Calculation and Matching a Turbocharger

By

Bara' Mohammed Joulani

Husam Mohammed Khatib

Supervisor:

Dr. Zuhdi Salhab

Submitted to the College of Engineering

in partial fulfillment of the requirements for the degree of

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Mechanical Engineering Department

Hebron – Palestine

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Chair of the Department Signature

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Abstract

Internal Combustion engines are considered the most famous Source of energy that is used in water and land vehicles and some trains.

And From the early years of discovering this source there was a tough competition between firms, concentrating, basically on developing and increasing number of vehicle and the dilemma of environmental crisis, firms were forced to search for solutions that aim to increase efficiency and at the sometime decrease emissions.one of the idea, and it may be the most important one, is the addition of turbocharger to the internal combustion engines.

In this project, the engine of the Volkswagen Beetle 1300cc was selected through calculations we have reached the right size for the turbocharger including improved engine efficiency the nearest turbocharger on the market, the GT1548, was matched.

تعتبر محركات الاحتراق الداخلي اكثر مصدر طاقة شيوعا للمركبات البرية والمائية وبعض القطارات .

ومنذ اختراعها كان هناك تنافس كبير بين الشركات يتمحور حول تطويرها وزيادة كفاءتها .

ومع زيادة عدد المركبات وتفاقم الازمة البيئية أدى ذلك الى التضييق على الشركات ما

استوجب البحث عن حلول تهدف الى زيادة كفاءة المحركات والتقليل من الانبعاثات.

ولعل من ابرز الحلول التي تم التوصل اليها إضافة شاحن توربيني لمحركات الاحتراق الداخلي .

ويعمل الشاحن التوربيني على زيادة كفاءة محركات الاحتراق الداخلي والتقليل من انبعاثاتها.

وفي هذا المشروع وقع الاختيار على محرك سيارة فولكس فاجن بيتل 1300cc ومن خلال العمليات

الحسابية توصلنا الى الحجم المناسب للشاحن التوربيني , ومنها تحسين كفاءة المحرك , وتمت

مطابقة اقرب شاحن توربيني موجود في السوق وهو GT1548.

الاهداء

إلى المعلم الأول .. إلى قائد هذه الأمة وقدوتها .. رسولنا محمد صلوات الله وسلامه عليه

إلى من كلله الله بالهيبة والوقار .. إلى من علمني العطاء دون انتظار .. إلى من أحمل إسمه بكل افتخار .. أرجو من الله أن يمد في عمرك لترى ثماراً قد حان قطافها بعد طول انتظار وستبقى كلماتك نجوماً أهتدي بها اليوم وفي الغد وإلى الأبد .
والدي العزيز

إلى ملاكي في الحياة .. إلى معنى الحب والحنان والتفاني .. إلى بسمة الحياة وسر التميز
إلى من كان دعائها سر نجاحي وحنانها بلسم جراحي إلى أغلى الأحباب.
أمي الغالي

إلى الشموع التي تحترق لتنير لنا الطريق .. إلى منهل العلم والمعرفة .. إلى من عبرنا على أيديهم وبمساعدتهم ورعايتهم إلى بر الأمان .. إلى من علمونا حروفاً من ذهب وكلمات من درر.
أساتذتنا الأفاضل

إلى الأسود القابعة خلف القضبان .. إلى من ضحوا بحريتهم من أجل حرية غيرهم.
الأسرى الأبطال

إلى من هم أكرم منا مكانة .. إلى من ضحوا بدمائهم في سبيل تحرير هذا الوطن.
الشهداء الأبرار

إلى من سرنا سوياً نشق الطريق معاً نحو النجاح والإبداع .
الزملاء والزميلات

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1 Chapter 1: Introduction

1.1 Introduction

Since 1860, when the first internal combustion engine was manufactured, research, studies and experiments have been initiated to obtain the highest efficiency of such engines and to improve their performance.

From here, we have generated the idea of matching a turbocharger to a motor that does not contain it, thus obtaining the maximum engine efficiency.

Adding a turbocharger to the engine enables us to improve the performance of the engine and get more horsepower compared to engines that do not have a turbocharger, which matches the principle and purpose of the turbocharger.

The principle of the turbocharger is to increase the amount of air and fuel entering the combustion chambers during the intake stroke, using the exhaust gas generated by the combustion, as will be explained later.

Increasing the amount of air and fuel in the combustion chambers increases combustion efficiency, which positively affects the resulting exhaust gases, which in turn protect the environment and reduces pollution in the air.

In this project, the 1973 Volkswagen Beetle gasoline engine(1300 L) was chosen to matching turbocharger , which is mainly aims to develop an old car engine while preserving its old classic form.

1.2 Literature review

TURBOCHARGER-DESIGN EFFECTS ON GASOLINE-ENGINE PERFORMANCE ,T. Koralcianitis and T. Sadoi , 2005

CONCLUSIONS

‘’ Theoretical turbocharger matching is useful to approach a range of turbocharger frames, but final testing is essential to investigate the effect of different turbochargers on the overall designpoint and off-design-point piston-engine cycle performance. Different turbochargers are advantageous to an engine for different types of operation. The performance of a turbocharger is determined by the combination of compressor and turbine specification, and the turbine is not as sensitive to engine matching as the compressor.’’ [3]

Design and Performance of a Gas-Turbine Engine from an Automobile Turbocharger , Lauren Tsa.i , 2004

CONCLUSION

‘’ The purpose of this project is to design and manufacture a gas turbine engine. The gas turbines designed and manufactured for this project show the operation of the Brayton cycle. The Brayton cycle is a suitable course for illustration because it contains a set of standard components, which are used in many other energy conversion applications. The Brayton cycle consists of a compressor, heat exchanger, turbine and other heat exchanger. The gas turbine engine operates an open version of the Brayton cycle and allows students to measure the temperature and pressure changes associated with each component of the system.’’[4]

Thermodynamic and experimental researches on matching strategies of the pre-turbine steam injection and the Miller cycle applied on a turbocharged diesel engine , Sipeng Zhu, Sheng Liu, Shuan Qu, Kangyao Deng , 2017

CONCLUSION

‘’ In this paper, the thermodynamic processes of pre-turbine steam injection and the first Miller cycle were studied, followed by matching strategies based on a non-dimensional matching map. Experiments are also underway to demonstrate the advantages of this new system applied to a turbocharged diesel engine. The results showed that the steam mass flow rate had a much greater effect on the air supply characteristics of the turbocharger compared to steam temperature, while the Miller cycle rate had a significant impact on the engine's air intake characteristics ’’ [5]

1.3 Time Schedule for the Project

As shown in Table 1.1 and Table 1.2 the time schedules for the project in steps. The time needed for the project is Thirty weeks., Table 1.1 and Table 1.2 list the tasks of the project respectively, and the needed tasks for each in gray.

First semester															
No. of week / Task	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Identification of project idea															
Writing project name and abstract and proposal															
Literature review															
Drawing the parts and maps															
Writing the project and reference															

Table 1 : Time schedule for the project

Second semester															
No. of week / Task	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Determine Mistakes	■	■													
Generate Solutions			■	■											
Calculatio n					■	■	■	■	■	■	■	■	■		
Writing and Documentation							■	■	■	■	■	■	■	■	■

Table 2 : Time schedule for the project

2 Chapter2: Technical Background

2.1 Internal Combustion Engines

An internal combustion engine is a machine that converts chemical energy to mechanical energy by burning the mixture (fuel and air) where the mixture is burned in a sealed chamber called the combustion chamber located at the top of the engine.

Figure 1: Internal Combustion Engines [1]

2.1.1.1 The Cylinder Head

It is a part of the internal combustion engine which is located at the top of the engine and the top of the cylinder block, as it contains a combustion chamber, forming a combustion chamber. The head can also be a place for fitting valves, spark plugs and fuel injectors. Contains columns, camshafts, valves, and all other connected devices, such as springs and reservoirs.

Figure 2: Cylinder Head [1]

2.1.1.2 The Engine Block

The engine block is the part that contains the internal combustion engine parts of the pistons, connecting rods, crankshaft, all bearings and other related devices.

Figure 3: Engine Block [1]

2.1.1.3 **The Piston**

The piston is a piece shaped disk inside a cylinder motor that hangs on a connecting rod.

Figure 4: piston [1]

2.1.1.4 **The Connecting Rod**

The connecting rod is connected by a clamp to a crankshaft ,It turns the frequency movement into a rotary motion.

Figure 5: connecting rod [1]

2.1.1.5 **The Crankshaft**

Crankshaft is a part that transforms the reciprocating motion into rotational motion. The crankshaft is installed on a number of key bearings and a dimension to one end.

Figure 6: Crankshaft [1]

2.1.2 **Theory of Operation**

in a 4 cylinder engine, it is the principle of working through the stroke. In the first stroke (Intake) , the piston begins at top dead center (T.D.C.) and ends at bottom dead center (B.D.C.), intake valve must be in the open position while the piston pulls an air-fuel mixture into the cylinder by producing vacuum pressure into the cylinder through its downward motion. The piston is moving down as air is being sucked in by the downward motion against the piston.

At the end of the Intake and when the piston is T.D.C. The Compression stroke begins where In this stroke the piston compresses the air-fuel mixture in preparation for ignition during the power stroke (below). Both the intake and exhaust valves are closed during this stage.

The crankshaft completed a complete revolution at 360 degrees. While the piston before T.D.C. Shortly after the compression stroke the Combustion stroke Start where , the compressed air mixture is ignited by a torch plug (in a gasoline engine) or high pressure heat (diesel engines), and the piston is strongly re-energized to B.D.C. This stroke produces the mechanical work of the engine to turn the crankshaft.

At the end of the explosion process, the piston enters and moves the exhaust valve open and the combustion gases are driven by the piston.

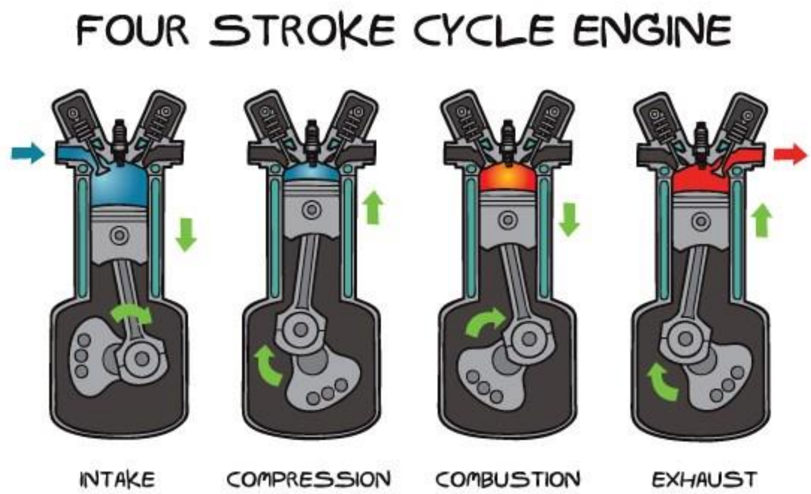


Figure 7: four stroke engine [1]

2.1.3 **Areas of Improvement Efficiency**

Engine efficiency can be increased in several different ways. One way is to add a turbocharger to the motor that increases the compression ratio within the engine to improve its efficiency. However, high pressure ratio can lead to blasting within the engine. Therefore, there are restrictions on determining the compression ratio of the engine, which depend on the fuel octane that will be in use and the amount of airflow through the engine, in each revolution.

2.1.4 **Basic Operation**

The turbocharger is a rotary mechanical device containing the turbine and the compressor where it is connected through a column. The turbocharger operates through the exhaust gas, where the exhaust gas circulates the turbine and the result of this spin leads to the spin of the compressor which causes the air pressure in the engine. This increase in air pressure tends to increase the ratio of the mixture that enters the engine to the same size of the cylinder, which increases the engine efficiency.

3 Chapter3 :Turbocharger

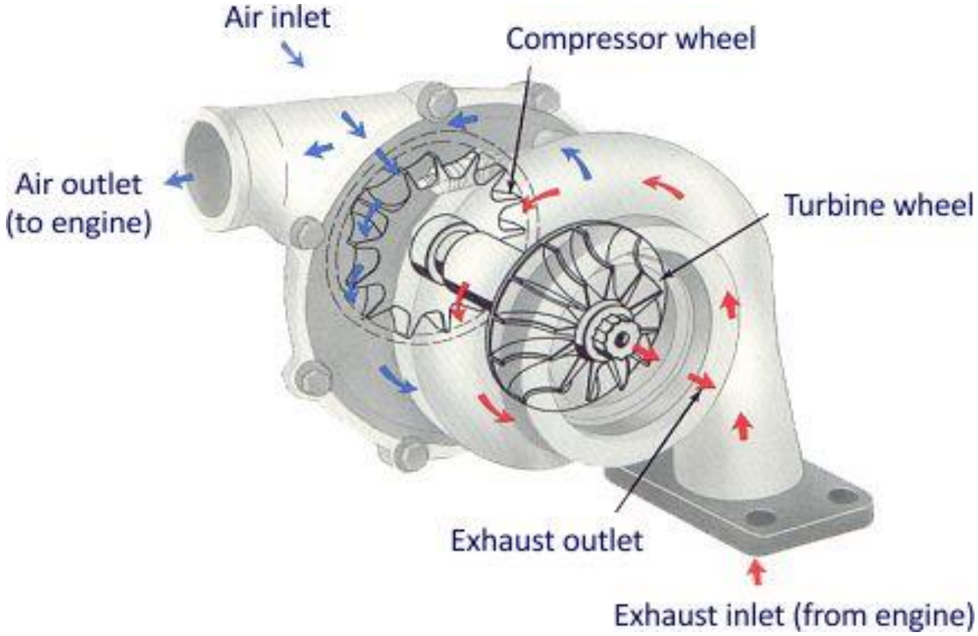


Figure 8 : Turbocharger [1]

3.1 **Turbocharger part**

The turbocharger consists of two main parts: the compressor part, the turbine part and the shaft , the rest of the parts, bearings, sealants and oil control components.

3.1.1 **The Compressor**

The compressor part consists of: the compressor wheel and the compressor cover. Within these components, there are many important design types and specific features such as the diffuser, which is a ring chamber with a number of rotors forming a series of different paths in the manifold. It directs the airflow from the impeller to the manifold at an angle designed to hold the maximum amount of energy transferred by the impeller and also connects the air to the manifold quickly and with pressure to be used in the combustion chambers.
Turbocharger compressors have design limits for their efficient functioning of air pressure. Each compressor contains optimum flow efficiency, maximum flow capacity (choke), and a pressure

point where, down it, will not flow when a certain amount of mass or will stop.
The compressor works 76% efficient; basically it has the ability to compress air at a constant efficiency level of 76%. The efficient efficiency of the compressor will never reach 100 percent, because there are factors that add unwanted heat.

3.1.1.1 The Compressor Wheel

The air compressor wheels are made of different aluminum alloy, which is the process of compressing the air and take the movement of the penis connecting them to the wheel of the Turbine.
The turbocharger wheel is called a beam pressure because it takes the fresh air and speeds it radially, or turns to 90 degrees.
The compressor wheel contains a number of critical areas, resulting in many of the available edges to adjust the flow parameters and match the compressor correctly with the engine.

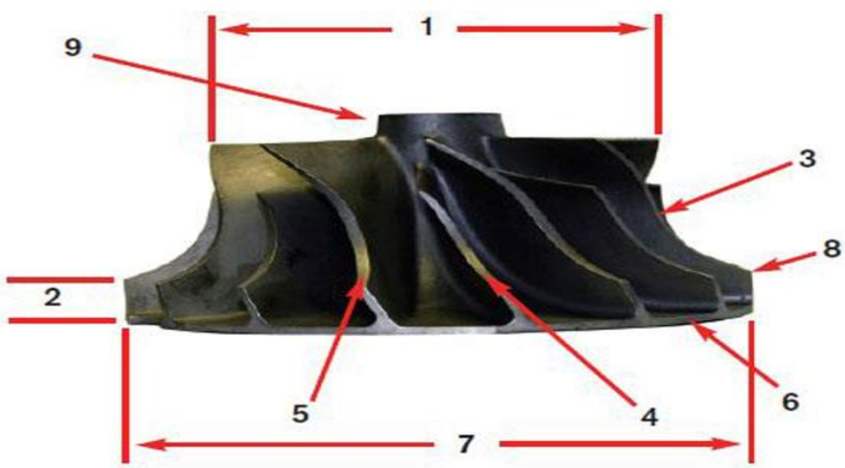


Figure 9 : Compressor wheel [1]

The compressor wheel has several important areas of design consideration: (1) inducer diameter, (2) tip height or tip width, (3) wheel contour, (4) splitter blade, (5) full blade, (6) backwall, (7) wheel diameter, tip diameter, or exducer diameter, (8) tip, impellor, or exducer, and (9) nose.

3.1.1.2 The Compressor Cover

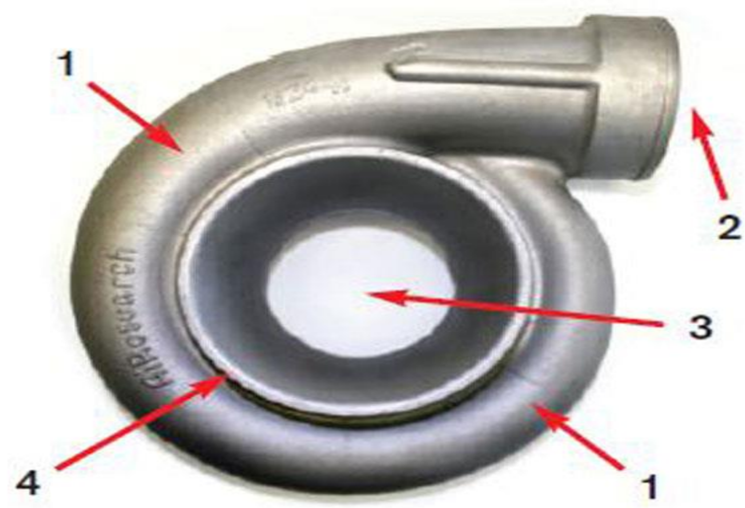


Figure 10 : The Compressor Cover [1]

It is made of aluminum alloy. The compressor portion of the compressor is an important part of the overall compressor cover design.
Compressor covers have an A / R rating or space above a radius relationship. However, compressors are not very sensitive to A / R rates.

3.1.2 The Turbine

The turbine is the first part that moves through the turbocharger, converting the energy in the exhaust into a mechanical energy to run
the compressor wheel, which in turn supplies the air to the engine. Thus, the turbine is where everything starts.
The turbine contains two main components:
The lid is called the turbine housing and the turbine wheel.

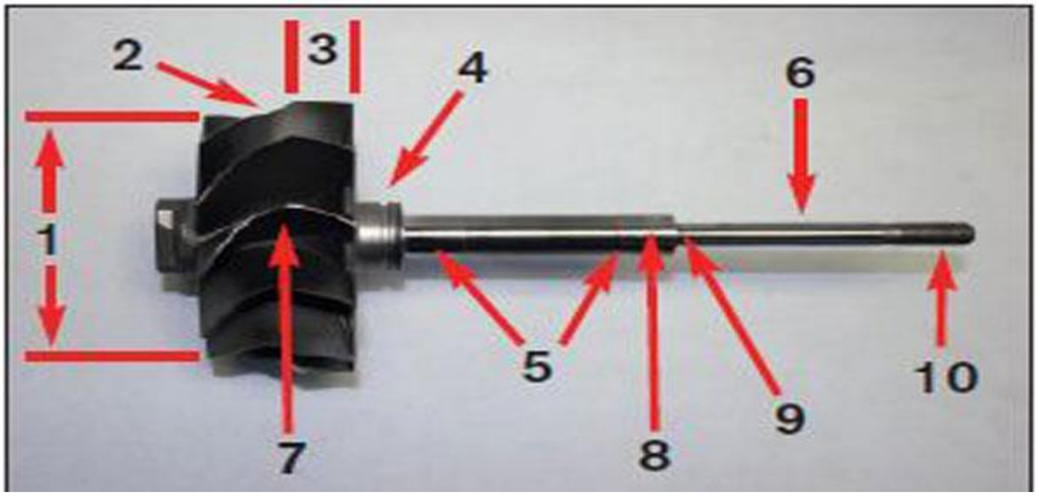


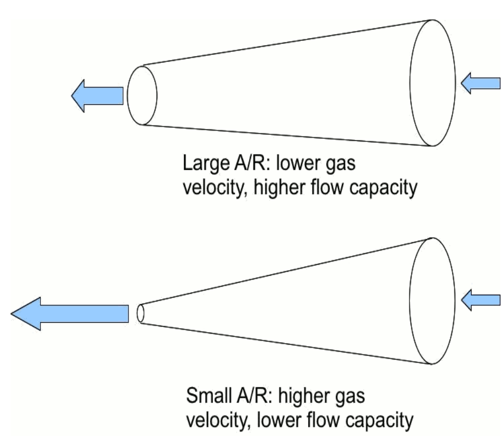
Figure 11 : the turbine shaft and wheel [1]

3.1.2.1 The Turbine Housing

It shall be made of steel or iron. Exhaust gases enter the turbine foot, a lip connected to the exhaust manifold. The gases the flow through the voltage through the turbine wheels to the outlet.

Figure 12 : Turbine Housing [1]

3.1.2.2 Turbine A/R ratio



The turbine housing is sized to optimize the pressure and flow as in the garden hose analogy. The larger the A/R is, the larger the swallowing volume. Therefore the smaller the A/R of the housing, the higher the pressure becomes in the turbine stage.

the “A/R ratio.” The ratio is where “A” is the area of the volute at the tongue of the turbine housing. The “R” is the radius from the center of the axis of rotation to the centroid of the volute.

Figure 13 : A/R ratio [1]

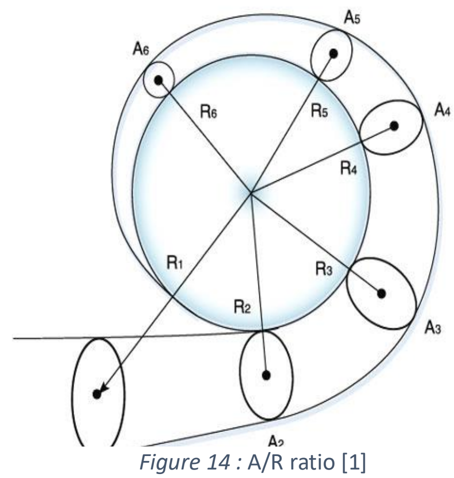


Figure 14 : A/R ratio [1]

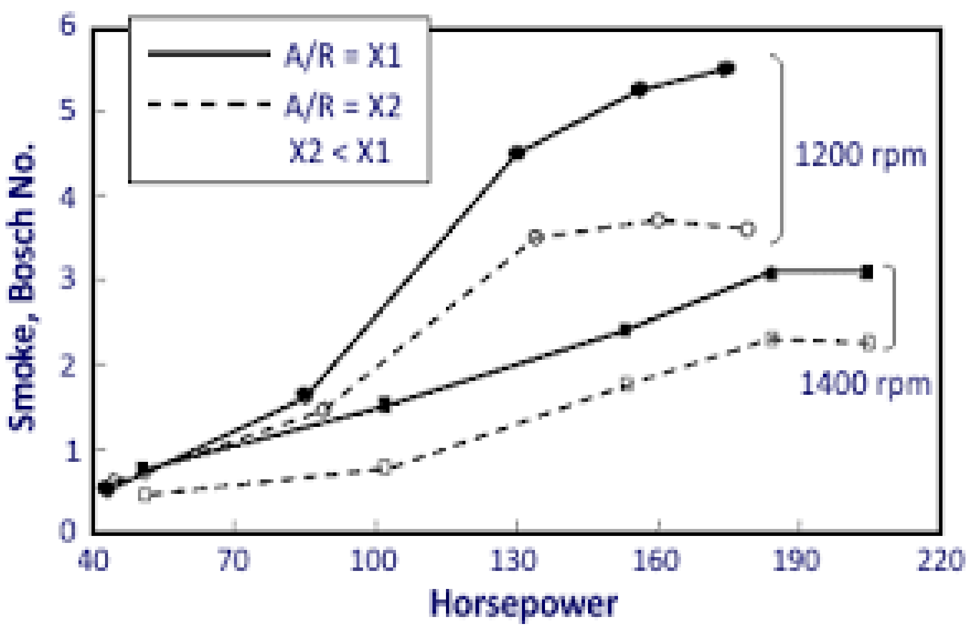


Figure 15 : A/R ratio with horsepower [1]

3.2 TYPES OF TURBOCHARGER :

3.2.1 Single-Turbo

Figure 16 : Single-Turbo [1]

Single turbocharger. Depending on the size of the elements inside the turbo, the characteristics of the torque can be achieved quite differently .Large turbos will bring on high top-end power, but smaller turbos will provide better low-end grunt as they spool faster.

3.2.2 Twin-Turbo

Just like a single turbocharger. It is used in the case of V6 or V8 engines, this can be done by setting one turbo to work with each cylinder bank, and provides better torque at low cycles (reducing turbo lage).

3.2.3 Twin-Scroll Turbo

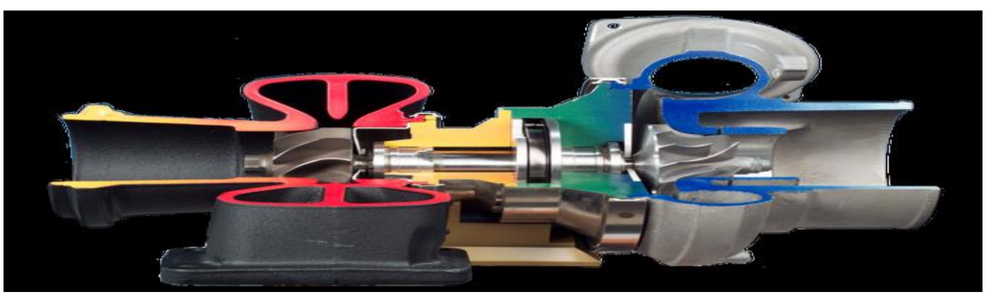


Figure 17 : Twin-Scroll Turbo [1]

The twin-turbocharged turbocharger requires a turbocharged ventricular cover and a manifold that works on the correct engine cylinder pairs with each roll. Independently. For example, in a four-cylinder engine (with the release order 1-3-4-2), cylinders 1 and 4 may be fed on one scroll of the turbo, while 2 and 3 cylinders feed on a separate scroll. This design provides a more efficient connection to exhaust gas to turbo. Where it provides higher air density per cylinder.

3.2.4 Variable Geometry Turbocharger (VGT)

A loop of dynamically shaped rotors in the turbine container at the turbine entrance. These rotors rotate to change the angle of the gas vortex and cross sectional area. These internal rotors change the ratio of the area to the turbine radius (A / R) to match the number of cycles per minute of the engines, thus providing the highest performance. When the number of cycles per minute decreases, the low A / R ratio allows the turbine to accumulate rapidly by increasing the speed of the exhaust gas. At higher cycles, the ratio of A / R increases, allowing increased airflow. This results in a low support threshold that reduces turbo delay and provides a wide and smooth torque range. As it is widely used in engines that operate on diesel because the degrees of exhaust gases are few compared to others.

3.2.5 Variable Twin-Scroll Turbocharger

The turbocharger combines the advantages of double turbocharged turbo and variable turbo geometry. This can be done by using a valve that can redirect the exhaust air flow to just a single pass, or by changing the quantity opened by the valve that can allow the exhaust gases to split into both passes. It is more suitable for engines that operate in gasoline .

3.2.6 Electric Turbochargers

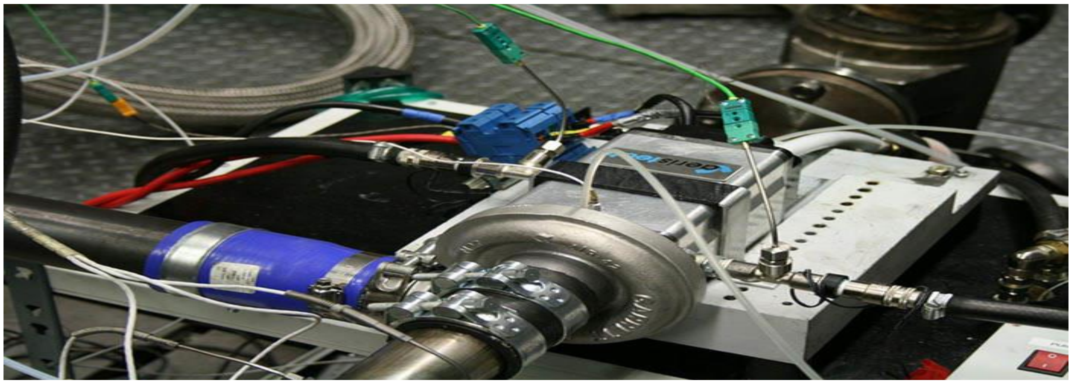


Figure 18 : Electric Turbochargers [1]

It is added to the normal turbocharger to minimize the slow-down of the turbocharger at the start of operation. This is accomplished by adding an electric motor circulating in the turbocharger from the beginning.

3.2.6.1 Small vs. Large Turbocharger

- One sure way to reduce the inertia of the turbine and compressor is to make the turbocharger smaller. A small turbocharger will provide boost more quickly and at lower engine speeds, but may not be able to provide much boost at higher engine speeds when a really large volume of air is going into the engine. It is also in danger of spinning too quickly at higher engine speeds, when lots of exhaust is passing through the turbine.
- A large turbocharger can provide lots of boost at high engine speeds, but may have bad turbo lag because of how long it takes to accelerate its heavier turbine and compressor.

3.2.7 Wastegates:

A wastegate is a mechanical device that is used to bypass part of the exhaust gases produced by the engine, so that it does not flow through the turbine housing for a turbocharger. In this way, the rotary speed of the turbocharger can be controlled and therefore the pressure is released from the compressor.

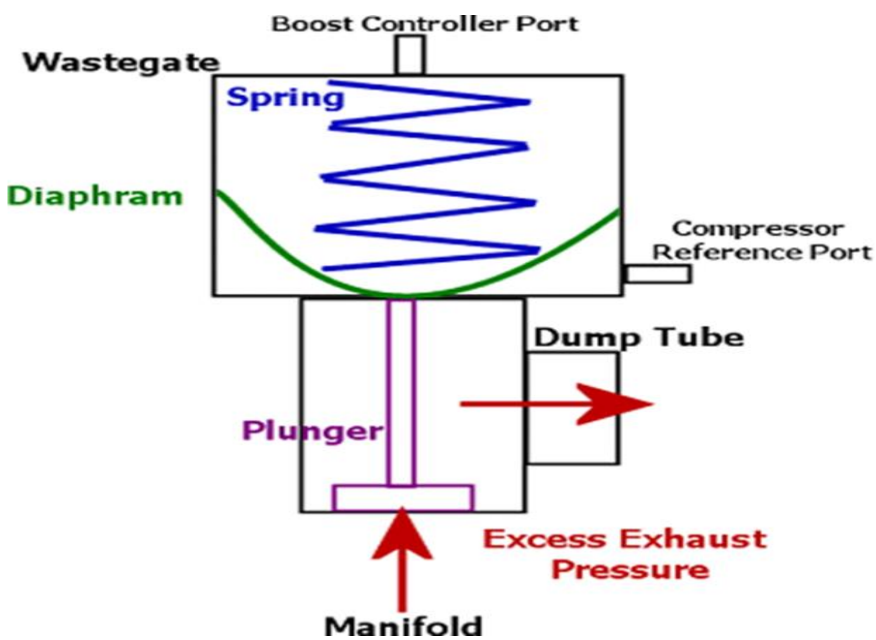


Figure 19 : Typical Wastegate Configuration [1]

4 Chapter 4 : Compressor and Turbine calculations

4.1 Compressor calculations

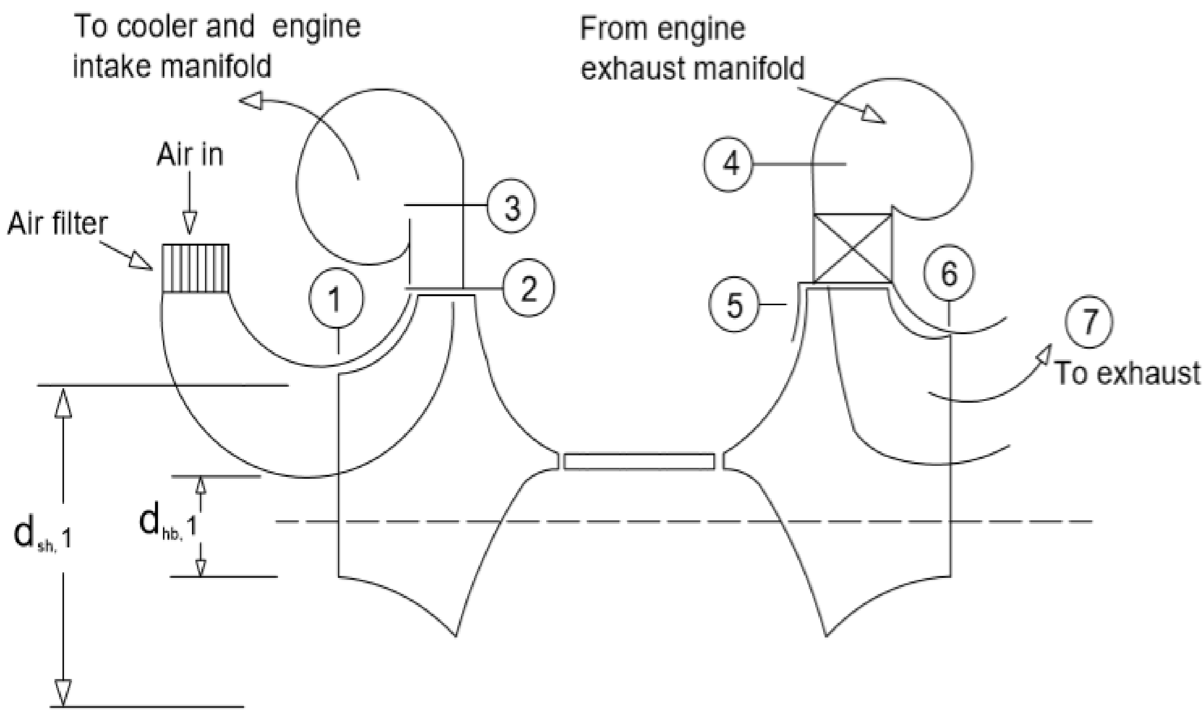


Figure 20 : Turbocharge

4.1.1 enthalpy

$$\frac{T_{0,3}}{T_{0,1}} = \left[\frac{P_{0,3}}{P_{0,1}}\right]^{\left[\left(\frac{R}{C_p}\right)\frac{1}{\eta_{p,c,ts}}\right]} \qquad (4\text{-}1)$$

$T_{0,3}$: outlet stagnation temperature

$T_{0,1}$: Inlet stagnation temperature (300 K)

$P_{0,3}$: outlet stagnation pressure

$P_{0,1}$: Inlet stagnation pressure ($1 * 10^5\ N/M^2$)

R : gas constant (287 $J \cdot kg^{-1} \cdot K^{-1}$)

C_p :Specific heat capacity (1010 J/(kg K))

$\eta_{p,c,ts}$: Polytropic efficiency (70%)

$$\frac{T_{0,3}}{T_{0,1}} = \left(\frac{2.5}{1}\right)^{\left[\left(\frac{287}{1010}\right)\ \frac{1}{0.7}\right]}$$

$$\frac{T_{0,3}}{T_{0,1}} = 2.5^{0.4059405941} = 1.450574383$$

$\Delta T_{0,1-3}$ = 0.4049405941 * 300 = 135.1723148 K

$$\Delta h_{0,1-3} = C_p * \Delta T_{0,1-3} \qquad (4\text{-}2)$$

= 1010 * 135.1723148

= 136524.038 J/Kg

$\Delta h_{0,1-3}$: Enthalpy

$(\Delta h_{0,1-3})^{3/4}$ = 7102.428064 J/Kg

4.1.2 Inlet volume flow rate

Now Guess inlet axial velocity(C_X) = 110 m/s

$$\frac{C_X}{\sqrt{RT_{0,1}}} = \frac{110}{\sqrt{287}*300} \quad (4\text{-}3)$$

= 0.3748789971

4.1.3 Mach number

This equation has been used to find the Mach number :

$$\frac{C_X}{\sqrt{RT_{0,1}}} = \sqrt{2}\frac{C_p}{R}\left[1-\left[1+\frac{M^2}{2\left[\frac{C_p}{R}-1\right]}\right]^{-1}\right] \quad (4\text{-}4)$$

$$\frac{110}{\sqrt{287}*300} = \sqrt{2}\frac{1010}{287}\left[1-\left[1+\frac{M^2}{2\left[\frac{1010}{287}-1\right]}\right]^{-1}\right]$$

M = 0.3204045291

M: Mach number AT point 1

Velocity Function vs Mach Number for Perfect Gases

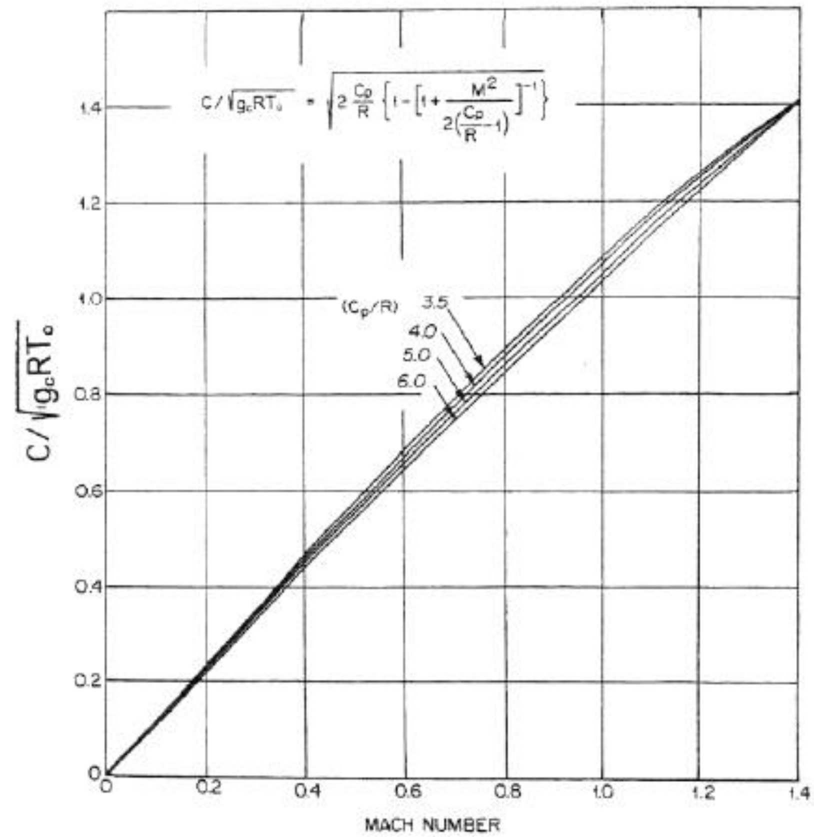


Figure 21 : Velocity Function vs Mach Number for Perfect Gases [2]

4.1.4 Rotation speed

$$\frac{\rho_0}{\rho_{st}} = \left[1 + \frac{M^2}{2\left[\frac{Cp}{R} - 1\right]}\right]^{\left[\frac{Cp}{R} - 1\right]} \qquad (4\text{-}5)$$

ρ_0 : density or air

ρ_{st} : static density

$$\frac{\rho_0}{\rho_{st}} = \left[1 + \frac{0.3204045291^2}{2\left[\frac{1010}{287} - 1\right]}\right]^{\left[\frac{1010}{287} - 1\right]}$$

$$\rho_0 = \frac{P_{0,1}}{RT_{0,1}} \qquad (4\text{-}6)$$

$= \frac{1*10^5}{287*300}$ = 1.161440186 Kg/M^3

$\rho_{st} = \frac{1.161440186}{1.052126928}$ = 1.103897405 Kg/M^3

$$\dot{m} = \rho_{st}\ \dot{v}_1 \qquad (4\text{-}7)$$

$\dot{m}$: Mass flow rate, kg/s

$\dot{v}_1$: volume flow rate, m^3/s

$\dot{v}_1 = \frac{1}{1.103897405}$ = 0.9058812852

$$N = \frac{60\left(\Delta h_{0,1-3}\right)^{3/4} N_S}{2\pi\sqrt{\dot{v}_1}} \qquad (4\text{-}8)$$

N_S : specific speed (0.6) According to the reference

N= $\frac{255687.4103}{5.950197294}$ = 42755.6814 rev/min

4.1.5 Diameter of impeller (d_2)

From Wiesner's correlation :

$$\sigma_w = \frac{1-\sqrt{\cos\beta_2}}{Z^{0,7}} \quad (4\text{-}9)$$

σ_w:Slip Factor

β_2 : Blade angle at periphery

Z : Number of rotor blades

$$\sigma_w = \frac{1-\sqrt{0,8660254038}}{17^{0,7}} \quad = 0.8719289203$$

$$\frac{C_{u,2}}{u_2} = \left[\frac{\tan\beta_2}{\tan\alpha_{c2}} + \frac{1}{\sigma_w}\right]^{-1} \quad (4\text{-}10)$$

$C_{u,2}$: Outlet Velocity from the compressor

u_2 : Outlet Velocity from the compressor

β_2 : Blade angle at periphery

α_{c2} : Flow angle at rotor exit

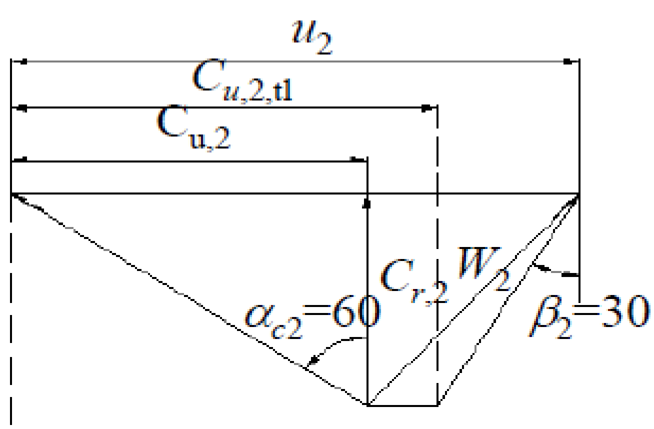


Figure 22 : Outlet Velocity Triangle [2]

$$\frac{C_{u,2}}{u_2} = \left[\frac{\tan 30_2}{\tan 60} + \frac{1}{0.8719289203}\right]^{-1}$$

$$= 0.6755771644 = \psi$$

ψ : loading coefficient

$$u_2 = \left[\frac{(\triangle h_0)}{\psi}\right]^{0.5} \quad (4\text{-}11)$$

$$= \left[\frac{136524.038}{0.6755771644}\right]^{0.5} = 449.5386964 \text{ m/s}$$

$$d_2 = \frac{60\, u_2}{\pi N} \quad (4\text{-}12)$$

$$= \frac{60*449.5386964}{\pi*42755.68142} = 200.8050484 \text{ mm}$$

4.1.6 Calculation of Optimum Shroud Parameter

$d_{sh,1}$(mm)	90	100	105	110	115	120	125
$u_{sh,1}$ (m/s) $(N\pi/60)\, d_{sh,1}($	201.481402	223.868	235.0616	246.25505	257.44845	268.641808	279.835285
$A_a(m^2)$ $\left(\pi\, d_{sh,1}{}^2/4\right) * (1 - 0.6^2)$	4.0715	5.026	5.5422675	6.082123377	6.648207	7.23	7.854
$\dot{m}\,\sqrt{RT_{0,1}/A_a P_{0.1}}$	0.7206	0.5838	0.529436	0.4824433785	0.4413641	0.4053	0.3736
M_1	0.86	0.61	0.508027	0.3694	0.404787	0.4053	0.3736
$\rho_{st,1}/\rho_{0,1}$	0.70818	0.836	0.881748	0.93489515	0.9227547	0.934	0.944
C_x(m/s) $= 0.86088/A_a\,(\rho_{st,1}/\rho_{0,1})$	298.67	204.93	176.16135	151.4	140.33041	127.31	116.06
$W_{sh,1} = \sqrt{C_x{}^2 + u_{sh,1}{}^2}$	360.275646	303.5017081	293.7461383	289.0735341	293.0735341	297.2814996	302.9483583

Table 3 : 4.1.6 Calculation of Optimum Shroud Parameter

$d_{sh,1}$: dimeter AT shroud

$W_{sh,1}$: relative velocity at shroud at entry

Choose a value of $d_{sh,1}$ =(110mm) because in this point itis minimum value of

$W_{sh,1}$ = 289.0735341 m/s because itis minimum value

4.1.7 Impeller-Outlet and diffuser-inlet width, *b*:

$$C_2 = \frac{\psi u_2}{\sin \alpha_{c,2}} \qquad (4\text{-}13)$$

$$= \frac{0.6455771644 * 449.5386964}{\sin 60}$$

C_2 : Absolute Velocity At point 2

= 340.3996334 m/s

$T_{0,2} = T_{0,1} * 2.5^{0.4059405941}$

= 435.1723148 K

$$\frac{C_2}{\sqrt{RT_{0,2}}} = \frac{110}{\sqrt{278*435.1723148}}$$

= 0.9632022894

$$\frac{C_2}{\sqrt{RT_{0,2}}} = \sqrt{2}\frac{C_p}{R}\left[1-\left[1+\frac{M^2}{2\left[\frac{C_p}{R}-1\right]}\right]^{-1}\right]$$

$$= \frac{110}{\sqrt{278*435.1723148}} = \sqrt{2}\frac{1010}{287}\left[1-\left[1+\frac{M^2}{2\left[\frac{1010}{287}-1\right]}\right]^{-1}\right]$$

M_2= 0.8746464285

M_2=: Mach number At point number (2)

$$\frac{\rho_{0,2}}{\rho_{st,2}} = \left[1+\frac{{M_2}^2}{2\left[\frac{C_p}{R}-1\right]}\right]^{\left[\frac{C_p}{R}-1\right]} \qquad (4\text{-}14)$$

ρ_0 : density or air At point number (2)

ρ_{st} : static density At point number (2)

$$\frac{\rho_{0,2}}{\rho_{st,2}} = \left[1 + \frac{0.8746464285^2}{2\left[\frac{1010}{287} - 1\right]}\right]^{\left[\frac{1010}{287} - 1\right]}$$

$$\frac{\rho_{st,2}}{\rho_{0,2}} = 0.7003935618$$

$$\frac{P_{0,2}}{P_{0,1}} = \left[\frac{T_{0,2}}{T_{0,1}}\right]^{\left[\frac{C_p}{R}^{\eta_{p,c,tt\ 1-2}}\right]} \qquad (4\text{-}15)$$

$\eta_{c,tt\ 1-2}$: Polytropic efficiency in compressor from point 1-2

$$\frac{P_{0,2}}{P_{0,1}} = \left[\frac{435.1723}{300}\right]^{\left[3.52^{0.73}\right]}$$

$$P_{0,2} = 2.505971597 * 10^{5} \text{ N}/m^2$$

$$\rho_0 = \frac{P_{0,2}}{RT_{0,2}} \qquad (4\text{-}16)$$

$$\rho_{0,2} = \frac{2.505971597*10^{5}}{287*435.1723} = 2.006471541 \text{ Kg}/m^3$$

$\rho_{st,2}$ = 1,405319731 Kg/m^3

$$C_{r,2} = C_2 \cos 60 \qquad (4\text{-}17)$$

$C_{r,2}$: flow velocity

= 340.3996334 * cos 60 = 170.1998167 m/s

$$b = \frac{\dot{m}}{(\pi d_2 \rho_{st,2} C_{r,2})} \qquad (4\text{-}18)$$

$$= \frac{1}{\pi*200.8050484*1.405319731*170.1998167} = 6.62737077 \text{ mm}$$

4.1.8 Diameter of hub ($d_{hb,1}$)

Choose a value of $\frac{d_{hp,1}}{d_{sh,1}} = 0.6$ range (0.6 – 0.85) from the reference

$d_{hp,1}$: diameter hub

$d_{hp,1} = 0.6 * 110$ = 66 mm

4.1.9 Radial Diffuser :

$$R_{e,2=}\frac{C_2(d_2/2)\rho_{st,2}}{\mu_{st,2}} \quad (4\text{-}19)$$

$R_{e,2}$: Radial Diffuser Stability

$\mu_{st,2}$: Dynamic viscosity of air from table at $T_{st,2\,=\,350\,K}$

$R_{e,2} = \frac{48029.58776}{2.02*10^{-5}}$ = $2377.702364 * 10^6$

$\frac{b}{r_2} = \frac{6.62737077}{100.4025242}$ = 0.06600800949

$r_3/r_2 = 2.1$ From the Stability limits in Jansen's curves

80 percent of 2.1 =1.68

$$d_{3\,=}\,1.68 * d_2 \quad (4\text{-}20)$$

= 1.68 * 200.8050484 = 337.3524813 mm

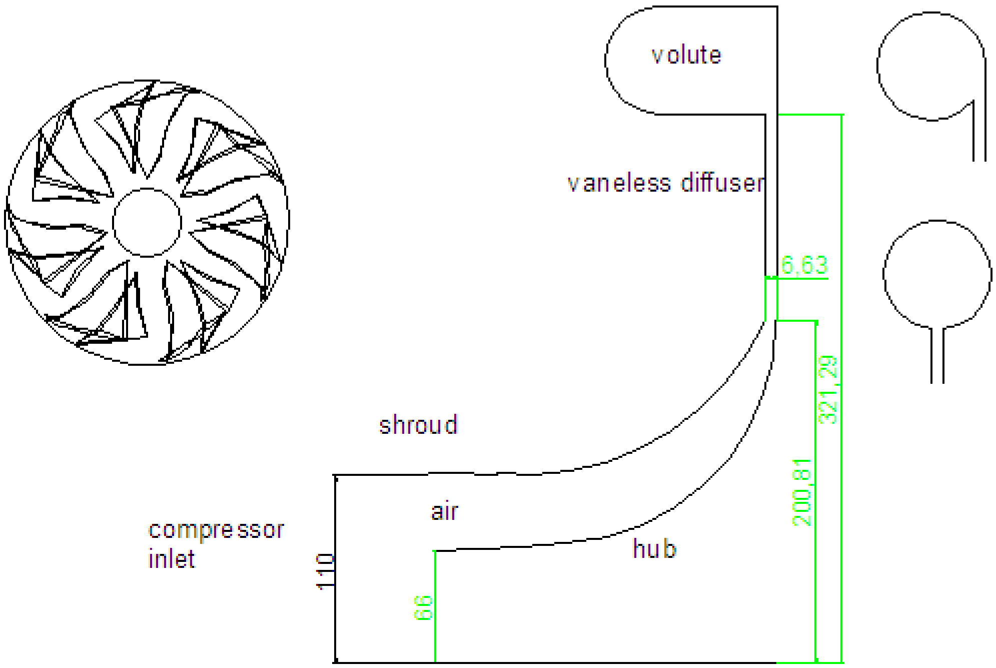


Figure 23 : compressor

4.2 Turbine calculations

$$Q = engine\ size * \frac{rpm}{2} \quad (4\text{-}21)$$

Q : volume flow rate

Q = 1.3 * $\frac{4000}{2}$ = 43.44 $\frac{m^3}{s}$

$$A_5 = \frac{\pi}{4} {d_5}^2 \quad (4\text{-}22)$$

A_5 : aria at point 5

d_5 : dimeter at point 5

= $\frac{\pi}{4}$ 0.195^2 = 0.0283528737 m^2

$$u_5 = \frac{Q}{A_5} \quad (4\text{-}23)$$

u_5 : velocity at point 5

= 1450.873622 m/s

$$\Delta h = \psi * {u_5}^2 \quad (4\text{-}24)$$

= 0.834 * 1450.873622^{2} = 1755598.578

$$\varphi = \frac{\psi}{tan\ 70} \quad (4\text{-}25)$$

= 0.3035511754

$$N_{S=} \frac{2\pi\ N}{60} * \frac{\sqrt{Q}}{(\Delta h)^{3/4}} \quad (4\text{-}26)$$

= 0.6110797322

$$\frac{b_5}{d_5} = \frac{N_s \tan \alpha_{c,5} \psi^{0.5}}{4\pi} \quad (4\text{-}27)$$

= 0.07455951122

b_5 = 14.53910469 mm

Choose a value of $\frac{d_{sh,6}}{d_5} = 0.6$

$d_{sh,6}$ = 0.6 * 195 = 117 mm

$$\sqrt{1 - \Lambda^2} = \sqrt{1 - \left(\frac{d_{hb,6}}{d_{sh,6}}\right)^2} = \frac{\Psi^{3/4} N_s}{\sqrt{\pi\varphi}} \quad (4\text{-}28)$$

$$= \frac{(0.834)^{3/4} * 0.6110797322}{\sqrt{\pi 0.3035511745}} = 0.5714654909$$

Λ = 0.8377123343

$$d_{hb,1} = \Lambda * d_{sh,1} \quad (4\text{-}29)$$

= 98.01234312 mm

5 Turbocharge matching

We have created the matching by calculating the ratio of Trim and A / R in the turbines and engine displacement.

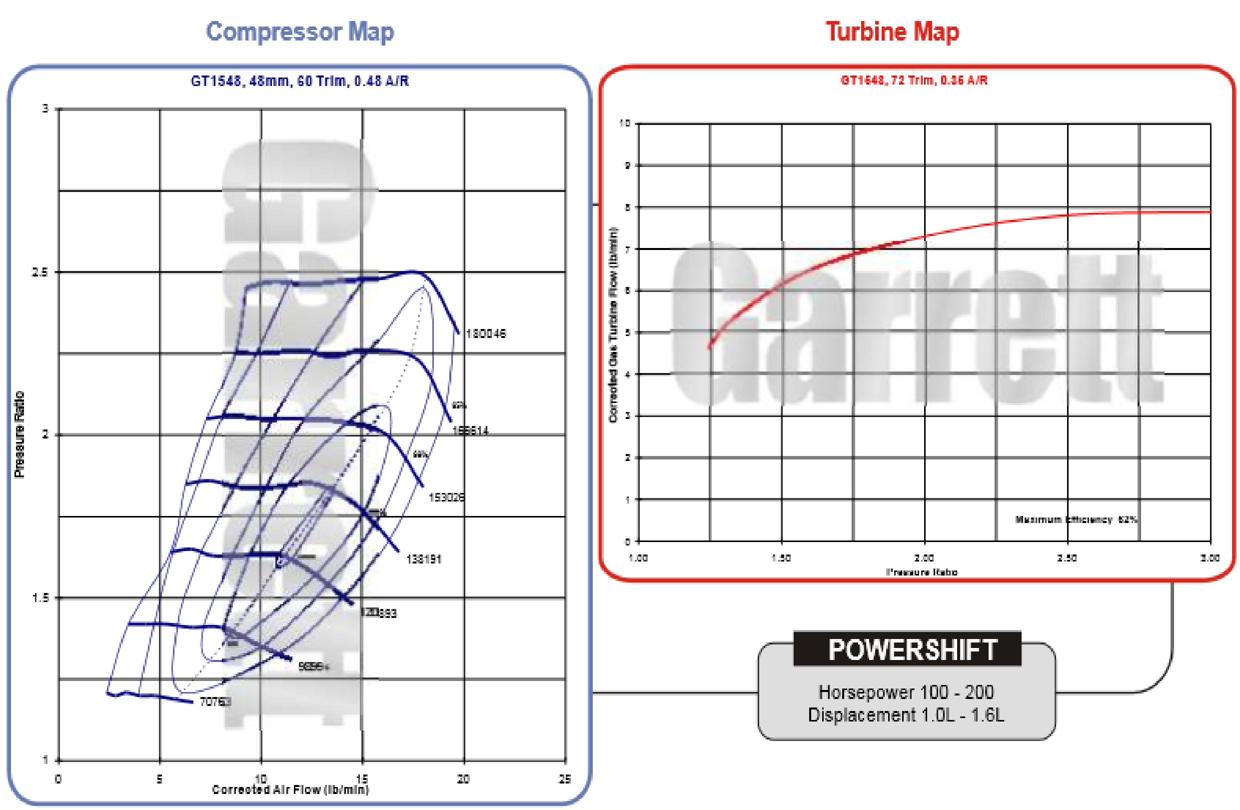
engine displacement = 1300 L

$$Trim = \left(\frac{d_{hb,1}}{d_{sh,1}}\right)^2 * 100\% \tag{5-1}$$

$$= \left(\frac{98.01234312}{117}\right)^2 * 100\% = 70.1761955$$

$$A/R = \frac{0.0283528737}{0.1465} \approx 0.19353 \tag{5-2}$$

From the Garrett catalog it was found that the nearest turbocharger is GT1548 (Trim 72 , A/R 0.35) .



GT1544		COMPRESSOR				TURBINE			
Turbo	CHRA	Wh Dia Ind	Exd	Trim	A/R	Wh Dia	Trim	A/R	Type
466755-3	431876-93	37.2mm	48.0mm	60	0.48	41.2mm	72	0.35	Wastegated

Figure 24 : GT1548 [1]

6 Conclusion

6.1 Conclusion

The summary of the project is centered on calculating the dimensions of the turbine fan and the compressor. Through the knowledge of these dimensions, the turbine charger was matched by catologues. The appropriate turbocharger was selected.

7 Reference

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3. Korakianitis, T. and T. Sadoi, *Turbocharger-design effects on gasoline-engine performance.* Journal of Engineering for Gas Turbines and Power(Transactions of the ASME), 2005. **127**(3): p. 525-530.
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8 Appendix A

Ideal Gas Properties of Air

Temp. (K)	c_v (kJ/kg-K)	c_p (kJ/kg-K)	u (kJ/kg)	h (kJ/kg)	$\int_{T_{ref}}^{T} \frac{c_p(T)}{T} dT$ (kJ/kg-K)
200	0.7153	1.002	142.7	200.1	5.299
220	0.7155	1.003	157.0	220.2	5.394
240	0.7158	1.003	171.3	240.2	5.481
260	0.7162	1.003	185.6	260.3	5.562
280	0.7168	1.004	200.0	280.3	5.636
300	0.7177	1.005	214.3	300.4	5.705
320	0.7188	1.006	228.7	320.5	5.770
340	0.7202	1.007	243.1	340.7	5.831
360	0.7219	1.009	257.5	360.8	5.889
380	0.7239	1.011	272.0	381.0	5.944
400	0.7262	1.013	286.5	401.3	5.995
420	0.7289	1.016	301.0	421.6	6.045
440	0.7318	1.019	315.6	441.9	6.092
460	0.7350	1.022	330.3	462.3	6.137
480	0.7385	1.026	345.0	482.8	6.181
500	0.7423	1.029	359.8	503.3	6.223
520	0.7462	1.033	374.7	524.0	6.263
540	0.7504	1.037	389.7	544.7	6.302
560	0.7547	1.042	404.7	565.5	6.340
580	0.7592	1.046	419.9	586.3	6.377
600	0.7638	1.051	435.1	607.3	6.412
620	0.7685	1.055	450.4	628.4	6.447
640	0.7732	1.060	465.8	649.5	6.480
660	0.7780	1.065	481.3	670.8	6.513
680	0.7828	1.070	497.0	692.1	6.545
700	0.7876	1.075	512.7	713.6	6.576
720	0.7925	1.079	528.5	735.1	6.606
740	0.7973	1.084	544.4	756.8	6.636
760	0.8020	1.089	560.3	778.5	6.665
780	0.8068	1.094	576.4	800.3	6.693
800	0.8114	1.098	592.6	822.2	6.721
820	0.8160	1.103	608.9	844.3	6.748
840	0.8206	1.108	625.3	866.4	6.775
860	0.8250	1.112	641.7	888.6	6.801
880	0.8294	1.116	658.3	910.8	6.827
900	0.8337	1.121	674.9	933.2	6.852
920	0.8379	1.125	691.6	955.7	6.876
940	0.8420	1.129	708.4	978.2	6.901
960	0.8460	1.133	725.3	1001	6.924
980	0.8500	1.137	742.2	1024	6.948
1000	0.8538	1.141	759.3	1046	6.971
1020	0.8575	1.145	776.4	1069	6.993
1040	0.8612	1.148	793.6	1092	7.016
1060	0.8648	1.152	810.8	1115	7.038
1080	0.8682	1.155	828.2	1138	7.059
1100	0.8716	1.159	845.6	1161	7.080
1120	0.8749	1.162	863.0	1185	7.101
1140	0.8782	1.165	880.6	1208	7.122

Temp. (K)	c_v (kJ/kg-K)	c_p (kJ/kg-K)	u (kJ/kg)	h (kJ/kg)	$\int_{T_{ref}}^{T} \frac{c_p(T)}{T} dT$ (kJ/kg-K)
1160	0.8813	1.168	898.2	1231	7.142
1180	0.8843	1.171	915.8	1255	7.162
1200	0.8873	1.174	933.5	1278	7.182
1220	0.8902	1.177	951.3	1301	7.201
1240	0.8930	1.180	969.1	1325	7.220
1260	0.8958	1.183	987.0	1349	7.239
1280	0.8985	1.185	1005	1372	7.258
1300	0.9011	1.188	1023	1396	7.276
1320	0.9036	1.191	1041	1420	7.294
1340	0.9061	1.193	1059	1444	7.312
1360	0.9085	1.196	1077	1468	7.330
1380	0.9109	1.198	1095	1492	7.347
1400	0.9132	1.200	1114	1516	7.365
1420	0.9154	1.202	1132	1540	7.382
1440	0.9176	1.205	1150	1564	7.398
1460	0.9197	1.207	1169	1588	7.415
1480	0.9218	1.209	1187	1612	7.432
1500	0.9239	1.211	1206	1636	7.448
1520	0.9259	1.213	1224	1660	7.464
1540	0.9278	1.215	1243	1685	7.480
1560	0.9297	1.217	1261	1709	7.495
1580	0.9316	1.219	1280	1733	7.511
1600	0.9334	1.220	1298	1758	7.526
1620	0.9352	1.222	1317	1782	7.541
1640	0.9369	1.224	1336	1807	7.556
1660	0.9386	1.226	1355	1831	7.571
1680	0.9403	1.227	1373	1856	7.586
1700	0.9419	1.229	1392	1880	7.600
1720	0.9435	1.231	1411	1905	7.615
1740	0.9451	1.232	1430	1929	7.629
1760	0.9466	1.234	1449	1954	7.643
1780	0.9481	1.235	1468	1979	7.657
1800	0.9496	1.237	1487	2003	7.671
1820	0.9511	1.238	1506	2028	7.684
1840	0.9525	1.240	1525	2053	7.698
1860	0.9539	1.241	1544	2078	7.711
1880	0.9553	1.242	1563	2103	7.725
1900	0.9566	1.244	1582	2127	7.738
1920	0.9579	1.245	1601	2152	7.751
1940	0.9592	1.246	1620	2177	7.764
1960	0.9605	1.248	1640	2202	7.776
1980	0.9618	1.249	1659	2227	7.789
2000	0.9630	1.250	1678	2252	7.802
2020	0.9633	1.250	1698	2277	7.814
2040	0.9645	1.252	1717	2303	7.826
2060	0.9656	1.253	1736	2328	7.839
2080	0.9668	1.254	1756	2353	7.851
2100	0.9679	1.255	1775	2378	7.863

Temp. (K)	c_V (kJ/kg-K)	c_p (kJ/kg-K)	u (kJ/kg)	h (kJ/kg)	$\int_{T_{ref}}^{T} \frac{c_p(T)}{T} dT$ (kJ/kg-K)
2120	0.9689	1.256	1794	2403	7.875
2140	0.9700	1.257	1814	2428	7.886
2160	0.9711	1.258	1833	2453	7.898
2180	0.9721	1.259	1853	2478	7.910
2200	0.9731	1.260	1872	2503	7.921
2220	0.9741	1.261	1891	2529	7.933
2240	0.9751	1.262	1911	2554	7.944
2260	0.9761	1.263	1930	2579	7.955
2280	0.9770	1.264	1950	2604	7.966
2300	0.9779	1.265	1970	2630	7.977
2320	0.9789	1.266	1989	2655	7.988
2340	0.9798	1.267	2009	2680	7.999
2360	0.9807	1.268	2028	2706	8.010
2380	0.9815	1.269	2048	2731	8.021
2400	0.9824	1.269	2068	2756	8.031
2420	0.9833	1.270	2087	2782	8.042
2440	0.9841	1.271	2107	2807	8.052
2460	0.9850	1.272	2127	2833	8.063
2480	0.9858	1.273	2146	2858	8.073
2500	0.9866	1.274	2166	2884	8.083
2520	0.9874	1.274	2186	2909	8.093
2540	0.9882	1.275	2206	2935	8.103
2560	0.9890	1.276	2225	2960	8.113
2580	0.9897	1.277	2245	2986	8.123
2600	0.9905	1.278	2265	3011	8.133
2650	0.9924	1.279	2314	3075	8.158
2700	0.9942	1.281	2364	3139	8.182
2750	0.9960	1.283	2414	3203	8.205
2800	0.9977	1.285	2464	3267	8.228
2850	0.9994	1.286	2514	3332	8.251
2900	1.001	1.288	2564	3396	8.273
2950	1.003	1.290	2614	3460	8.295
3000	1.004	1.291	2664	3525	8.317
3050	1.006	1.293	2714	3590	8.338
3100	1.007	1.294	2765	3654	8.359
3150	1.009	1.296	2815	3719	8.380
3200	1.010	1.297	2865	3784	8.401
3250	1.012	1.299	2916	3849	8.421
3300	1.013	1.300	2967	3914	8.441
3350	1.015	1.302	3017	3979	8.460
3400	1.016	1.303	3068	4044	8.479
3450	1.017	1.304	3119	4109	8.498
3500	1.019	1.306	3170	4174	8.517

9 Appendix B

Viscosity of Air dynamic and kinematic

TABLE C.1 Physical Properties of Air at Standard Atmospheric Pressure (SI Units)[a]

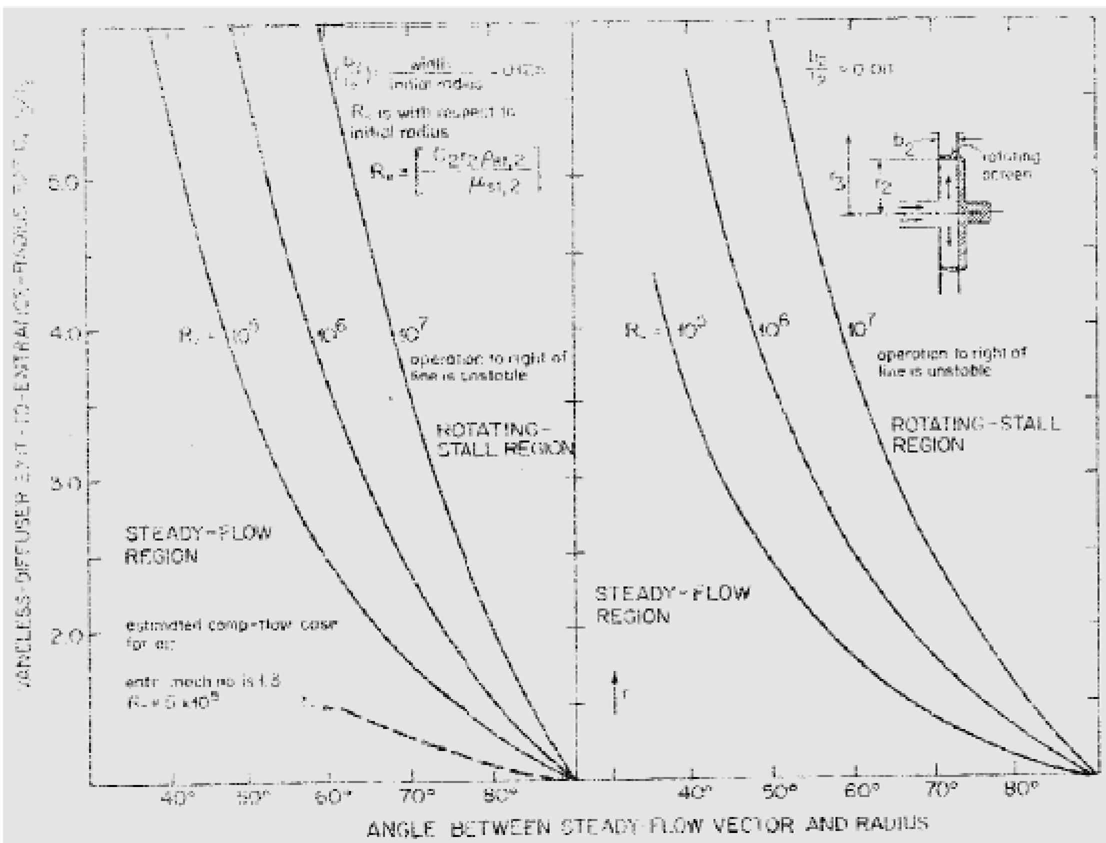
Temperature (°C)	Density, ρ (kg/m^3)	Dynamic viscosity, μ ($N \cdot s/m^2$)	Kinematic viscosity, ν (m^2/s)	Specific heat ratio, γ
−40	1.514	1.57 E − 5	1.04 E − 5	1.401
−20	1.395	1.63 E − 5	1.17 E − 5	1.401
0	1.292	1.71 E − 5	1.32 E − 5	1.401
5	1.269	1.73 E − 5	1.36 E − 5	1.401
10	1.247	1.76 E − 5	1.41 E − 5	1.401
15	1.225	1.80 E − 5	1.47 E − 5	1.401
20	1.204	1.82 E − 5	1.51 E − 5	1.401
25	1.184	1.85 E − 5	1.56 E − 5	1.401
30	1.165	1.86 E − 5	1.60 E − 5	1.400
40	1.127	1.87 E − 5	1.66 E − 5	1.400
50	1.109	1.95 E − 5	1.76 E − 5	1.400
60	1.060	1.97 E − 5	1.86 E − 5	1.399
70	1.029	2.03 E − 5	1.97 E − 5	1.399
80	0.9996	2.07 E − 5	2.07 E − 5	1.399
90	0.9721	2.14 E − 5	2.20 E − 5	1.398
100	0.9461	2.17 E − 5	2.29 E − 5	1.397
200	0.7461	2.53 E − 5	3.39 E − 5	1.390
300	0.6159	2.98 E − 5	4.84 E − 5	1.379
400	0.5243	3.32 E − 5	6.34 E − 5	1.368
500	0.4565	3.64 E − 5	7.97 E − 5	1.357
1000	0.2772	5.04 E − 5	1.82 E − 4	1.321

[a] Based on data from R. D. Blevins, *Applied Fluid Dynamics Handbook*, Van Nostrand Reinhold Co., Inc., New York, 1984.

10 Appendix C

Stable operating range of vaneless diffusers (Jansen, 1964)

Radial Diffuser Stability (… contd.)



Stable operating range of vaneless diffusers (Jansen, 1964)