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Automotive Superchargers and Turbochargers

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INTRODUCTION

Supercharging uses a mechanically driven "air pump" to artificially supply the engine cylinders with a greater amount of air/fuel mixture than would be taken in under normal atmospheric conditions. The basic concept in most supercharging devices is to increase the outlet pressure over the inlet pressure and therefore the density of the air delivered to an engine above ambient atmospheric conditions. This in turn increases the mass of air drawn into the cylinders during each intake stroke. Then, if the optimum air-to-fuel ratio is still maintained, more fuel can be combusted to produce more power output.

A supercharged engine of a given displacement size can produce significantly more power and torque compared to the same engine when normally aspirated, that is, not supercharged. Conversely, a smaller and

* Retired

lighter supercharged engine can produce the same power as a larger and heavier normally aspirated engine (Fig. 1). The smaller engine (1) consumes less fuel under idle conditions, (2) has less mass and inertial loading, (3) has lower frictional and partial-load pumping losses, and (4) produces less emissions under partial-load conditions. Smaller supercharged engines in turn allow smaller, more compact vehicles without sacrificing performance. Supercharging can compensate for the power loss that occurs with the less dense air available as a vehicle climbs in altitude.

A supercharger typically consists of a compressor to increase the pressure of the air inflow, which is usually driven off the engine crankshaft either by belts or gears (Fig. 2). A turbo supercharger, or simply a turbocharger, consists of a centrifugal compressor that is driven directly by a gas turbine, which in turn is driven by engine exhaust gases. It should be noted that a turbocharger is a form of supercharger. However, common terminology will be used here. A mechanically driven compressor will be called a supercharger, while an exhaust-driven turbine/centrifugal compressor combination is a turbocharger (Fig. 3).

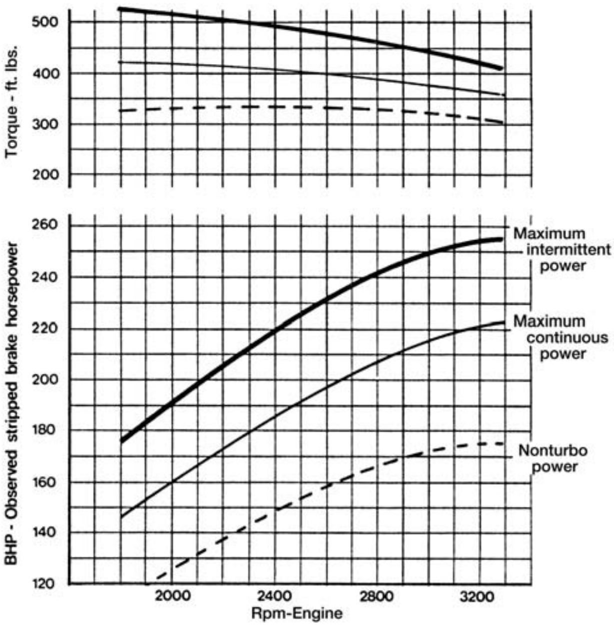


Figure 1 Comparison of the performance between a turbo supercharged (turbocharged) diesel V-8 engine and a naturally aspirated engine (nonsuper charged). (Courtesy Hypermax Engineering, Inc.)

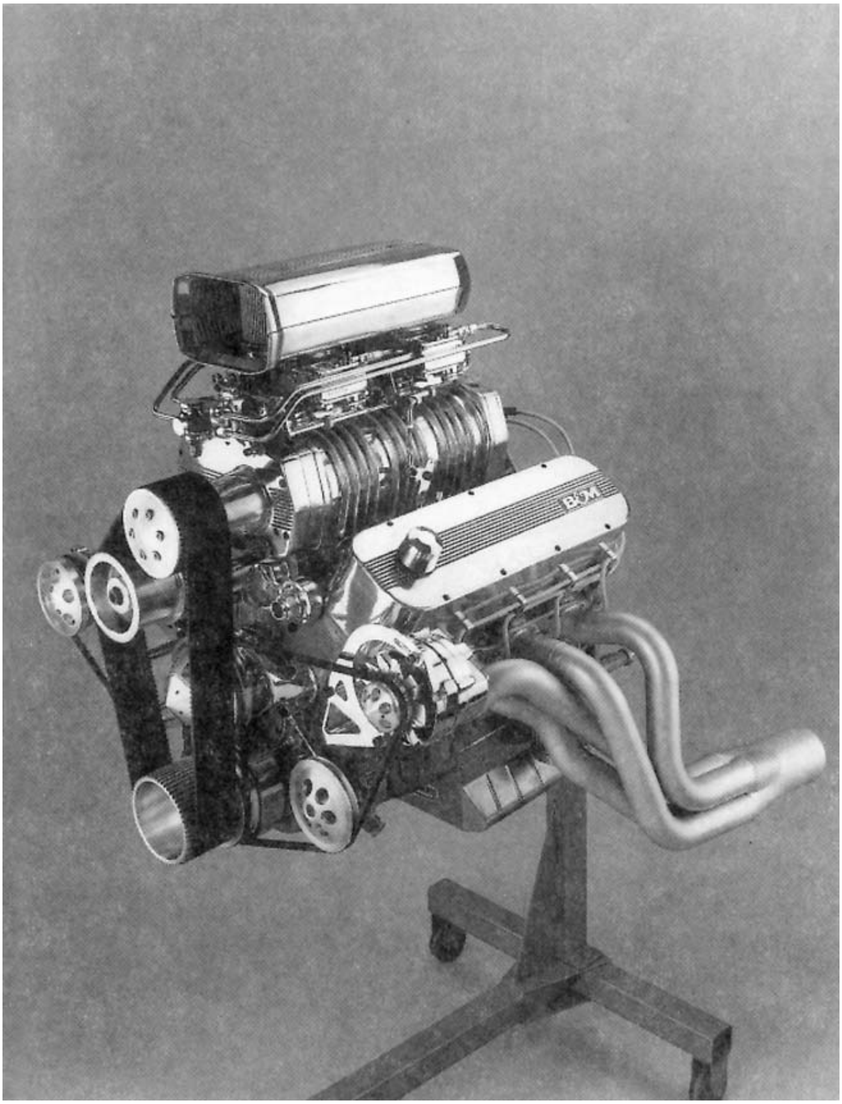
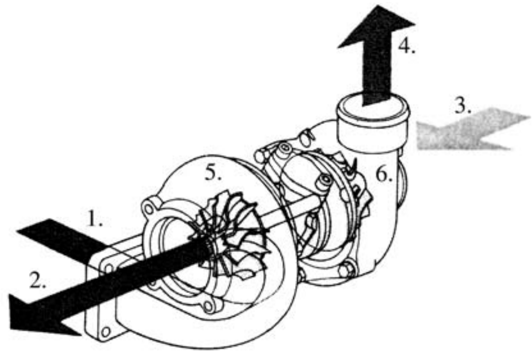


Figure 2 A Roots-type supercharger driven off the crankshaft by a belt drive. (Courtesy B&M Automotive Products.)



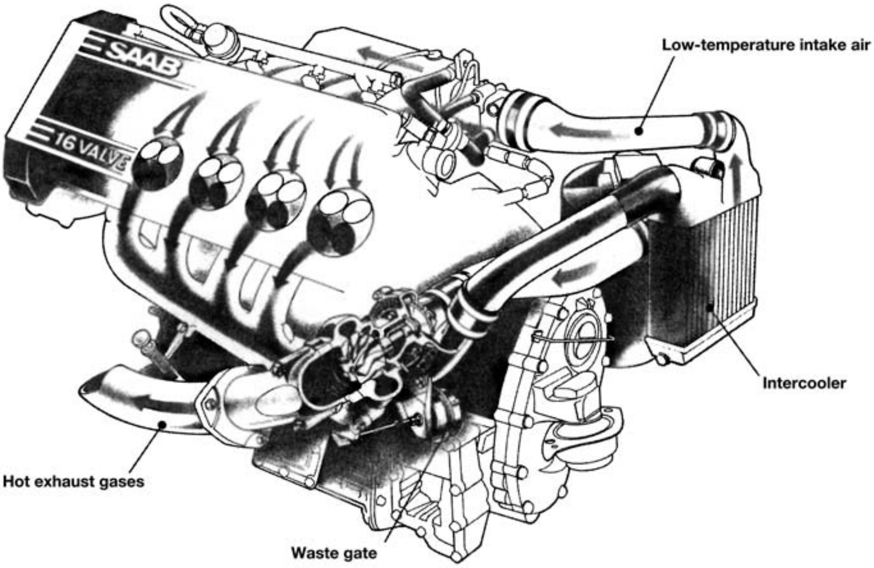


Figure 3 Typical turbocharger installation on a spark ignition engine. After the air has been compressed in the turbocharger, it is cooled as it flows through the intercooler. The wastegate is a valve that controls the boost pressure. (Courtesy Saab Automobile AB.)

The supercharger was in routine use many years before the turbocharger, primarily because it took longer to develop the materials needed for the turbochargers, which operate at very high speeds and temperatures. Many of the materials that allowed mass-produced turbochargers were developed for aircraft gas turbine engines.

Superchargers driven directly off the engine rather than by exhaust gases are extremely responsive in providing an instantaneous increase in power. In contrast, the turbocharger's rotating parts, the turbine and compressor, take a finite time to be spun up to operating speed after the throttle is opened. This time is commonly referred to as turbo lag (Fig. 4). Turbo lag is a result of incompatibility in matching the engine, a volume machine, with a speed machine, the turbocharger. Engineers have done a commendable job in reducing turbo lag, but it has not been entirely eliminated.

Being mechanically driven, usually off the crankshaft via belts or gears, the supercharger can be a complex piece of machinery. Therefore, it is usually more expensive to manufacture than a turbocharger. It often is more difficult to find space for superchargers in the engine compartment. By operating at very high speeds, turbochargers are very compact and require

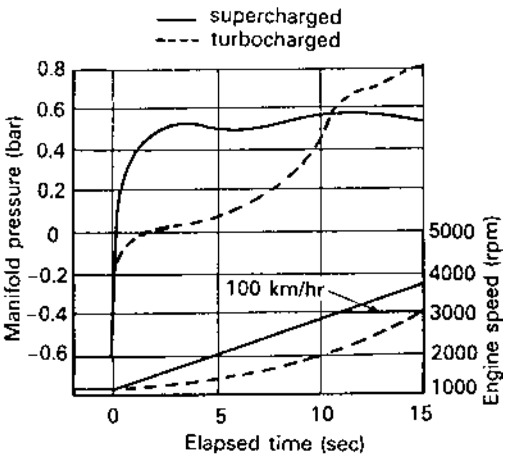


Figure 4 Comparison of a 2.5-L engine fitted with a Roots-type supercharger versus the same engine fitted with a turbocharger. Note that the supercharger provides a high manifold (boost) pressure almost instantaneously. The turbocharger takes longer to reach maximum boost pressure because of "turbo lag." Therefore, engine speed rises faster with the supercharger compared to the turbocharger. (Courtesy Porsche AG.)

significantly less space in the engine compartment. However, supercharger concepts, such as belt-driven centifugal compressors and pressure-wave superchargers, can rival turbochargers in the space they consume.

The supercharger is usually less efficient, drawing off more power from the engine and adding to fuel consumption. At first, it may seem that you are getting something for nothing with a turbocharger, since it is powered by the energy of the exhaust gases, which normally would be wasted out of exhaust pipe. However, closer examination shows the turbine in the turbocharger does increase to back pressure in the exhaust manifold, which in turn results in reduced engine power output. Fortunately, this loss of power in driving the turbine is relatively small in relation to the power increase obtained at the flywheel. Also, there are frictional losses, mainly in turbine and compressor shaft bearings, though again these are relatively small. A supercharger is more likely to be noisier, often with a characteristic whine, than a carefully designed and maintained turbocharger. The two are compared in Table 1.

A BRIEF HISTORY OF SUPERCHARGING AND TURBOCHARGING

Supercharging is almost as old as the internal combustion engine itself. Rudolf Diesel produced his first supercharger design in 1896, only about 20 years after Dr. Nickolaus August Otto had invented his four-stroke engine. In 1901, Sir Dugald Clark found that an engine produced more power when the volume of air entering a cylinder was increased artificially. Around the turn of the century, Rateau in France developed the centrifugal compressor, and in 1902 Louis Renault patented a system using a centrifugal fan to blow air into the carburetor inlet.

In 1907, American Lee Chadwick with J. T. Nicholls used a single-stage centrifugal compressor to put the carburetor under pressure to increase volumetric efficiency. This concept was extended to a three-stage, belt-driven centrifugal compressor that supplied the carburetor with pressurized air. In 1908, Chadwick's car equipped with a blower won the Great Despair hill climb in Pennsylvania. In the next 2 years, the car won more races, and the replicas produced were the first 100-mph cars sold to the public. In 1905, Swiss Alfred Buchi developed the first successful modern turbocharger driven by engine exhaust gases.

Supercharger technology advanced rapidly during World War I to allow fighters and bombers to fly at higher altitudes. The supercharger compressed the thin air of high altitudes so the engines could get sufficient air for proper combustion. Besides increasing the amount of air supplied to

Table 1 Superchargers Versus Turbochargers

Parameter	Supercharger	Turbocharger
Performance	30–40% Power increase over naturally aspirated engine Increased low-RPM torque No turbo lag—dramatic improvement in startup and passing acceleration, particularly with automatic transmission	30–40% Power increase over naturally aspirated engine
Temperatures		Increased underhood temperatures may require component upgrading
Efficiency loss	Mechanical friction	Increased engine back pressure
Packaging	Fairly simple if belt-driven	Major revisions required to package new exhaust system
Lubrication system	Self-contained	Uses engine oil and coolant
System noise	Pressure pulsation and gear noise	High-frequency whine and wastegate

Note. Based on information supplied by Eaton Corporation.

the engine at high altitudes, supercharging provided extra power for takeoffs and climbs. Initially, developments centered around the Roots-type positive displacement blower, but were superceded by more efficient centrifugal compressors.

Supercharging was pursued between the wars, first for military aircraft and then for commercial airliners like the Boeing 247 and Douglas DC-3 that could fly above most adverse weather. The Boeing 307 Stratoliner also had a pressurized cabin achieved by as a byproduct of supercharging. While most engine makers concentrated on superchargers driven via gears off to the engine crankshaft, General Electric had developed a successful aircraft turbo supercharger or turbocharger by 1925.

Meanwhile, supercharging was picked up by auto racers by the early 1920s. In 1921, a six-cylinder Mercedes 28/95 using a Roots-type blower won the Coppa Florio. In 1923, Fiat was the first to supercharge a Grand Prix race car, first using a Wittig vane-type supercharger, but switching to a Roots-type blower. The latter used a charge air cooler before the carburetor.

The Alfa Romeo P2's supercharged straight eight delivered a then-impressive 140 hp at 5,500 rpm. Mercedes entered a supercharged car in the 1923 Indianapolis, and a Duesenberg with a supercharged engine first won the Indy 500 in 1924. The Duesenberg used a centrifugal compressor and was the first supercharging system to suck air through the carburetor. Now the latent heat of vaporization of the fuel could be used to cool the blower and the fuel/air mixture. This resulted in a greater mass of mixture being forced into the engine. Incidently, Fred and Augie Duesenberg and Harry Miller were major forces in developing supercharged cars in the United States. Between 1925 and 1938, most Grand Prix cars including Alfa Romeos, Auto-Unions, Bugattis, Delages, and Mercedes were supercharged. In 1937–1939, Grand Prix racing was completely dominated by the enormously powerful Mercedes and Auto Union supercharged cars. The Auto Unions with 6-L, 16-cylinder engines developed as much as 520 hp and the Mercedes W 125 produced 646 hp, both with two-stage Roots-type blowers. With the exception of Mercedes, who persisted in downstream carburetor positioning until 1937, the practice of mounting the carburetor in front of the supercharger became normal. The W 125's straight eight's power output would not be matched until the turbocharged Indianapolis, CanAm, and Grand Prix race cars of the 1970s and 1980s. Supercharging was also available on top-of-the-line road cars of the 1930s in both Europe and the United States. Automakers like Alfa Romeo, Auburn, Bentley, Buggati, Cord, Delage, Duesenberg, Mercedes Benz, and a few more had supercharged models in their catalogs.

Aviation supercharging again moved ahead during World War II, and turbochargers proved themselves in aircraft like the B-17 Flying Fortress, B-24 Liberators, P-38 Lightning, and P-47 Thunderbolt. By the early 1950s, the piston engine had reached its zenith with the Wright Turbo-Compound, an 18-cylinder engine that could produce up to 3,700 hp and allowed airliners like the Douglas DC-7 and Lockheed Super Constellation to fly across the continent or the Atlantic Ocean nonstop. Supercharger and turbocharger technology was also used to improve the power/weight ratios and obtain maximum power from large diesel engines for ships and for electrical power generation. Turbocharged diesel engines for trucks were introduced in the mid-1950s.

There was a brief return of supercharged race cars in Europe right after World War II, However, by 1950–1951 unsupercharged Ferraris dominated racing and the use of supercharged cars declined, abetted by a change in Grand Prix rules in 1952 limiting supercharged cars to 500 cm^3. Turbocharging would appear in this racing venue in about 23 years. In 1951, a turbocharged Cummins Diesel Special appeared in the Indianapolis 500, and turbocharged Offenhausers made their debut here in 1966. By the mid-

1960s, turbocharging was used by USAC racers, and of course supercharging and turbocharging were used and improved by drag racers.

Kaiser was the first U.S. automaker in 1954 and 1955 to market a supercharged engined car after the war. It used McCulloch superchargers to boost the output of its 226.2-cubic-inch displacement (cid) six-cylinder engines from 118 to 140 hp. Studebaker-Packard also used McCulloch/Packard superchargers on the 289-cid V-8 used in the 1957 and 1958 Packards and Studebakers as well in the 1963–1964 Studebaker Avanti. For 1957 only, Ford offered a McCulloch/Paxton supercharger on its 312-cid V-8 that was conservatively rated at 300 hp. There was also a NASCAR version of the engine that developed 340 hp for stock-car racing.

The 1962–1963 Oldsmobile Jetfire was the first production car to use a turbocharger. Oldsmobile engineers added an AiResearch turbocharger to the Buick-built 215 V-8. The Turbo-Rocket engine produced 215 hp compared to 155 hp for the normally aspirated version of the engine. The Turbo-Rocket engine also had an ultrahigh compression ratio of 10.25:1. Indeed, combining turbocharging with a high compression ratio was the Jetfire's downfall because it led to severe denotation problems. To cure this problem, Oldsmobile used a specially formulated Turbo-Rocket fluid, really a mixture of water and alcohol, which was injected into the engine. Chevrolet offered a TRW turbocharger for its 1962–1966 Corvairs with their rear-mounted air-cooled six-cylinder engines with more success.

By the mid- to late 1970s, turbocharging became a commonly used means to boost the power of small-displacement engines that were being used to achieve better fuel economy. Porsches, BMWs, Saabs, Buicks, and many others were available with turbocharged engines.

Finally, in the late 1950s, General Motors investigated "air-injection supercharging" (Fig. 5) as an alternative to a mechanically driven supercharger or exhaust-driven turbocharger. Air was stored at very high pressures in air bottles that were pressurized by a couple of engine-driven air compressors. The high-pressure air was injected directly into the cylinder through a separately actuated poppet valve in the cylinder head during the compression stroke after the intake valve was closed. Additional fuel was also added through the carburetor. Tests on a 2.7-L, six-cylinder engine showed that air-injection supercharging increased horsepower output about 2.5 times. Additionally, direct air injection reduced both temperatures in the cylinder and increased the turbulence, both allowing higher cylinder pressures without knocking. While the concept looked good on paper and in tests, there were several drawbacks. The air bottles took up almost all the trunk space, even they though there was only sufficient air for a couple of minutes of supercharging. The system was quite complex and costly, and it

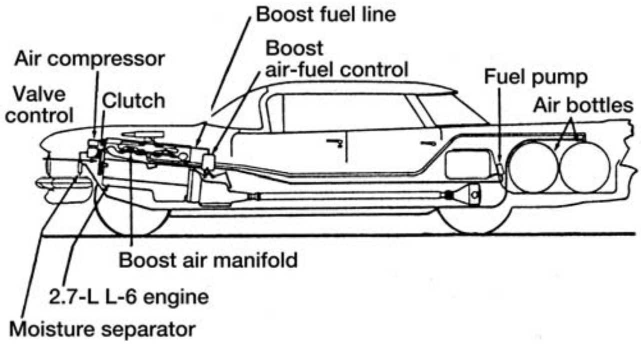


Figure 5 Schematic of the "air-injection supercharging" system tested by General Motors in the late 1950s.

took at least a half hour to bring the tanks up to full charge using the engine-driven compressors. The 3,000-psi air bottles could also become "bombs" in a crash.

SUPERCHARGERS

There are basically two distinct categories, classified according to compressor type, of mechanically driven superchargers that have been used with production gasoline engines. First there are the positive-displacement types, which are characterized by the Roots, rotary piston, or vane-type superchargers. During each revolution, the positive-displacement supercharger pushes the air charge into a zone of higher pressure.

The second type is a dynamic or kinetic supercharger that uses a mechanically drive radial- or axial-flow compressor. These turbomachines convert the mechanical energy into kinetic energy by accelerating the air charge. This increased kinetic energy is converted to increased pressure by using of a diffuser. Incidentally, the turbocharger is a kinetic turbomachine except the drive energy comes from an exhaust-driven turbine rather than being mechanically driven by the engine crankshaft via belt or gears. Indeed, it is also called a turbosupercharger.

Roots Blower

The Roots-type blower was invented and originally marketed by P. H. & F. M. Roots of Connersville, Indiana, in 1866. This positive-displacement supercharger is probably the best known and at one time the most widely used mechanically driven supercharger for automotive applications.

The basic Roots blower uses two counterrotating lobed rotors that mesh together. Air entering the intake port is trapped between a rotor lobe and the casing while it is carried around to the discharge port. The two rotors do not actually touch, because they are driven in proper phase by externally mounted gears (Fig. 6).

The Roots-type supercharger is more correctly called a blower rather than a compressor because the volume of air trapped by a lobe does not change while being transferred from the inlet to discharge port at constant pressure. Compression does not occur internally in the supercharger, but in the downstream restriction through area reduction. When the air charge is exposed to the outlet port, part of this higher-pressure air flows back into the supercharger to equalize the local pressure. The meshing rotors seal the pressurized volume as they rotate back to the inlet port.

The volume of air pumped during each revolution is a function of the length and diameter of the rotor. These dimensions, along with the lobe profile and port shape, control the flow rate pattern over the rpm operating range. While the basic design will work with two lobes per rotor, three- and four-lobe versions have been used.

Advantages of the Roots-type blower include a balanced rotating system and a relatively low operating temperature. Disadvantages include somewhat reduced reliability at high rotational speeds. Also, the tight tolerances require precise engineering and manufacturing, leading to

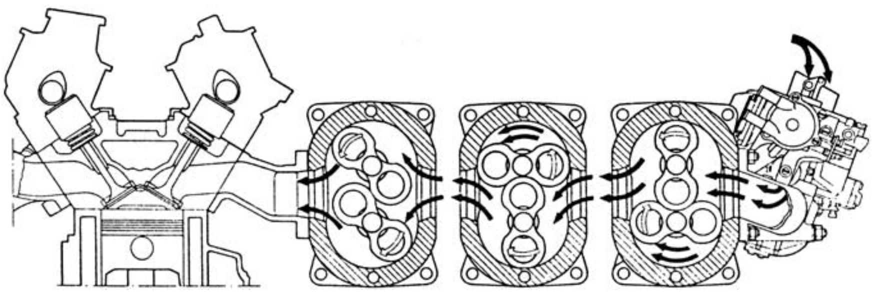


Figure 6 Operation of a twin-rotor Roots blower. The air is trapped between the lobes and the casing while it is carried from inlet to outlet. Note the area reduction after the discharge port that provides the pressure boost. (Courtesy Fiat/Lancia.)

relatively high unit costs. Frictional losses are relatively low with Roots blowers because the only rotating contact is between the synchronizing gears, which usually run in their own lubrication chamber. However, because of compression difficulties, the overall efficiency of Roots blowers can be below 50%. Low friction and a design that eliminates out-of-balance forces allow the Roots-type blower to run at speeds up to 10,000 rpm. Speed limitations result from rotor flexing that can cause rotor lobes to clash or due to excessive thrust loading can cause the rotors to operate out of phase or bearings to spin in their housings (Fig. 7).

High noise level has been a long-time disadvantage of Roots-type superchargers, which produce a very distinctive noise, especially at low

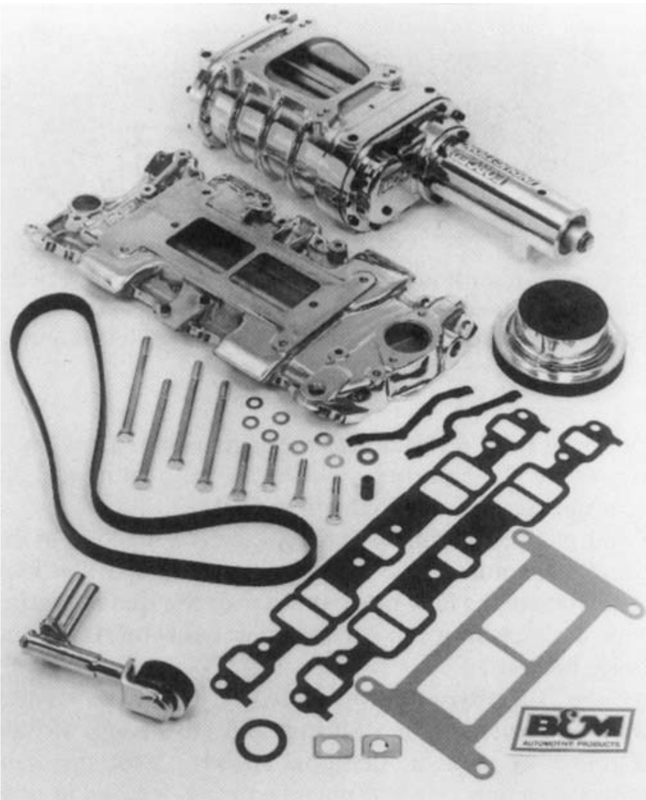


Figure 7 The Roots-type blower is a rather complex piece of machinery. The twin rotor lobes can be seen in the actual supercharger in the upper portion of the picture. (Courtesy B&M Automotive Products.)

speeds. The sound is caused by the surging or pulsing intake and discharge at both the inlet and discharge manifolds, repectively, as the lobes pass the delivery port. This noise can be reduced substantially by using three or more lobe rotors. Another, more satisfactory approach uses rotors with a helically shaped rotor. The helix layout is along the longitudinal axis. This configuration reduces the noise pulsations because the air delivery is smoother with less pressure spikes. The twisted rotor design provides nearly constant inlet and outlet volumes and pressures while minimizing losses in volumetric efficiency. In addition to twisted rotors, attention must be paid to the shape of the inlet and outlet ports to obtain optimum noise reduction. Besides reducing noise, twisted designs can also have higher efficiency.

The volumetric efficiency of a naturally aspirated engine is defined as the ratio of the actual volume of air drawn into the cylinder during one induction stroke to the geometric or theoretical swept volume of the cylinder. Losses in volumetric efficiency can be attributed to filters, carburetor or throttle valves, manifold ducting, port and valve restrictions, plus residual exhaust gases left from the previous cycle. Supercharging and turbocharging can improve volumetric efficiency, even above 100% in some cases.

Normally, boost pressure in mechanical superchargers such as the Roots-type blower is controlled through use of a clutch that can be disengaged or has controlled slippage. A bypass valve can also be used to relieve excessive pressures. The disadvantage of the latter technique is that it dissipates some of the energy that went into moving the air, representing a loss in efficiency and a waste of fuel. However, proper valve design can minimize this loss.

Rotary Piston Compressors

Rotary piston machines are generally true compressors in which the volume of a chamber changes between inlet and outlet resulting in an internal pressure rise. Many rotary piston compressors have been proposed, usually with an intersecting axis rotary piston configuration where the axis of the moving elements and the housing are not parallel and may operate with an oscillating motion. However, very few have been used in automotive applications to any extent, mainly because they are usually quite complex and require precise machining, leading to high manufacturing cost. Providing continual sealing during the complete operation cycle is one of the main design challenges.

Probably the most successful rotary piston design is the Kuhnle, Kopp & Kausch Ro-charger (Fig. 8). The design uses a twin-lobed piston that rotates eccentrically with respect to the unit's main axis of rotation. A three-

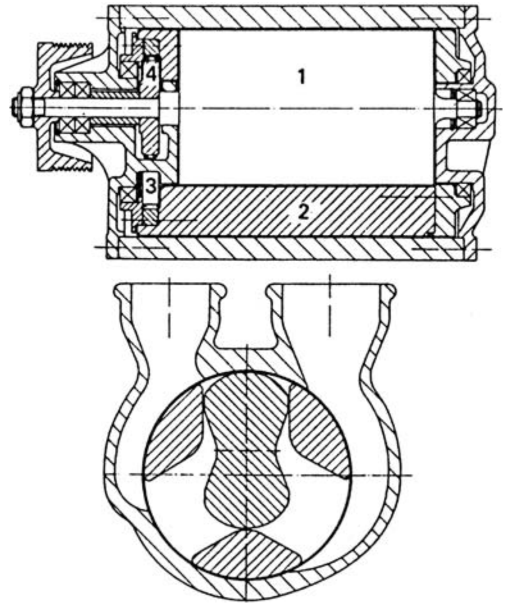


Figure 8a Layout of the Ro-charger rotary piston supercharger: 1, twin-lobe rotating position; 2, triple-chamber rotating cylinder; 3, ring gear that drives cylinder; 4, sprocket gear, which meshes with ring gear to drive piston drive shaft. (Courtesy Kuhnle, Kopp & Kausch AG.)

chamber cylinder with three inlet/outlet ports rotates on the main axis inside the Ro-charger housing. A ring gear and sprocket drive the outer cylinder and inner piston, using a 3:2 ratio to synchronize these elements. The Ro-charger is belt-driven by the engine crankshaft.

There are three compressions per cycle. Looking at one compression cycle, the cycle starts when the inlet port is open to the intake manifold. Here the chamber has its maximum volume as air enters it. As the cylinder rotates, a piston lobe rotates into the chamber and the internal volume decreases, increasing the air pressure. By the time the rotating cylinder reaches the outlet port, the volume is at a minimum and the delivery pressure is maximized. Future rotation forces the air from the chamber through the outlet port and the process repeats itself.

Positive sealing is not possible with the Ro-charger, so pressure losses are mininized by accurate machining of the components. This results in

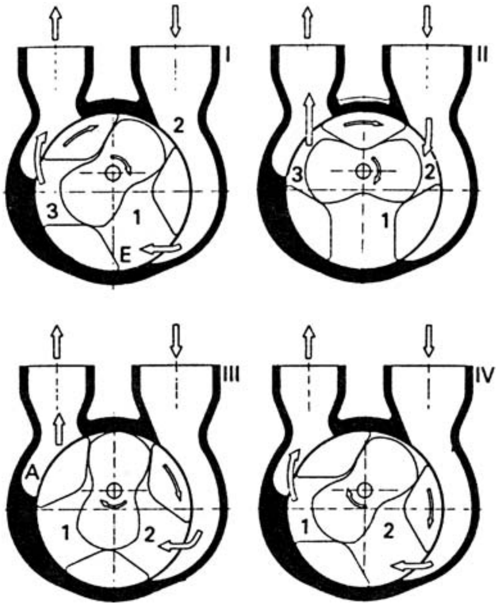


Figure 8b How the Ro-charger operates: Phase I, the cylinder is positioned so chamber 1 is fully opened to the intake port and air is drawn in. Phase II, chamber 1 is at maximum volume. Phase III, piston moves into chamber 1 to compress the air charge. Phase IV, chamber 1 is positioned to it is opened to the outlet port and air is discharged to the intake manifold. The cycle repeats itself as chamber 2 is now opened to the intake port.

minimum leakage between the various elements. The Ro-charger is a very compact unit relative to the volume of air handled.

Vane-Type Compressors

Vane-type compressors are closely related to piston types, but they have been used somewhat more frequently in practical automotive applications. A common feature of vane-type compressors is that they use a number of sliding vanes. These vanes divide the housing into compartments whose volume changes as the vanes rotate. The volume change occurs since the vanes are attached to a rotor or other device that rotates with an eccentric motion. At the beginning of the cycle, the compartment is open to the inlet manifold and the resulting vacuum draws in the air charge. As the device rotates, the compartment volume decreases in size to compress the charge. As the chamber volume reaches a minimum, it is opened to the discharge port and therefore the intake manifold [Fig. 9(a)].

One important advantage of piston and vane-type compressors over Roots-type blowers is that the compression occurs within the compressor itself. Internal compression increases the density and thus the weight of the air charge delivered during each revolution. The temperature rise for a given weight of charge air is also reduced. The pressure differential between the discharge port and the inlet manifold is reduced, improving the pumping efficiency. This means less power is required to drive the supercharger to achieve a given pressure increase.

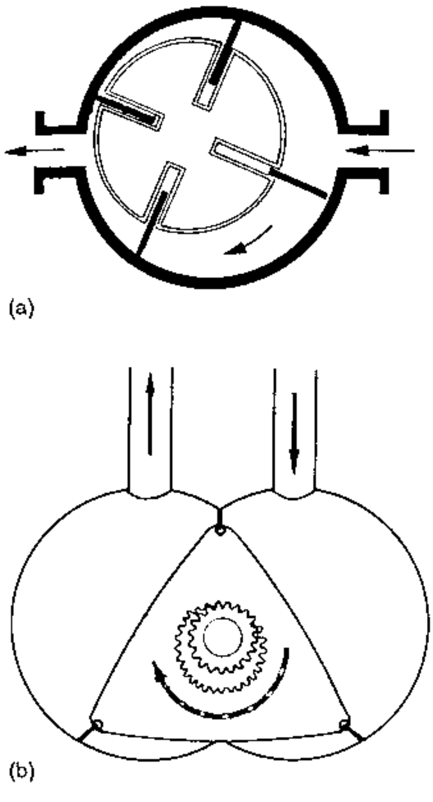


Figure 9 (a) Operating principles of a sliding vane-type supercharger. (b) A sliding vane-type supercharger that operates on the same principle as a Wankel rotor engine.

The vanes in many designs are prevented from actually touching the housing walls, so there is a very small clearance to reduce friction and eliminate vane tip wear. One of the problems with some vane-type compressors is the requirement to actively lubricate the sliding vane, adding to the complexity and resulting in possible oil contamination of the air charge. New materials can eliminate the need for lubrication. One successful design uses the same operating principles as the Wankel rotary automotive engine [Fig. 9(b)].

Screw-Type Compressors

The concept of using one or more helices within an enclosure for compression dates back to the 1800s. One of the more successful was the Lysholm screw compressor (Fig. 10). In this device, the twin helical rotors, one with four helixes, the other with six, rotate at different speeds, actually a 6:4 ratio, so they mesh as they rotate. The clearance volume between the two screws decreases along their length to perform the compression. The Lysholm screw compressor is quite compact and provides a pulseless discharge. However, it does require close-tolerance machining and assembly, since positive sealing is not possible. Both actively lubricated and dry-running designs have been used, the latter in automotive applications.

G-Laders

The G-lader mechanical supercharger was developed in the 1980s by Volkswagen as an economical competitor to the highly popular turbochargers (Fig. 11). The design was originally patented by Frenchman L. Creux in 1905. The design gets its name from the G-shaped spiral blower element that is moved eccentrically within another similar but fixed spiral. The blower spiral is rocked by another eccentric control shaft in such a manner that inlet and outlet ports are uncovered at the appropriate time as the volume of space between the spirals decreases. There is a duplicate chamber, operating out of phase to give a smoother as well as a higher volume of air charge.

Centrifugal Superchargers

The centrifugal supercharger, a kinetic turbomachine, uses a centrifugal compressor like that used by the turbocharger, except it is mechanically driven. Therefore, many of the features and characteristics covered in detail

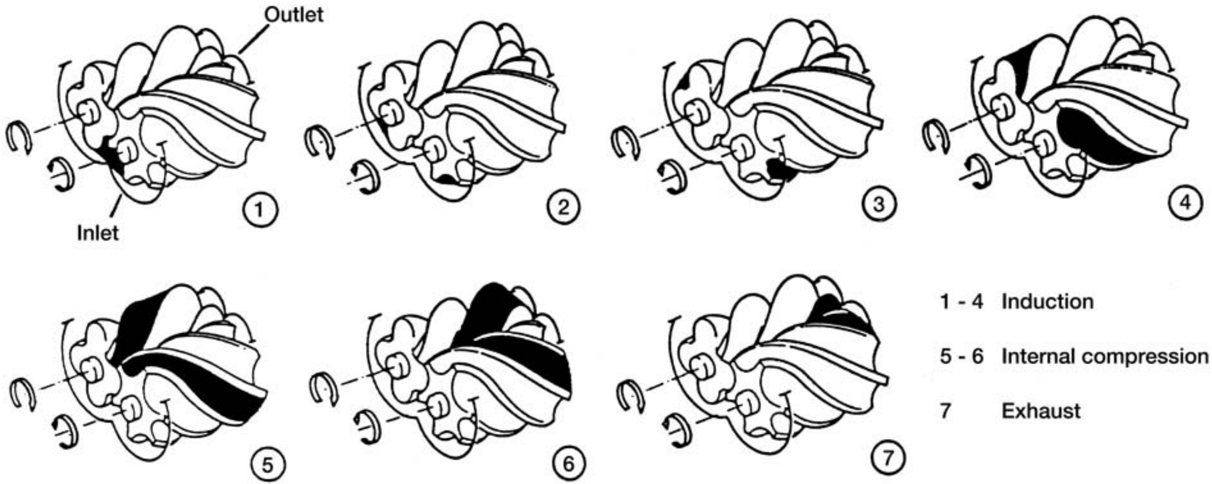


Figure 10 The Lysholm screw compressor uses two screw-type rotors with a different number of helices and turning at different speeds. Because of these differences, the volume of air trapped between the screws is decreased along their length to perform the compression. (Courtesy Fleming Thermodynamics, Ltd.)

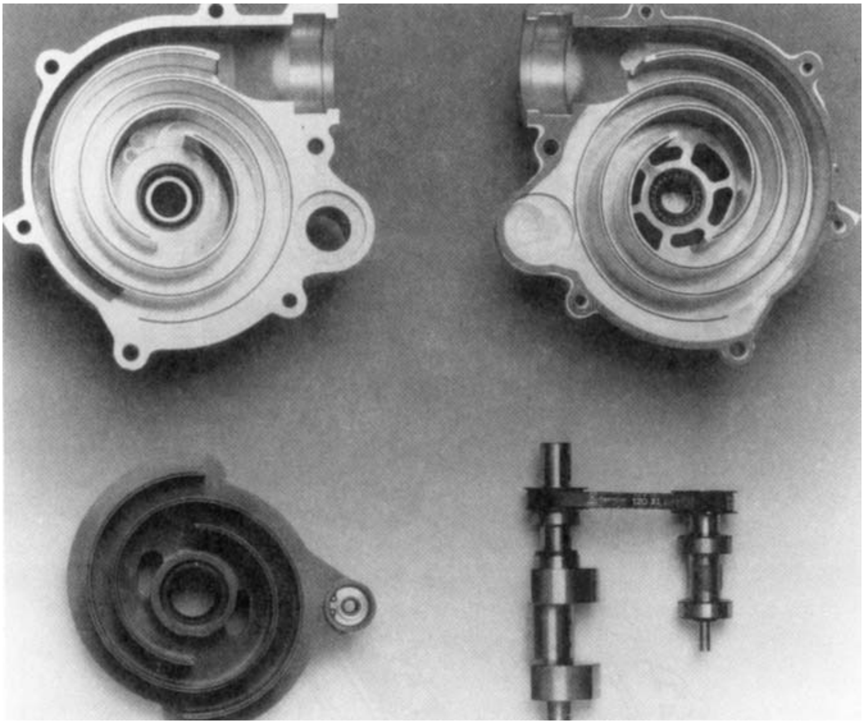


Figure 11a Key components of the G-lader supercharger. At the top are the two halves of the housing with the fixed G-shaped spiral chambers. In the lower left is the G-shaped rotating element that mates with the spiral in the housing. The shaft element that provides the oscillating motion is shown in the right corner. (Courtesy Volkswagen AG.)

in the next section on turbochargers apply to the mechanically driven compressor as well.

While a single-stage radial-flow compressor can provide a high pressure ratio, very high rotational speeds are required if the compressor is to be kept to a reasonable size. Unlike positive-displacement-type compressors, kinetic devices do not displace the same volume of air during each revolution. Therefore, in a centrifugal compressor, output pressure is proportional to the square of the rotational speed. Compressor speeds of up to 25,000 rpm may be needed before any noticeable positive boost pressure above atmospheric pressure is achieved. Since the input speed to the

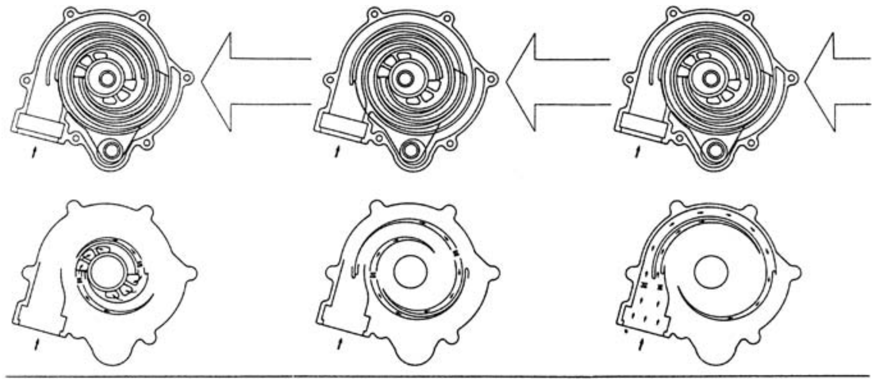


Figure 11b Principle of operation of the G-lader supercharger. Air brought in through the inlet port is compressed by the oscillation of the fixed and movable G-spirals before it exits the centrally outlet port at higher pressure. (Courtesy Volkswagen AG.)

supercharger is that of the engine crankshaft, roughly between 2,000 to 6,000 rpm, a high-ratio step-up in speed is required to reach maximum compressor speeds of up to 100,000 rpm or higher, as well as providing boost at low engine speeds.

The air volume required by a reciprocating engine is proportional to the speed of the engine. Thus there must be a matching of a proportional requirement with a squared air supply, which can become a problem. The usual solution to the matching challenge is to use a variable-speed drive. For instance, the venerable and still popular Paxton supercharger uses a gearless, planetary ball drive to provide smooth speed multiplication. The ZF-TURMAT centrifugal supercharger uses a combination of a variable-speed belt drive drive and 1:15 ratio planetary step-up gearing. In addition, the ZF-TURMAT can be turned off and on via an electromagnetic friction clutch on the input side of the supercharger.

Centrifugal superchargers are a popular alternative to turbochargers because they too are very compact relative to the volume of air handled. They also are relatively inexpensive to manufacture, are easy to install, and require minimum space in the engine compartment. The installation of a centrifugal supercharger does not require modification of the exhaust system like a turbocharger. However, the belt drive system used to drive the alternator, water pump, air conditioner compressor, and power steering pump may require revision (Fig. 12).

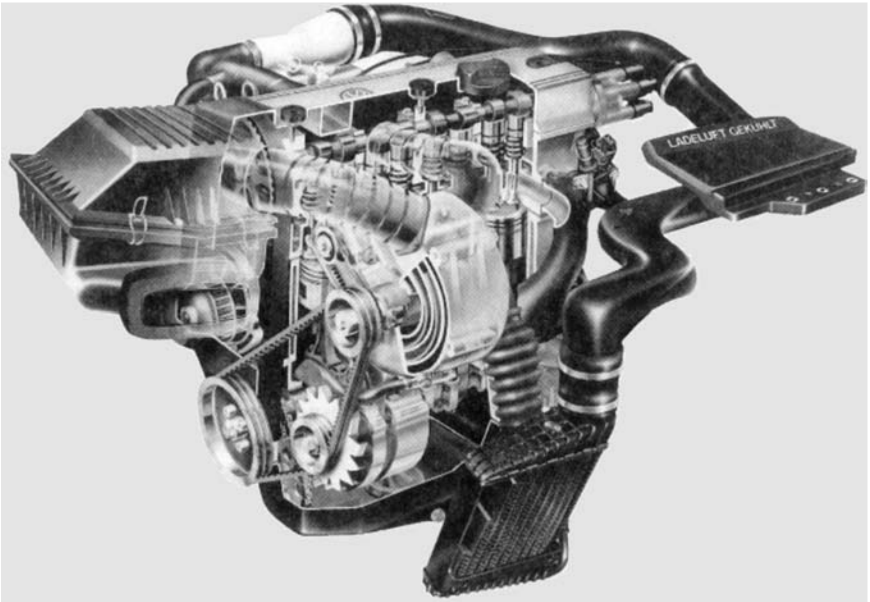


Figure 11c G-Lader supercharger is driven by the engine's crankshaft via a toothed belt drive. Air is taken in at the top of the supercharger. The compressed air leaving the central port passes through the air-to-air intercooler in the lower center of the picture. (Courtesy Volkswagen AG.)

Because the centrifugal supercharger runs cooler than an exhaust-driven turbocharger, an intercooler may not be needed. Also, heat shields and other thermal protection provisions associated with the installation of a turbocharger are not required. Most important, the centrifugal compressor provides virtually a near-instant boost without turbo lag. Finally, unlike many positive-displacement superchargers, the centrifugal supercharger is immune to backfire problems.

Maximum Compression Ratios

The boost pressure resulting from supercharging or turbocharging effectively increases the pressure in the cylinders. This has the same effect as increasing the compression ratio. There is a maximum compression ratio for spark ignition engines because excessive combustion chamber pressure can lead to denotation and potential engine damage (Fig. 13). This problem

Figure 12a Installation of a centrifugal supercharger requires some revision of the belt drive system so it can be driven off the crankshaft pulley. (Courtesy Paxton Superchargers.)

can be alleviated by using a fuel with a higher octane rating or by decreasing the effective compression ratio. The latter can be done by using a thicker head gasket or a decompression plate. More satisfactory solutions include removing metal from cylinder head to increase the volume of the combustion chamber, or better yet replacing pistons with ones that result in a lower compression ratio. Water injection into the cylinder can also help.

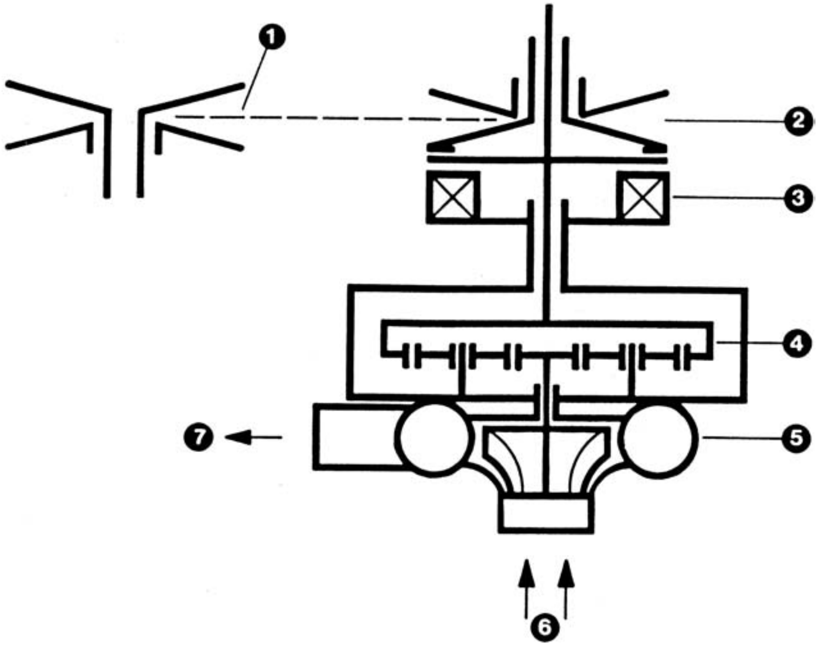


Figure 12b Schematic of centrifugal supercharger installation: 1, primary pulley for V-belt drive on engine crankshaft; 2, secondary pulley on turbocharger; 3, electromagnetic clutch; 4, planetary step-up gears; 5, centrifugal compressor; 6, air inlet; 7, air discharge. (Courtesy Zahnradfabrik Friedrichshafen AG.)

TURBOCHARGERS

Although there are several significantly different types of mechanical superchargers, exhaust-driven turbochargers used in automotive applications use the same basic design layout with differences in component design.

Turbochargers consist of a centrifugal compressor with radial outflow through either a vaneless or vaned diffuser housing (Fig. 14). The diffuser housing usually has a tapered constant-velocity volute collector that leads to a diffusing outlet duct that routes the higher-pressure air to the engine cylinders. The turbocharger turbine uses a centripetal radial-inflow exhaust-gas turbine, surrounded by a vaneless or vaned turbine housing connected to the exhaust manifold. The compressor and turbine wheels are mounted on a

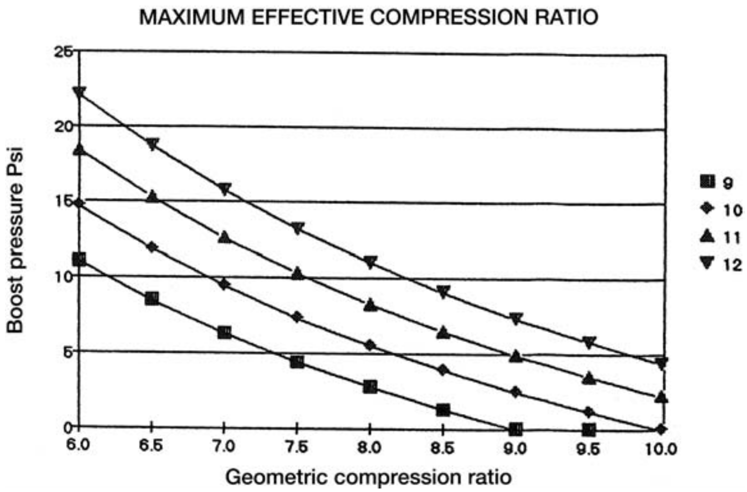


Figure 13 The geometric compression ratio is the ratio for a normally aspirated engine. The four maximum effective compression ratios—9, 10, 11, and 12 shown here—are the values where denotation could occur. This ratio depends heavily on the octane ratio of the fuel used.

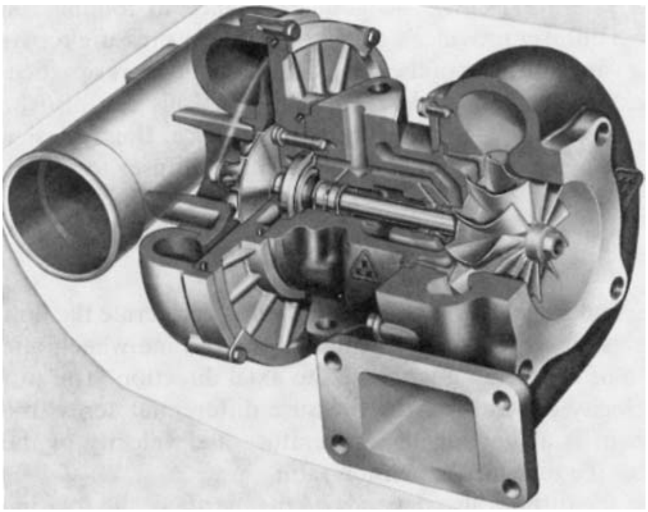


Figure 14a Example of a turbocharger for a spark ignition engine. (Courtesy Kuhnle, Kopp & Kausch AG.)

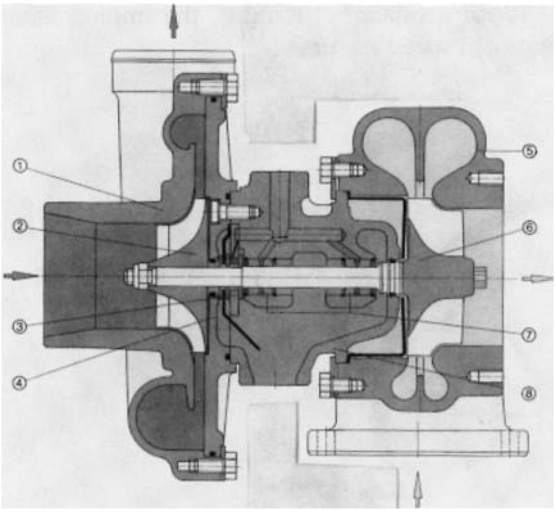


Figure 14b Components of the typical turbocharger: 1, compressor housing; 2, compressor wheel; 3, thrust bearing; 4, compressor backplate; 5, turbine housing; 6, turbine wheel; 7, bearings; 8, bearing housing. (Courtesy Kuhnle, Kopp & Kausch AG.)

common shaft and rotate at the same radial velocity. The shaft runs on bearings and usually uses thrust rings and has oil seals at both ends. Because of the high speeds involved, the unit requires very precise balancing.

In a radial-flow turbocharger, the gas flow nearest the hub will be parallel with the rotation axis and the flow will be at right angles to the hub at the perimeter of the compressor or turbine wheel. For the compressor, inlet air flows parallel to the hub and exits at right angles to the hub. For the turbine, the exhaust gases enter the turbine at the outer perimeter at right angles and are turned so they exist parallel to the hub. Some turbochargers use a mixed-flow design where flow at the wheel perimeter is less than at right angles to the hub axis, so that it is a combination of axial and centrifugal flow, that is, "mixed."

In axial-flow turbomachines, the direction of flow of the working gases remains essentially parallel to the axis of rotation. While used in very large engine applications, they are rarely, if ever, used in automobiles or trucks. Therefore, axial-flow turbochargers are not discussed further.

Compressor Design

The compressor impeller spins at very high speeds so that centrifugal forces accelerate the air to very high velocities so they attain a high level of kinetic energy. This high kinetic energy is then converted to an increase in static pressure in the diffuser section, which has a gradually increasing cross-sectional area. As pressure increases, the air velocity is significantly reduced. A collector or volute then collects the air flow around the perimeter of the diffuser and delivers it to the exit duct, where it flows to the engine cylinders.

There are several compressor designs in use. The simplest uses straight blades, which are easy and inexpensive to manufacture. However, straight blades result in relatively low compressor efficiencies because the shock waves are produced at the blade tips. Shock loss can be reduced by using curved inducer blades. Here the angle of curvature at the leading edge is chosen so it has exactly the same angle as the air entering the compressor. Finally, backward-curved inducer blades can be used. Here the blades are curved backward from the direction of rotation. For a given pressure rise, backward-curved blading requires a higher tip speed but the exiting velocity is lower. This means a greater percentage of the overall pressure rise takes place within the impeller and less occurs in the diffuser. Curved blades are more difficult to produce and are subjected to greater bending stress at the blade roots. Currently, backward-curved blades offer the highest peak compressor efficiency. Shrouded impellers have been used because they reduce air-flow recirculation within the blade, leading to maximum efficiency. They are now rarely used because they are difficult to manufacture, tend to be weaker, and are subject to dirt accumulation.

Early compressor diffusers used a series of divergent nozzles to convert kinetic energy to pressure rise via volume expansion. Nozzles have been replaced in current turbochargers by a ring of angled vanes that provide the required divergent passages. Vaned diffusers are usually used with compressor impellers with radial blades. The simplest, and now the most popular, design uses a vaneless diffuser usually with parallel walls. The volume of space between the walls increases with increasing radial distance from the impeller tip. The vaneless diffuser allows a broad operating range, low manufacturing cost, and good resistance to fouling and erosion. The vaneless diffuser provides a significantly lower-pressure recovery compared with a vaned diffuser of the same diameter and works particularly well with backward-curved blades. The diffuser walls can also be designed with a nonlinear area increase with radial distance. In any design it is important to ensure smooth internal surfaces in the diffuser to reduce frictional losses.

Turbine Design

The turbine housing and wheel together are designed to accelerate the flow of exhaust gases, transfer their energy to rotating the turbine wheel, and change the direction of flow from a tangential to axial direction. The turbine's rotational velocity depends on the pressure differential across the turbine, which in turn is a function of the temperature and velocity of the exhaust gas as well as the turbine's expansion ratio.

The velocity of the turbine also depends on the inertia of the rotating parts, namely, the turbine and compressor wheel, and the A/R ratio of the turbine housing. The A/R ratio is the area of the turbine inlet measured at its narrowest point, or throat, divided by the distance from this point to the turbine's rotational axis (Fig. 15). A large A/R ratio means the exhaust gases impinge on the turbine wheel at a shallow angle, resulting in a relatively low rotational speed. With a smaller A/R ratio, the impingement angle is increased so the rotational speed increases.

If the turbine housing entry has a circular cross section, calculation of the A/R ratio is quite straightforward. Otherwise the A/R ratio can be calculated using both conservation of mass and angular momentum principles. The latter results in the requirement that turbine housing volute provide a free vortex flow field at the tip of the turbine wheel, meaning the tangential velocity must vary from its magnitude at the inlet throat of the housing to the velocity needed to match the turbine wheel tip speed. The calculations assume two-dimensional flow and neglects both friction and

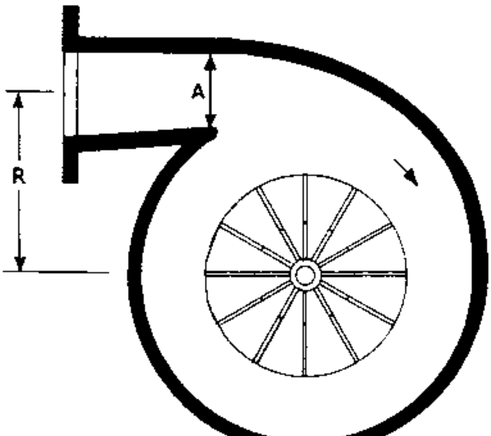


Figure 15 Definition of A/R ratio parameters (circular cross section).

compressibility effects. Referring to Fig. 16:

$$A/R = Q/K$$

Where

Q = flow rate of exhaust gases = AV_t.
A = cross-sectional area of the volute at the tongue.
V = tangential velocity component at the tongue.
$K = 2\pi R_w^2 N_w$.
R_w = wheel tip radius.
N_w = wheel tip velocity.

Finally, R_{DC} is defined as the dynamic center and determines the location that divides the area of the scroll into two areas, each supplying half the flow volume:

$$R_{DC} = A/(A/R)$$

Normally, the turbine freely turns at least 25,000 rpm under cruise conditions. Then when the throttle is opened wider, the carburetor or fuel injection system supplies more fuel, which is combusted, producing more exhaust gases. This additional thermal energy accelerates the turbine and compressor wheels so the compressor supplies more pressurized air to the combustion chambers. The rate of acceleration of the wheels depends on the moment of inertia of the rotating components, the A/R ratio of the turbine housing, and the amount of thermal energy available in the exhaust gases. Turbo, turbine, or throttle lag is the time delay while the turbocharger's rotating parts accelerate in response to increased gas flow that results from increased throttle opening. This lag results in a delay of full power delivery. The lag time depends on the amount of engine load and speed as well as the

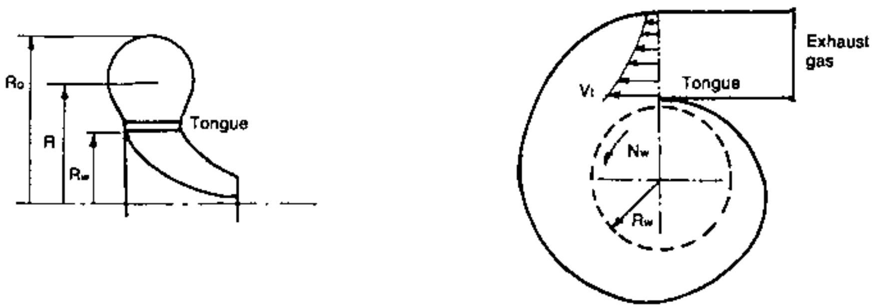


Figure 16 Definition of A/R ratio parameters (general configuration).

inertia of the wheel assembly. Minimum lag results with small-diameter turbine wheels that have a minimum of mass located near the wheel's perimeter.

In order to increase the response of small turbochargers, multiple entry passages, usually two, are sometimes used. The multiple entry design improves the low-speed turbine response by taking advantage of the natural exhaust gas pulse energy created when the exhaust valves open.

The earliest turbochargers used radial blades in the gas turbine, which led to rather larger turbine wheel diameters that were also heavy and had high inertia. Both installation-size constraints and performance response dictate small, lightweight turbines with low intertia. Turbine design has tended toward achieving more axial flow in the turbine.

A turbine with nozzles is the equivalent of the vaned diffuser in a compressor. Its main purpose is to add a swirling motion to the exhaust gases before they enter the turbine wheel. While the gas velocity increases as the gases pass through the nozzle vanes, in practice with small cross-sectional turbine inlet ports and those with multiple entry ports, the inlet velocities are already very high so only a small amount of further acceleration takes place within the nozzle vanes. Actually it is possible to dispense with the nozzle vanes entirely. Then the turbine housing has to be shaped so it produces the required initial swirling motion. Like vaneless compressors, nozzleless turbines are now very common because they are less expensive to manufacture and are very reliable while still providing acceptable efficiencies.

One desirable option is the variable-nozzle turbine such as the VNT (Variable Nozzle Turbocharger) developed by Chrysler for its four-cylinder engines (Fig. 17). Here the nozzle vanes pivot to vary the area between vanes, which changes the power output characteristics of the turbine. In the VNT, 12 aerodynamic vanes pivot individually on a nozzle ring. The vanes are moved by the unison ring to adjust the flow of exhaust gases to the turbine wheel in response to control inputs from the system computer. At idle or while cruising when turbo boost is not needed, the vanes are held at the 40–50% open position, resulting in minimum exhaust restriction. When the throttle is opened, the vanes are momentarily positioned to restrict flow to increase exhaust gas velocity to rapidly accelerate the turbine to increase boost at low rpm for greater transient low-speed torque and better response. As engine and power speed increase, the exhaust flow increases and the vanes are reopened to obtain higher flow for maximum boost and power.

The disadvantages of variable nozzles include their greater complexity, and thus added expense, and need to provide linkages or other techniques to move the nozzles. The latter is complicated because power servos are needed

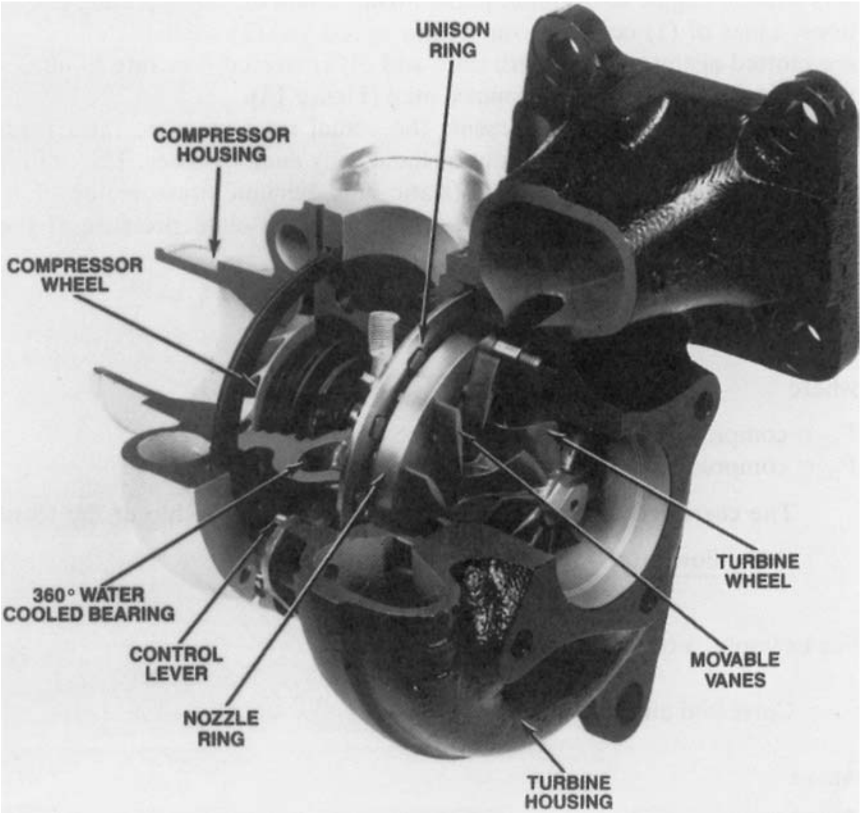


Figure 17 Variable nozzle turbocharger. (Courtesy Chrysler Corporation.)

to move vanes under high gas pressure forces, and mechanical parts must operate reliably in a very-high-temperature environment without lubrication.

Compressor Performance Maps

Compressor performance maps are used to describe the performance of particular centrifugal compressors. They are also used to match compressors with an engine to optimize performance under actual operating conditions. Lines of (1) constant compressor speed and (2) adiabatic efficiency are plotted against (3) pressure ratio and (4) corrected flow rate to obtain the typical compressor performance map (Fig. 18).

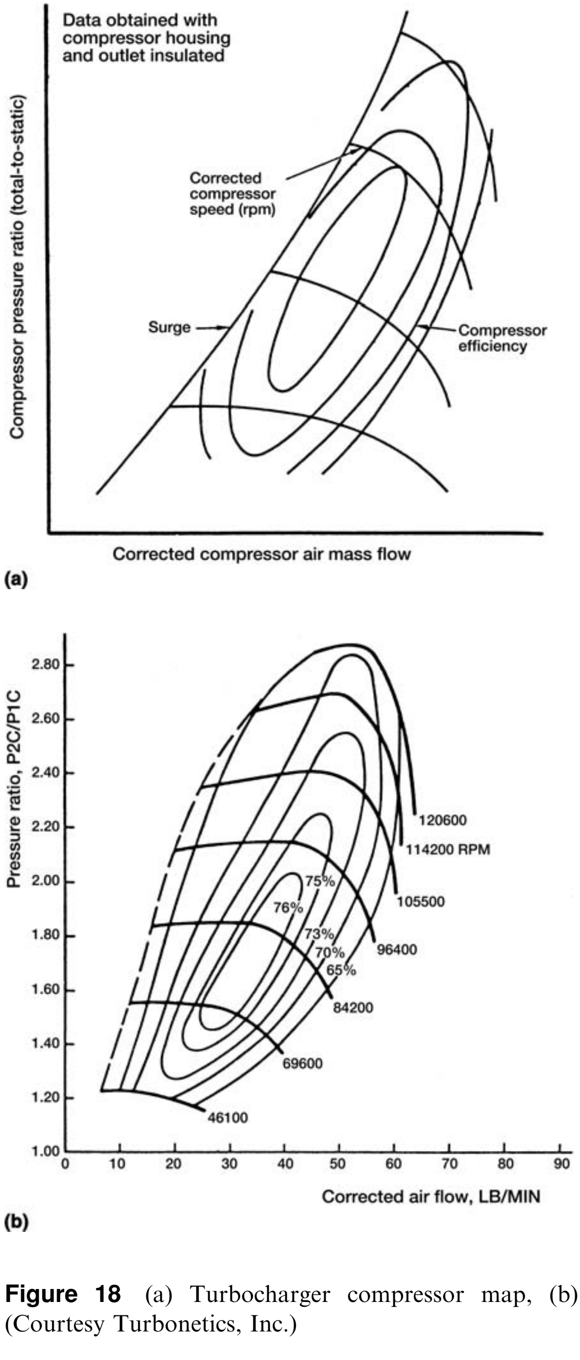


Figure 18 (a) Turbocharger compressor map, (b) typical compressor map. (Courtesy Turbonetics, Inc.)

The pressure ratio represents the actual total pressure ratio from compressor inlet to outlet without including any ducting losses. The ratio is determined by dividing the total (static plus dynamic pressure) absolute pressure at the compressor outlet by the total absolute pressure at the compressor inlet:

$$\text{pressure ratio} = \frac{P_{2C}}{P_{1C}}$$

Where

P_{1C} = compressor inlet air total pressure (in. Hg abs.).
P_{2C} = compressor discharge air total passage (in. Hg abs.).

The corrected air flow is corrected using a relationship of the form

$$\frac{(\text{mass flow rate}) \times (\text{absolute temperature})^{1/2}}{(\text{pressure})}$$

For example, a typical correction is

$$\text{corrected air flow} = \frac{W_a (T_{1C}/545)^{1/2}}{P_{1C/28.4}}$$

Where

T_{1C} = compressor inlet air total temperature (degrees R).
W_a = compressor discharge air flow (lb/min).

Likewise, the compressor speed is corrected using a a correction of the form

$$\frac{(\text{compressor speed})}{(\text{temperature})^{1/2}}$$

A typical correction is

$$\text{corrected compressor speed} = \frac{N}{(T_{1C}/545)^{1/2}}$$

where N is the compressor speed (rpm).

The basis for the adiabatic compression efficiency is an adiabatic compression in which the pressure increases without any thermal loss or gain. However, the temperature increases with increased pressure. This ideal temperature is compared with the actual higher temperature obtained in practice due to energy losses to determine the adiabatic efficiency.

Theoretically this efficiency is:

$$\text{compressor efficiency} = \frac{\begin{array}{c}\text{isentropic enthalpy rise across compressor stage}\\ \text{through compressor pressure ratio}\end{array}}{\text{actual enthalpy rise across compressor stage}}$$

or

$$\text{compressor adiabatic efficiency} = \frac{T_{1C}Y}{T_{2C} - T_{1C}}$$

Where

T_{2C} = compressor outlet air total temperature (degrees R).
$Y = (P_{2C}/P_{1C})^{0.283} - 1.$

There are three main areas on the centrifugal compressor map. The central area is the stable operating zone. The surge line in the low flow rate area at the left of Fig. 18(a) defines an area of pressure and flow rate where compressor operation is unstable. Surging occurs when the mass flow rate through the compressor is reduced while maintaining a constant pressure ratio until a point is reached where local flow reversal occurs in the boundary layers. If the flow rate is further reduced, complete reversal occurs. This will relieve the adverse pressure gradient until a new flow regime is established at a lower pressure ratio. The flow rate will again build up to the initial condition, and this flow instability will continue at a fixed frequency and in some cases can become quite violent.

The compressor map shows a tradeoff in compressor design. The narrower the operating range of the compressor, the sharper the surge. However, the broader the range, the lower the peak efficiency.

Turbine Performance Map

The performance map for the turbine shows the lines of constant corrected speed, corrected exhaust gas flow, and overall turbine efficiency plotted against turbine pressure ratio (Fig. 19). The total-to-static pressure ratio is used here since the actual exit conditions cannot be predicted for a particular installation.

$$\text{pressure ratio (expansion ratio)} = \frac{P_{1T}}{P_{2T}}$$

Where

P_{1T} = inlet gas total absolute pressure (in. Hg).

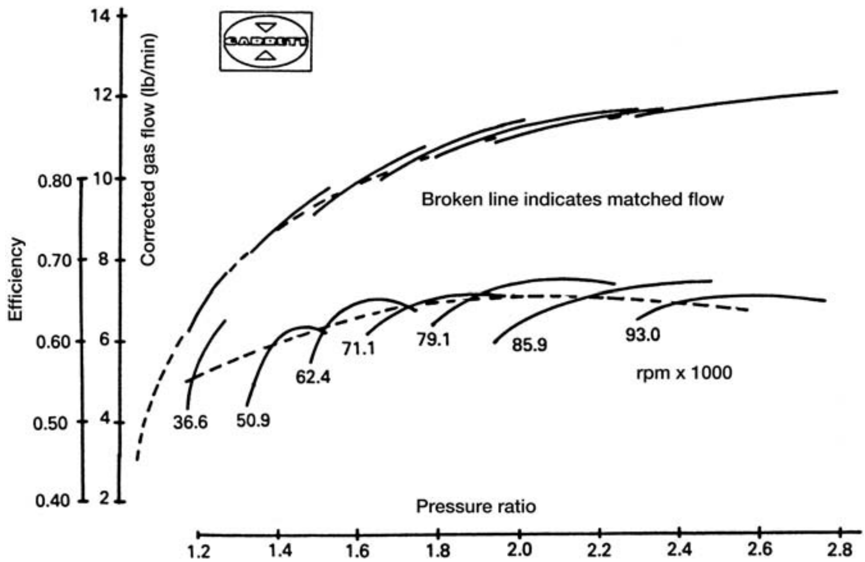


Figure 19 Typical turbine performance map. Because the gas flow versus pressure ratio is hardly different at different turbine speeds, it can be quite accurately approximated by a single curve. (Courtesy Allied Signal/Garrett Turbocharger Division.)

P_{2T} = outlet gas static absolute pressure (in. Hg).

The turbine gas flow is

$$\text{corrected gas flow} = \frac{W_g (T_{1T})^{1/2}}{P_{1T}}$$

Where

T_{1T} = turbine inlet gas total pressure (degrees R).
W_g = turbine gas flow (lb/min).

The corrected turbine speed is

$$\text{corrected turbine speed} = \frac{N}{(T_{1T})^{1/2}}$$

where N is the turbine speed (rpm).

The overall turbine efficiency includes both the turbine efficiency and the mechanical efficiency, which includes bearing losses.

$$\text{combined efficiency} = \frac{\text{actual enthalpy rise across compressor}}{\text{isentropic enthalpy drop through turbine expansion ratio}}$$

Altitude Effects

The turbine and compressor speed depends on the pressure differential between the turbine inlet and outlet. The greater this pressure differential between the exhaust manifold and atmospheric pressure, the higher the turbine and compressor speed. Since the manifold pressure remains relatively constant, as atmospheric pressure decreases, such as when climbing in altitude, the increase in speed will cause the compressor to increase boost pressure. Thus a turbocharger is largely self-compensating with respect to altitude.

Boost Control

It is necessary to limit the maximum boost pressure provided by a turbocharger. For example, the typical turbocharger used on conventional automobiles and light trucks is designed to provide a maximum boost pressure of 5–10 psi. By comparison, turbochargers for motorsports competition (road racing, hill climbing, rallying, oval track, etc.) use boost pressure up to 25 psi, while short-duration events like drag racing and tractor pulling can tolerate boost pressures in the 20–75 psi range.

There are three reasons for controlling boost pressure in turbocharged systems. First, excessive boost pressures can lead to preignition and denotation in the combustion chamber, leading to severe engine damage. Second, turbocharger speeds have to be limited to prevent self-destruction. Finally, speeds have to be controlled so that the turbocharger can be matched to the engine to obtain the desired performance characteristics over the wide operating range found in automotive applications.

The earliest boost controls consisted of fixed restrictions in the inlet or exhaust ducting, or even both. Incidentally, even the air filter or carburetor choke can provide a measure of boost control. Fixed restrictions provide progressive control that varies as the square of the exhaust flow. For example, when the restriction limits the maximum boost pressure, there is also a corresponding restriction at lower rpm, resulting in less efficient engine performance and poorer throttle response.

Currently, the most common method of controlling turbocharger boost is through the use of a wastegate. An externally mounted wastegate is mounted on or near the exhaust manifold or turbine housing. The wastegate provides a flow path through which exhaust gases can bypass the turbine, going directly to the discharge exhaust pipe. A common form of wastegate uses a poppet valve connected to a diaphragm (Fig. 20). The chamber in front of the diaphragm is connected by a pressure feed line that senses boost pressure downstream of the compressor housing in the intake manifold. When the boost pressure exceeds a threshold value, the pressure on the diaphragm begans to overcome the pressure exerted by the spring on the otherside of the diaphragm, thus compressing the spring and opening the poppet valve. This lets an increasing percentage of the exhaust gas travel to the turbine exhaust pipe without passing through the turbine, halting the acceleration of the turbine and the subsequent increase in boost pressure. The amount of boost pressure is determined by the strength of the spring, sometimes referred to as the cracking pressure. Springs can be changed to vary boost pressure. Some more expensive wastegates have the capability to

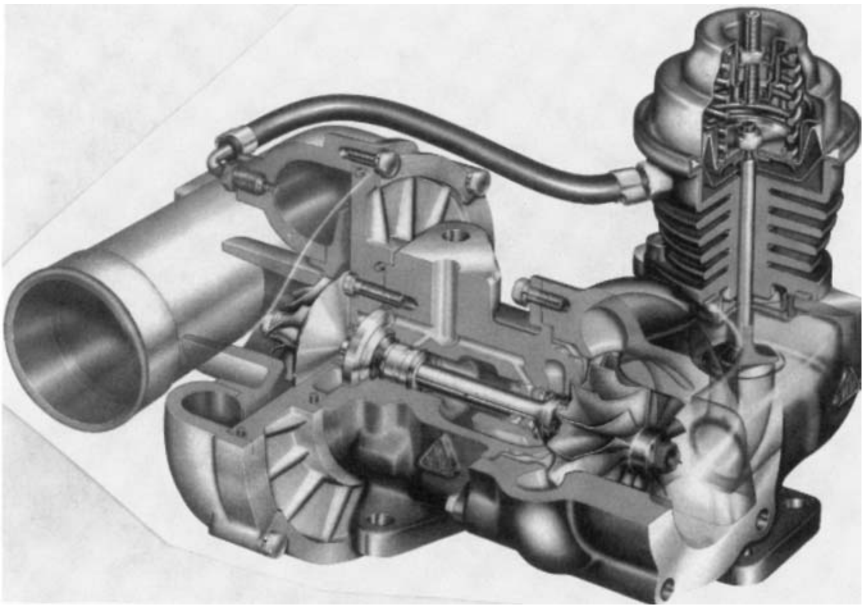


Figure 20 Turbocharger fitted with an integral boost pressure control valve. Boost pressure sensed in the discharge outlet from the compressor is transmitted via the tubing to the diaphragm side of the control valve. When boost pressure exceeds the spring force, the poppet valve is opened so a portion of the exhaust gas bypasses the turbine. (Courtesy Kuhnle, Kopp & Kausch AG.)

adjust the spring force to vary the boost pressure that will open the valve. Using a wastegate reduces overall fuel consumption efficiency because as the title implies, energy is "wasted" when exhaust gases bypass the turbine.

There are several alternate wastegate designs. For example, a swinging arm-type valve can be used (Fig. 21). In smaller engine applications, the wastegate is integrated into the turbine housing, thus eliminating the need for external high-temperature exhaust-line joints that can lead to problems.

The other class of boost control devices involves a variable-geometry turbine that effectively changes the A/R ratio and thus the turbine wheel speed and in turn the boost pressure. The simplest of these devices uses a turbine housing scroll with twin inlet passages, one of which can be closed

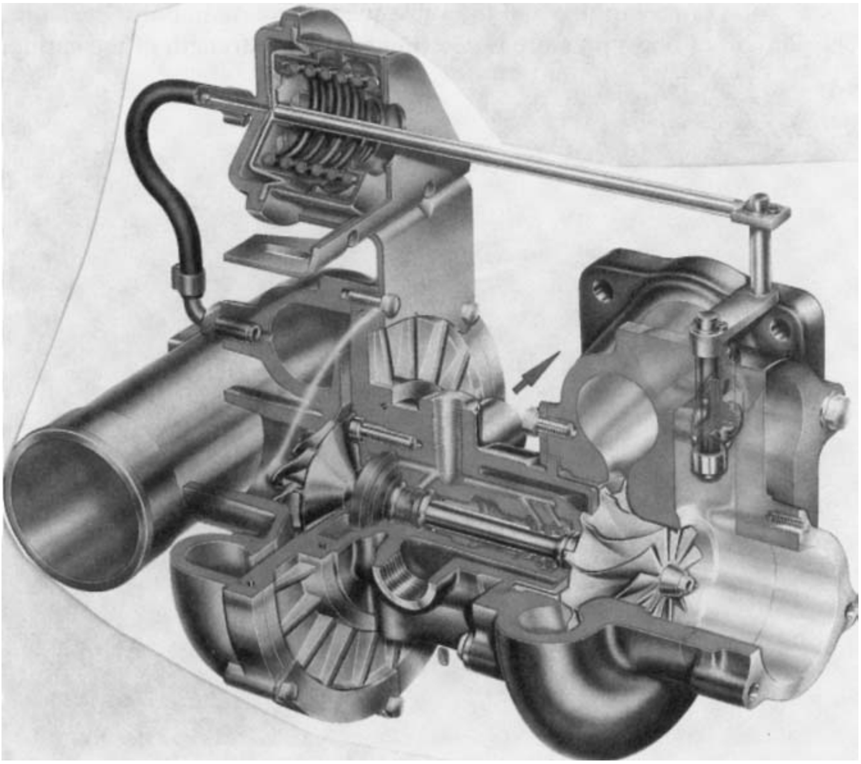


Figure 21 Turbocharger fitted with swinging arm-type valve. Boost pressure sensed in the discharge outlet from the compressor is transmitted via the tubing to the diaphragm side of the control valve. When boost pressure exceeds the spring force, the shaft opens the swing valve so a portion of the exhaust gas bypasses the turbine.

off by a selector valve. This design has only two modes of operation, exhaust gas flow with one or both passages open, and therefore only two A/R ratios are available. Disadvantages of this approach include reduced turbine efficiency caused by wake loss and additional surface friction, plus high-temperature stresses on the thin dividing wall. An alternate version of this concept uses a horizontally sliding gate that closes off part of the turbine housing volute at low speeds.

One more flexible variable-nozzle boost control concept uses a single movable vane placed at the turbine inlet (Fig. 22). This device changes the A/R ratio by changing the effective throat area over a relatively large range. The design is potentially quite reliable because of the limited number of moving parts and offers a relatively large operating range. There is a reduction in efficiency because of gas leakage through the side gap between the vane and housing, wake loss, and a nonuniform velocity distribution.

A more complex variable-geometry design uses multiple nozzle vanes located on the perimeter of the turbine housing. (The VNT design previously mentioned is one example.) The individual vanes are linked together so they mechanically move in unison. Changing the passage area between the vanes is used to control the turbine speed. While more complex, more costly to manufacture, and somewhat less reliable, the design provides excellent efficiency and a very wide operating range. A larger turbine

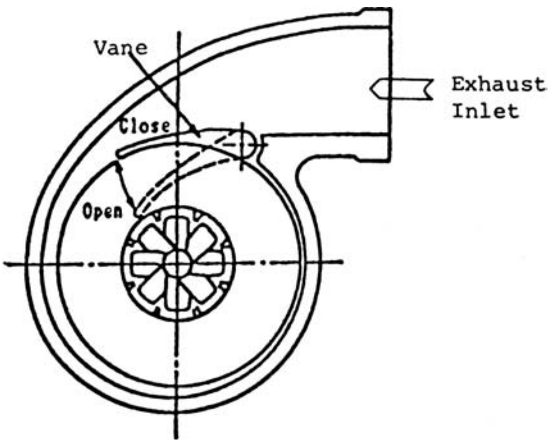


Figure 22a Single vane used to control exhaust gas velocity supplied to turbine wheel is a simple means of controlling boost pressure. (Courtesy Mitsubishi Heavy Industries America, Inc.)

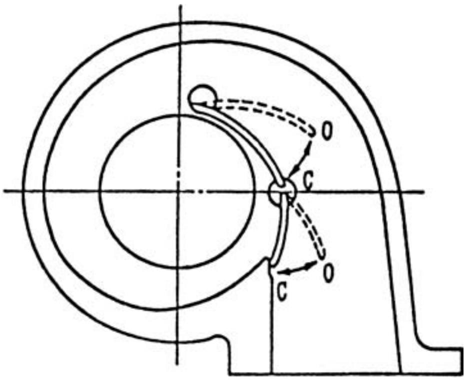


Figure 22b Two vanes are used to provide a form of variable inlet turbine. (Courtesy Mitsubishi Heavy Industries America, Inc.)

housing is also needed to house the vanes, and the substantial exhaust gas forces on the vanes require a heavy-duty servo system to move them. While variable-geometry designs provide boost control, they are often used with wastegates in order to match the requirements for boost control over the speed ranges found in passenger-car applications.

Bearing Design

There are several possible designs for bearing location. Early turbocharger designs often used a straddle design where bearings were located at the end of the common shaft for the turbine and compressor wheels. The design allows sufficient space for sealing and is conducive to water cooling of the bearings. An alternative, older design uses a single overhung layout where bearings are only located on one side, usually on the cooler compressor side. The large bending moment resulting from the cantilever design leads to significant bending stresses on the shaft.

Most current automotive turbochargers use a double overhung layout where bearings are located between the two wheels. The design results in a low-cost, lightweight, compact configuration. Bearing sealing and temperature differential problems are quite solvable. Most of these small-size turbochargers use sleeve-type bearings that are both cooled and lubricated by oil from the engine lubrication system.

Intercoolers

As the compressor increases boost pressure and charge density, the temperature of the air also increases. This increase in temperature results in an increase in air volume, thus decreasing the charge density a bit. However, a bigger problem is that higher air temperatures can cause excessive thermal stress on engine componentry and produce conditions that can lead to denotation and preignition. The solution now often used is the installation of an intercooler.

The temperature increase in the compressor (Fig. 23) can be estimated by the relationship

$$T_2 = T_1(P_2/P_1)^{0.286}$$

Where

T_1 = absolute temperature of air entering compressor.
T_2 = absolute temperature of air leaving compressor.
P_1 = absolute pressure on inlet side of compressor.
P_2 = absolute pressure on outlet side of compressor.

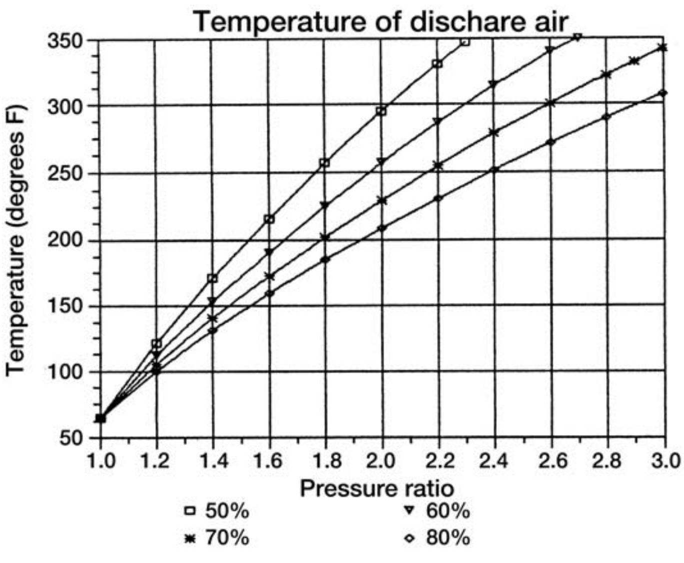


Figure 23 Increase in air discharge temperature as a function of boost pressure ratio for various turbocharger efficiencies.

However, the temperature has to be modified by the adiabatic efficiency of the compressor. Therefore, the actual temperature increase is

$$\Delta T = \frac{(T_2 - T_1)}{n_{\mathrm{ab}}}$$

where n_{ab} is the compressor's adiabatic efficiency.

Intercooling, aftercooling, and charge air cooling are all terms used to describe the method of cooling the air inlet charge by installing a heat exchanger in the inlet manifolding between the compressor discharge and the engine. The intercooler is used to cool the charge air to increase the density of the charge and therefore its mass flow into the engine. The greater mass flow of air allows a greater quantity of fuel/air mixture converted to power. Intercooling also reduces the thermal loading on engine parts like valves and pistons. Reducing charge air temperature also reduces the problem of preignition and denotation in spark ignition engines.

Intercooler heat exchangers are usually of the air-to-charge-air or water-to-air type. Unlike a radiator, the core through which air flows should offer as little restriction as possible, and at the same time it must present the maximum surface area with cooling medium. There are several cooler configurations for automotive applications. Most common systems use an air-to-air heat exchanger located in front of the main radiator, or alternately at any convenient location at the front of the vehicle where it can take best advantage of the airstream. Since turbocharging is used normally when the vehicle is moving, the ram-air effect should be sufficient, eliminating the need for a fan. Plastic or rubber ducting can be used to bring air from the compressor and deliver it to the engine after being cooled by passing through the intercooler.

In more rarely used air-to-water intercooler designs, the cooling water can be simply normal engine coolant that is circulated through the intercooler. More complex air-to-water designs use a separate cooling circuit. Besides the intercooler for cooling the air charge, there are another water-to-air radiator to reject this heat and a pump to circulate the coolant.

Intercooling does result in some disadvantages. When air flows through the heat exchanger, there is a pressure drop offsetting some of the density increase resulting from cooling. Also, space must be provided in the engine compartment for the heat exchanger and associated plumbing. Excessive cooling can lead to condensation in the inlet manifold. Finally, intercoolers add to the expense of the vehicle.

TURBOCHARGING DIESEL ENGINES

Turbochargers are widely used on compression ignition (diesel) engines in both automobiles and trucks. Although operating on the same principles, there are some differences compared to turbocharging a spark ignition (gasoline) engine. Automotive diesel engines can be supercharged, but the practice is rare in production applications. The performance improvements gained by turbocharging a diesel engine are so dramatic that today most automobile and light-truck diesels are turbocharged, as are virtually all heavy-duty, over-the-road truck engines.

In the diesel engine, only air is drawn into the cylinder. The air is compressed, increasing its temperature above the point needed to ignite the separately injected fuel. Combustion occurs relatively slowly as the fuel mixes with the high-temperature air. The power output is controlled by varying the amount of fuel injected. Unlike a spark ignition engine where the air/fuel mixture ratio is nearly constant at or very near the stoichiometric value, in a diesel engine it is only necessary to have sufficient air to assure complete combustion of the fuel injected. Excess air is always acceptable, and indeed desirable, in a diesel engine. The power output of a diesel engine is dependent on the amount of fuel combusted, and the engine's maximum power output at any speed is determined by the amount of air available to burn the fuel efficiently. Insufficient intake air for a given load condition will result in a rapid decrease in efficiency. The maximum air mass flow for a normally aspirated diesel engine is limited by engine cylinder displacement. To increase power output, either the engine speed or displacement has to be increased.

A turbocharger can substantially increase air mass flow and density. When additional fuel is added to the increased air mass, the resultant power increase is limited basically only by the mechanical and thermal capabilities of the engine. A turbocharger for an automotive diesel should allow engine operation over a relatively wide speed range for the desired flexibility needed for highway use. The turbocharger should provide boost from as low an engine speed as possible, for example, 1,200 rpm, without causing excessive back pressure in the exhaust manifold. High back pressures can be reflected back to the engine, causing high cylinder pressures and an increase in fuel consumption.

The excess air supplied by a turbocharger can reduce combustion chamber and exhaust temperatures, providing higher power output without the need for very high-temperature-resistant, and thus more expensive, materials. Also, the fuel control system can be simplified when the engine is supplied with excess air. Excess air can scavenge the cylinders at the end of the exhaust stroke to remove the last traces of exhaust gases. Finally, excess air can reduce carbon monoxide and unburned hydrocarbon emissions. The lower combustion temperatures that result with excess air lead to a reduction in nitrogen oxide emissions.

Operating a diesel's engine's turbocharger continuously to provide boost under full-load conditions can result in excessive thermal loads on the engine, leading to reduced engine life. An intercooler can be used to reduce this damaging heat load. Unlike intercoolers on turbocharged or supercharged gasoline engines, which are added to extract more power by eliminating denotation, the diesel's intercooler only helps reduce operating temperatures for durability and reliability.

The absence of denotation problems allows much higher boost pressures even when diesels normally operate at much higher compression ratios. Currently, maximum pressures are limited by the mechanical strength of engine components as well as by the maximum injection pressures that can be provided by an efficient and reliable fuel pump that must work against the high combustion chamber pressures.

One technique that can be used to extract more energy from a turbocharger is to make use of the high-pressure, high-velocity exhaust pulses generated each time an exhaust valve opens. This peak pulse pressure can be many times higher than the average exhaust manifold pressure. To utilize these pressure pulses, the exhaust system must be designed so that the high-pressure peaks are not canceled out by the low pressures, as is the case when only a single-inlet turbine housing is used. The pulse pressure technique requires from each cylinder head a separate exhaust connection that delivers the exhaust gases to a special divided inlet housing.

Turbocompounding

Additional energy can be extracted from the exhaust by using turbocompounding. While this can be done in several ways, one successful turbocompound diesel truck engine is marketed by Saab Scandia (Fig. 24). Here a second power turbine is located downstream from a conventional turbocharger in the gas stream to extract the additional thermal energy before it is exhausted out the exhaust pipe. This second turbine is mechanically connected to the crankshaft to utilize the power extracted. Because this second turbine can run at speeds of up to 65,000 rpm, gearing can present serious engineering challenges, since high turbine speeds must be matched to relatively low crankshaft speeds with the flexibility to match the instantaneous engine rpm needed for changing road speeds. Therefore, Saab Scandia uses a fluid coupling to provide both the speed and flexibility required. According to Saab Scandia, turbocompounding

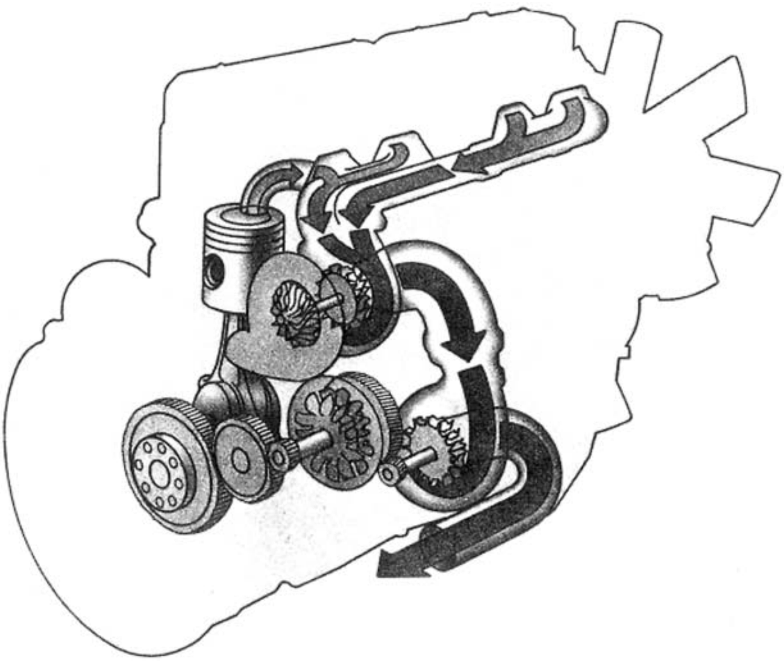


Figure 24 Turbocompound diesel engine. The exhaust gas first drives a normal turbocharger. Then the exhaust gases drive a second turbine, which is attached to a gear drive so it can rotate a fluid coupling. The fluid coupling transfers energy to the crankshaft through additional gearing. (Courtesy Saab Scandia.)

increases the efficiency of its already intercooled turbocharged diesel truck engine by an additional 2%, for a total efficiency of 46%.

Pressure-Wave Supercharging

Pressure-wave supercharging can combine the advantages of both a mechanically driven supercharger and an exhaust-driven turbocharger. Burghard conceived a pressure-wave machine in 1910, Brown Boveri of Switzerland began working on the concept as early as 1909, and by 1940/1942 Claude Seippel had invented the COMPREX design (Fig. 25). The COMPREX supercharger is a simple machine; the underlying thermodynamic processes involved are very complex. Therefore, while much development work was done and prototypes built, a successful pressure-wave supercharger for an automotive diesel engine had to wait for the

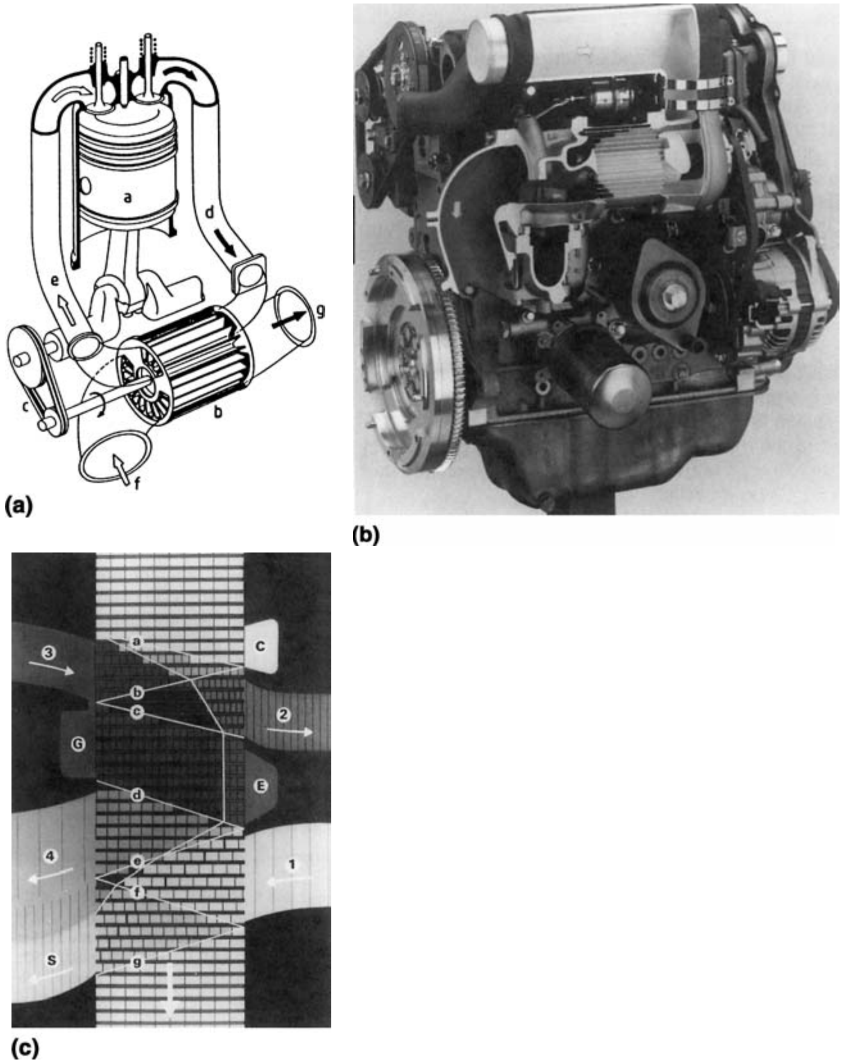


Figure 25 (a) Schematic of the COMPREX supercharger: a, combustion chamber; b, rotor with cells; c, belt drive to drive rotor; d, exhaust gas from combustion chamber; e, high-pressure air charge; f, ambient air pressure; g, exhaust gases delivered to exhaust system. (b) COMPREX installed on a small diesel engine. (c) Pressure-wave (a–g) process in the COMPREX supercharger: 1, suction air entering COMPREX rotor for next cycle; 2, compressed air charge to engine; 3, exhaust gases from engine acting on air charge; 4, exhaust gases discharged; S, Scavenging air; C, compression pocket; E, expansion pocket; G, gas pocket. (Courtesy COMPREX AG.)

advent of high-speed digital computers to perform the huge number of computations and complex modeling needed to optimize the design. COMPREX AG, a Swiss company, now builds the pressure-wave superchargers that have been on Japanese Mazda 626 Capella models since 1987.

Like a supercharger, the COMPREX is driven off the engine crankshaft at a speed proportional to engine speed. Typically, the COMPREX runs at about 30,000 rpm using a toothed belt drive system that increases the speed five to eight times. Unlike a normal supercharger, no energy is transferred from the crankshaft except to overcome friction and inertia. Only about 1% of the engine output is required to overcome friction and windage losses.

The COMPREX consists of an engine-driven rotor located between two end casings connected to intake and outlet ducts for the charge air and exhaust gases, respectively. The inlet and outlet ports for air are located at one end and the exhaust inlet and outlet ports are at the other end. The rotor has axial vanes that form passages called cells, which are opened at both ends. Ambient air enters the cells through the inlet port and is carried around by the vanes until it is opposite the exhaust gas port. At this point, the high-pressure exhaust gases rush into the chamber to compress the air charge, which is pushed toward the opposite end and forced through the high-pressure air duct in the air casing to the engine. During this process the exhaust gases are reflected back out of the exhaust port and the inlet charge is directed to the engine. The pressure wave created by the exit of the exhaust gas through the exhaust port allows a fresh inlet charge to enter the inlet port, and so the cycles continues. There is no contact between the casings and rotor faces, but the gap is very small to minimize leakage.

Before the exhaust gas penetrates the rotor to the extent that it would flow out with the compressed air, the rotor cells are brought into connection with the outlet duct in the gas casing by the rotation of the rotor. The exhaust gas flows out of the rotor, expands, and the suction produced by the powerful flow is able to draw in fresh air through the low-pressure duct of the air casing and into the rotor cells.

Because of the very short period that the inlet charge and exhaust gas are in contact without any physical barrier, there is no contamination of air by the hot exhaust and heat is not transferred. The concept takes advantage of the physical fact that when two gases with different pressures are brought into direct contact, equalization of pressure occurs faster than the mixing of the gases. This is especially true if the gases are guided in narrow channels or cells.

Since energy transfer is accomplished by pressure waves traveling at the speed of sound, the COMPREX provides instantaneous response. With this plus the fact that the exhaust gas does not have to drive any machinery, there is essentially no turbo lag.

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