The Turbomustangs.com complete Turbocharging guide

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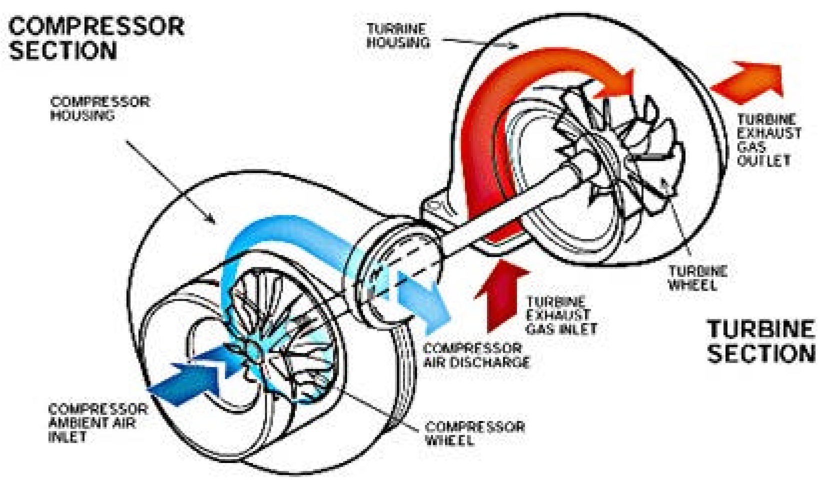
Sources: Garrett, turbodriven.com, and Grapeaperacing.com

Many Thanks to Spencer Brown for his contribution to our site!!!

Holset on fire.



turbo breakdown

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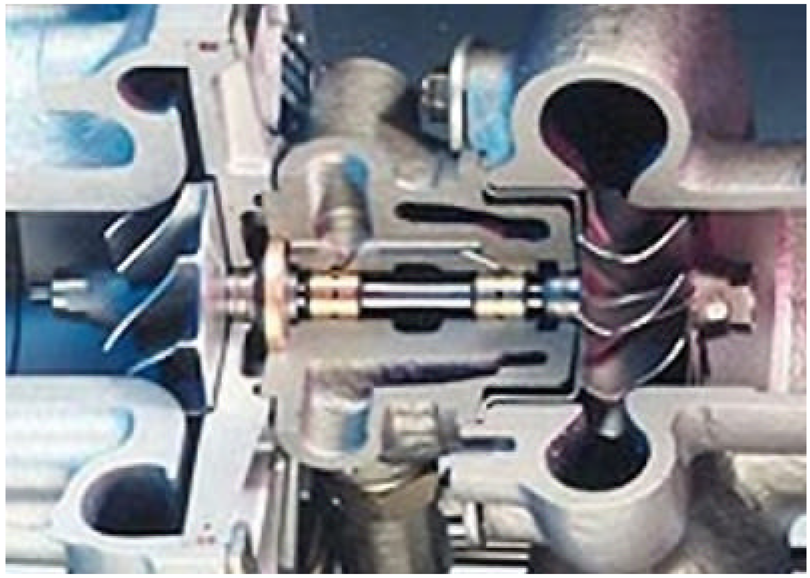
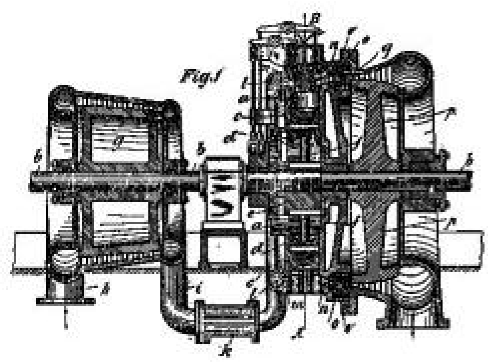


Cutaway of a TO4

(courtesy of Borgwarner turbosystems and Garrett)

Turbocharger History

The first exhaust-driven supercharger was developed by Dr. Alfred J. Buchi of Switzerland between 1909 and 1912, long before Garrett products entered the turbocharger picture. Dr. Buchi was Chief Engineer of Sulzer Brothers Research Department and in 1915 proposed the first prototype of a turbocharged diesel engine, but his ideas gained little or no acceptance at that time.



General Electric began developing turbochargers during the late 1910's. In 1920, a LePere bi-plane that was equipped with a Liberty engine and a General Electric turbocharger set a new altitude record of 33,113 feet (10092m).

Turbochargers were used sparingly on aircraft in World War I, but their development occurred on a widening scale in the 1930's and 1940's - first in Europe and then in the United States. In the United States, General Electric developed turbochargers for military aircraft, and in World War II, thousands were used on fighter aircraft and bombers, such as the B-17. The Garrett Corporation, formed in 1936 by J. C. "Cliff" Garrett, supplied the charge air cooler (aftercooler) for the B-17, located between the General Electric turbocharger and the Pratt and Whitney engine.

In the late 1940's and early 1950's, Garrett was heavily committed to the design of small gas turbine engines from 20 - 90 horse power (15 - 67 kw). The engineers had developed a good background in the metallurgy of housings, high speed seals, radial inflow turbines, and centrifugal compressors.

On September 27, 1954, Cliff Garrett made the decision to separate the turbocharger group from the Gas Turbine department due to commercial diesel turbocharger opportunities. That was the beginning of the new AiResearch Industrial Division - for turbocharger design and manufacturing. AiResearch Industrial Division would later be named Garrett Automotive.

The Chevrolet Corvair Monza and the Oldsmobile Jetfire were the first turbo-powered passenger cars, and made their debut on the US market in 1962/63. Despite maximum technical outlay, however, their poor reliability caused them to disappear quickly from the market.

The internal combustion engine is an air consuming machine. This is because the fuel that is burned requires air with which it can mix to complete the combustion cycle. Once the air/fuel ratio reaches a certain point, the addition of more fuel will not produce more power, but only black smoke or unburned fuel into the atmosphere. The more dense the smoke, the more the engine is being over fueled. Therefore, increasing the fuel delivery beyond the air/fuel ratio limit results in excessive fuel consumption, pollution, high exhaust temperature (diesel) or low exhaust temperature (gasoline), and shortened engine life.

After the first oil crisis in 1973, turbocharging became more acceptable in commercial diesel applications. Until then, the high investment costs of turbocharging were offset only by fuel cost savings, which were minimal. Increasingly stringent emission regulations in the late 80's resulted in an increase in the number of turbocharged truck engines, so that today, virtually every truck engine is turbocharged.

In the 70's, with the turbocharger's entry into motor sports, especially into Formula I racing, the turbocharged passenger car engine became very popular. The word "turbo" became quite fashionable. At that time, almost every automobile manufacturer offered at least one top model equipped with a turbocharged petrol engine. However, this phenomenon disappeared after a few years because although the turbocharged petrol engine was more powerful, it was not economical. Furthermore, the "turbo-lag", the delayed response of the turbochargers, was at that time still relatively large and not accepted by most customers.

The real breakthrough in passenger car turbocharging was achieved in 1978 with the introduction of the first turbocharged diesel engine passenger car in the Mercedes-Benz 300 SD, followed by the VW Golf Turbodiesel in 1981. By means of the turbocharger, the diesel engine passenger car's efficiency could be increased, with almost petrol engine "driveability", and the emissions significantly reduced.

Today, the turbocharging of petrol engines is no longer primarily seen from the performance perspective, but is rather viewed as a means of reducing fuel consumption and, consequently, environmental pollution on account of lower carbon dioxide (CO2) emissions. Currently, the primary reason for turbocharging is the use of the exhaust gas energy to reduce fuel consumption and emissions.

Turbocharger Theory

A turbo charger is basically an exhaust gas driven air compressor and can be best understood if it is divided into its two basic parts, the exhaust gas driven turbine and its housing, and the air compressor and its housing. I did say divided didn't I. Well I should have said like a set of Siamese twins because each of them perform different functions but, because they are joined together at the hip via a common shaft, the function of one impacts the function of the other. How? Take a perfectly set up compressor section and mate it with an incorrect turbine section, or visa versa, and you end up with with our Siamese twins trying to go in different directions. The result is that our Siamese twins end up wasting all of their energy fighting each other and go nowhere.

When considering a turbo charger most folks tend to look at the maximum CFM rating of the compressor and ignore everything else under the assumption that the compressor and the exhaust turbine are perfectly matched out of the box. I will grant you that in stock factory applications that is probably close to the truth but, in all out performance applications, nothing could be further from the truth because of the extremes of operation in a performance application.

The goal in a performance application is to get the exhaust turbine up to speed as quickly as possible however, it must be mated to a compressor wheel that will generate as much pressure as it can as soon as possible. This is a contradiction because the exhaust turbine generates the drive power and the compressor consumes that power. The larger the compressor and the higher the pressure (boost) we want, the quicker the power from the exhaust turbine is used up. Put in a larger exhaust turbine and it will take the engine longer to develop enough hot expanding exhaust gas to spin it, slowing down the compressor and causing turbo lag. At this point I am going to repeat something stated earlier, do not think of a turbo charger as a bolt on piece of equipment, think of it as a system.

The turbine is powered by hot expanding exhaust gas, a lot of hot expanding exhaust gas, the more and the hotter the expanding exhaust gas the better. I am sure many of you have seen pictures of turbo charged engines with cherry red hot exhaust systems and turbo housings. The captions under most of these types of pictures proclaim outstanding horse power numbers. What most of the articles related to these pictures do not tell you is that the engine was under an extreme load. A load so heavy that the engine was almost at its stall point for a prolonged period of time. A condition that most turbo charged engines will never see.

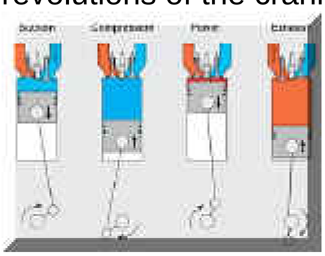
The real point I am trying to make is that the exhaust turbine will not generate enough power to turn the air compressor fast enough for it to work properly unless the engine is feeding the exhaust turbine a lot of hot expanding exhaust gas, a condition that can only be created when the engine is under a load. There is where the selection of transmission gear ratios and the ring and pinion ratio play a critical part. The fact that the engine must be under a load is the reason why, no matter how high you rev a turbo charged engine with no load on it, you will not see the boost gauge move.

This is also where the term 'turbo lag' came from. Turbo lag is basically the amount of time it takes from the time you place a load on the engine (stomp the gas peddle to the floor and dump the clutch or, get full converter lock up with your automatic trans) until the time the engine develops enough hot expanding exhaust gas to spin the turbine fast enough for the compressor to do its job.

Effectively, a turbo charged engine is a normally aspirated engine until the turbine and compressor spin up. To minimize turbo lag, it is imperative that the turbine and the compressor are properly matched to the engine as well as the engine being properly matched to the transmission gears, the ring and pinion gears, and the tires.

Principles of Turbocharging

To better understand the technique of turbocharging, it is useful to be familiar with the internal combustion engine's principles of operation. Today, most passenger car and commercial diesel engines are four-stroke piston engines controlled by intake and exhaust valves. One operating cycle consists of four strokes during two complete revolutions of the crankshaft.



Schematic of a four stroke piston engine

Suction (charge exchange stroke)
When the piston moves down, air (diesel engine or direct injection petrol engine) or a fuel/air mixture (petrol engine) is drawn through the intake valve.

Compression (power stroke)
The cylinder volume is compressed.

Expansion (power stroke)
In the petrol engine, the fuel/air mixture is ignited by a spark plug, whereas in the diesel engine fuel is injected under high pressure and the mixture ignites spontaneously.

Exhaust (charge exchange stroke)
The exhaust gas is expelled when the piston moves up.

These simple operating principles provide various possibilities of increasing the engine's power output:

Swept volume enlargement

Enlargement of the swept volume allows for an increase in power output, as more air is available in a larger combustion chamber and thus more fuel can be burnt. This enlargement can be achieved by increasing either the number of cylinders or the volume of each individual cylinder. In general, this results in larger and heavier engines. As far as fuel consumption and emissions are concerned, no significant advantages can be expected.

Increase in engine rpm

Another possibility for increasing the engine's power output is to increase its speed. This is done by increasing the number of firing strokes per time unit. Because of mechanical stability limits, however, this kind of output improvement is limited. Furthermore, the increasing speed makes the frictional and pumping losses increase exponentially and the engine efficiency drops.

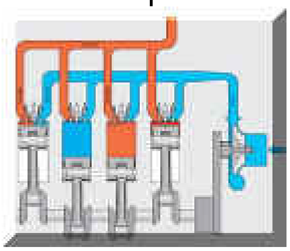
Turbocharging

In the above-described procedures, the engine operates as a naturally aspirated engine. The combustion air is drawn directly into the cylinder during the intake stroke. In turbocharged engines, the combustion air is already pre-compressed before being supplied to the engine. The engine aspirates the same volume of air, but due to the higher pressure, more air mass is supplied into the combustion chamber. Consequently, more fuel can be burnt, so that the engine's power output increases related to the same speed and swept volume.

Basically, one must distinguish between mechanically supercharged and exhaust gas turbocharged engines.

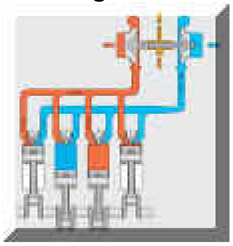
Mechanical supercharging

With mechanical supercharging, the combustion air is compressed by a compressor driven directly by the engine. However, the power output increase is partly lost due to the parasitic losses from driving the compressor. The power to drive a mechanical turbocharger is up to 15 % of the engine output. Therefore, fuel consumption is higher when compared with a naturally aspirated engine with the same power output.


Schematic of a mechanically supercharged four-cylinder engine

Exhaust gas turbocharging

In exhaust gas turbocharging, some of the exhaust gas energy, which would normally be wasted, is used to drive a turbine. Mounted on the same shaft as the turbine is a compressor which draws in the combustion air, compresses it, and then supplies it to the engine. There is no mechanical coupling to the engine.


Schematic of an exhaust gas turbocharged four-cylinder

Common Terms

Adiabatic Efficiency

A 100% adiabatic efficiency means that there is no gain or loss of heat during compression. Most turbochargers will have a 65-75% adiabatic efficiency. Some narrow range turbo's can get higher, these types of turbo's work well in engines that operate over a narrow rpm range. In general the wide range turbo's don't have as good peak efficiency, but have better average efficiency and work better on engine that operate over a wide rpm range.

Pressure Ratio

This is the inlet pressure compared to the outlet pressure of the turbocharger's compressor. For single stage turbo's, the inlet pressure will usually be atmospheric (14.7 psi) and the outlet will be atmospheric + boost pressure. For staged turbo's the inlet pressure will be the outlet pressure of the turbo before it + atmospheric, and the outlet will be inlet pressure + additional boost from that turbo.

Density Ratio

Turbochargers compress the air to make it more dense, this is what allows more oxygen in the engine and give the potential to make more power. The density of the inlet air compared to the density of the outlet air is the density ratio.

Turbine

The side of the turbocharger that converts the energy of the exhaust into mechanical energy to turn the compressor.

Compressor

The side of the turbocharger that compresses the incoming air charge and directs it to your engine.

Cartridge

This is the center section of your turbocharger. It houses the bearings for your turbocharger, they have oil passages to lubricate the bearings and some have water jackets for water cooling.

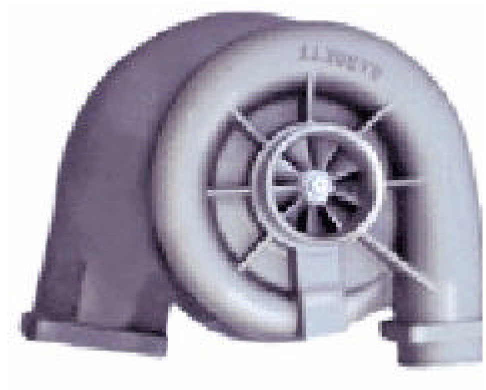
Intercooler

When intake air is compressed by a turbocharger it is also heated, even more so than when supercharging due to the turbo being heated by the exhaust. Hot intake air is not good for power and will increase the chance of detonation. An intercooler reduces the intake temperature by pushing the air through a heat exchanger (much like a small radiator) that absorbs some of the heat out of the charge. With less heat, you'll need less boost pressure to get the desired power and decrease the chance of detonation. Anything that reduces the intake temperature is a big plus in a supercharged engine.

Turbo Lag

A turbocharger uses a centrifugal compressor, which needs rpm to make boost, and it is driven off the exhaust pressure, so it cannot make instant boost. It is especially hard to make boost at low rpm. The turbo takes time to accelerate before full boost comes in, it is this delay that is known as turbo lag. To limit lag, it is important to make the rotating parts of the turbocharger as light as possible. Larger turbo's for high boost applications will also have more lag that smaller turbo's, due to the increase in centrifugal mass. Impeller design, and the whole engine combo also have a large effect on the amount of lag. Turbo lag is often confused with the term boost threshold, but they are not the same thing, lag is nothing more the the delay from when the throttle is opened to the time noticeable boost is achieved.

Turbo Boost



Usually measured in pounds per square inch, it is the pressure the turbocharger makes in the intake manifold. One of the ways to increase airflow through a passage is to increase the pressure differential across the passage. By boosting the intake manifold pressure, airflow into the engine will increase, making more power potential. Boost is also measured in Bar. One Bar equals 14.7 psi.

Boost Threshold

Unlike turbo lag, which is the delay of boost, boost threshold is the lowest possible rpm at which there can be noticeable boost. A low boost threshold is important when accelerating from very low rpm, but at higher rpm, lag is the delay that you feel when you go from light to hard throttle settings.

Wastegate

The wastegate is a valve that allows the exhaust gasses to bypass the turbine. The waste gate relies on boost pressure to open it. Spliced into the wastegate pressure feed there must be some form of pressure bleed. By bleeding pressure to the wastegate, it is possible to control the amount of boost by reducing the pressure at the wastegate.

Turbo Cool Down

A turbocharger is cooled by engine oil, and in many cases, engine coolant as well. Turbo's get very hot when making boost, when you shut the engine down the oil and coolant stop flowing. If you shut the engine down when the turbo is hot, the oil can burn and build up in the unit (known as "coking") and eventually cause it to leak oil (this is the most common turbocharger problem). It is a good idea to let the engine idle for at least 2 minutes after any time you ran under boost. This will cool the turbo down and help prevent coking.

Advantages of Exhaust Gas Turbocharging

Compared with a naturally aspirated engine of identical power output, the fuel consumption of a turbo engine is lower, as some of the normally wasted exhaust energy contributes to the engine's efficiency. Due to the lower volumetric displacement of the turbo engine, frictional and thermal losses are less.

The power-to-weight ratio, i.e. kilowatt (power output)/kilograms (engine weight), of the exhaust gas turbocharged engine is much better than that of the naturally aspirated engine.

The turbo engine's installation space requirement is smaller than that of a naturally aspirated engine with the same power output.

A turbocharged engine's torque characteristic can be improved. Due to the so-called "maxi dyne characteristic" (a very high torque increase at low engine speeds), close to full power output is maintained well below rated engine speed. Therefore, climbing a hill requires fewer gear changes and speed loss is lower.

The high-altitude performance of a turbocharged engine is significantly better. Because of the lower air pressure at high altitudes, the power loss of a naturally aspirated engine is considerable. In contrast, the performance of the turbine improves at altitude as a result of the greater pressure difference between the virtually constant pressure upstream of the turbine and the lower ambient pressure at outlet. The lower air density at the compressor inlet is largely equalized. Hence, the engine has barely any power loss.

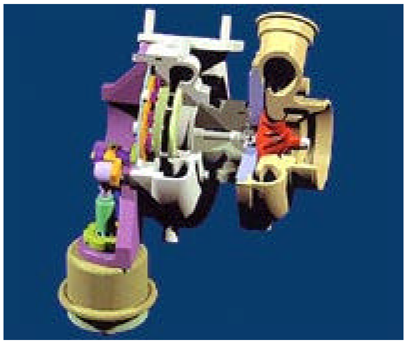
Because of reduced overall size, the sound-radiating outer surface of a turbo engine is smaller, it is therefore less noisy than a naturally aspirated engine with identical output. The turbocharger itself acts as an additional silencer.

Development, Matching and Testing

Development

As turbochargers have to meet different requirements with regard to map height, map width, efficiency characteristics, moment of inertia of the rotor and conditions of use, new compressor and turbine types are continually being developed for various engine applications. Furthermore, different regional legal emission regulations lead to different technical solutions.

The compressor and turbine wheels have the greatest influence on the turbocharger's operational characteristics. These wheels are designed by means of computer programs which allow a three-dimensional calculation of the air and exhaust gas flows. The wheel strength is simultaneously optimized by means of the finite-element method (FEM), and durability calculated on the basis of realistic driving cycles.



CAD-assembled model of a turbocharger

Despite today's advanced computer technology and detailed calculation programs, it is testing which finally decides on the quality of the new aerodynamic components. The fine adjustment and checking of results is therefore carried out on turbocharger test stands.

Matching

The vital components of a turbocharger are the turbine and the compressor. Both are turbo-machines which, with the help of modeling laws, can be manufactured in various sizes with similar characteristics. Thus, by enlarging and reducing, the turbocharger range is established, allowing the optimal turbocharger frame size to be made available for various engine sizes. However, the transferability to other frame sizes is restricted, as not all characteristics can be scaled dimensionally. Furthermore, requirements vary in accordance with each engine size, so that it is not always possible to use the same wheel or housing geometries.

The model similarity and modular design principle, however, permit the development of turbochargers which are individually tailored to every engine. This starts with the selection of the appropriate compressor on the basis of the required boost pressure characteristic curve. Ideally, the full-load curve should be such that the compressor efficiency is at its maximum in the main operating range of the engine. The distance to the surge line should be sufficiently large.

The thermodynamic matching of the turbocharger is implemented by means of mass flow and energy balances. The air delivered by the compressor and the fuel fed to the engine constitute the turbine mass flow rate. In steady-state operation, the turbine and compressor power outputs are identical (free wheel condition). The matching calculation is iterative, based on compressor and turbine maps, as well as the most important engine data.

The matching calculation can be very precise when using computer programs for the calculated engine and turbocharger simulation. Such programs include mass, energy and material balances for all cylinders and the connected pipe work. The turbocharger enters into the calculation in the form of maps. Furthermore, such programs include a number of empirical equations to describe interrelationships which are difficult to express in an analytical way.

Testing

The turbocharger has to operate as reliably and for as long as the engine. Before a turbocharger is released for series production, it has to undergo a number of tests. This test program includes tests of individual turbocharger components, tests on the turbocharger test stand and a test on the engine. Some tests from this complex testing program are described below in detail.

Containment test

If a compressor or turbine wheel bursts, the remaining parts of the wheel must not penetrate the compressor or turbine housing. To achieve this, the shaft and turbine wheel assembly is accelerated to such a high speed that the respective wheel bursts. After bursting, the housing's containment safety is assessed. The burst speed is typically 50 % above the maximum permissible speed.

Low-Cycle Fatigue Test (LCF test)

The LCF test is a load test of the compressor or turbine wheel resulting in the component's destruction. It is used to determine the wheel material load limits. The compressor or turbine wheel is installed on an overspeed test stand. The wheel is accelerated by means of an electric motor until the specified tip speed is reached and then slowed down. On the basis of the results and the component's S/N curve, the expected lifetime can be calculated for every load cycle.

Rotor dynamic measurement

The rotational movement of the rotor is affected by the pulsating gas forces on the turbine. Through its own residual imbalance and through the mechanical vibrations of the engine, it is stimulated to vibrate. Large amplitudes may therefore occur within the bearing clearance and lead to instabilities, especially when the lubricating oil pressures are too low and the oil temperatures too high. At worst, this will result in metallic contact and abnormal mechanical wear.

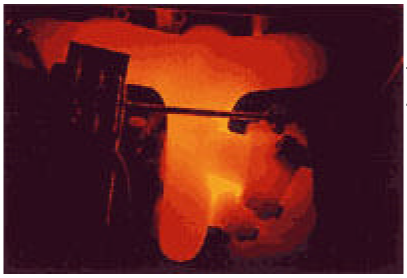
The motion of the rotor is measured and recorded by contactless transducers located in the suction area of the compressor by means of the eddy current method. In all conditions and at all operating points, the rotor amplitudes should not exceed 80 % of maximum possible values. The motion of the rotor must not show any instability.

Start-stop test

The temperature drop in the turbocharger between the gases at the hot turbine side and at the cold compressor inlet can amount to as much as 1000 °C in a distance of only a few centimeters. During the engine's operation, the lubricating oil passing through the bearing cools the center housing so that no critical component temperatures occur. After the engine has been shut down, especially from high loads, heat can accumulate in the center housing, resulting in coking of the lubricating oil. It is therefore of vital importance to determine the maximum component temperatures at the critical points, to avoid the formation of lacquer and carbonized oil in the turbine-side bearing area and on the piston ring.

After the engine has been shut down at the full-load operating point, the turbocharger's heat build-up is measured. After a specified number of cycles, the turbocharger components are inspected. Only when the maximum permissible component temperatures are not exceeded and the carbonized oil quantities around the bearing are found to be low, is this test considered passed.

Cyclic endurance test



During engine operation, the waste gate is exposed to high thermal and mechanical loads. During the waste gate test, these loads are simulated on the test stand.

The checking of all components and the determination of the rates of wear are included in the cycle test. In this test, the turbocharger is run on the engine for several hundred hours at varying load points. The rates of wear are determined by detailed measurements of the individual components, before and after the test.

Recommendations for Servicing and Care

What is good for a turbocharger?

The turbocharger is designed such that it will usually last as long as the engine. It does not require any special maintenance; and inspection is limited to a few periodic checks.

To ensure that the turbocharger's lifetime corresponds to that of the engine, the following engine manufacturer's service instructions must be strictly observed:

- Oil change intervals
- Oil filter system maintenance
- Oil pressure control
- Air filter system maintenance

What is bad for a turbocharger?

90 % of all turbocharger failures are due to the following causes:

- Penetration of foreign bodies into the turbine or the compressor
- Dirt in the oil
- Inadequate oil supply (oil pressure/filter system)
- High exhaust gas temperatures (ignition system/injection system)
- These failures can be avoided by regular maintenance. When maintaining the air filter system, for example, care should be taken that no tramp material gets into the turbocharger.

Failure diagnosis

If the engine does not operate properly, one should not assume that the turbocharger is the cause of failure. It often happens that fully functioning turbochargers are replaced even though the failure does not lie here, but with the engine.

Only after all these points have been checked should one check the turbocharger for faults. Since the turbocharger components are manufactured on high-precision machines to close tolerances and the wheels rotate up to 300,000 rpm, turbochargers should be inspected by qualified specialists only.

Turbocharger Turbine

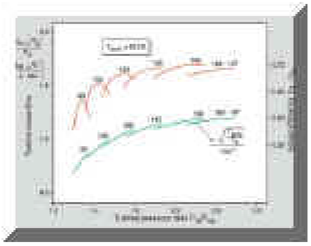
The turbocharger turbine, which consists of a turbine wheel and a turbine housing, converts the engine exhaust gas into mechanical energy to drive the compressor. The gas, which is restricted by the turbine's flow cross-sectional area, results in a pressure and temperature drop between the inlet and outlet. This pressure drop is converted by the turbine into kinetic energy to drive the turbine wheel.

There are two main turbine types: axial and radial flow. In the axial-flow type, flow through the wheel is only in the axial direction. In radial-flow turbines, gas inflow is centripetal, i.e. in a radial direction from the outside in, and gas outflow in an axial direction.

Up to a wheel diameter of about 160 mm, only radial-flow turbines are used. This corresponds to an engine power of approximately 1000 kW per turbocharger. From 300 mm onwards, only axial-flow turbines are used. Between these two values, both variants are possible.

As the radial-flow turbine is the most popular type for automotive applications, the following description is limited to the design and function of this turbine type.
In the volute of such radial or centripetal turbines, exhaust gas pressure is converted into kinetic energy and the exhaust gas at the wheel circumference is directed at constant velocity to the turbine wheel. Energy transfer from kinetic energy into shaft power takes place in the turbine wheel, which is designed so that nearly all the kinetic energy is converted by the time the gas reaches the wheel outlet.

Operating characteristics
The turbine performance increases as the pressure drop between the inlet and outlet increases, i.e. when more exhaust gas is dammed upstream of the turbine as a result of a higher engine speed, or in the case of an exhaust gas temperature rise due to higher exhaust gas energy.



Turbocharger turbine map

The turbine's characteristic behaviors is determined by the specific flow cross-section, the throat cross-section, in the transition area of the inlet channel to the volute. By reducing this throat cross-section, more exhaust gas is dammed upstream of the turbine and the turbine performance increases as a result of the higher pressure ratio. A smaller flow cross-section therefore results in higher boost pressures.
The turbine's flow cross-sectional area can be easily varied by changing the turbine housing.

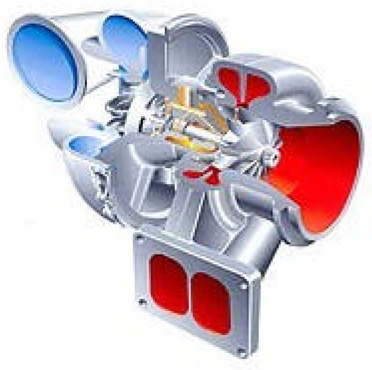
Besides the turbine housing flow cross-sectional area, the exit area at the wheel inlet also influences the turbine's mass flow capacity. The machining of a turbine wheel cast contour allows the cross-sectional area and, therefore, the boost pressure, to be adjusted. A contour enlargement results in a larger flow cross-sectional area of the turbine.

Turbines with variable turbine geometry change the flow cross-section between volute channel and wheel inlet. The exit area to the turbine wheel is changed by variable guide vanes or a variable sliding ring covering a part of the cross-section.

In practice, the operating characteristics of exhaust gas turbocharger turbines are described by maps showing the flow parameters plotted against the turbine pressure ratio. The turbine map shows the mass flow curves and the turbine efficiency for various speeds. To simplify the map, the mass flow curves, as well as the efficiency, can be shown by a mean curve

For a high overall turbocharger efficiency, the co-ordination of compressor and turbine wheel diameters is of vital importance. The position of the operating point on the compressor map determines the turbocharger speed. The turbine wheel diameter has to be such that the turbine efficiency is maximized in this operating range.

Twin-entry turbines

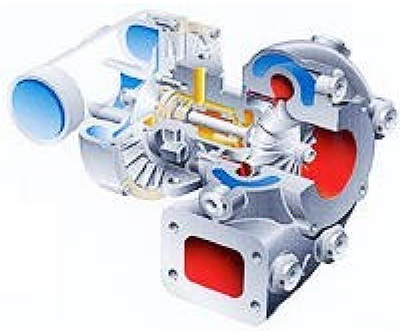


Turbocharger with twin-entry turbine

The turbine is rarely subjected to constant exhaust pressure. In pulse turbocharged commercial diesel engines, twin-entry turbines allow exhaust gas pulsations to be optimized, because a higher turbine pressure ratio is reached in a shorter time. Thus, through the increasing pressure ratio, the efficiency rises, improving the all-important time interval when a high, more efficient mass flow is passing through the turbine. As a result of this improved exhaust gas energy utilization, the engine's boost pressure characteristics and, hence, torque behavior is improved, particularly at low engine speeds.

To prevent the various cylinders from interfering with each other during the charge exchange cycles, three cylinders are connected into one exhaust gas manifold. Twin-entry turbines then allow the exhaust gas flow to be fed separately through the turbine.

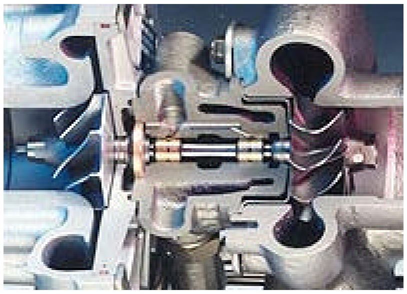
Water-cooled turbine housings



Turbocharger with water-cooled turbine housing for marine applications

Safety aspects also have to be taken into account in turbocharger design. In ship engine rooms, for instance, hot surfaces have to be avoided because of fire risks. Therefore, water-cooled turbocharger turbine housings or housings coated with insulating material are used for marine applications.

Turbocharger Bearing System



Turbocharger bearing system (cut-away model)

The turbocharger shaft and turbine wheel assembly rotates at speeds up to 300,000 rpm. Turbocharger life should correspond to that of the engine, which could be 1,000,000 km for a commercial vehicle. Only sleeve bearings specially designed for turbochargers can meet these high requirements at a reasonable cost.

Radial bearing system
With a sleeve bearing, the shaft turns without friction on an oil film in the sleeve bearing bushing. For the turbocharger, the oil supply comes from the engine oil circuit. The bearing system is designed such that brass floating bushings, rotating at about half shaft speed, are situated between the stationary center housing and the rotating shaft. This allows these high speed bearings to be adapted such that there is no metal contact between shaft and bearings at any of the operating points. Besides the lubricating function, the oil film in the bearing clearances also has a damping function, which contributes to the stability of the shaft and turbine wheel assembly. The hydrodynamic load-carrying capacity and the bearing damping characteristics are optimized by the clearances. The lubricating oil thickness for the inner clearances is therefore selected with respect to the bearing strength, whereas the outer clearances are designed with regard to the bearing damping. The bearing clearances are only a few hundredths of a millimeter.

The one-piece bearing system is a special form of a sleeve bearing system. The shaft turns within a stationary bushing, which is oil scavenged from the outside. The outer bearing clearance can be designed specifically for the bearing damping, as no rotation takes place.

Axial-thrust bearing system
Neither the fully floating bushing bearings nor the single-piece fixed floating bushing bearing system support forces in axial direction. As the gas forces acting on the compressor and turbine wheels in axial direction are of differing strengths, the shaft and turbine wheel assembly is displaced in an axial direction. The axial bearing, a sliding surface bearing with tapered lands, absorbs these forces. Two small discs fixed on the

shaft serve as contact surfaces. The axial bearing is fixed in the center housing. An oil-deflecting plate prevents the oil from entering the shaft sealing area.

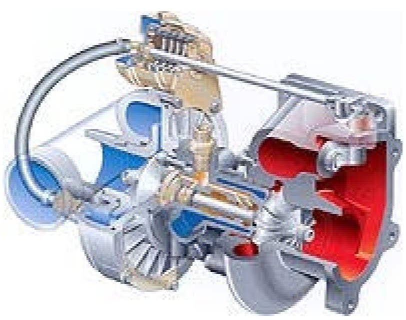
Oil drain

The lubricating oil flows into the turbocharger at a pressure of approximately 4 bar. As the oil drains off at low pressure, the oil drain pipe diameter must be much larger than the oil inlet pipe. The oil flow through the bearing should, whenever possible, be vertical from top to bottom. The oil drain pipe should be returned into the crankcase above the engine oil level. Any obstruction in the oil drain pipe will result in back pressure in the bearing system. The oil then passes through the sealing rings into the compressor and the turbine.

Sealing

The center housing must be sealed against the hot turbine exhaust gas and against oil loss from the center housing. A piston ring is installed in a groove on the rotor shaft on both the turbine and compressor side. These rings do not rotate, but are firmly clamped in the center housing. This contactless type of sealing, a form of labyrinth seal, makes oil leakage more difficult due to multiple flow reversals, and ensures that only small quantities of exhaust gas escape into the crankcase.

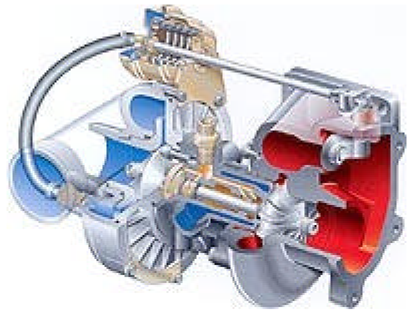
Water-cooling



Turbocharger for passenger car gasoline applications with water-cooled bearing housing

Petrol engines, where the exhaust gas temperatures are 200 to 300 °C higher than in diesel engines, are generally equipped with water-cooled center housings. During operation of the engine, the center housing is integrated into the cooling circuit of the engine. After the engine's shutdown, the residual heat is carried away by means of a small cooling circuit, which is driven by a thermostatically controlled electric water pump.

Turbocharger Compressor

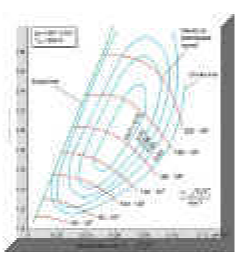


Turbocharger compressors are generally centrifugal compressors consisting of three essential components: compressor wheel, diffuser, and housing. With the rotational speed of the wheel, air is drawn in axially, accelerated to high velocity and then expelled in a radial direction.

The diffuser slows down the high-velocity air, largely without losses, so that both pressure and temperature rise. The diffuser is formed by the compressor backplate and a part of the volute housing, which in its turn collects the air and slows it down further before it reaches the compressor exit.

Operating characteristics
The compressor operating behavior is generally defined by maps showing the relationship between pressure ratio and volume or mass flow rate. The useable section of the map relating to centrifugal compressors is limited by the surge and choke lines and the maximum permissible compressor speed.

Surge line
The map width is limited on the left by the surge line. This is basically "stalling" of the air flow at the compressor inlet. With too small a volume flow and too high a pressure ratio, the flow can no longer adhere to the suction side of the blades, with the result that the discharge process is interrupted. The air flow through the compressor is reversed until a stable pressure ratio with positive volume flow rate is reached, the pressure builds up again and the cycle repeats. This flow instability continues at a fixed frequency and the resultant noise is known as "surging".



Compressor map of a turbocharger for passenger car applications

Choke line:

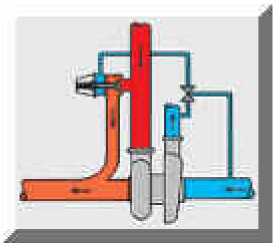
The maximum centrifugal compressor volume flow rate is normally limited by the cross-section at the compressor inlet. When the flow at the wheel inlet reaches sonic velocity, no further flow rate increase is possible. The choke line can be recognized by the steeply descending speed lines at the right on the compressor map.

Turbocharger Control System

The drivability of passenger car turbo engines must meet the same high requirements as naturally aspirated engines of the same power output. That means, full boost pressure must be available at low engine speeds. This can only be achieved with a boost pressure control system on the turbine side.

Control by turbine-side bypass
The turbine-side bypass is the simplest form of boost pressure control. The turbine size is chosen such that torque characteristic requirements at low engine speeds can be met and good vehicle drivability achieved. With this design, more exhaust gas than required to produce the necessary boost pressure is supplied to the turbine shortly before the maximum torque is reached. Therefore, once a specific boost pressure is achieved, part of the exhaust gas flow is fed around the turbine via a bypass. The wastegate which opens or closes the bypass is usually operated by a spring-loaded diaphragm in response to the boost pressure.

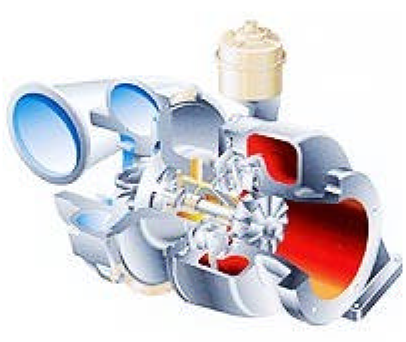
Today, electronic boost pressure control systems are increasingly used in modern passenger car diesel and petrol engines. When compared with purely pneumatic control, which can only function as a full-load pressure limiter, a flexible boost pressure control allows an optimal part-load boost pressure setting. This operates in accordance with various parameters such as charge air temperature, degree of timing advance and fuel quality. The operation of the flap corresponds to that of the previously described actuator. The actuator diaphragm is subjected to a modulated control pressure instead of full boost pressure.



Boost pressure control of a turbocharged petrol engine by proportional control pressure

This control pressure is lower than the boost pressure and generated by a proportional valve. This ensures that the diaphragm is subjected to the boost pressure and the pressure at the compressor inlet in varying proportions. The proportional valve is controlled by the engine electronics. For diesel engines, a vacuum-regulated actuator is used for electronic boost pressure control.

Variable turbine geometry
The variable turbine geometry allows the turbine flow cross-section to be varied in accordance with the engine operating point. This allows the entire exhaust gas energy to be utilized and the turbine flow cross-section to be set optimally for each operating point. As a result, the efficiency of the turbocharger and hence that of the engine is higher than that achieved with the bypass control.



Turbocharger for truck applications with variable turbine geometry (VTG)

Flow cross-section control through variable guide vanes: VTG
Variable guide vanes between the volute housing and the turbine wheel have an effect on the pressure build-up behavior and, therefore, on the turbine power output. At low engine speeds, the flow cross-section is reduced by closing the guide vanes. The boost pressure and hence the engine torque rise as a result of the higher pressure drop between turbine inlet and outlet. At high engine speeds, the guide vanes gradually open. The required boost pressure is achieved at a low turbine pressure ratio and the engine's fuel consumption reduced. During vehicle acceleration from low speeds the guide vanes close to gain maximum energy of the exhaust gas. With increasing speed, the vanes open and adapt to the corresponding operating point.

Today, the exhaust gas temperature of modern high-output diesel engines amounts to up to 830 °C. The precise and reliable guide vane movement in the hot exhaust gas flow puts high demands on materials and requires tolerances within the turbine to be exactly defined. Irrespective of the turbocharger frame size, the guide vanes need a minimum clearance to ensure reliable operation over the whole vehicle lifetime.

Intercooler Theory

An intercooler is a heat exchanger. That means there are two or more fluids that don't physically touch each other but a transfer heat or energy takes place between them.

At wide open throttle and full boost the hot compressed air coming from a turbocharger is probably between 250 and 350 deg F depending on the particular turbo, boost pressure, outside air temperature, etc.. We want to cool it down, which reduces its volume so we can pack more air molecules into the cylinders and reduce the engine's likelihood of detonation.

How does an intercooler work? Hot air from the turbo flows through tubes inside the intercooler. The turbo air transfers heat to the tubes, warming the tubes and cooling the turbo air. Outside air (or water) passes over the tubes and between fins that are attached to the tubes. Heat is transferred from the hot tubes and fins to the cool outside air. This heats the outside air while cooling the tubes. This is how the turbo air is cooled down. Heat goes from the turbo air to the tubes to the outside air.

There are some useful equations which will help us understand the factors involved in transferring heat. These equations are good for any heat transfer problem, such as radiators and a/c condensers, not just intercoolers. After we look at these equations and see what's important and what's not, we can talk about what all this means.

Equation 1

The first equation describes the overall heat transfer that occurs.
Q = U x A x DTlm

Q is the amount of energy that is transferred.
U is called the heat transfer coefficient. It is a measure of how well the exchanger transfers heat. The bigger the number, the better the transfer.
A is the heat transfer area, or the surface area of the intercooler tubes and fins that is exposed to the outside air.
DTlm is called the log mean temperature difference. It is an indication of the "driving force", or the overall average difference in temperature between the hot and cold fluids. The equation for this is:

$$DTlm = \frac{(DT1-DT2) * F}{\ln(DT1/DT2)}$$

where **DT1** = turbo air temperature in - outside air temperature out
DT2 = turbo air temperature out - outside air temperature in
F = a correction factor, see below

Note:

The outside air that passes through the fins on the passenger side of the intercooler comes out hotter than the air passing through the fins on the drivers side of the intercooler. If you captured the air passing through all the fins and mixed it up, the temperature of this mix is the "outside air temperature out".

F is a correction factor that accounts for the fact that the cooling air coming out of the back of the intercooler is cooler on one side than the other.

To calculate this correction factor, calculate "P" and "R":

P = turbo air temp out - turbo air temp in
outside air temp in - turbo air temp in

R = outside air temp in - outside air temp out
turbo air temp out - turbo air temp in

This overall heat transfer equation shows us how to get better intercooler performance. To get colder air out of the intercooler we need to transfer more heat, or make Q bigger in other words. To make Q bigger we have to make U, A, or DTlm bigger, so that when you multiply them all together you get a bigger number. More on that later.

Equation 2

We also have an equation for checking the amount of heat lost or gained by the fluid on one side of the heat exchanger (i.e., just the turbo air or just the outside air):
Q = m x Cp x DT

Q is the energy transferred. It will have the exact same value as the Q in the first equation. If 5000 BTU are transferred from turbo air to outside air, then Q = 5000 for this equation AND the first equation.
m is the mass flow rate (lbs/minute) of fluid, in this case either turbo air or outside air depending on which side you're looking at.
Cp is the heat capacity of the air. This is a measure of the amount of energy that the fluid will absorb for every degree of temperature that it goes up. It is about 0.25 for air and 1.0 for water. Air doesn't do a great job of absorbing heat. If you put 10 BTU into a pound of air the temperature of it goes up about 40 degrees. If you put 10 BTU into a pound of water, the temperature only goes up about 10 degrees! Water is a great energy absorber. That's why we use water for radiators instead of some other fluid.
DT is the difference in temperature between the inlet and outlet. If the air is 200 deg going in and 125 deg coming out, then DT = 200 - 125 = 75. Again, on the cooling air side the outlet temperature is the average "mix" temperature.

If you know 3 of the 4 main variables on one side of the exchanger (the amount of heat transferred, the inlet and outlet temperatures, and the fluid's flow rate) then this equation is used to figure out the 4th. For example, if you know the amount of heat transferred, the inlet temperature, and the flow rate you can calculate the outlet temperature. Since you can't measure everything, this equation is used to figure out what you don't know.

Caveat

These equations are all for steady state heat transfer, which we probably don't really see too much under the conditions that we are most interested in - drag race! Cruising on the highway you would definitely see steady state. Perhaps at the big end of the track you may see it too, I don't know. The material of the intercooler itself will rise in temperature when you hit full throttle, absorbing more heat than what these equations would lead you to believe. For example, at steady state idle the intercooler body may be at 100 deg F. At steady state full throttle it may be 175 deg F. The energy it takes to heat it up to that temperature comes from the turbo outlet air, and so the cooling of that air is what is removed by both the flowing outside air and the absorption of the intercooler body. How long does it take to get to the new steady state? Beats me, but the graphs I've seen of intercooler outlet temperatures over the course of a quarter mile run lead me to believe that it is approached before you get to the end of the quarter mile, since the intercooler outlet temperatures reached a steady level.

So, now that we've got these equations, what do they really tell us?

The difference between the intercooler outlet temperature and the outside air temperature is called the approach. If it is 100 degrees outside and your intercooler cools the air going into the intake manifold down to 140 degrees, then you have an approach of 40 degrees (140 - 100 = 40). To get a better (smaller) approach you have to have more area or a better U, but there is a problem with diminishing returns. Lets rearrange the first equation to Q/DTlm = U x A. Every time DTlm goes down (get a better temperature approach) then Q goes up (transfer more heat, get a colder outlet temperature), and dividing Q by DTlm gets bigger a lot faster than U x A does. The upshot of that is we have a situation of diminishing returns; for every degree of a better approach you need more and more U x A to get there. Start with a 30 deg approach and go to 20 and you have to improve U x A by some amount, to go from 20 to 10 you need to increase U x A by an even bigger amount.

I would consider an approach of 20 degrees to be pretty good. In industrial heat exchangers it starts to get uneconomical to do better somewhere around there, the exchanger starts to get too big to justify the added expense. The only practical way of making the DTlm bigger on an existing intercooler is to only drive on cold days; if you buy a better intercooler you naturally get a better DTlm.

You can transfer more heat (and have cooler outlet temps) with more heat transfer area. That means buying a new intercooler with more tubes, more fins, longer tubes, or all three. This is what most aftermarket intercoolers strive for. Big front mounts, intercooler and a half, etc... are all increasing the area.

A practical consideration is the fin count. The area of the fins is included in the heat transfer area; more fins means more area. If you try to pack too many fins into the intercooler the heat transfer area does go up, which is good, but the cooling air flow over the fins goes down, which is bad. Looking at the 2nd equation, Q=m*Cp*DT, when the fin count is too high then the airflow ("m") drops. For a given Q that you are trying to reach then you have to have a bigger DT, which means you have to heat up the air more. Then that affects the DTlm in the first equation, making it smaller, and lowering the overall heat transfer. So there is an optimum to be found. Starting off with bare tubes you add fins and the heat transfer goes up because you're increasing the area, and keep adding fins until it starts to choke off the cooling air flow and heat transfer starts going back down. At that point you have to add more tubes or make them longer to get more heat transfer out of the increased area.

Make U go up. You can increase the U by adding or improving "tabulators" inside the tubes. These are the fins inside the tubes which cause the air to swirl inside the tube and makes it transfer its heat to the tube more efficiently. One of the best ways to increase the U is to clean the tubes out. Oil film inside the tubes acts as an insulator or thermal barrier. It keeps heat from moving from the air to the tube wall. This is expressed in our equation as a lower U. Lower U means Lower Q's which mean hotter turbo air temperatures coming out of the intercooler.

Air-to-water. If we use water as the cooling medium instead of the outside air, we can see a big improvement for several reasons: Water can absorb more energy with a lower temperature rise. This improves our DTlm, makes it bigger, which makes Q go up and outlet temps go down. A well designed water cooler exchanger also has a much bigger U, which also helps Q go up. And since both DTlm and U went up, you can make the area A smaller which makes it easier to fit the intercooler in the engine compartment. Of course there are some practical drawbacks. The need for a water circulation system is one. a big one is cooling the water down after it is heated, which means another radiator. This leads to another problem; You heat the water and cool it down with the outside air. You can't get it as cool as the outside air, but maybe you can get it within 20 degrees of it. Now you are cooling the turbo air with water that is 20 degrees hotter than the outside air, and you can only get within 15 degrees of that temperature so coming out of the intercooler you have turbo air that is 35 degrees hotter than the outside. You could have easily done that with an air to air intercooler. But if you put ice water in your holding tank and circulate that, then maybe the air temp coming out of the intercooler is 15 degrees above that or 45 to50 degrees. But after the water warms up you're back to the hot air again. Great for racing but not as good for the street.

Lower the inlet temperature. The less the turbo has to work to compress the air the lower the temperature the air coming out of the turbo is. This actually hurts DTlm, but the cooler going in the cooler coming out. You can work the turbo less by running lower boost, by improving the pressure drop between the air filter and the turbo, or by having a more efficient compressor wheel. You can also reduce the pressure drop in the intercooler, which allows you to run the same boost in the intake manifold while while having a lower turbo discharge pressure. If you can drop the turbo outlet pressure by 2 psi, or raise the turbo inlet pressure by 1 psi, that will drop the turbo discharge temperature by about 16 degrees. If the turbo air is going into the intercooler 16 degrees colder then it may come out only 10 degrees colder than before, but that is still better than it was.

Pressure Drop

Another aspect of intercoolers to be considered is pressure drop. The pressure read by a boost gauge is the pressure in the intake manifold. It is not the same as the pressure that the turbocharger itself puts out. To get a fluid, such as air, to flow there must be a difference in pressure from one end to the other. Consider a straw that is sitting on the table. It doesn't having anything moving through it until you pick it up, stick it in your mouth, and change the pressure at one end (either by blowing or sucking). In the same way the turbo outlet pressure is higher than the intake manifold pressure, and will always be higher than the intake pressure, because there must be a pressure difference for the air to move.

The difference in pressure required for a given amount of air to move from turbo to intake manifold is an indication of the hydraulic restriction of the intercooler, the up pipe, and the throttle body. Let's say you are trying to move 255 gram/sec of air through a stock intercooler, up pipe, and throttle body and there is a 4 psi difference that is pushing it along (I'm just making up numbers here). If your boost gauge reads 15 psi, that means the turbo is actually putting up 19 psi. Now you buy a PT-70 and slap on some Champion heads. Now you are moving 450 gm/sec of air. At 15 psi boost in the intake manifold the turbo now has to put up 23 psi, because the pressure drop required to get the higher air flow is now 8 psi instead of the 4 that we had before. More flow with the same equipment means higher pressure drop. So we put on a new front mount intercooler. It has a lower pressure drop, pressure drop is now 4 psi, so the turbo is putting up 19 psi again. Now we add the 65 mm throttle body and the pressure drop is now 3 psi. Then we add the 2.5 inch up pipe, and it drops to 2.5 psi. Now to make 15 psi boost the turbo only has to put up 17.5 psi. The difference in turbo outlet temperature between 23 psi and 17.5 psi is about 40 deg (assuming a constant efficiency)! So you can see how just by reducing the pressure drop we can lower the temperatures while still running the same amount of boost.

I have seen some misunderstandings regarding intercooler pressure drop and how it relates to heat transfer. For example, one vendor's catalog implies that if you had little or no pressure drop then you would have no heat transfer. This is incorrect. Pressure drop and heat transfer are relatively independent, you can have good heat transfer in an intercooler that has a small pressure drop if it is designed correctly. It is easier to have good heat transfer when there is a larger pressure drop because the fluid's turbulence helps the heat transfer coefficient (U), but I have seen industrial coolers that are designed to have less than 0.2 psi of drop while flowing a heck of a lot more air, so it is certainly feasible.

Pressure drop is important because the higher the turbo discharge pressure is the higher the temperature of the turbo air. When we drop the turbo discharge pressure we also drop the temperature of the air coming out of the turbo. When we do that we also drop the intercooler outlet temperature, although not as much, but hey, every little bit helps. This lower pressure drop is part of the benefit offered by new, bigger front mount intercoolers, by bigger up pipes; and by bigger throttle bodies. You can also make the turbo work less hard by improving the inlet side to it. K&N air filters, free flowing MAF pipes, removing the MAF itself when switching to an aftermarket fuel injection system, these all reduce the pressure drop in the turbo inlet system which makes the compressor work less to produce the same boost which will reduce the turbo discharge temperature (among other, and probably greater, benefits).

What about my Intercooler?

Wondering if your intercooler is up to snuff? The big test: measure your intercooler outlet temperature! When I did this I got a K type thermocouple, the thin wire kind, slid it under the throttle body/up pipe hose and down into the center of the up pipe, and went for a drive. On an 80 to 85 deg day I got a WOT temperature of 140 deg, for a 55 to 60 deg approach. That tells me that I need more intercooler. If I can get the temperature down to 100 deg, the air density in the intake manifold goes up by 7%, so I should flow 7% more air and presumably make 7% more hp. On a 350 hp engine that is 25 hp increase. On a 450 hp engine that's a 30 hp increase. Damn, where's my check book…

Another check is pressure drop. Best way to check it is to find a pressure differential gauge, which has 2 lines instead of the single line a normal pressure gauge has. It checks the difference between the 2 spots it is hooked up to, as opposed to checking the difference in pressure between the spot it is hooked up to and atmospheric pressure, which is how a normal pressure gauge works.

Hook one line of the gauge to the turbo outlet and one to (preferably) the intercooler outlet. The turbo outlet/intercooler inlet pressure is easy, just tee into the wastegate supply line off the compressor housing. It would be nice to get the intercooler outlet pressure directly, but there's no convenient spot to hook up to. Hooking into the intake manifold (such as via the line to the boost gauge) is quite convenient, but gives the total pressure drop: intercooler + up pipe + throttle body. That'll give you a pretty good idea though.

Instead of the differential pressure gauge you could use 2 boost gauges, one in each spot, but then you have to worry about whether both gauges are calibrated the same, try to read both at the same time while driving fast, etc AND you may spring (i.e., ruin) the gauge on the turbo outlet since when you close the throttle you get a big pressure spike that your normal boost gauge never sees.

If you find more than 4 or 5 psi difference between the intercooler inlet and intake manifold (and I'm just giving an educated guess here, you'd probably want to refer to one of the intercooler manufacturers for a better number) then I would suspect that a larger, lower pressure drop intercooler would offer you some gains.

A/R Explained

TERMS

A/R =The Area / Radius **ratio**
A = Area of throat at intake
R = radius of scroll in the housing of either the compressor or exhaust turbine

TRIM -turbine **wheel trim** can effect an increase or decrease in turbine pressure for a given housing A/R

Compressor Maps

T3 "40"
T3 "45"

T3 "50

T3 "60"

T3 Super 60

T04B S-3

T04B V1/V2

T04B H-3

60-1

62-1

T04E "40"

T04E "46"

TS04

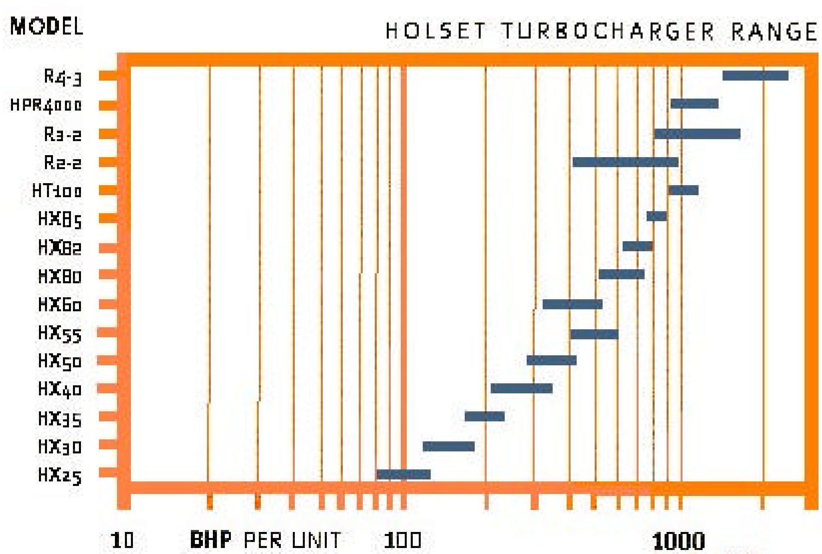
T61

T64

T66

T04E "50"

A little info on your Holsets



Selecting a Turbocharger Compressor

Compressor Selection

When using the formula's below, you will need to use compressor flow maps and work with the formulas until you size the compressor that will work for your application. Compressor flow maps are available from the manufacturer, do a search on the web, or use the maps I have provided. On the flow maps, the airflow requirements should fall somewhere between the surge line and the 60% efficiency line, the goal should be to get in the peak efficiency range at the point of your power peak. In this article I will walk through an example as I explain it.

Engine Airflow Requirements

In order to select a turbocharger, you must know how much air it must flow to reach your goal. You first need to figure the cubic feet per minute of air flowing through the engine at maximum rpm. The the formula to to this for a 4 stroke engine is:

(CID × RPM) ÷3456 = CFM

For a 2 stroke you divide by 1728 rather than 3456. Lets assume that you are turbocharging a 302 cubic inch engine That will redline at 6000 rpm.

(302 × 6000) ÷ 3456 = 524.3 CFM

The engine will flow 524.3 CFM of air assuming a 100% volumetric efficiency. Most street engines will have an 80-90% VE, so the CFM will need to be adjusted. Lets assume our 302 has an 85% VE.

$524.3 \times 0.85 = 445.7$ CFM

Our 302 will actually flow 445.7 CFM with an 85% VE.

Pressure Ratio

The pressure ratio is simply the pressure in, compared to the pressure out of the turbocharger. The pressure in is usually atmospheric pressure, but may be slightly lower if the intake system before the turbo is restrictive, the inlet pressure could be higher than atmospheric if there is more than 1 turbocharger in series. In that case the inlet let pressure will be the outlet pressure of the turbo before it. If we want 10 psi of boost with atmospheric pressure as the inlet pressure, the formula would look like this:

$(10 + 14.7) \div 14.7 = 1.68:1$ pressure ratio

Temperature Rise

A compressor will raise the temperature of air as it compresses it. As temperature increases, the volume of air also increases. There is an ideal temperature rise which is a temperature rise equivalent to the amount of work that it takes to compress the air. The formula to figure the ideal outlet temperature is:

$$T_2 = T_1 (P_2 \div P_1)^{0.283}$$

Where:
T_2 = Outlet Temperature °R
T_1 = Inlet Temperature °R
°R = °F + 460
P_1 = Inlet Pressure Absolute
P_2 = Outlet Pressure Absolute

Lets assume that the inlet temperature is 75° F and we're going to want 10 psi of boost pressure. To figure T_1 in °R, you will do this:

$T_1 = 75 + 460 = 535$°R

The P_1 inlet pressure will be atmospheric in our case and the P_2 outlet pressure will be 10 psi above atmospheric. Atmospheric pressure is 14.7 psi, so the inlet pressure will be 14.7 psi, to figure the outlet pressure add the boost pressure to the inlet pressure.

$P_2 = 14.7 + 10 = 24.7$ psi

For our example, we now have everything we need to figure out the ideal outlet temperature. We must plug this info into out formula to figure out T_2:

$T_1 = 75$

P_1 = 14.7
P_2 = 24.7

The formula will now look like this:

$T_2 = 535\ (24.7 \div 14.7)^{0.283} = 620\ °R$

You then need to subtract 460 to get °F, so simply do this:

620 - 460 = 160 °F Ideal Outlet Temperature

This is a temperature rise of 85 °F

Adiabatic Efficiency

The above formula assumes a 100% adiabatic efficiency (AE), no loss or gain of heat. The actual temperature rise will certainly be higher than that. How much higher will depend on the adiabatic efficiency of the compressor, usually 60-75%. To figure the actual outlet temperature, you need this formula:

Ideal Outlet Temperature Rise ÷ AE = Actual Outlet Temperature Rise

Lets assume the compressor we are looking at has a 70% adiabatic efficiency at the pressure ratio and flow range we're dealing with. The outlet temperature will then be 30% higher than ideal. So at 70% it using our example, we'd need to do this:

85 ÷ 0.7 = 121 °F Actual Outlet Temperature Rise

Now we must add the temperature rise to the inlet temperature:

75 + 121 = 196 °F Actual Outlet Temperature

Density Ratio

As air is heated it expands and becomes less dense. This makes an increase in volume and flow. To compare the inlet to outlet air flow, you must know the density ratio. To figure out this ratio, use this formula:

(Inlet °R ÷ Outlet °R) × (Outlet Pressure ÷ Inlet Pressure) = Density Ratio

We have everything we need to figure this out. For our 302 example the formula will look like this:

(535 ÷ 656) × (24.7 ÷ 14.7) = 1.37 Density Ratio

Compressor Inlet Airflow

Using all the above information, you can figure out what the actual inlet flow in in CFM. Do do this, use this formula:

Outlet CFM × Density Ratio = Actual Inlet CFM

Using the same 302 in our examples, it would look like this:

447.5 CFM × 1.37 = 610.6 CFM Inlet Air Flow

That is about a 37% increase in airflow and the potential for 37% more power. When comparing to a compressor flow map that is in Pounds per Minute (lbs/min), multiply CFM by 0.069 to convert CFM to lbs/min.

610.6 CFM × 0.069 = 42.1 lbs/min

Now you can use these formula's along with flow maps to select a compressor to match your engine. You should play with a few adiabatic efficiency numbers and pressure ratios to get good results. For twin turbo's, remember that each turbo will only flow 1/2 the total airflow.

Using Your Numbers

A turbocharger compressor map has two axis. On the x-axis (the horizontal one) is the airflow, often in lbs/minute. On the y-axis is the pressure ratio, usually as "1+boost pressure", in bar. Inside the map there are plots for turbine rpm, more or less horizontal lines, efficiency (oval rings) and most often also surge limit - a dotted line.

To use the map, you need to know the airflow you will have through the engine.

Using this value, you can use a map. Draw a line from your air flow (lbs/min) on the x-axis, and a line from the pressure ratio (psi + 14.7 ÷14.7). The point of intersection will hopefully be inside one of the higher efficiency rings, about 70%.

You should always have the intersection to the right of the surge limit, otherwise it is no good.

The way to do this is to plot 5-10 intersections (different rpms and boost pressures) in different maps. By having maps for different turbos, and trying different boost pressures and rpm, you can get an idea of how it's going to work.

Remember this only gives an estimate, you might have to resort to trial and error to get exactly spot on. This way, however, you can be reasonably sure you are in the right ballpark.

1 bar = 14.50377 PSI

1 PSI = 0.06894757 bar

Compressor Formula's

CFM of Airflow

CFM = ((CID × RPM) ÷ 3456) × VE

Where:
VE = Volumetric Efficiency
CID = Cubic Inch Displacement

Pressure Ratio

Pressure Ratio = $(P_2 + P_1) \div P_1$

Where:
P_1= Inlet Pressure Absolute
P_2= Outlet Pressure Absolute

Ideal Outlet Temperature Rise °R

$T_2 = T_1(P_2 \div P_1)^{0.283}$

Where:
T_2= Outlet Temperature °R
T_1= Inlet Temperature °R
°R = °F + 460
P_1= Inlet Pressure Absolute
P_2= Outlet Pressure Absolute

Actual Outlet Temperature Rise

OTR = $(T_2 \div AE) - 460$

Where:
OTR = Outlet Temperature Rise °F
T_2= Outlet Temperature °R
AE = Adiabatic Efficiency

Actual Outlet Temperature

Actual Outlet Temp = Inlet Temp + Outlet Temp Rise

Density Ratio

Density Ratio = $(T_1 \div T_2) \times (P_2 \div P_1)$

Where:
T_2= Outlet Temperature °R
T_1= Inlet Temperature °R
°R = °F + 460
P_1= Inlet Pressure Absolute
P_2= Outlet Pressure Absolute

Actual Compressor Inlet Flow

Actual Inlet CFM = Outlet CFM × Density Ratio

CFM to lbs/min

lbs/min = CFM × 0.069

Turbocharger Camshafts

Pressure Differential

Unlike a supercharger that is driven directly form the crankshaft, a turbo is driven by exhaust gas velocity. Turbochargers are an exhaust restriction (which raises the exhaust gas pressure), but since they use energy that would otherwise be wasted, they are much more efficient than a belt driven supercharger. Normally when the exhaust valve opens, there is still useable pressure in the cylinder that needs to be dumped so it will not resist the piston trying to go back up the bore. That pressure makes high exhaust gas velocity. With a turbocharged engine, this is the energy that is used to spin the turbine.

With a well matched turbo / engine combo, boost pressure should be higher than exhaust gas pressure at the low side of the power band (near peak torque). As the engine nears peak hp, the pressure differential will get nearer 1:1. At some point the pressures in the intake and exhaust will be equal then crossover making the exhaust a higher pressure than the intake. At peak hp there will usually be more exhaust gas pressure than boost pressure. The ultimate goal is to have as little exhaust backpressure possible for the desired boost.

If the turbocharger is matched well to the engine combination, the camshaft selection will not need to be much different than that of a supercharged engine. The problem is that most factory turbo engines have turbo's that are sized too small and will usually have more back pressure than boost pressure over much of the useable power band. Car manufactures do this in an attempt to reduce turbo lag. When a turbocharger is too small, it will be a bigger restriction in the exhaust, causing more back pressure. A big mistake of turbo owners is to crank the boost up as high as they can thinking they are going faster, but in reality, chances are that they are just killing the efficiency of the turbo and most gains are lost. If you want to run higher boost levels and back pressure is a problem, cam timing can be altered to give respectable power increases for much cheaper than a new turbocharger. Before you go increasing boost and changing cams, remember that the oxygen content into the engine will increase power, not boost pressure. A good flowing head with a good intercooler can make a lot of power without high boost. You may not need more boost to get the power you want.

Valve Overlap

If your one of many factory turbo car owners with a turbo sized too small, there will be higher exhaust pressure than intake, you should see that if both valves are open at the same time, the flow would reverse. Any valve overlap is a no no if you're looking for higher boost with a restrictive turbine housing. The exhaust valve will usually close very close to TDC, but there is will still be more pressure on the cylinder than in the intake. You must allow the piston to travel down the bore until the pressure is equalized. If the cylinder pressure is lower than the intake manifold pressure, no reverse flow will take place. This means that the intake valve needs to open 20-35° ATDC, depending on the amount of boost you're using. Most street turbo's will work well when the valve opens close to 20° ATDC, only when boost gets near 30 psi will you need to delay it as much as 35° ATDC. In low boost applications (under 15 psi or so), opening the valve closer to TDC and maybe keeping the exhaust valve open a little after TDC is a compromise for better throttle response before the boost comes on. As you increase boost, you will need to delay the opening of the intake valve to avoid reversion. You want the intake valve to open as soon as possible, in an ideal situation, the intake valve should open when the pressure in the cylinder is equal to boost pressure. This can cause a little confusion with cam overlap. If the exhaust valve closes before the intake opens, the overlap will be considered negative. If the exhaust valve closed at TDC and the intake opened at 20° ATDC there would be -20° of overlap. In this type situation, pumping losses are quite large, although the turbo will still use less power than a crank driven supercharger.

If you have a well matched turbo for the engine and application, it is a different deal altogether. A well matched turbine housing on the turbo will usually work well with cams with a lobe separation in the 112-114° area. If there is more pressure in the intake than in the exhaust, a camshaft suited for superchargers or nitrous will usually works well. When the exhaust backpressure is lower than the intake, reversion is not a problem, actually just the opposite is a problem. More pressure in the intake can blow fresh intake charge right out the exhaust valve. This can be a serious problem with a turbo motor since the charge will burn in the exhaust raising temperatures of the exhaust valves and turbo. This is also a problem with superchargers, which is why supercharger cam profiles usually work well with turbo's. In this type situation, the power required to turn the turbine is nearly 100% recovered energy that would have normally been dumped out the tailpipe, basically free power. Many will argue that nothing is free and you need pressure to spin the turbine and this must make pumping losses. They are wrong because a turbo is not getting anything for free at all, it is just making the engine more efficient. It is true that there are pumping losses, but on the other hand there are pumping gains as well. If the exhaust back pressure is lower than the intake, the intake pressure makes more force on the intake stroke to help push the piston down. At the same time another piston is on it's exhaust stroke. So the intake pressure is more than canceling out the exhaust pressure. Not free, just more efficient.

Valve Lift

By delaying the opening of the intake, the duration of the cam will be much shorter. A short duration intake works well with a turbo, but the problem is that sufficient lift is hard to get from such a short duration. This is where high ratio rockers can really pay off. A cam for a turbo engine can delay the intake opening by over 40° compared to an cam for a normally aspirated engine. This makes for much less valve lift when the piston is at peak velocity (somewhere near 75° ATDC), any help to get the valve open faster will make large improvements.

Roller Camshafts

Turbo motors place a large flow demand at low valve lifts, and roller cams cannot accelerate the valve opening as fast as a flat tappet. They do catch up and pass a flat tappet after about 20° or so, but up until that point the favor goes toward the flat tappet cam. The area where rollers really help in turbo motors (and supercharged) is cutting frictional losses. Any forced induction engine will need more spring force on the intakes. If you run a lot of boost, you'll need quite a bit more spring force to control the valves. As spring forces gets higher, the life of the cam gets reduced. A roller tappet can withstand more than twice the spring pressure as a flat tappet with no problems. On the exhaust side, it's not the springs that put the loads on the cam lobes. The problem there is that there is still so much cylinder pressure trying to hold that valve closed. This puts tremendous pressure on the exhaust lobes. So when high boost levels are used, consider a roller cam. I would definitely consider a roller cam on engines making more than 20 lbs. of boost.

Turbocharger Placement

Heat

The turbo(s) mustn't come near anything that will be affected by heat and there must be plenty of room around them for the plumbing as well (which gets hot too).. This applies to the compressor housings as well. The compressor housings may not run hot while the engine is running due to inlet air cooling them, but when you shut down heat will soak through the cartridge and make them almost as hot as the turbine housings, so take this into consideration. When an engine shuts down, there is little to no air movement under hood, so things can get hotter. Even an idling engine at least has the cooling fan causing some airflow under the hood.

Heat Retention

The power used to power a turbocharger is exhaust gas velocity. When temperature drops in an exhaust system, so does it's velocity. This is important when considering placement of a turbocharger because the farther the turbo is from the engine, the greater the temperature drop will be. In this respect, putting the turbo's as close to the engine as practical will give best turbo performance. Sometimes is it not practical to have a turbo very close to the engine, in those cases, making the manifold (or header) from a material with a strong fatigue resistance will allow insulating of the piping.

Plumbing

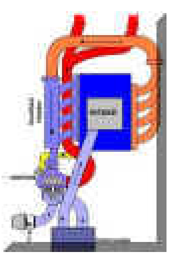
A turbocharger requires quite a bit of plumbing. There is turbine inlet and outlet, compressor inlet and outlet, oil feed and drain, and sometimes coolant in and out. As with any type of plumbing, sharp bends cause a restriction, so it's best to put the turbo where the bending required will be least restrictive. Also, as with any type of plumbing, high pressure sides are less affected by restriction than low pressure sides. In other words is you have a choice of a 90 degree elbow at the compressor inlet or the compressor outlet. It will do less harm on the higher pressure outlet side. It is harder for a pump to pull than it is too push. With the oil system, A 90 degree fitting at the pressure side will harm very little since most engines oiling systems can supply more pressure than the turbo needs to begin with, but a restriction on the drain side can back up oil in the cartridge cause big problems. And for the turbine side, the exhaust gasses entering the turbine will (should) be much higher than in the exhaust system after the turbo. The idea here is not to try redesigning the chassis around a "correct" turbo plumbing system, the idea is to compromise the fewest of these things for the space you have to work with. This may require relocating some under hood components and/or protecting others. For me, I relocated my Duraspark box and some wiring. What I did, which may not be an acceptable option to some people, is I converted to manual brakes and steering. By removing the power booster and power steering pump, I gained a lot of room for a better turbo system.

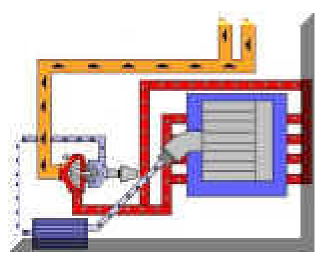
Other

Once you find the best place to mount the turbo, consider the effect it will have when you do routine maintenance on the car. If you have a solid cam that requires periodic valve adjustments, it would be nice to be able to remove the valve cover(s) without removing a turbo. Spark plugs are another thing, make sure you can get to them without too much trouble. When you are mocking up a manifold, plugs and wires should be in place and checked for enough clearance. Many V8's have the dipstick on one side or the other, it would really suck if you couldn't check your oil. You can take a few pictures of the engine bay before you start taking it apart for quick reference as to what might be in the way. It's easy to forget the little things that make a big difference.

Basic Turbo System

These are two examples of the basic turbo system





Turbocharger Troubleshooting

All too frequently, serviceable turbochargers are removed from engines before the cause of the problem has been determined. Always inspect and assess turbocharger condition before removal from the engine.

Should removal of the turbo become necessary, try to determine if the connections were tight and without leaks while you are removing the hoses, clamps or connections. Once disassembly has been completed, it may be difficult or impossible to substantiate the conditions that caused the problem.

Problems experienced in the field can most often be corrected by system troubleshooting. Immediate or early failure of a replacement turbocharger may be related to:

1. The incomplete correction of the problem that caused the need for the replacement.

2. Problems introduced during the replacement.

3. A defective turbocharger.

A turbocharger that has operated successfully is very unlikely to be found defective at a later date. Speed and

temperature normally seen in turbocharger operation usually identify defects very quickly. Installation or engine system problems can also show up immediately upon replacement. Don't be too quick to blame the turbo for operational problems if the turbocharger spins freely and has not rubbed the housing.

It should be emphasized that a turbocharger does not basically change the operating characteristics of an engine. A turbocharger is not a power source within itself. The turbo's only function is to supply a greater volume of compressed air to the engine so that more fuel can be burned to produce more power. It can function only as dictated by the flow, pressure and temperature in the engine exhaust gas.

Turbochargers are an integral component of a complete operating system. Only by convenience is a turbo external or 'bolt-on" in installation. It is no less dedicated than an engine's camshaft or pistons. Understanding how a turbocharger is part of a complete engine management system is essential in successfully diagnosing and repairing problems. Likewise, a better understanding of some of a turbocharger's features can be helpful when determining that a turbo is damaged or defective, and installing it correctly the first time, every time.

Verify that the turbocharger is the correct configuration for the application. Assembly and component part numbers may both need attention. This is particularly important because the matching" process requires subtle component differences. Part number checks are necessary because some of the possible discrepancies will not be apparent to the untrained eye.

A turbocharger cannot correct or over-come such things as malfunctions or deficiencies in the engine fuel system, timing, plugged air cleaners, faulty liners, etc. Therefore, if a turbocharged engine system has malfunctioned and the turbocharger has been examined and determined to be operational, proceed with troubleshooting, as though the engine were non-turbocharged. Simply replacing a good turbocharger with another will not correct basic engine deficiencies.

Common Symptoms

Turbochargers and engines have common problem symptoms.

Engine Lacks Power

Engine Exhaust Smoke

Oil Consumption

Noisy Operation

As you see, any of these symptoms could be the result of an internal engine problem and might not involve the turbocharger at all.

The following on-engine troubleshooting guide is designed to quickly determine turbocharger condition and prevent unnecessary removal.

On Engine Troubleshooting

Many of the problem causing conditions will appear in direct or inverse proportion to the power output. For example, there may be a problem at idle that is unnoticeable at full power or visa versa. The following procedures are an overall evaluation involving varying operational conditions. On-engine troubleshooting will also help to expose any external or engine related causes of turbocharger failure that must be corrected to prevent the failure of a replacement unit.

The most efficient way to troubleshoot a performance complaint is to proceed through all of the steps in the order presented before making a final determination of the service required. It is extremely important that all in-service problem areas are examined before any single one is corrected.

In some instances, corrective service may lead you to turbocharger Damage Analysis. And, depending on the results of these inspections and/or the turbocharger model, you may also have to measure the bearing clearances or test the wastegate device. Those inspections, as well as a detailed analysis of problems that may be exposed here, are also covered in turbocharger Damage Analysis.

On-engine troubleshooting consists of several basic steps that should be taken before the turbocharger is removed from the engine. Any external or engine-related faults found must be corrected before a replacement turbocharger is installed.

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running, make sure it is cool before beginning.

Warning! Operating the turbocharger without the inlet duct and air filter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

The basic steps to troubleshooting are as follows:

VISUAL AND MECHANICAL CHECKS

Inspect the turbocharger exterior and installation. Listen for unusual mechanical noises. Visually check and test for leaks, blockage, high heat, restrictions or conditions that have allowed wheels to contact the housings. Leaks that are seemingly small and insignificant at idle or low power can greatly affect air/fuel ratios and pressures within the housing and full power. At full power, those leaks become problematic.

a. Listen for unusual mechanical noise and watch for vibration.

b. Listen for a high pitched noise. It can indicate air or gas leaks.

c. Listen for noise level cycling. It can indicate a restriction in the air cleaner or ducting.

d. Inspect for missing or loose nuts, bolts, clamps and washers.

e. Inspect for loose or damaged intake and exhaust manifolds and their ducting and clamps.

f. Inspect for damaged or restricted oil supply and drain lines.

g. Inspect for cracked or deteriorating turbocharger housings.

h. Inspect for external oil or coolant leakage; external din deposits (indicates air, oil, exhaust or coolant leakage).

i. Inspect for obvious heat discoloration.

j. Inspect for obviously restricted air filter.

k. Check the wastegate for free movement and damage. Be sure that hoses are in good condition and that the connections are tight. Check the calibration and control system according to the original equipment specifications.

I. Verify that the turbocharger is the correct configuration for the application.

Remember, correcting these problems does not in it self remove any residues that were the indicators of the problem. The remaining residues often cause inaccurate turbocharger evaluation. Incorrect turbocharger evaluation may result after the situation has been corrected and the residues remain. For example, an air filter replaced just previous to your inspection would lead you to conclude that air blockage is not the problem even though the residue indicates blockage.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, then the unit should be replaced at this time and corrective actions taken to prevent reoccurrence.

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running make sure it is cool before beginning.

Warning! Operating the turbocharger without the inlet duct and air filter connected can result In personal injury. Equipment damage may result from foreign objects entering the turbocharger.

TURBINE WHEEL AND TURBINE HOUSING CHECKS

Remove the ducting from the turbine outlet. Using an inspection light:

Inspect the turbine for evidence of foreign object damage. This is usually not easily visible from the turbine outlet unless the damage is severe. Determine the source of the object and check for possible engine damage. Figure 20 highlights where turbine wheel rub frequently occurs.

Turn the rotating assembly by hand and feel for dragging or binding; also check by pushing the assembly sideways while turning. The wheel should turn freely and without any rubbing or scraping noises. If there are obvious signs of wheel nib or that the turbine housing has been operated at excessive temperatures, then the turbocharger is damaged and must be replaced. If you are still not sure whether the wheel is rubbing, inspect the bearing clearances after completing this section.

Look for evidence of oil leakage. If oil deposits are found, then determine whether the oil is from the engine or from the turbocharger center housing. Some oil residues may be cleaned; heavy oil residues may require replacement. If the oil is from the center housing, then remove the oil drain line and look into the turbocharger drain opening and drain line with an inspection light. Check for an oily, sludge build-up on the shaft between the bearing journals, in the drain cavity, and in the drain line.

Check the following to determine the cause of the problem and effect corrections as necessary:

a. Restricted draining or high crankcase pressure can raise the pressure of the center housing drain area above the pressure in the turbine housing forcing oil in that direction.

b. PCV flow control valves on spark ignition engines must operate as one way check valves when boost is developed. This reverses the direction of flow in the ventilation system. A partially closed PCV allows manifold boost to pressurize the crankcase.

c. Damaged oil drain line.

d. Improper line routing (more than 35 degrees from vertical or any sharp bends) or routings close to exhaust manifolds.

e. Submerged drain line from too high an oil level or equipment operated at extreme angle.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, the unit should be replaced at this time and corrective actions taken to prevent reoccurrence.

Refer to the engine manufacturer’s service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running make sure it is cool before beginning.

Warning' Operating the turbocharger without the inlet duct and airfilter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

COMPRESSOR WHEEL AND COMPRESSOR HOUSING CHECKS

Remove the ducting from the compressor inlet. Using an inspection light:

Inspect the compressor for evidence of foreign object damage. If the wheel is damaged, the foreign object probably entered through the intake system. Remember that the origin of foreign object damage should be identified. Foreign objects usually come from human error or deteriorated intake systems. Determine the source of the object, clean the system, and check for possible engine damage.

Turn the rotating assembly by hand and feel for dragging or binding; also check

by pushing the assembly sideways while turning. Look for any evidence of wheel rub. Wheel rub can be caused by loose, distorted, or binding housings as well as damaged bearings. If there is still any doubt, inspect the bearing clearances.

Look for evidence of oil leakage. The compressor-side is most sensitive to a restricted air inlet. Oil in the compressor outlet does not prove turbocharger seal leakage. Oil from crankcase ventilation or other oil sources can be confused with compressor-side oil leaks. Figure 21 shows how the crankcase ventilation system is tied to the compressor side of the turbocharger. The turbocharger compressor can take oil vapor and expel it as liquid oil. General engine condition greatly affects engine crankcase ventilation system operation. Follow manufacturer's recommendations. Other factors that can cause oil leakage into the compressor are detailed in the Troubleshooting charts. Compressor oil leaks can result in oil accumulations in the charge-air cooler. When all problems have been corrected this oil can be transferred into the engine. If oil accumulation occurs, it will require draining and cleaning of the charge-air cooler.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, the unit should be replaced at this time and

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running, make sure it is cool before beginning.

Warning: Operating the turbocharger without the inlet duct and airfilter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

ROTATING ASSEMBLY CHECK

Check for signs of a sludged or coked center housing. A sludged or coked center housing will not likely be evident by inspecting the end housings. Evidence of this condition will be found in advanced cases by looking for oil deposits in the oil inlet. Also check the oil drain area.

Turn the rotating assembly by hand and feel for dragging or binding; also check by pushing the assembly sideways while turning. Look for any evidence of wheel rub. Wheel rub can be caused by loose, distorted, or binding housings as well as damaged bearings. If there is still any doubt, inspect the bearing clearances.

Look for evidence of leakage; either oil and/or coolant.

a. Loose or improper connections.

b. Improper gaskets or gasket material.

c. Casting porosity.

d. Improperly drilled holes.

CHECK RADIAL AND AXIAL BEARING CLEARANCES

If none of the previous steps have revealed any turbocharger faults, or if the evidence is not conclusive, this procedure will show if the unit is worn or damaged internally to the point of needing replacement.

Radial Journal Bearing Clearance Note: Due to the unique location of the internal opening in the center housing casting on models T45 and T51, access to the shaft wheel at this point is difficult.

Check the radial clearance of the journal bearings as follows:

For all models, except T45 and T51, attach the turbocharger gage set to the unit so that the dial indicator plunger extends through the oil drain port and contacts the shaft of the turbine wheel assembly.

For models T45 and T51 only, place the special curved end of the gage arm in contact with the wheel shaft through the oil outlet port and the internal opening in the casting.

a. The dial indicator shaft is then placed in contact with the exposed portion of the gage arm at a point equidistant from the gage arm pivot and a point of contact with the wheel shaft, with the arm kept in contact with the shaft by the spring action of the dial indicator plunger.

b. Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go away from the dial indicator plunger.

c. Set the dial indicator to zero.

d. Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go toward the dial indicator plunger. Note the maximum shaft movement shown on the indicator dial.

e. To make sure that the dial indicator reading is the maximum possible, roll the wheels slightly in one direction and then the other while applying pressure.

f. Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go away from the dial indicator plunger. Make sure the dial indicator pointer returns to zero.

g. Repeat steps -b' through "f" several times to make sure that the maximum bearing radial clearance, as indicated by the maximum shaft movement, has been measured.

h. Compare the maximum clearance measured to the specification for bearing radial clearance for the model turbocharger being tested, as found in the specifications section of your catalog. If the measurement is within the specification, the journal bearings are in good condition. If the measurement is not within the specification, the turbocharger is worn or damaged internally and must be replaced.

Axial (Thrust) Bearing Clearance Check the axial clearance of the thrust bearing as follows:

a. Clean the hub end of the turbine wheel assembly.

b. Attach a turbocharger gage set to the turbine end of the turbocharger so that the dial indicator plunger rests on the hub end of the turbine wheel assembly.

c. Manually apply pressure to the compressor wheel and turbine wheel assembly to move the assembly as far as it will go away from the turbine end of the turbocharger (away from the dial indicator plunger).

d. Set the dial indicator to zero.

e. Manually apply pressure to the compressor wheel and turbine wheel assembly to move the assembly as far as it will go toward the turbine end of the turbocharger (toward the dial indicator plunger). Note the maximum shaft movement shown on the indicator dial.

f. Repeat steps 'c" through 'e" several times to make sure that the maximum bearing axial clearance, as indicated by the maximum turbine wheel assembly movement, has been measured.

g. Compare the minimum and maximum clearance measured to the specification for bearing axial clearance for the model turbocharger being tested, as found in the specifications section of your catalog.

If the measurement is within the specification, the thrust bearing is in good condition. If no other faults have been found in previous steps, the turbocharger is likely not at fault in the complaint. Troubleshoot the engine as instructed in the engine manufacturer's service manual.

If the turbocharger was recently replaced or overhauled, make certain that the proper unit was installed or that the right parts were used in the overhaul. A turbocharger can appear to be right for the installation, but if the turbine and compressor components are not identical to those recommended by the engine manufacturer, performance and service life can suffer.

If the measurement is out of specification, the turbocharger is worn or damaged internally and must be replaced.

WASTEGATE ASSEMBLY CHECK

Wastegates may be an integral part of the turbine housing or a separate device plumbed into the exhaust system. Actuator's are connected directly to the compressor outlet or work in conjunction with the engine management system. Engine manufacturers supply specific information on wastegates because of their arrangement within the engine management system.

Actuators spring pre-load may be high enough to make checking for free movement by hand difficult or impossible. Visually check for obstacles that can prevent movement or closure. Inverted exhaust pipe connection studs can prevent some wastegates from opening. Stress relief cracking may be found around the relief port in turbine housings. Cracks that do not extend beyond the wastegate valve do not present a problem.

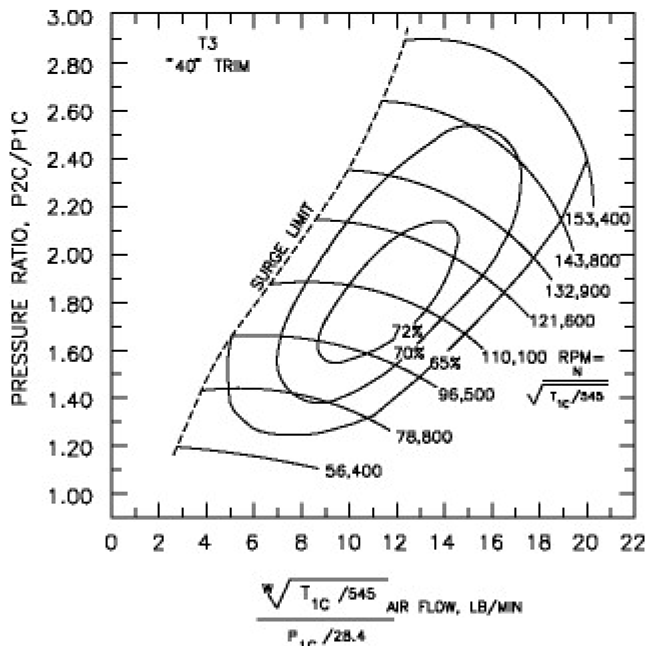
CAUTION! When checking an actuator do not over pressurize because the diaphragm may become dam-aged Swing valve actuators should move smoothly and show no decay when subjected to calibration pressures. Many poppet valve units have a hollow stem that opens in the guide giving a small leak when pressurized Poppet valve units also depend to some degree on engine vibration to overcome static friction When checking this type unit, light tapping of the housing will usually provide an accurate calibration check.

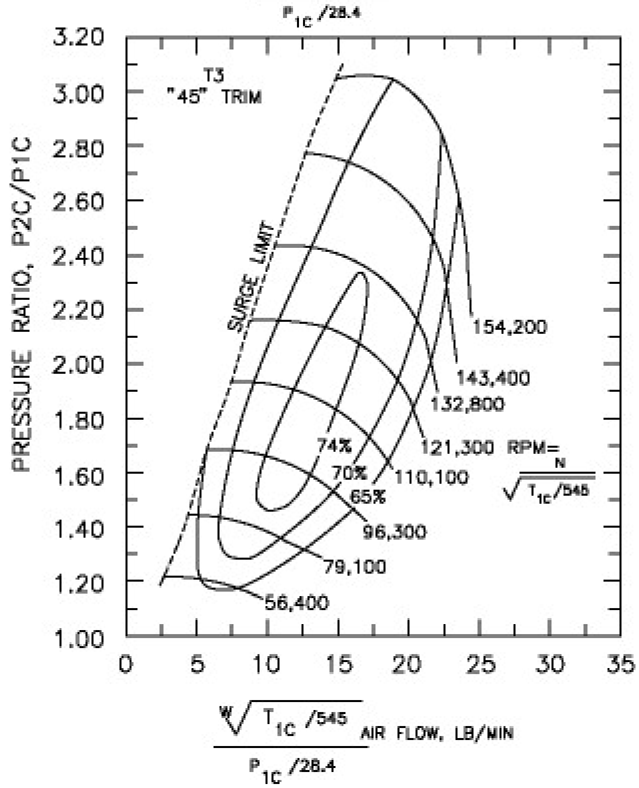
Original equipment specifications for calibration should be closely followed because they are established to interact with the entire engine management system. The calibration pressure is not necessarily a reflection of expected manifold boost because other pressures act on the valve. In many cases pressure to the actuator is overridden by the engine control system to vary the amount of boost depending upon conditions.

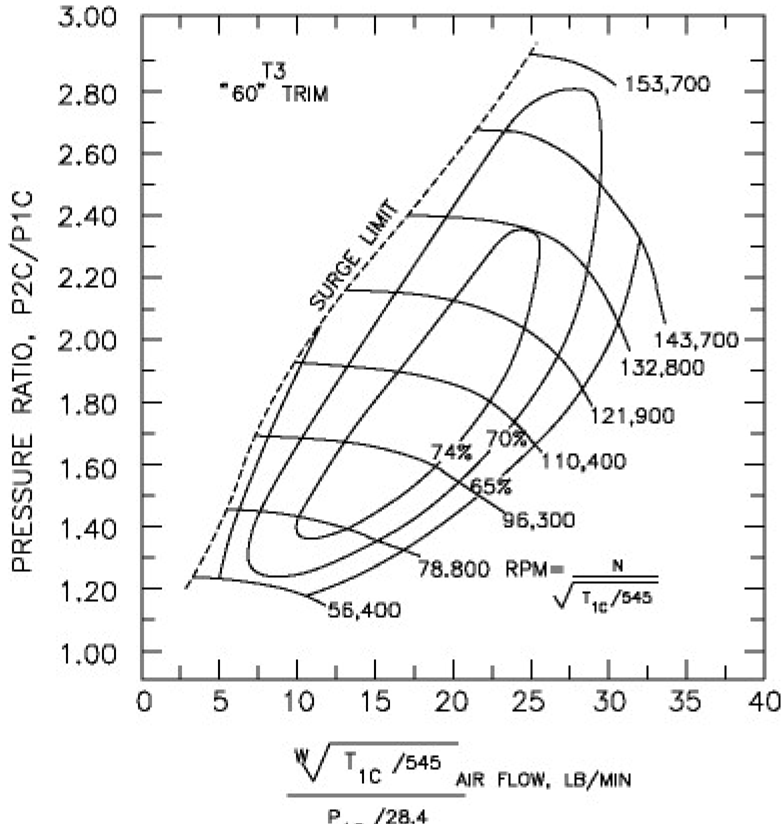
Many actuators are mounted on a bracket away from high temperatures. Problems may come from bending of the brackets or rods. High temperatures from exhaust leaks, corrosion, or other loose or damaged components can also result in wastegate problems.

When all else fails ask somebody that knows.

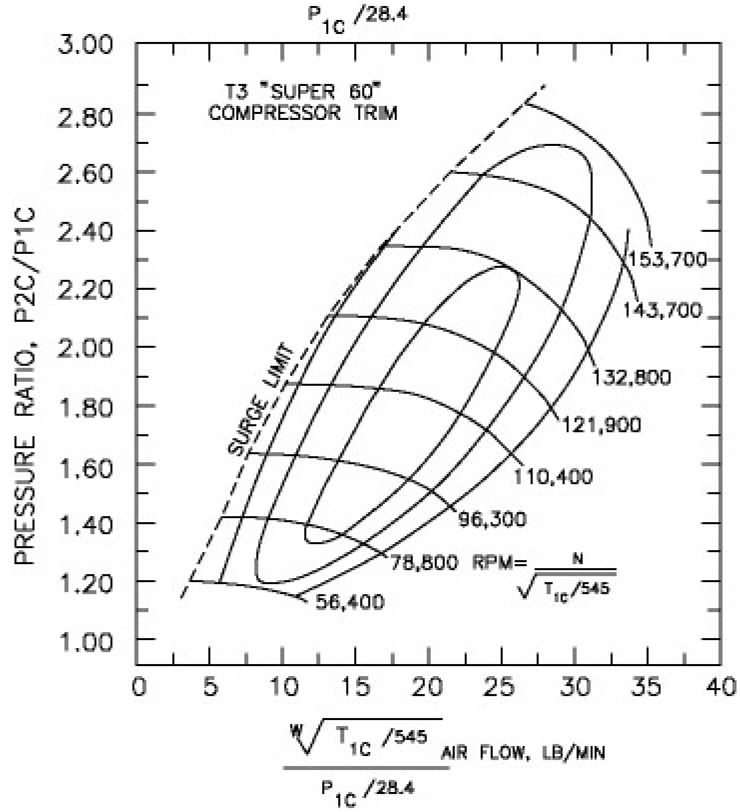
This document was compiled by Spencer Brown. Half of it was written by Spencer Brown and the rest of it contains data from: Garrett, turbodriven.com, and Grapeaperacing.com Please visit those sites for more turbo information.


T3
"40" TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
153,400
143,800
132,900
121,600
110,100 RPM=
96,500
78,800
56,400
72%
70%
65%
AIR FLOW, LB/MIN

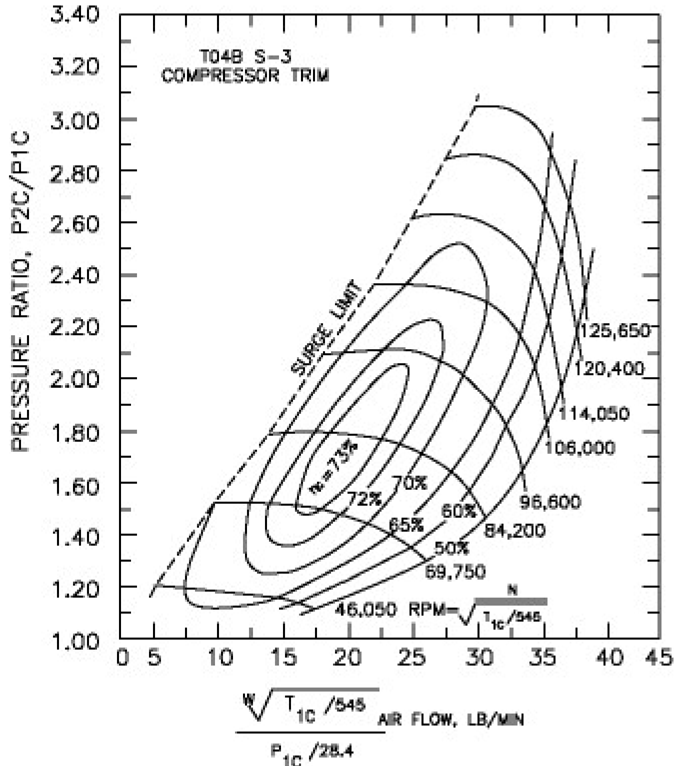

T3
"45" TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
154,200
143,400
132,800
121,300 RPM=
110,100
96,300
79,100
56,400
74%
70%
65%
AIR FLOW, LB/MIN

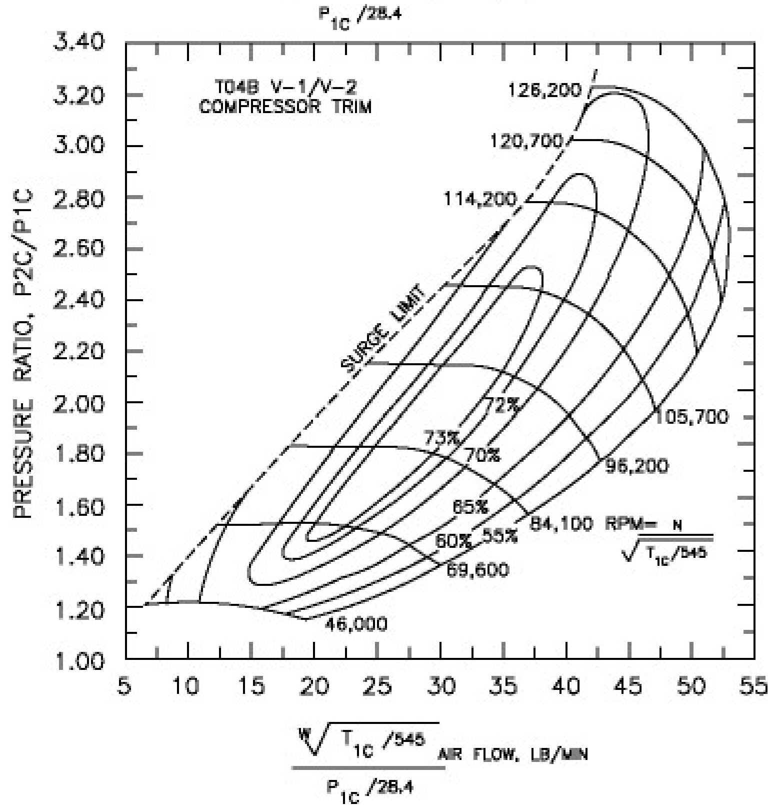


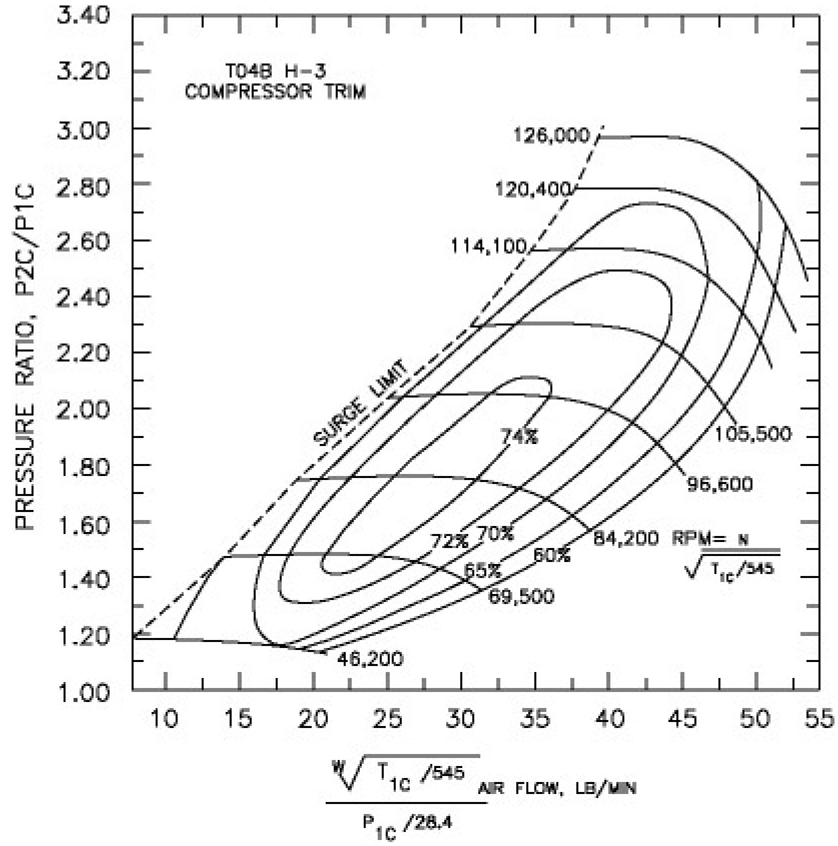
T3
"60" TRIM
SURGE LIMIT
153,700
143,700
132,800
121,900
110,400
96,300
78,800
56,400
74%
70%
65%
RPM= N / √(T1C/545)
PRESSURE RATIO, P2C/P1C
W √(T1C/545) / P1C/28.4 AIR FLOW, LB/MIN

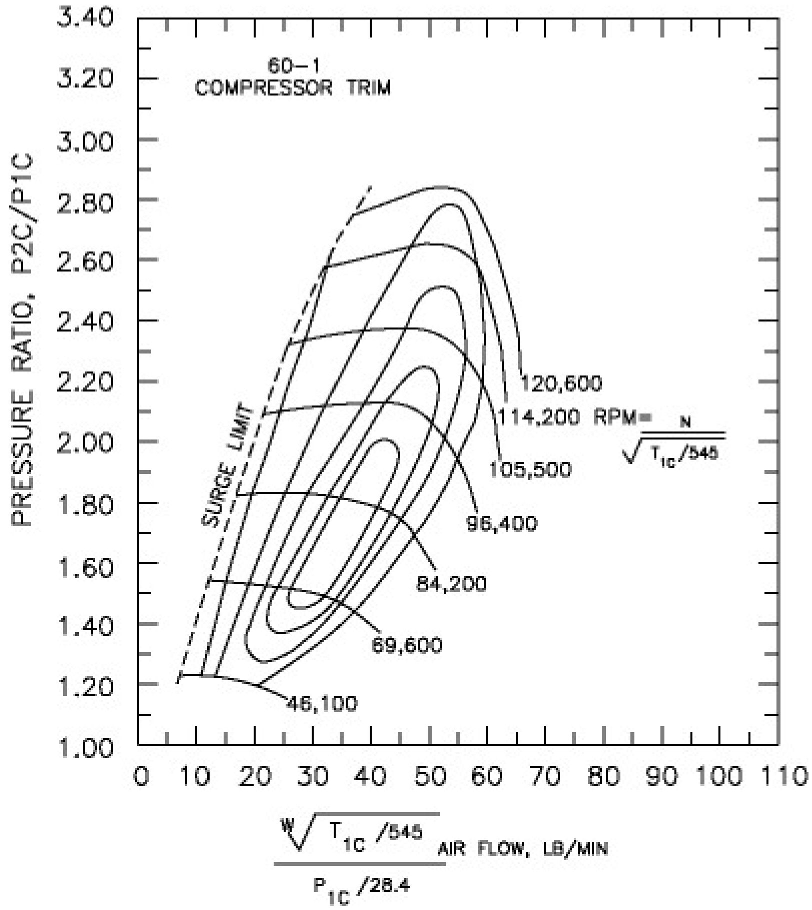


T3 "SUPER 60"
COMPRESSOR TRIM
SURGE LIMIT
153,700
143,700
132,800
121,900
110,400
96,300
78,800
56,400
RPM= N / √(T1C/545)
PRESSURE RATIO, P2C/P1C
W √(T1C/545) / P1C/28.4 AIR FLOW, LB/MIN

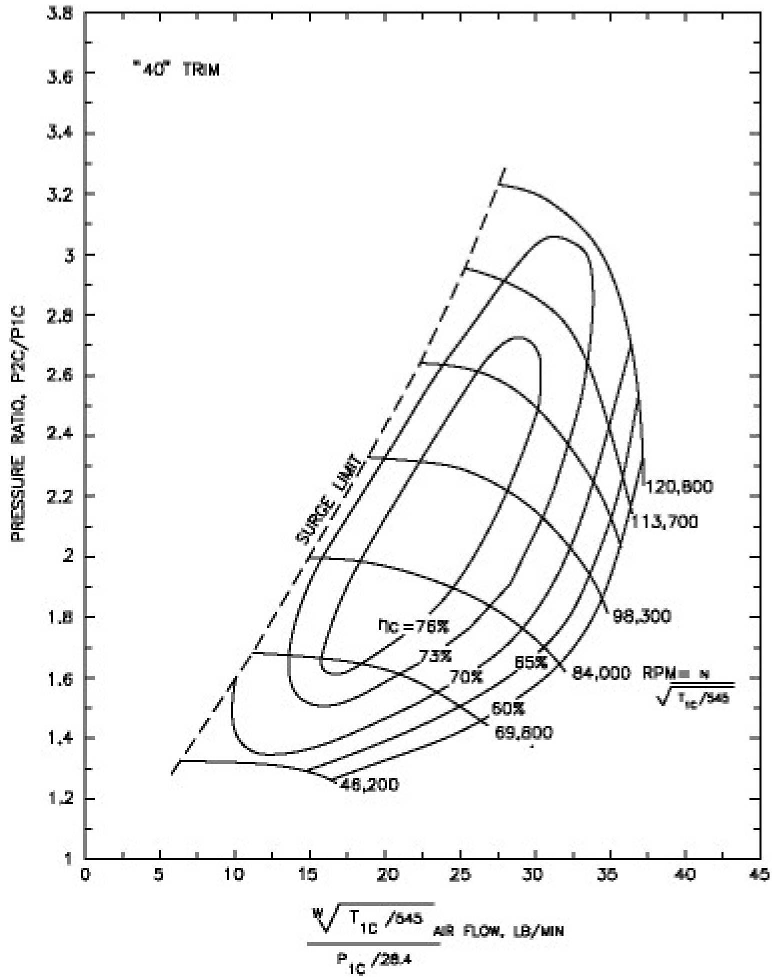

T04B S-3
COMPRESSOR TRIM
SURGE LIMIT
PRESSURE RATIO, P2C/P1C
125,650
120,400
114,050
106,000
96,600
84,200
69,750
46,050 RPM=
73%
72%
70%
65%
60%
50%
AIR FLOW, LB/MIN

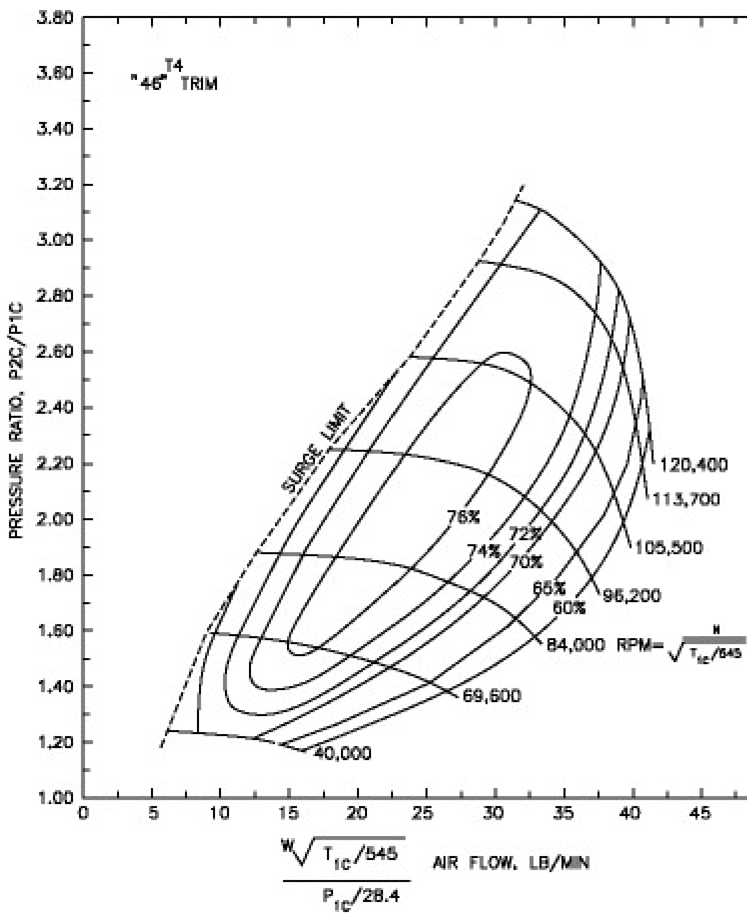

T04B V-1/V-2
COMPRESSOR TRIM
SURGE LIMIT
PRESSURE RATIO, P2C/P1C
126,200
120,700
114,200
105,700
96,200
84,100 RPM=
69,600
46,000
73%
72%
70%
65%
60%
55%
AIR FLOW, LB/MIN


T04B H-3
COMPRESSOR TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
126,000
120,400
114,100
105,500
96,600
84,200 RPM= N / √(T1C/545)
69,500
46,200
74%
72%
70%
65%
60%
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
10
15
20
25
30
35
40
45
50
55
W √(T1C /545) / P1C /28.4 AIR FLOW, LB/MIN

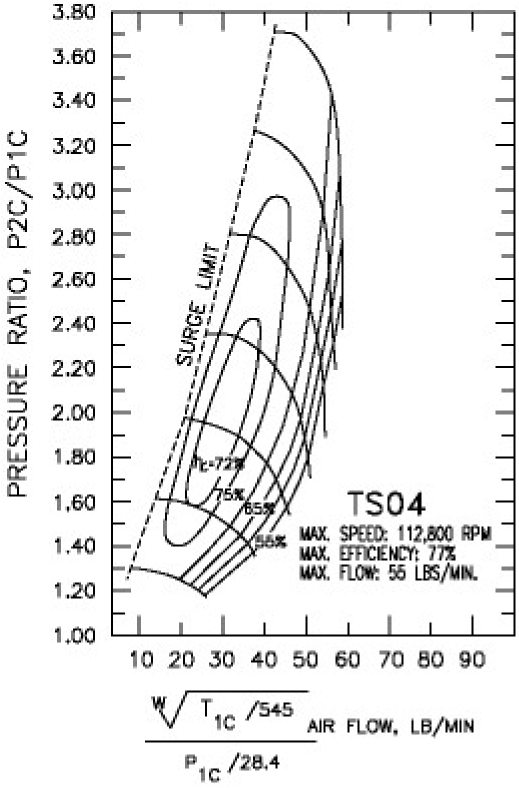


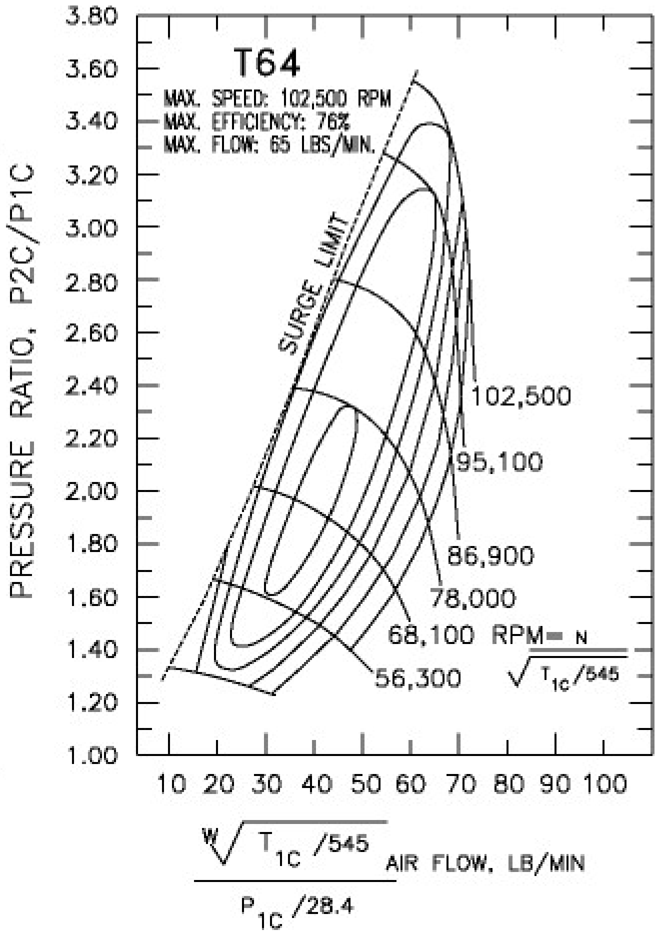
60-1
COMPRESSOR TRIM
PRESSURE RATIO, P2C/P1C
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
SURGE LIMIT
120,600
114,200 RPM= N / √(T1C/545)
105,500
96,400
84,200
69,600
46,100
0 10 20 30 40 50 60 70 80 90 100 110
W√(T1C/545) / (P1C/28.4) AIR FLOW, LB/MIN

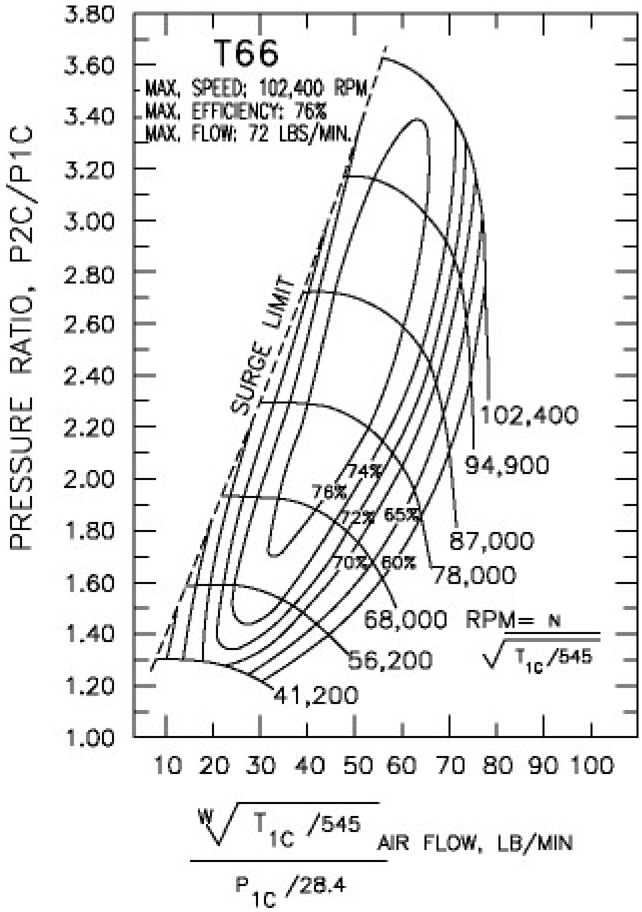

"40" TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
η_C =76%
73%
70%
65%
60%
120,800
113,700
98,300
84,000 RPM= N / √(T1C/545)
69,800
46,200
W √(T1C /545) / P1C /28.4 AIR FLOW, LB/MIN
3.8
3.6
3.4
3.2
3
2.8
2.6
2.4
2.2
2
1.8
1.6
1.4
1.2
1
0
5
10
15
20
25
30
35
40
45

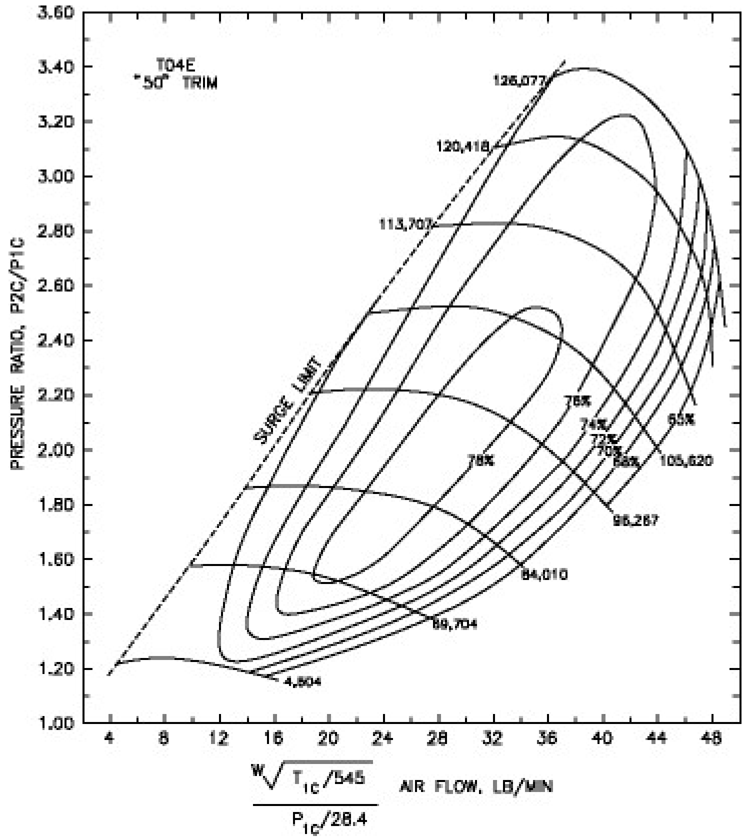


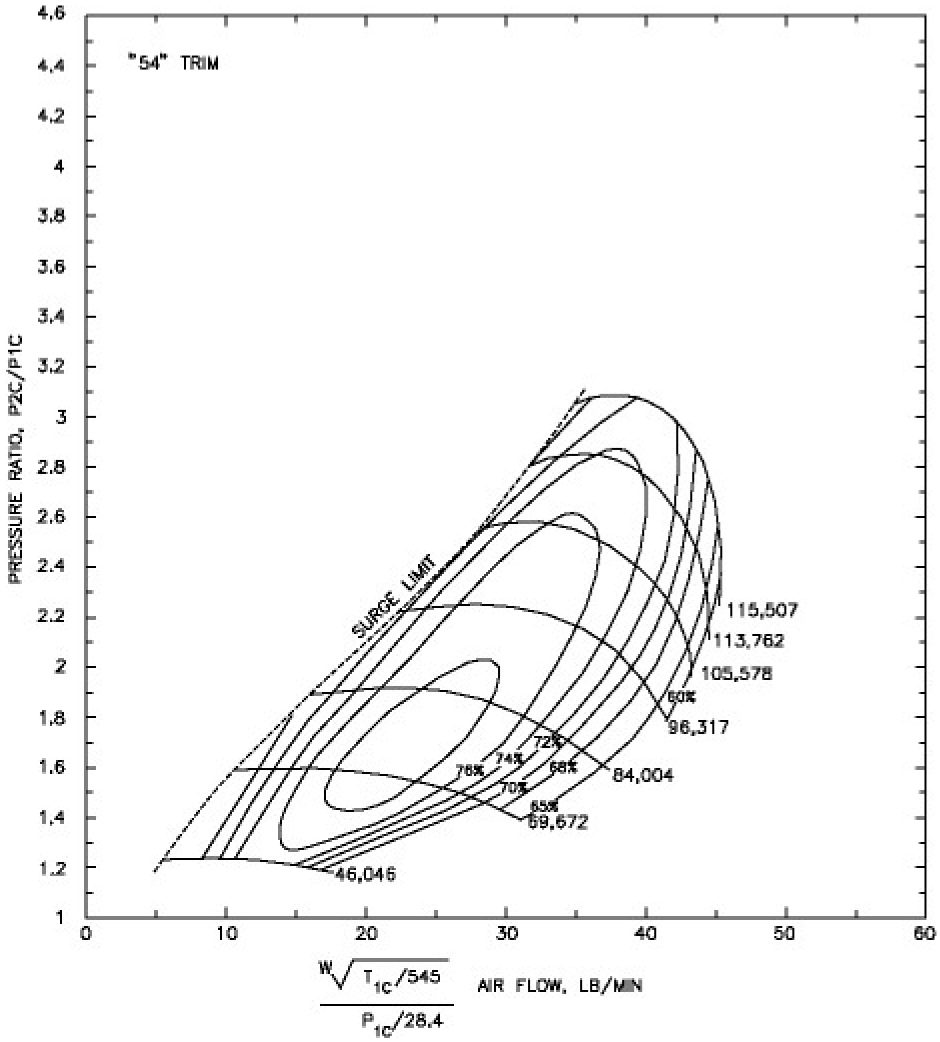
"46" T4 TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
76%
74%
72%
70%
65%
60%
120,400
113,700
105,500
96,200
84,000
69,600
40,000
RPM=
3.80
3.60
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
0
5
10
15
20
25
30
35
40
45
50
AIR FLOW, LB/MIN


TS04
MAX. SPEED: 112,800 RPM
MAX. EFFICIENCY: 77%
MAX. FLOW: 55 LBS/MIN.
SURGE LIMIT
PRESSURE RATIO, P2C/P1C
75%
65%
55%
AIR FLOW, LB/MIN

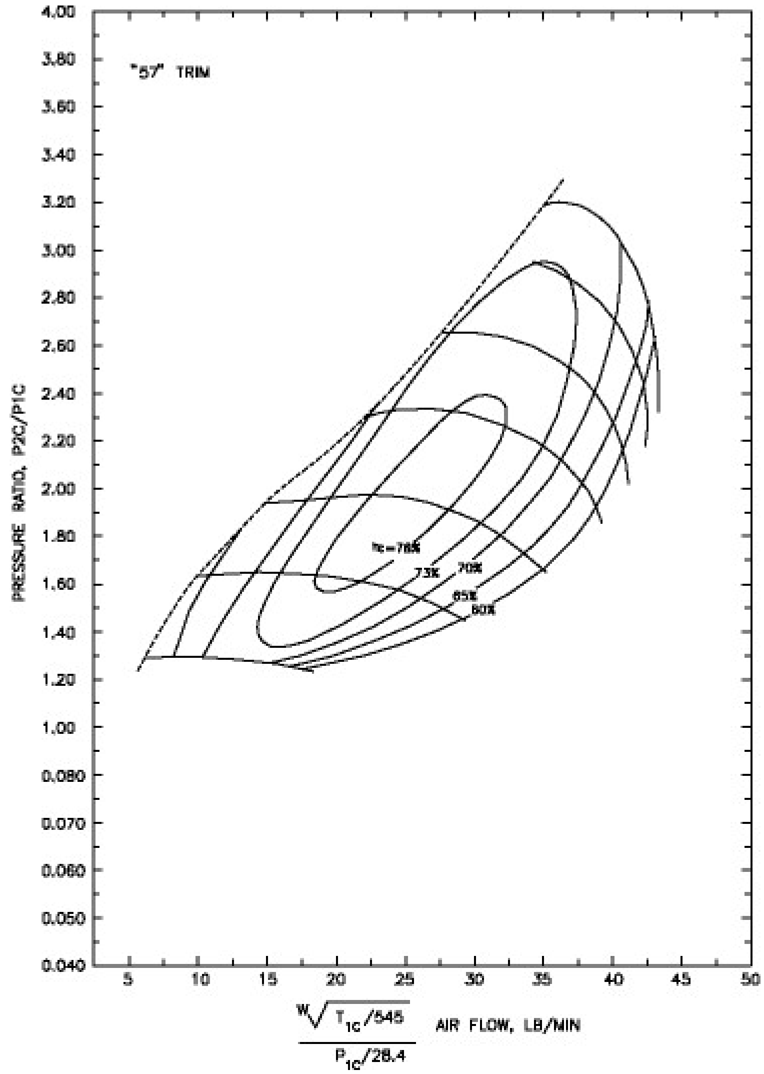

T64
MAX. SPEED: 102,500 RPM
MAX. EFFICIENCY: 76%
MAX. FLOW: 65 LBS/MIN.
SURGE LIMIT
PRESSURE RATIO, P2C/P1C
102,500
95,100
86,900
78,000
68,100 RPM
56,300
AIR FLOW, LB/MIN

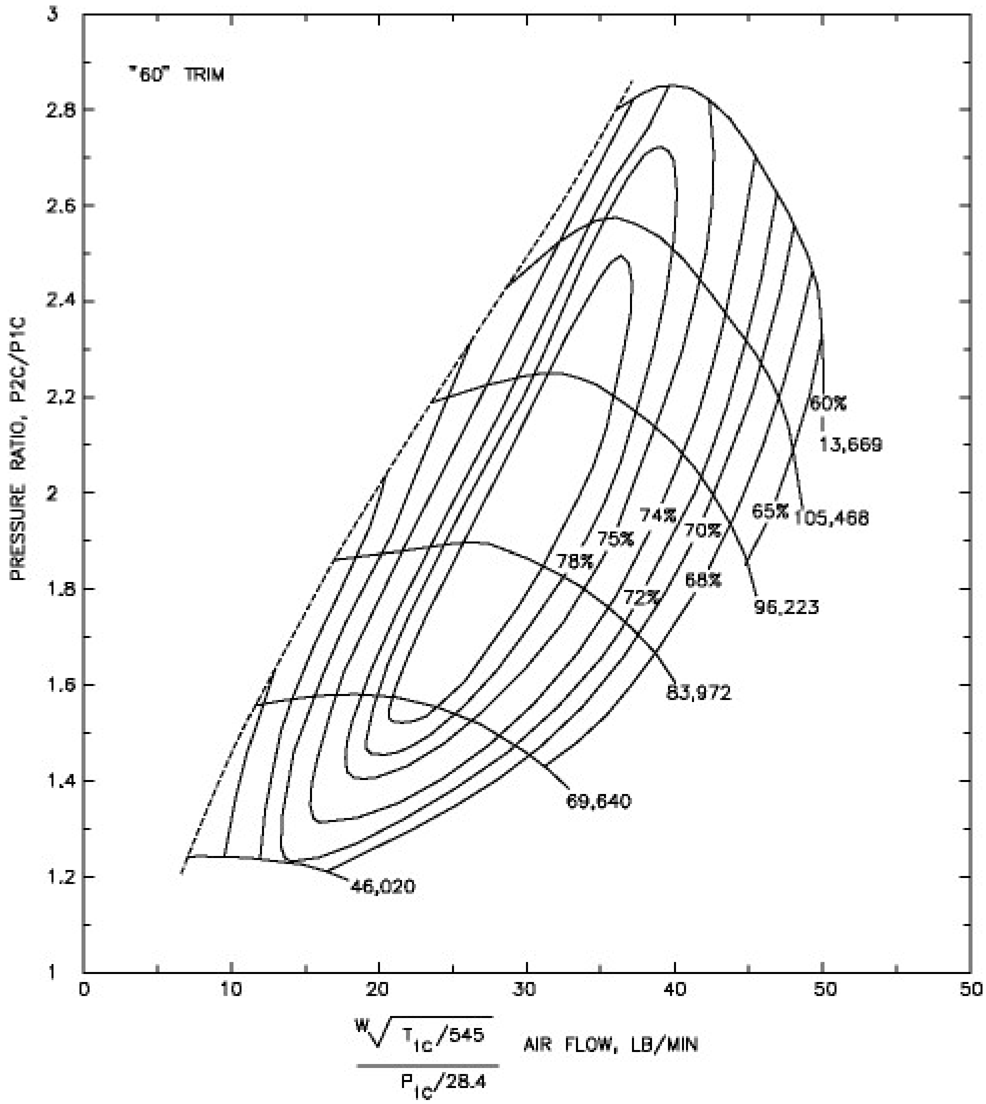

T66
MAX. SPEED: 102,400 RPM
MAX. EFFICIENCY: 76%
MAX. FLOW: 72 LBS/MIN.
SURGE LIMIT
PRESSURE RATIO, P2C/P1C
102,400
94,900
87,000
78,000
68,000
56,200
41,200
74%
76%
72%
65%
70%
60%
RPM= N / √(T1C/545)
W√(T1C/545) / (P1C/28.4) AIR FLOW, LB/MIN

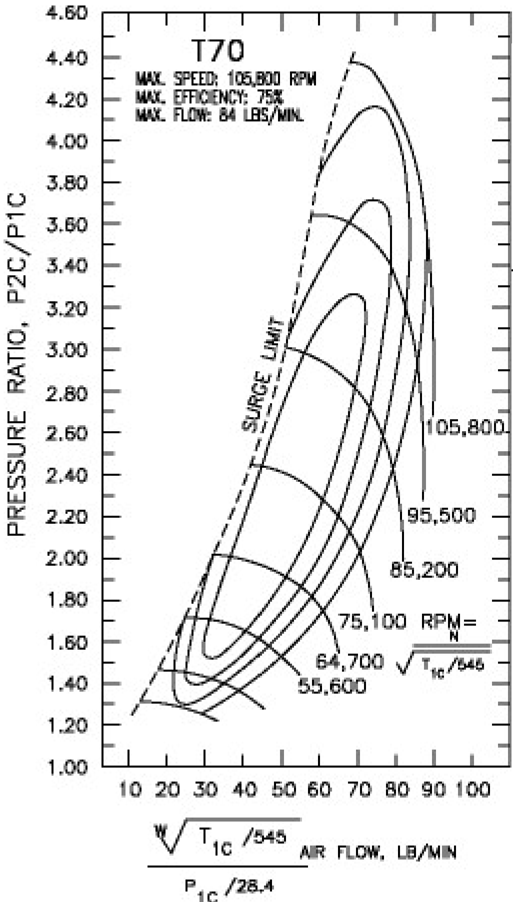

T04E
"50" TRIM
PRESSURE RATIO, P2C/P1C
AIR FLOW, LB/MIN
SURGE LIMIT
126,077
120,418
113,707
105,620
96,267
84,010
69,704
4,604
76%
74%
72%
70%
68%
65%
78%
3.60
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
4
8
12
16
20
24
28
32
36
40
44
48

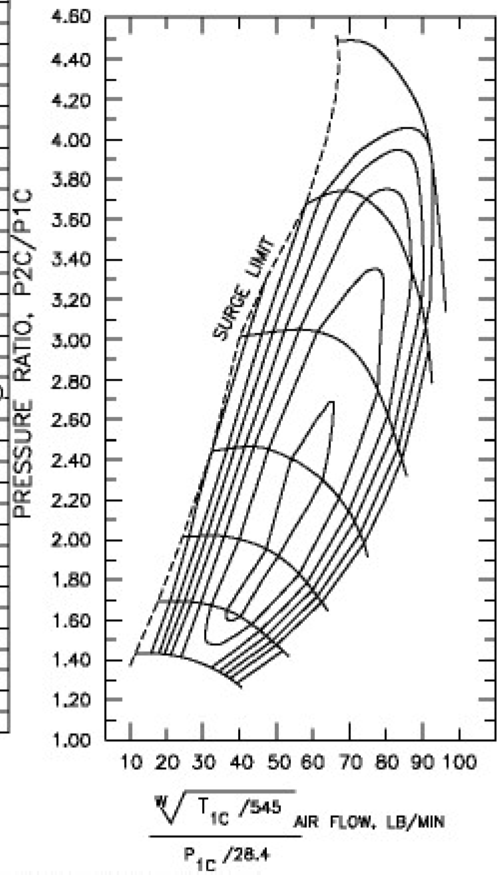


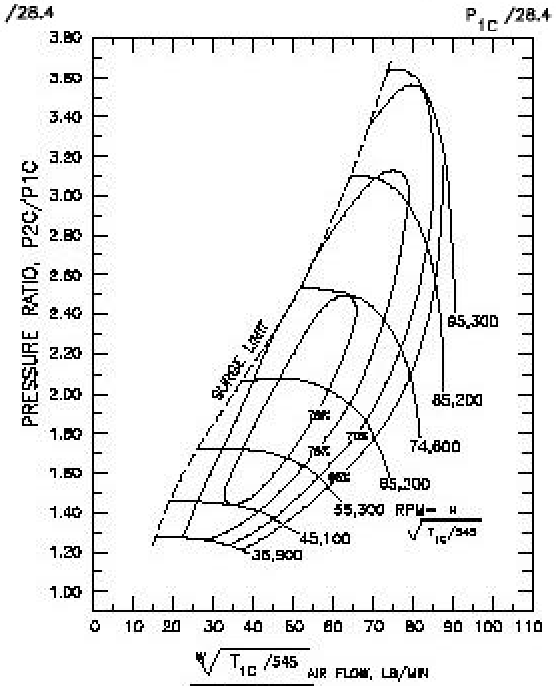
"54" TRIM
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
115,507
113,762
105,578
96,317
84,004
69,672
46,046
80%
78%
76%
74%
72%
70%
68%
65%
W √ T1C/545 / P1C/28.4 AIR FLOW, LB/MIN


"57" TRIM
PRESSURE RATIO, P2C/P1C
4.00
3.80
3.60
3.40
3.20
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
0.080
0.070
0.060
0.050
0.040
5
10
15
20
25
30
35
40
45
50
ηc=76%
73%
70%
65%
60%
W√T1C/545 / P1C/28.4
AIR FLOW, LB/MIN


"60" TRIM
PRESSURE RATIO, P2C/P1C
60%
13,669
105,468
65%
74%
75%
70%
78%
68%
72%
96,223
83,972
69,640
46,020
W √(T1C/545) / P1C/28.4 AIR FLOW, LB/MIN


T70
MAX. SPEED: 105,800 RPM
MAX. EFFICIENCY: 75%
MAX. FLOW: 84 LBS/MIN.
PRESSURE RATIO, P2C/P1C
SURGE LIMIT
105,800
95,500
85,200
75,100
64,700
55,600
RPM= N / √(T1C/545)
W √(T1C/545) / (P1C/28.4) AIR FLOW, LB/MIN


PRESSURE RATIO, P2C/P1C
SURGE LIMIT
W √(T1C/545) / (P1C/28.4) AIR FLOW, LB/MIN


PRESSURE RATIO, P2C/P1C
SURGE LIMIT
95,300
85,200
74,600
65,200
55,300
45,100
35,900
RPM= N / √(T1C/545)
W √(T1C/545) / (P1C/28.4) AIR FLOW, LB/MIN