

Turbo Charger Analysis

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Contents

1. Introduction	2
2. Research	3
3. Methodology	5
4. Results	7
5. Analysis	9
6. Conclusion	12
7. Bibliography	13
8. Appendices	15

1. Introduction

Turbochargers have been in use for decades as a relatively easy and inexpensive way to improve the volumetric efficiency of an internal combustion engine. They have become especially popular in recent years due to the emission standards becoming stricter, and their versatility in being able to pass drive cycles whilst maintaining drivability on the road. Turbochargers can be described as an early mechanical type of energy recovery system, as they use the heat energy wasted in the exhaust.

Parkers (2018) claim that:

The benefits these small turbocharged powerplants offer are higher levels of fuel efficiency and lower CO₂ emissions without sacrificing the speed buyers have become accustomed to. Reducing CO₂ output benefits car buyers in terms of lower VED car tax, and allows car makers to meet ever-tighter exhaust gas legislation.

In this report, the basics of a turbocharger system will be studied, the flow of air through the system will be analysed and a strip down of a turbo will be documented. In addition, a case study will be analysed, and suggestions presented for suitable compressor choices and intercooler design.

The aims of this report are to explain the fundamentals of turbocharging systems and how they are being adapted into modern technology, as well as successfully selecting appropriate components for a case study turbo application and justify those choices.

2. Research

A basic turbocharger uses exhaust gases to drive a turbine at very high speeds. This turbine connects to a compressor wheel, which pressurises the intake air. This provides the cylinders with a much higher mass of air, so more fuel can be burnt at once. This gives more power and less fuel consumption, as well as reduced noise pollution due to the intake noise muffling effect. (Vousden, 2016)

An intercooler is used to cool the compressed intake air while an oil feed cools the turbocharger unit itself. A wastegate is often used to control the boost pressure and prevent over boosting. (HoltsAuto, 2017)

The boost pressure must be carefully controlled, and this can be done in multiple ways. A simple method is using a wastegate as mentioned above. The boost pressure acts against a diaphragm (appendices 37 and 41 show the airflow through this system), which opens the wastegate to allow exhaust gases to bypass the turbine. This in turn limits the speed of the turbine. Modern electronic systems provide adaptive control of the boost pressure, taking into consideration the various operating conditions of the engine. This is done again with a diaphragm, but the control pressure is modulated by a control unit. Variable geometry turbos can also be used – these control boost pressure by changing the cross-section through which the exhaust gases flow into the turbine (see appendix 38). This gives a higher efficiency of the system than a wastegate controlled turbo. The cross-section is varied using vanes which open up and close to manipulate the velocity of the gases. (BorgWarner, N.D.)

When throttle input is reduced, the turbine maintains inertia and continues to generate boost pressure. This can put strain on components, as the pressure has nowhere to go, and can even attempt to go backwards through the compressor wheel. To prevent this, manufacturers fit compressor bypass valves or blow off valves, which release this pressure, either into the beginning of the intake system or the atmosphere respectively. (Hyde, 2010)

Intercoolers are fitted to turbocharging systems as compressing the intake air causes a rise in temperature. The intercooler cools the intake air down to densify it again and maintain the benefits of increasing the charge pressure. (Valeo, N.D.)

One concern of turbocharger systems is the delay in power response, known as turbo lag. This is when the turbine is at a low speed and the throttle is opened suddenly. It takes time for the turbine to spin up to speed, therefore creating a delay before boost pressure is built up and full power is reached. This can make turbocharged vehicles more difficult to drive. Manufacturers have developed a few different ways of reducing this lag period. One of these is by using multiple turbochargers (for example in twin-turbo or bi-turbo setups). Variable geometry turbos are also commonly in use, which can control the velocity of exhaust gases through the turbine. Other systems developed by Audi and Volvo inject compressed air into the turbine to spin it up to speed faster. (Parkers, 2018)

Within motorsport, hybrid-electric turbos are also used. This is where a motor-generator is attached to the spindle shaft. This acts as a boost controller as it can slow down the turbine, but without wasting energy of the gases which would normally escape via the wastegate. This energy is converted into electrical energy and stored in a battery. The energy can then be

used to power a drive motor, or to speed up the turbine in order to build boost pressure quicker and eliminate turbo lag. (F1, 2015)

The flow of gases through the entire system is illustrated in appendix 39. Ambient air enters the system through the compressor inlet (normally via ducting from the front of the vehicle and an air filter/airbox). It is then compressed by the compressor wheel and flows through the intercooler to recover the lost density due to the generated heat. From the intercooler it flows through the throttle body and into the inlet manifold / cylinders. Once combustion has taken place, the gases normally flow through the manifold and straight into the turbine (or some through the wastegate). Once the gases leave the turbine, they flow through the rest of the exhaust system and out into the atmosphere.

Within the turbocharger unit itself, the flow is shown in appendix 40. The air enters the compressor axially and once compressed, leaves circumferentially. On the turbine side, this is the opposite – gases enter the turbine circumferentially and speed up the turbine, then leave axially into the exhaust system.

Appendix 42 shows some computational fluid dynamics analysis results in the form of streamlines. This shows that the gases flowing into the turbine and compressor both present smooth laminar flow. Once the gases have passed through the wheels, the flow is somewhat disturbed – some turbulence can be observed, and a small level of helical flow is induced. This will have a negative impact on the exhaust flow, as this will cause a higher pressure and slowing down of the gases after the turbine. On the compressor side, it can be seen as beneficial due to the increased atomisation of fuel in turbulent flow, however the overall effects will be negative. The energy losses in the boundary layer will be much higher with this type of flow, and downstream it will interrupt with the intercooler, causing further restrictions. This explains why the turbocharger is not a fully efficient device.

3. Methodology

A strip down and rebuild of a BorgWarner CJ5G-6K682-BA turbo, mounted as original equipment to Ford petrol engines, was conducted in the workshop. This is a water-cooled unit with a wastegate for boost control. According to BorgWarner (N.D.), their range is “extremely affordable” yet “these robust turbochargers meet the highest quality standards”. This would explain the reasoning behind Ford’s choice for such a high-volume engine.

Initially, the turbo was a complete unit still attached to the exhaust manifold and oil pipes. The full strip down procedure was documented in appendices 1-36, which will be referred to throughout this section. Appendices 1 to 4 show the turbocharger unit before disassembly.

Firstly, the compressor bypass valve was removed from the unit (see appendix 5). This is held in place with three bolts and an O ring seals the valve to the housing. This valve diverts the pressure back to prior the throttle body, when the throttle opening is reduced (EuroSport Tuning, 2015). Next, the pressure hoses were removed, which provide control pressure to the wastegate actuator diaphragm, by removing the hose clips (see appendix 6). Following this, the coolant pipes were removed (see appendix 7) paying attention to the location of the washers – one either side of the fitting (see appendix 8). The exhaust manifold was then removed from the unit by undoing the three nuts securing it to the turbine housing (see appendices 9 and 10). The oil pipes were then removed – the low-pressure pipe was attached with two torx screws with a paper gasket (see appendix 11), while the high-pressure pipe had a bolt fitting (see appendix 12). Appendix 13 shows the bolt and copper washers for the high-pressure oil pipe, similar to the coolant pipe fittings. Next, the wastegate actuator was removed by removing the two nuts securing it to the housing (see appendix 14). The actuator rod was then removed from the wastegate flap lever by removing the securing clip and undoing the end nut (see appendix 15). The compressor housing was then removed by undoing the four securing nuts (see appendix 16) but it was important to also mark the orientation so that it is reassembled in the same way, due to the packaging in the vehicle (see appendix 17). An O ring seal sits inside the housing and seals it to the surface of the spindle housing – in this case, the O ring was split (see appendix 18). Appendix 19 shows the rest of the unit with the compressor wheel exposed, alongside the compressor housing. The next step was to remove the turbine housing. This was fitted using a clamp, tightened by a bolt and nut (see appendix 20). Once the clamp is sufficiently loosened, the housing can be separated from the spindle housing – Appendices 21 and 22 show these components laid out separately. The wastegate flap and lever cannot be separated from the turbine housing post-production – the flap can be seen inside the housing in appendix 23. The spindle shaft was then removed, which together with the turbine are one piece. The compressor wheel is separate and is held on with a nut. The turbine must be held in place while this nut is loosened (see appendix 24). Care had to be taken as the nut features a left-hand thread (see appendix 25). Appendix 26 shows the spindle shaft and its housing laid out next to each other as they would be assembled, while appendix 27 shows the bearings on the spindle shaft – one of these remained in the housing when the shaft was removed. The shaft required tapping out of the housing carefully, and this left the compressor wheel loose on top of the housing. A washer was located between the compressor wheel and the housing (see appendix 28). The next stage was to remove the plate covering the internals of the housing – this was done by removing the two torx screws (see appendix 29). Appendix 30 shows this plate removed, along with the brass plate seen in more detail in appendix 31. The internal oil galleries in the housing can be seen in appendix 32. The

complete spindle shaft with all bearings and both wheels is shown in appendix 33. Appendices 34, 35 and 36 show the oil and coolant connections into the housing with more detail inside the housing.

The turbocharger unit was fully stripped at this point. Reassembly was simply a reversal of the strip down, paying attention to alignment of parts and positioning of washers and seals, as well as observing appropriate torque settings.

4. Calculations

Firstly, to determine a choice of compressor wheel, the pressure ratio (PR) and volumetric flow rate (VFR) must be calculated. The pressure ratio is defined by:

$$PR = \frac{\text{Boost Pressure}}{\text{Atmospheric Pressure}}$$

The given boost pressure for this case study is 12psi, while atmospheric pressure may be taken as 14.7psi. Therefore, the pressure ratio can be calculated as:

$$PR = \frac{14.7 + 12}{14.7}$$

$$\underline{PR = 1.816}$$

To calculate the turbocharged VFR, the figure for the naturally aspirated engine must be found first. This requires the total engine displacement (V) to be calculated. The specifications are given as a V8 engine with a bore of 95mm and a stroke of 80mm (L), a volumetric efficiency of 85% (η_v) and a maximum engine speed of 9500rpm ($N_{\max}$). The displacement (V) can be calculated by:

$$V = \pi \frac{\phi^2}{4} Ln$$

The number of cylinders is denoted by n.

$$V = \pi \frac{95^2}{4} \times 80 \times 8$$

$$\underline{V = 4.536 \times 10^{-3} m^3}$$

The VFR for the naturally aspirated engine can then be calculated based on:

$$VFR_{N/A} = \frac{V \times \eta_v \times N}{2 \times 60}$$

Using the figures above:

$$VFR_{N/A} = \frac{4.536 \times 10^{-3} \times 0.85 \times 9500}{2 \times 60}$$

$$VFR_{N/A} = 0.3053 m^3/s$$

To reach the turbocharged VFR figure, this is simply multiplied by the pressure ratio:

$$VFR_T = 0.3053 \times 1.816$$

$$\underline{VFR_T = 0.5544 m^3/s}$$

This will need to be plotted on the compressor maps supplied, however the units used on those maps are in pounds per minute. To convert from m³/s to lbs/min, a multiplier of 148.33 can be used:

$$0.5544 \times 148.33 = 82.238 \text{ lbs/min}$$

A compressor will be selected in the analysis section – to do this, the flow rate needs to be plotted against the pressure ratio on the compressor maps. The VFR above is true at 100% throttle opening. The VFR at 50% @ PR must also be plotted, along with the VFR at 20% @ PR=1. These VFR figures (in lbs/min) are:

$$0.5VFR_T = 41.119$$

$$0.2VFR_T = 16.448$$

5. Analysis

In this section, the 62-1 compressor will be referred to as compressor 1, the T3 “Super 60” as compressor 2, the T04B H-3 as compressor 3, and the last unlabelled compressor as compressor 4.

When plotting the calculated figures, it became apparent that the airflow figures for this engine were outside of the operating range of three of the compressors. These figures were calculated based on a single turbocharger feeding all eight cylinders. A twin turbo setup could be considered for this engine, with each smaller turbo feeding each bank. This was the approach taken for compressors 1 and 3, with the VFR figures halved. The figures for twin turbos were still just outside of the range of compressor 2, therefore this option was not considered at all. Compressor 4 was considered for a single turbo setup. All of the plots on the maps can be found in appendix 43.

Compressor 1 could be safely used for this engine as the plot is well within the surge limit. The compressor efficiency ranges between approximately 70-76%, with the peak at maximum load. It would run at up to around 84,000rpm.

Compressor 3 would operate at a wide range of efficiencies, between approximately 67-74%. Peak efficiency is at around 60-70% throttle, and the maximum speed would be around 92,000rpm. One area of the plot is situated on the surge limit line therefore it would not be safe to use this compressor.

Compressor 4's plot sits just within the surge limit line, making it a safe choice. The efficiency would be between 74-80%, with a peak at approximately 75% throttle. The speed would reach a maximum of approximately 52,000rpm.

Based on these plots, there are two possible choices with different setups, both being advantageous depending on the vehicle's usage. For a road application, compressor 4 would be the optimal choice. It is the most efficient compressor from the given choices, and the efficiency distribution is more suited to road driving, as throttle opening will be at around 75% much more often than wide open. It is also a simpler and cheaper installation, hence more suited to high volume production. The low compressor speed range also reduces the stress on components, improving reliability.

The other option is a twin turbo setup using compressor 1. This is more suited towards a motorsport application. Although this compressor is less efficient overall, the efficiency at maximum load is higher and this is where most race vehicles spend most of their time. The added complexity, cost and higher rpm are not problematic for a race application, as performance will be preferable, and maintenance will be at a much higher level than for a road vehicle.

It will be assumed that the vehicle will be used in a race application, therefore the selection will be a twin turbo setup using compressor 1. The ambient temperature is given as 27°C, therefore the compressor outlet air temperature can be calculated, assuming adiabatic compression. All temperatures must be converted to Kelvin for the calculations therefore the ambient temperature will be taken as 300K. The compressor efficiency is 76% at maximum load.

$$\Delta T = \frac{(PR^{0.28} \times T_{ci}) - T_{ci}}{\eta_c}$$

$$\Delta T = \frac{(1.816^{0.28} \times 300) - 300}{0.76}$$

$$\Delta T = 71.77K$$

This gives the temperature difference, which can be added to the ambient / compressor inlet temperature to give a compressor outlet temperature of 371.77K (98.77°C).

If a 60% efficient intercooler was fitted to the system, the intercooler outlet temperature can be calculated as:

$$T_{io} = -\eta_i(T_{co} - T_{ci}) + T_{co}$$

$$T_{io} = -0.6(371.77 - 300) + 371.77$$

$$T_{io} = 328.71K (55.71°C)$$

This gives a temperature difference of 43.06 degrees across the intercooler. Density recovery can be calculated based on this:

$$DR = \frac{T_{co}}{T_{io}} - 1$$

$$DR = \frac{371.77}{328.71} - 1$$

$$DR = 0.131 (13.3\%)$$

To estimate the internal flow area of a suitable intercooler, the graph in figure 44 was used. An area of 58 square inches would be required for this application. The charge face area should be 129 square inches (having factored in 45% of the face being available for flow).

To complete a turbocharger installation on the vehicle several considerations must be made other than the compressor and intercooler selections. These will depend on the type of vehicle and its usage, as well as packaging possibilities. Intercooler location and exact dimensions will be dependant on where the engine is located on the vehicle and other packaging constraints. Air to water intercoolers may be beneficial on applications where packaging is difficult to resolve. Intercooler location and dimensions will also affect the vehicle's aerodynamics, so this must be considered when designing the system. In terms of the intercooler itself, there are also some design considerations to be made, such as the type of join between the channels and face, and the use of turbulators.

Heat levels are a major concern when installing a turbocharger system, as there is much more concentrated heat in the engine bay area. This must be carefully managed to keep critical temperatures within range and prevent any component failures or premature wear due to temperatures. This is also related to space within the engine bay.

In terms of the engine, all of the involved components must be able to withstand the heat and pressures associated with the turbocharger. The fuel pump must be able to supply sufficient pressure, and the fuel injectors may have to be replaced to ensure there is enough fuel reaching the cylinders. A suitable control unit will also need to be fitted to the engine and fully calibrated for the application. The turbocharger unit also requires lubrication – an oil feed must be fitted from the engine which supplies sufficient pressure, and a return line must be fitted to the sump.

For motorsport applications, regulations must also be considered. Some series may not allow forced induction systems, or may have specific restrictions or engine capacity multipliers.

6. Conclusion

To conclude, the aims of the report have been successfully met, as turbocharging systems have been explained along with modern technology adaptations. A compressor wheel has been selected for the case study with justifications and basic intercooler dimensions have been estimated along with temperature calculations.

These calculations are only based on a limited amount of provided data and only serve as estimations. In reality, this engine would need to be tested to validate all these figures and to make any necessary adjustments to the system. The design and component selection of a turbocharging system becomes much more complex when actually faced with a real vehicle as there are many factors to consider which affect one another. A turbocharger installation will interfere with a number of systems on a vehicle and all of these will need to be adjusted and/or modified in order to accommodate the installation of a turbocharger.

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8. Appendices

Appendix 1 – BorgWarner Turbocharger prior to strip down (Kot, 2019)



Appendix 2 – BorgWarner Turbocharger prior to strip down (Kot, 2019)



Appendix 3 – BorgWarner Turbocharger prior to strip down (Kot, 2019)



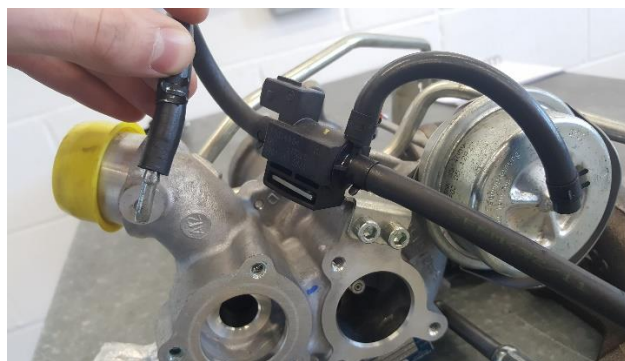
Appendix 4 – BorgWarner Turbocharger prior to strip down (Kot, 2019)



Appendix 5 – Compressor bypass valve removal (Kot, 2019)



Appendix 6 – Pressure hoses attaching to wastegate diaphragm (Kot, 2019)



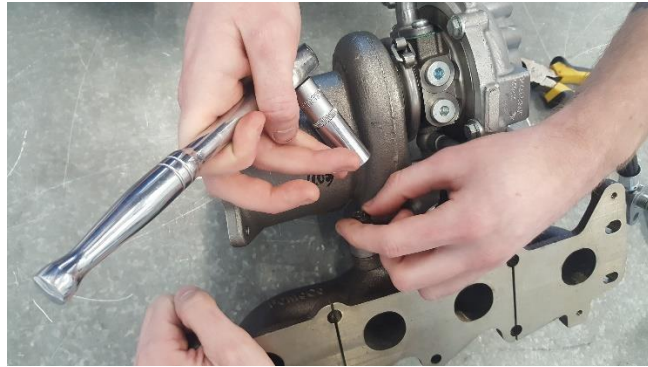
Appendix 7 – Coolant pipe location (Kot, 2019)



Appendix 8 – Coolant pipe bolt and copper washers layout (Kot, 2019)



Appendix 9 – Manifold removal from the turbocharger (Kot, 2019)



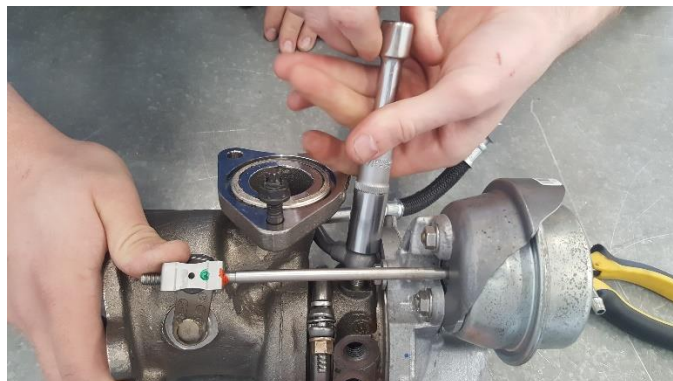
Appendix 10 – Studs and nuts securing the manifold to the turbocharger (Kot, 2019)



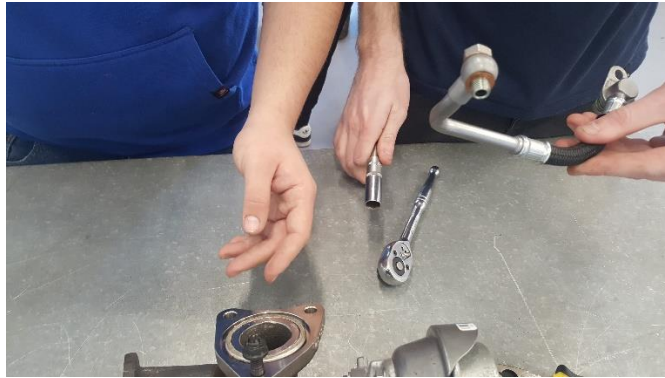
Appendix 11 – Low pressure oil pipe removal (Kot, 2019)



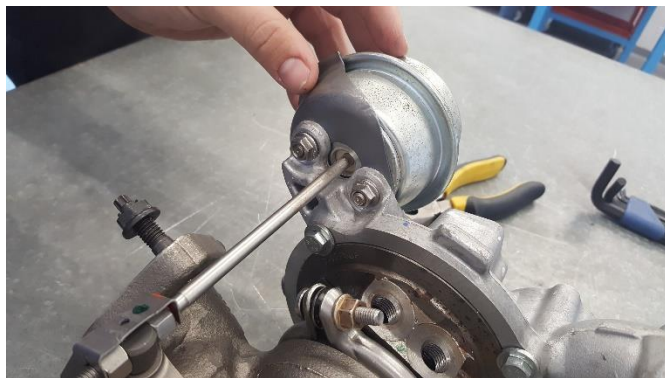
Appendix 12 – High pressure oil pipe removal (Kot, 2019)



Appendix 13 – High pressure oil pipe removal and bolt/washers layout (Kot, 2019)



Appendix 14 – Wastegate actuator removal (Kot, 2019)



Appendix 15 – Wastegate actuator rod removal from lever (Kot, 2019)



Appendix 16 – Compressor housing removal (Kot, 2019)



Appendix 17 – Marking the orientation of the compressor housing (Kot, 2019)



Appendix 18 – Compressor housing removal (Kot, 2019)



Appendix 19 – Compressor housing removal (Kot, 2019)



Appendix 20 – Turbine housing removal (Kot, 2019)



Appendix 21 – Turbine housing removal (Kot, 2019)



Appendix 22 – Turbine housing removal (Kot, 2019)



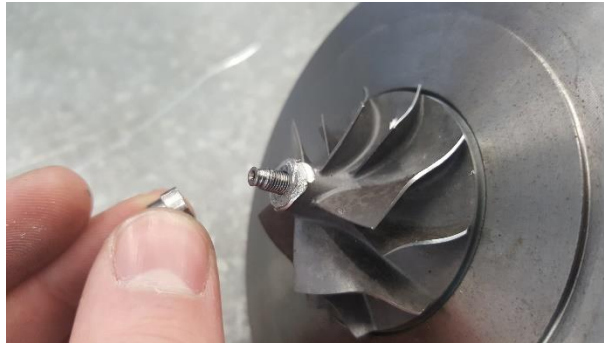
Appendix 23 – Wastegate flap (Kot, 2019)



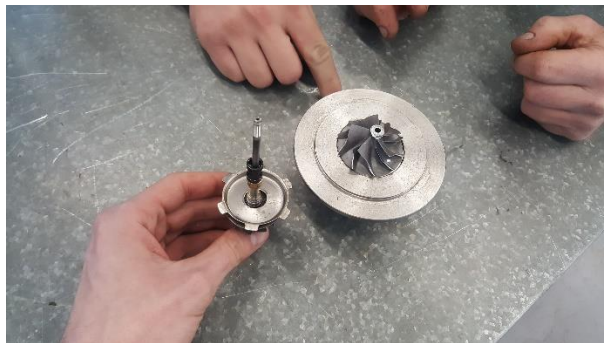
Appendix 24 – Spindle shaft removal (Kot, 2019)



Appendix 25 – Left hand thread on spindle shaft (Kot, 2019)



Appendix 26 – Spindle shaft removal (Kot, 2019)



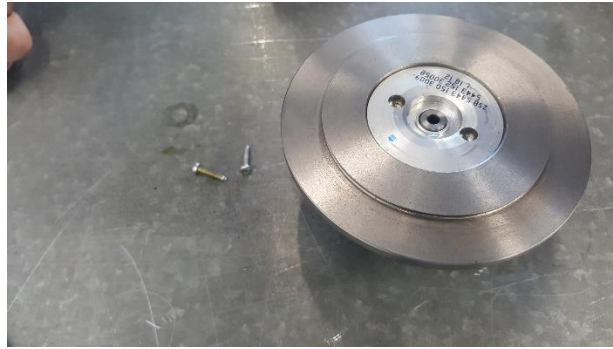
Appendix 27 – Spindle shaft bearings (Kot, 2019)



Appendix 28 – Washer under compressor wheel (Kot, 2019)



Appendix 29 – Cover plate removal (Kot, 2019)



Appendix 30 – Cover plate removal (Kot, 2019)



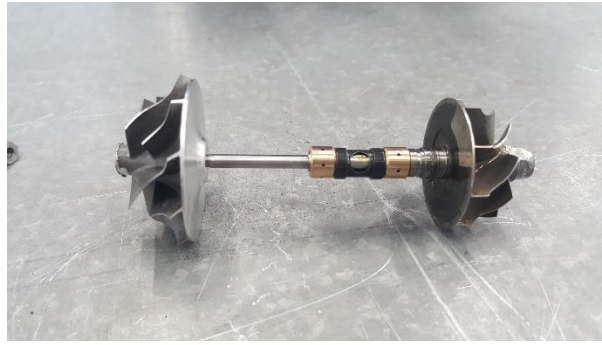
Appendix 31 – Brass element (Kot, 2019)



Appendix 32 – Oil galleries within the housing (Kot, 2019)



Appendix 33 – Spindle shaft assembly (Kot, 2019)



Appendix 34 – Low pressure oil return pipe fitting on housing (Kot, 2019)



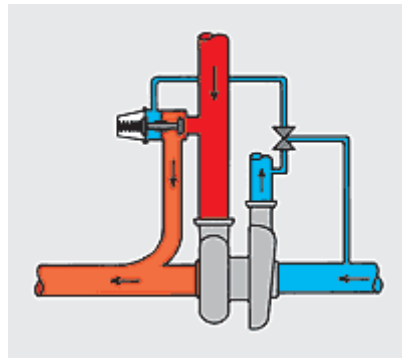
Appendix 35 – High pressure oil pipe fitting on housing (Kot, 2019)



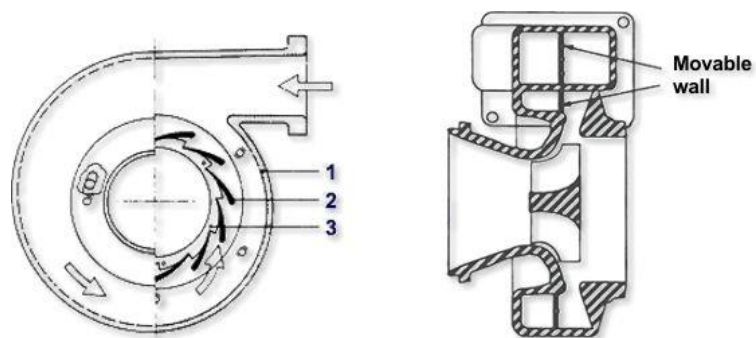
Appendix 36 – Coolant pipe fittings on housing (Kot, 2019)



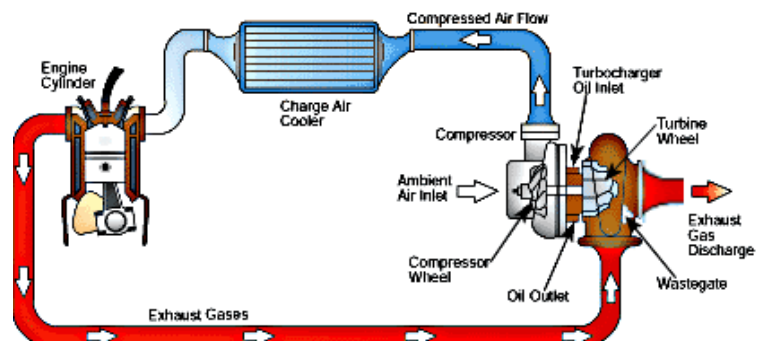
Appendix 37 – Flow to the wastegate actuator (BorgWarner, N.D.)



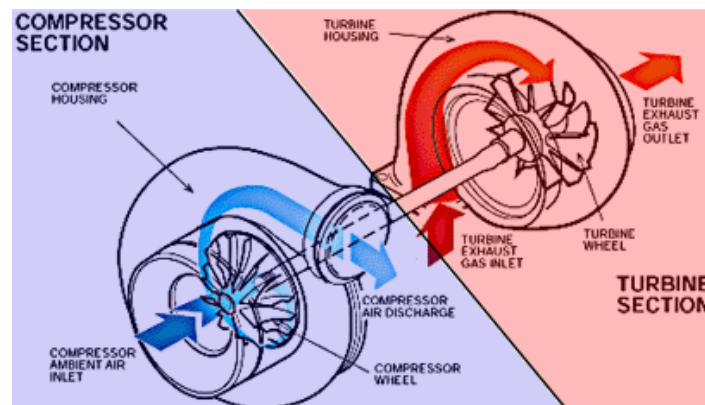
Appendix 38 – Variable geometry turbo diagram (Jääskeläinen, 2016)



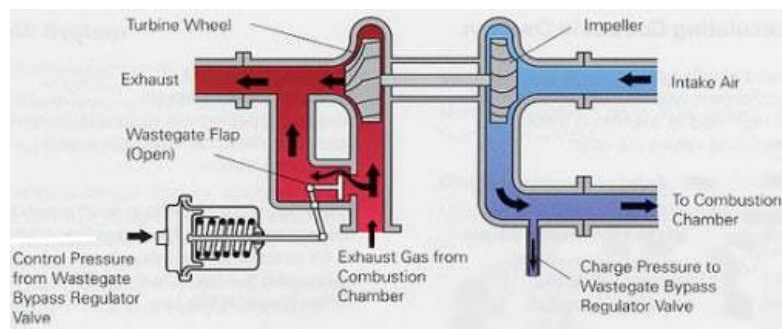
Appendix 39 – Flow through a turbocharged system (Formula 1 Dictionary, N.D.)



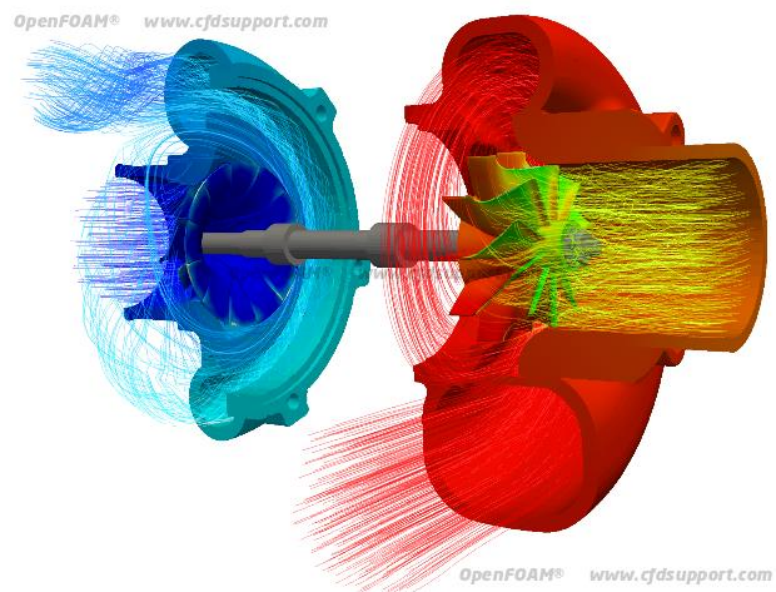
Appendix 40 – Flow through a turbocharger unit (Formula 1 Dictionary, N.D.)



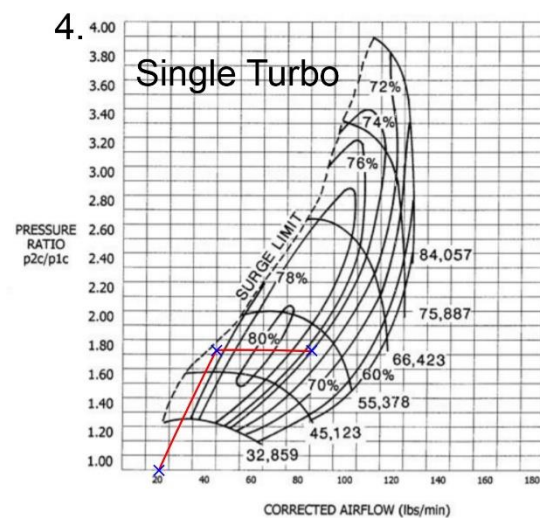
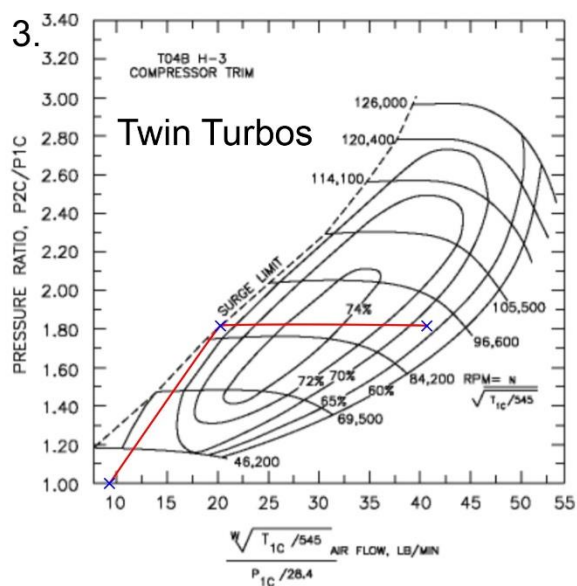
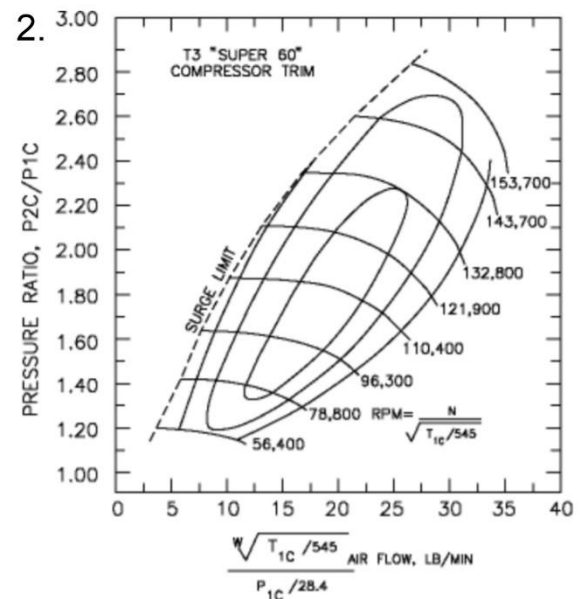
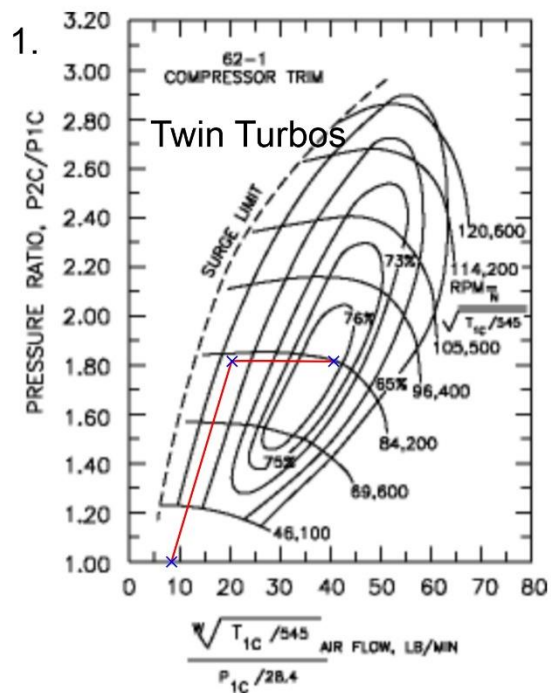
Appendix 41 – Flow through a wastegate (Formula 1 Dictionary, N.D.)



Appendix 42 – Computational Fluid Dynamics study of a turbocharger (CFD Support, N.D.)



Appendix 43 – Compressor map plots for case study (maps from Turbonetics, N.D.)



Appendix 44 – Intercooler sizing (graph from Bell, 1997)

