**Turbo Tech 101 (Basic)**

**How a Turbo System Works**

Engine power is proportional to the amount of air and fuel that can get into the cylinders. All things being equal, larger engines flow more air and as such will produce more power. If we want our small engine to perform like a big engine, or simply make our bigger engine produce more power, our ultimate objective is to draw more air into the cylinder. By installing a Garrett turbocharger, the power and performance of an engine can be dramatically increased.

So how does a turbocharger get more air into the engine? Let us first look at the schematic below:

![Turbocharger Schematic](image)

The components that make up a typical turbocharger system are:

- The air filter (not shown) through which ambient air passes before entering the compressor (1)
- The air is then compressed which raises the air’s density (mass / unit volume) (2)
- Many turbocharged engines have a charge air cooler (aka intercooler) (3) that cools the compressed air to further increase its density and to increase resistance to detonation
- After passing through the intake manifold (4), the air enters the engine’s cylinders, which contain a fixed volume. Since the air is at elevated density, each cylinder can draw in an increased mass flow rate of air. Higher air mass flow rate allows a higher fuel flow rate (with similar air/fuel ratio). Combusting more fuel results in more power being produced for a given size or displacement.
After the fuel is burned in the cylinder it is exhausted during the cylinder’s exhaust stroke into the exhaust manifold (5).

The high temperature gas then continues on to the turbine (6). The turbine creates backpressure on the engine which means engine exhaust pressure is higher than atmospheric pressure.

A pressure and temperature drop occurs (expansion) across the turbine (7), which harnesses the exhaust gas’ energy to provide the power necessary to drive the compressor.

**What are the components of a turbocharger?**

The layout of the turbocharger in a given application is critical to a properly performing system. Intake and exhaust plumbing is often driven primarily by packaging constraints. We will explore exhaust manifolds in more detail in subsequent tutorials; however, it is important to understand the need for a compressor bypass valve (commonly referred to as a Blow-Off valve) on the intake tract and a Wastegates for the exhaust flow.

**Other Components**

**Blow-Off (Bypass) Valves**

The Blow-Off valve (BOV) is a pressure relief device on the intake tract to prevent the turbo’s compressor from going into surge. The BOV should be installed between the compressor discharge and the throttle body, preferably downstream of the charge air cooler (if equipped). When the throttle is closed rapidly, the airflow is quickly reduced, causing flow instability and pressure fluctuations. These rapidly cycling pressure fluctuations are the audible evidence of surge. Surge can eventually lead to thrust bearing failure due to the high loads associated with it.

Blow-Off valves use a combination of manifold pressure signal and spring force to detect when the throttle is closed. When the throttle is closed rapidly, the BOV vents boost in the intake tract to atmosphere to relieve the pressure; helping to eliminate the phenomenon of surge.
**Wastegates**

On the exhaust side, a Wastegates provides us a means to control the boost pressure of the engine. Some commercial diesel applications do not use a Wastegates at all. This type of system is called a free-floating turbocharger.

However, the vast majority of gasoline performance applications require a Wastegates. There are two (2) configurations of Wastegates, **internal** or **external**. Both internal and external Wastegates provide a means to bypass exhaust flow from the turbine wheel. Bypassing this energy (e.g. exhaust flow) reduces the power driving the turbine wheel to match the power required for a given boost level. Similar to the BOV, the Wastegates uses boost pressure and spring force to regulate the flow bypassing the turbine.

**Internal** Wastegates are built into the turbine housing and consist of a “flapper” valve, crank arm, rod end, and pneumatic actuator. It is important to connect this actuator only to boost pressure; i.e. it is not designed to handle vacuum and as such should not be referenced to an intake manifold.

**External** Wastegates are added to the exhaust plumbing on the exhaust manifold or header. The advantage of external Wastegates is that the bypassed flow can be reintroduced into the exhaust stream further downstream of the turbine. This tends to improve the turbine’s performance. On racing applications, this Wastegated exhaust flow can be vented directly to atmosphere.
Oil & Water Plumbing

The intake and exhaust plumbing often receives the focus leaving the oil and water plumbing neglected.

Garrett ball bearing turbochargers require less oil than journal bearing turbos. Therefore an oil inlet restrictor is recommended if you have oil pressure over about 60 psig. The oil outlet should be plumbed to the oil pan above the oil level (for wet sump systems). Since the oil drain is gravity fed, it is important that the oil outlet points downward, and that the drain tube does not become horizontal or go “uphill” at any point.

Following a hot shutdown of a turbocharger, heat soak begins. This means that the heat in the head, exhaust manifold, and turbine housing finds its way to the turbo’s center housing, raising its temperature. These extreme temperatures in the center housing can result in oil coking.

To minimize the effects of heat soak-back, water-cooled center housings were introduced. These use coolant from the engine to act as a heat sink after engine shutdown, preventing the oil from coking. The water lines utilize a thermal siphon effect to reduce the peak heat soak-back temperature after key-off. The layout of the pipes should minimize peaks and troughs with the (cool) water inlet on the low side. To help this along, it is advantageous to tilt the turbocharger about 25° about the axis of shaft rotation.

Many Garrett turbos are water-cooled for enhanced durability.

Which Turbocharger is Right for Me or more affectionately known as My Turbo & Me

Selecting the proper turbocharger for your specific application requires many inputs. With decades of collective turbocharging experience, the Garrett Performance Distributors can assist in selecting the right turbocharger for your application.

The primary input in determining which turbocharger is appropriate is to have a target horsepower in mind. This should be as realistic as possible for the application. Remember that engine power is generally proportional to air and fuel flow. Thus, once you have a target power level identified, you begin to hone in on the turbocharger size, which is highly dependent on airflow requirements.

Other important factors include the type of application. An autocross car, for example, requires rapid boost response. A smaller turbocharger or smaller turbine housing would be most suitable for this application. While this will trade off ultimate power due to
increased exhaust backpressure at higher engine speeds, boost response of the small turbo will be excellent.

Alternatively, on a car dedicated to track days, peak horsepower is a higher priority than low-end torque. Plus, engine speeds tend to be consistently higher. Here, a larger turbocharger or turbine housing will provide reduced backpressure but less-immediate low-end response. This is a welcome tradeoff given the intended operating conditions.

Selecting the turbocharger for your application goes beyond “how much boost” you want to run. Defining your target power level and the primary use for the application are the first steps in enabling your Garrett Performance Distributor to select the right turbocharger for you.

**Journal Bearings vs. Ball Bearings**
The journal bearing has long been the brawn of the turbocharger, however a ball-bearing cartridge is now an affordable technology advancement that provides significant performance improvements to the turbocharger.

Ball bearing innovation began as a result of work with the Garrett Motorsports group for several racing series where it received the term the ‘cartridge ball bearing’. The cartridge is a single sleeve system that contains a set of angular contact ball bearings on either end, whereas the traditional bearing system contains a set of journal bearings and a thrust bearing.

Turbo Response – When driving a vehicle with the cartridge ball bearing turbocharger, you will find exceptionally crisp and strong throttle response. Garrett Ball Bearing turbochargers spool up 15% faster than traditional journal bearings. This produces an improved response that can be converted to quicker 0-60 mph speed. In fact, some professional drivers of Garrett ball-bearing turbocharged engines report that they feel like they are driving a big, normally aspirated engine.

Tests run on CART turbos have shown that ball-bearings have up to half of the power consumption of traditional bearings. The result is faster time to boost which translates into better drivability and acceleration.

On-engine performance is also better in the steady-state for the Garrett Cartridge Ball Bearing.
Reduced Oil Flow – The ball bearing design reduces the required amount of oil required to provide adequate lubrication. This lower oil volume reduces the chance for seal leakage. Also, the ball bearing is more tolerant of marginal lube conditions, and diminishes the possibility of turbocharger failure on engine shut down.

Improved Rotordynamics and Durability – The ball bearing cartridge gives better damping and control over shaft motion, allowing enhanced reliability for both everyday and extreme driving conditions. In addition, the opposed angular contact bearing cartridge eliminates the need for the thrust bearing commonly a weak link in the turbo bearing system.

Competitor Ball Bearing Options – Another option one will find is a hybrid ball bearing. This consists of replacing only the compressor side journal bearing with a single angular contact ball bearing. Since the single bearing can only take thrust in one direction, a thrust bearing is still necessary and drag in the turbine side journal bearing is unchanged. With the Garrett ball bearing cartridge the rotor-group is entirely supported by the ball bearings, maximizing efficiency, performance, and durability.

Ball Bearings in Original Equipment – Pumping up the MAZDASPEED Protegé’s heart rate is a Garrett T25 turbocharger system. With Garrett technology on board, the vehicle gains increased acceleration without sacrificing overall efficiency and it has received many rave reviews from the world’s top automotive press for it’s unprecedented performance.
**Turbo Systems 102 (Advanced)**

Please thoroughly review and have a good understanding of Turbo Systems 101 - Basic prior to reading this section. The following areas will be covered in the Turbo System 102 - Advanced section:

1. Wheel trim topic coverage
2. Understanding turbine housing A/R and housing sizing
3. Different types of manifolds (advantages/disadvantages log style vs. equal length)
4. Compression ratio with boost
5. Air/Fuel Ratio tuning: Rich v. Lean, why lean makes more power but is more dangerous

**1. Wheel trim topic coverage**

Trim is a common term used when talking about or describing turbochargers. For example, you may hear someone say "I have a GT2871R '56 Trim' turbocharger. What is 'Trim?' Trim is a term to express the relationship between the inducer* and exducer* of both turbine and compressor wheels. More accurately, it is an area ratio.

* The inducer diameter is defined as the diameter where the air enters the wheel, whereas the exducer diameter is defined as the diameter where the air exits the wheel.

Based on aerodynamics and air entry paths, the inducer for a compressor wheel is the smaller diameter. For turbine wheels, the inducer is the larger diameter (see Figure 1.)

![Figure 1. Illustration of the inducer and exducer diameter of compressor and turbine wheels](image)

**Example #1:** GT2871R turbocharger (Garrett part number 743347-2) has a compressor wheel with the below dimensions. What is the trim of the compressor wheel?

* Inducer diameter = 53.1mm
* Exducer diameter = 71.0mm

\[
\text{Trim} = \left( \frac{\text{inducer}}{\text{exducer}} \right)^2 \times 100
\]

\[
\text{Trim} = \left( \frac{53.1}{71.0} \right)^2 \times 100
\]

\[
\text{Trim} = 56
\]

**Example #2:** GT2871R turbocharger (part # 743347-1) has a compressor wheel with an exducer diameter of 71.0mm and a trim of 48. What is the inducer diameter of the compressor wheel?

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*The inducer diameter is defined as the diameter where the air enters the wheel, whereas the exducer diameter is defined as the diameter where the air exits the wheel.*
The trim of a wheel, whether compressor or turbine, affects performance by shifting the airflow capacity. All other factors held constant, a higher trim wheel will flow more than a smaller trim wheel.

However, it is important to note that very often all other factors are not held constant. So just because a wheel is a larger trim does not necessarily mean that it will flow more.

2. Understanding housing sizing: A/R

A/R (Area/Radius) describes a geometric characteristic of all compressor and turbine housings. Technically, it is defined as:

\[
\text{A/R} = \frac{\text{Area}}{\text{Radius}}
\]

which is the inlet (or, for compressor housings, the discharge) cross-sectional area divided by the radius from the turbo centerline to the centroid of that area (see Figure 2.).
The A/R parameter has different effects on the compressor and turbine performance, as outlined below.

**Compressor A/R** - Compressor performance is comparatively insensitive to changes in A/R. Larger A/R housings are sometimes used to optimize performance of low boost applications, and smaller A/R are used for high boost applications. However, as this influence of A/R on compressor performance is minor, there are not A/R options available for compressor housings.

**Turbine A/R** - Turbine performance is greatly affected by changing the A/R of the housing, as it is used to adjust the flow capacity of the turbine. Using a smaller A/R will increase the exhaust gas velocity into the turbine wheel. This provides increased turbine power at lower engine speeds, resulting in a quicker boost rise. However, a small A/R also causes the flow to enter the wheel more tangentially, which reduces the ultimate flow capacity of the turbine wheel. This will tend to increase exhaust backpressure and hence reduce the engine's ability to "breathe" effectively at high RPM, adversely affecting peak engine power.

Conversely, using a larger A/R will lower exhaust gas velocity, and delay boost rise. The flow in a larger A/R housing enters the wheel in a more radial fashion, increasing the wheel's effective flow capacity, resulting in lower backpressure and better power at higher engine speeds.

When deciding between A/R options, be realistic with the intended vehicle use and use the A/R to bias the performance toward the desired powerband characteristic.

Here's a simplistic look at comparing turbine housing geometry with different applications. By comparing different turbine housing A/R, it is often possible to determine the intended use of the system.

Imagine two 3.5L engines both using GT30R turbochargers. The only difference between the two engines is a different turbine housing A/R; otherwise the two engines are identical:
1. Engine #1 has turbine housing with an A/R of 0.63
2. Engine #2 has a turbine housing with an A/R of 1.06.

What can we infer about the intended use and the turbocharger matching for each engine?

**Engine #1:** This engine is using a smaller A/R turbine housing (0.63) thus biased more towards low-end torque and optimal boost response. Many would describe this as being more "fun" to drive on the street, as normal daily driving habits tend to favor transient response. However, at higher engine speeds, this smaller A/R housing will result in high backpressure, which can result in a loss of top end power. This type of engine performance is desirable for street applications where the low speed boost response and transient conditions are more important than top end power.

**Engine #2:** This engine is using a larger A/R turbine housing (1.06) and is biased towards peak horsepower, while sacrificing transient response and torque at very low engine speeds. The larger A/R turbine housing will continue to minimize backpressure at high rpm, to the benefit of engine peak power. On the other hand, this will also raise the engine speed at which the turbo can provide boost, increasing time to boost. The performance of Engine #2 is more desirable for racing applications than Engine #1 where the engine will be operating at high engine speeds most of the time.

3. **Different types of manifolds (advantages/disadvantages log style vs. equal length)**

There are two different types of turbocharger manifolds; cast log style (see Figure 3.) and welded tubular style (see Figure 4.).

![Cast log style turbocharger manifold](image)
Manifold design on turbocharged applications is deceptively complex as there many factors to take into account and trade off.

General design tips for best overall performance are to:

- Maximize the radius of the bends that make up the exhaust primaries to maintain pulse energy
- Make the exhaust primaries equal length to balance exhaust reversion across all cylinders
- Avoid rapid area changes to maintain pulse energy to the turbine
- At the collector, introduce flow from all runners at a narrow angle to minimize "turning" of the flow in the collector
- For better boost response, minimize the exhaust volume between the exhaust ports and the turbine inlet
- For best power, tuned primary lengths can be used

Cast manifolds are commonly found on OEM applications, whereas welded tubular manifolds are found almost exclusively on aftermarket and race applications. Both manifold types have their advantages and disadvantages. Cast manifolds are generally very durable and are usually dedicated to one application. They require special tooling for the casting and machining of specific features on the manifold. This tooling can be expensive.

On the other hand, welded tubular manifolds can be custom-made for a specific application without special tooling requirements. The manufacturer typically cuts pre-bent steel U-bends into the desired geometry and then welds all of the components together. Welded tubular manifolds are a very effective solution. One item of note is durability of this design. Because of the welded joints, thinner wall sections, and reduced stiffness, these types of manifolds are often susceptible to cracking due to thermal expansion/contraction and vibration. Properly constructed tubular manifolds can last a long time, however. In addition, tubular manifolds can offer a substantial performance advantage over a log-type manifold.

A design feature that can be common to both manifold types is a "DIVIDED MANIFOLD", typically employed with "DIVIDED" or "twin-scroll" turbine housings. Divided exhaust manifolds can be incorporated into either a cast or welded tubular manifolds (see Figure 5. and Figure 6.).
The concept is to DIVIDE or separate the cylinders whose cycles interfere with one another to best utilize the engine's exhaust pulse energy.

For example, on a four-cylinder engine with firing order 1-3-4-2, cylinder #1 is ending its expansion stroke and opening its exhaust valve while cylinder #2 still has its exhaust valve open (cylinder #2 is in its overlap period). In an undivided exhaust manifold, this pressure pulse from cylinder #1's exhaust blowdown event is much more likely to contaminate cylinder #2 with high pressure exhaust gas. Not only does this hurt cylinder #2's ability to breathe properly, but this pulse energy would have been better utilized in the turbine.

The proper grouping for this engine is to keep complementary cylinders grouped together-- #1 and #4 are complementary; as are cylinders #2 and #3.

Because of the better utilization of the exhaust pulse energy, the turbine's performance is improved and boost increases more quickly.

**4. Compression ratio with boost**

Before discussing compression ratio and boost, it is important to understand engine knock, also known as detonation. Knock is a dangerous condition caused by uncontrolled combustion of the air/fuel mixture. This abnormal combustion causes rapid spikes in cylinder pressure which can result in engine damage.
Three primary factors that influence engine knock are:

1. **Knock resistance characteristics (knock limit) of the engine:** Since every engine is vastly different when it comes to knock resistance, there is no single answer to "how much." Design features such as combustion chamber geometry, spark plug location, bore size and compression ratio all affect the knock characteristics of an engine.

2. **Ambient air conditions:** For the turbocharger application, both ambient air conditions and engine inlet conditions affect maximum boost. Hot air and high cylinder pressure increases the tendency of an engine to knock. When an engine is boosted, the intake air temperature increases, thus increasing the tendency to knock. Charge air cooling (e.g. an intercooler) addresses this concern by cooling the compressed air produced by the turbocharger.

3. **Octane rating of the fuel being used:** octane is a measure of a fuel's ability to resist knock. The octane rating for pump gas ranges from 85 to 94, while racing fuel would be well above 100. The higher the octane rating of the fuel, the more resistant to knock. Since knock can be damaging to an engine, it is important to use fuel of sufficient octane for the application. Generally speaking, the more boost run, the higher the octane requirement.

This cannot be overstated: engine calibration of fuel and spark plays an enormous role in dictating knock behavior of an engine. See Section 5 below for more details.

Now that we have introduced knock/detonation, contributing factors and ways to decrease the likelihood of detonation, let's talk about compression ratio. Compression ratio is defined as:

\[
\text{Compression Ratio} = \frac{\text{displacement volume} + \text{clearance volume}}{\text{clearance volume}}
\]

or

\[
CR = \frac{V_d + V_c}{V_c}
\]

where

- CR = compression ratio
- \(V_d\) = displacement volume
- \(V_c\) = clearance volume

Geometry of cylinder, piston, connecting rod, and crankshaft where \(B = \) bore and \(L = \) stroke.
The compression ratio from the factory will be different for naturally aspirated engines and boosted engines. For example, a stock Honda S2000 has a compression ratio of 11.1:1, whereas a turbocharged Subaru Impreza WRX has a compression ratio of 8.0:1.

There are numerous factors that affect the maximum allowable compression ratio. There is no single correct answer for every application. Generally, compression ratio should be set as high as feasible without encountering detonation at the maximum load condition. Compression ratio that is too low will result in an engine that is a bit sluggish in off-boost operation. However, if it is too high this can lead to serious knock-related engine problems.

Factors that influence the compression ratio include: fuel anti-knock properties (octane rating), boost pressure, intake air temperature, combustion chamber design, ignition timing, valve events, and exhaust backpressure. Many modern normally-aspirated engines have well-designed combustion chambers that, with appropriate tuning, will allow modest boost levels with no change to compression ratio. For higher power targets with more boost, compression ratio should be adjusted to compensate.

There are a handful of ways to reduce compression ratio, some better than others. Least desirable is adding a spacer between the block and the head. These spacers reduce the amount a "quench" designed into an engine's combustion chambers, and can alter cam timing as well. Spacers are, however, relatively simple and inexpensive.

A better option, if more expensive and time-consuming to install, is to use lower-compression pistons. These will have no adverse effects on cam timing or the head's ability to seal, and allow proper quench regions in the combustion chambers.

5. Air/Fuel Ratio tuning: Rich v. Lean, why lean makes more power but is more dangerous

When discussing engine tuning the 'Air/Fuel Ratio' (AFR) is one of the main topics. Proper AFR calibration is critical to performance and durability of the engine and it's components. The AFR defines the ratio of the amount of air consumed by the engine compared to the amount of fuel.

A 'Stoichiometric' AFR has the correct amount of air and fuel to produce a chemically complete combustion event. For gasoline engines, the stoichiometric, A/F ratio is 14.7:1, which means 14.7 parts of air to one part of fuel. The stoichiometric AFR depends on fuel type-- for alcohol it is 6.4:1 and 14.5:1 for diesel.

So what is meant by a rich or lean AFR? A lower AFR number contains less air than the 14.7:1 stoichiometric AFR, therefore it is a richer mixture. Conversely, a higher AFR number contains more air and therefore it is a leaner mixture.

For Example:
15.0:1 = Lean
14.7:1 = Stoichiometric
13.0:1 = Rich

Leaner AFR results in higher temperatures as the mixture is combusted. Generally, normally-aspirated spark-ignition (SI) gasoline engines produce maximum power just slightly rich of stoichiometric. However, in practice it is kept between 12:1 and 13:1 in order to keep exhaust gas temperatures in check and to account for variances in fuel quality. This is a realistic full-load AFR on a normally-aspirated engine but can be dangerously lean with a highly-boosted engine.

Let's take a closer look. As the air-fuel mixture is ignited by the spark plug, a flame front propagates from the spark plug. The now-burning mixture raises the cylinder pressure and temperature, peaking at some point in the combustion process.

The turbocharger increases the density of the air resulting in a denser mixture. The denser mixture raises the peak cylinder pressure, therefore increasing the probability of knock. As the AFR is leaned out, the temperature of the burning gases increases, which also increases the probability of knock. This is why it is imperative to run richer AFR on a boosted engine at full load. Doing so will reduce the likelihood of knock, and will also keep temperatures under control.

There are actually three ways to reduce the probability of knock at full load on a turbocharged engine: reduce boost, adjust the AFR to richer mixture, and retard ignition timing. These three parameters need to be optimized together to yield the highest reliable power.

For further in-depth calculations of pressure ratio, mass flow, and turbocharger selection, please read Turbo Systems 103 Expert tutorial.
Turbo Tech 103 (Expert)

This article is a bit more involved and will describe parts of the compressor map, how to estimate pressure ratio and mass flow rate for your engine, and how to plot the points on the maps to help choose the right turbocharger. Have your calculator handy!!

1 Parts of the Compressor Map:
◊ The compressor map is a graph that describes a particular compressor’s performance characteristics, including efficiency, mass flow range, boost pressure capability, and turbo speed. Shown below is a figure that identifies aspects of a typical compressor map:

◊ Pressure Ratio
• Pressure Ratio \( \Pi_c \) is defined as the Absolute outlet pressure divided by the Absolute inlet pressure.

\[
\Pi_c = \frac{P_{2c}}{P_{1c}}
\]

Where:
- \( \Pi_c \) = Pressure Ratio
- \( P_{2c} \) = Compressor Discharge Pressure
- \( P_{1c} \) = Compressor Inlet Pressure

◊ It is important to use units of Absolute Pressure for both \( P_{1c} \) and \( P_{2c} \). Remember that Absolute Pressure at sea level is 14.7 psia (in units of psia, the a refers to “absolute”). This is referred to as standard atmospheric pressure at standard conditions.

• Gauge Pressure (in units of psig, the g refers to “gauge”) measures the pressure above atmospheric, so a gauge pressure reading at atmospheric conditions will read zero. Boost gauges measure the manifold pressure relative to atmospheric pressure, and thus are measuring Gauge Pressure. This is important when determining \( P_{2c} \). For example, a reading of 12 psig on a boost gauge means that the air pressure in the manifold is 12 psi above atmospheric pressure. For a day at standard atmospheric conditions,
12 psig + 14.7 psia = 26.7 psi absolute pressure in the manifold

- The pressure ratio at this condition can now be calculated:

\[
\frac{26.7 \text{ psia}}{14.7 \text{ psia}} = 1.82
\]

- However, this assumes there is no adverse impact of the air filter assembly at the compressor inlet.
- In determining pressure ratio, the absolute pressure at the compressor inlet (P\text{2c}) is often LESS than the ambient pressure, especially at high load. Why is this? Any restriction (caused by the air filter or restrictive ducting) will result in a “depression,” or pressure loss, upstream of the compressor that needs to be accounted for when determining pressure ratio. This depression can be 1 psig or more on some intake systems. In this case P\text{1c} on a standard day is:

\[
14.7 \text{ psia} - 1 \text{ psig} = 13.7 \text{ psia at compressor inlet}
\]

- Taking into account the 1 psig intake depression, the pressure ratio is now:

\[
\frac{(12 \text{ psig} + 14.7 \text{ psia})}{13.7 \text{ psia}} = 1.95.
\]

- That’s great, but what if you’re not at sea level? In this case, simply substitute the actual atmospheric pressure in place of the 14.7 psi in the equations above to give a more accurate calculation. At higher elevations, this can have a significant effect on pressure ratio.

For example, at Denver’s 5000 feet elevation, the atmospheric pressure is typically around 12.4 psia. In this case, the pressure ratio calculation, taking into account the intake depression, is:

\[
\frac{(12 \text{ psig} + 12.4 \text{ psia})}{(12.4 \text{ psia} - 1 \text{ psig})} = 2.14.
\]

Compared to the 1.82 pressure ratio calculated originally, this is a big difference.

- As you can see in the above examples, pressure ratio depends on a lot more than just boost.

◊ **Mass Flow Rate**

- Mass Flow Rate is the mass of air flowing through a compressor (and engine!) over a given period of time and is commonly expressed as lb/min (pounds per minute). Mass flow can be physically measured, but in many cases it is sufficient to estimate the mass flow for choosing the proper turbo.
- Many people use Volumetric Flow Rate (expressed in cubic feet per minute, CFM or ft\(^3\)/min) instead of mass flow rate. Volumetric flow rate can be converted to mass flow by multiplying by the air density. Air density at sea level is 0.076 lb/ft\(^3\).
- What is my mass flow rate? As a very general rule, turbocharged gasoline engines will generate 9.5-10.5 horsepower (as measured at the flywheel) for each lb/min of airflow. So, an engine with a target peak horsepower of 400 Hp will require 36-44 lb/min of airflow to achieve that target. This is just a rough first approximation to help narrow the turbo selection options.
Surge Line

- **Surge** is the left hand boundary of the compressor map. Operation to the left of this line represents a region of flow instability. This region is characterized by mild flutter to wildly fluctuating boost and "barking" from the compressor. Continued operation within this region can lead to premature turbo failure due to heavy thrust loading.

- Surge is most commonly experienced when one of two situations exist. The first and most damaging is surge under load. It can be an indication that your compressor is too large. Surge is also commonly experienced when the throttle is quickly closed after boosting. This occurs because mass flow is drastically reduced as the throttle is closed, but the turbo is still spinning and generating boost. This immediately drives the operating point to the far left of the compressor map, right into surge.

Surge will decay once the turbo speed finally slows enough to reduce the boost and move the operating point back into the stable region. This situation is commonly addressed by using a Blow-Off Valves (BOV) or bypass valve. A BOV functions to vent intake pressure to atmosphere so that the mass flow ramps down smoothly, keeping the compressor out of surge. In the case of a recirculating bypass valve, the airflow is recirculated back to the compressor inlet.

- **A Ported Shroud compressor (see Fig. 2)** is a feature that is incorporated into the compressor housing. It functions to move the surge line further to the left (see Fig. 3) by allowing some airflow to exit the wheel through the port to keep surge from occurring. This provides additional useable range and allows a larger compressor to be used for higher flow requirements without risking running the compressor into a dangerous surge condition. The presence of the ported shroud usually has a minor negative impact on compressor efficiency.

The Choke Line

- **The Choke Line** is the right hand boundary of the compressor map. For Garrett maps, the choke line is typically defined by the point where the efficiency drops below 58%. In addition to the rapid drop of compressor efficiency past this point, the turbo speed will also be approaching or exceeding the allowable limit. If your actual or predicted operation is beyond this limit, a larger compressor is necessary.

- **Turbo Speed Lines** are lines of constant turbo speed. Turbo speed for points between these lines can be estimated by interpolation. As turbo speed increases, the pressure ratio increases and/or mass flow increases. As indicated above in the choke line description, the turbo speed lines are very close together at the far right edge of the map. Once a compressor is operating past the choke limit, turbo speed increases very quickly and a turbo over-speed condition is very likely.

- **Efficiency Islands** are concentric regions on the maps that represent the compressor efficiency at any point on the map. The smallest island near the center of the map is the highest or peak efficiency island. As the rings move out from there, the efficiency drops by the indicated amount until the surge and choke limits are reached.

2. Plotting Your Data on the Compressor Map

In this section, methods to calculate mass flow rate and boost pressure required to meet a horsepower target are
presented. This data will then be used to choose the appropriate compressor and turbocharger. Having a horsepower target in mind is a vital part of the process. In addition to being necessary for calculating mass flow and boost pressure, a horsepower target is required for choosing the right fuel injectors, fuel pump and regulator, and other engine components.

◇ Estimating Required Air Mass Flow and Boost Pressures to reach a Horsepower target.

- Things you need to know:
  - Horsepower Target
  - Engine displacement
  - Maximum RPM
  - Ambient conditions (temperature and barometric pressure. Barometric pressure is usually given as inches of mercury and can be converted to psi by dividing by 2)

- Things you need to estimate:
  - Engine Volumetric Efficiency. Typical numbers for peak Volumetric Efficiency (VE) range in the 95%-99% for modern 4-valve heads, to 88% - 95% for 2-valve designs. If you have a torque curve for your engine, you can use this to estimate VE at various engine speeds. On a well-tuned engine, the VE will peak at the torque peak, and this number can be used to scale the VE at other engine speeds. A 4-valve engine will typically have higher VE over more of its rev range than a two-valve engine.
  - Intake Manifold Temperature. Compressors with higher efficiency give lower manifold temperatures. Manifold temperatures of intercooled setups are typically 100 - 130 degrees F, while non-intercooled values can reach from 175-300 degrees F.
  - Brake Specific Fuel Consumption (BSFC). BSFC describes the fuel flow rate required to generate each horsepower. General values of BSFC for turbocharged gasoline engines range from 0.50 to 0.60 and higher. The units of BSFC are \( \frac{lb}{hp \cdot h} \). Lower BSFC means that the engine requires less fuel to generate a given horsepower. Race fuels and aggressive tuning are required to reach the low end of the BSFC range described above.

For the equations below, we will divide BSFC by 60 to convert from hours to minutes.

To plot the compressor operating point, first calculate airflow:

\[
Wa = \frac{HP \cdot A/F \cdot BSFC}{60}
\]

Where:
- \( Wa \) = Airflow actual (lb/min)
- \( HP \) = Horsepower Target (flywheel)
- \( A/F \) = Air/Fuel Ratio
- \( BSFC/60 = \frac{lb}{hp \cdot min} \) = Brake Specific Fuel Consumption (\( \frac{lb}{hp \cdot h} \)) ÷ 60 (to convert from hours to minutes)

EXAMPLE:

I have an engine that I would like to use to make 400Hp, I want to choose an air/fuel ratio of 12 and use a BSFC of 0.55. Plugging these numbers into the formula from above:

\[
Wa = 400 \cdot 12 \cdot 0.55 / 60 = 44.0 lb/min
\]

Thus, a compressor map that has the capability of at least 44 pounds per minute of airflow capacity is a good starting point.

Note that nowhere in this calculation did we enter any engine displacement or RPM numbers. This means that for
any engine, in order to make 400 Hp, it needs to flow about 44 lb/min (this assumes that BSFC remains constant across all engine types).

Naturally, a smaller displacement engine will require more boost or higher engine speed to meet this target than a larger engine will. So how much boost pressure would be required?

Calculate required manifold pressure required to meet the horsepower, or flow target:

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

Where:
- \( MAP_{req} \) = Manifold Absolute Pressure (psia) required to meet the horsepower target
- \( Wa \) = Airflow\text{actual}(lb/min)
- \( R \) = Gas Constant = 639.6
- \( T_m \) = Intake Manifold Temperature (degrees F)
- \( VE \) = Volumetric Efficiency
- \( N \) = Engine speed (RPM)
- \( Vd \) = engine displacement (Cubic Inches, convert from liters to CI by multiplying by 61.02, ex. 2.0 liters * 61.02 = 122 CI)

EXAMPLE:

To continue the example above, let’s consider a 2.0 liter engine with the following description:

- \( Wa = 44 \) lb/min as previously calculated
- \( T_m = 130 \) degrees F
- \( VE = 92\% \) at peak power
- \( N = 7200 \) RPM
- \( Vd = 2.0 \) liters * 61.02 = 122 CI

\[ MAP_{req} = \frac{44 \times 639.6 \times (460 + 130)}{0.92 \times 7200/2 \times 122} \]

\[ = 41.1 \text{ psia} \] (remember, this is absolute pressure. Subtract atmospheric pressure to get gauge pressure (aka boost):

\[ 41.1 \text{ psia} - 14.7 \text{ psia (at sea level)} = 26.4 \text{ psig boost} \]

As a comparison let’s repeat the calculation for a larger displacement 5.0L (4942 cc/302 CI) engine.

Where:
- \( Wa = 44 \) lb/min as previously calculated
- \( T_m = 130 \) degrees F
- \( VE = 85\% \) at peak power (it is a pushrod V-8)
- \( N = 6000 \) RPM
- \( Vd = 4.942\times61.02 = 302 \) CI

\[ MAP_{req} = \frac{44 \times 639.6 \times (460 + 130)}{0.85 \times 6000/2 \times 302} \]

\[ = 21.6 \text{ psia} \] (or 6.9 psig boost)
This example illustrates in order to reach the horsepower target of 400 hp, a larger engine requires lower manifold pressure but still needs 44lb/min of airflow. This can have a very significant effect on choosing the correct compressor.

With Mass Flow and Manifold Pressure, we are nearly ready to plot the data on the compressor map. The next step is to determine how much pressure loss exists between the compressor and the manifold. The best way to do this is to measure the pressure drop with a data acquisition system, but many times that is not practical.

Depending upon flow rate, charge air cooler characteristics, piping size, number/quality of the bends, throttle body restriction, etc., the plumbing pressure drop can be estimated. This can be 1 psi or less for a very well designed system. On certain restrictive OEM setups, especially those that have now higher-than-stock airflow levels, the pressure drop can be 4 psi or greater.

For our examples we will assume that there is a 2 psi loss. So to determine the Compressor Discharge Pressure ($P_{2c}$), 2 psi will be added to the manifold pressure calculated above.

$$P_{2c} = MAP + \Delta P_{loss}$$

Where:
- $P_{2c}$ = Compressor Discharge Pressure (psia)
- MAP = Manifold Absolute Pressure (psia)
- $\Delta P_{loss}$ = Pressure Loss Between the Compressor and the Manifold (psi)

For the 2.0 L engine:

$$P_{2c} = 43.1 + 2$$

$$= 43.1 \text{ psia}$$

For the 5.0 L engine:

$$P_{2c} = 21.6 + 2$$

$$= 23.6 \text{ psia}$$

Remember our discussion on inlet depression in the Pressure Ratio discussion earlier, we said that a typical value might be 1 psi, so that is what will be used in this calculation. For this example, assume that we are at sea level, so ambient pressure is 14.7 psia.

We will need to subtract the 1 psi pressure loss from the ambient pressure to determine the Compressor Inlet Pressure ($P_{1c}$).

$$P_{1c} = P_{amb} - \Delta P_{loss}$$

Where:
- $P_{1c}$ = Compressor Inlet Pressure (psia)
- $P_{amb}$ = Ambient Air pressure (psia)
- $\Delta P_{loss}$ = Pressure Loss due to Air Filter/Piping (psi)

$$P_{1c} = 14.7 - 1$$

$$= 13.7 \text{ psia}$$

With this, we can calculate Pressure Ratio ($\Pi c$) using the equation.

$$\Pi c = \frac{P_{2c}}{P_{1c}}$$
For the 2.0 L engine:

\[ \Pi_c = \frac{43.1}{13.7} = 3.14 \]

For the 5.0 L engine:

\[ \Pi_c = \frac{23.6}{13.7} = 1.72 \]

We now have enough information to plot these operating points on the compressor map. First we will try a GT2860RS. This turbo has a 60mm, 60 trim compressor wheel.

Clearly this compressor is too small, as both points are positioned far to the right and beyond the compressor’s choke line.

Another potential candidate might be the GT3076R. This turbo has a 76mm, 56 trim compressor wheel:
This is much better; at least both points are on the map! Let’s look at each point in more detail.

For the 2.0L engine this point is in a very efficient area of the map, but since it is in the center of the map, there would be a concern that at a lower engine speeds that it would be near or over the surge line. This might be ok for a high-rpm-biased powerband that might be used on a racing application, but a street application would be better served by a different compressor.

For the 5.0L engine, this looks like a very good street-biased powerband, with the lower engine speeds passing through the highest efficiency zone on the map, and plenty of margin to stay clear of surge. One area of concern would be turbo overspeed when revving the engine past peak power. A larger compressor would place the operating point nearer to the center of the map and would give some additional benefit to a high-rpm-biased powerband. We’ll look at a larger compressor for the 5.0L after we figure out a good street match for the 2.0L engine.

So now lets look at a GT3071R, which uses a 71mm, 56 trim compressor wheel.
For the 2.0L engine, this is a much more mid-range-oriented compressor. The operating point is shifted a bit towards the choke side of the map and this provides additional surge margin. The lower engine speeds will now pass through the higher efficiency zones and give excellent performance and response.

For the 5.0L engine, the compressor is clearly too small and would not be considered.

Now that we have arrived at an acceptable compressor for the 2.0L engine, let's calculate a lower rpm point to put on the map to better get a feel for what the engine operating line will look like. We can calculate this using the following formula:

\[
W_a = \frac{MAP \times VE \times \frac{N}{2} \times V_d}{R \times (460 + T_m)}
\]

We'll choose the engine speed at which we would expect to see peak torque, based on experience or an educated guess. In this case we'll choose 5000rpm.

Where:
- \( W_a \) = Airflow_{actual} (lb/min)
- \( MAP \) = Manifold Absolute Pressure (psia) = 35.1 psia
- \( R \) = Gas Constant = 639.6
- \( T_m \) = Intake Manifold Temperature (degrees F) = 130
- \( VE \) = Volumetric Efficiency = 0.98
- \( N \) = Engine speed (RPM) = 5000rpm
- \( V_d \) = engine displacement (Cubic Inches, convert from liters to CI by multiplying by 61, ex. 2.0 liters * 61 = 122 CI)
\[
\dot{W}_c = \frac{43.1 \times 0.98 \times 5000 / 2 \times 122}{639.6 \times (460 + 130)}
\]

= 34.1 lb/min

Plotting this on the GT3071R compressor map gives the following operating points.

This gives a good representation of the operating line at that boost level, which is well suited to this map. At engine speeds lower than 5000rpm the boost pressure will be lower, and the pressure ratio would be lower, to keep the compressor out of surge.

Back to the 5.0 L engine. Let’s look at a larger compressor’s map. This time we will try a GT3582R with an 82mm, 56 trim compressor.
Here, compared to the GT3076R, we can see that this point is not quite so deep into choke and will give better high-rpm performance than the 76mm wheel. A further increase in wheel size would give even better high-rpm performance, but at the cost of low- and mid-range response and drivability.

Hopefully this has given a basic idea of what a compressor map displays and how to choose a compressor. As you can see, a few simple estimations and calculations can provide a good basis for compressor selection. If real data is available to be substituted in place of estimation, more accurate results can be generated.