, power output, and other performance metrics. Simulink's simulation capabilities enable to observe how changes in various system parameters affect the overall engine performance and efficiency.

Simulink also facilitates the integration of control systems into the engine model. By simulating this integrated control system, we can evaluate its effectiveness in maintaining stable and efficient operation.

Overall, Simulink provides a powerful and versatile platform for modeling, simulating, and analyzing our turbocompressor system.

3.3 IMPLEMENTATION OF THE MODEL

In this subsection, our aim is to integrate the original engine model with the Simulink model. This integration allows us to simulate the engine's behavior under various operating conditions and evaluate its performance in a controlled virtual environment. After the integration, we will be able to try to couple the turbocompressor.

3.3.1 SI ENGINE DYNAMOMETER

To develop our model, we will use a function in Simulink called SI Engine Dynamometer Reference Application, which can be used to study the behavior of the engine.

The spark-ignition (SI) engine dynamometer reference application is a powerful tool for calibrating, validating, and optimizing the engine controller and plant model parameters. By utilizing this application, we will configure the engine controller settings, validate the model against real-world data, and optimize parameters for enhanced performance.

This comprehensive analysis ensures accurate representation of the engine's behavior before integrating it into a larger vehicle model. The integration of the calibrated and optimized engine model enables a thorough examination of the vehicle's overall performance, allowing for informed design decisions and improvements in drivability, fuel economy, and emissions.

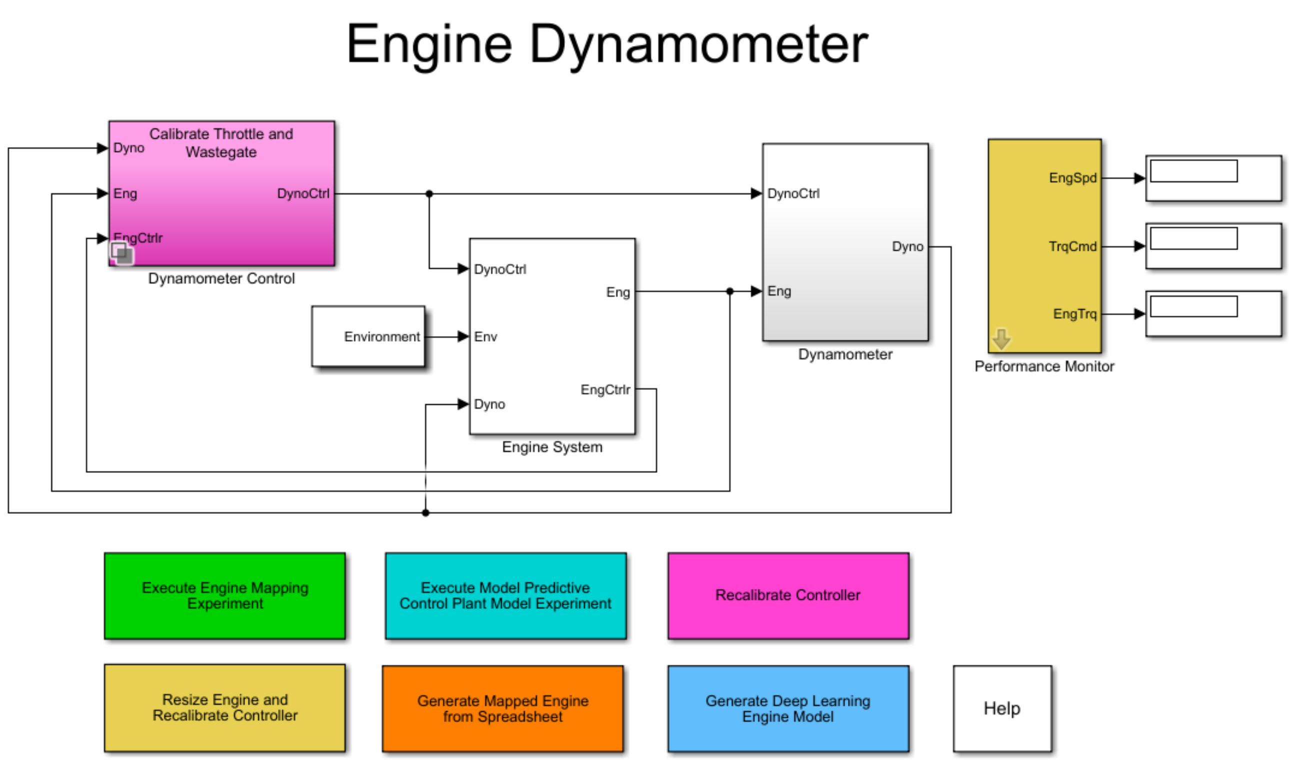

Figure 12: SI Engine Dynamometer Reference Application

Within the Engine System, when configured without the turbocompressor, the following model can be observed:

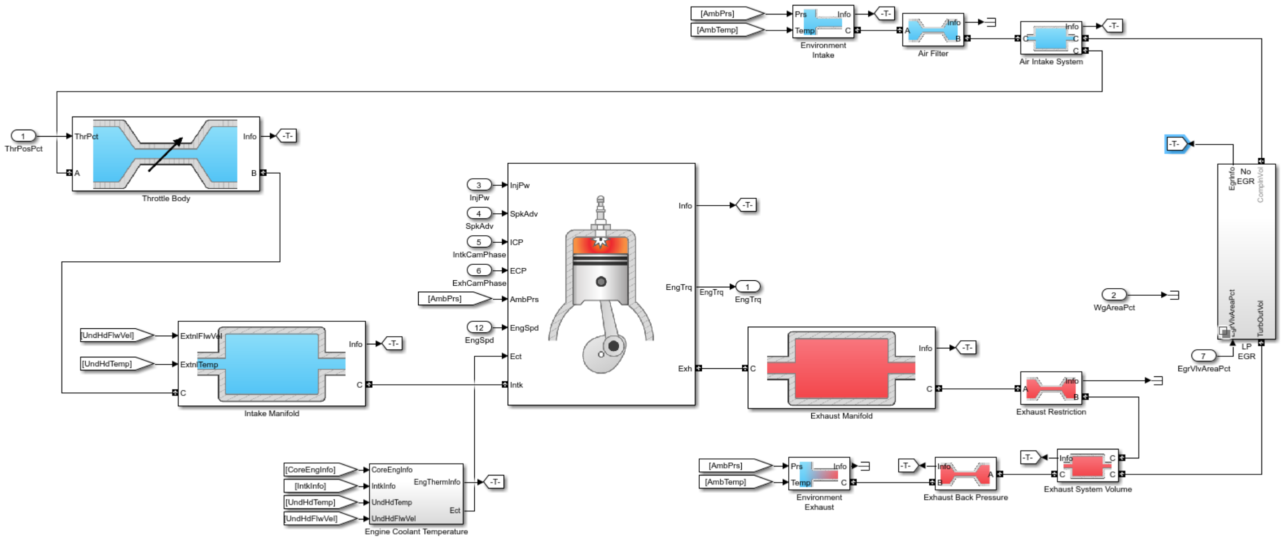

Figure 13: Engine system without Turbocompressor

3.3.2 SI CORE ENGINE: CALIBRATION

The most important component in this model is the ***SI Core Engine.*** The ***SI Core Engine*** block implements a spark-ignition (SI) engine from intake to exhaust port. You can use the block in larger vehicle models, hardware-in-the-loop (HIL) engine control design, or vehicle-level fuel economy and performance simulations. The ***SI Core Engine*** block calculates:

- Brake torque
- Fuel flow
- Port gas mass flow, including exhaust gas recirculation (EGR)
- Air-fuel ratio (AFR)
- Exhaust temperature and exhaust mass flow rate
- Engine-out (EO) exhaust emissions

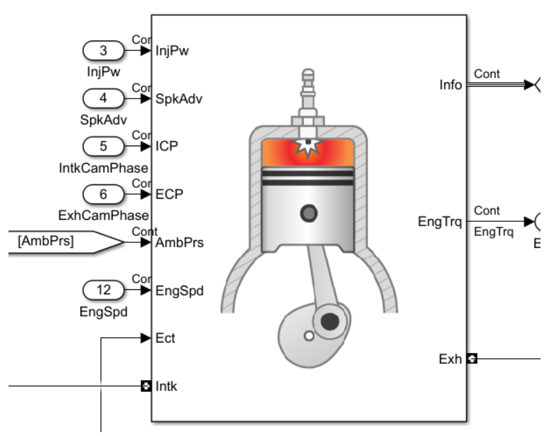

Figure 14: SI Core Engine

In order to fit the duke KTM 690 into the SI Core Engine Model, we must re-calibrate the engine to the duke's specifications:

DUKE KTM 690 M14	
Displacement volume	690 cc
Engine Type	Internal Combustion
Number of cycles	4
Number of cylinders	1
Bore	102mm
Stroke	84.5mm
Number of intake valves	2
Number of exhaust valves	2
Maximum Power	52kw (70cv) at 7500rpm
Maximum Torque	70Nm at 5500 rpm

Table 6: Properties of Duke KTM 690 M14

For recalibrating the engine, we are using the function ***Resize engine and re-calibrate controller***, whit this function, we can resize engine plant (and engine core) and re-calibrate engine controller based on desired maximum power level, displacement and number of cylinders.

We have measured and plotted the performance characteristics of the engine after the recalibration. These curves represent how power output and torque, or fuel consumption, change with respect to engine speed:

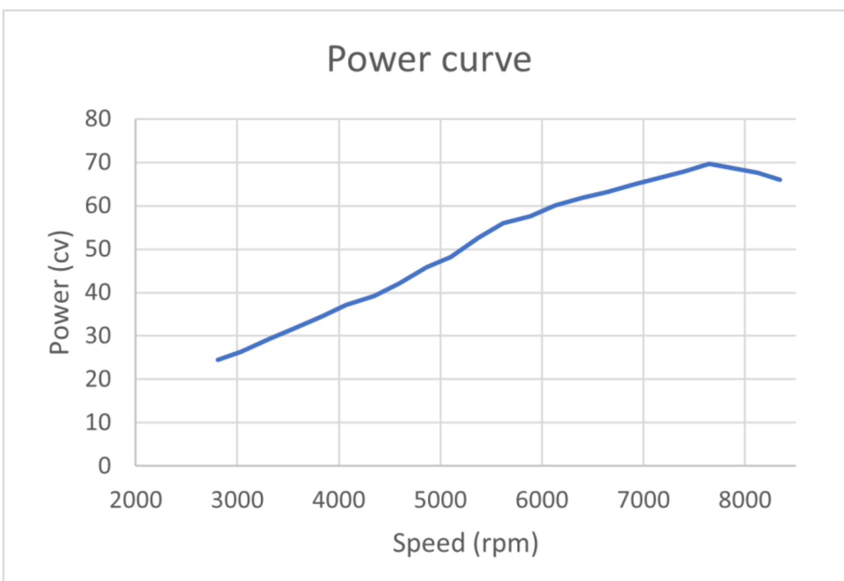

Figure 15: Power curve after recalibrating

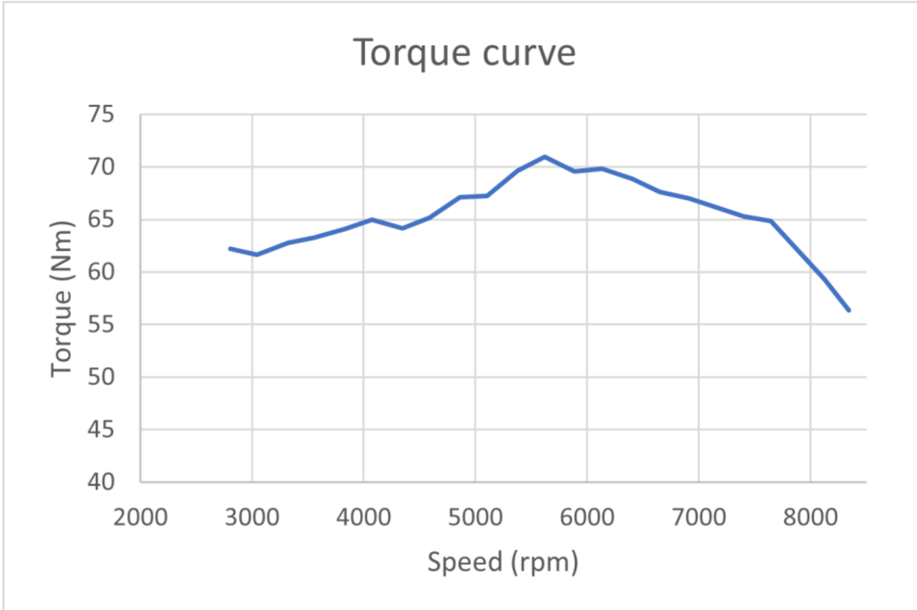

Figure 16: Torque curve after recalibrating

3.3.3 SI CORE ENGINE: COMPARISON

When comparing the curves of a motor calculated with Simulink to the curves provided by the manufacturer, it is possible to observe slight differences between them. While these curves may not be exactly the same, they often exhibit a high degree of similarity. There are several reasons why these differences can occur:

- Assumptions and Simplifications: When creating a simulation model using Simulink, certain assumptions and simplifications are made to represent the motor's behavior.

- Modeling Accuracy: Although these models aim to capture the motor's characteristics accurately, there may still be small difference due to limitations in the modeling process.

- Measurement Variability: The manufacturer's curves are typically obtained through numerous tests on physical prototypes that involve a variety of measurement techniques and instruments, which can introduce some level of variability.

- Formula SAE restriction: In our model, we have incorporated certain restrictions imposed by the Formula SAE rules, with the primary focus being on the inclusion of an air restrictor. According to the regulations, this air restrictor must have a fixed area of 19mm2. This restriction can have a slight impact on the power output, especially at lower speeds.

However, despite these differences, it is important to note that if the curves obtained through Simulink closely resemble the manufacturer's curves, it indicates that the simulation model is providing a good approximation of the motor's behavior. This similarity can be appreciated in the following graphs:

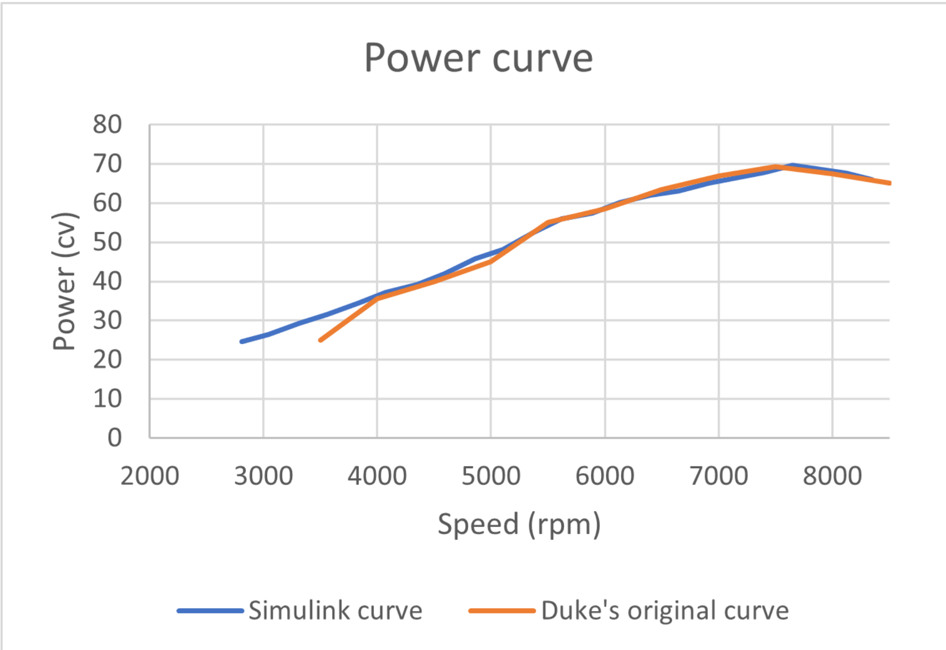

Figure 17: Power comparison Simulink vs Duke’s

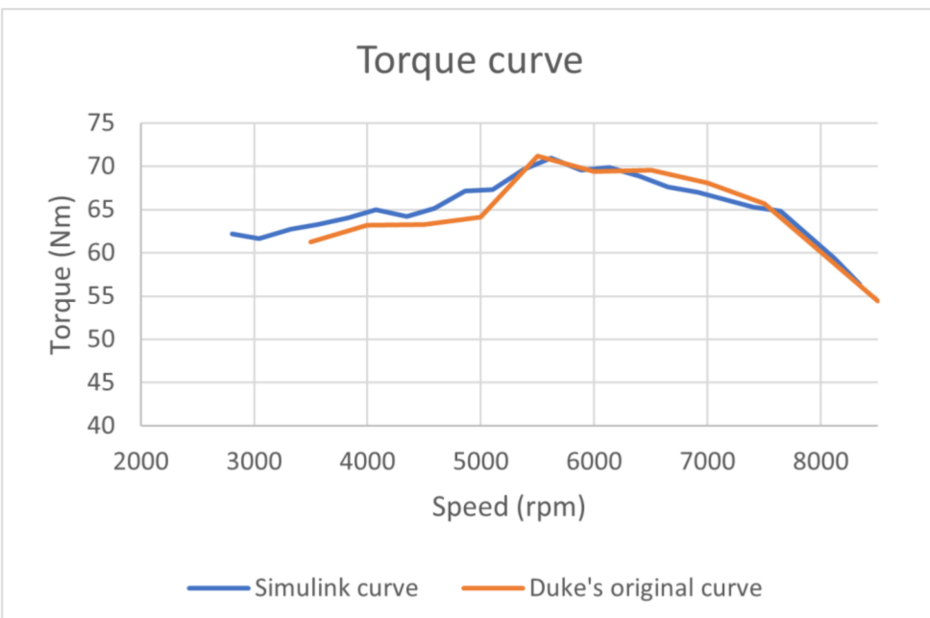

Figure 18: Power comparison Simulink vs Duke’s

3.3.4 TURBOCHARGED MODEL

Once we have confirmed the correctness of the engine model, the next step is to configure the turbocharger. This configuration aims to analyze the potential performance improvements that can be achieved by integrating a turbocharger into the system.

We can incorporate these blocks (turbine, compressor…) separately or we can use a tool of this model that allows us to add all the Turbocompressor system.

After incorporating the turbocharger, the model assumes the following configuration:

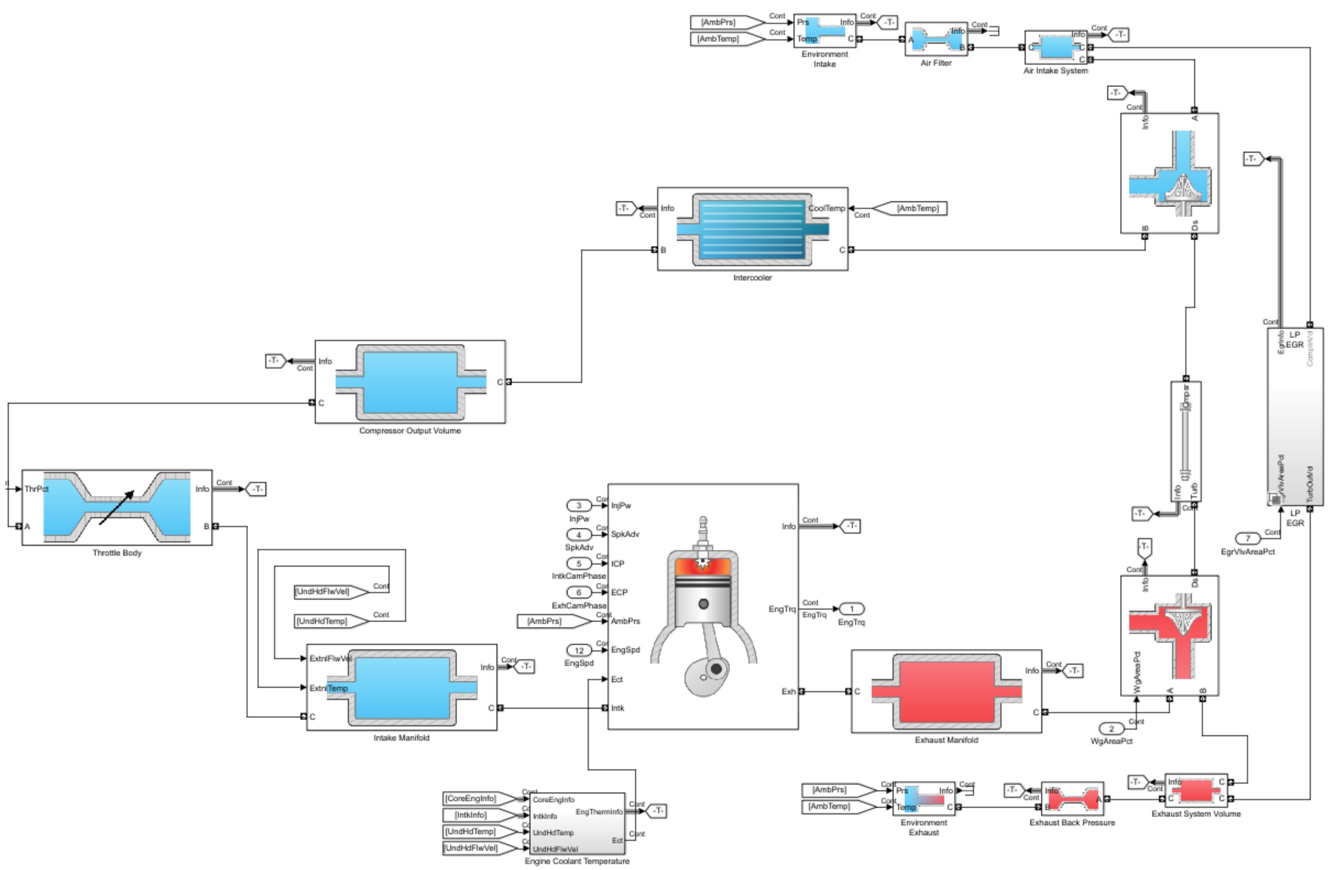

Figure 19: Engine system without Turbocompressor

In order to incorporate the Honeywell-Garret Turbocompressor into the model, a detailed configuration process is required for both the turbine and compressor components. We must do this configuration separately as we will see in the following sections.

3.3.5 TURBINE

The *Turbine block* employs the principles of mass and energy conservation to determine the flow rates of mass and heat in turbines. Can have fixed or variable geometry.

By using a wastegate valve, the block can be configured to redirect the gas flow away from the turbine. Two-way ports are used to establish connections with the inlet and outlet control volumes, as well as the drive shaft. To calculate the mass flow rate and turbine efficiency, you can specify lookup tables. It's important to note that the block does not support reverse mass flow.

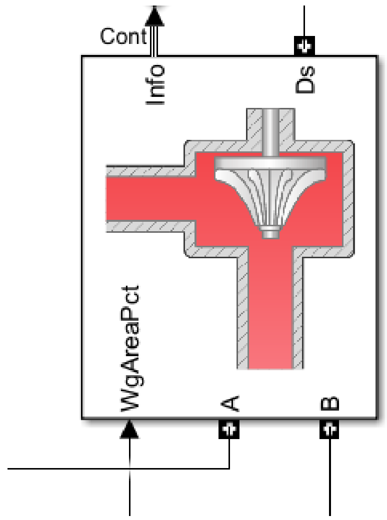

Figure 20: Turbine Block

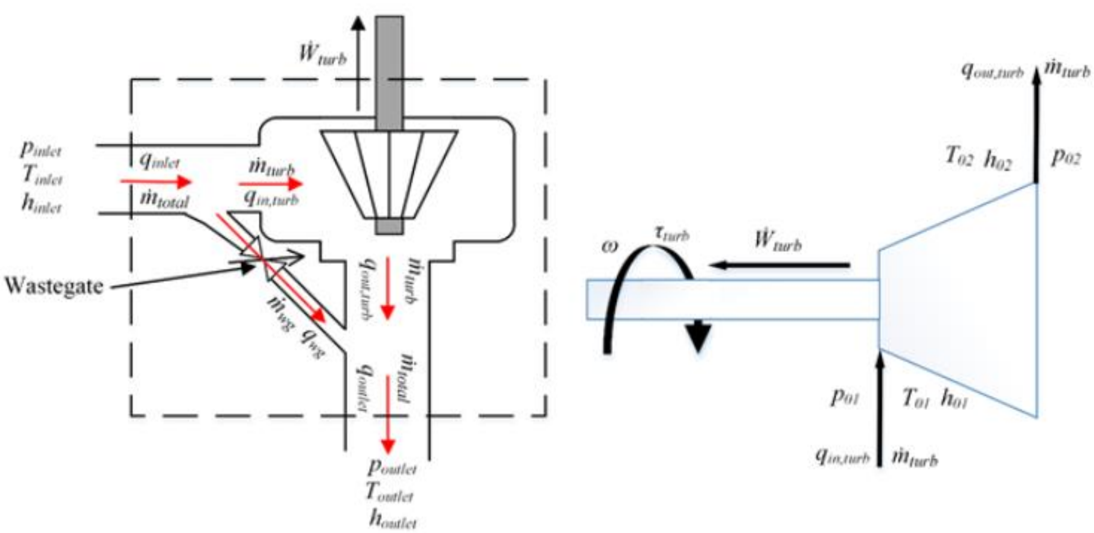

Figure 21: Turbine scheme

We have calibrated the performance maps to virtually calibrate the corrected mass flow rate and turbine efficiency lookup tables using measured data. This data comes from the turbine's map:

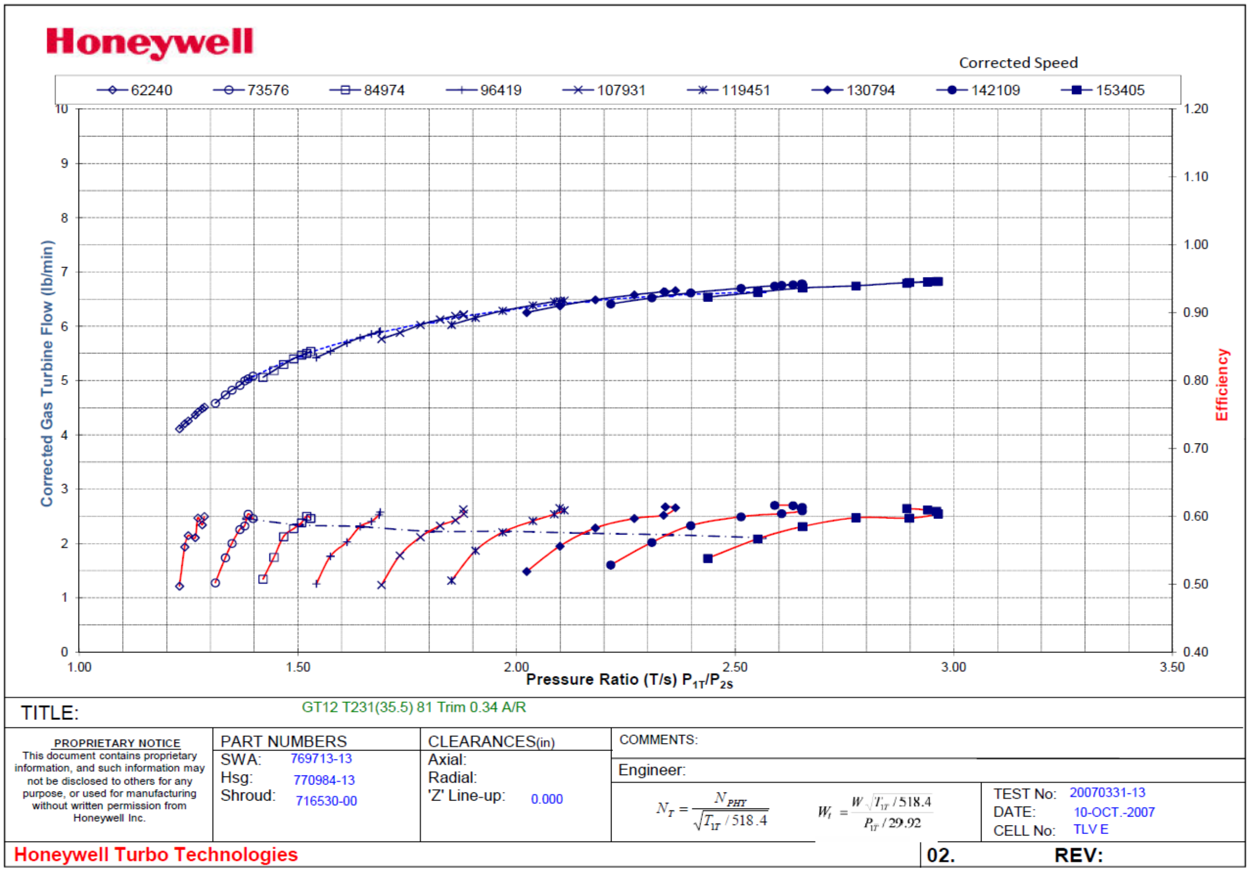

Figure 22: Turbine GT1238 map

To utilize the "Calibrate Performance Maps" tool, we need to extract specific data from the Honeywell-Garret map, including:

- Mass flow rate
- Speed
- Efficiency
- Pressure ratio

We need to derive two matrices: one for the mass flow rate and another for the efficiency. These matrices are constructed as a function of both the pressure ratio and speed. By obtaining this data, we gain valuable insights into how the turbine operates under different conditions, allowing us to accurately model its behavior.

Once we have obtained these matrices, we can use the "Calibrate Performance Maps" tool, which facilitates the integration of our turbine into the Simulink model.

3.3.4 COMPRESSOR

The *Compressor block* serves to simulate the boosting effect in an engine by utilizing the energy from the drive shaft to raise the pressure in the intake manifold. This block is an integral component of supercharger and turbocharger models. It employs two-way ports to establish connections with the inlet and outlet control volumes, as well as the drive shaft. These control volumes provide the necessary parameters such as pressure, temperature, and specific enthalpy, which are used by the compressor to calculate the flow rates of mass and energy.

In order to determine the torque and flow rates, the drive shaft supplies the compressor with the rotational speed. Compressor manufacturers typically furnish mass flow rate and efficiency tables that are dependent on corrected speed and pressure ratio.

To compute the mass flow rate and efficiency, you have the flexibility to specify the lookup tables. It's important to note that the block does not support the occurrence of reverse mass flow.

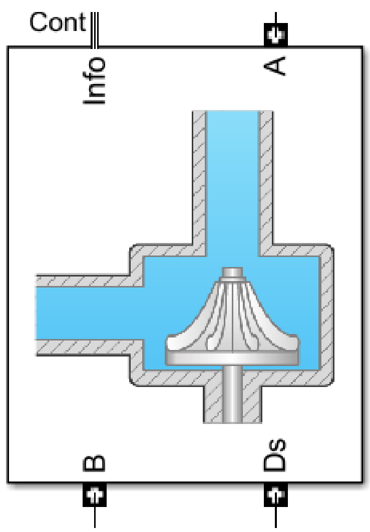

Figure 23: Compressor Block

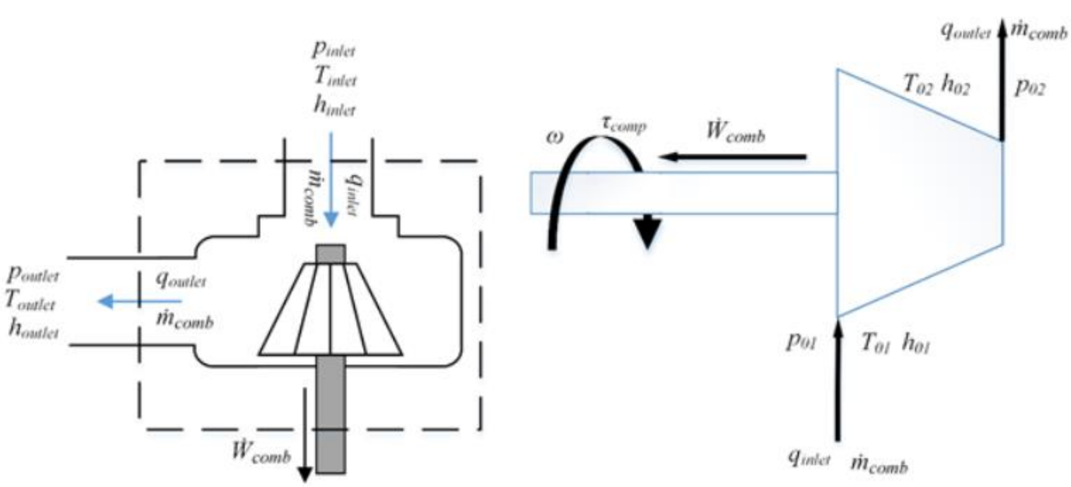

Figure 24: Turbine scheme

We have calibrated the performance maps using the same tool as for the turbine. To use it, we must know compressor’s map data:

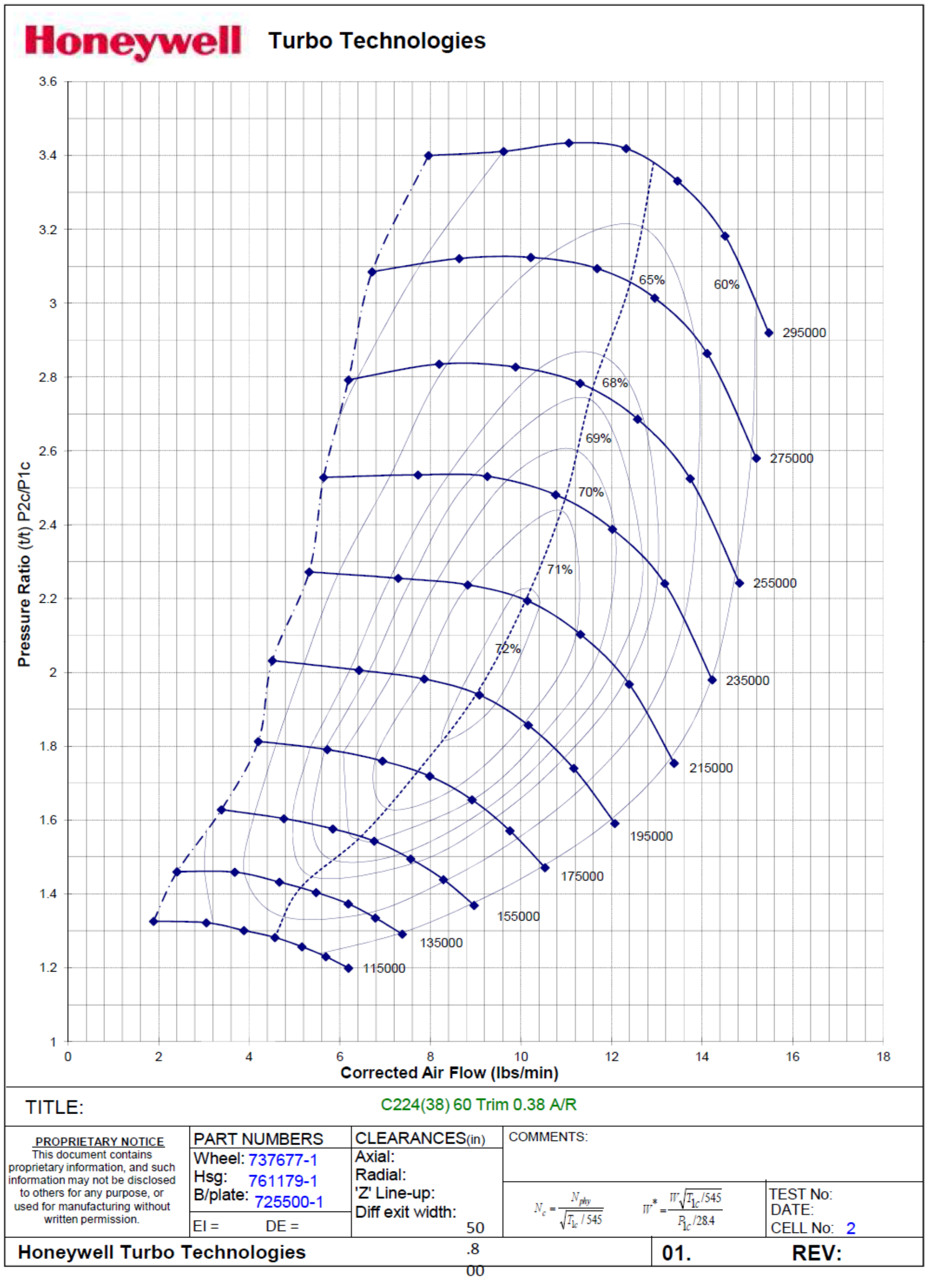

Figure 25: Compressor GT1238 map

We will do exactly the same as with the turbine. We need to derive those two matrices: one for the mass flow rate and another for the efficiency, constructed as a function of both the pressure ratio and speed.

Once we have obtained these matrices, we can use the "Calibrate Performance Maps" tool, which facilitates the integration of our compressor into the Simulink model.

CHAPTER 4: RESULTS

In this chapter, we will analyze and compare the results obtained from configuring the engine system with and without the Turbocompressor in Simulink.

Our focus will be on evaluating the performance enhancements achieved through the integration of the Turbocompressor. We will assess key parameters such as power output and torque to understand the impact of the Turbocompressor on overall engine performance.

Additionally, we will examine the efficiency of the Turbocompressor system by analyzing factors such as volumetric efficiency.

This analysis will provide valuable insights for optimizing the Turbocompressor system and futures improvements of the engine performance.

4.1 TURBOCHARGED MODEL RESULTS

In this section, we will analyze and discuss the power and torque curves of the engine with the Turbocharger. These curves are essential tools for understanding the performance of an engine with forced induction.

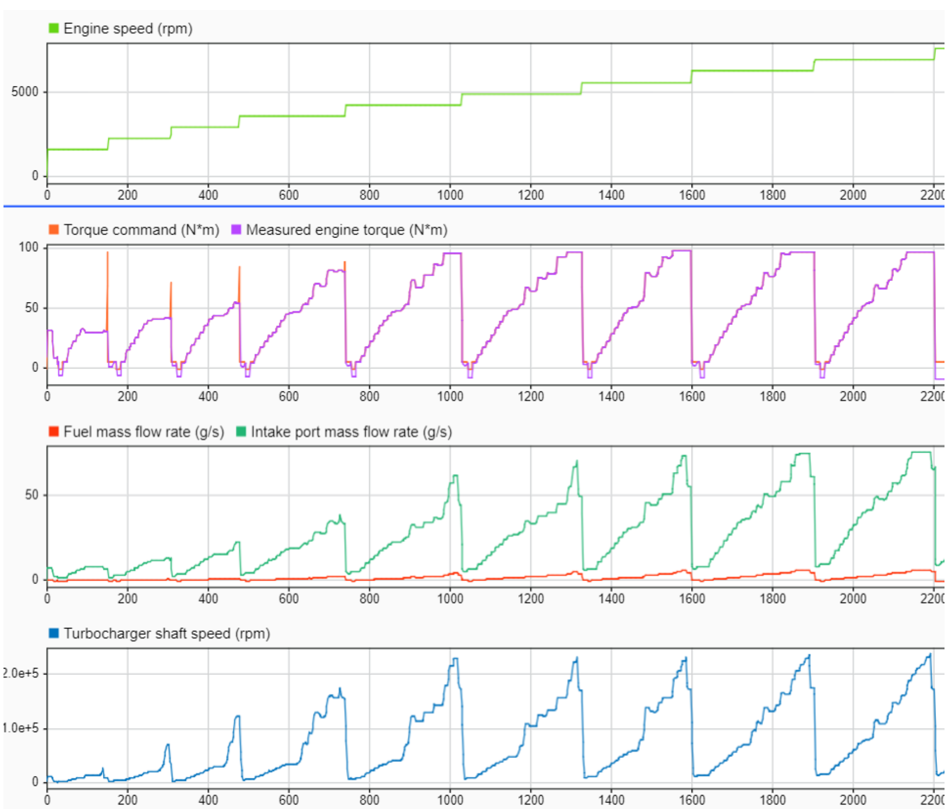

Figure 26: Results of the turbocharged engine

The power curve explains the amount of energy generated by the engine as engine speed increases. This curve is crucial for evaluating acceleration capability and power delivery across different speed ranges. We can identify the engine's peak power point of approximately 93.8cv at 8100rpm, indicating the optimal speed for achieving the best performance.

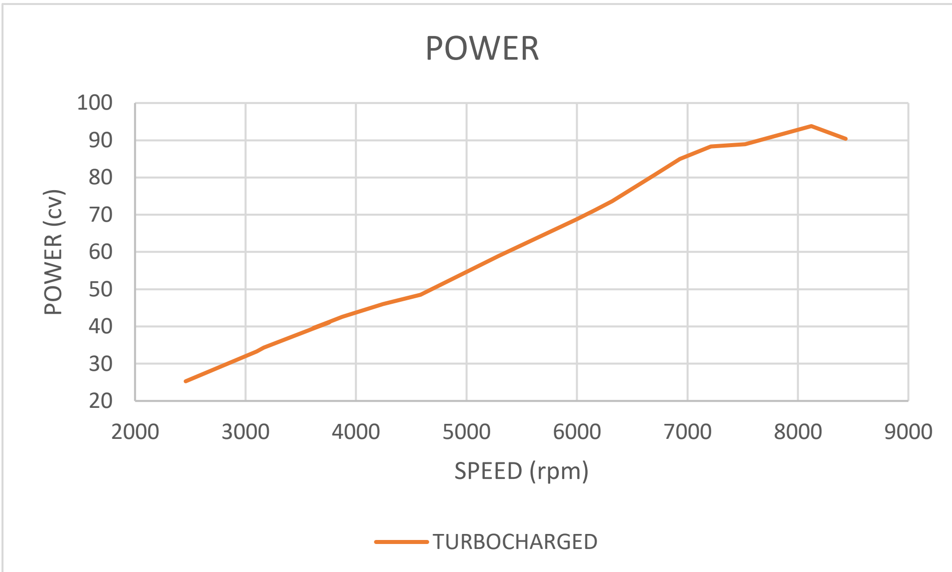

Figure 27: Power output of the turbocharged engine

On the other hand, the torque curve illustrates the engine's rotational force or pulling capability in relation to engine speed. This curve is particularly important for understanding the engine's responsiveness to varying load demands. It allows us to identify the speed range where the engine delivers maximum torque, indicating its ability provide a smooth and powerful driving experience. We can find this maximum point of approximately 88Nm at 7000rpm.

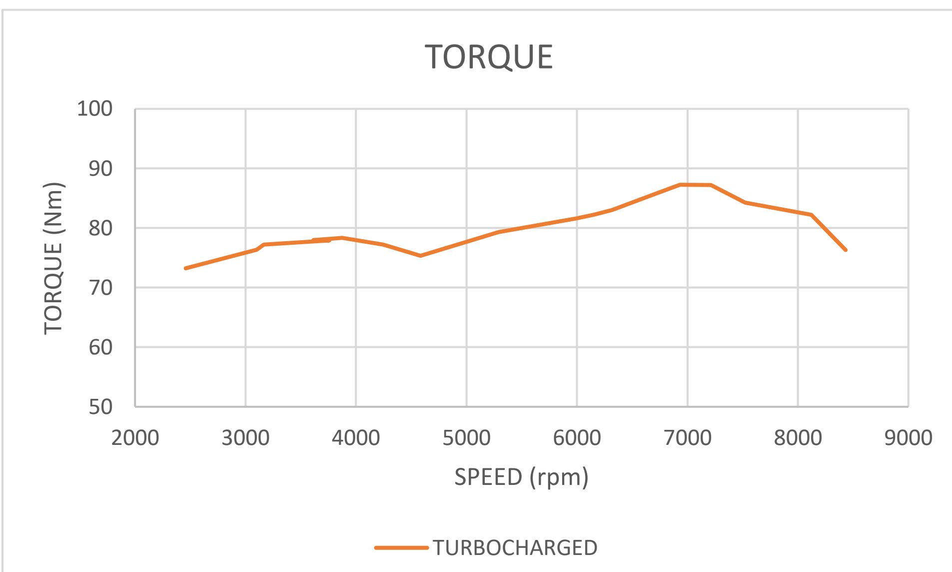

Figure 28: Torque of the turbocharged engine

Note: Remember that the Power output and the Torque are directly related with the following equation:

$$\boldsymbol{P(W)} = \frac{\boldsymbol{T(Nm) \times N \times 2 \times \Pi}}{\boldsymbol{60}}$$

4.2 COMPARITION OF TURBOCHARGED ENGINE AND NATURALLY ASPIRATED ENGINE

After calibrating the turbo engine model, a comparison was made between the performance of the naturally aspirated configuration and the turbocharged configuration. This analysis focused on power output, torque delivery and the filling coefficient. The comparison highlights the significant performance enhancements achieved through turbocharging, demonstrating its positive impact on power, efficiency, and overall engine performance.

4.2.1 POWER OUTPUT

When comparing the power output curves of the naturally aspirated and turbocharged engines, notable differences emerge.

The naturally aspirated engine reaches a maximum of approximately 70cv at 7500rpm, while the turbocharged engine achieves a significantly higher peak power of approx. 92cv at 8100rpm.

This comparison clearly demonstrates the substantial power advantage provided by turbocharging. With its ability to deliver greater power output at higher engine speeds, the turbocharged engine exhibits enhanced performance and potential for improved acceleration and overall driving experience.

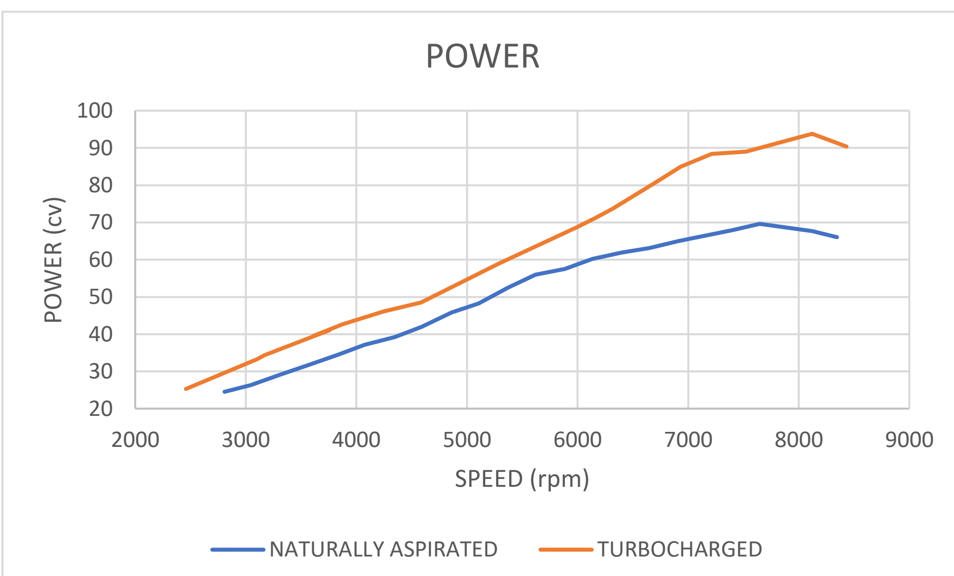

Figure 29: Power comparison between naturally aspirated engine and turbocharged engine

4.2.2 TORQUE

When comparing the torque curves of the naturally aspirated and turbocharged engines, distinctions can be observed. The naturally aspirated engine reaches its peak torque of approximately 70 Nm at 5500 RPM, while the turbocharged engine exhibits a higher maximum torque of approximately 88 Nm at 7000 RPM.

This contrast highlights the advantage of turbocharging in delivering greater torque output across a wider range of engine speeds.

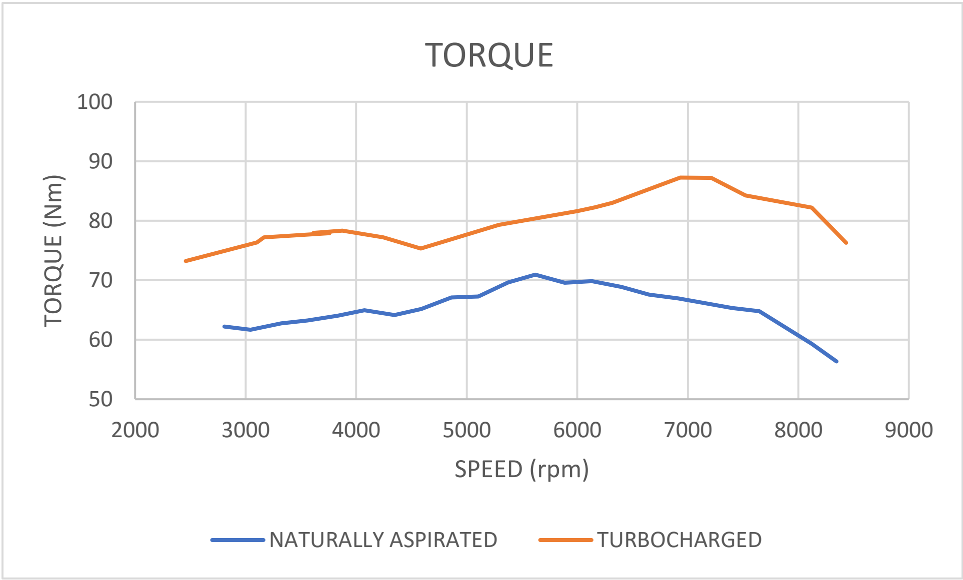

Figure 30: Torque comparison between naturally aspirated engine and turbocharged engine

The relationship between torque and power is essential in understanding an engine's performance characteristics. Torque represents the rotational force generated by the engine, while power reflects the rate at which work is done. Torque provides the initial force to overcome resistance and accelerate the vehicle, while power indicates how quickly the work can be accomplished.

Higher torque values, as seen in the turbocharged engine, contribute to improved acceleration and pulling power. Additionally, the higher peak torque achieved at a higher engine speed suggests that the turbocharged engine can deliver more power throughout its operating range. This translates to enhanced performance, allowing for faster acceleration and better overall driving dynamics.

In summary, comparing the torque curves of the naturally aspirated and turbocharged engines reveals the significant advantage of turbocharging in terms of both maximum torque output and the speed range over which high torque is available. Understanding the relationship between torque and power has been necessary for evaluating the engine's performance capabilities and predicting its behavior in various driving scenarios.

4.2.3 VOLUMETRIC EFFICIENCY

Analyzing the volumetric efficiency is crucial because it provides valuable insights into an engine's ability to intake and exhaust air efficiently. A high volumetric efficiency indicates that the engine is effectively utilizing the available air, which leads to improved combustion and increased power output.

We can measure the volumetric efficiency by calculating the filling coefficient. This coefficient (that is a synonym of volumetric efficiency) is defined as the ratio between the actual mass of air introduced into the cylinder per cycle, represented by ***ma***, and the theoretical mass ***mt.***

In order to calculate the actual mass, we will use the following equation:

$$m_a = \dot{m}_a \times \frac{\varepsilon}{N}$$

On the other hand, to calculate the theoretical mass:

$$m_t = V_d \times \rho_a$$

By calculating this coefficient, the results can be observed at the following graph:

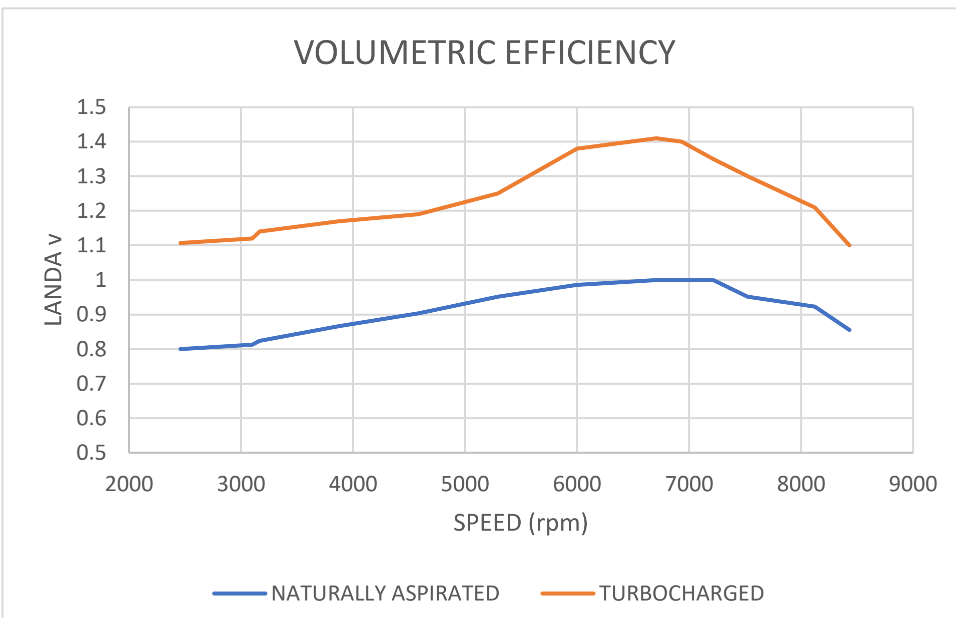

Figure 31: Volumetric efficiency between naturally aspirated engine and turbocharged engine

When comparing the volumetric efficiency curves of the naturally aspirated and turbocharged engines, main differences can be observed.

In the case of the <u>naturally aspirated</u> engine, the volumetric efficiency curve reaches a maximum value of nearly 1 at 7000 RPM. This indicates that the engine is effectively filling its cylinders with air, achieving close to optimal intake efficiency.

On the other hand, the turbocharged engine exhibits a significantly higher volumetric efficiency, with a maximum value of approximately 1.4 at 7000 RPM. This higher value suggests that the turbocharger is successfully compressing the incoming air, allowing for a greater amount of air to enter the cylinders compared to the naturally aspirated engine.

The reason why the naturally aspirated engine cannot achieve a volumetric efficiency greater than 1 is due to its reliance on atmospheric pressure alone.

The air intake process is limited by atmospheric pressure, and the engine can only draw in a certain volume of air per cycle based on the displacement of the cylinders. As a result, the volumetric efficiency reaches a maximum value of around 1, indicating that the engine is effectively utilizing the available atmospheric air.

In contrast, the turbocharged engine benefits from forced induction, where the turbocharger compresses the incoming air before it enters the cylinders. This compression increases the density of the air, allowing a larger mass of air to be introduced into the cylinders.

As a result, the volumetric efficiency can exceed 1, indicating that the turbocharged engine can intake a greater volume of air than its naturally aspirated counterpart.

In summary, this comparison of volumetric efficiency curves reveals the advantage of turbocharging in improving the engine's ability to intake air. The turbocharger's compression mechanism allows the turbocharged engine to achieve higher volumetric efficiency values, indicating improved air intake and enhanced overall performance compared to a naturally aspirated engine.

4.2 SIMULATION WITH CRITICAL AMBIENT CONDITIONS

The simulations conducted so far have taken into consideration the prevailing environmental conditions under which the engine operates:

• The pressure ($pamb$) is maintained at 1 bar.

• The temperature ($Tamb$) is set at 25°C.

As the vehicle will be participating in competitions held in various environments, subject to different temperature and pressure conditions, it was deemed necessary to simulate the engine's performance under the following additional conditions, we will try with these ones:

• The pressure ($pamb$) is reduced to 0.8 bar.

• The temperature ($Tamb$) is increased to 40°C.

After the simulation, we get the following results:

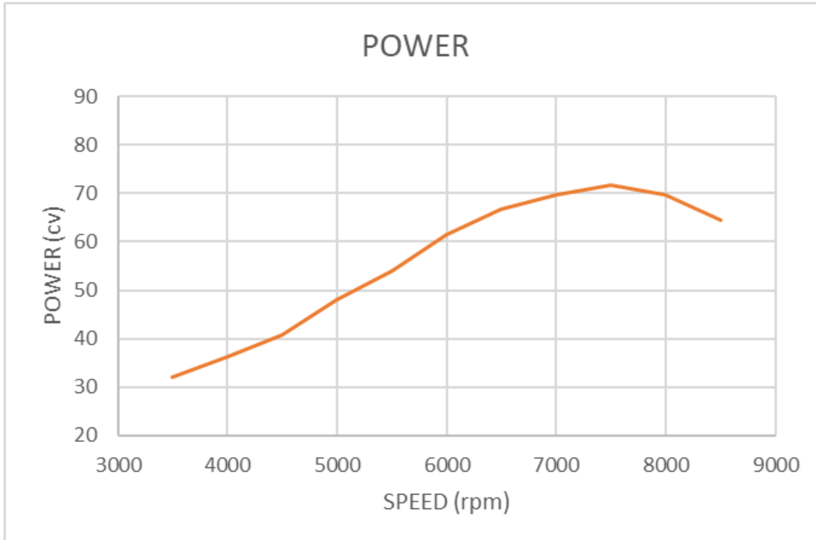

Figure 32: Power of an engine with turbocharger in critical ambient conditions

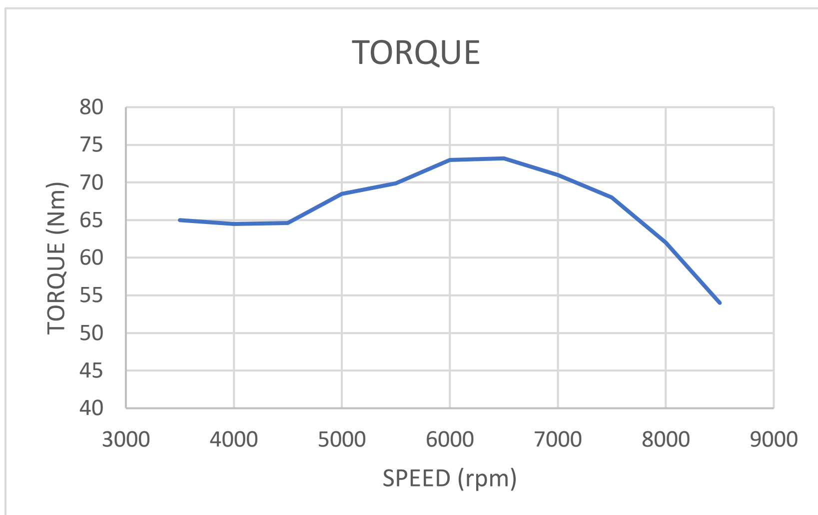

Figure 33: Power of an engine with turbocharger in critical ambient conditions

Under critical conditions of 0.8 bar pressure and 40°C temperature, the torque curve and power output of the turbocharged engine decrease. This can be attributed to several factors.

Firstly, the reduced air density resulting from lower pressure limits the amount of air available for combustion, leading to a decrease in power generation.

Secondly, the higher temperature can cause inefficient combustion, resulting in incomplete combustion and reduced engine efficiency.

Lastly, these conditions impose a heavier workload on the turbocharger, which may struggle to maintain optimal performance. As a result, both the torque and power curves exhibit a downward trend, as we can observe in the graphs, indicating a decrease in overall engine performance under these critical environmental conditions.

CHAPTER 5: CONCLUSION

The main goal of this thesis work was to study the turbocharging and its application to a single-cylinder high-performance engine operating according to the otto cycle that will be used for a Formula SAE car.

In Chapter 1, we have introduced various topics related to the internal combustion engine and the Formula SAE competition. We began by discussing the basics of combustion and engine operation. We then delved into the concept of forced induction, specifically, we explored the principle of turbocharging, a popular method of forced induction.

Additionally, we provided an overview of the Formula SAE competition and its rules. Understanding these fundamental concepts sets the foundation for the subsequent chapters, where we further explore the application of turbocharging and its impact on engine performance.

Chapter 2 illustrated the methodology for component design and selection. This analysis was carried out in conjunction with the requirements not only of the engine department but of the entire Scuderia Tor Vergata, seeking compromises among all the team's needs.

The use of MATLAB-Simulink software has been of fundamental importance in simulating the engine, both in its original configuration and in the supercharged configuration, as we can see in Chapter 3. The study of the engine in its original configuration allowed for the calibration of the system in order to validate the reliability of its results, in the absence of engine bench tests that were not available at the time of the study.

The results presented in Chapter 4 were the product of an extensive process of fine-tuning the model and adjusting turbocharger pressures to ensure a smooth simulation without encountering compressor surges.

The outcomes conclusively demonstrate the significant performance improvements and power enhancements achievable through turbocharging. However, it is important to recognize that despite the evident advantages, there are practical limitations that hinder the immediate practical implementation of this solution.

These findings offer valuable insights into the potential of turbocharging as a viable approach for boosting power, but they also emphasize the need for further exploration and in-depth analysis to overcome the existing challenges and find feasible solutions.

As this represents a Scuderia Tor Vergata's approach to the issue of forced induction, this study serves as a starting point for future investigations. With the aid of three-dimensional computational software, it will be possible to refine the modeling of the phenomena that occur in engine operation, aiming to find solutions to potential challenges that may arise.

This study lays the groundwork for further exploration and advancements in the understanding of forced induction systems, paving the way for continuous improvement in engine performance and efficiency.

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