2 The theory of turbocharging

Contents

2.1 Introduction

The purpose of supercharging is to increase the mass of air trapped in the cylinders of the engine, by raising air density. This allows more fuel to be burnt, increasing the power output of the engine, for a given swept volume of the cylinders. Thus the power to weight and volume ratios of the engine increase. Since more fuel is burnt to achieve the power increase, the efficiency of the engine cycle remains unchanged.

A compressor is used to achieve the increase in air density. Two methods of supercharging can be distinguished by the method used to drive the compressor. If the compressor is driven from the crankshaft of the engine, the system is called 'mechanically driven supercharging' or often just 'supercharging'. If the compressor is driven by a turbine, which itself is driven by the exhaust gas from the cylinders, the system is called 'turbocharging'. The shaft of the turbocharger links the compressor and turbine, but is not connected to the crankshaft of the engine (except on some experimental 'compound' engines, see Chapter 3). Thus the power developed by the turbine dictates the compressor operating point, since it must equal that absorbed by the compressor.

The essential components of the 'turbocharger' are the turbine, compressor, connecting shaft, bearings and housings. The advantage of the turbocharger, over a mechanically driven supercharger, is that the power required to drive the compressor is exctracted from exhaust gas energy rather than the crankshaft. Thus turbocharging is more efficient than mechanical supercharging. However the turbine imposes a flow restriction in the exhaust system, and therefore the exhaust manifold pressure will be greater than atmospheric pressure. If sufficient energy can be extracted from the exhaust gas, and converted into compressor work, then the system can be designed such that the compressor delivery pressure exceeds that at turbine inlet, and the inlet and exhaust processes are not adversely affected.

The process of compression raises temperature as well as pressure. Since the objective is to increase inlet air density, charger air coolers (heat exchangers) are often used to cool the air between compressor delivery and the cylinders, so that the pressure increase is achieved with the maximum rise in density.

Figure 2.1 shows the ideal dual combustion cycle of a diesel

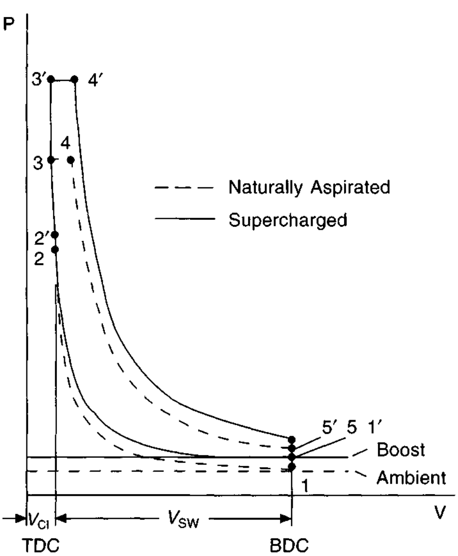

Figure 2.1 Comparison of supercharged and naturally aspirated air standard dual combustion cycles having the same compression ratio

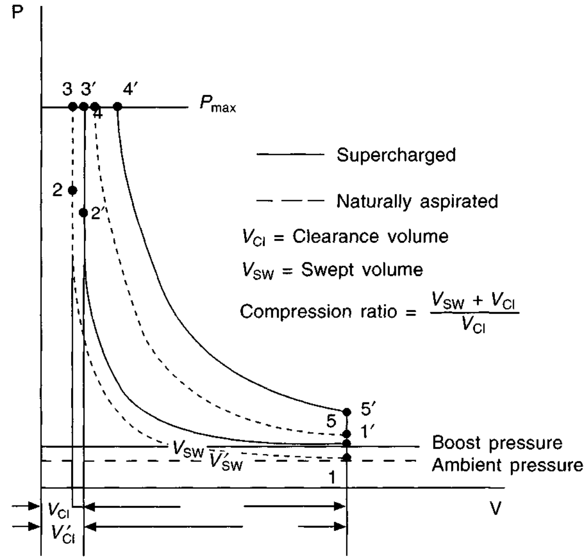

Figure 2.2 Comparison of supercharged and naturally aspirated air standard dual combustion cycle having same maximum pressure but different compression ratio

engine in naturally aspirated and turbocharged form. Since the inlet and exhaust pressures are above ambient, and more fuel is burnt in the engine, the cylinder pressure throughout the cycle, and particularly during combustion, is substantially higher for the turbocharged cycle. The compression ratio of the engine must be reduced to prevent an excessive maximum cylinder pressure being reached. *Figure 2.2* compares naturally aspirated and turbocharged ideal dual combustion cycles, when compression ratio is adjusted for the same maximum cylinder pressure. Since reducing compression ratio lowers cycle efficiency, and may make the engine difficult to start, there is a limit to how low a compression ratio can be used in practice.

Turbocharging increases power by increasing the work done per engine cycle. Thus brake mean effective pressure (b.m.e.p.) increases. *Figure 2.3* shows trends in b.m.e.p. for four-stroke and two-stroke engines, and the compressor pressure ratios used. The increase in b.m.e.p. in the 1960s occurred due to the widespread adoption of turbochargers on industrial, marine and rail traction engines. This trend continued in the 1960s as charge air cooling became more popular and engines were redesigned to accept higher compressor pressure ratios. Although these trends have slowed in recent years, b.m.e.p.'s are still increasing and many experimental engines have run at much higher values than those shown in *Figure 2.3*. It is inevitable that ratings will increase further in the search for lower manufacturing cost per horse-power. The alternative approach of increasing power output by increasing speed is unattractive, due to the rapid rise of mechanical and aerodynamic losses, and the corresponding fall in brake thermal efficiency.

Turbocharging of large two-stroke engines, four-stroke medium speed engines, and high speed truck and passenger car diesel engines all make different requirements on the turbocharger. In the following section, different types of turbochargers will be described, followed by a description of the different types of turbocharging systems used to deliver exhaust gas to the turbine. Later sections deal with charge air cooling and the application of turbochargers to the different classes of engine described above. Chapter 2 is restricted to analysis of conventional turbocharging systems. Non-conventional systems are presented in Chapter 3.

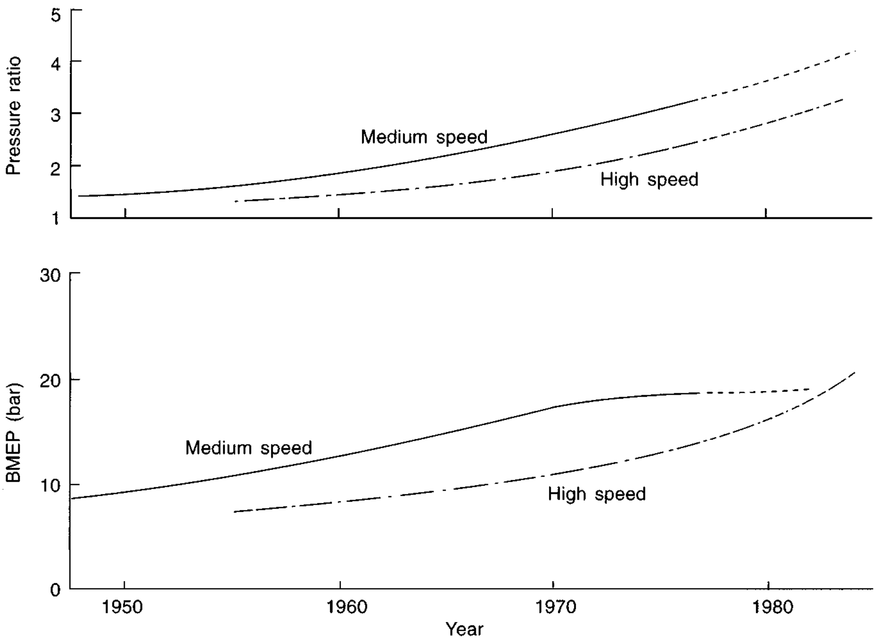

Figure 2.3 **(a)** Four-stroke diesel engine–increase in b.m.e.p. and compressor ratio

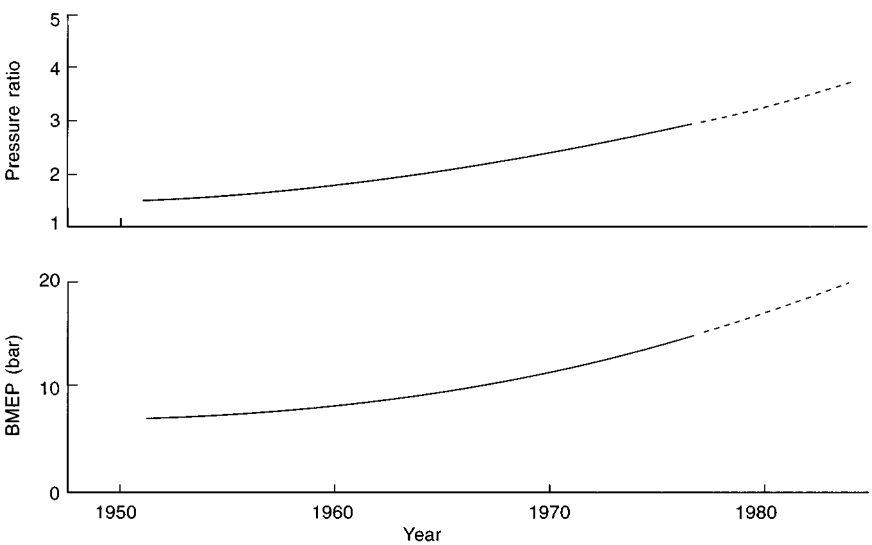

Figure 2.3 **(b)** Two-stroke diesel engine–increase in b.m.e.p. and compressor pressure ratio

2.2 Turbocharging

2.2.1 Turbochargers for automotive diesel engines

Turbochargers in this class are used for passenger car diesel engines rated at 45 kW upwards to larger special heavy truck and construction vehicles rated at up to 600 kW. The most important design factors are cost, reliability and performance. To keep cost low, the design must be simple, hence a single stage radial flow compressor, and a radial flow turbine are mounted on a common shaft with an inboard bearing system (*Figure 2.4*). This arrangement simplifies the design of inlet and exhaust casing and reduces the total weight of the turbocharger.

2.2.1.1 Compressor

The compressor impeller is an aluminium alloy (LM-16-WP or C-355T61) investment casting, with a gravity die-cast aluminium housing (LM-27-M). The design of the impeller is a compromise between aerodynamic requirements, mechanical strength and foundry capabilities. To achieve high efficiency, and minimum flow blockage, very thin and sharp impeller vanes are required, thickening at the root (impeller hub) for stress reasons. It is

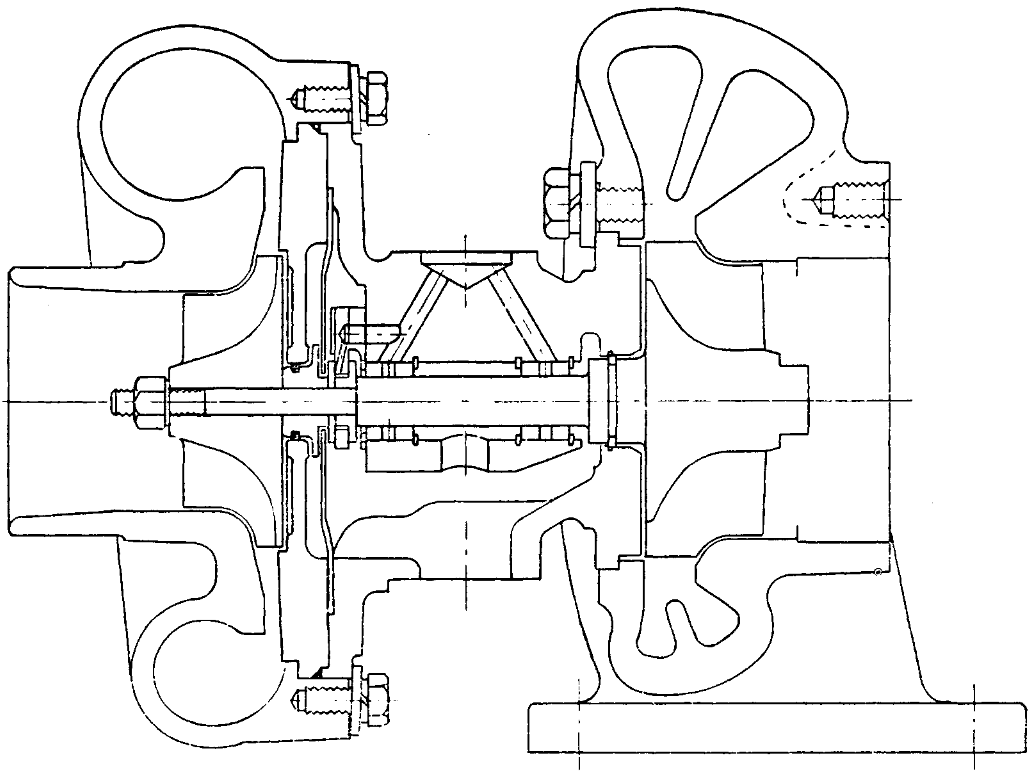

Figure 2.4 An automotive diesel engine turbocharger

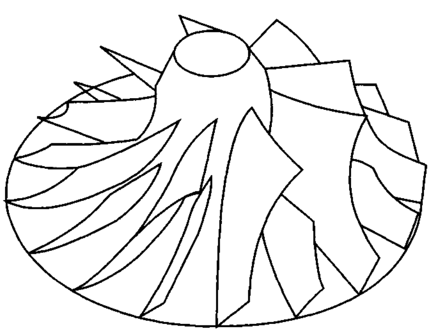

Figure 2.5 Automotive turbocharger compressor impeller, with splitter blades.

common practice to use splitter blades (*Figure 2.5*) that start part way through the inducer, in order to maintain good flow guidance near the impeller tip without excessive flow blockage at the eye. Until recently the impeller vanes have been purely radial so that blades were not subjected to bending stress. However most recent designs incorporate backswept blades at the impeller tip since this has been shown to give better flow control and reduces flow distortion transmitted through from impeller to diffuser.

Typical design point pressure ratios fall in the range of 2 to 2.5:1, requiring impeller tip speeds of 300 to 350 m/s, hence small units of typically 0.08 m tip diameter rotate at 72 000 to 83 000 rev/min. In order to match wide differences in air flow requirements from one engine to another, a range of compressor impellers is available to fit the same turbocharger. These will be produced from one or two impeller castings, but with different tip widths and eye diameters generated by machining as shown in *Figure 2.6*, and matched with appropriate compressor housings. Usually up to ten or more alternative 'trims' are available but since the impeller tip diameter is unchanged and the hub diameter at the impeller eye is fixed by the shaft diameter, the flow passage variations alter the efficiency as well as flow characteristics of the impeller.

The compressor can be a loose or slight interference fit on the shaft, clamped by the compressor end nut. Impellers of most turbochargers are balanced before assembly onto the shaft, so that components can be interchanged without rebalancing.

Vaneless diffusers (*Figure 2.4*) are used on all except very high pressure ratio compressors. Relative to the alternative vaned designs, the vaneless diffuser is slightly less efficient due to a longer gas flow path and poorer flow guidance, but has a substantially wider range of high operating efficiency. This is important in truck and passenger car applications where engine speed, and therefore mass flow range, is large. The volute acts not merely as a collector of air leaving the diffuser, but is usually designed to achieve a small amount of additional diffusion in its delivery duct. Generally the volute slightly overhangs the diffuser (*Figure 2.4*) in order to reduce the overall diameter of the turbocharger. The volute and impeller casing are invariably formed as a single component.

2.2.1.2 Turbine

Radial inflow turbines are universally used, usually friction or electron beam welded to the shaft. The turbine wheel must sustain the same high rotational speed of the compressor and operate at gas temperatures up to 900 K. The turbines are investment cast in high temperature creep resistant steels, such as 713 C Inconel. Its properties exceed the requirements but it

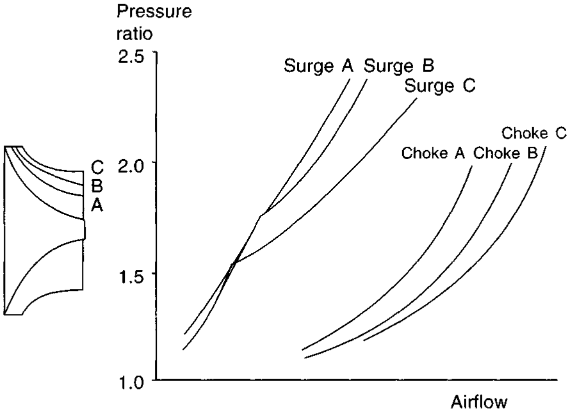

Figure 2.6 A range of compressor 'trims', machined from the same impeller casting

is a readily available material, from the gas turbine industry. The turbine housing must stand the same high gas temperature as the impeller but is not subject to rotating stress. However it should be strong enough to sustain a rotor burst. Sand or shell cast SG iron (spheroidal graphite nodular) is used in most applications, and is free of scaling at temperatures up to 900 K. Ni-resist is used for higher temperatures but is more expensive, and prone to cracking.

Vaneless stators are used except for a very few high pressure ratio applications. These are cheaper, and enable the gas angle at stator exit to vary to some extent with the mass flow, giving high efficiency over a very wide flow range. On the rare occasions when stator nozzles are incorporated in turbocharger turbines of this size, a nozzle ring is cast in Ni-resist or the nozzle vanes are fabricated from nickel-chromium alloys. The gas angle at rotor inlet is controlled by the nozzel vane angle or the geometry of the vaneless housing. In the latter case (*Figure 2.7*) the inlet scroll and vaneless stator are combined. The cross-sectional area of a section A defines the tangential gas velocity for a given mass flow rate and inlet density. The scroll is designed to spill the mass evenly around the circumference of the rotor. Again, for a given mass flow rate and density the radial component of velocity entering the impeller is fixed by the cross-sectional area at the rotor tip. This, and area A control the flow angle at rotor entry. Altering this flow angle alters the effective flow capacity of the turbine. Thus a range of casings with varying area A, or nozzle stator rings with varying blade angle, are available to match each rotor design to the requirements of a particular engine.

Gas flow through a turbine is predominantly accelerating, whereas in a compressor the gas diffuses. Fluid dynamic flow control is much easier to achieve in an accelerating flow, hence turbine design is less critical than compressor design. The turbine is more tolerant of mass flow variation, hence a single rotor may be used with a number of different area vaneless casings. Only a few turbine rotor trim variations are needed to cover the requirements of a large range of engine sizes.

Most turbine casings have twin entries as shown in *Figure 2.4*. The twin entries are used to separate the exhaust gas pulses coming from cylinders whose exhaust valves are open at the same time. This for example, the exhaust manifold of a typical six-cylinder engine will have two pipes, each connecting three cylinders to a twin entry turbine. Single entry casings are used on passenger car engines in order that a single waste-gate may be used to bypass some gas around the turbine at high engine

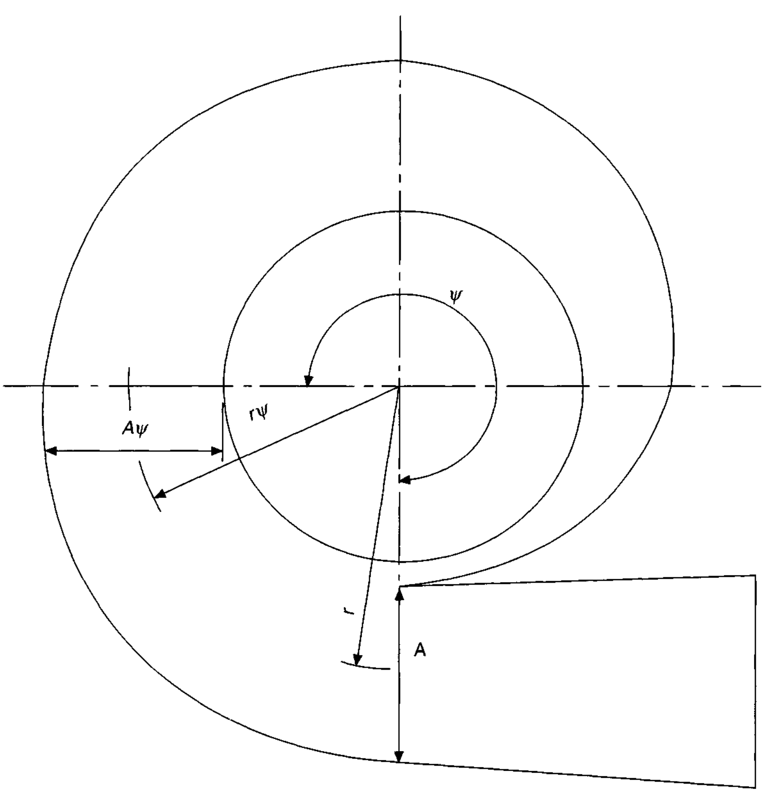

Figure 2.7 Geometric specification of a vaneless radial turbine housing (volute)

speeds. This reduces turbine and compressor power and hence boost pressure, avoiding the tendency for the turbocharger to overboost the engine at full speed and load (see section 2.6.7).

Since flow guidance is less critical than in the compressor, the turbine impeller has fewer blades. For stress and manufacturing reasons, the vanes are thicker than compressor vanes. They must withstand the pulsating gas pressure developed in the exhaust system of the engine. To reduce rotating inertia the back of the impeller is cut away between the blades (*Figure 2.4*). Although this slightly reduces efficiency, the benefit in terms of reduced stress and faster acceleration is substantial. Considerable rotor to housing clearance is essential, due to thermal effects, large thrust bearing clearance and the build-up of tolerances between components. Thus efficiencies are lower than those normal in gas turbine practice, due to leakage and clearance losses.

The turbine inlet flange mounts the turbine to the engine via the exhaust manifold. This avoids the need for expansion joints in the exhaust manifold.

2.2.1.3 Bearings and centre casing

For low cost, simplicity and ease of maintenance, the bearing system must be designed to use the lubricating oil of the engine. All automotive turbochargers use simple journal bearings, since ball bearings are more expensive, have a short life at very high speeds and are difficult to replace. The inboard (between compressor and turbine) bearing location imposes a short distance between bearings which, when combined with very light radial loads and a heavy overhung mass at one end only (the turbine), leads to complex bearing behaviour. In addition the very high rotational speed dictates that the rotor runs through both the first and second critical vibration frequencies of the rotating assembly. Thus bearing design is primarily concerned with the stability of the system.

Two fully floating sleeve bearings are used, free to rotate at a speed less than the rotor speed, and determined by bearing clearances. The outer oil film imparts an additional degree of damping into the bearing system, but both the shaft to sleeve to housing clearances are large for improved stability. Typical clearances are 0.02 to 0.05 mm between shaft and sleeve and 0.07 to 0.1 mm between sleeve and housing. These large clearances mean that oil filtration down to about 20 μm only is required; easily met by full flow paper engine oil filters. A large oil flow rate is required, due to large clearances and the need to cool the turbine end bearing. Bearing material is typically leaded bronze with added tin flashing.

Axial thrust loads are relatively light, except when an exhaust brake is used, upsetting the compressor-turbine pressure balance. A simple thrust assembly is located at the cooler, compressor end, with flat or tapered lands, made from sintered leaded bronze.

The power required to overcome bearing losses is quite significant, typically being 5 to 10% of turbine power at full speed. More important is the fact that this percentage increases at lower engine speeds, which is a disadvantage in vehicle applications where high boost pressure is difficult to achieve at low engine speed. Lower viscosity oil would help but is incompatible with the requirements of the engine.

The bearing sleeves run directly in a bore in the high grade grey iron centre casting of the turbocharger. This casting also acts as an oil drain and holds the compressor and turbine housings. For good oil drainage the casing must be installed with the oil inlet and drain at top and bottom, in the vertical position, with the rotor shaft horizontal. The oil seals at compressor and turbine ends are a difficult design problem, due to the need to keep frictional losses low, and to large movements due to large bearing clearance and adverse pressure gradients under some conditions. The piston ring seals shown in *Figure 2.8* are typical, the rings being a press fit in the housing. They act as a labyrinth seal due to small clearance between the stationary ring and the sides and base of the groove in the rotor shaft. In practice they leak unless care is taken to keep as much oil as possible away from them. Rotating flingers are designed into the shaft for this reason and an oil shield is frequently fitted at the compressor since a depression can be generated in the compressor when the engine is idling.

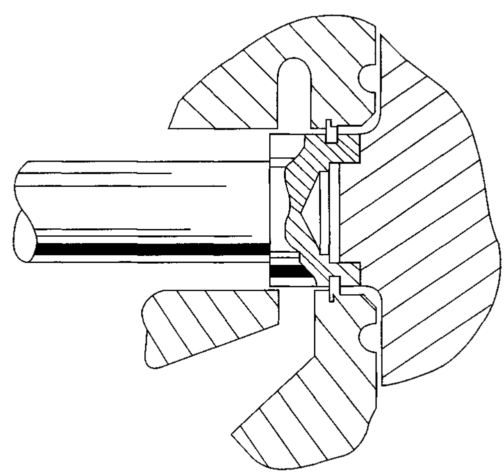

Figure 2.8 Piston ring oil seal, located in a stepped bore

The compressor and turbine housings can usually be rotated so that the compressor delivery and turbine inlet can be located in convenient positions. Set screws or stainless steel V-clamps are used to hold the components together.

The performance characteristics of typical small turbochargers are presented in section 2.3.3.

2.2.2 Small industrial and marine engine turbochargers

Turbochargers designed for small industrial and marine engines, though larger than those of large truck engines, are similar in concept to the automotive turbochargers described above. Radial flow compressor and turbines are used, with an inboard bearing arrangement (*Figure 2.9*). Apart from the larger size, they are required to have greater durability and higher efficiency. Thus the designs are usually more complex and expensive.

Engines designed for these applications operate over a smaller speed range than truck engines, and at greater b.m.e.p., hence higher compressor pressure ratio. It follows that the flow range required from the compressor is smaller, hence vaned diffusers are used. Vaned turbine stator nozzles are also used. This results in higher design point compressor and turbine efficiency. A range of diffuser nozzle angles and turbine stator blade angles are available for matching a basic turbocharger to a particular engine.

The maximum size is governed by precision casting limitations for the radial flow turbine rotor, currently about 300 mm, although most units in this class are smaller. Turbine housings are simple volutes designed to deliver the flow evenly around the circumference of the stator nozzle ring, the latter generating the design gas flow angle at rotor inlet. The turbine housings are supplied in uncooled or water cooled form. Although cooling is undesirable thermodynamically, it is sometimes required for safety reasons due to the potential danger of hot exposed surfaces in small engine rooms.

Bearings are of similar design to those of automotive units, except that clearances, relative to turbocharger size, are smaller. Sometimes cooling air is bled from the compressor to the rear

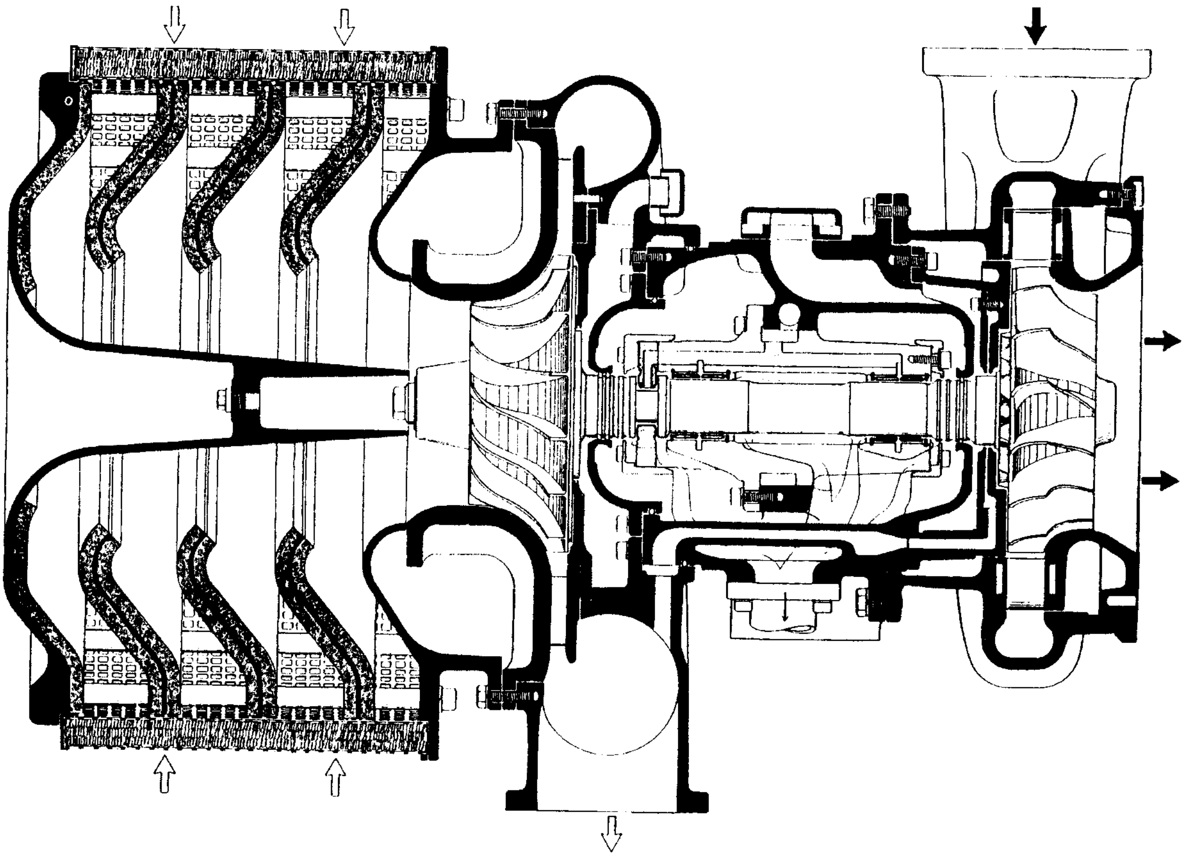

Figure 2.9 An industrial engine turbocharger with radial compressor and turbine (Brown Boveri RR series)

of the turbine hub and bearing area. This also helps prevent exhaust gas leaking down the back of the turbine wheel and reaching the bearings. These techniques help keep the hot end bearing cool, preventing serious oil oxidation deposits. Like the smaller units previously described, the lubricating oil system of the engine is also used for the turbocharger. Since bearing clearances are smaller, rotor movements are small and conventional labyrinth oil seals can be used at the compressor and turbine ends of the rotor shaft.

Turbochargers of this type are made in relatively small numbers, by batch production, hence their cost is high relative to automotive units.

2.2.3 Large industrial and marine engine turbochargers

These turbochargers are characterized by having axial flow, single stage, turbines and are fitted to the majority of large industrial and marine engines, both four- and two-stroke. The duty cycles of these engines are more arduous than that of automotive engines and they tend to spend much more of their operating time at high load. Furthermore the consequences of failure are more serious, particularly on a marine engine. As a result, although every attempt is made to keep the designs simple, the primary objectives are a very high level of reliability, high efficiency and versatility to cover a great range of engine types and sizes at reasonable cost. However, design variations from one manufacturer to another are greater than is the case with smaller turbochargers.

Figure 2.10 is a cross-section of a typical large turbocharger, with a radial flow compressor and axial flow turbine. The compressor impeller is made in two separate parts, the inducer and main part of the impeller. The inducer is usually machined from a steel casting or an aluminium forging, and is splined or keyed to the shaft. The impeller is machined from an aluminium forging except for very high pressure ratio requirements when titanium is used due to its superior high temperature properties. The advantage of the two piece compressor is ease of machining, but an additional benefit is some impeller vane damping provided by friction at the inducer-impeller contact surfaces. Compressor diffusers are vaned for high efficiency.

The turbine disc is machined either as an integral part of the shaft or is shrunk on to the shaft. The rotor blades may be cast, forged or machined from a high temperature creep-resistant steel such as Nimonic 80A or 90. Welded joints or 'fir-tree' roots are used to fix them to the disc, the latter design being more common on high pressure units since they provide a degree of vibration damping and allow a wider selection of blade and disc materials to be considered. Additional vibration damping can be provided by wire lacing the blades. The turbocharger manufacturer will offer a range of 'trims' or flow capacities with each basic design of turbocharger by varying blade (stator and rotor) height and stator blade angle.

A disadvantage of the axial flow turbine is that it complicates the design of the gas inlet and outlet. The gas inflow section is particularly important hence this is usually located on the end, allowing generous curvature in the inlet ducts to the stator blades for minimum flow distortion and loss. The turbine exit duct acts merely as a collector, hence a compact design can be used,

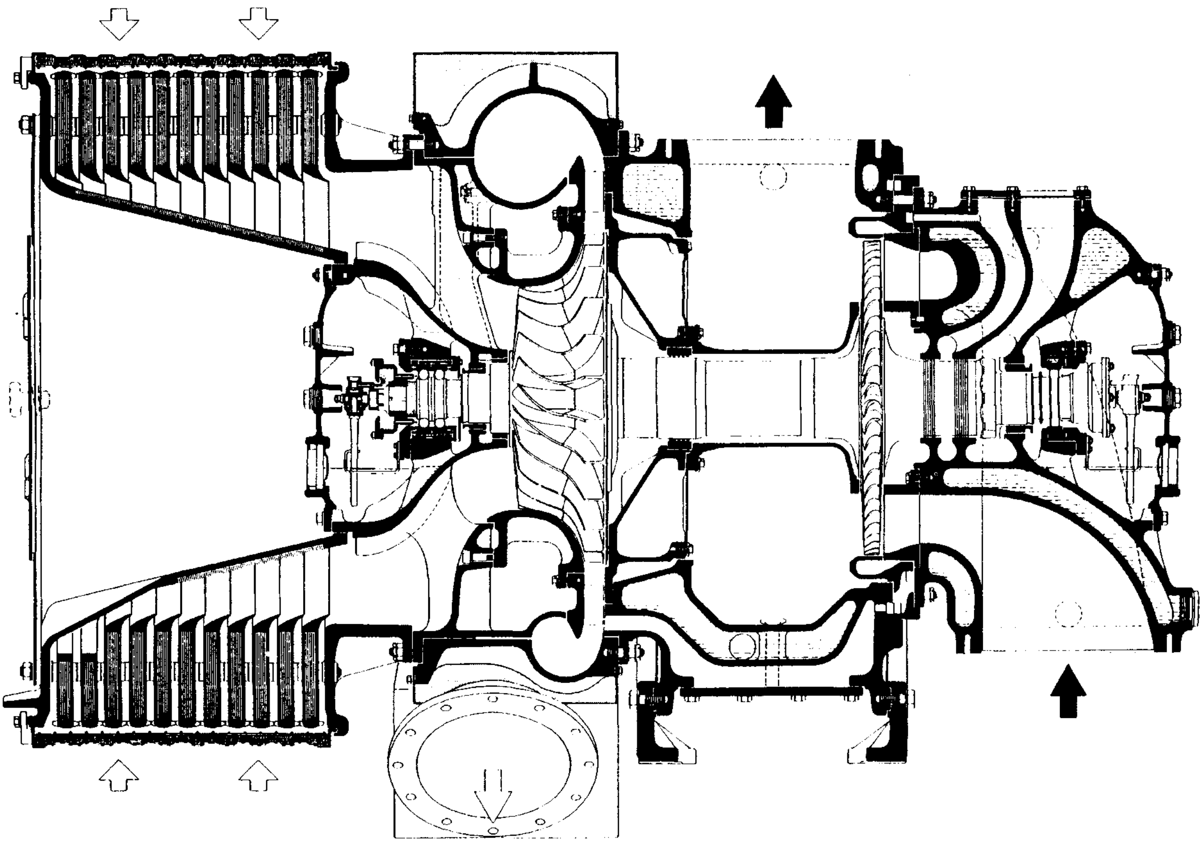

Figure 2.10 A large turbocharger with radial compressor and axial turbine (Brown Boveri)

minimizing turbocharger length. However, a recent trend is to utilize some exhaust diffusion to increase turbine expansion ratio and power output.

Most of the larger turbochargers in this class have outboard rolling element bearings (i.e. outside the compressor and turbine, *Figure 2.10*), with their own oil supply, and resilient mountings to prevent brinelling. The advantages of this are stable shaft mounting and low dynamic loads due to the wide bearing spacing, small bearing diameter, low rolling resistance and good access for bearing maintenance. The use of separate oil supplies for the turbocharger and engine enables a lower viscosity oil to be used, further reducing bearing friction. Low pressure ratio turbochargers use simple rotating steel discs, partially immersed in the oil, to pick up and deliver the oil to the bearings, but with higher bearing loads and speeds, gear pumps are used to spray oil on the bearings. Plain or sleeve bearings are sometimes available as an option and are preferred for durability although their frictional losses are greater.

Turbocharger design is simpler with inboard bearings since this gives greater freedom to design low loss intake ducts. Fewer components are required and the turbocharger is shorter, lighter and cheaper as a result (*Figure 2.11*). The disadvantage is a less stable bearing system and higher bearing loads. Fully floating sleeve and multi-lobe plain bearings are used, with well damped mountings for stability; the rotors must still be carefully balanced. Relative to rolling element bearings, higher oil pressure and greater oil flow rates are required and the combination of large diameter and width means that frictional losses are greater.

With either bearing system, the turbine outlet casing is the main structure to which the other components are bolted, and incorporates mountings to the engine. The casing is usually water cooled. Bolted to it is the water cooled turbine inlet casing, incorporating the bearing housing (for outboard bearings) and its oil reservoir. Single, two, three and four entry turbine inlets are available, manufactured from high grade cast iron. Between turbine inlet and outlet casings, provision is made for mounting the turbine stator nozzle ring. The compressor inlet and outlet casings are aluminium alloy castings.

The compressor inlet casing incorporates webs to support the bearing housing if outboard bearings are used. These webs must be carefully designed to be far enough away from the impeller to avoid impeller vane excitation. The casing also houses a combined air filter and silencer on most larger turbochargers (*Figure 2.10*). Sound waves originating at the compressor intake are reflected and reduced in intensity by baffles lined with sound absorbing material.

2.3 Turbocharger performance

The performance of turbochargers can be defined by the pressure ratio, mass flow rate and efficiency characteristics of the compressor and turbine, plus the mechanical efficiency of the bearing unit. In this section we will look at the efficiency of compressors and turbines leading into a description of typical turbocharger performance maps.

2.3.1 Compressor and turbine efficiency

The work output from (or input to) a turbomachine can be found from the first law of thermodynamics. From this law the steady

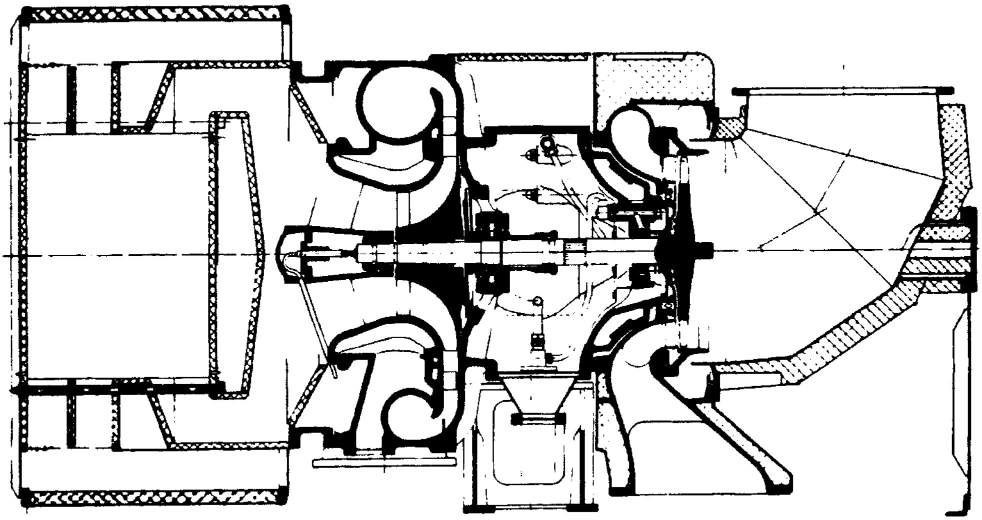

Figure 2.11 A large turbocharger with axial turbine but inboard bearings (M.A.N.)

flow energy equation may be derived. A turbomachine has one inlet and one outlet port. The steady flow energy equation becomes:

$$\dot{Q} - \dot{W} = \dot{m}[(h_2 + KE_2 + PE_2) - (h_1 + KE_1 + PE_1)] \qquad (2.1)$$

where $\dot{Q}$ = heat transfer rate (+ ve to the system);
$\dot{W}$ = work transfer rate (+ ve by the system);
$\dot{m}$ = mass flow rate;
h = specific enthalpy;
KE = specific kinetic energy;
PE = specific potential energy;
suffixes 1, 2 = inlet and outlet ports respectively.

Denoting the stagnation enthalpy (h_0) as

$$h_0 = h + KE \qquad (2.2)$$

and neglecting changes in potential energy and heat transfer, since these terms are small, this becomes

$$-\dot{W} = \dot{m}[(h_{02} - h_{01}) \qquad (2.3)$$

Both air and exhaust gas are considered as perfect gases. Hence they obey the equation of state.

$$Pv = RT \qquad (2.4)$$

where P, v, R and T denote pressure (absolute), specific volume, gas constant and temperature respectively. The specific heat capacity at constant pressure (Cp) for a perfect gas is given by:

$$Cp = dh/dT \qquad (2.5)$$

Thus eqn (2.3) becomes

$$-\dot{W} = \dot{m}Cp[(To_2 - To_1) \qquad (2.6)$$

where To denotes stagnation (or 'total') temperature, the temperature of a gas if brought to rest. Relative to the free stream temperature (T) of a gas moving at velocity V,

$$To = T + V^2/2Cp \qquad (2.7)$$

The second law of thermodynamics tells us that specific entropy is related to the specific heat transfer

$$ds > dQ/T \qquad (2.8)$$

The second law can also be used to show that ideal adiabatic compression or expansion takes place at constant entropy.

One definition of the efficiency of a compressor is the power required for ideal, adiabatic compression divided by the actual power required in a non-ideal, non-adiabatic compressor, working with the same inlet pressure and temperature and outlet pressure.

$$\eta_{is} = \frac{\text{isentropic power}}{\text{actual power}}$$

Hence η_{is} is termed the isentropic efficiency of the compressor. From eqns (2.3), (2.6) and (2.8),

$$\eta_{isTT} = \frac{ho_{2_{is}} - ho_1}{ho_2 - ho_1} \qquad (2.9)$$

and

$$\eta_{isTT} = \frac{To_{2_{is}} - To_1}{To_2 - To_1} \qquad (2.10)$$

where suffices 'is' and 'TT' denote 'isentropic' and 'total to total', meaning an efficiency based on total temperature values. Note that the work required by a non-ideal compressor exceeds that of an isentropic compressor, hence the exit air temperature To_2 is higher than $To_{2_{is}}$.

For isentropic compression, pressure and temperature are related by the expression

$$\frac{Po_2}{Po_1} = \left(\frac{To_{2_{is}}}{To_1}\right)^{\gamma/(\gamma-1)} \quad \text{where } \gamma = Cp/Cv \qquad (2.11)$$

Hence eqn (2.10) may be rearranged as

$$\eta_{isTT} = \frac{(Po_2/Po_1)^{(\gamma-1)/\gamma} - 1}{\frac{To_2}{To_1} - 1} \qquad (2.12)$$

An evaluation of compressor efficiency based on total-to-total temperature rise, implicity assumes that the kinetic energy leaving the compressor can be made use of in the following machine components. This is true in a gas turbine since the gas velocity is maintained through the combustion chamber to the turbine, where it does useful work. However, air delivered from the turbocharger compressor to the inlet manifold of an engine is brought almost to rest, without doing useful work. This loss of kinetic energy should be considered as a loss of compressor efficiency relative to the ideal of a negligible exit gas velocity. At inlet to the compressor, air is accelerated from rest, into the compressor eye without introducing inefficiencies, hence the inlet ambient temperature can be used (total and static temperatures are equal). Thus a more appropriate definition of compressor efficiency is

$$\eta_{isTS} = \frac{(Po_2/Po_1)^{(\gamma-1)/\gamma} - 1}{To_2/To_1 - 1} \tag{2.13}$$

This is the total-to-static isentropic efficiency and will usually be a few percentage points lower than the total-to-total isentropic efficiency. Unfortunately some turbocharger manufacturers quote total-to-total values, some without declaring the basis of their measurement.

Manipulation of eqns (2.6), (2.8) and (2.11) gives the following relationships for compressor power ($\dot{W}$)

$$-\dot{W}_c = \dot{m}CpTo_1\left[\left(\frac{Po_2}{Po_1}\right)^{(\gamma-1)/\gamma} - 1\right]/\eta_{isTS} \tag{2.14}$$

The negative sign results purely from the thermodynamic sign convention of work being done by the system being considered positive, and work done on the system as negative. Thus the power required to drive the compressor is a function of the mass flow rate ($\dot{m}$), inlet air temperature, (To_1), pressure ratio (Po_2/Po_1), compressor efficiency (η_{is}) and specific heat at constant pressure. Equations (2.13) and (2.14) show that low compressor efficiency not only increases the power requirement for a given pressure ratio, but also increases the delivery temperature (T_2) and therefore reduces the air density leaving the compressor. It is important to achieve high compressor efficiency for both reasons.

The isentropic efficiency of the turbine may be expressed as the actual power output divided by that obtained from an ideal adiabatic (isentropic) turbine operating with the same inlet pressure and temperature.

$$\eta_{is} = \frac{\text{actual power}}{\text{isentropic power}} \text{ (turbine)} \tag{2.15}$$

This expression may be developed in a similar manner to the compressor to give

$$\eta_{isTT} = \frac{1 - To_4/To_3}{1 - (Po_4/Po_3)^{(\gamma-1)/\gamma}} \tag{2.16}$$

and

$$\eta_{isTS} = \frac{1 - To_4/To_3}{1 - (Po_4/Po_3)^{(\gamma-1)/\gamma}} \tag{2.17}$$

Kinetic energy leaving the turbine is wasted through the exhaust pipe, hence the total-to-static efficiency is again most appropriate, although not always quoted by the turbocharger manufacturer.

The power output of the turbine is given by:

$$\dot{W}_t = \dot{m}CpTo_3\left[1 - \left(\frac{P_4}{Po_3}\right)^{(\gamma-1)/\gamma}\right]\eta_{isTS} \tag{2.18}$$

Thus the power developed by the turbine is a function of its inlet temperature (To_3) mass flow rate ($\dot{m}$), expansion ratio (P_4/Po_3), efficiency (η_{isTC}) and specific heat capacity of the exhaust gas (Cp).

2.3.2 Non-dimensional representation of compressor and turbine characteristics

The mass flow rate, efficiency and temperature rise ($\Delta T = T_{02} - T_{01}$) of a compressor or turbine can be expressed as a function of all possible influencing parameters, as follows:

$$\dot{m} = \eta_{is}, \Delta T = f(Po_1, Po_2, To_1, N, D, R, \gamma, \mu) \tag{2.19}$$

where N, D and μ are rotational speed, diameter and the kinematic viscosity of the gas respectively, and $\gamma = Cp/Cv$. These can be reduced, using dimensional analysis, to the following non-dimensional groups:

$$\frac{\dot{m}\sqrt{To_1 R}}{Po_1 D^2}, \eta_{is}, \frac{\Delta T}{To_1} = f\left(\frac{ND}{\sqrt{RTo_1}}, \frac{Po_2}{Po_1}, \frac{\dot{m}}{\mu D}, \gamma\right) \tag{2.20}$$

For the compressor, the value of γ remains constant, except for a very small variation with temperature, hence the last term may be ignored. γ does vary with air-fuel ratio, but its influence on turbine performance is small and is therefore also ignored. Fortunately the Reynolds number ($\dot{m}/\mu D$) also has only a small effect on performance and can be ignored. A relationship between η_{is}, $\Delta T/To_1$ and Po_2/Po_1 has already been given (eqns 2.12 and 2.16), hence eqn (2.20) may be reduced to

$$\frac{\dot{m}\sqrt{To_1 R}}{Po_1 D^2}, \eta_{is} = f\left(\frac{ND}{\sqrt{RTo_1}}, \frac{Po_2}{Po_1}\right) \tag{2.21}$$

(For a turbine, suffixes 3 and 4 replace 1 and 2).

For a particular turbocharger, the diameter remains constant and, for turbocharger applications the gas constant remains fixed, hence the variation in performance with running conditions is given by

$$\frac{\dot{m}\sqrt{To_1}}{Po_1}, \eta_{is} = f\left(\frac{N}{\sqrt{To_1}}\right), \frac{Po_2}{Po_1} \tag{2.22}$$

The complete performance map of the turbomachine (*Figure 2.12*) can be shown by plotting a graph of $m\sqrt{To_1}/Po_1$ against

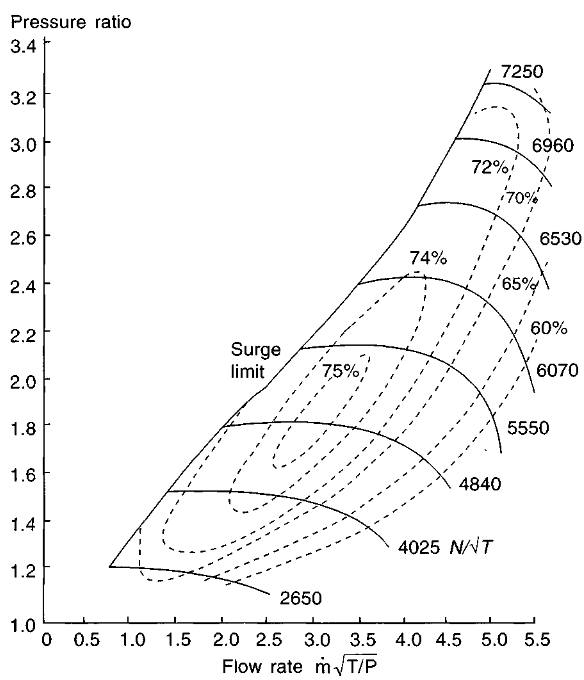

Figure 2.12 Turbocharger compressor performance map

Po_2/Po_1 showing lines of constant $N/\sqrt{To_1}$ and efficiency (η_{is}). The advantage of this presentation is that it uniquely describes the performance of the turbomachine, regardless of inlet conditions (pressure and temperature). However, the terms in eqn (2.22) are no longer truly dimensionless.

2.3.3 Compressor performance

A typical performance map, from a turbocharger compressor designed for a medium speed engine, is shown in *Figure 2.12.* The central area is the stable operating zone, bounded by the surge line on the left (low mass flow rates), and a regime of high rotational speed and low efficiency on the right (high mass flow rate).

A detailed explanation of the causes of surge has yet to be fully accepted, but it is clear that when the mass flow rate through the compressor is reduced whilst maintaining a constant pressure ratio, a point arises at which local flow reversal occurs in the boundary layers. Complete flow reversal can develop in a 'stall cell' somewhere in the compressor, with normal flow elsewhere. Once several stall cells have developed the complete flow pattern can break down hence mass flow and pressure fall. A stable flow pattern becomes re-established at a lower pressure ratio, allowing the mass flow to build up again to the initial value. The flow instability repeats in a surge cycle. The compressor must not be asked to work in this region of operation but it must be realized that surge is influenced by the complete intake, compressor and inlet manifold system. Thus the surge line drawn by the turbocharger manufacturer is only a guide, and in practice it varies from one engine installation to another.

The area of high rotational speed and low efficiency is a result of choking of the limiting flow area in the compressor. Extra mass flow can only be achieved by increasing rotational speed, which must be limited by stress constraints. If the diffuser chokes, rather than the rotor, then compressor speed eventually rises substantially with little increase in airflow. This is likely to be the case with a vaned diffuser, as fitted to the compressor whose characteristics are shown in *Figure 2.12.*

Constant efficiency loops are also shown in *Figure 2.12.* Note that these tend to be parallel to the surge line.

2.3.4 Turbine performance

An axial flow turbine characteristic based on the same pressure ratio versus mass flow parameter is shown in *Figure 2.13.* The most evident feature is the way that the lines of constant speed parameter ($N/\sqrt{To}$) converge to a single line of almost constant mass flow parameter. This flow limit is caused by the gas reaching sonic velocity and choking the inlet casing or stator nozzle blades. This choked flow will remain constant (for constant inlet conditions), regardless of rotor speed. At pressure ratios below the choking condition, the effective flow area of the turbine, and hence mass flow rate, will be influenced by the rotor. Thus rotational speed influences mass flow rate unless the stator chokes.

In a radial flow turbine, rotational speed of the rotor influences the pressure at stator exit, due to centrifugal effects. Thus the overall pressure ratio (stator inlet to rotor exit) at which the stator chokes is dependent on rotor speed. *Figure 2.14* shows pressure ratio versus mass flow rate curves for a radial flow turbine, with stator nozzles, illustrating the variation of choked pressure ratio and mass flow rate with rotor speed.

Nozzleless radial turbines exhibit the widest variation of mass flow with rotor speed. However, it is rare for the data obtained by the turbocharger manufacturer to cover the full pressure ratio versus mass flow range along each constant speed line, even for these turbines. It reduces test time and is rarely required

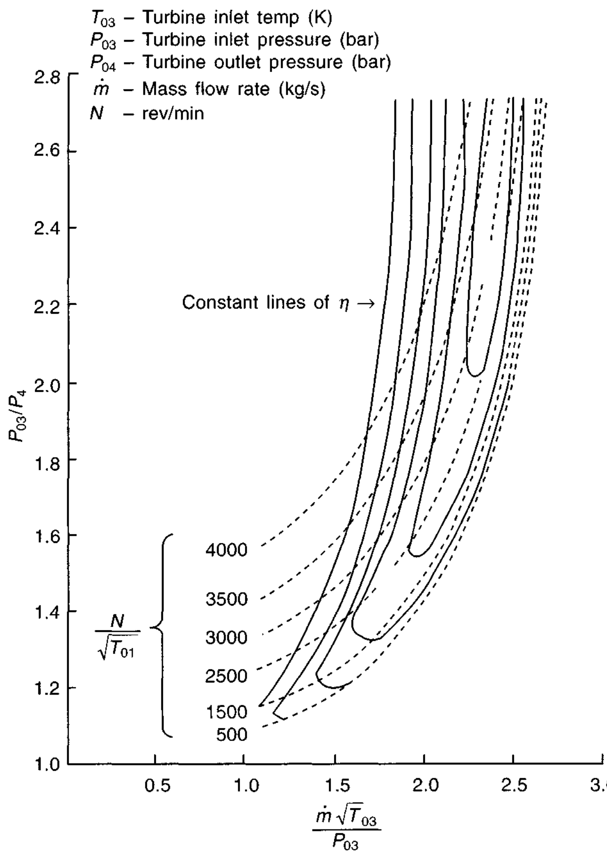

Figure 2.14 Radial flow turbine performance map

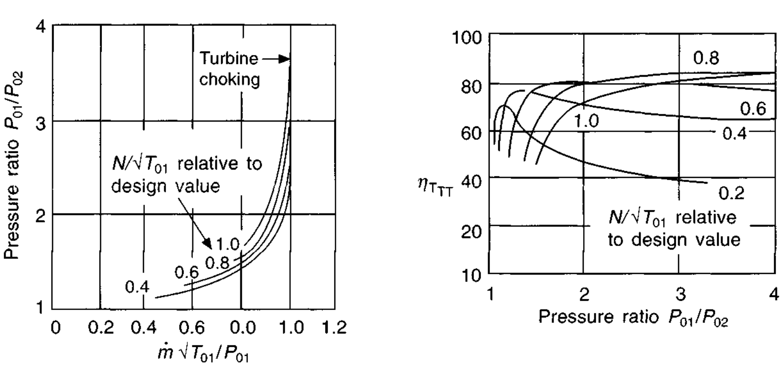

Figure 2.13 Axial flow turbine performance map

by the user, for the following reason. For a given turbocharger, it is possible to link the compressor and turbine characteristics and plot the equilibrium running line on compressor and turbine maps. This comes from balancing the compressor and turbine power eqns (2.14 and 2.18), and imposing a common rotational speed, factors that are implicit in any experimental turbocharger test. This equilibrium line is shown in *Figure 2.15* and runs across from a low speed line at low pressure ratio to a high speed line at high pressure ratio. The equilibrium line is based on steady-state testing and theory, and is not necessarily the operating regime if the turbine inlet conditions are pulsating.

All turbocharger manufacturers present compressor data in the form described in section 2.3.3, but there is less uniformity of turbine data. For example, some manufacturers follow gas turbine practice and transpose the ordinate and abscissa of *Figures 2.13* and *2.14* to those of *Figure 2.15*. Further variations occur in presentation of turbine efficiency data. The problem arises from the fact that the operational area of the turbine occupies such a restricted area on the pressure ratio versus mass flow parameter map. It is possible, but inconvenient, to superimpose lines of constant efficiency with a radial turbine (*Figure 2.14*), but quite impossible for an axial flow turbine. It is simpler to present efficiency on a separate diagram, but different manufacturers tend to use different diagrams.

Most common is a plot of turbine efficiency against blade speed ratio (u/c). u is the tip speed of a radial turbine rotor or the blade velocity at the mean blade height of an axial rotor. The rotor blade velocity (u) is non-dimensionalized by dividing by a theoretical velocity (c) that would be achieved by the gas if it expanded isentropically from the turbine inlet condition to the turbine exit pressure. Lines of constant pressure ratio or speed parameter ($N\sqrt{To}$) are superimposed on the map (*Figure 2.16*). This presentation happens to be useful for the turbocharger

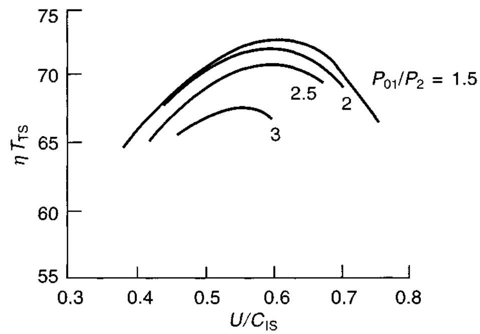

Figure 2.16 Turbine efficiency map

manufacturer when matching compressor and turbine rotor diameters but is inconvenient for an engine manufacturer. He desires a graph whose axes represent parameters relevant to his engine. Thus some turbocharger manufacturers plot turbine efficiency against pressure ratio, showing lines of constant turbocharger speed (*Figure 2.15*).

In practice diagrams such as *Figures 2.15* and *2.16* show the product of turbine and mechanical efficiency (bearing losses predominate) since it is difficult to separate them. Indeed it is surprisingly difficult to achieve accurate turbine efficiency measurements, due to heat transfer, non-uniform and unsteady flow effects, etc. As a result most turbocharger manufacturers have developed their own standard test techniques whose results can be reliably compared across their range of turbochargers. However these results cannot be directly compared with those of another manufacturer who, for example, tests with a different standard turbine inlet temperature and therefore measures a

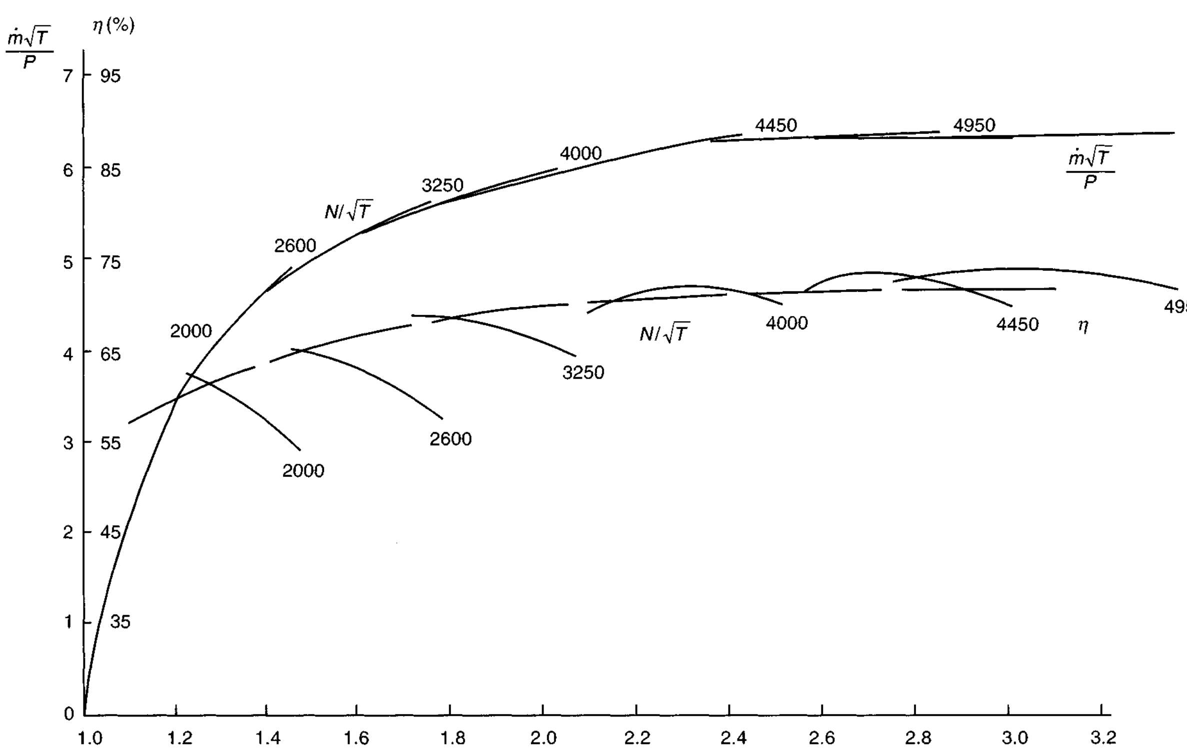

Figure 2.15 Turbine performance map, alternative presentation

different isentropic efficiency due to different heat transfer. As a result several turbocharger manufacturers do not reveal their turbine efficiency data since unrepresentative comparisons with those of another manufacturer could easily be made by a user.

2.4 Turbocharging systems–principles

Successful design of a turbocharged diesel engine is highly dependent on the choice of system for delivering exhaust gas energy from the exhaust valves or ports, to the turbine, and its utilization in the turbine.

Virtually all the energy of the gas leaving the cylinders arrives at the turbine. Some is lost on the way, due to heat transfer to the surroundings, but this is unlikely to exceed 5% unless water cooled exhaust manifolds are used, and will usually be much less. However, the design of the exhaust manifolds between the exhaust valve and turbine influence the proportion of exhaust gas energy that is available to do useful work in the turbine. An important parameter is the pressure in the exhaust system. Equation (2.18) shows that turbine power increases with pressure ratio (Po_3/P_4), hence the exhaust manifold pressure should be high. However, this implies that the pistion has to push the combustion products out of the cylinder against a high 'back-pressure', reducing the potential power output of the engine. Various turbocharging systems have been proposed to rationalize these apparently conflicting requirements. The most commonly used will be described herein. More complex systems that have been developed for special purpose applications are described in Chapter 3.

2.4.1 The energy in the exhaust system

The ideal thermodynamic cycles of engine operation were presented in Chapter 1. *Figure 2.17* shows the energy potentially available in the exhaust system, with an ideal cycle. The exhaust valve opens at BDC, point 5, where the cylinder pressure is much greater than the ambient pressure at the end of the exhaust pipe. If the contents of cylinder at EVO were somehow allowed to expand isentropically and reversibly down to the ambient pressure (to point 6), then the work that *could* be done is represented by the cross-hatched area 5-6-1. This work could be recovered by allowing the piston to move further than normal as shown in *Figure 2.17*. However, this requires an engine with an exceptionally long stroke and in practice it is found that the additional piston friction offsets the work gained by an ultra-long expansion stroke.

The work represented by area 5-6-1 is therefore potentially available to a turbocharger turbine placed in the exhaust manifold. It is called the 'blow-down' energy, since it involves the combustion products being 'blown-down' from cylinder pressure at point 5 to atmospheric pressure at point 6, when the exhaust valve opens. *Figure 2.17* represents a naturally aspirated engine. Consider now an ideal turbocharged four-stroke engine, as shown in *Figure 2.18*. Turbocharging raises the inlet manifold pressure, hence the inlet process (12–1) is at pressure P_1, where P_1, is above ambient pressure P_a. The 'blow-down' energy is represented by area 5-8-9. The exhaust manifold pressure (P_7) is also above the ambient pressure P_a. The exhaust process from the cylinder is represented by line 5, 13, 11, where 5, 13 is the 'blow-down' period when the exhaust valve opens and high pressure gas expands out into the exhaust manifold. Process 13, 11 represents the remainder of the exhaust process, when the piston moves from BDC to TDC displacing most of the gas from the cylinder to exhaust manifold. This gas is above ambient pressure and therefore also has the potential to expand down to ambient pressure whilst doing useful work. The potential work

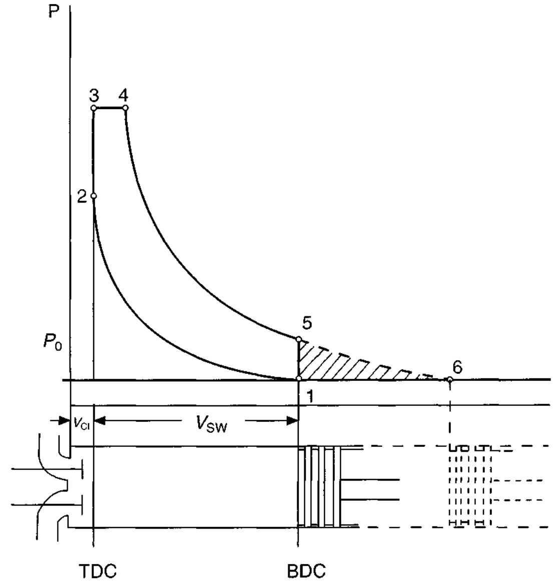

Figure 2.17 Ideal limited pressure cycle–naturally aspirated

that could be done is represented by the cross-hatched area 13-9-10-11. This work is done by the piston but could be recovered by a turbine in the exhaust. It will be called the piston pumping component of exhaust energy.

The maximum possible energy available to drive a turbine will be sum of areas 5-8-9 and 13-9-10-11, but it is impossible to devise a practical system that will harness all this energy. To achieve this, the turbine inlet pressure must instantaneously rise to P_5 when the exhaust valve opens, followed by isentropic expansion of the exhaust gas through P_7 to the ambient pressure ($P_8 = P_a$). During the displacement part of the exhaust process, the turbine inlet pressure would have to be held at P_7. Such a series of processes is impractical.

Consider a simpler process that would occur if a larger chamber were fitted between the engine and turbine inlet in order to damp down the pulsations in exhaust gas flow. The turbine acts as a flow restrictor creating a constant pressure (P_7) in the exhaust manifold chamber. The available energy at the turbine is given by area 7-8-10-11. This is the ideal 'constant pressure turbocharging system'. Next consider an alternative system, in which a turbine wheel is placed directly downstream of the engine, very close to the exhaust valve. The gas would expand directly through the turbine along line 5-6-7-8, assuming isentropic expansion and no losses in the exhaust port. If the turbine were sufficiently large, both cylinder and turbine inlet pressure would drop to equal ambient pressure before the piston has moved significantly from BDC. Thus the piston pumping work would be zero during the ideal exhaust stroke and area 5-8-9 represents the available energy at the turbine. This is the ideal 'pulse turbocharging system'.

In practice the systems commonly used and referred to as constant pressure and pulse systems are based on these principles but are far from ideal. They will be described below.

Although the diagrams illustrating the energy available at the turbine are based on the four-stroke engine cycle, similar diagrams can be constructed for two-stroke engines. Apart from the change in valve or port timing, the work done by the piston is replaced by energy transferred from the compressor to scavenge air. For clarity, scavenge in the overlap period in the case of a four-stroke engine, has been ignored.