Comp Turbo Turbochargers

The turbochargers manufactured and sold by Comp Turbo, embody the latest in small turbocharger design technology. The three main components that contribute to the turbocharger overall efficiency and it's performance on the engine are the compressor, bearing system and the turbine.

Compressor Design

Referring now to the compressor component, a primary design objective is to obtain the most mass flow through small diameter wheels, thereby minimizing the inertia of the rotating assembly. The mass flow through the compressor wheel is controlled by the net axial flow area at the inducer inlet. Cutting back alternate inducer vanes opens up the flow area at the base of the vanes and allows the hub diameter to be minimized. Obviously, a large inducer vane outside diameter, along with the small hub, maximizes the net inlet axial flow area. Usually, several inducer vane outside diameters are employed to produce several different flow ranges from a single wheel casting. The inducer vanes are made as sharp as possible along their entrance edges to minimize entrance losses and this contributes to maximizing the net inlet flow area and the flow range of the compressor.

It is usual to limit the inducer vane outside diameter to about 75% of the wheel O.D. to limit the stress at the base of the vanes. Exceeding the 75% limit can increase the vane base stress to values that can exceed the material properties of cast wheels and force the wheel to be machined from a billet. This is an unnecessary expense since 75% inducer wheels made from economical casting material have adequate flow range for essentially all commercial applications. There is no reason to use a full bladed inducer. The evolution of small compressor performance took a giant step forward with the development of wheels that employ alternately cut back inducer vanes.

It is desirable to select a relatively large number of compressor wheel vanes to maximize the pressure ratio capability of a given size wheel. The exit velocity of the compressed air can never reach the exit velocity of the vanes, and this difference is termed wheel “slip”. To illustrate this phenomenon, an approximation of wheel slip can be calculated by using the Stodola equation from the literature:

Wheel slip = 1 – ?/N where N is the number of vanes.
A 14-vane wheel would have a slip factor of 1 – ?/14 = .776.
An 11-vane wheel slip factor would be 1 – ?/11 = .714.

This comparison indicates that the 14-vane wheel will have a significantly higher pressure ratio capability than an 11-vane wheel because of its greater air exit velocity. The even vane number of the 14-vane wheel allows the wheel to have all the advantages of alternate cut back vanes. A wheel with 11 full vanes could always have its flow range and pressure capability increased by adding a vane and alternately cutting back the inducer vanes.

Due to “slip”, the tangential component of the air exit velocity is less than the wheel speed and the relative velocity, W2, dictates the design of a backward curve in the vanes to match the relative exit velocity so that the vane wake loss is minimized. Designing back sweep into the vanes as they near the exit or O.D. improves both the efficiency and the flow range of the compressor.

Consideration of all the foregoing design factors results in the availability of broad range compressors with maximum efficiencies approaching 80%, while still retaining relatively small size to minimize rotational inertia.

Referring to the compressor casing, a re-circulation slot can be designed into the casing located just inboard of the inducer inlet. This feature can produce a lower surge line and broaden the flow range of the compressor at high pressure ratio. The re-circulation slot is an outgrowth of work done at NASA, where the flow range of axial flow compressors was enhanced by several types of tip treatment. The re-circulation slot has been a useful addition to small compressor design technology.

Bearing System Design

Referring now to the turbocharger bearing system, Comp Turbo turbochargers utilize the latest in high-speed ball bearing technology. The acceleration rate of a turbocharger is a function of the rotor inertia and the friction losses in the bearing system. Conventional commercial turbochargers use floating sleeve bearing systems that are a result of years of experimental development. The floating sleeve bearings have an inner and outer oil film fed by lube oil under pressure from the engine’s lubricating oil system. They must also employ a separate stationary thrust bearing that is fed lube oil under pressure from the engine. The friction loss attributed to a stationary thrust bearing is proportional to the fourth power of the radius and can amount to several horsepower at the high speed at which turbochargers operate.

The oil films in conventional floating sleeve bearings have significant viscosity that produces appreciable friction losses due to oil film shear when the turbocharger rotor is accelerated and running at high speed. The friction losses in the sleeve bearing systems and in the stationary thrust bearings result in mechanical efficiencies in the middle 90% range in conventional turbochargers.

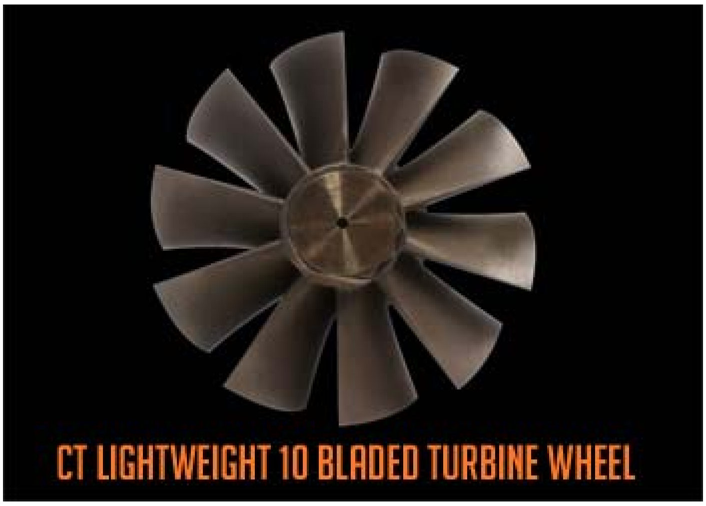
The Comp Turbo turbochargers use a ball bearing system that does not need a separate thrust bearing since the ball bearings carry both the radial load and the axial thrust loads. There is little or no oil film shear in ball bearings that operate with rolling friction only so that Comp Turbo turbochargers accelerate much faster than conventional turbochargers that use sleeve bearing systems. The Comp Turbo bearing system is a proprietary design that is unique in the industry. It utilizes full compliment angular contact ball bearings with ceramic balls. Compared with steel balls, ceramic balls in ball bearing have a number of advantages.

According to a prominent ball bearing manufacturer, bearing service life is two to five times longer than steel balls, they run at lower operating temperatures and allow running speeds to be as much as 50% higher. Also, since the surface finish of ceramic balls is almost perfectly smooth, they have lower friction losses and lower vibration levels. And, since there is less heat buildup during high-speed operation, they exhibit reduced ball skidding and have a longer fatigue life.

All these characteristics make ceramic ball bearings ideal for use in turbochargers where they must operate at very high speeds and survive in a high temperature environment. The full compliment bearings do not use a cage to position the balls and this additional feature, combined with the ceramic material, provides a combination that has minimal friction losses. The mechanical efficiency of Comp Turbo turbochargers that use ceramic ball bearings can approach the high 90% range and this contributes to rotor acceleration rates that have been shown to be faster than competition.

In the proprietary Comp Turbo ball bearing system, the angular contact bearings are mounted in an elongated steel cylinder that is free to rotate in the bearing housing. The outside diameter of the cylinder is fed with lube oil and this outer oil film provides a cushion against shock and vibration. Two angular contact bearings are mounted in tandem on the compressor end of the cylinder in an arrangement that carries rotor thrust in both axial directions. A single angular contact bearing is mounted under pre-load on the turbine end of the cylinder and is free to move axially with shaft elongation when heat is conducted down the shaft from the hot turbine wheel. The elongated steel cylinder containing the angular contact bearings represents the complete bearing system and can be inserted and/or removed as an assembly, making the Comp Turbo turbocharger fully serviceable and rebuildable.

Turbine Design



The Comp Turbo turbine wheels are a unique design in that they have vanes that are constant in outside diameter from inlet to exit. This design feature maximizes the flow capacity of a given size wheel and allows the use of reasonably small turbine wheels on large-size engines. One of the largest losses in small turbine wheel design is the leaving gas velocity, which is unrecoverable energy and is dissipated in the atmosphere when the exhaust gas leaves the turbine casing. The Comp Turbo full-bladed turbine wheels have minimal leaving velocity due to the large exit area, thus their leaving losses are minimized, leading to higher turbine efficiency and greater flow range than contoured turbine wheels. Conventional twin flow and undivided turbine casings are available to match different engine exhaust manifold systems.

Application Technology

Racing applications require turbochargers that build boost pressure as rapidly as possible, thus allowing the engine to develop high torque at low engine speed and with boost capability that can produce very high maximum power output. Comp Turbo turbochargers do exactly that. For example, when mounted on one dragster, the Comp Turbo turbocharger produced 1.7 bar boost in two tenths of a second and developed 650HP ready for takeoff. Now, that's phenomenal response and very impressive.

In street applications, the acceleration rate of a vehicle equipped with a Comp Turbo turbocharger is enhanced and moves the engine out of inefficient operating regimes more rapidly. An improvement in number of gallons of fuel used is the usual result when a vehicle is accelerated faster. Under steady-state operation, the lower HP losses in the Comp Turbo turbocharger ball bearing system means more power is available to the turbocharger compressor, which results in higher intake manifold pressure. In most cases, higher boost pressure can make an additional contribution to improving engine fuel consumption.

Comp Turbo can supply turbochargers with various compressor and turbine wheel trims to tailor their performance to exactly match specific engine application requirements, whether they be racing, street, off-highway, or stationary. Models available are described and listed in this catalog. Comp Turbo also offers a rebuild service for conventional turbochargers as well as a conversion service that can substitute a ball bearing system for conventional sleeve bearings in many commercial turbocharger models. The sale of turbocharger spare parts and accessories, such as waste gates, is also available.

COMP TURBO TECHNOLOGY BULLETIN NO. 1

TURBOCHARGER THRUST BEARING COMPARISON

The floating sleeve bearing systems used in many current turbochargers must include a stationary thrust bearing that carries rotor thrust loads transmitted to it from collars mounted on the rotor shaft. The design of the stationary thrust bearing is the result of many years of development to arrive at configurations that were capable of carrying the rotor thrust loads at high speeds without failure. Their capability to function satisfactorily at the very high speeds and high boost pressures required of turbochargers used in racing applications might be considered borderline.

The stationary thrust bearings are the source of the largest horsepower loss in the floating sleeve bearing systems. A formula for calculating the power loss in a stationary thrust bearing can be derived from basic theory resulting in the following expression for HP.

HP = .1363

where:

- N = shaft speed – RPM
- R2 = outside radius of thrust surface – in.
- R1 = inside radius of thrust surface – in.
- C = oil film thickness – in.

(The derivation of the formula can be furnished upon request.)

Note that the HP loss is a function of the fourth power of the radii.

The dimensions of a typical stationary thrust bearing in a turbocharger running at 90,000 RPM producing a boost pressure of approximately 45 psig might be as follows:

N = 90,000 RPM R2 = .555" R1 = .338" C = .0005 in2

The calculated HP loss of the loaded side of the thrust bearing is:

HP1 = .1363 = 1.806 HP

Since the thrust bearing has an unloaded side with an assumed clearance of .004", the loss attributed to this side is:

HPUN = .1363 = .374 HP

Thus, the total loss of this thrust bearing is 1.806 + .374 = 2.18 HP which is significant. Since the loss is proportional to the square of the speed at 100,000 RPM, the HP loss goes up to 2.69 HP. The mechanical efficiency of the turbocharger using floating sleeve bearings and the typical thrust bearing described above would be approximately .969.

The Comp Turbo turbochargers use angular contact, full compliment ball bearings with ceramic balls. There is no cage necessary to space the balls, and any friction loss attributed to a cage is eliminated. In the patented TRIPLEX CERAMIC™ ball bearing system, the rotor thrust is carried by a single angular contact bearing that has a static thrust load capacity of 607 lbs. (Barden Catalog). Since the thrust loads in turbochargers usually range between 50 and 100#, the single angular contact bearing can carry these thrust loads easily within its load-carrying capacity, making it ideal for use in racing turbochargers. The TRIPLEX CERAMIC™ ball bearing system has a calculated loss of approximately 1 HP, making the mechanical efficiency approximately .991.

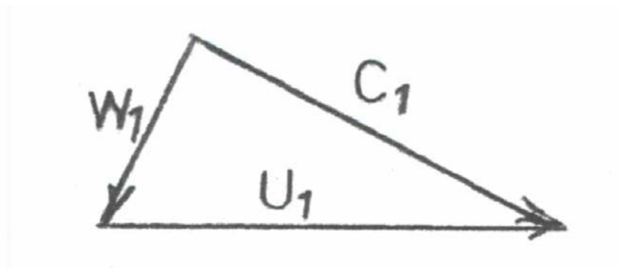
The power loss in a turbocharger bearing system has a direct effect on the rotor acceleration. The turbocharger with the highest mechanical efficiency will accelerate the fastest, such as the Comp Turbo turbochargers with the TRIPLEX CERAMIC™ ball bearings. These turbochargers also produce a higher boost pressure since the high mechanical efficiency results in more turbine power available to drive the compressor.

Experience the acceleration and the outstanding performance of Comp Turbo Technology racing turbochargers!

COMP TURBO TECHNOLOGY, INC.
TECHNICAL BULLETIN NO. 2

EXHAUST MANIFOLD DESIGN FOR TURBOCHARGED ENGINES

The configuration of the exhaust manifolds used on turbocharged engines can have a significant effect on the performance of the engine. The turbine casing of the turbocharger has a relatively small throat area in its nozzle section in order to generate a high exhaust gas velocity at the entrance of the turbine wheel. This high entrance velocity is necessary to enable the turbine to generate the power needed to drive the compressor wheel. A typical entrance velocity triangle is illustrated below:



where:

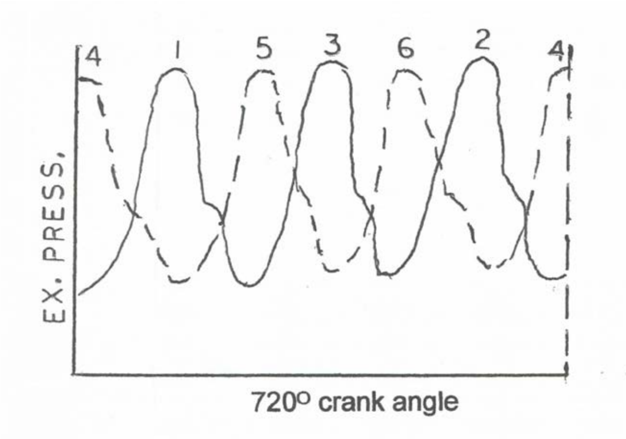
U1 = turbine wheel tip speed
C1 = exhaust gas entrance velocity
W1 = gas velocity as seen by the rotating wheel

The small throat area of the turbine casing presents a restriction to the exhaust gas flow from the cylinders and results in a high pressure in the exhaust manifold ahead of the turbine casing. Since the engine pistons in 4-cycle engines must act against this pressure when evacuating the cylinders, the level of pressure in the exhaust manifold causes a parasitic loss in engine power.

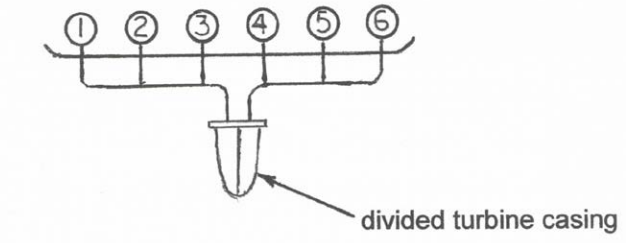
If all the engine cylinders exhaust into a common manifold, the pressure in the manifold will remain at a high level for all the engine's pistons when they are pushing remaining exhaust gases out of the cylinders on their exhaust upstroke.

Thus, it is desirable to separate the exhaust manifold into several branches so that no successive exhaust pulse enters into a common branch. For example, in an in-line 6-cylinder engine that has a firing order of 1-5-3-6-2-4, it is advantageous to divide the manifold into two branches, allowing cylinders 1, 2 and 3 to exhaust into one branch and cylinders 4, 5 and 6 to exhaust into the other branch. This allows the pressure level from cylinder number 1 to fall to a low level before cylinder number 3 exhausts into that branch, etc. for each remaining cylinder. The result of this manifold division is a low average pressure in the manifold branches that reduces the pumping loss of the engine, increases power output, lowers fuel consumption, and reduces smoke on acceleration.

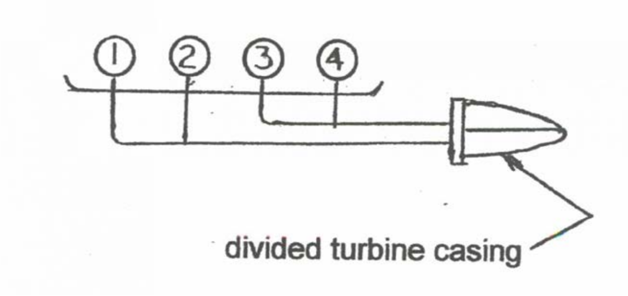
A schematic diagram that illustrates the exhaust pressure variation in a 6-cylinder engine divided manifold follows:



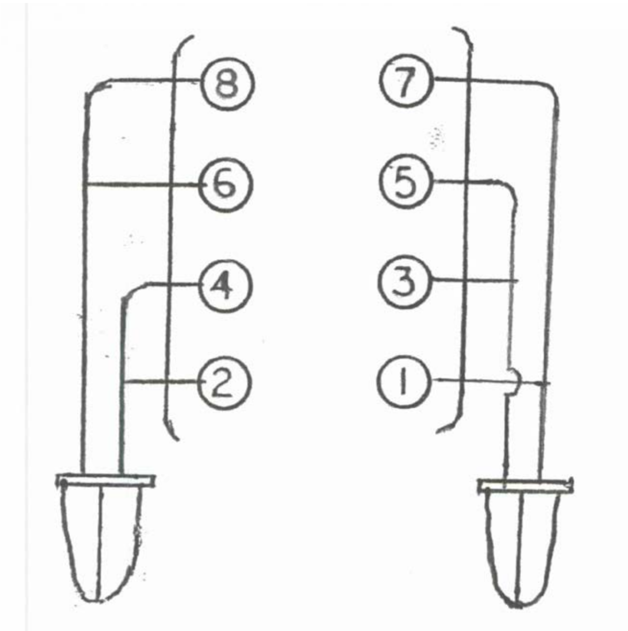
By separating the exhaust pulses by the manifold division, the pressure in each branch is allowed to fall to a low value before the next cylinder exhausts in that branch. The solid line in the above diagram represents the pressure in the 1-2-3 cylinder branch whereas the dotted line represents the pressure in the 4-5-6 cylinder branch. This manifold division is illustrated below with each branch connected to one opening in a divided turbine casing.



In a 4-cylinder engine with a firing or4der of 1-3-2-4, the cylinders should be divided with cylinders 1 and 2 in one branch, and cylinders 3 and 4 in another. This is schematically illustrated below:



In the case of a V-8 engine, the division of the manifolds becomes more complicated. There are a number of different firing orders that can be successfully used in 4-cycle V-8 engines. One commonly used firing order is 1-8-4-3-6-5-7-2. These cylinders should be divided into four branches using two turbochargers with divided turbine casings; one on each side of the engine. This can be illustrated as follows:



The above manifold division will work equally as well with several other firing orders. These are:

1-8-7-2-6-5-4-3
1-5-4-8-7-2-6-3
1-6-2-5-8-3-7-4
1-2-7-8-4-5-6-3

If peak engine performance is desired or required, then divided exhaust manifolds are an absolute necessity; either fabricated or cast. Compared with undivided manifolds, properly dividing the exhaust manifolds used on turbocharged engines will make a very noticeable improvement in engine and vehicle performance.

COMP TURBO TECHNOLOGY, INC.
TECHNICAL BULLETIN NO. 3

How a Turbocharger Works

Approximately 30% of the fuel energy put into an engine to produce horsepower is normally wasted out the engine exhaust pipe. The purpose of the device called a turbocharger is to capture some of this wasted energy and put it back into the engine in the form of high engine intake manifold pressure.

The turbine component of the turbocharger consists of only two parts; the turbine casing and the turbine wheel. The turbine casing is usually a ductile iron casing in the shape of a volute that completely surrounds the radial turbine wheel. The inlet section of the casing is a convergent passage that increases the velocity of the exhaust gas to a very high level before it is distributed around the periphery of the turbine wheel by the volute shape of the casing. The high velocity exhaust gas, that is also at a very high temperature, expands down to near ambient pressure as it flows inward through the turbine wheel passages and on out the exhaust pipe, usually through a muffler. This expansion process causes the turbine wheel to rotate and generates the horsepower needed to drive the compressor wheel mounted on the same shaft. The turbine casing is mounted on the engine exhaust manifold and supports the turbocharger on the engine.

The compressor component of the turbocharger consists of three parts; the compressor wheel, the compressor casing, and the compressor back plate. The compressor wheel, being forced to rotate by the turbine wheel, is designed to draw in air from the atmosphere, usually through an air cleaner, and performs two functions. First, the air that is drawn in from the atmosphere flows radially outward through the converging passages between the blades of the wheel. The flow through the converging passages causes the air pressure to increase and, concurrently, accelerates the air to a very high velocity at the wheel periphery. This high velocity air exits the wheel tangentially and enters the diffuser section of the compressor.

The diffuser is formed outboard of the wheel by the compressor casing wall and the wall of the compressor back plate. These two walls form a narrow parallel passage that accepts the high velocity air exiting the compressor wheel and directs the air into the compressor casing, that is usually a volute-shaped casting. The airflow through the radially increasing area of the diffuser is slowed down from its high wheel exit velocity, and this deceleration results in a second increase in the pressure of the air. The outlet of the compressor casing volute is connected to the intake air system of the engine.

The compression process that raises the air pressure in two stages as the air passes through the compressor wheel and diffuser, also causes an increase in the air temperature. This increase is termed the "heat of compression" and is a normal result of a compression process.

The compression process in a turbocharger is usually defined by the ratio of the ambient pressure (barometer), divided into the compressor outlet pressure. That is:

$$P_R = 1 + \frac{P_2}{P_1}$$

WHERE: PR=PRESSURE RATIO
P1=BAROMETRIC PRESSURE
P2=OUTLET GAUGE PRESSURE

The compressed air temperature rise versus pressure ratio for a range of compressor efficiencies is plotted on the following chart, using an initial ambient air temperature of 80F.

It is interesting to note from the chart that a pressure ratio of 2.5 will result in an air outlet temperature of close to 300 F, if the compressor efficiency is 72%. Rather than supply an engine with 300 F intake air, an air-to-air heat exchanger is usually employed in order to reduce the compressed air temperature to as low a value as possible before it is introduced into the engine cylinders. An illustration of an air-to-air after-cooled engine is illustrated in the diagram that follows.

Removal of most of the heat of the compression by the use of n air-to-air after cooler increases the engine intake air density, which allows more fuel to be burned in the engine cylinders, generating a higher level of output horsepower.

Summarizing in brief, the turbocharger turbine uses energy in the engine exhaust to drive the air compressor mounted on the same shaft. The turbocharger compressor provides the engine with high pressure air, enabling the engine to produce more horsepower that it would be capable of if it were naturally aspirated rather than turbocharged.

Another important function of the turbocharger is its ability to compensate for ambient air density loss as the engine is operated at high altitude. It is usual to de-rate a naturally aspirated engine's horsepower by 3% for each 1000 ft.. altitude. The turbocharger compensates for the ambient air density loss by increasing its operational speed as the engine is taken to higher altitudes than sea level. It is usual to expect sea level performance from a turbocharged engine at higher altitudes with no necessity for de-rating the engine's horsepower.

Finally, engine fuel consumption is improved since the turbocharger utilizes energy from the engine's exhaust gas that is normally wasted. An engine's horsepower output can be more than doubled by turbocharging. The cost of a turbocharger is small compared with all the advantages of turbocharging

Comp Turbo Technology Tech Bulletin No. 4

Matching A Turbocharger To An Engine

The primary objective in matching a turbocharger to an engine is the selection of a compressor that covers the engine air requirement at the highest possible compressor efficiency. The engine air requirement must be either known or estimated before a match can be made. If the air requirement is not known, a rough estimate can be made using the following equation:

$$Q = V_E \times Q_D \times P_r \times \frac{(540^o)}{T_3}$$

Where:

Q = compressor inlet volume flow - CFM
V_E = engine volumetric efficiency
Q_D = engine displacement volume flow - CFM
P_r = estimated total pressure ration
540°= ambient air temperature = 80°F + 460° = 540°R
T_3 =intake manifold air temperature - °R

The volumetric efficiency V_E of a turbocharger engine is usually assumed to be 1.0 for a gasoline engine that has a reasonably low allowable boost pressure. Commercial diesel engines are boosted to higher levels so that a volume efficiency of 1.10 can be assumed. For racing engines that are boosted to very high levels, the volumetric efficiency should be assumed to be 1.25 or higher for very highly rated engines.

Since many compressor performance maps are plotted using inlet air mass flow, W in pounds per minute, Q in cubic feet per minute can be converted to W in pounds per minute by multiplying Q by an ambient air density of .0735 pounds per cubic foot.

The displacement airflow Q_D in CFM can be obtained as follows:

$$Q_D = \frac{D_E}{1728} \times \frac{RPM}{2} \quad \text{for a 4-cycle engine}$$

Q_D = displacement airflow - CFM
D_E = engine displacement - cu.in.
RPM = rated engine speed

P_r is obtained by dividing the compressor outlet pressure by the inlet pressure. Total pressure values are usually used when plotting compressor maps, which means that the compressor outlet pressure is the static outlet pressure plus the outlet air velocity converted to pressure. The inlet total pressure is the barometric pressure minus the inlet vacuum, converted to pressure, measured at the compressor wheel inlet.

A rough estimate, however, of the pressure ratio can be made by estimating the intake manifold pressure needed to develop the rated engine horsepower and dividing it by the barometric pressure at sea level.

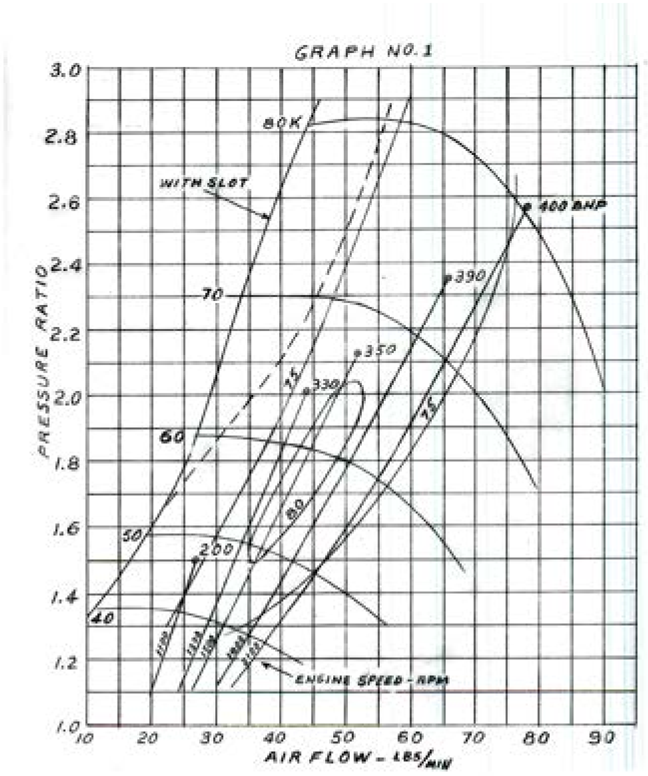
The value of the intake manifold temperature, T_3, can be obtained from the graph of temperature versus pressure ratio presented in our Bulletin No. 3, if the engine is not aftercooled. If the engine is equipped with an air-to-air aftercooler, the value of T_3 can be estimated as 120°F for an ambient air temperature of 80°F and an assumed aftercooler effectiveness of 80%. The value of T_3 for substitution in the above equation must be converted to absolute temperature by adding 460°.

Once the rated engine pressure ratio and inlet airflow have been estimated, a compressor performance map can be selected to match these values at a reasonably high compressor efficiency. This point must be far enough out on the compressor flow range to allow for a reasonable torque rise (for example 20%) as the engine speed is reduced at full throttle. The maximum compressor efficiency should occur in the middle of the engine operating range.

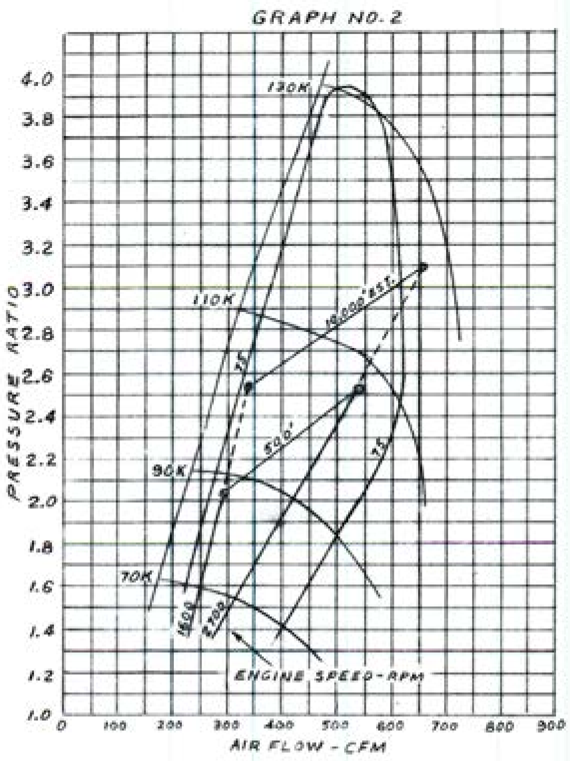
Examples of correctly matched compressors to specific engine air requirements are shown as follows:

Graph No 1. This Graph shows the performance of a compressor with a 3.8" diameter compressor wheel matched to the air requirement of an 855 cu.in. diesel engine with a 20% torque rise. Note that the engine air requirement stays within the 75%compressor efficiency envelop and utilizes the maximum efficiency of 78% in the middle of the engine air requirement range.

This compressor map also shows the effect on the compressor surge line by adding the well known recirculation slot in the compressor casing just inboard of the compressor wheel inducer inlet. The dotted line shows the surge line of the compressor without the recirculation slot. In this case, due to the broadening of the compressor flow range at high speeds, the recirculation slot might prevent the torque peak turbocharger speed from crossing the surge line when the engine is operated at high altitudes.

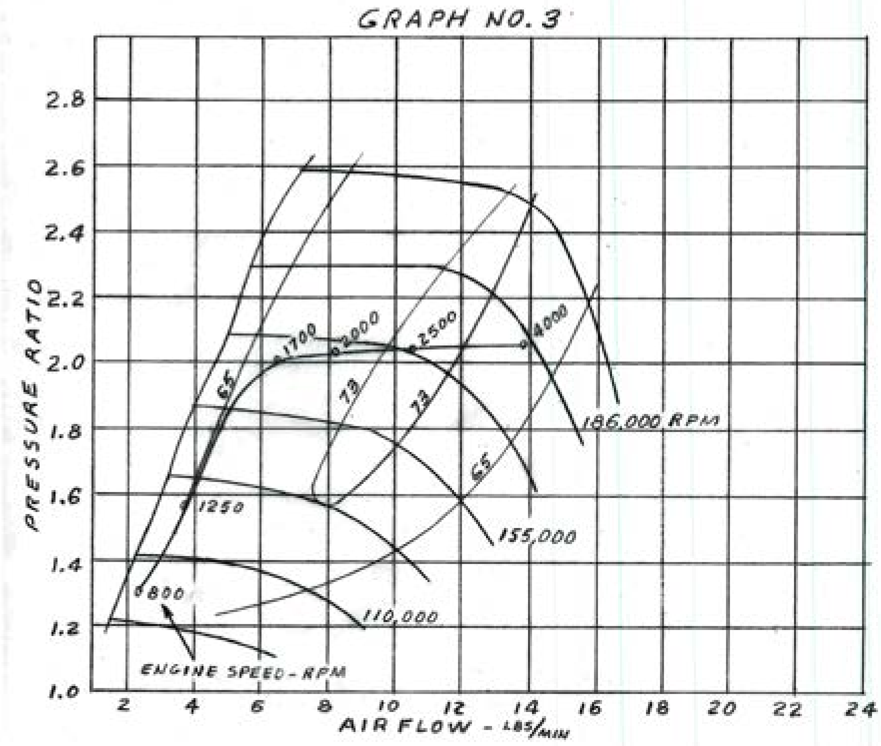


Graph No. 2. This graph shows the performance of a compressor with a 2.9" diameter compressor wheel properly matched to the air requirement of a 5.9 liter diesel engine . Here again, the engine air requirement stays well within the 75% compressor efficiency envelop at sea level. The estimated increase in turbocharger speed as the engine is operated at altitudes above sea level is indicated on the graph. Since the ambient air pressure decreases as altitude increases, the turbocharger turbine has the benefit of a lower back pressure at high altitudes and the turbine can develop more horsepower due to a greater expansion ratio. This results in the turbocharger operating at a higher speed and allows the compressor to deliver a greater volume of the less dense ambient air to the engine cylinders. The higher turbocharger speed at altitudes higher than sea level compensates for the drop in air density and allows the engine to be rated at sea level power rather than be de-rated as it is operated at altitudes higher than sea level.



Graph No. 3. Passenger car engines need to develop high torque at low engine speed to facilitate fast acceleration of the vehicle. Thus, the matching of a turbocharger to a passenger car engine entails the use of a turbocharger that produces high boost at low engine speeds. Graph No. 3 shows the match of a compressor with a 1.96" diameter compressor wheel that has a flow range broad enough to cover the air requirement of a passenger car engine.

Since the compressor is forced to provide high boost at the low engine speeds, it would exceed its maximum allowable operating speed at high engine speeds unless it is prevented from doing so by providing an exhaust gas bypass valve in the turbine casing. The bypass valve, or waste gate , limits the turbocharger speed and holds the boost pressure nearly constant from torque peak to full rated engine speed. In Graph No. 3, the boost level is nearly constant from 1700 RPM to 4000 RPM engine speed by the use of a waste gate that bypasses exhaust gas around the turbine wheel and limits the turbocharger speed over the high speed range of the engine.



The matching of the turbine component of the turbocharger consists of selecting a turbine casing size that will operate the turbocharger at a speed that produces the intake manifold pressure required to reach the rated horsepower of the engine. Several turbine casing sizes are usually tested on the engine to find a size that operates the turbocharger at the desired boost level without imposing an undesirable back pressure on the engine which would have a negative effect on fuel consumption.

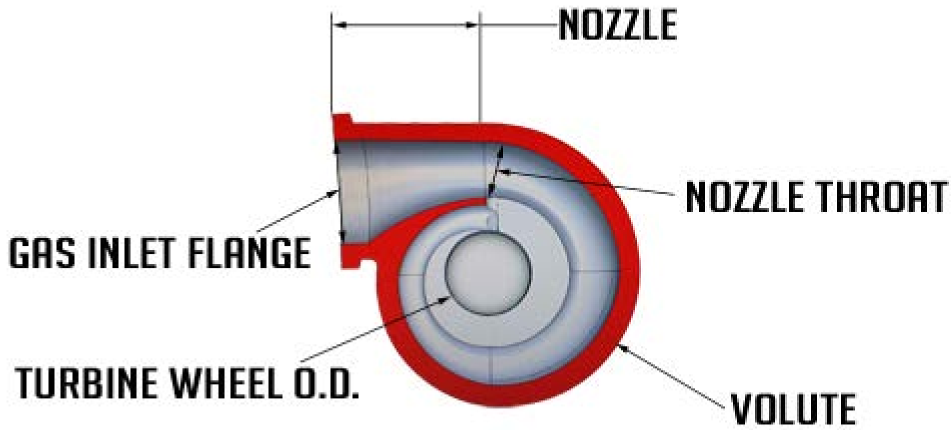
For additional information about matching a turbocharger to engines of various types, contact the Comp Turbo Technology, Inc. Technical Department.

Comp Turbo Technology Tech Bulletin No. 5

The Significance of The Turbine Housing A/R

Turbocharger turbine casings perform the function of increasing the engine exhaust gas velocity and then distributing the high velocity exhaust gas around the periphery of the turbine wheel. In order to perform these functions, the design of the turbine casing must include an initial nozzle section to increase the exhaust gas velocity, followed by a volute section to distribute and maintain the high velocity of the exhaust gas around 360° of the turbine wheel outside diameter.

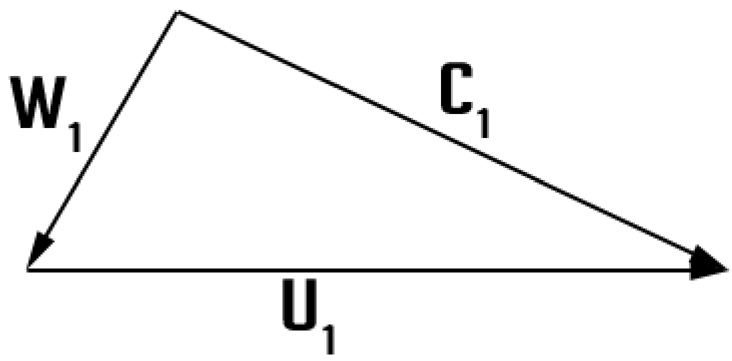
A cross section of a typical turbine casing is diagrammatically illustrated below:



The turbine inlet flange is connected to the engine exhaust manifold and mounts the turbocharger on the engine. As can be seen in the above diagram, the nozzle section of the casing decreases in cross sectional area from the inlet flange to the nozzle throat. The nozzle throat area, in square inches, perpendicular to the gas flow is the "A" of the casing A/R. The radius, in inches, to the center of the throat area is the "R" of the casing A/R.

Divided turbine casings are designed the same as undivided casings: i.e., they both have nozzle sections to increase the exhaust gas velocity and volutes to distribute the high velocity gas around the turbine wheel outside diameter. The divided casings have parallel gas passages created by a central dividing wall that extends from the Turbine inlet flange and continues around 360° of the volute. The divided casings also have an A/R designation where the "A" is the sum of the areas of the two parallel nozzle passages at their throats, at the termination of their nozzle sections.

A velocity diagram at the entrance of the turbine wheel is shown below:



C_1 is the exhaust gas velocity generated by the nozzle section of the turbine casing. U_1 is the tip velocity of the turbine wheel and W_1 is the relative gas velocity entering the turbine wheel blades.

A large area, A, at the nozzle throat produces a lower value of exhaust gas velocity and results in a lower value of C_1 entering the wheel. Thus, the turbocharger will be running at a lower RPM, the value of U_1 will be lower, and the velocity triangle will be of similar shape.

A small area, A, at the nozzle throat produces a higher value of C_1, and this higher gas velocity entering the turbine wheel blades drives the turbine to a higher speed when mounted on the engine. The value of U_1 will be higher and the turbine inlet triangle remains a similar shape, indicating little change in turbine efficiency.

Small values of the turbine casing A/R that operate the turbocharger at high speeds (and high boost pressures) cause a high back pressure to exist in the engine exhaust manifold and very small values will result in a negative differential pressure across the engine; from boost pressure to back pressure. This hurts engine fuel consumption.

Large values of the turbine casing A/R will operate the turbocharger at lower speeds, produce lower boost pressures, and normally will produce a positive differential pressure across the engine. This improves engine fuel consumption.

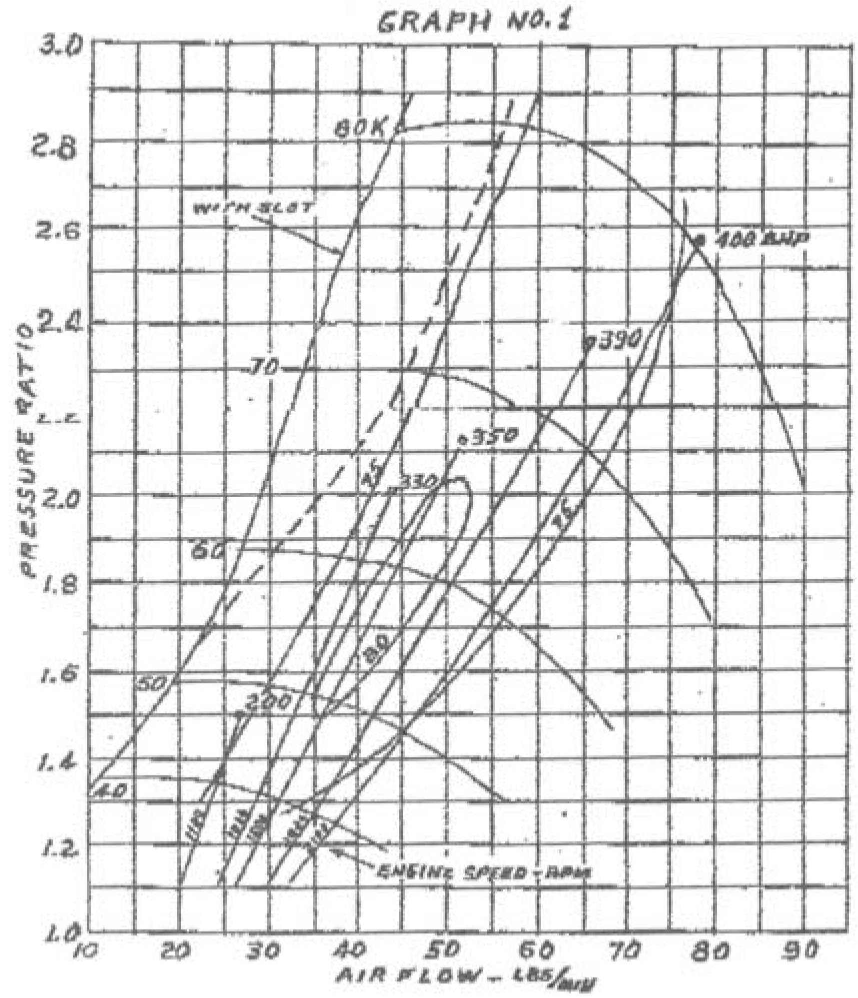
In summary, the value of the turbine casing A/R is an indication of the size of the casing and provides a means of adjusting the speed of the turbocharger when mounted on an engine. Intake manifold boost pressure can be adjusted higher or lower by selecting appropriate A/R values of the turbine casing.

Comp Turbo Technology Inc. Tech Bulletin No. 6

The Effect of Turbocharger Overall Efficiency On Engine Performance

The overall efficiency of a turbocharger has an important effect on the performance of a turbocharged engine. When the overall efficiency is high, less turbine power is required to drive the compressor in order to deliver a desired boost pressure to the engine intake manifold. A relatively larger turbine casting A/R (see our Bulletin No. 5 for a description of A/R) can be used, and this results in a lowered average pressure in the exhaust manifold. Since the pistons act against this lowered pressure on their upstroke when pushing remaining exhaust gas out of the engine cylinders, the parasitic loss of the engine is minimized and the engine fuel consumption is improved. Therefore, it is desirable to strive for the highest possible turbocharger overall efficiency when matching a turbocharger to an engine. The overall efficiency of a turbocharger is obtained by multiplying the compressor efficiency, mechanical efficiency, and turbine efficiency together to produce a single value of efficiency.

The compressor efficiency can be found from the compressor's performance map. These maps are produced by turbocharger manufacturers by running the turbocharger on a performance test stand and measuring the performance over the complete air flow and speed range. Several of these maps are presented in our Bulletin No. 4 on matching a turbocharger to an engine. The Graph Number 1 in Bulletin No. 4 us reproduced below and shows an engine air requirement superimposed on a compressor map, where the compressor efficiency remains over 75% over the complete engine operating range and peaks at 80% in the center of the range. This graph shows an ideally matched engine air requirement with a turbocharger compressor flow range. The maximum compressor efficiency would be taken from the compressor map when calculating a turbocharger overall efficiency.



Next, the mechanical efficiency of the turbocharger can be determined by analysis of the bearing system.

The friction loss in the floating sleeve bearings used in most commercial turbochargers can be calculated using the following formula:

HP = .1363(NS/ 10000)2 L (1/1000C1 / D13 + 1000C2 / D23)

Where:
N = turbocharger speed -RPM
L = bearing length – in.
C1 = I.D. Clearance – in.
C2 = O.D clearance – in.
D1 = inside diameter – in.
D2 = outside diameter – in.

The power loss should be calculated for both minimum and maximum bearing clearances. The minimum bearing clearances produce the higher power loss.

The power loss of the stationary thrust bearings used in commercial turbochargers can be found using the following formula:

HPT = .1363 (N/10000) 2 R24 – R14 /1000C

Where:
N = turbocharger speed – RPM
R1 = inside radius of thrust surface – in.
R2 = outside radius of thrust surface -in.
C = oil film thickness – in.

The loss of the loaded side of the thrust bearing and the loss of the unloaded side must both be calculated and added together to obtain the total HP loss (see our Bulletin No. 1).

The mechanical efficiency of a typical floating sleeve bearing system can be calculated using the above formula and should be of the order of 96%.

The full compliment ball bearings with ceramic balls used in Comp Turbo turbochargers have minimal friction loss since there is no oil film shear in the bearings. The rolling friction loss is very low and the mechanical efficiency of Comp Turbo ball bearing turbochargers is of the order of 99%.

The efficiency of the small gas turbines used in turbochargers has been very difficult to quantify. Some testing has been done by turbocharger manufacturers using a small high-speed dynamometer to measure the HP output of the turbine and determine the efficiency on steady hot gas flow. The values thus obtained are applicable to undivided exhaust manifold systems, however, most turbocharged engines used divided exhaust manifolds to take advantage of their superior performance. In the case of a divided manifold system, the turbine must operate on pulsating exhaust gas flow and, as a result, the turbine efficiency is a variable with time and changes as the magnitude of the exhaust pulses change. A detailed discussion of turbine efficiency is complicated and will be the subject of a subsequent technical bulletin.

In summary, if an average turbine efficiency of 75% is assumed, a Comp Turbo turbocharger can have an overall efficiency of .80 x .99 x .75, which equals 59.4%. A value of turbocharger overall efficiency of 60% has been the objective of turbocharger designers and developers since their inception.

Comp Turbo Technology Inc Technical Bulletin No 7

The Determination of Turbocharger Turbine Efficiency

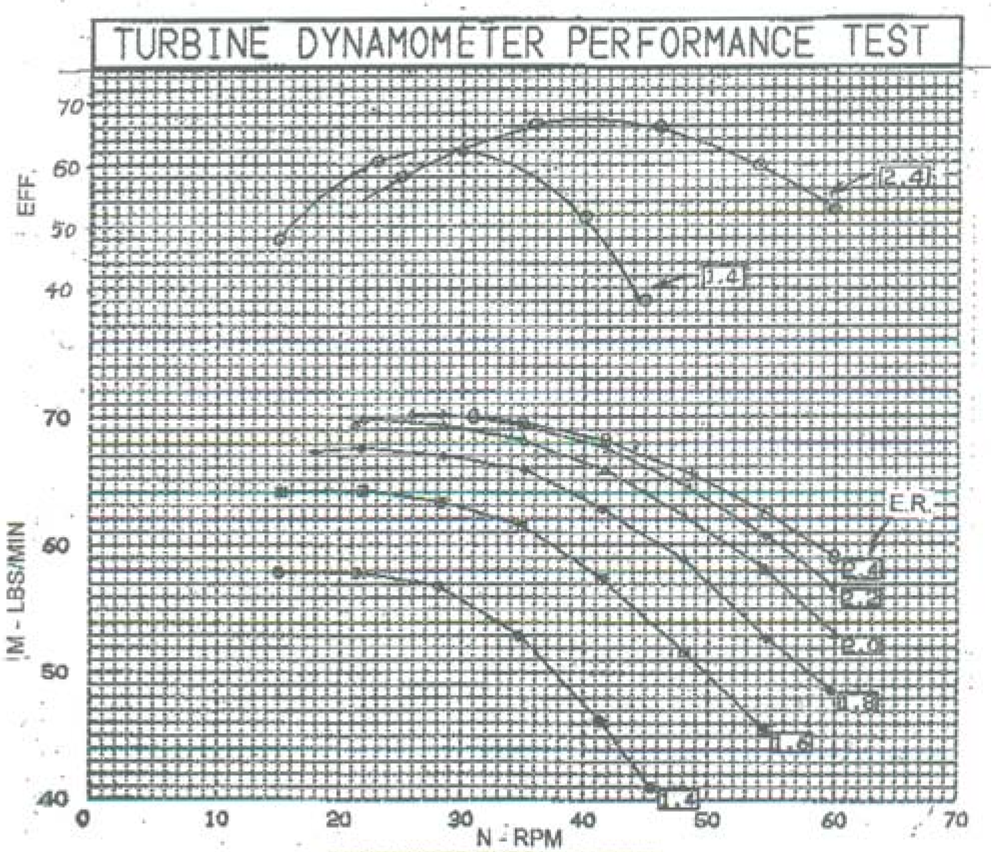
Our Technical Bulletin No 7 described how the overall efficiency of the turbocharger affects the performance of a turbocharged engine. The overall efficiency is obtained by multiplying the compressor efficiency, mechanical efficiency, and turbine efficiency together, resulting in a single value of overall efficiency.

The Compressor efficiency can be accurately determined by operating the turbocharger on a test stand and measuring its performance over the entire operating field of the compressor. Mechanical efficiency can be found easily by calculation using empirical equations for the power losses in the turbocharger bearing system. (see our Bulletin No. 7) This leaves the turbine efficiency, which has been very difficult to quantify, and several methods have been employed to obtain accurate values for use in calculating turbocharger overall efficiency.

In the early stages of design of small turbochargers, attempts were made to determine turbine efficiencies by using temperature, pressure, and mass flow values taken when the turbocharged engine was run in the laboratory coupled to a dynamometer. Date taken by this method consistently produced turbine efficiencies of over 100% and these attempts were subsequently abandoned.

The following attempts to accurately find turbine efficiencies involved connecting the turbine component to a high-speed dynamometer, where the power output could be measured along with accurate values of mass flow, exhaust temperature and pressure under conditions of steady exhaust gas flow.

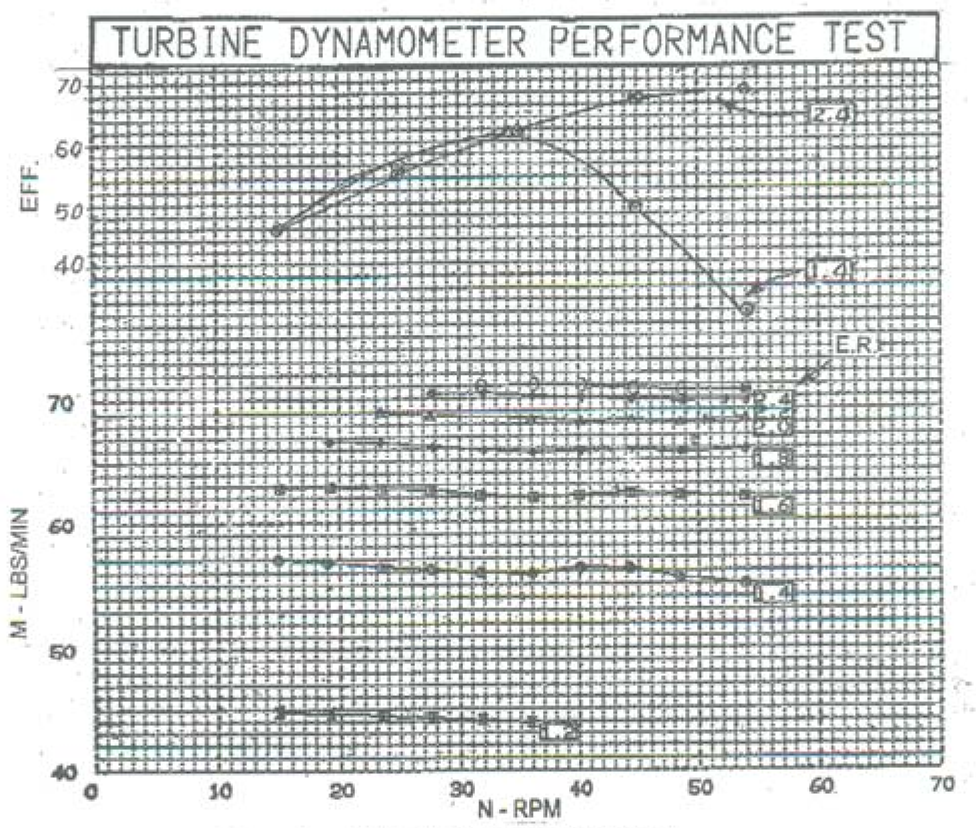
GRAPH No. 1



CONTOURED TURBINE WHEEL

Graph No. 1 shows a turbine dynamometer performance test of a contoured turbine wheel that has an exit diameter smaller than the O.D. of the wheel. Most commercial turbochargers use this type of wheel. The Graph vividly shows that the wheel has a choking effect on its mass flow capability and a drop off in turbine efficiency occurs as the mass flow falls off each of the expansion ratios tested.

GRAPH No. 2

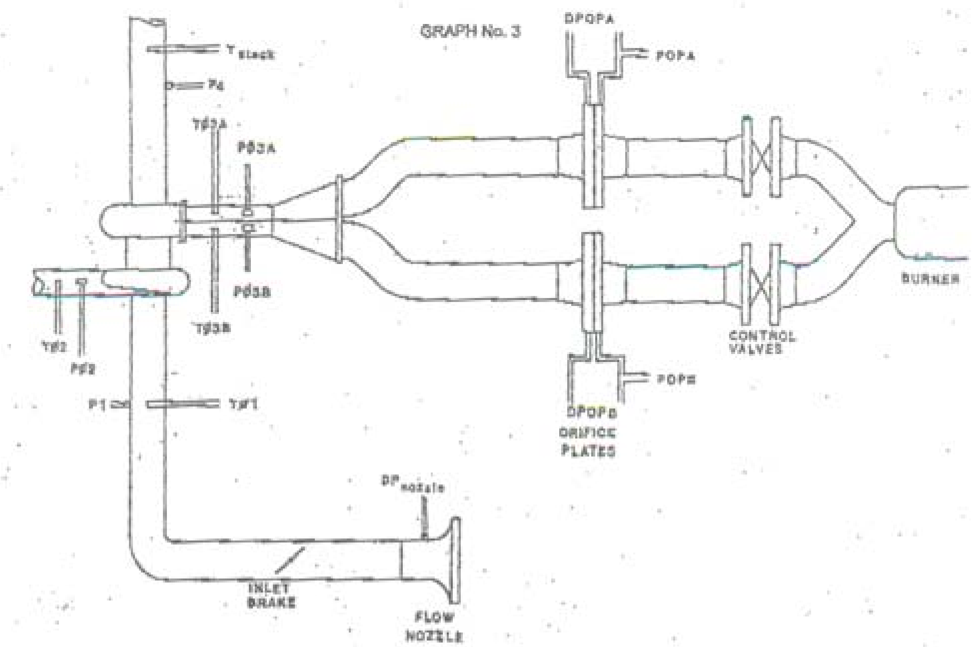


FULL-BLADED TURBINE WHEEL

For comparison purposes, Graph No. 2 shows a turbine dynamometer performance test of a similar sized wheel which has its exits diameter the same as the wheel O.D., knows as a full-bladed wheel. The improvement in mass flow (lbs,/min) is dramatic in that the full-bladed wheel does not choke over its entire performance range. In addition, the turbine efficiency does not fall off as the wheel is operated at high speed and high expansion ratios. Comparing efficiencies at 54,000 RPM and 2.4 expansion ratio, the full-bladed wheel reaches 68%, whereas the contoured wheel only makes 60%. The performance characteristic shown on Graph No. 2 is ideal for racing applications where the turbocharger is operated at very high speed and pressure ratio. Full-bladed turbine wheels are used in many Comp Turbo turbochargers.

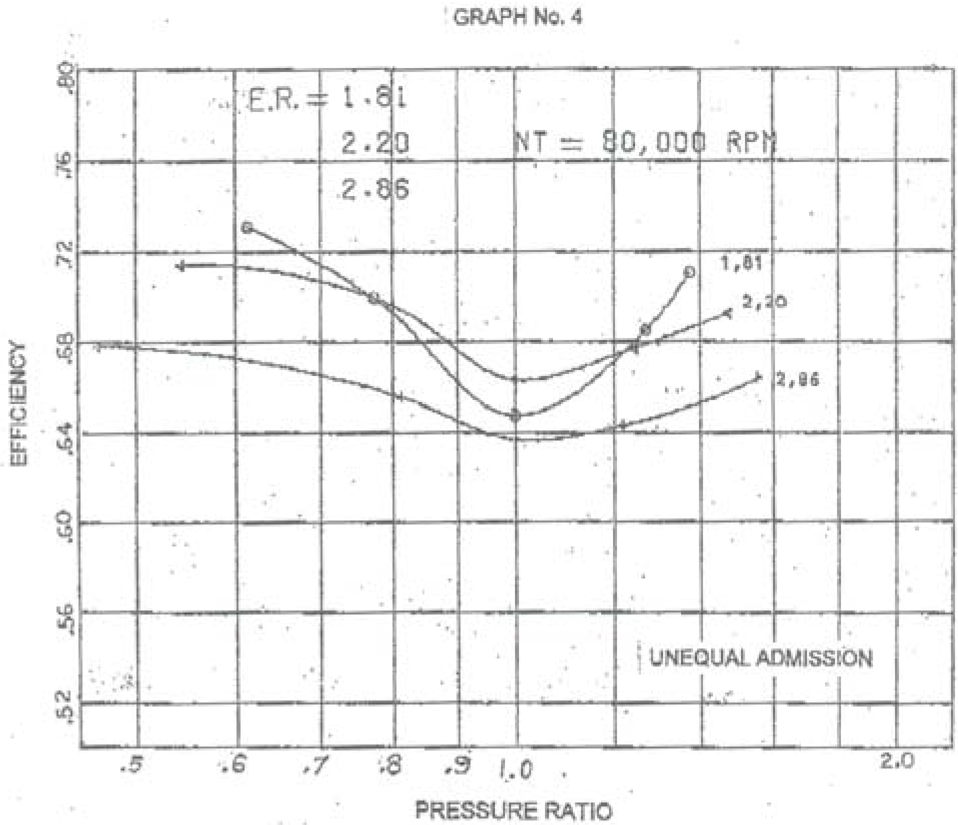
As stated previously, the turbine efficiencies obtained from the turbine dynamometer tests correspond to steady exhaust gas flow conditions. These values are useful only when an undivided exhaust manifold is used on a turbocharged engine. In this case, the exhaust gas flow to the turbine wheel approaches steady flow. Most turbocharged engines employ a divided manifold and a divided turbine casting on the turbochargers. There are significant engine performance advantages resulting from the use of divided manifolds, which are described in detail in our Bulletin No. 2.

By dividing the exhaust manifold and turbine casing, the exhaust pulses from the engine cylinders are separated and present the turbine wheel with a highly pulsating flow. (see diagram in Bulletin No.2) Thus, the turbine efficiencies obtained from dynamometer tests are not applicable to turbines operating on pulsating flow.



UNEQUAL ADMISSION STAND

In an attempt to quantify turbine efficiencies on pulsating flow, a university in Europe devised a test apparatus that could measure the turbine efficiency under simulated pulsating flow. This test apparatus is shown on Graph No. 3. The turbine inlet gas flow is divided to be similar to the gas flow in a divided manifold and the pressure could be controlled in each inlet pipe to mimic pulsating gas flow. This apparatus had the ability to run simulated pulses under steady flow conditions, and thereby obtain accurate turbine efficiencies under simulated gas flow.



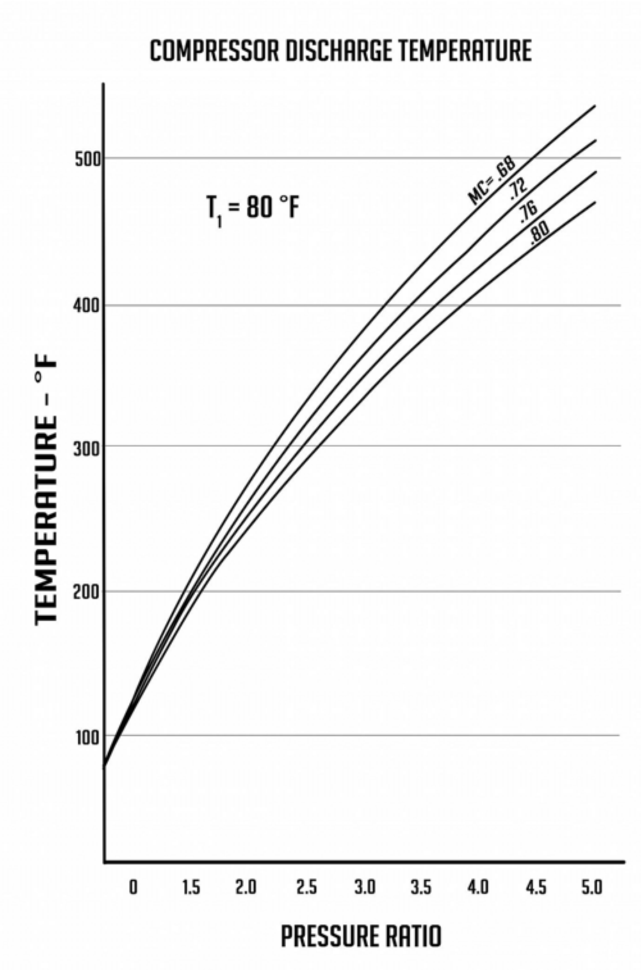
The results of this pioneering effort were unique and surprising. Graph No. 4 is a typical example of results obtained from running a turbocharger turbine on the unequal admission test apparatus. Equal gas flow in the separated pipes is represented by the1.0 pressure ration. Pressure ratios less than 1.0 indicate unequal flow in the two gas inlet pipes. It was very interesting to discover that turbine efficiencies are higher when the simulated exhaust gas pulse from the engine cylinders is high, and the turbine efficiencies are higher than with equal flow in the two gas inlet pipes. The pulse turbine efficiencies are continually changing and get higher as the simulated pulse gets higher. When the exhaust pulse is highest during cylinder blow down, the instantaneous turbine efficiency will be maximized. This discovery contributes to explaining why the divided manifolds produce better engine performance than undivided manifolds.

The conclusion that can be reached form consideration of the information and data given in this bulletin is primarily that a turbocharger with a divided turbine casing mounted on a divided manifold and that employs a full-bladed turbine wheel will produce superior engine performance in racing and other commercial applications.

Comp Turbo Technology Inc Technical Bulletin No 8

The Advantages of Air to Air Aftercooling

One way to substantially improve the performance of a turbocharged engine is to lower the intake air temperature that has been increased by the compression process in the turbocharger compressor. This increase in air temperature has been presented in Bulletin No. 3 on a graph that shows air temperature versus compressor pressure ratio with compressor efficiency as a parameter. This graph is being reproduced here as Figure 1.



For example referring to the graph, a pressure ratio of 3.0 (approx. 30 psig) will heat the intake air to 350°F if the compressor efficiency is 76%. Thus, engine performance will be much improved if the compressed air temperature can be reduced to a much lower value before it enters the engine cylinders.

Charge air cooling has been used for many years, initially employing water-to-air heat exchangers, using engine coolant as the cooling medium. However, except for marine engines, the water-to-air charge cooling has been replaced by air-to-air charge air cooling due to greater cooling effect provided by the use of ambient air as the cooling medium.

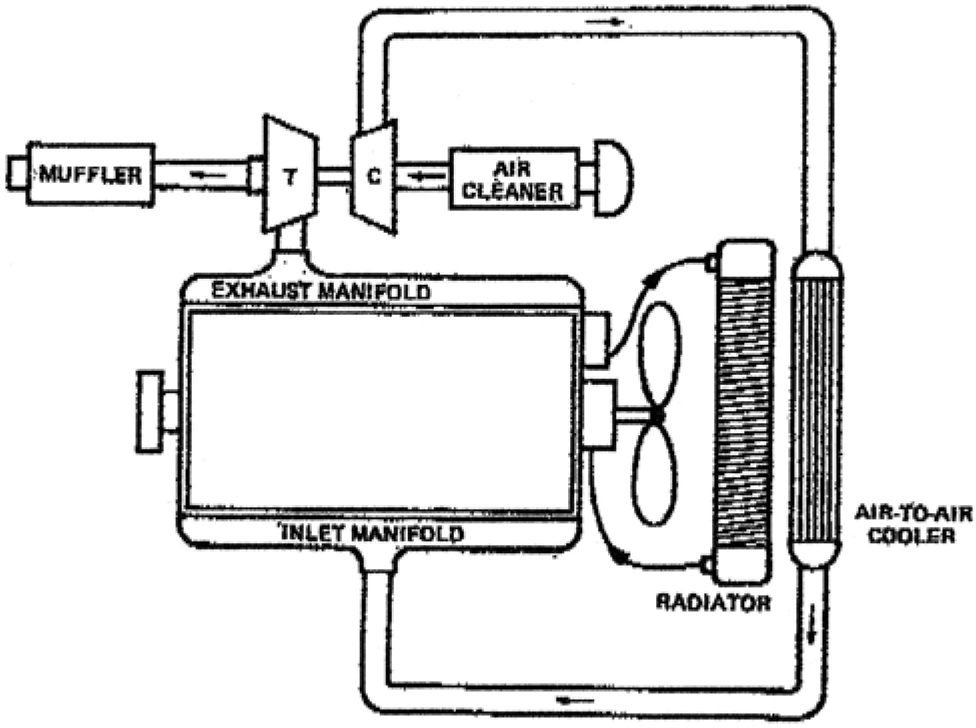


Fig. 2

The air-to-air aftercooling system is shown diagrammatically in Figure 2. It is uncomplicated, has no moving parts, and is in widespread use in commercial applications. To illustrate the potential of the system, referring to Figure 1, the ambient air temperature has been taken as 80°F. A pressure ratio of 3.0 produces a compressor outlet temperature of 350°F. Thus the maximum cooling effect will be 350-80=270°F. If the air-to-air heat exchanger has an effectiveness of .80, then the temperature of the air will be reduced by 216°F (270X.80). The compressed air entering the engine intake manifold will be 134°F (350-216).

If engine coolant were used as the cooling medium, the coolant temperature can typically be 180°F. Then the maximum cooling effect becomes 350°-180°=170°F.

Using the same heat exchanger effectiveness of .80 the compressed air temperature will be reduced by 136°F (170X.80). The air temperature entering the intake manifold will be 214°F (350-136). Comparing this temperature of 214°F with the 134°F produced by the air-to-air system, the advantage of the latter system is obvious.

Lowering the engine intake manifold temperature can result in large improvements in engine performance. Due to the large increase in intake air density, more fuel can be burned, resulting in higher horsepower output. Greater air density in the cylinders can increase combustion efficiency and lower engine fuel consumption. Exhaust temperatures are lower which lowers engine emissions and smoke in the engine exhaust is improved. Air-to-air aftercooling produces lower combustion gas temperatures in the cylinders. This can result in longer engine life since the potential for thermal stress is reduced.

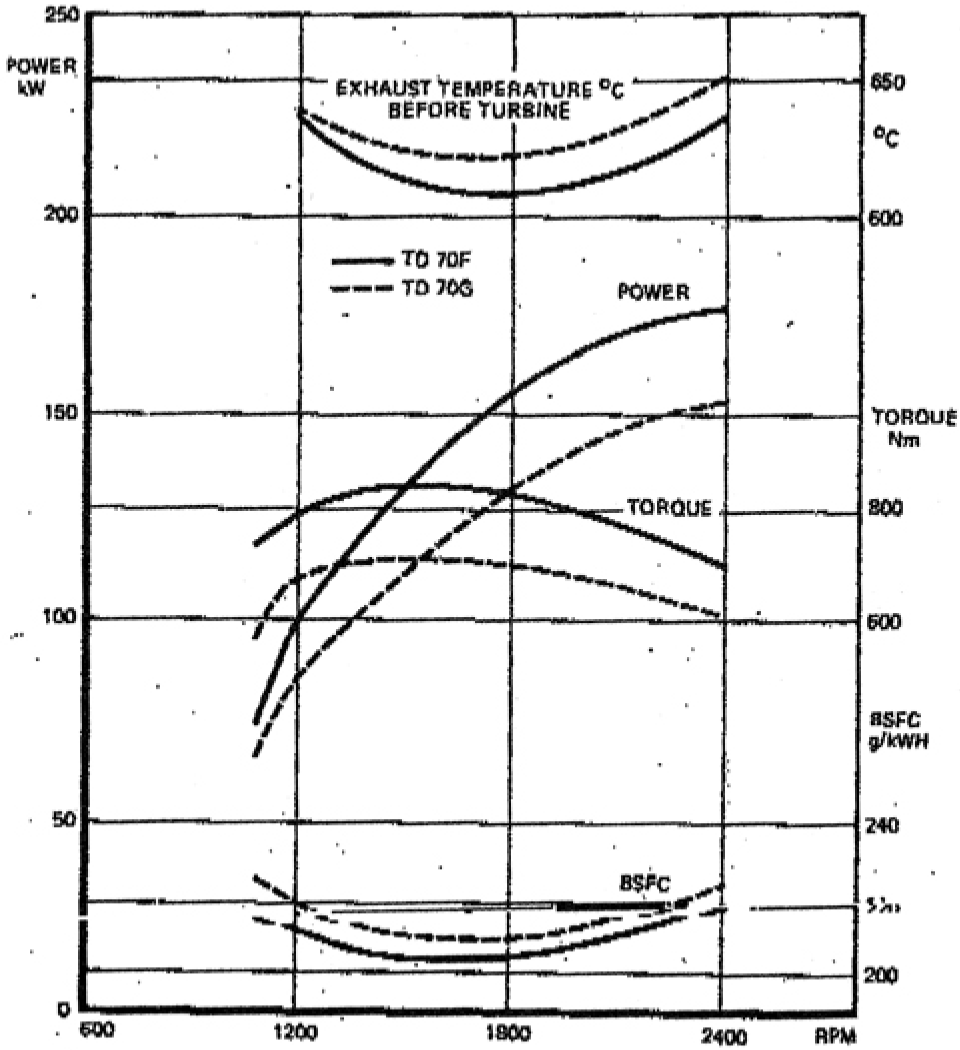


FIG. 3

As an illustration of the performance improvement of a 6.7 liter engine, normally rated at 155 HP, Figure 3 shows the actual performance of a the non-cooled engine compared with the air-to-air aftercooled version. In addition to the higher HP output, exhaust temperature is lower and fuel consumption has been improved. The engine performance shown in Figure 3 is indicative of an engine with a very modest power rating. Performance improvements of highly rated engines would be more dramatic.

The application of air-to-air charge air cooling to turbocharging is no doubt the biggest step in the development of turbocharged engines since the turbocharger itself was introduced.

If additional information or assistance is needed in the application of air-to-air aftercooling to an engine, contact a technical staff member of Comp Turbo Technology Inc.

Comp Turbo Technology Inc Technical Bulletin No 9

The Advantages of the Comp Turbo Triplex Ceramic™ Proprietary Bearing Systems

There have been numerous attempts in the past, almost from the very beginning, to design a successful ball bearing system for small turbochargers. The minimal friction loss and rapid acceleration potential of ball bearings has made them attractive for automotive turbocharger application. Almost all double ball bearing systems that have been designed by various turbocharger manufactures have failed to meet necessary durability requirements. The one system that has met the necessary durability requirements is the Comp Turbo TRIPLEX CERAMIC™ triple ball bearing system. This system has now been in commercial production for over seven years and has established an enviable reputation for performance and durability. It continues to outperform competitive system in the field in Comp Turbo turbochargers.

The TRIPLEX CERMAIC™ bearing system utilizes three full compliment, angular contact ball bearings with ceramic balls. Two bearings are mounted back-to-back in the (cool) compressor end of a rotatable steel cylinder with one bearing mounted slidably against a pre-load spring in the (hot) turbine end of the cylinder. The steel cylinder rides on an outer lube oil film in the bearing housing that cushions the bearings against shock and vibration. All three inner races are clamped against a shoulder on the shaft and rotate with it. When the turbine end of the shaft expands axially due to heat conducted from the hot turbine wheel, the turbine end bearing can move axially with the shaft due to the sliding fit in the bearing carrier. Axial thrust is taken in both directions by the tandem bearings. The TRIPLEX CERAMIC™ bearing system is illustrated in Fig. 1.

Compared with steel balls, ceramic balls in ball bearings have proven to have a number of advantages. Bearing service life is two to five times longer and, since they are lighter and run at lower temperatures, allowable running speeds are as much as 50% higher. The surface finish of the balls is almost perfectly smooth so that friction losses are lower. They exhibit reduced skidding and have a longer fatigue life. These advantages have been established by a well-known ball bearing manufacturer and have been borne out by Comp Turbo turbochargers' operational experience. The minimal friction losses of the ceramic balls are enhanced by the absence of a cage in the full compliment bearings and the resulting mechanical efficiency of the system can approach 99%.

The acceleration rate of a turbocharger is a function of the rotor inertia and the frictional drag of the bearing system. Floating sleeve bearing systems have significant frictional looses due to oil film shear in the inner and outer diameter clearances and in the loaded and unloaded sides of the stationary thrust bearing. (See our Bulletin No. 1 for calculations of frictional losses in sleeve bearing systems.) Since the only frictional losses in the TRIPLEX CERAMIC™ bearing system are from rolling friction, the acceleration rate of the turbocharger rotor is maximized.

OIL-LESS BEARING SYSTEMS

Sleeve bearing systems in turbochargers cannot operate without a continuous supply of lube oil from the engine. In very cold weather, lube oil becomes very viscous and, if it does not reach the turbocharger sleeve bearing promptly on engine startup, the bearings will fail. Comp Turbo Technology, Inc. has developed the first automotive turbocharger that does not require lube oil form the engine. The ball bearings are lubricated by a high temperature grease and they can be operated in very cold temperatures without problems.

Since there is no lube oil to drain back to the engine crankcase, the turbocharger can be mounted on the engine at a variety of angles, including vertically. Fig. 2 illustrates a Comp Turbo oil-less turbocharger mounted on a snowmobile engine in a vertical position.

Removing lube oil from turbochargers has additional advantages in that there can be no oil leakage past the turbocharger seals into the compressor and/or the turbine housing. Also, there is no lube oil in the bearing housing that can carbonize if the engine is shut down hot. Since there is no lube oil shear in the bearing system, frictional losses are reduced an additional amount from bearing systems that use lube oil.

The oil-less bearing system uses the identical parts used in the oil-lubricated TRIPLEX CERAMIC™ bearing system, except that the cylindrical bearing carrier is mounted in the bearing housing on two widely spaced "O" rings. The bearing housing is water cooled and the space between the "O" rings is open to the cooling medium. Thus, the O.D. of the bearing carrier is cooled to carry away heat generated in the ball bearings. Since the cylindrical bearing carrier is easily removed from the bearing housing, the ball bearing can be re-greased at appropriate intervals to extend their service life indefinitely.

The TRIPLEX CERAMIC™ bearing system is available in both the lube oil version and the oil-less version and represents a major step ahead in automotive turbocharging. Both systems are proprietary and protected by recently granted patents. These developments have projected Comp Turbo Technology, Inc. to be well ahead in ball bearing technology for turbochargers.

Comp Turbo Technology Inc. Tech Bulletin No. 10

Turbine Wheel Design

The design of the turbine wheel for a turbocharger must follow the establishment of a design point for the compressor. The compressor design point consists of the selection of the air mass flow and pressure ratio needed to match specific turbocharged engine air requirements. The mass flow of exhaust gas that the turbine wheel must be designed to operate on is the compressor mass flow plus the small additional quantity of mass added by the products of combustion of the fuel burned in the engine cylinder. Thus, the design mass flow for the turbine wheel is usually estimated at about 1.03 times the compressor mass flow.

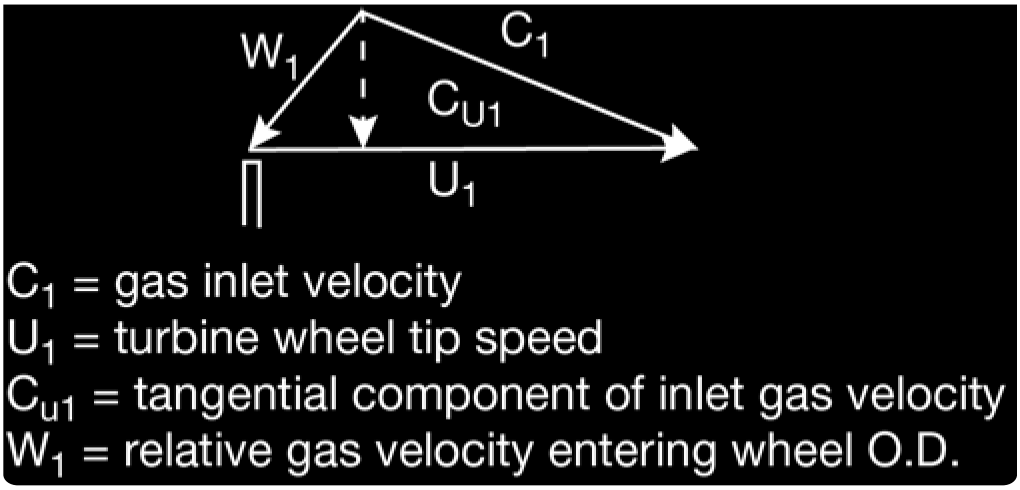
Comp Turbo Bulletin No. 2 deals with exhaust manifold design and illustrates the pulsating exhaust gas flow that the turbine wheel is subjected to when the turbine with a divided turbine casing is mounted on a divided engine exhaust manifold. The turbine wheel must develop the horsepower needed to drive the compressor wheel at its design flow and pressure ratio and is designed on a steady flow basis with some appropriate allowances in the design for operation on pulsating exhaust gas flow.

The turbine power to be developed must include the bearing system losses and an estimate of friction and windage losses. Thus,

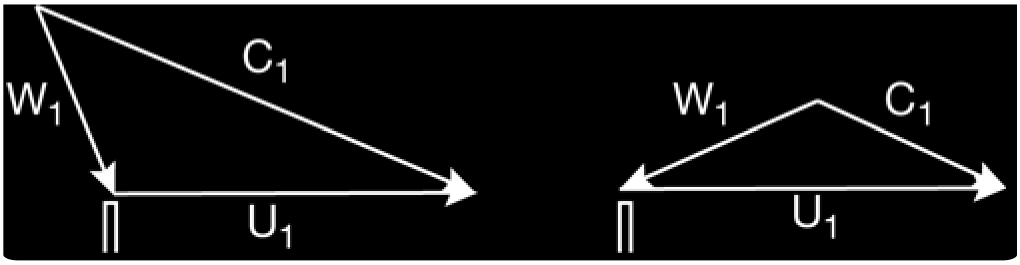
$HP_T = HP_C + HP_B + HP_{FW}$ Where:

HP_T = Turbine Power to be developed
HP_C = Power required to drive compressor
HP_B = Bearing system loss
HP_{FW} = Friction and windage loss

The turbine inlet velocity triangle can be established once the turbine wheel tip speed has been calculated. A typical turbine inlet velocity triangle is illustrated below:

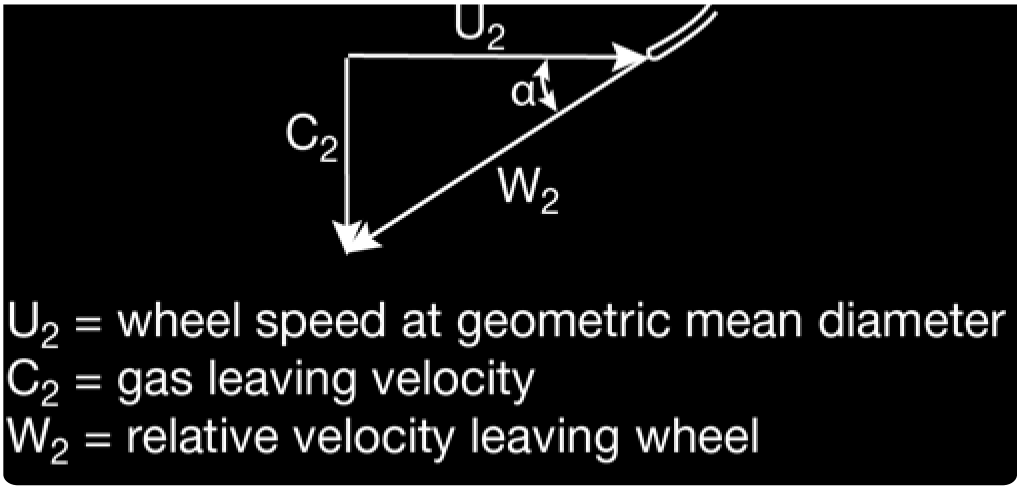


The inlet triangle shown above represents steady flow conditions. In reality, the pulsating exhaust gas flow will cause the value of C_1 to vary with time. The inlet velocity triangles shown below illustrate the minimum and maximum values of the entering gas velocity.



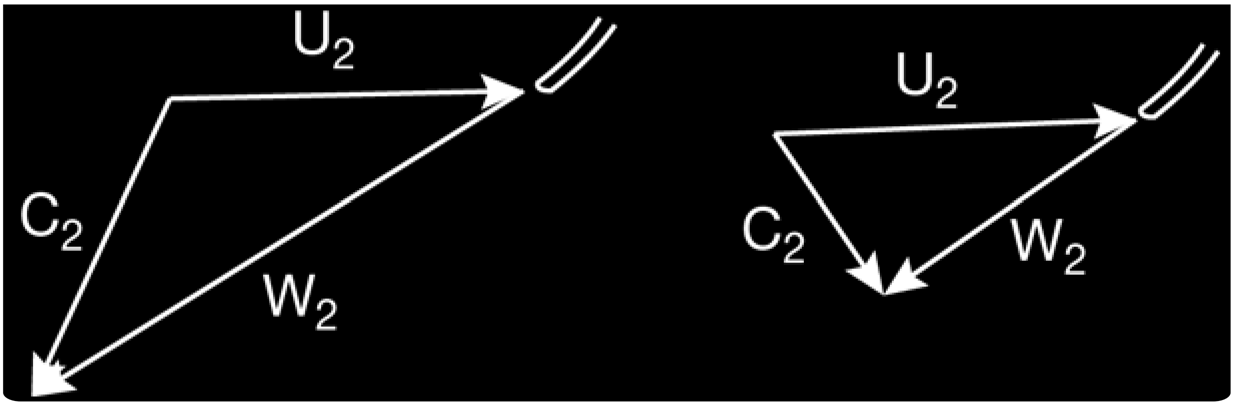
The bulk of the exhaust gas flow occurs at the peak of the pulse. To minimize entrance losses, the design of the wheel should position the maximum values of the relative entering gas velocity to be as close to radial as possible.

The turbine exit conditions have and important effect on the turbine efficiency. A typical exit velocity triangle at the geometric mean diameter is shown below:



The angle a, at the design point flow, determines the wheel blade angle at the exit. The value of C_2 represents unrecoverable energy that is lost in the atmosphere when the gas leaves the wheel and this establishes the value of the leaving loss.

The value of C_2 will vary with time because of the pulsating flow of the exhaust gas. The exit velocity triangles shown below illustrate the increased magnitude of the leaving loss at the peak of the pulse and the decreased magnitude in the valley between pulses.



Some Comp Turbo full bladed turbine wheels have exit blade diameters that are equal to the inlet diameter and, due to this design feature, the gas leaving velocity C_2 is minimized, leading to the lowest possible leaving loss and this maximizes the turbine efficiency.

Our Bulletin No. 8 on the subject of turbocharger turbine efficiency shows the comparison of efficiencies of a contoured turbine and a full-bladed wheel operating on steady flow. The advantage of the full-bladed wheel with minimized leaving loss is characterized by the high efficiencies at very high mass flows. Graph No. 4 in Bulletin No. 8 shows actual test data of a pulsating flow turbine where efficiencies were found to be higher when the pulsations are maximum.

Aftermarket Turbo Cross Reference Guide

Our model #	Garrett Model#	Precision Model #	Turbonetics #
CT2 4747 TBB	GT2860RS (GT28RS) Disco Potato	PTB225-4828B	N/A
CT2 4947 TBB	GT28RS / GT2871R (49mm)/GT2876R	PTB225-5128B	N/A
CT2 5147 TBB	GT2871R (51mm)	PTB225-5128B	N/A
CT4 5347 TBB	GT2876R (2540R HKS)	N/A	N/A
CT3 5356 TBB	GT3071R	PTB225-5130B	TN 400
CT3 5558 TBB	GT3076R (GT30R/GT3037)	PTB305-5530B	TN 450
CT4 5558 TBB	GT3076R (GT30R/GT3037)	PTB405-5530B	N/A
CT3 5858 TBB	GT3076R (GT30R/GT3037)	PTB305-5830B	N/A
CT4 5858 TBB	GT3076R (GT30R/GT3037)	PTB405-5830B	TN 500
CT3 6262 TBB	GT3582R (GT35R-GT35)	PTB305-6235B	TN Hi-Fi 550
CT4 6262 TBB	GT3582R (GT35R-GT35)	PTB405-6235B	Billet GT-K 600
CT43 6467 TBB	GT4088R	PT6466 CEA	Billet GT-K 700
CT43 6767 TBB	GT4094R	PT6766 CEA	Billet GT-K 750
CT43 6467 JB360	GT4088	PT6466 CEA	Billet GT-K 700
CT43 6767 JB 360	GT4094	PT6766 CEA	Billet GT-K 750
CT5 7083 TBB	GT4294	PT7175 CEA	HP-72
CT5 7083 TBB	GT4294R	PT 7175 CEA	HP-72
CT5 7083 TBB	GTX4294R	PT 7175 CEA	HP-72
CT5 7483 TBB	GT4202R	PT7675 GT42	HP-76
CT5 7783 TBB	GTX4202	PT7685 GT42 CEA	HP-78
CT6 8087 TBB	GT4508R	PT8285 GT42 CEA	TNX80-91 Y2K
CT6 8087 TBB	GTX4508R	PT8284 CEA	TNX80-91 Y2K
CT6 8087 TBB	GT4708R	PT8285 GT42 CEA	TNX80-91 Y2K
CT6 8894 TBB	GT4718R	PT8685 GEN2 CEA	TNX8889
CT6 8894 TBB	GTX4718R	PT8891 GEN2 CEA	TNX8889
CT6 88112 TBB	GT5518R	GEN2 Pro Mod 88 CEA	TNX8801
CT6 88112 TBB	GTX5518R	GEN2 Pro Mod 88 CEA	TNX8801
CT6 91112 TBB	GT5533R	GEN2 Pro Mod 91 CEA	TNX 91-106
CT6 94112 TBB	GT5533R	PT94 CEA	N/A
CT6 94112 TBB	GTX5533R	GEN2 Pro Mod 94 CEA	N/A
CT6 106112 TBB	GT5541R	PT106 CEA TNX0606	N/A
All Billet Wheels	X=Billet Wheel	CEA= Billet wheel	

Comp Turbo Compressor Housings	Comp Turbo Turbine Housings
CT2 - 3" inlet/ 2" outlet	CT2 - T2 turbine housing options
	.64 .86 Internal gate 5 bolt discharge
CT3 - 3" inlet/ 2" outlet	.64 .86 Tial V-band SS housing
	CT2/CT3 - T3 Turbine housing options
CT4 - 4" inlet/ 2.5" outlet	T3 .36 .48 .63 or .82 4 bolt/5 bolt or V-band 2.5", 3" or 3.5"
	Tial SS V-band in and out .63 .82 1.06
CT43 - 4" inlet/ 3" outlet	CT3/CT4/CT43 - T4 Turbine housing options
	Divided .58 .70 .84 1.15 1.32 1.52 V-band 3",3.5" or 4"
CT5 - 5" inlet/ 3.5" v-band outlet	Undivided .68 .81 .96 V-band 3",3.5" or 4"
	Tial SS V-band in and out .63, .82, or 1.06
CT6 - 6" inlet/ 3.5" v-band outlet	CT5/CT6 - Mid Frame T4 Turbine options
	.90 1.0 1.10 1.25 Divided V-band 4"
	Tial SS V-band in and out 1.02 1.16 1.30
	CT5/CT6 - T6 Large frame turbine housing options
	.96 1.08 1.23 1.39 Open V-band 5"
	Tial SS V-band in and out 1.15 1.30