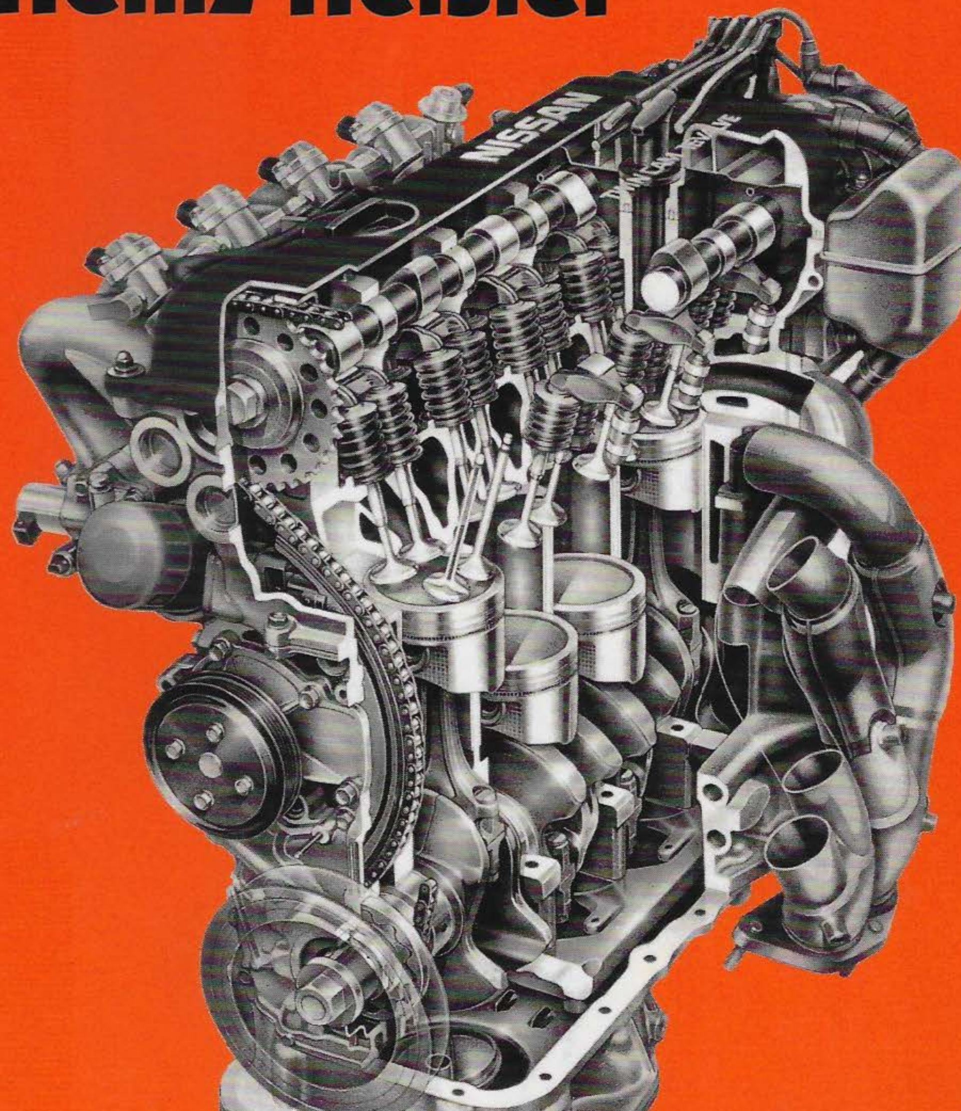
Advanced engine technology

Heinz Heisler



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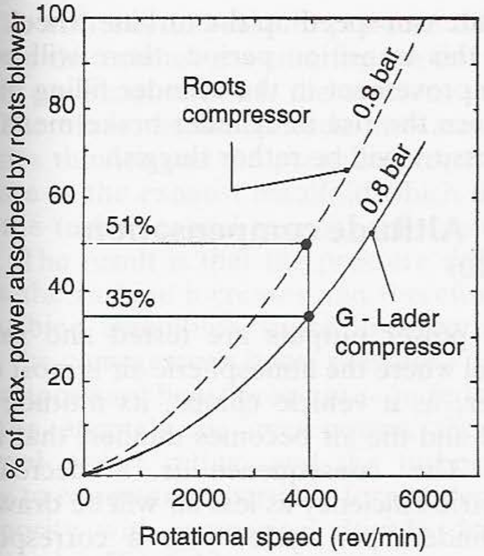


Fig. 6.37 Comparative power absorbed by Root's- and 'G'-Lader-type superchargers relative to rotational speed

mum boost pressure is limited to just under 0.8 bar.

The adiabatic compressor efficiency rises rapidly with an increase in boost pressure, reaching a peak of about 68%, and then gradually decreases (Fig. 6.24).

It is claimed that there is very little temperature rise during the compression process so that with the low amount of power absorbed by the compressor and the relatively cool delivery of the boost charge, it makes this type of supercharger very efficient and extremely suitable for boosting small- to medium-sized car petrol engines.

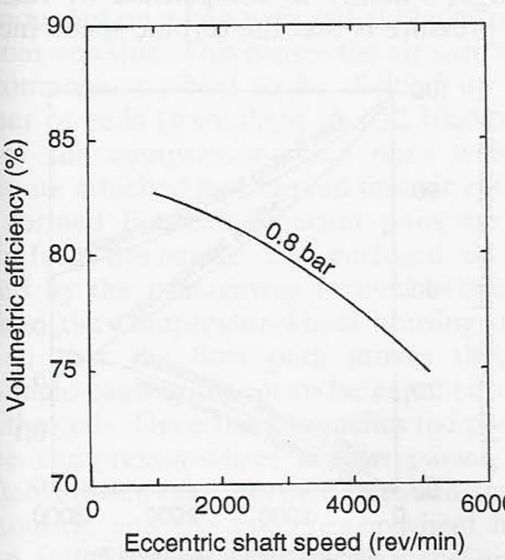


Fig. 6.38 Relationship between volumetric efficiency and eccentric shaft speed at constant boost pressure for a 'G'-Lader supercharger

6.7 Turbochargers

6.7.1 Introduction

A typical petrol engine may harness up to 30% of the energy contained in the fuel supplied to do useful work under optimum conditions but the reminaing 70% of this energy is lost in the following way:

7%	heat energy to friction, pumping and dynamic movement
9%	heat energy to surrounding air
16%	heat energy to engine's coolant system
38%	heat energy to outgoing exhaust gases

Thus, the vast majority of energy, for design reasons, is allowed to escape to the atmosphere through the exhaust system.

A turbocharger utilizes a portion of the energy contained in the exhaust gas—when it is released by the opening of the exhaust valve towards the end of the power stroke (something like 50° before BDC)—to drive a turbine wheel which simultaneously propels a centrifugal compressor wheel.

The turbocharger relies solely on extracting up to a third of the wasted energy passing out from the engine's cylinders to impart power to the turbine wheel and compressor wheel assembly. However, this does produce a penalty by increasing the manifold's back-pressure and so making it more difficult for each successive burnt charge to be expelled from the cylinders. It therefore impedes the clearing process in the cylinders during the exhaust strokes.

The ideal available energy which can be used to drive the turbocharger comes from the blow-down energy transfer which takes place when the exhaust valve opens and the gas expands down to atmospheric pressure (Fig. 6.39). This blow-down energy is represented by the loop area 4, 5 and 6 whereas the boost pressure energy used to fill the cylinder is represented by the rectangular area 0, 1, 6 and 7.

Turbocharged engines produce higher cylinder volumetric efficiencies compared with the normally aspirated induction systems. Therefore, there will be higher peak cylinder pressures which increase the mechanical loading of the engine components and could cause detonation in petrol engines. Therefore, it is usual to reduce the engine's compression ratio by a factor of one or

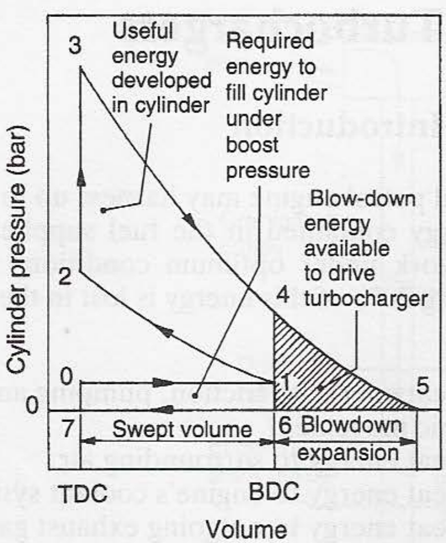


Fig. 6.39 Petrol engine cycle pressure–volume diagram showing available exhaust gas energy

two. Thus, a compression ratio of 10:1 for a normally aspirated engine would be derated to 9:1 for a low boost pressure or even reduced to 8:1 if a medium to high boost pressure is to be introduced. Similarly, for a direct injection diesel engine which might normally have a compression ratio of 16:1, when lightly turbocharged the compression ratio may be lowered to 15:1 and, if much higher supercharged inlet pressures are to be used, the compression ratio may have to be brought down to something like 14:1.

The compression of the charge entering the cells of the impellor depends upon the centrifugal force effect which increases with the square of the rotational speed of the impellor wheel. Consequently, under light load and low engine speed conditions the energy released with the exhaust gases will be relatively small and is therefore insufficient to drive the turbine assembly at very high speeds. Correspondingly, there will be very little extra boost pressure to make any marked improvement to the engine's torque and power output in the low-speed range of the engine. Thus, in effect, the turbocharged engine will operate with almost no boost pressure and with a reduced compression ratio compared with the equivalent naturally aspirated engine. Hence, in the very low speed range, the turbocharged engine may have torque and power outputs and fuel consumption values which are inferior to the unsupercharged engine.

Another inherent undesirable characteristic of turbochargers is that when the engine is suddenly accelerated there will be a small time delay before the extra energy discharged into the turbine housing volute can speed up the turbine wheel. Thus, during this transition period, there will be very little improvement in the cylinder filling process, and hence the rise in cylinder brake mean effective pressure will be rather sluggish.

6.7.2 Altitude compensation

(Fig. 6.40)

Engine power outputs are tested and rated at sea-level where the atmospheric air is most dense; however, as a vehicle climbs, its altitude is increased and the air becomes thinner, that is, less dense. The consequence is a decrease in volumetric efficiency as less air will be drawn into the cylinders per cycle, with a corresponding reduction in engine power since power is directly related to the actual mass of charge burnt in the cylinder's every power stroke. A naturally aspirated engine will have its power output reduced by approximately 13% if it is operated approximately 1000 m above sea-level. Supercharging the cylinders enables the engine's rated power above sea-level to be maintained or even exceeded.

With a turbocharged engine there will still be some power loss with the engine operating at high altitudes, but the loss will be far less than if the engine breathing depended only on natural aspiration. As can be seen (Fig. 6.40) at 1000 m the power loss is only 8% compared with the naturally aspirated engine where the power decrease is roughly 13%. The reason for the turbocharger's ability to compensate by raising its boost pressure is that the turbine speed increases

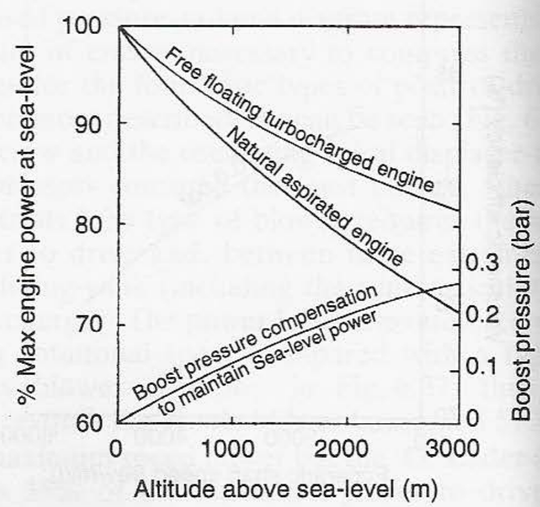


Fig. 6.40 Effect of altitude on rated engine power at sea level for both naturally-aspirated and turbocharged engines

directly with any increase in pressure difference between the exhaust gas entering the turbine and its exit pressure, which is the ambient air pressure. Therefore, as the altitude increases the air becomes thinner and the pressure drops, but the pressure in the exhaust manifold which impinges onto the turbine wheel remains substantially the same. The result is that the pressure differential across the turbine increases and therefore raises the turbine assembly's speed and, correspondingly, the compressor's boost pressure.

The necessary boost pressure required to maintain the engine's sea-level power, the loss of sea-level power rating, and the turbocharger's ability to compensate partially for the decrease in air density with increasing altitude driving, is compared in Fig. 6.40.

6.7.3 Description of turbocharger

(Fig. 6.41)

A turbocharger comprises three major components, an exhaust-gas driven turbine and housing, a centrifugal compressor wheel and housing and an interconnecting support spindle mounted on a pair of fully floating plain bearings, which are themselves encased in a central bearing housing made from nickel cast iron (Fig. 6.41).

Compressor

(Figs 6.41 and 6.42)

The impellor compressor wheel (Fig. 6.42) is an aluminium alloy casting which takes the form of a disc mounted on a hub with radial blades projecting from one side. This causes the air surrounding the compressor wheel to be divided up into a number of cells (something like 12 blades). The hub of the compressor-wheel onto which the blades are attached is so curved that air enters the cells formed between adjacent pairs of blades axially from the centre. The enclosed air is then divided by the passageway formed between the hub and the compressor-wheel housing internal wall so that the flow path moves through a right-angle causing the air to be expelled radially from the cells. Once the air reaches the periphery of the compressor-wheel it then passes to the parallel diffuser (gap) formed between the bearing housing and the compressor-wheel housing (Fig. 6.41). From the diffuser gap the air flows into a circular volute-shaped collector, which is a constant expansion passage from some starting point to its exit.

Turbine

(Figs 6.41 and 6.42)

The exhaust gas temperature at the inlet to the turbine wheel, under light-load to full-load high-speed operating conditions, may range between 600°C to 900°C. Consequently, the turbine is usually made from a high-temperature heat-resistant nickel-based alloy such as 'Inconel'. Ceramic materials such as sintered silicon nitride are being developed as an alternative to nickel based alloys, and these materials have the advantage of weighing less than half that of suitable metallic materials. These ceramic materials have a higher specific strength and a much lower coefficient of expansion than their counterparts, they are also capable of operating under the same full-load exhaust gas temperature conditions at similar maximum rotational speeds as the nickel-based alloys.

The turbine wheel (Fig. 6.42) takes the form of a hub supporting a disc at one end and a number of radial blades which project axially and radially from both the hub and disc respectively. The outer edges of the blades are curved backwards to trap the impinging exhaust gases.

Exhaust gases from the manifold enter the spherical graphite cast-iron turbine housing flange entrance, the gases then flow around either a single or twin volute passageway surrounding the turbine wheel (Fig. 6.41). The gases are then forced tangentially inwards from the throat of the turbine housing so that they impinge onto the blade faces. The flow path then directs the gases gradually through a right-angle so that they come out axially from the centre of the turbine hub, and they are then expelled into the exhaust pipe system.

In most turbocharger designs the turbine and spindle are joined together by some sort of welding process such as inert gas welding, resistance welding, or electron beam welding. The general trend these days is to attach the turbine to the spindle by the friction welding solid phase technique (Fig. 6.42). The turbine and spindle are brought together under load, with one part revolving against the other so that frictional heat is generated at the interface. When the joint area is sufficiently plastic as a result of the increase in temperature the rotation is stopped and the end force increased to forge and consolidate the metallic bonds. The surface films and inclusions that might interfere with the formation of these bonds are broken up by friction and removed from the weld area in a radial direction owing to

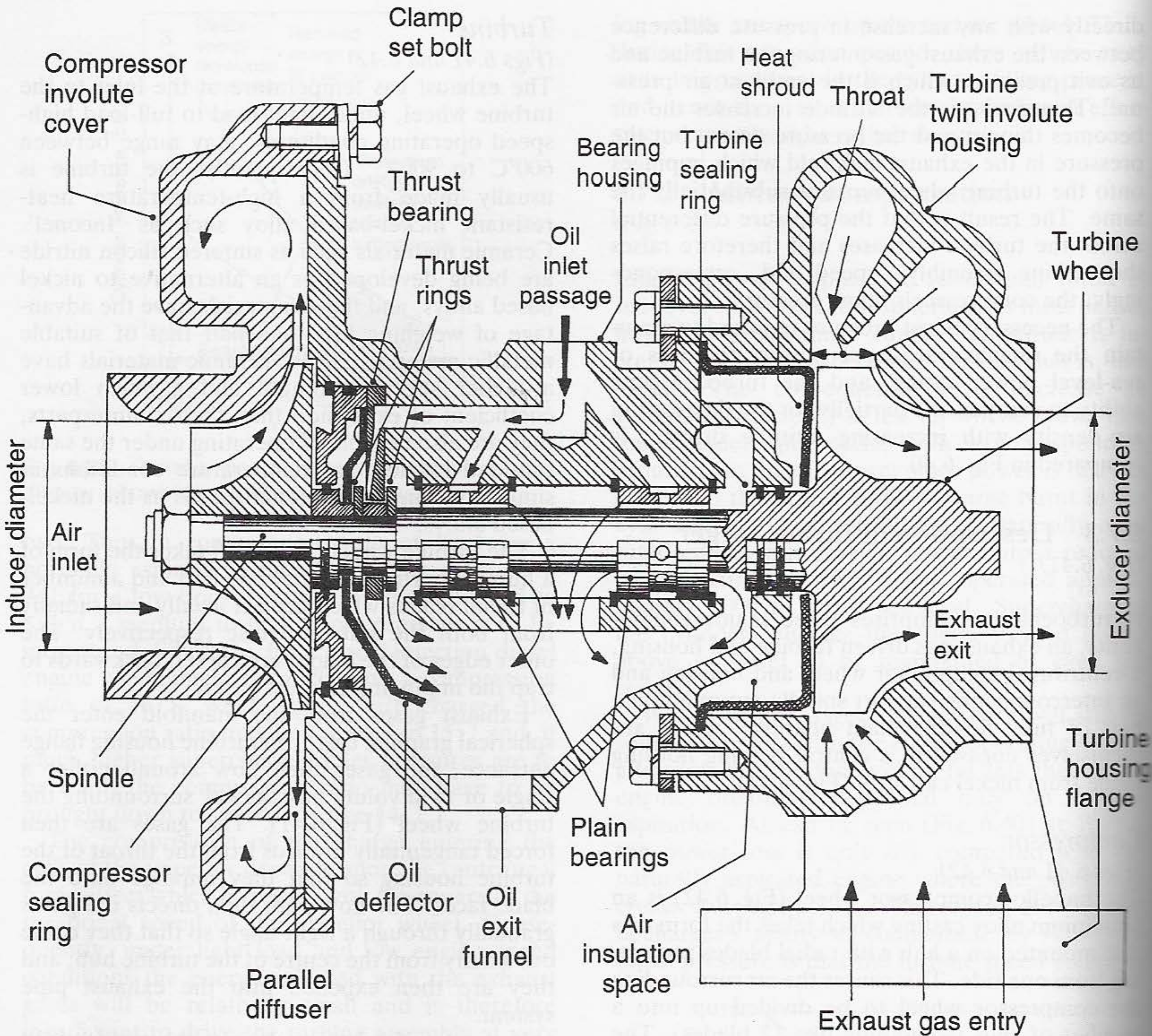


Fig. 6.41 Turbocharger construction

marked plastic deformation on the surfaces. The burrs surrounding the joint are then ground away leaving a high-quality joint.

Spindle and bearing assembly
(Figs 6.41 and 6.42)
The turbine wheel and steel spindle can be welded together in a vacuum by an electron beam. This joining process produces a neat joint in which the heat flow areas are held to a minimum so that there is very little distortion and machining is kept to the minimum. The medium carbon steel spindle is induction hardened where the bearings contact the spindle. The hollow space between the turbine wheel and the spindle (Fig. 6.42 sectioned-view) prevents heat being transferred through the centre of the spindle. Thus, heat is carried away along the outer section of the spindle, which can readily be cooled by the lubricating oil. The spindle is supported on a pair of free-floating phosphorus bronze plain bearings and around the outside of each bearing shell are six radial holes and, depending upon design, there may be a circumferential groove machined to distribute the lubrication oil. The rotation of the spindle assembly in conjunction with the oil supply from the engine's lubrication system

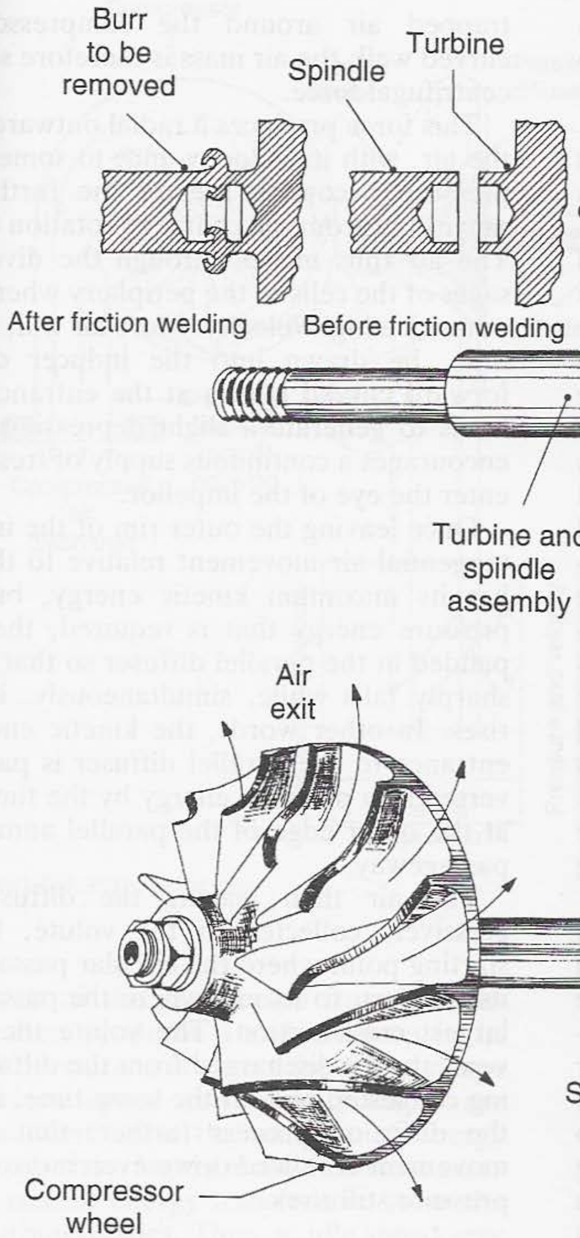


Fig. 6.42 Compressor and turbine wheel assembly

causes the viscous drag to rotate these bearings at approximately one-third of the spindle's rotational speed.

The large-diameter shouldered section next to the turbine-wheel is grooved to house two piston rings and a similar single piston ring is positioned on a grooved collar at the opposite end of the spindle next to the compressor wheel (Fig. 6.41). These piston-type rings remain stationary when the spindle rotates, their function being to prevent exhaust gas or compressed air entering the bearing housing chamber. Oil should be prevented from reaching the piston rings as any film, foam or splash entering the seal region will leak out. The spindle bearing at the turbine wheel end is separated from the large diameter shouldered section of the spindle by a cast-in-oil drain slot cavity so that oil, spurting from the end of the bearing, spills into this cavity and then drains down to the oil exit funnel. In contrast, at the compressor wheel end, an oil deflector, fixed to the thrust bearing, protects the piston ring seal from contamination by shielding the bearing oil spray from the piston rings, the oil is then permitted to drain down to the bearing housing exit.

During operation, the exhaust gas impinging onto the turbine-wheel, and the compressed air reaction on the compressor wheel under varying speed and load conditions, produces a certain amount of end thrust as the spindle will want to move axially first in one direction and then the other.

Provision for end float or thrust control is provided by a thrust spacer (Fig. 6.41) which has a deep central groove. This thrust spacer is sandwiched between the stepped spindle shoulder and the piston-ring spacer. A phosphorus bronze thrust bearing plate, which is attached to the bearing housing, slips into the central groove. Consequently, the walls on each side of the thrust-plate become thrust-rings, whilst the reduced diameter portion between them forms a spacer sleeve, its width being critical to control any end movement of the compressor-wheel and the turbine wheel.

To minimize the heat transference from the turbine-wheel exhaust gas flow path to the bearing housing, a space is created between the turbine-wheel and the bearing housing. This air space is enclosed by a stainless-steel heat shroud pressing, shaped in the form of a cup, which is located immediately behind the turbine-wheel. This relatively large air gap provides an effective heat barrier, and therefore insulates the bearing housing from the hot turbine assembly.

Both radial plain bearings and the axial end thrust bearing are supplied with ample oil from the engine's lubrication system via drillings made in the bearing housing and in the bearings themselves (Fig. 6.41). The oil supply has two major functions: firstly, to lubricate the bearings so that a hydrodynamic oil film can be established so that, in effect, the shaft and bearings are floating on oil; and secondly, to remove excess heat from the bearing assembly. Thus, it is just as important to be able to return the circulating oil to the engine's sump as it is to flood the bearings in the first place with lubricant.

In some turbochargers, the bearing housing incorporates a liquid coolant jacket through which coolant from the engine's cooling system is made to circulate via a pair of flexible inlet and outlet pipes.

6.7.4 The operating principles of compressor and turbine

The operating principle of the compressor (Fig. 6.43)

With the spindle assembly rotating, the air cells formed between adjacent blades sweep the entrapped air around the compressor housing curved wall: the air mass is therefore subjected to centrifugal force.

This force produces a radial outward motion to the air, with its velocity and, to some extent, its pressure becoming greater the further the air moves out from the centre of rotation (Fig. 6.43). The air thus moves through the diverging passages of the cells to the periphery where it is flung out with a high velocity. More air will, at the same time, be drawn into the inducer due to the forward curved blades at the entrance, and this tends to generate a slight depression. Hence, it encourages a continuous supply of fresh charge to enter the eye of the impellor.

Once leaving the outer rim of the impellor the tangential air movement relative to the impellor has its maximum kinetic energy, but, as it is pressure energy that is required, the air is expanded in the parallel diffuser so that its velocity sharply falls while, simultaneously, its pressure rises. In other words, the kinetic energy at the entrance to the parallel diffuser is partially converted into pressure energy by the time it arrives at the outer edge of the parallel annular-shaped passageway.

The air then leaving the diffuser is progressively collected in the volute, from some starting point where the circular passageway is at its smallest, to its exit where the passage is at its largest cross-section. The volute therefore prevents the air discharged from the diffuser becoming congested and, at the same time, it continues the diffusion process further; that is, the air movement is slowed down even more whereas its pressure still rises.

The operating principle of the turbine (Fig. 6.43)

Exhaust gas from the engine's cylinders is expelled via the exhaust manifold into the turbine volute circular decreasing cross-section passageway, at a very high velocity, where it is directed tangentially inwards through the throat of the turbine housing. The released gas kinetic energy impinges on the turbine-blades, thereby imparting energy to the turbine-wheel as it passes through the cells formed between adjacent blades with a corresponding decrease in both gas velocity and pressure (Fig. 6.43). The exhaust gas with its rapidly decreasing energy moves radially inwards and, at the same time, its flow path moves through a right-angle so that it passes axially along the hub before leaving the turbine housing.

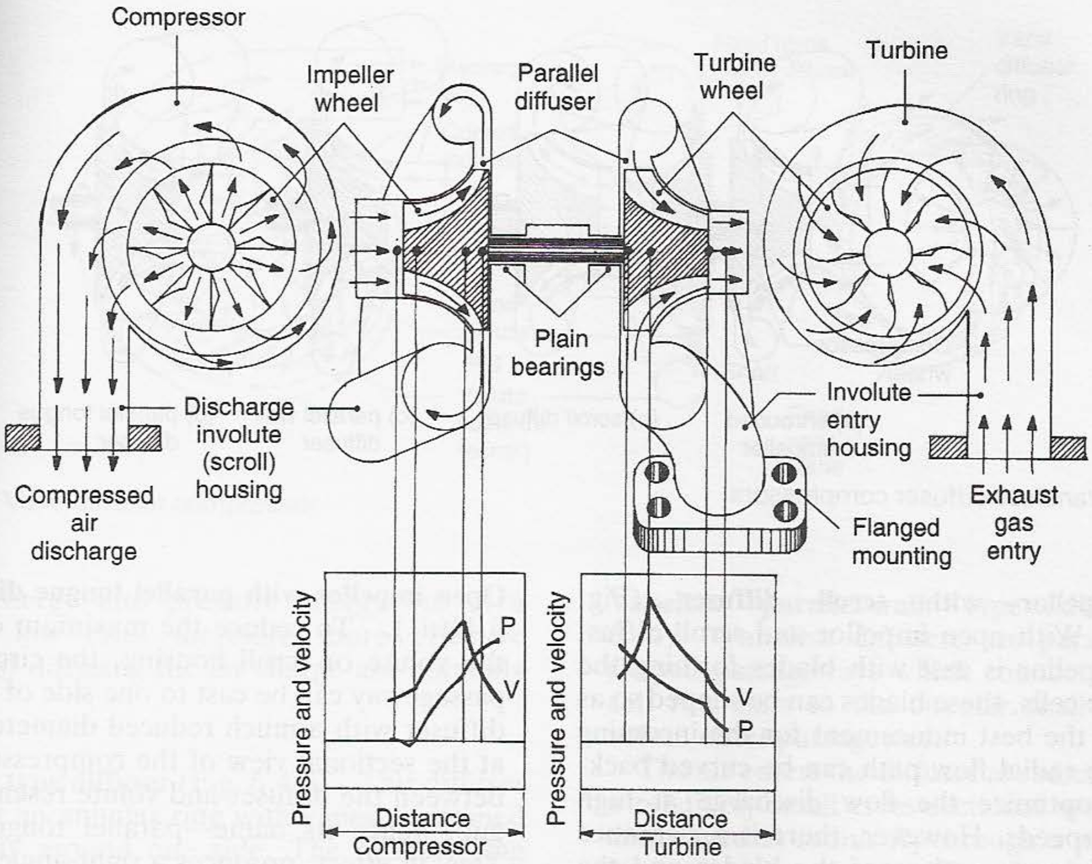


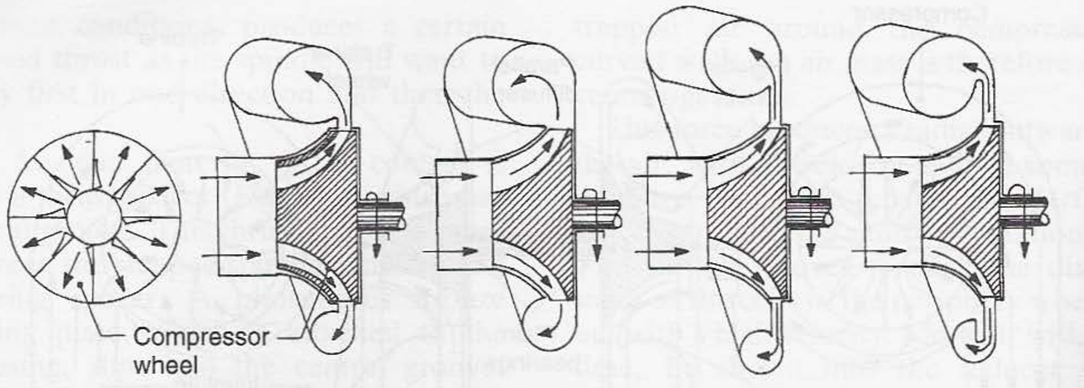
Fig. 6.43 Turbocharger principle

The expansion of the gas ejected from the turbine-wheel then produces a sudden drop in its velocity and pressure as it enters the silencer pipe system. Turbine speed and boost pressure are largely dependent upon the amount of energy contained in the hot, highly mobile exhaust gases and on the rate of energy transference from the gas to the turbine-blades. Thus, at idle speed very little fuel is supplied to the engine, and therefore the energy content in the outgoing exhaust gas will also be very low whereas, with increased engine speed and load conditions, considerably more fuel is consumed by the engine, which in turn releases proportionally more energy to the escaping exhaust gases. Hence, at light load and low speed, the turbine assembly speed can be something like 30000 to 50000 rev/min, whereas at high speed and high load operating conditions the spindle and wheel assembly can revolve at speeds up to 120000 to 150000 rev/min, depending upon design and application.

6.7.5 Compressor impellor and housing design

Compressor and housing arrangements

Closed or shrouded impellor with scroll diffuser (Fig. 6.44(a)). The impellor may be closed or shrouded; that is, the impellor is cast so that the cells or channels are completely enclosed. This construction eliminates direct leakage as the induced air is flung radially outwards in the cells. However, it is difficult to cast radial cells so that they curve backwards and also provide an axial angled inlet at the eye of the impellor. Other important disadvantages which must be considered are that the mass of the shroud is supported by the blades, such that at high rotational speeds the blades are subjected to severe centrifugal stresses. In addition, the shroud which is away from the central hub raises the impellor wheel's moment of inertia and thus impedes its ability to accelerate or decelerate rapidly. This design has been succeeded by the open-cell type impellor.



(a) shrouded impeller (b) scroll diffuser (c) parallel wall diffuser (d) parallel tongue diffuser

Fig. 6.44 Vaneless diffuser compressors

Open impellor with scroll diffuser (Fig. 6.44(b)). With open impellor and scroll diffuser, the impellor is cast with blades forming the walls of the cells, these blades can be shaped so as to provide the best inducement for the incoming air and the radial flow path can be curved backwards to optimize the flow discharge at high rotational speeds. However, there is a clearance between the outer edges of the blades and the internal curved walls of the housing which encloses the rotating cells. This gap, small as it is, will be responsible for leakage losses under high boost pressure operating conditions. With this arrangement, the kinetic energy of the air at the blade tips is converted into pressure energy by directly entering into the relatively large scroll volume. In other words, the air flung out at the rim of the impellor enters the scroll where it is diffused by the relatively large mass of air already occupying the circular passageway. Thus, the intermingling of the discharged air causes its velocity to decrease and its pressure to increase, which goes someway towards producing the desired flow conditions for the charge.

Open impellor with parallel wall diffuser (Fig. 6.44(c)). If a more positive method of converting the kinetic energy to pressure energy is required, a parallel annular space between the impellor and the volute or scroll will enlarge the circular passage from the entrance at the impellor rim to where it merges with the discharge volute. Thus, as the air moves outwards in a semi-spiral and radial direction the air will expand, thus causing its speed to reduce while its pressure rises.

Open impellor with parallel tongue diffuser (Fig. 6.44(d)). To reduce the maximum diameter of the volute or scroll housing, the circular volute passageway can be cast to one side of the parallel diffuser with a much reduced diameter. Looking at the sectional view of the compressor, the wall between the diffuser and volute resembles a tongue; hence its name—parallel tongue diffuser. This, in effect, produces a right-angled flow-path similar to the normal parallel wall diffuser design but the overall dimensions of the housing will be more compact.

Compressor diffusers

(Figs 6.45 and 6.46)

The object of a compressor diffuser is to convert the air's kinetic energy produced at the rim of the impellor, to pressure energy by expanding it so that its velocity falls, thereby causing its pressure to rise.

The are three types of diffusers:

a) the scroll diffuser
b) the vane ring diffuser
c) the vaneless parallel wall diffuser

The scroll-type diffuser (Fig. 6.44(b)). The scroll or volute is the circular passageway of varying cross-sectional area which directly surrounds the impellor rim. Thus, when the air is flowing through the impellor radial cells towards the periphery of the blades it will have attained a very high velocity. The air will then be flung in a spiral flow path directly into the scroll, and the much larger circular passageway then enables the air so that it reduces speed and, at the same time, raises pressure. This form of diffuser ...

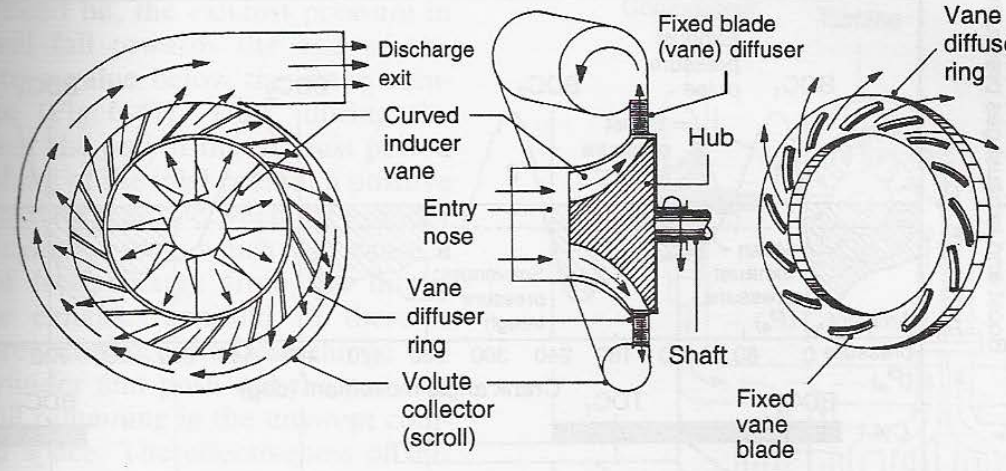


Fig. 6.45 Vane-diffuser compressor

velocity energy into pressure energy but at a relatively low rate, therefore more effective methods of diffusing the air charge are normally selected.

Vane-ring type diffuser (Fig. 6.45). This diffuser consists of an annular ring with vanes positioned tangentially around one side. The diffuser ring with its diverging multi-passageways joins the impellor cell periphery outlets to the circular, variable cross-section, volute collector. The vanes are so positioned that they guide the air discharge from the impellor rim to the volute in a tangential direction through passages of increasing cross-section. Thus, since the energy contained by the air cannot be destroyed, the effect of the expanding passages will be to slow down the air movement; therefore, if energy is to be retained the air pressure will rise.

Vaneless parallel-wall type diffuser (Fig. 6.46). Vaneless diffusers are parallel annular-shaped passageways which connect the impellor cell rim exits to the circular snail-shell shaped volute outer passageway.

The air enters the diffuser at radius R_1, through a relatively small cross-sectional area A_1, and is discharged at radius R_2 through a proportionally larger cross-section A_2. Thus, by similar triangles

$$\frac{A_1}{R_1} = \frac{A_2}{R_2}$$

therefore

$$A_1 = A_2 \frac{R_1}{R_2}$$

Hence, if R_1 is half that of R_2, then A_1 will be half the cross-sectional area of A_2 and vice versa.

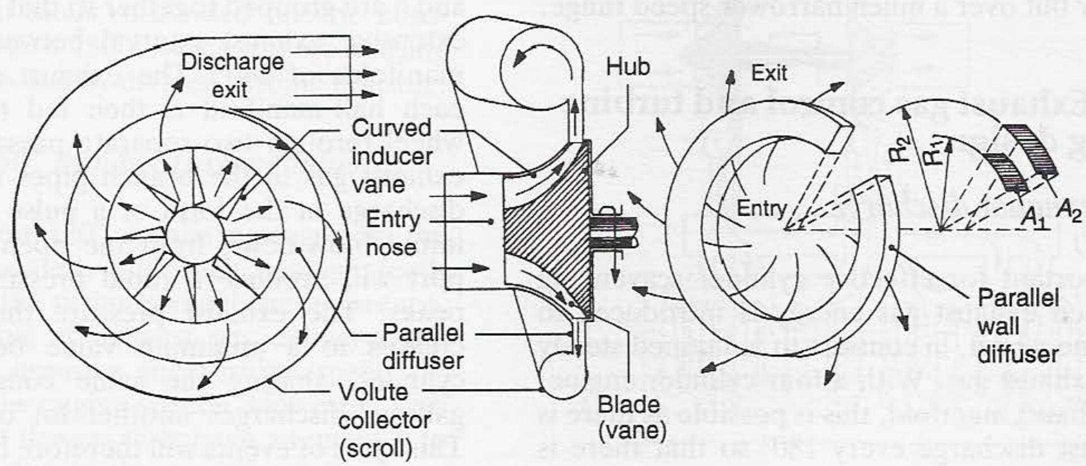


Fig. 6.46 Vaneless diffuser compressor with parallel wall diffuser illustrating expansion of flow area from inlet to exit

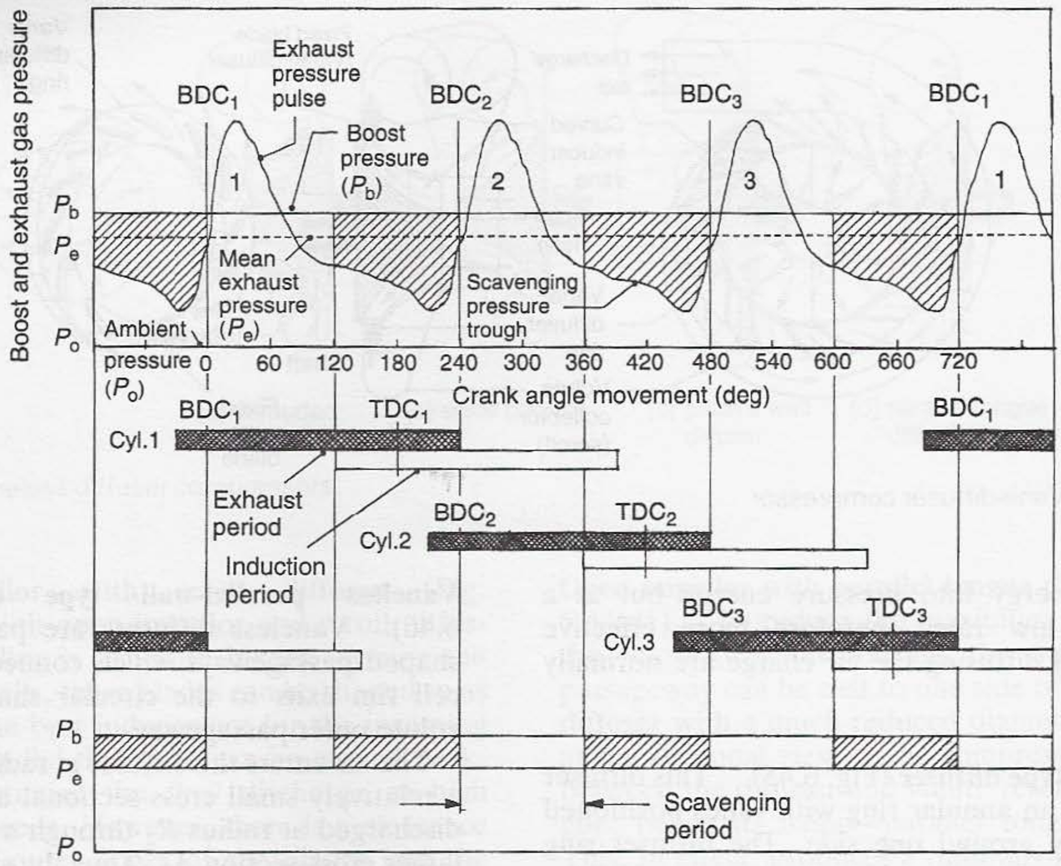


Fig. 6.47 Exhaust gas pressure variation in activated six-cylinder turbocharged engine manifold

Accordingly, the air leaving the impellor and passing through the parallel diffuser will reduce its speed in proportion to the increase in the annular passageway cross-sectional area. In contrast, the air discharge pressure rises.

Parallel-wall diffusers have a broad operating speed range over which a moderate compressor efficiency is maintained, whereas the vane ring diffuser operates the compressor at a fairly high efficiency but over a much narrower speed range.

6.7.6 Exhaust gas control and turbine housing design

Pulsed exhaust discharge
(Fig. 6.47)

It is important for effective cylinder scavenging that pulsed exhaust gas energy is introduced to the turbine wheel, in contrast to a damped steady flow of exhaust gas. With a four-cylinder engine, single-exhaust manifold, this is possible as there is an exhaust discharge every 180° so that there is very little exhaust gas interference between cylinders.

However, with more than four cylinders, exhaust gas will discharge at shorter intervals than the 180°; that is, for 5, 6 and 8-cylinder engines the intervals between exhaust discharges will be 144°, 120° and 90° respectively. To overcome exhaust gas interference in the manifold, manifolds are sub-divided so that, in the case of an in-line six-cylinder engine, cylinders 1, 2 and 3 are grouped together and, similarly, cylinders 4, 5 and 6 are grouped together so that there is now an extensive exhaust interval between sub-divided manifolds of 240°. The exhaust discharge from each half-manifold is then fed to the turbine-wheel through two separate passageways. If the exhaust gas in the branch pipes is permitted to discharge in the form of a pulse (Fig. 6.47) the initial blow-down from the open exhaust valve port will produce a rapid pressure rise until it peaks. The exhaust pressure then quickly decreases to a minimum value before the next cylinder, sharing the same common manifold gallery, discharges another lot of exhaust gas. This cycle of events will therefore be continuously repeating. By reducing or even eliminating inter-cylinder exhaust gas interference, by sub-dividing the manifold if need be, the exhaust pressure in the manifold will fall towards the end of the exhaust stroke to a value below the mean compressor pressure (Fig. 6.47). Thus, during the valve overlap near the end of the exhaust period and at the beginning of the inlet period, a positive pressure difference will exist between the cylinder intake and the cylinder exit, which will cause a blow-through of fresh charge from the intake manifold to the exhaust manifold. If there is sufficient pressure difference the fresh charge will rush into the cylinder and push out the residual exhaust gases still remaining in the unswept combustion chamber space. The effectiveness of this scavenging action will also depend on the engine speed and the actual valve opening area during the time of valve overlap.

Divided turbine housing passageways

The turbine housing for a four-cylinder engine normally has a single volute circular passageway into which the four merged branch pipes discharge their individual exhaust gas pulses at intervals of 180° through a 360° throat entrance to the turbine wheel.

When there are more than four cylinders, better turbine response is obtained by dividing the exhaust manifold into two halves. The discharge from each half manifold is then fed to the turbine wheel through two separate passageways formed in the turbine housing. Alternating exhaust gas pulses from each group of branches discharge at relatively prolonged intervals through the throat of the turbine housing in a pulsed jet stream against the turbine-blades. This pulse impingement of the exhaust gas from individual branch pipes is more effective than the resultant steady discharge from several cylinders.

There are two kinds of divided turbine housings, the double-flow 180° divided turbine housing and the twin-flow axially divided turbine housing.

Double flow 180° divided turbine housing
(Fig. 6.48)

In the double-flow 180° circular passageway, the throats are separated, and each passage feeds only one half of the turbine-wheel circumference. With this method of exhaust gas delivery, the pulsed gas will improve the turbine speed response at very low engine speeds. However, with this arrangement the gas does have a tendency to reverse its flow due to the centrifugal force exceeding the inward pulse thrust when there are

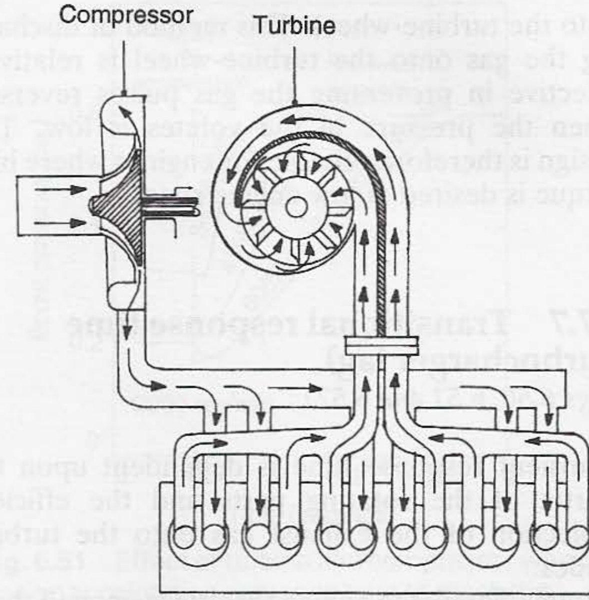


Fig. 6.48 Double-flow 180° divided turbo-housing

only low pressure pulses in the housing passageways.

Twin flow axially divided turbine housing
(Fig. 6.49)

In the twin-flow axially divided turbine housing each of the two volute passages commences feeding into a common throat which completely surrounds the circumference of the turbine wheel. With this configuration, exhaust gas is pulsed inwards alternating from each volute passage through the 360° throat which discharges directly

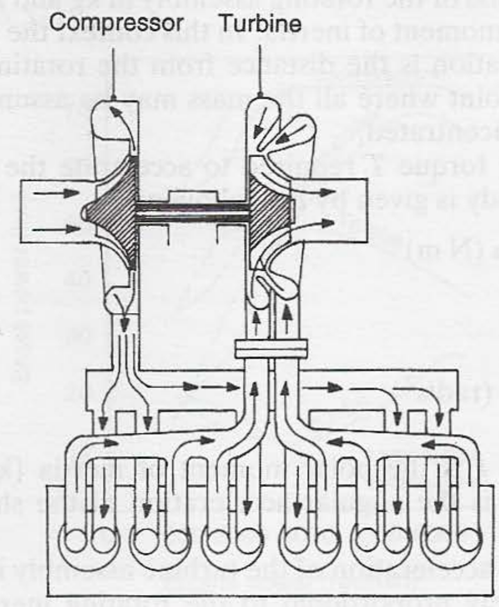


Fig. 6.49 Twin-flow (twin-scroll) axially divided turbine housing

onto the turbine-wheel. This method of discharging the gas onto the turbine-wheel is relatively effective in preventing the gas pulses reversing when the pressure in the volutes is low. This design is therefore suitable for engines where high torque is desired at low engine speeds.

6.7.7 Transitional response time (turbocharger lag)

(Figs 6.50, 6.51 and 6.52)

Transient response time is dependent upon the inertia of the rotating parts and the efficient projection of the exhaust gas onto the turbine blades.

Immediately the engine throttle is opened there will be an increased flow of mixture entering the cylinders with a corresponding exit of exhaust gas, which is directed onto the turbine-blades causing the wheel assembly to accelerate rapidly.

The time taken for the turbine and compressor assembly to attain the maximum operating speed is dominated by the overall efficiency of the turbocharger and the polar moment of inertia of the rotating assembly.

The polar moment of inertia I is the reluctance of the rotating body to change speed, which may be represented by

$$I = mk^2 \text{ (kg m}^2\text{)} \qquad (5)$$

where k is the radius of gyration in metres, m is the mass of the rotating assembly in kg and I is the polar moment of inertia. In this context the radius of gyration is the distance from the rotating axis to a point where all the mass may be assumed to be concentrated.

The torque T required to accelerate the rotating body is given by the following

$$T = I\alpha \text{ (N m)} \qquad (6)$$

thus

$$\alpha = \frac{T}{I} \text{ (rad/s}^2\text{)} \qquad (7)$$

where I is the polar moment of inertia (kg m^2) and α is the angular acceleration of the shaft in rad/s^2.

The acceleration of the turbine assembly is thus inversely proportional to the rotating inertia so that halving the polar moment of inertia will double the acceleration and vice versa.

Example

Compare the moment of inertia for two different-sized turbine-wheels from the following data.

a) For a large turbine—diameter 75 mm, mass 1.28 kg and radius of gyration of 20 mm.
b) For a small turbine—diameter 50 mm, mass 1.0 kg and radius of gyration of 16 mm.

a) $I_{75} = mk^2 = 1.28(20\times10^{-3})^2$
$= 1.28\times400\times10^{-6}$
$= 512\times10^{-6}$ (kg m^2)

b) $I_{50} = mk^2 = 1.0(16\times10^{-3})^2$
$= 1.0(16\times10^{-3})^2$
$= 1.0\times256\times10^{-6}$
$= 256\times10^{6}$ kgm^2

Relative polar moment of inertia =

$$= \frac{512\times10^{-6}}{256\times10^{-6}} = 2:1$$

This comparison illustrates that only a small reduction in turbine diameter considerably reduces the polar moment of inertia and, in this example, reducing the turbine diameter by one third, halves the polar moment of inertia for the turbine. Thus, using two small turbochargers instead of one large unit for a V-cylinder banked engine goes someway towards reducing transient response time (turbocharger lag).

Turbocharger lag will depend to some extent upon the excess torque available from the turbine-wheel over that required to drive the compressor with the air flow and boost pressure existing at that instant. Therefore, a small turbine volute housing attached directly to a short and small diameter passageway (minimum volume) manifold is desirable as this will provide an undamped exhaust gas pulse directly and effectively to the turbine blades, thereby producing the least time lag.

The rotational speed characteristics of the turbine and compressor assembly relative to the engine speed for wide-open throttle operation and for normal road load driving conditions are shown in Fig. 6.50. It can be seen that with wide-open throttle (maximum acceleration) the extra exhaust gas energy released to the turbine housing produces a much earlier rise in turbo speed than for road load throttle operating conditions where the throttle opening will be, say, between a third and two-thirds less. Note the kink at about 3000 rev/min on the wide-open throttle curve and then the reduced rate of turbo speed increase as the wastegate begins to open and bypasses a portion of the exhaust gas from the

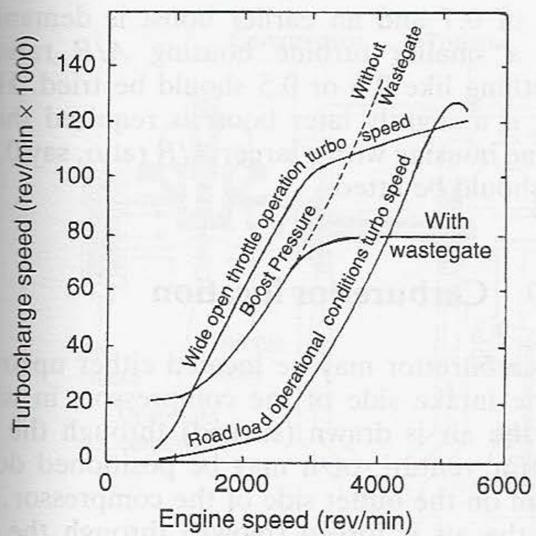


Fig. 6.50 Effect of engine speed on turbocharger spin speed and boost-pressure for wide-open throttle and road load operating conditions

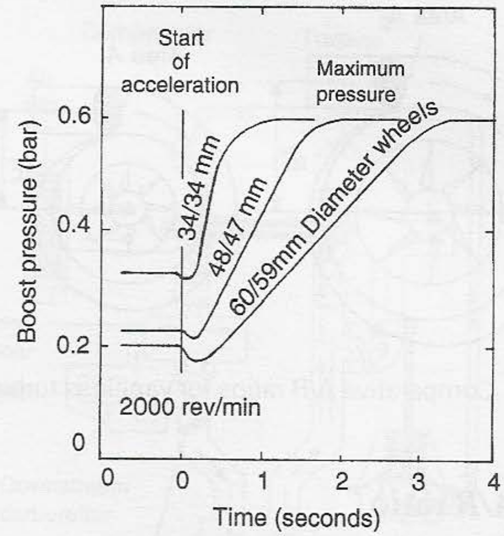


Fig. 6.51 Effect of turbine and compressor wheel size on acceleration response time to reach the designed maximum boost pressure

turbine blades. In the case of the road load throttle opening curve, the turbo speed does not commence to rise until the engine's speed has increased to about 2500 rev/min, it then rises continuously and only bends over as the engine speed approaches its maximum, indicating that the wastegate only is then required to open.

The maximum turbocharger rotational speeds obtainable are related to the turbine and compressor wheel sizes—small diameter wheels are able and do operate effectively at much higher rotational speeds than do large diameter wheels.

Likewise, small turbine and compressor wheel combinations are capable of much higher acceleration rates due to their low inertia compared with larger wheels. Consequently, small-diameter wheels can reach their maximum boost pressure before the wastegate opens much earlier than larger wheels. These facts are demonstrated in Fig. 6.51. Here, three different-sized turbochargers have been tested from a starting engine speed of 2000 rev/min and their boost pressure-rise against accelerating time has been recorded. As can be seen, the large turbocharger takes between 3.0–3.5 seconds to reach full boost pressure from 2000 rev/min, the medium-sized turbo takes only 1.5–2.0 seconds, whereas the small turbo only needs 0.5–1.0 seconds to attain its maximum boost pressure. These large, medium and small turbine and compressor wheel diameters have maximum rotational speeds respectively as follows: 60/59 mm–150 000 rev/min, 48/47–180 000 rev/min, and 34/34–270 000 rev/min.

However, the reduced transitional response time with the smaller-sized wheels is partly offset by the reduction in efficiency caused by the proportional increase of leakage past the wheel as its diameter decreases, and also due to the increase in exhaust gas back pressure imposed by the reduced flow path created between the turbine blades.

Improvements in low-speed boost pressure by using a smaller and faster turbocharger can raise the torque and power developed in the engine's lower speed range and, at the same time, can reduce the specific fuel consumption (see Fig. 6.52).

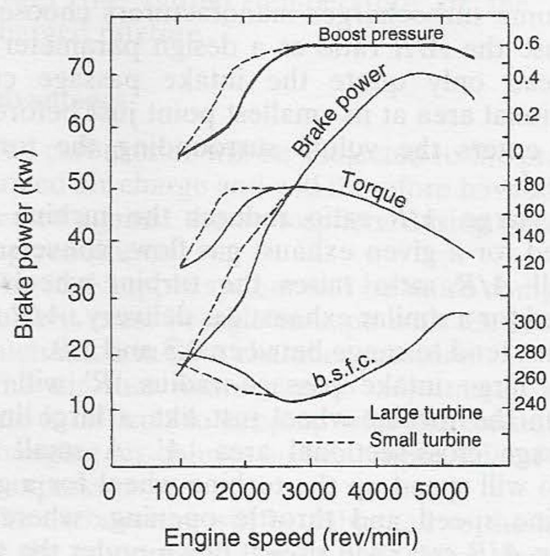


Fig. 6.52 Effect of reducing diameter of turbine-wheel on engine low-speed performance

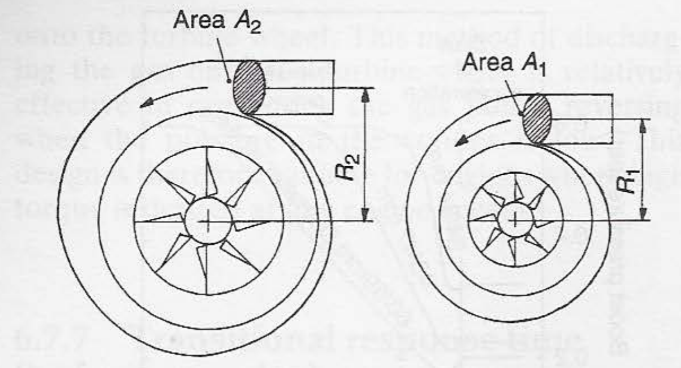


Fig. 6.53 Comparative A/R ratios for vaneless turbine housings

6.7.8 *A/R* ratio

(Fig. 6.53)

The speed and acceleration of the turbine and compressor wheel assembly is influenced by a number of factors, but one of the most critical and important controlling parameters is the A/R ratio.

The A/R ratio (Fig. 6.53) is the smallest cross-sectional area (CSA) of the intake passages in the turbine housing before the flow path spreads around the circumferential throat leading to the turbine-wheel divided by the distance from the centre of the turbine-wheel to the centroid of area 'A' i.e.

$$A/R \text{ ratio} = \frac{\text{smallest CSA of passage leading to volute}}{\text{distance between CSA centroid and centre of shaft}}$$

Some turbocharger manufacturers choose not to use the A/R ratio as a design parameter and instead only quote the intake passage cross-sectional area at its smallest point just before the gas enters the volute surrounding the turbine wheel.

A large A/R ratio reduces the turbine spin speed for a given exhaust gas flow, conversely a small A/R ratio raises the turbine-wheel spin speed for a similar exhaust gas delivery. A/R ratio values tend to range between 0.3 and 1.0.

A large intake passage radius 'R' will slow down the turbine-wheel just like a large intake passage cross-sectional area 'A'. A small A/R ratio will speed up the turbine-wheel for a given engine speed and throttle opening, whereas a large A/R ratio will slow it down under the same operating conditions.

For example, if a turbine housing has an A/R ratio of 0.7 and an earlier boost is demanded, then a smaller turbine housing A/R ratio of something like 0.6 or 0.5 should be tried. However, if a slightly later boost is required then a turbine housing with a larger A/R ratio, say 0.8 or 0.9, should be fitted.

6.7.9 Carburettor location

The carburettor may be located either upstream on the intake side of the compressor, in which case the air is drawn (sucked) through the carburettor venturi, or it may be positioned downstream on the outlet side of the compressor, and here the air is forced (blown) through the carburettor venturi. The merits and limitation of both arrangements are compared as follows.

Carburettor positioned upstream of compressor (suck through)

(Fig. 6.54(a))

Advantages

a) The carburettor operates at ambient pressure and therefore can still maintain the standard fuel pump system
b) The carburettor operates under normal temperature conditions
c) The carburettor does not require any modification except a matching of the jet sizes to cope with the greater volume of air flow
d) The carburettor tuning is easier than for downstream mounted carburettors
e) Charge mixture distribution under most driving conditions is generally good

Disadvantages

a) The turbocharger must have a special seal to avoid drawing oil into the compressor housing during part throttle high vacuum operating conditions
b) The longer intake flow path and the vacuum before the compressor may cause the fuel to condense on the cool manifold walls, which can upset the mixture distribution, particularly at low operating speed conditions
c) Under certain operating conditions the throttle response may be impaired due to the wetting of the extended intake passage walls
d) Segregation of liquid fuel from the air charge under transient driving conditions can occur

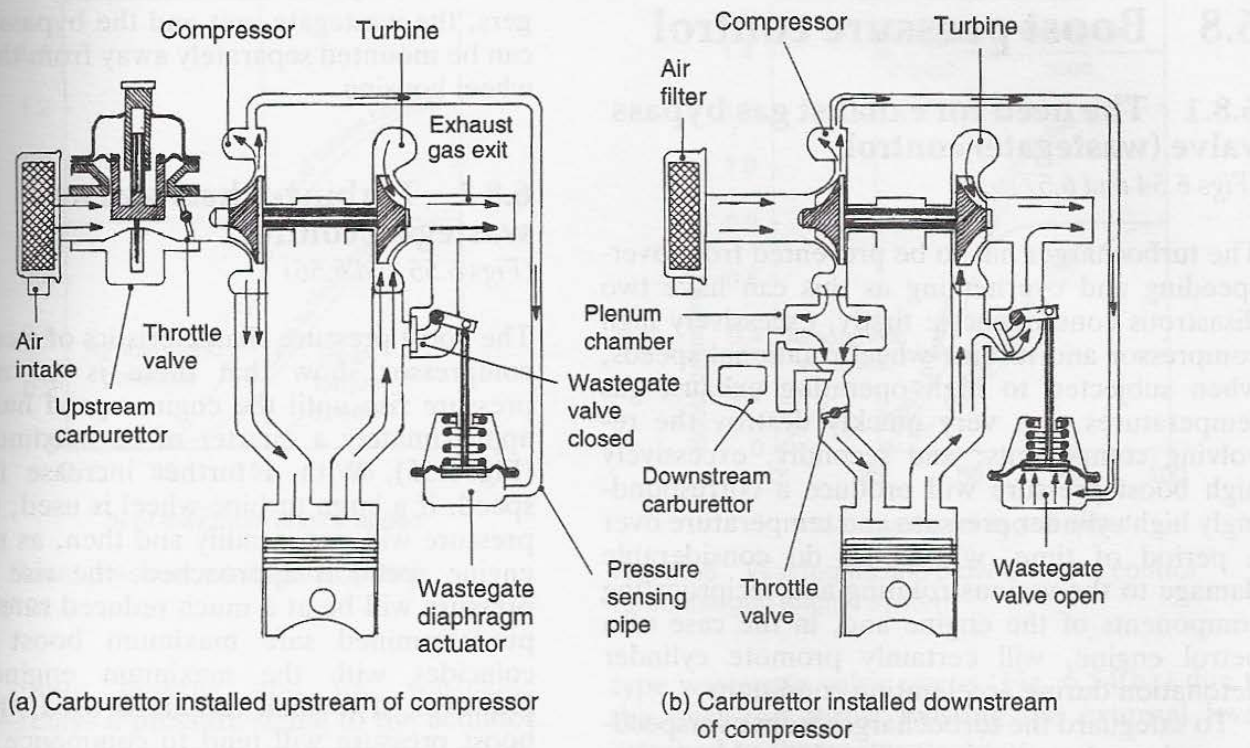


Fig. 6.54 Turbocharged engine layout with carburettor position either upstream or downstream of the compressor

the compressor volute housing, which may cause the air–fuel mixture ratio to be unstable

e) The cooling effect of the liquid fuel is transferred to the air stream when it is still cool and not when it has been compressed and heated, where it would provide some degree of intercooling
f) The greater pressure differential across the compressor, with the carburettor's venturi ahead of the compressor, will increase the temperature of the compressed charge

Carburettor positioned downstream of compressor (blow through)

(Fig. 6.54(b))

Advantages

a) The danger of high vacuum levels in the compressor housing under part throttle conditions is greatly reduced so that the need for special seals on the compressor end of the spindle is avoided
b) The carburettor can remain in the same position as for the standard naturally aspirated engine layout
c) Discharging liquid fuel on the heated output side of the compressor provides good air–fuel mixture distribution and fuel vaporization under cold-start and warm-up operating conditions
d) Discharging liquid fuel into the compressed and heated air charge provides a certain amount of intercooling of the charge before it enters the cylinder
e) With the carburettor on the downstream side of the compressor the pressure difference across it will be smaller, which will marginally lower the output temperature of the discharged mixture

Disadvantages

a) The carburettor will be subjected to the pressurized air charge and will therefore have to be sealed against the atmosphere during operating conditions
b) The fuel supply system will be more complex since it must be able to cope with the severe fluctuation in float chamber pressure
c) The air–fuel mixture ratio adjustment may find it difficult to keep up with the constantly changing air density on the output side of the compressor
d) The carburettor will be subjected to the heat of the compressed charge as it flows through the venturi. Therefore, means must be provided to cool the carburettor assembly.

6.8 Boost pressure control

6.8.1 The need for exhaust gas bypass valve (wastegate) control

(Figs 6.54 and 6.57)

The turbocharger has to be prevented from overspeeding and overheating as this can have two disastrous consequences: firstly, excessively high compressor and turbine-wheel rotational speeds, when subjected to high operating exhaust gas temperatures, can very quickly destroy the revolving components; and secondly, excessively high boost pressure will produce a correspondingly high cylinder pressure and temperature over a period of time, which can do considerable damage to the various rotating and reciprocating components of the engine and, in the case of a petrol engine, will certainly promote cylinder detonation during acceleration conditions.

To safeguard the turbocharger from overspeeding and overheating, a portion of the exhaust gas expelled from the cylinders under high engine-load and/or speed operating conditions is deliberately made to bypass the turbine housing and instead flow directly to the exhaust pipe. Under extreme operating conditions, something like 30% to 40% of the exhaust gas can be diverted away from the turbine throat with the effect that the turbine will not increase its speed and the output boost pressure will remain approximately constant with any further rise in engine speed.

The exhaust gas bypass passage opening is controlled by a wastegate in the form of either a poppet-type valve (Fig. 6.57) or a swinging-flap type valve (Fig. 6.54). Both types of wastegate valves are normally operated by a diaphragm actuator controlled by either the boost pressure from the volute impellor housing or by the exhaust manifold gas pressure.

With the poppet-type valve wastegate the long stem of the valve is connected directly to the diaphragm actuator, this stem is usually enclosed in a finned housing to improve the heat dissipation from the valve and actuator assembly. Conversely, the swinging-flap type wastegate is operated by a short external lever which is linked to the diaphragm actuator by a long push-rod, so that the actuator is practically insulated from the exhaust gas heat.

The wastegate and bypass passageways for small turbochargers can be integral with the turbine-wheel housing or, for the larger turbochargers, the wastegate unit and the bypass passages can be mounted separately away from the turbine wheel housing.

6.8.2 Turbine-wheel size and wastegate control

(Figs 6.55 and 6.56)

The boost pressure characteristics of a centrifugal compressor show that there is no noticeable pressure rise until the engine speed has risen to approximately a quarter of its maximum speed (Fig. 6.55). With a further increase in engine speed, if a large turbine-wheel is used, the boost pressure will rise steadily and then, as maximum engine speed is approached, the rise in boost pressure will be at a much reduced rate until the predetermined safe maximum boost pressure coincides with the maximum engine speed. However, if a small turbine is used, the rise in boost pressure will tend to commence at about the same engine speed, but its rise will be much steeper, and it will increase to a much higher pressure due to the lighter and smaller turbine-wheel's ability to increase its speed in a shorter time and for the wheel to reach a much higher spin speed.

Without a wastegate to limit the exhaust gas energy going to the turbine housing, the small turbine-wheel spin-speed would rise beyond the safe maximum burst speed of the rotating assembly and therefore could not be utilized. If, however, a portion of the exhaust gas bypasses the turbine housing when the desired boost pressure is reached then the small turbine-wheel will develop a positive and potent boost pressure much earlier than a large turbine-wheel (Fig. 6.56). Thus, once the desired boost pressure has been obtained the wastegate is made to open, and the bypass passage will therefore divert a sufficient quantity of the discharged exhaust gas from the turbine housing so that the boost pressure remains fairly constant throughout the upper speed range of the engine.

6.8.3 Boost-pressure controlled wastegate

(Figs 6.54(a and b) and 6.57(a and b))

With the boost pressure sensing method of control, a rubber pipe—connected between the com-

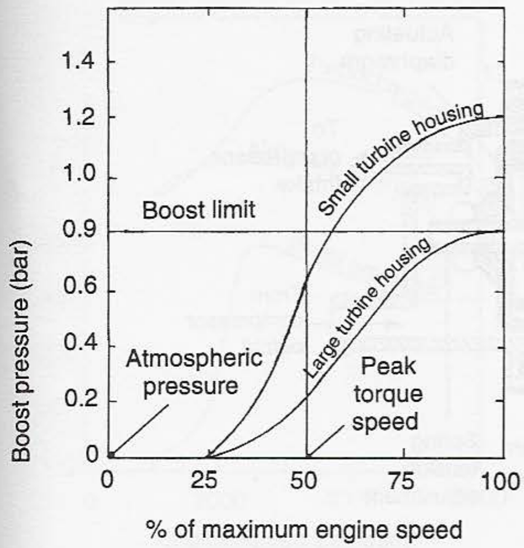


Fig. 6.55 Boost pressure wastegate control characteristics

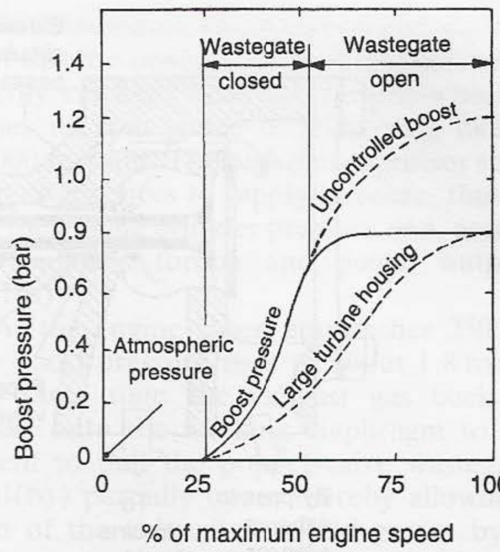


Fig. 6.56 Wastegate and blow-off valve control turbocharged engine system

pressor volute housing and the wastegate actuator—relays a pressure signal to the actuator diaphragm. Under normal steady driving conditions the wastegate valve is closed (Figs 6.54(a) and 6.57(a)), however, when this pressure reaches the predetermined value (say 1.6 bar) the charge pressure acting on the diaphragm's cross-sectional-area will be sufficient to force back the return-spring. If a poppet-type wastegate valve is used (Fig. 6.57(b)) it will be pushed into the open position permitting exhaust gas to bypass the turbine housing. Alternatively, the swing-flap type wastegate valve opens (Fig. 6.54(b)) due to the push-rod thrust twisting the external lever attached to the wastegate pivot.

Wastegate assembly air cooling system

(Fig. 6.58)

With an integral turbine housing and poppet-valve type wastegate there is a tendency for the poppet-valve and diaphragm actuator assembly to overheat. One way of overcoming this problem is to circulate a portion of the boost air supplied to

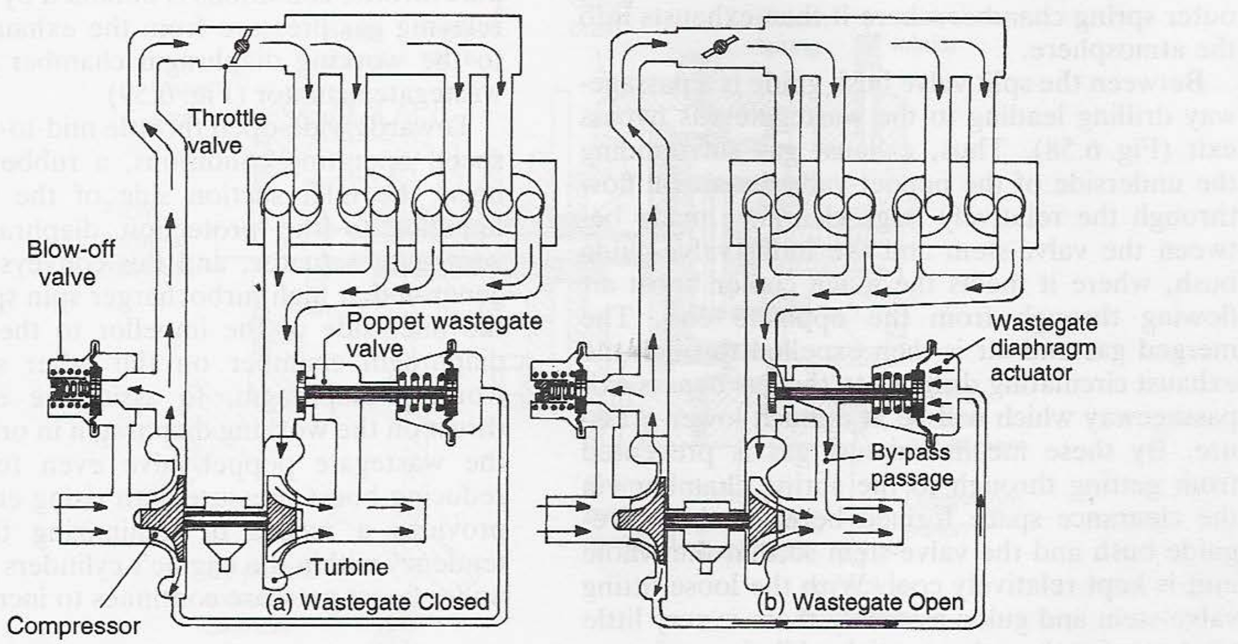


Fig. 6.57 Matching turbine housing size to meet engine requirements

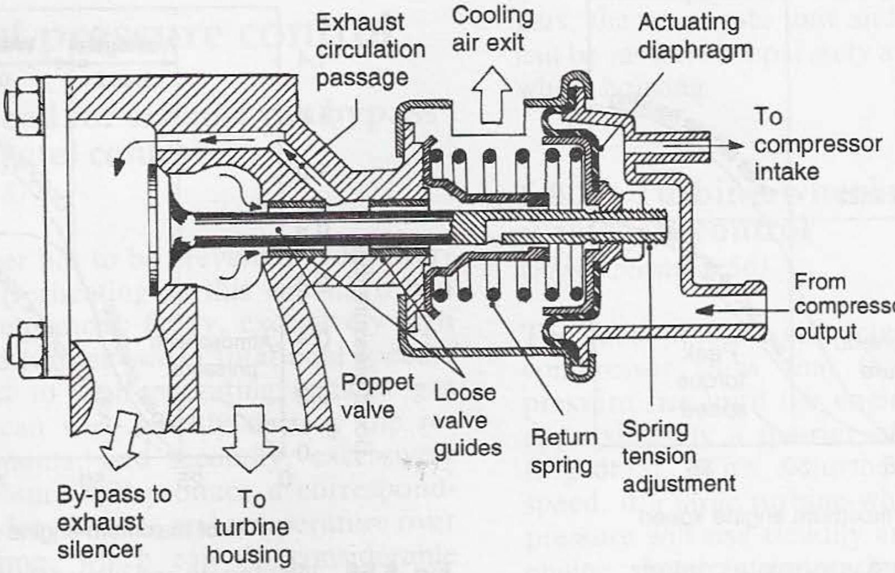


Fig. 6.58 Poppet-valve type integral wastegate with cooling air circulation and boost pressure actuated

the wastegate actuator diaphragm chamber back to the intake side of the compressor. At the same time some of the compressed air charge enters the central drilling in the valve-stem and comes out on one side in a funnel-shaped chamber surrounding the outer exposed end of the valve-guide (Fig. 6.58). Compressed air enters the relatively large clearance space formed between the valve-stem and guide so that it provides a cooling influence to the valve-stem assembly, the bulk of the air, however, will flow from the mouth of the funnel chamber into the outer spring chamber where it then exhausts into the atmosphere.

Between the split valve bush guide is a passageway drilling leading to the wastegate gas bypass exit (Fig. 6.58). Thus, exhaust gas surrounding the underside of the poppet-valve head will flow through the relatively large clearance made between the valve-stem and the inner valve-guide bush, where it meets the much cooler boost air flowing through from the opposite end. The merged gas and air is then expelled through the exhaust circulating drillings to the gas bypass exit passageway which will be at a much lower pressure. By these means exhaust gas is prevented from getting through to the spring chamber via the clearance space formed between the valve-guide bush and the valve-stem so that the whole unit is kept relatively cool. With the loose fitting valve-stem and guide assembly there is very little tendency for the valve to stick while in service.

6.8.4 Exhaust back-pressure and vacuum controlled wastegate

(Figs 6.59, 6.60 and 6.61(a–c))

This method of opening and closing the wastegate provides a steep rise in boost pressure to approximately 1.8 bar at 2500 rev/min followed by a steady decline of boost down to roughly 1.6 bar at 5000 rev/min, after which the charge pressure remains very nearly constant (Fig. 6.61).

The initial wastegate opening under mid-speed part throttle conditions is obtained by a steel pipe relaying gas pressure from the exhaust manifold to the working diaphragm chamber side of the wastegate actuator (Fig. 6.59).

Towards wide-open throttle mid-to-high engine speed operating conditions, a rubber pipe connects the inlet suction side of the compressor impellor to the protection diaphragm of the wastegate actuator, and this conveys a vacuum, generated at high turbocharger spin speeds, from the inlet side of the impellor to the protection diaphragm chamber on the other side of the working diaphragm, to assist the exhaust gas thrust on the working diaphragm in order to open the wastegate poppet-valve even further. The reducing boost pressure with rising engine speed provides a means of minimizing the pinking tendency within the engine's cylinders as its speed under boost pressure continues to increase.

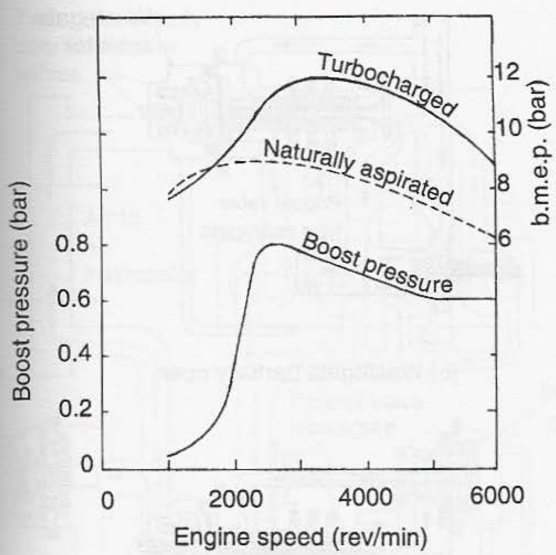


Fig. 6.59 The effects of turbocharging on the brake mean effective pressure characteristics at different engine speeds

Stages in wastegate operation

(Fig. 6.61(a–c))

During idle and low part throttle engine speeds the turbocharger spin speed may range between 5000–20000 rev/min, these compressor wheel speeds are much too low in producing any pressure changes and, therefore, the fresh charge will be drawn into the induction manifold via the spaces between the impellor blades just like a naturally aspirated engine.

With increased engine speed and slightly wider part throttle operation, the increased exhaust gas energy released onto the turbine-wheel rapidly raises its spin speed to something like 30000–50000 rev/min. The higher compressor spin speed now commences to supply pressure, thus increasing the engine cylinder pressure and, accordingly, the engine's torque and power output (Fig. 6.61(a)).

As the engine speed approaches 2500 rev/min the boost pressure rises to about 1.8 bar, and at the same time the exhaust gas back-pressure thrust onto the working diaphragm will be sufficient to pull the poppet-valve wastegate (Fig. 6.61(b)) partially open, thereby allowing a portion of the exhaust manifold gas to bypass the turbine-wheel housing. This will regulate the amount of exhaust gas energy reaching the turbine so that the turbine-wheel maximum spin speed will automatically be restricted to roughly 100000–120000 rev/min and, correspondingly, the boost pressure will be limited to something of the order of 1.8 bar.

With still higher engine speeds and wider throttle openings the vacuum on the inlet side of the compressor wheel progressively rises so that the actuator will be subjected to a high exhaust gas back-pressure on the working diaphragm side and a relatively high vacuum on the opposite protection diaphragm side (Fig. 6.61(c)). The additive effect of both gas pressure and compress-

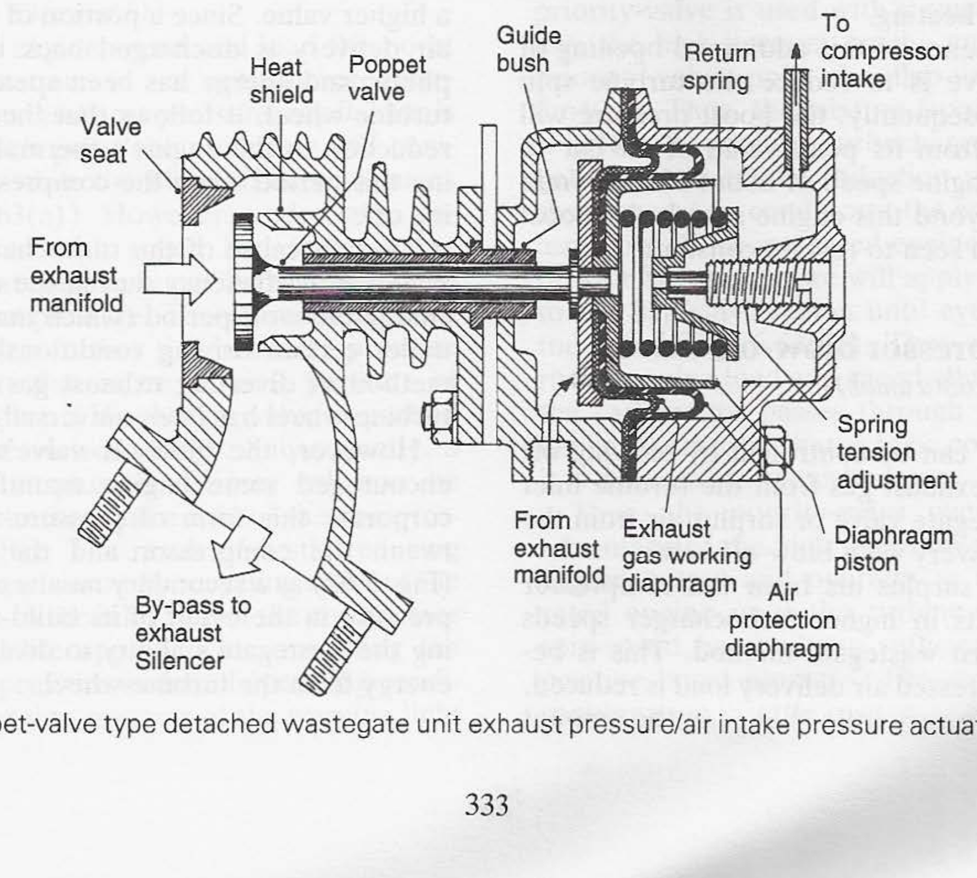


Fig. 6.60 Poppet-valve type detached wastegate unit exhaust pressure/air intake pressure actuated

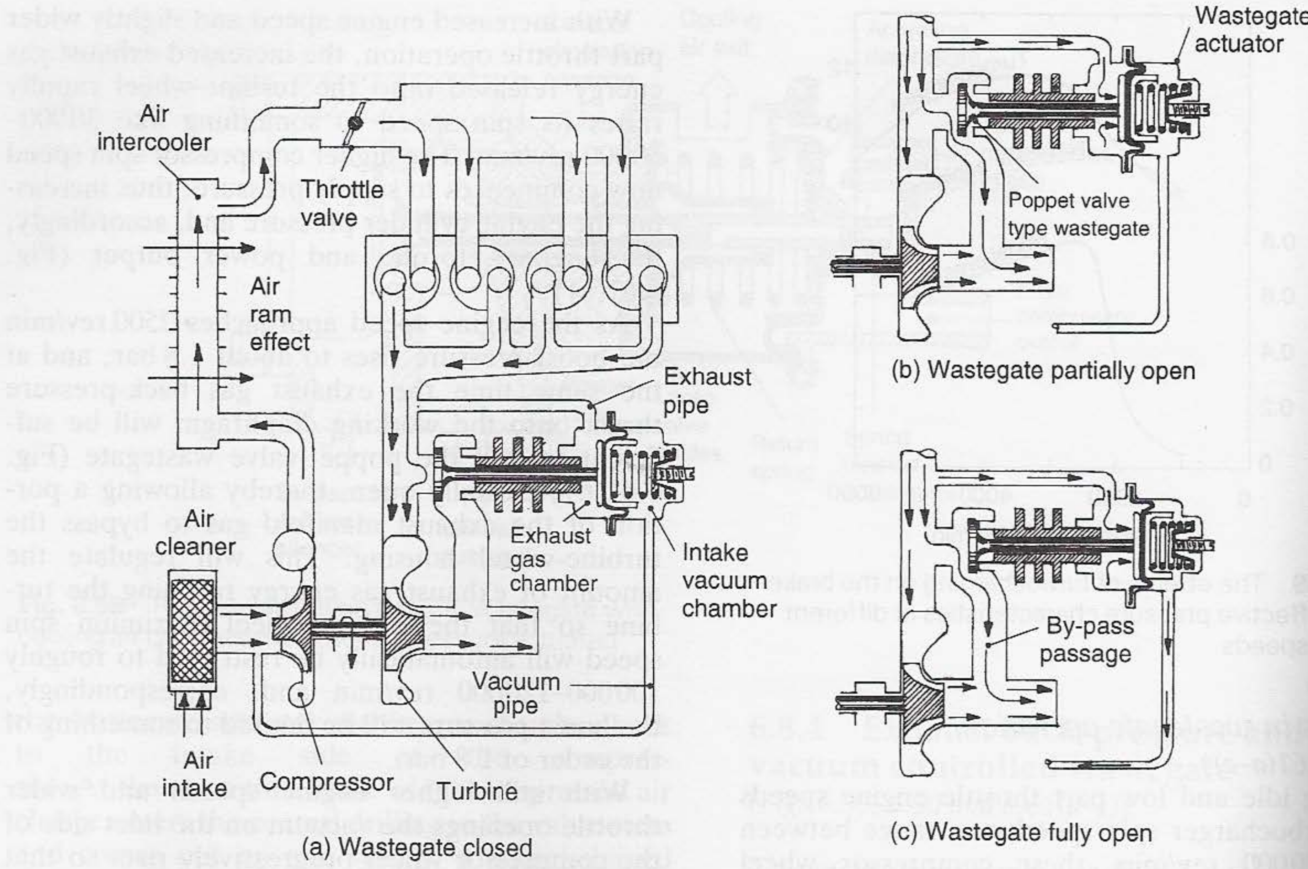


Fig. 6.61 Exhaust back-pressure and vacuum controlled wastegate system

or inlet vacuum will steadily move the poppet-valve fully away from its seat until it butts against the annular heat shield which protects the valve neck from overheating.

The consequence of this additional opening of the poppet-valve is to reduce the turbine spin speed and, subsequently, the boost pressure will also decrease from its peak value of 1.8 bar to 1.6 bar at an engine speed of about 5000 rev/min (Fig. 6.60). Beyond this engine speed, the boost pressure is then seen to remain constant.

6.8.5 Compressor blow-off valve

(Figs 6.62 and 6.63(a and b))

Boost pressure can be controlled by blowing off either surplus exhaust gas from the turbine inlet through a wastegate valve or surplus air from the compressor delivery via a blow-off valve.

Blowing off surplus air from the compressor discharge results in higher turbocharger speeds than the exhaust wastegate method. This is because the compressed air delivery load is reduced, but there will be very little change in the amount of gas energy passing through the turbine-wheel; consequently, the excess energy input to the turbine will raise the rotor assembly spin speed to a higher value. Since a portion of the compressed air delivery is discharged back into the atmosphere, and energy has been spent in driving the turbine-wheel, it follows that there will be some reduction in the engine's thermal efficiency during this period when the compressed air is blowing off.

Thus, because of the turbocharger's relatively low overall efficiency during the compressor discharge blow-off period (which may be prolonged under certain driving conditions) the wastegate method of diverting exhaust gas away from the turbine-wheel has been universally adopted.

However, the blow-off valve's simplicity has encouraged some engine manufacturers to incorporate this form of pressure-relief valve between the compressor and the inlet manifold (Fig. 6.62) as a secondary means of limiting boost pressure in the event of its build-up rate exceeding the wastegate's ability to divert sufficient gas energy from the turbine-wheel.

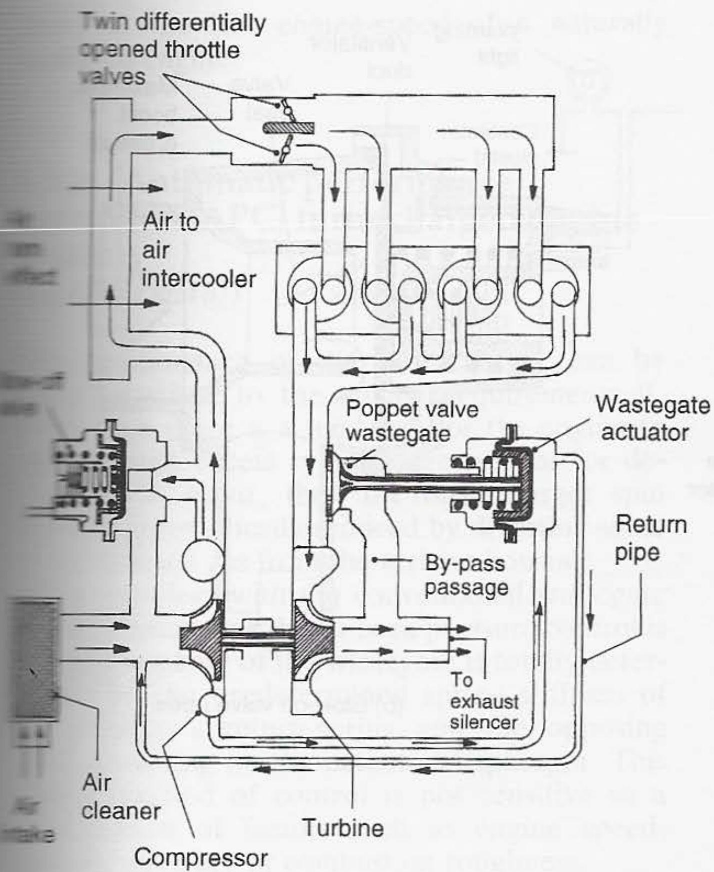


Fig. 6.62 Wastegate and blow-off valve boost pressure controlled

Stages in wastegate and blow-off valve operation

(Figs 6.62 and 6.63(a and b))

With increased engine speed and load the boost pressure will continue to rise until the predetermined maximum discharge pressure, say 0.7 bar is reached, at which point the wastegate will commence to open while the blow-off valve remains closed (Fig. 6.63(a)). However, under large engine loads or at high speed and large power outputs the wastegate may not be totally able to control the boost pressure. Therefore, under such conditions, with a further rise in charge pressure of only 0.1 bar, that is, to a boost pressure of 0.8 bar, the blow-off diaphragm valve is pushed open by the pressurized charge. Subsequently, a portion of the surplus air delivery will now be bypassed back to the intake side of the compressor (Fig. 6.63(b)). The result of firstly reducing turbine power and secondly recycling some of the air charge back to the inlet side of the compressor provides a reliable approach in controlling the delivery boost pressure under all operating conditions. If the warning contacts of the warning light closes (Fig. 6.63(b)) the light will be illuminated, indicating to the driver that excessively high boost pressure is being generated and that possibly the wastegate has become stuck in the closed position.

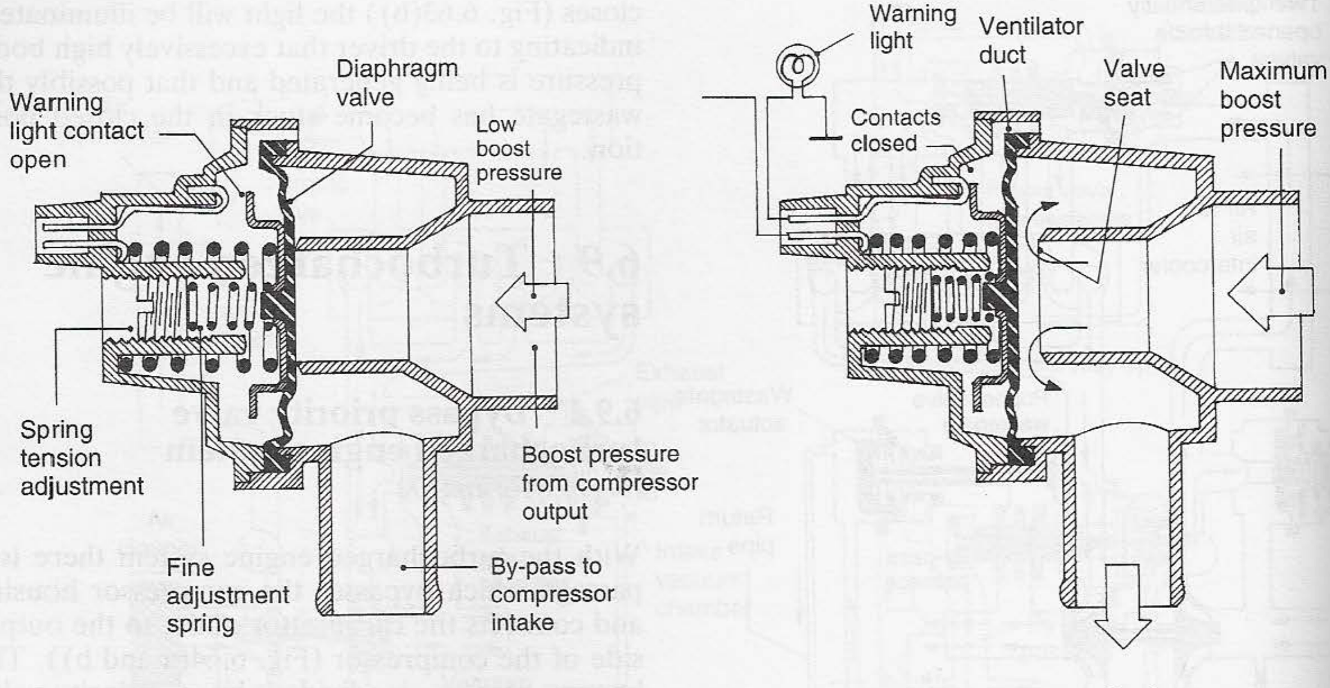
6.9 Turbocharged engine systems

6.9.1 Bypass priority valve turbocharged engine system

(Fig. 6.64(a and b))

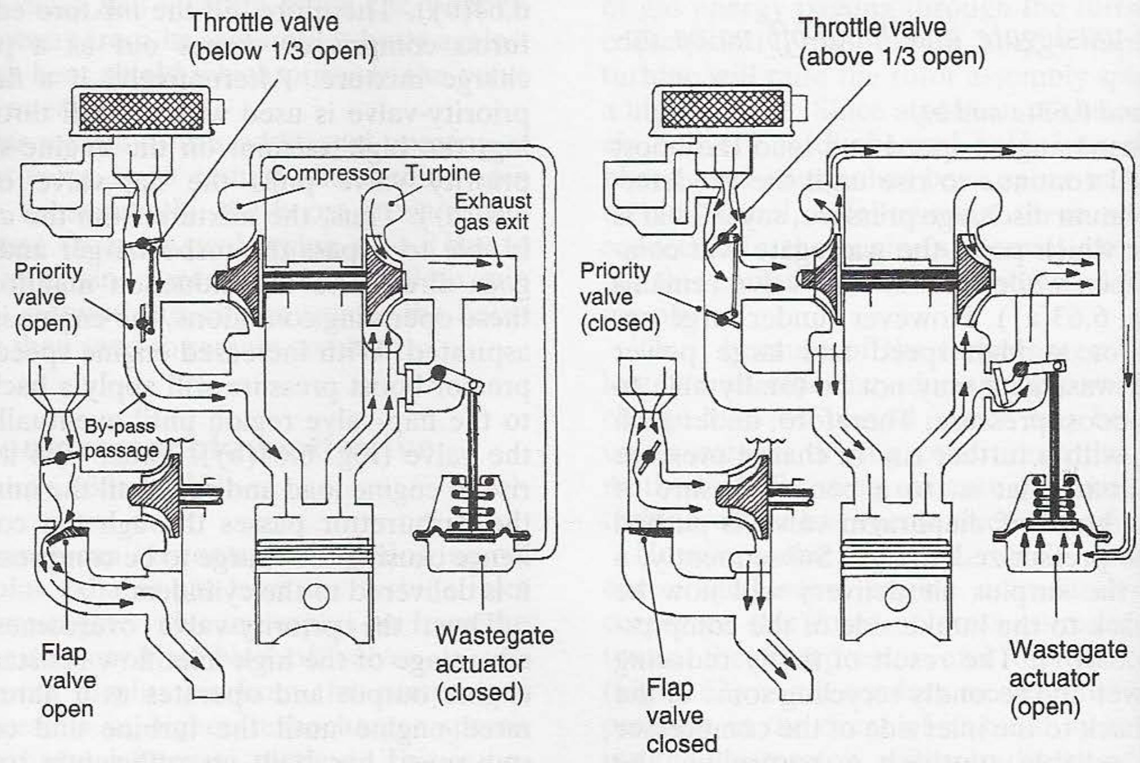
With the turbocharged engine system there is a passage which bypasses the compressor housing and connects the carburettor outlet to the output side of the compressor (Fig. 6.64(a and b)). The bypass passage is divided by a priority-valve which may be interconnected to the carburettor throttle valve or it may be a flexible flap-valve. When an interlinked priority valve is employed the valve remains open while the carburettor throttle opening is less than one-third fully open, the engine is then naturally aspirated (Fig. 6.64(a)). Above this throttle setting, however, the interlinkage closes the priority valve (Fig. 6.64(b)). Therefore, all the mixture entering the turbo compressor comes out as a pressurized charge mixture. Alternatively, if a flexible flap priority-valve is used with a small throttle opening, the high vacuum on the engine-side of the priority valve pulls the flap-valve open (Fig. 6.64(a)). Thus, the mixture from the carburettor is able to bypass the turbocharger and therefore goes directly to the induction manifold. Under these operating conditions, the engine is naturally aspirated. With increased engine speed the compressor boost pressure will apply a back pressure to the flap-valve region until eventually it closes the valve (Fig. 6.64(b)). Thus, with any further rise in engine load and speed all the mixture from the carburettor passes through the compressor, hence causing the charge to be compressed before it is delivered to the cylinders.

Thus, the priority-valve overcomes the disadvantage of the high inlet flow resistance at low engine output and operates as a naturally aspirated engine until the turbine and compressor spin-speed has built up sufficiently to deliver a positive boost pressure. The result is an almost spontaneous throttle response, with the fuel eco-



(a) Blow-off valve closed (b) Blow-off valve open

Fig. 6.63 Diaphragm-type blow-off valve



(a) Low speed light-load condition (b) High-speed heavy load condition

Fig. 6.64 Turbocharged engine layout with part throttle by-pass priority valve and passage energy at the low engine-speed of a naturally aspirated engine.

6.9.2 Automatic performance controlled (APC) turbocharged engine systems

(Fig. 6.65(a and b))

The performance of the turbocharger can be better matched to the engine requirements if, every time there is a tendency for the engine to operate with excess cylinder pressure or for detonation to occur, then the turbocharger spin speed is automatically reduced by diverting some of the exhaust gas from the turbine housing.

The problem with the conventional wastegate boost pressure or exhaust back pressure control is that the opening of the wastegate is totally determined by the predetermined spring stiffness of the actuator's return-spring and the opposing pressure acting on the actuator diaphragm. This simple method of control is not sensitive to a combination of factors such as engine speed, cylinder pressure or combustion roughness.

However, a more sensitive and accurate method of regulating the turbine spin speed, and therefore boost pressure, is to use a wastegate actuator with a relatively weak return-spring and then to control the delivery of boost pressure from the compressor to the actuator diaphragm via a wastegate solenoid control valve unit (Fig. 6.65(a and b)).

An electronic (microprocessor) control unit forms an important part of the automatic performance control system: its function is to receive inputs from the knock sensor pressure transmitter as well as the ignition pulse frequency as a measure of the engine speed. This information is then processed, an output signal is then passed to the solenoid control valve which will either open or close off the passage connecting the compressor to the wastegate actuator.

The actual engine speed and load combinations at which the wastegate opens will therefore depend upon the microprocessor instantly responding to input signals, such as those supplied by the pressure transmitter (which measures boost pressure) and signals from the piezometric knock sensor (which detects any incipient pinking). The outcome is an instant lowering in boost pressure under severe and abnormal operating conditions, which therefore protects the engine against any possible damage. An ignition retardation device may be installed to reduce the ignition advance whenever there is a tendency for the engine cylinder combustion process to become rough.

This boost control system enables the engine to use fuel, with the usual octane rating between RON 91 and 98 without reducing the engine's compression ratio by more than one, i.e. say from 9.5:1 to, say, 8.5:1.

Operating conditions

Moderate-speed light-load conditions (Fig. 6.65(a)). When the engine is operating under moderate speed and part throttle opening the wastegate solenoid control valve is made to open due to the energized solenoid but, with higher engine speeds and large throttle opening or under rough combustion (detonation), the microprocessor will cut off the electrical signal to the solenoid so that the wastegate control valve closes and therefore returns the boost pressure back to the intake side of the compressor, so that all the exhaust gas is directed towards the turbine, which has to provide the maximum boost relative to the turbocharger spin speed.

High-speed heavy-load conditions (Fig. 6.65(b)). With higher engine speeds and large throttle opening or under rough combustion (detonation) the microprocessor will cut off the electrical signal to the solenoid so that the wastegate control valve closes. Accordingly, the compressed air from the compressor will be conveyed to the wastegate actuator diaphragm via the control valve. This, therefore, pushes open the wastegate and thereby reduces the amount of exhaust gas energy reaching the turbine housing.

6.9.3 Dual-stage twin-volute turbine housing turbocharged engine system

(Fig. 6.66(a and b))

The twin scroll turbine housing has two volute passages surrounding the turbine-wheel which are separated by an integrally cast wall. Both of these passages feed into a common throat which is completely exposed to the periphery of the turbine-wheel. One of the twin exhaust gas inlet passages to the volute is equipped with a trapdoor like flap-valve (Fig. 6.66(a and b)). Thus, the turbine housing intake passage flow path has two cross-sectional areas, A_1 and A_2. However,

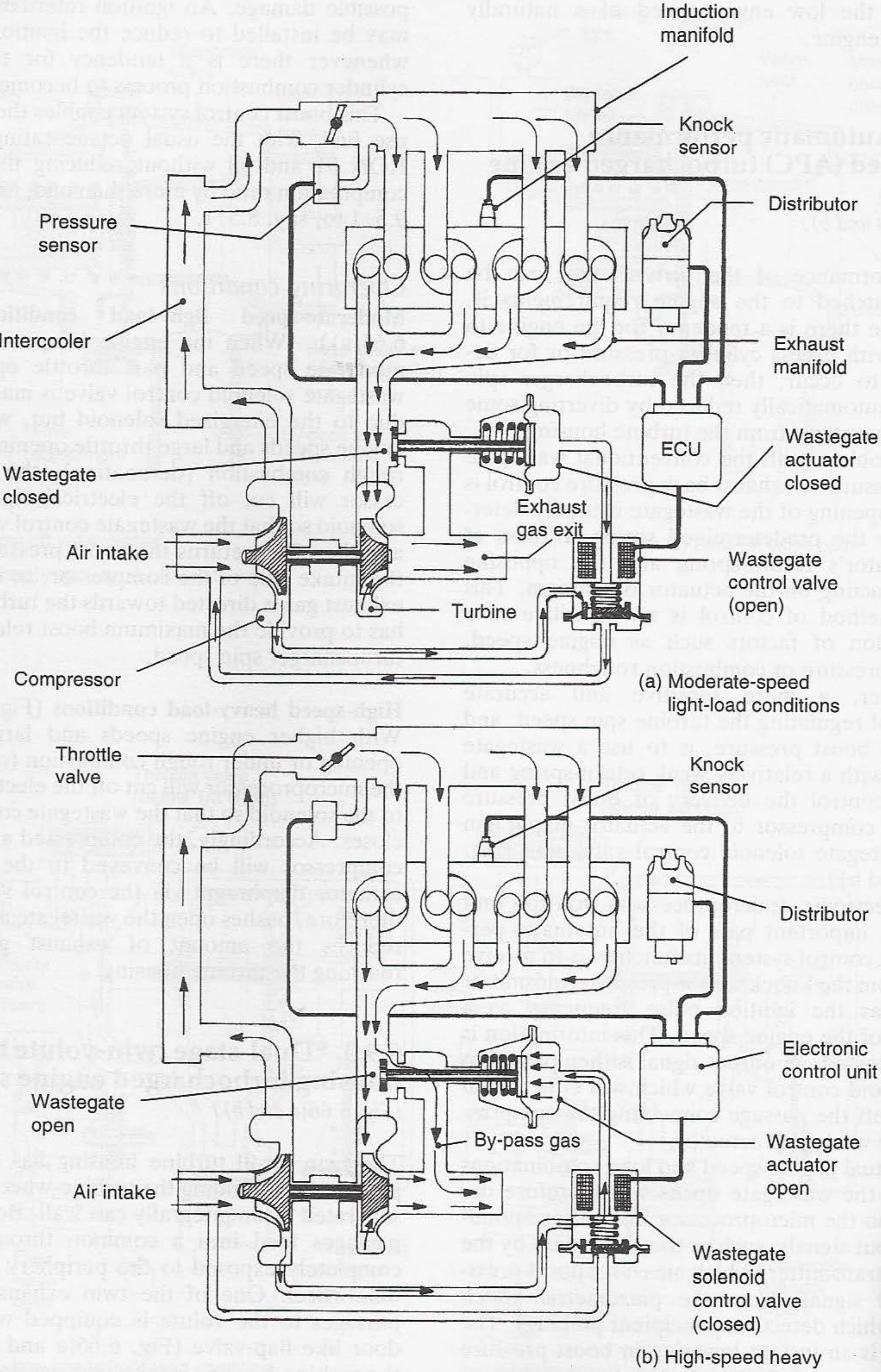


Fig. 6.65 Turbocharged engine system with knock sensor controlled wastegate

the gas can either go through the primary passage A_1 to the outer volute when the flap-valve is closed or through both the primary and secondary passages A_1 and A_2 respectively when the flap-valve in the secondary passage is open.

In both cases, the distance R between the turbine shaft centre and the centre of areas A_1 and A_2 is similar. Consequently, the turbine housing has two A/R ratios. That is, A_1/R ratio equal to 0.4:1 for the primary passage and $(A_1+A_2)/R$ equal to 1.0:1 for the combined primary and secondary passageways.

At low engine speeds (Fig. 6.66(a)) exhaust gas flows through the primary passage filling the outer volute and then discharges itself onto the turbine-wheel via the circumferential throat. The small cross-sectional area of the primary passage has a throttling effect which accelerates gas speed, and the volute and throat shape directs gas towards the turbine at roughly right-angles to the blades. This maximizes the thrust imparted to the turbine-wheel so that it rapidly increases the spin speed of the turbine and compressor assembly.

Once the engine has reached something like 2500 rev/min the change-over flap-valve opens (Fig. 6.66(b)). This now increases the flow-path cross-sectional area, and consequently the exhaust gas speed is reduced which will cause the gas to strike the turbine-blades at a more obtuse angle. With the slower gas speed the turbine will progressively slow down to a speed which matches the new flow conditions.

The small turbine housing primary passage provides good low engine speed response and minimizes turbo lag during transitional operating conditions, whereas the large turbine housing passages (primary and secondary) provide a measure of self-regulating speed and boost pressure control and, at the same time, reduce exhaust back-pressure, thereby improving cylinder filling and thermal efficiency.

The change-over flap-valve is operated by a diaphragm actuator which is, in turn, controlled by a change-over solenoid control valve and an electronic (microprocessor) control unit.

Operating conditions

(Fig. 6.66(a and b))

When the engine's speed and boost-pressure signal inputs to the microprocessor reach a predetermined set of combinations, an output signal to the change-over solenoid-controlled valve will open the valve so that boost is conveyed to the diaphragm actuator. This then forces open the change-over flap-valve (Fig. 6.66(b)). Accordingly, the flow path to the turbine will now be via both the primary and secondary passageways leading to the turbine's circumferential throat. If the engine speed and boost-pressure drop below the designed change-over from large to small intake cross-sectional area (Fig. 6.66(a)), then the change-over solenoid valve will be energized. This opens the control-valve so that the boost pressure will by bypassed back to the intake side of the compressor. It thus releases the diaphragm actuator so that the change-over flap-valve closes again. The turbine and compressor-wheel assembly will then be able to build up rapidly the boost pressure in the low-speed range or during a transition period where the engine is being accelerated.

Under excessive boost pressure or rough combustion the wastegate solenoid control-valve closes, thereby permitting boost pressure to push the actuator diaphragm against the return-spring until the wastegate flap-valve opens and redirects some of the exhaust gas away from the turbine housing.

6.9.4 Variable nozzle area single volute turbine housing turbocharger engine system

(Fig. 6.67(a and b))

The narrowing flow path leading to the circular volute passageway surrounding the rim of the turbine wheel is known as the nozzle and has a variable cross-sectional area provided by a curved tapering flap hinged at one end (Fig. 6.67(a and b)). The impellor wheel for the compressor is 53 mm in diameter, and is driven by a 49.9 mm diameter turbine wheel. The angular movement of the flap between the smallest to the largest nozzle cross-sectional area, amounts to 27°. With the flap nozzle closed the smallest cross-sectional area A in the nozzle region is 313 mm^2, whereas with the flap fully open it is enlarged to 858 mm^2. Thus, taking the distance between the centroid of the nozzle cross-sectional area and the centre of the turbine shaft as R, then the A/R ratio for the smallest nozzle is 0.21 going up to 0.77 for the largest opening.

The variable nozzle flap is operated by a diaphragm actuator, which is itself controlled by a variable flap control valve (pressure control mod-

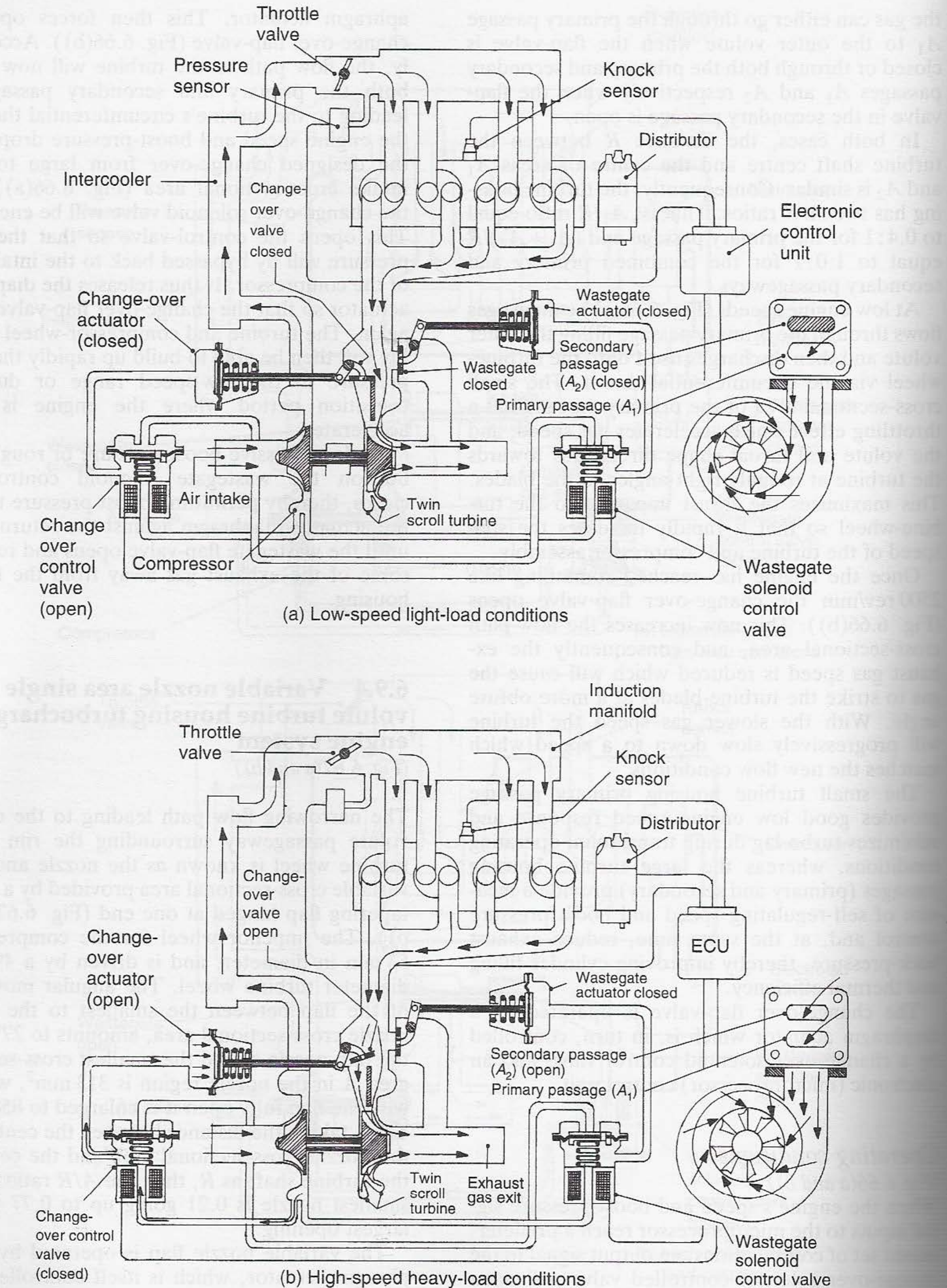


Fig. 6.66 Dual-stage scroll (volute) area turbocharged engine system

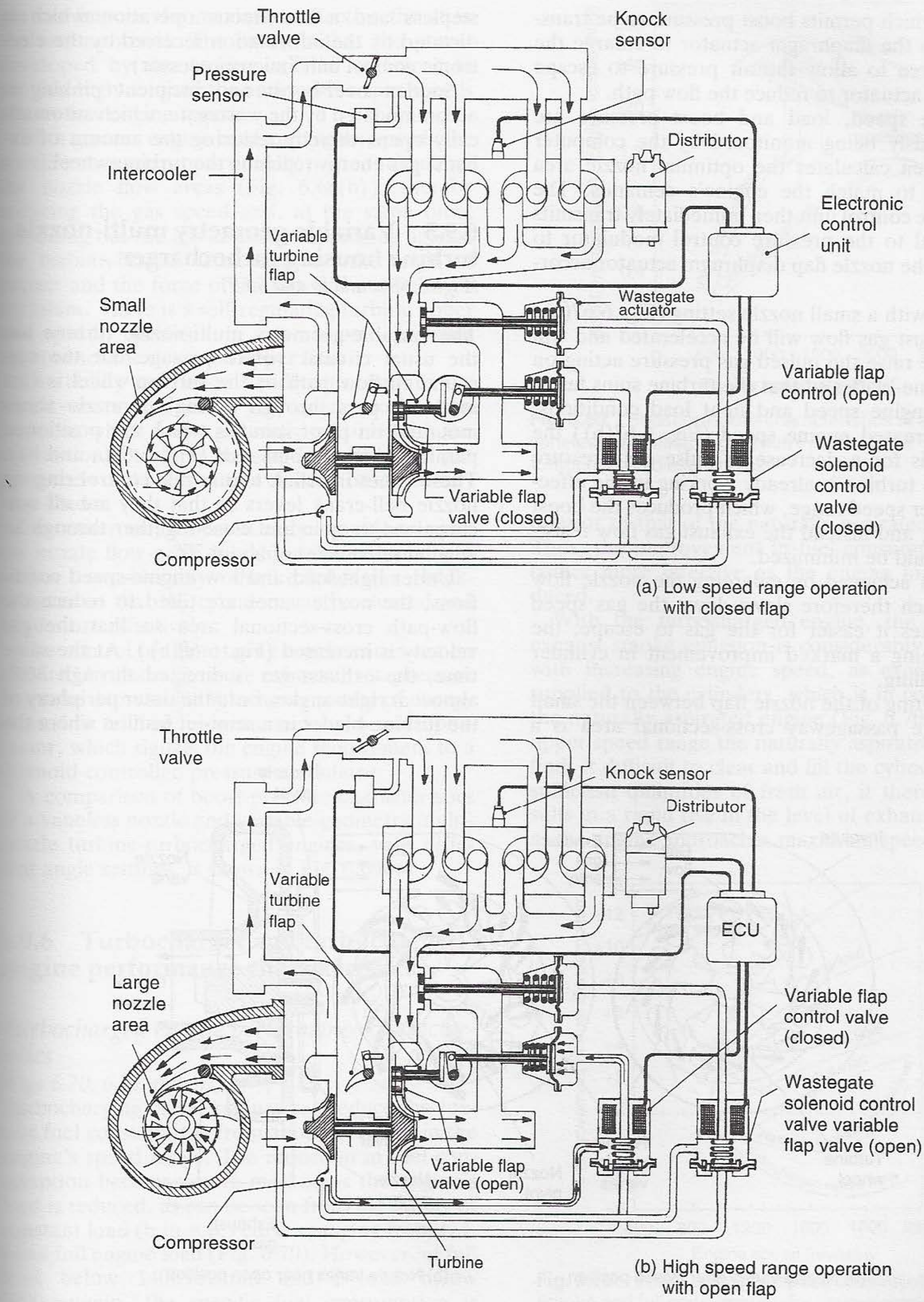


Fig. 6.67 Variable scroll (volute) area turbocharger engine system

ulator) which permits boost pressure to be transmitted to the diaphragm actuator to enlarge the nozzle area to allow the air pressure to escape from the actuator to reduce the flow path.

Engine speed, load and boost pressure are continuously being monitored by the computer which then calculates the optimum nozzle area required to match the engine's demands. The electronic control unit then immediately transmits the signal to the pressure control modulator to operate the nozzle flap diaphragm actuator accordingly.

Thus, with a small nozzle setting (Fig. 6.67(a)) the exhaust gas flow will be accelerated and will therefore raise the pulsed gas pressure acting on the turbine-blades so that the turbine spins faster at low engine speed and light load conditions. With increased engine speed (Fig. 6.67(b)) the priority is for a decrease in pulse gas pressure since the turbine is already working in the effective upper speed range, which produces the boost pressure, and instead the exhaust gas flow resistance should be minimized.

This is achieved by enlarging the nozzle flow area which therefore slows down the gas speed and makes it easier for the gas to escape, the result being a marked improvement in cylinder charge filling.

The tilting of the nozzle flap between the small and large passageway cross-sectional area is a stepless and a continuous operation which is dictated by the information received by the electronic control unit (microprocessor).

Sudden over-boost and incipient pinking is accommodated by the wastegate, which automatically opens thereby reducing the amount of exhaust gas energy reaching the turbine-wheel.

6.9.5 Variable geometry multi-nozzle turbine housing turbocharger

(Figs 6.68(a and b) and 6.69)

The variable-geometry multi-nozzle turbine has the usual circular volute passage but the surrounding flow path to the turbine-wheel is first made to pass through a ring of nozzle vanes mounted on pivot spindles which are positioned parallel to the turbine axis (Fig. 6.68(a and b)). These vanes interlink to a nozzle control ring via nozzle bell-crank levers so that they are all synchronized to open and close together through an angular movement of about 30°.

Under light-load and low engine-speed conditions, the nozzle vanes are tilted to reduce the flow-path cross-sectional area so that the gas velocity is increased (Fig. 6.68(a)). At the same time, the exhaust gas is directed through 360°, almost at right-angles, onto the outer periphery of the turbine-blades in a semi-jet fashion where the

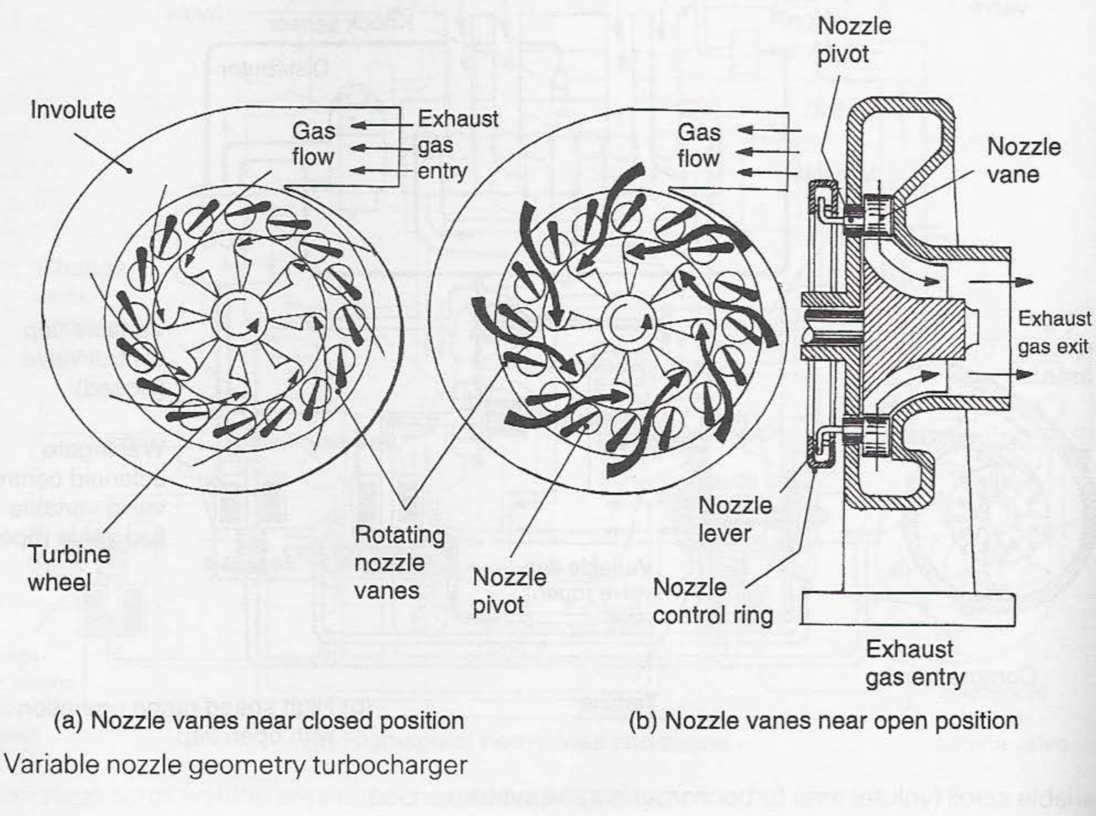


Fig. 6.68 Variable nozzle geometry turbocharger

gas can impose the maximum effective thrust to the turbine-wheel. The result is that the power developed by the turbine will be sufficient to provide high compressor and turbine spin speeds at very low engine speeds.

With greater engine load and higher speeds the nozzle-vanes are proportionally twisted to enlarge the nozzle flow areas (Fig. 6.68(b)), thereby reducing the gas speed and, at the same time, spreading out the gas discharge impinging against the turbine-blades. Consequently, the angle of impact and the force of contact will not be at its optimum. There is a self-regulating turbine upper spin-speed, particularly as boost pressure approaches the designed upper limit.

The ability to change rapidly the angle of the nozzle-vanes produces a very quick acceleration response which is not generally possible with the conventional turbine housing in which the gas from the volute discharges directly onto the turbine-blades.

The other major benefit of being able to open the nozzle flow-path at high engine outputs is to reduce considerably the exhaust-gas back-pressure so that the cylinders can be cleared and filled more effectively.

The control of the rotating nozzle-vanes can be actuated by boost pressure exhaust back-pressure or intake depression. Whichever source of energy is used it will be programmed by a microprocessor, which signals the engine requirement to a solenoid-controlled pressure modulator.

A comparison of boost-pressure characteristics of a vaneless nozzle and variable-geometry multi-nozzle turbine-turbocharged engines, with different angle settings, is shown in Fig. 6.69.

6.9.6 Turbocharger and turbocharged engine performance characteristics

Turbocharged engine performance characteristics
(Figs 6.70, 6.71 and 6.72)

Turbocharging diesel engines can reduce the specific fuel consumption from about 3 to 14% in the engine's speed range. The reduction in fuel consumption becomes more marked as the engine's load is reduced, as can be seen from the family of constant load (b.m.e.p.) curves ranging from $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$ and full engine load (Fig. 6.70). However, at full load below 1400 rev/min and $\frac{3}{4}$ load below 1000 rev/min, the specific fuel consumption is

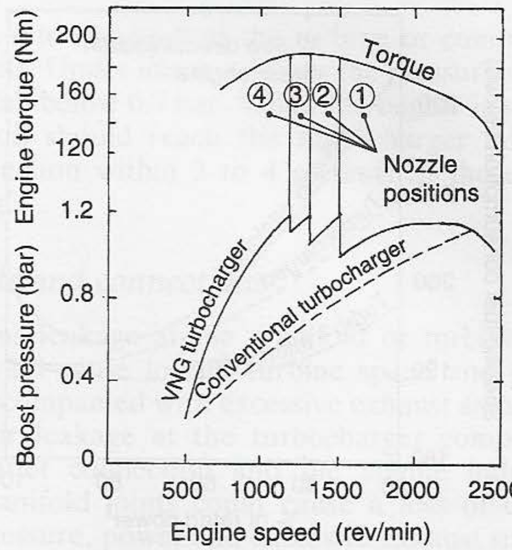


Fig. 6.69 Boost pressure characteristics of a vaneless and variable geometry multi-nozzle turbocharged diesel engine

inferior to that of the naturally aspirated engine. Thus, the improvement in fuel consumption becomes more effective as the engine load is reduced.

With the turbocharged engine, the level of exhaust smoke emission is considerably reduced with increasing engine speed, as excess air is supplied to the cylinders, which is in contrast to the naturally aspirated engine (Fig. 6.70). In the upper speed range the naturally aspirated engine finds it difficult to clear and fill the cylinders with sufficient quantities of fresh air, it therefore results in a rapid rise in the level of exhaust smoke as the engine approaches maximum speed.

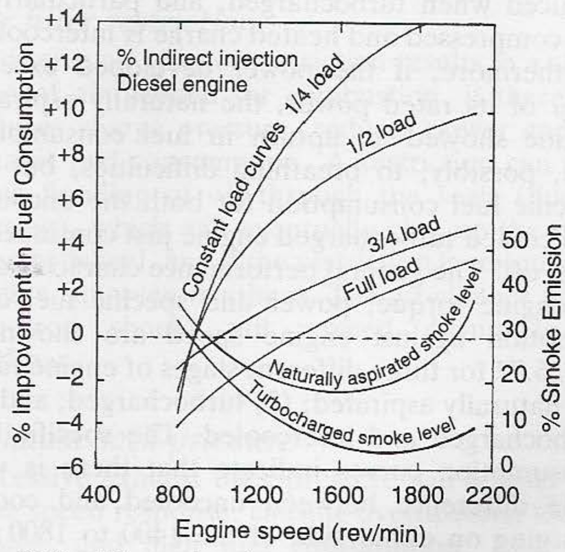


Fig. 6.70 Effect of engine speed on level of exhaust smoke and full consumption for various engine loads

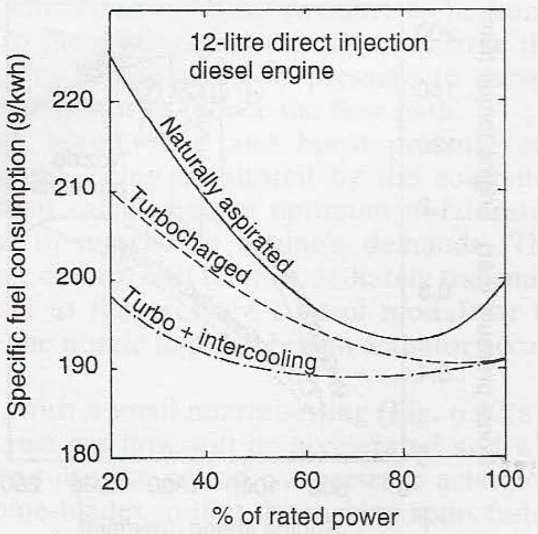


Fig. 6.71 Effect of varying the percentage of power on engine-specific fuel consumption for a naturally aspirated, turbocharged and turbocharged and intercooled state of tune

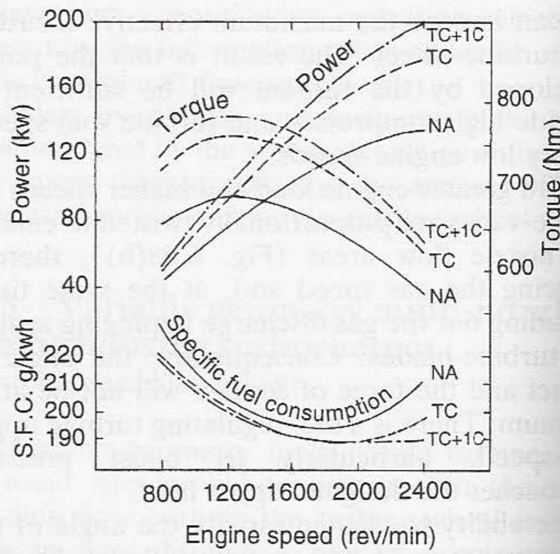


Fig. 6.72 Comparative performance for an engine operating under the following conditions: naturally-aspirated turbocharged and turbochaged and intercooled

Frictional losses rise rapidly with an increase in engine speed but do not rise in direct proportion to the engine load output. Thus, if the engine brake mean effective pressure (b.m.e.p.), and thus engine torque, is made to peak at lower engine speed where the mechanical losses are least, then there will be an improvement in the specific fuel consumption if the engine operates under these conditions. Figure 6.71 shows the benefits in specific fuel consumption compared with the naturally aspirated engine of an uncooled and intercooled turbocharged engine over the engine's operating power range. The graphs show that far less fuel is needed as the engine power is reduced when turbocharged, and particularly if the compressed and heated charge is intercooled. Furthermore, if the power developed exceeds 80% of its rated power, the naturally aspirated engine showed an upturn in fuel consumption due, possibly, to breathing difficulties, but the specific fuel consumption for both the uncooled and cooled turbocharged engine just continues to level off. The general performance characteristics of engine torque, power and specific fuel consumption against engine speed are shown in Fig. 6.72 for three different stages of engine tune: (1) naturally aspirated; (2) turbocharged; and (3) turbocharged and intercooled. The specific fuel consumption curves indicate that there is very little difference between uncooled and cooled charging on either side of the 1400 to 1800 rev/min speed band but the difference is more significant towards maximum speed.

Turbocharged petrol engines generally have reduced compression ratios to accommodate the high cylinder pressures and, under load, the ignition timing is automatically retarded to prevent detonation taking place, while at full load a rich mixture is necessary. Consequently, the turbocharged petrol engine's efficiency may not equal that of an equivalent sized naturally aspirated petrol engine although the engine's torque and power will be far superior.

6.10 Turbocharger fault diagnosis

6.10.1 Engine starting and stopping procedure

Starting

As soon as the engine fires and is revolving by its own power, remove one's foot from the accelerator pedal for about five seconds to permit the engine's lubricating pump to circulate oil to the turbocharger shaft and bearings before the turbine and compressor assembly becomes operational. If insufficient time is permitted for the oil to establish hydrodynamic lubrication conditions between the shaft and bearings then only boundary lubrication conditions prevail during the starting phase and the consequences will be a rapid wear rate.

Stopping
After completing a journey, particularly if the engine has been driven hard just before the vehicle stops, the engine should be allowed to idle for a short period until the turbine and compressor assembly have had time to reduce their speed to no-load rotational conditions. Hence, when the engine stops and the oil supply is cut off to the turbocharger, there will still be sufficient residual oil coverage between the shaft and bearings during the time it takes the rotating assembly to come to rest. If this precaution is not taken when stopping the engine, then it is highly likely that only boundary lubrication conditions will exist during the spin down time needed to bring the turbine and compressor assembly to a standstill.

One method used to ensure that there is adequate stopping time for the rotating assembly to slow down is to incorporate a time delay into the ignition switch circuit so that when the engine is switched off the engine will continue to run for a predetermined period before automatically cutting out.

6.10.2 Turbocharger service considerations

Faults
Symptoms which could indicate turbocharger malfunction are:

1 loss of power
2 excessive exhaust smoke
3 high fuel consumption
4 overheating
5 high exhaust temperature
6 oil leakage from turbocharger

Oil quality
Oil should be filtered below 15–20 μm. Only recommended oil specified by the manufacturer should be used. If inferior oil is used carbon deposits can be formed at the turbine end of the turbocharger causing excess wear of the sealing rings.

Oil supply
The minimum oil pressure when the engine is in load is 2.0 bar, however the pressure should not exceed 4.0 bar as this can force the oil past the seals into the back of the turbine or compressor wheels. Under idle conditions the pressure should not fall below 0.7 bar. When the engine is started the oil should reach the turbocharger inlet oil connection within 3 to 4 seconds of the engine firing.

Joints and connections

1 Gas leakage at the manifold or turbine inlet could cause loss of turbine speed and power accompanied with excessive exhaust smoke.
2 Air leakage at the turbocharger compressor outlet connection and the engine induction manifold joints could cause a loss of charge pressure, power and excessive exhaust smoke.

Oil leakage and restriction

1 A blocked air filter will produce a depression in the compressor wheel-chamber. This can cause oil to be drawn through the seals into the space behind the compressive-wheel with the result that it can clog up the compressor.
2 A damaged or badly fitted supply pipe connection will starve the turbine bearing assembly of oil causing excessive wear.
3 A blocked or kinked or otherwise damaged oil return drain-pipe may cause a build up of oil in the bearing housing.
4 High crank case pressure due to piston blow-by or clogged crankcase ventilation will restrict the flow of oil from the turbocharger bearing housing.

Air filter restriction
Excessive air cleaner restriction results in a shortage of air needed for combustion, it therefore reduces charge pressure, reduces power and increases fuel consumption. A restriction can produce flooding of oil through the seals (due to suction), which will eventually clog up the compressor-wheel, and if the restriction is serious, the engine can even overheat. The air-cleaner pressure-drop should not exceed 40 mm Hg or 0.05 bar.

Exhaust back-pressure
Excessive exhaust back-pressure will restrict turbine speed response, causing overheating, loss of power, and high fuel consumption. Excessive back-pressure can also force the gas through the seal and between the bearing and shaft; thus,

contamination will carburize the oil and clog the turbocharger components. Exhaust gas back-pressure should not exceed 40 mm Hg or 0.05 bar.

Boost pressure

Boost pressure should be checked every 50 000 km. The maximum boost pressure when the engine is running at maximum speed at full-load will vary according to the type of turbocharger and power unit. A typical boost pressure reading would be 0.8 ± 0.1 bar at 2100 rev/min for a large diesel engine.

Crankcase pressure

Crankcase pressure build-up should be checked as this indicates leakage of combustion gas from the cylinders, which will then contaminate the oil and shorten the service life of the various components. Crankcase pressure should not exceed 8 mm Hg or 0.01 bar.

Compressor housing checks

(Charts 1, 2 and 3 on p. 363)

Remove the inlet direct to the turbocharger periodically and check for the following.

1 Dirt and dust build-up on the impellor or in the housing, and for any signs of contact between the impellor and the housing. Excessive accumulation of dirt indicates either a leak in the ducting or a faulty air filter.
2 Observe if there is oil entering the compressor housing after the turbocharger has been operated for some time under load conditions. If there is, refer to chart 1.
3 Observe if there is oil in the inlet or outlet ducts or dripping from either housing. If there is refer to chart 2.
4 Spin the compressor-wheel by hand observing how freely it revolves. If the shaft tends to drag refer to chart 3.
5 Check for any unusual turbocharger vibration and noises when the engine is operating at rated output.

A bearing which is failing will produce a shrill whine over and above normal turbine whine. Usually, noises may result if there is improper clearance between the turbine-wheel and its housing.

6.10.3 Common turbocharging system faults

The most common turbocharger system faults come under the following headings:

1 noise
2 excessive smoke in exhaust gas
3 loss of power output
4 oil leakage from the turbocharger unit
5 damaged turbocharger malfunctioning

Noise

The probable causes are:

1 gas/air leakage
2 gas/air restriction
3 damaged turbocharger

Excessive smoke in exhaust gas and loss of power output

The probable causes can be similar for high levels of smoke and loss of power:

1 gas/air leakage
2 gas/air restriction
3 malfunctioning fuel pump and/or injectors
4 incorrect injection pump setting
5 dirt or soot deposits on compressor or turbine-wheel
6 oil leakage from engine
7 damaged turbocharger

Oil leakage

The probable causes are:

1 gas/air leakage
2 gas/air restriction
3 blocked crankcase ventilation
4 blocked oil return in the turbocharger
5 excessive idle
6 engine blow-by
7 damaged turbocharger

Damaged turbocharger

Damage to the turbocharger is usually due to worn bearings so that the turbine and/or compressor wheel fouls the housing.

An inspection can be made by removing the inlet connection and the exhaust pipe from the turbocharger unit. The rotating assembly should then be turned by hand to check whether it revolves easily or if a grating or scraping noise is experienced. If the bearings or seals are damaged then the cause must be traced.

Damaged compressor wheels can be caused by foreign particles bypassing the air filter so that they enter the compressor-wheel cells and then jam themselves between the blades and housing. They either severely damage the blade edges or bring the rotor assembly to a standstill.

Damaged turbine-wheels can be due to solid metallic objects being expelled from the engine cylinders and passing through the turbine-wheel passageways. These objects usually consist of engine components such as broken valves, pistons, piston rings and injector nozzles etc.

6.11 Intercooling

6.11.1 The need for intercooling

(Figs 6.73, 6.74, 6.75 and 6.76)

The density of air charge, that is, the mass per unit volume entering the cylinder, is of vital importance since squeezing more mass into the cylinder increases the power generated in the cylinder. With supercharged engines the air change entering the cylinder will be above the normal atmospheric density of air so that a comparison must be made between the actual density of charge in the cylinder to the air density under normal temperature and pressure (NTP) conditions (in which the temperature is taken as 16°C and the pressure as $101.3\,kN/m^2$ (1 bar)). The comparison used to relate different air densities in the cylinder to a known air density under normal temperature and pressure conditions is the air charge density ratio.

air charge density ratio =

$$A/R \text{ ratio} = \frac{\text{density of charge in cylinder under operating conditions}}{\text{density of charge in cylinder under NTP conditions}}$$

therefore air density ratio $= \dfrac{\rho}{\rho_{NTP}}$

The relationship between pressure ratio and air charge density ratio if the air temperature is held constant at three different temperatures is shown in Fig. 6.73. The graphs show that as the boost pressure ratio increases, the air density ratio increases likewise, however, the more the air is intercooled and its temperature reduced, the greater will be the rise in air charge density. Thus, if the air temperature is maintained at 30°C, the density at a pressure ratio of 2.2:1 will be about 2.1:1, whereas if the air temperature is kept constant at 90°C the air density ratio only rises to around 1.64:1. Well-designed intercoolers can hold the compressed air temperature to about 60°C.

The broken curve shows how the air density ratio increases if there is no intercooling. Here, it can be seen that as the boost pressure ratio increases beyond about 1.6:1 there is very little useful increase in the air density ratio for a considerable further increase in the pressure ratio.

When including the compressor efficiency it can be shown (Fig. 6.74), on a number of constant

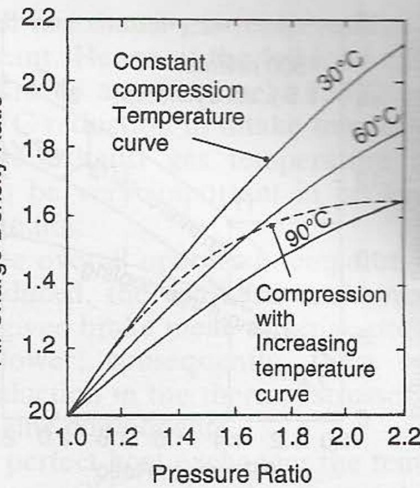


Fig. 6.73 Effect of pressure ratio on increase in charge density if the charge temperature is held constant or is allowed to rise

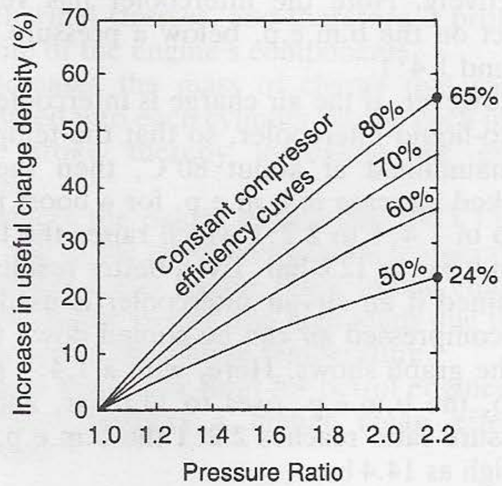


Fig. 6.74 Effect of pressure ratio on increase in useful charge density for various compressor efficiencies