10

Turbocharger Matching

10.1 Introduction

Naturally aspirated diesel engines are capable of operating over wide speed ranges. The maximum useful speed will usually be limited by poor volumetric efficiency, the inertia of the reciprocating parts or, in the case of some small high-speed engines, high frictional losses and poor combustion. An engine that is designed for variable speed operation will usually exhibit some deterioration in performance both at extreme low and high speeds. This is due to high gas frictional losses in the inlet valves and the use of valve timing optimised in the mid-speed range and a gradual mismatch between fuel injector characteristics and swirl. However, the useful speed range can be wide, since reciprocating machinery is well suited to cater for a wide range of mass flow rate.

The performance of turbomachines is very dependent on the gas angles at entry to the impeller, diffuser and turbine rotor. The blade angles are set to match these gas angles, but a correct match will only be obtained when the mass flow rate is correct for a specified rotor speed. Away from this 'design point' the gas angle will not match the blade angle and an incidence loss occurs due to separation and subsequent mixing of high and low-velocity fluid. These losses will increase with increasing incidence angle, hence turbomachines are not well suited for operation over a wide flow range. Their use as superchargers is due to their high design point efficiency and their ability to 'accept' high mass flow rates through small machines.

It is clear that a turbomachine is not ideally suited to operate in conjunction with a reciprocating machine, hence the combination of diesel engine and turbocharger must be planned with care. 'Matching' of the correct turbocharger to a diesel engine is of great importance and is vital for successful operation of a turbocharged diesel engine. The over-all objective of turbocharger matching is to fit a turbocharger with the most suitable characteristics to an engine in order to obtain the best over-all performance from that engine. The turbocharger will not be operating at its high efficiency flow area over the complete working range of the engine, if the latter is large (as in the case of an automotive engine). It follows that it is possible to 'match' the turbocharger correctly only at a particular point in the operating range of the engine. For example, if an engine is required to run for most of its life at constant speed and full load, the turbocharger will be chosen such that its high-efficiency operating area coincides with the pressure ratio and mass flow requirements of the engine at that condition. If the engine is required to operate over a broad speed and load range, then a compromise must be made when matching the turbocharger. This compromise will principally be governed by the duty for which the engine is required.

The basic size of the turbocharger will be determined by the quantity of air required by the engine. This will be a function of swept volume, speed, rating (or boost pressure), density of air in the inlet manifold, volumetric efficiency and scavenge flow. If these parameters are known an initial estimate of the air mass flow rate may be made: for a four-stroke engine

$$\dot{m} \approx \frac{N}{2} \times V_{sw} \times \rho_m \times \eta_{vol} \tag{10.1}$$

With little valve overlap, the volumetric efficiency will be less than unity and may be estimated from values obtained under naturally aspirated operating conditions or from previous experience. With large valve overlap, the clearance volume will be scavenged and some excess air will pass into the exhaust.

$$\eta_{vol} = f(\eta_{vol*}\,;\,\frac{CR}{CR-1}\,;ADR)$$

where η_{vol*} = volumetric efficiency, naturally aspirated. For a two-stroke engine (with scavenge pump assistance)

$$\dot{m} = N \times V_{sw} \times \rho_m \times \eta_{vol} \tag{10.2}$$

where V_{sw} is now the swept volume of the scavenge pump.

The density of the air in the inlet manifold will be given by

$$\rho_m = \frac{P_m}{RT_m} \approx \frac{P_2}{RT_2} \tag{10.3}$$

The boost pressure (P_m) will have been estimated for the engine to produce its target power output, subject to expected thermal and mechanical stresses. By assuming a value for the isentropic efficiency of the compressor (total-to-static) the boost temperature (T_m) may then be estimated

$$T_m = T_a + \frac{T_a}{\eta_c}\left[\left(\frac{P_m}{P_a}\right)^{(\gamma-1)/\gamma} - 1\right] \approx T_1 + \frac{T_1}{\eta_c}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right] \tag{10.4}$$

The isentropic efficiency of the compressor may be taken from compressor maps. If the compressor efficiency is quoted on the total-to-total basis, a correction of a few percentage points down on the total value is required.

For an engine using aftercooling the boost temperature (T_m) has to be reduced according to the equation

$$T_m = T_2(1 - \epsilon) + \epsilon T_w \tag{10.5}$$

From equations 10.1 to 10.5 and known values of ambient temperature and pressure, and required compressor pressure ratio, the mass flow rate at maximum power can be estimated.

By looking at the basic guide lines presented in the turbocharger manufacturers' literature (such as that shown in figure 10.1), or the complete compressor characteristic curves (figure 10.2), a basic 'frame size' of turbocharger may be selected. The final choice of compressor will be made bearing in mind the complete operating lines of the engine over its whole speed and load range, superimposed on the compressor characteristic. The compressor 'trim' or diffuser will be chosen to allow a sufficient margin from surge while ensuring that the operating lines pass through the high-efficiency area. This will be discussed in detail in section 10.2.

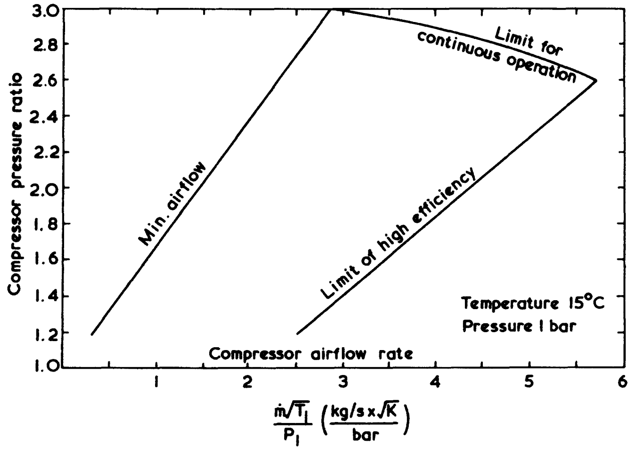

Figure 10.1 *Compressor airflow range*

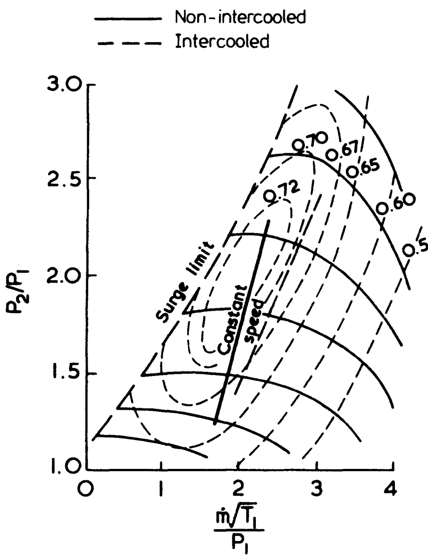

Figure 10.2 *Superimposition of engine running lines on compressor characteristics, constant engine speed, with and without charge air cooling*

Once the basic frame size and compressor have been established, the turbine must be matched by altering its nozzle ring, or volute if it is a radial flow machine. The effective turbine area will change, raising or lowering the energy available at the turbine and hence adjusting the boost pressure from the compressor. Although it is possible to calculate the over-all performance of a turbocharged engine and hence establish the most suitable turbocharger match, final matching will always take place on the test bed or, in some cases, when the engine is installed. In consequence, it is important when matching on a test bed to ensure that the correct air filter, intake and exhaust systems are used.

Turbocharger matching, particularly on a new engine, can be a lengthy process since many dependent parameters are involved. Although the basis is to match the turbocharger to the engine, it may well be necessary to improve the performance of the combination by engine design changes at the same time. Some such changes – adjustment of the fuel injection system, for example – are obviously essential. Others that may be desirable include alteration to the valve area and timing. It will not be possible to discuss all these factors in detail and the broad principles of matching fuel injection systems, swirl, optimisation of valve timing, etc., are not substantially different from those pertaining to naturally aspirated engines. Most large industrial highly rated turbocharged engines are designed as such from the outset today, and it is less common for a naturally aspirated design to be uprated by turbocharging in this class of engines. The designers will know from experience approximately what valve timing, injection pressures, etc., will be suitable, although factors such as these will be modified in the light of subsequent testing. Manufacturers of smaller engines however, sometimes turbocharge engines that were originally designed for naturally aspirated use. Fortunately, if modest power gains are sought (up to 50 per cent), major redesign is seldom necessary since the maximum cylinder pressure may be held down by an engine compression ratio reduction, and thermal stresses are not usually a major problem. However, if greater power gains are required the engine will probably require major redesign.

10.2 Air Flow Characteristics of Engine and Turbocharger

10.2.1 Four-stroke Engine

The air flow rate through a turbocharged (non-aftercooled) diesel engine will be a function of the engine speed, compressor delivery air density and the pressure differential between intake and exhaust manifolds during the period of valve overlap. If the engine is run at constant speed, but steadily increasing load, then the mass flow rate will increase approximately with the increasing charge density. The air flow through the engine may be superimposed on a turbocharger compressor characteristic, as shown in figure 10.2, the slope being governed by the density ratio. When matching a turbocharger to an engine with this operational requirement, the objective will be to choose a compressor such that the constant engine speed line falls through the middle of the high-efficiency area of the compressor map. If an aftercooler is fitted, then, as load increases, the cooling effect will increase charge density more rapidly for a corresponding boost pressure, hence the slope of the constant engine speed air flow line on the compressor characteristic will be less steep (figure 10.2). Again, to match the turbocharger for this duty, the objective will be for the operating line of the engine to pass through the high-efficiency region of the compressor characteristic.

Consider next an engine running at constant load (BMEP) but increasing speed. As the engine speed increases so will the volumetric flow rate of air. The effective flow area of the turbocharger turbine remains almost constant hence turbine inlet pressure will rise. As discussed in chapter 7, the result is an increase in energy available for expansion through the turbine and hence increased boost pressure at the compressor. Thus the constant load line of the engine will not lie horizontally on the compressor characteristic, but will rise with engine speed (figure 10.3), the slope depending on whether the engine is aftercooled or not. If the engine is required to operate over a range of speeds and load (for example, an automotive unit), then a set of constant speed and constant load lines may be drawn on the compressor characteristic to represent the operating range (figure 10.4). The complete engine characteristic must lie between the compressor surge line and the limit imposed by low efficiency or possibly turbocharger over-speed at high mass flow rates.

The margin between surge and the nearest point of engine operation must be sufficient to allow for three factors. Firstly, pulsations in the intake systems may well induce surge when the mean flow lies close to the nominal surge line. Secondly, if the air filter becomes excessively blocked in service, the air flow rate through the engine will reduce, but the turbine work will be maintained by a hotter exhaust as the air/fuel ratio gets richer. Thus boost pressure may not fall and some movement of the engine operating line towards surge may occur. A larger movement towards surge will result if the engine is operated at altitude. The effect of altitude operation on turbocharged engine performance is discussed

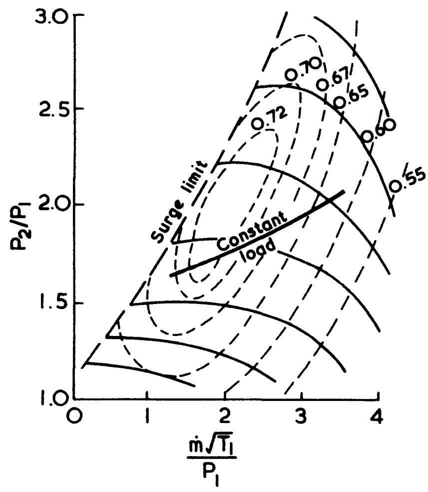

Figure 10.3 *Superimposition of engine running lines on compressor characteristics: constant engine load*

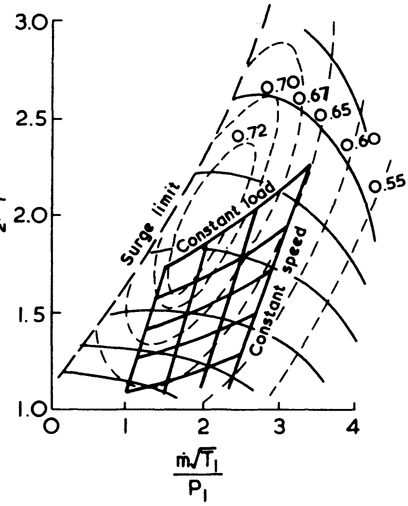

Figure 10.4 *Superimposition of engine running lines on compressor characteristics: constant engine load and speed lines*

in section 10.9. The combined effect of the three factors affecting surge margin will vary from one engine for one application, to another. In general, however, a margin of at least 10 per cent (of mass flow rate) between surge and the nearest engine operating line should be allowed. On engines with a small number of cylinders a 20 per cent margin will sometimes be required.

Generally, the turbocharger turbine can operate efficiently over a wider mass flow range than its compressor. It follows that it is more important to examine the engine air flow plotted on the compressor map than on the turbine map. This is indeed fortunate since if the turbine is operating under the pulse system with highly unsteady flow, it is not realistic to plot a 'mean' value on the turbine map. The result can be quite misleading, which is one of the reasons why turbocharger manufacturers are reluctant to offer turbine characteristic maps to many of their clients. To assess accuratley the operating area on a turbine map would require a plot of instantaneous gas flow and pressure ratio over a full range of engine operating conditions. The information is very difficult indeed to measure accurately. Probably the best that can be done [1] is to predict the instantaneous values using the methods outlined in chapter 15.

Figure 10.5 shows the turbine characteristic in terms of mass flow parameter against speed parameter, along lines of constant pressure ratio, with efficiency loops superimposed. This presentation enables the pulsating flow through the turbine to be illustrated, and its effect on turbine matching. The vertical lines denotes full load engine operation at four speeds. At any engine speed, the top point denotes the maximum amplitude point on the pressure wave passing through the turbine, and the bottom point corresponds to a pulse minimum. The line joining these points is vertical, since turbocharger speed varies only slightly at constant engine speed. At low engine speed (1000 rev/min), the pressure pulses are likely to be separated by short periods of zero flow, hence the pulse

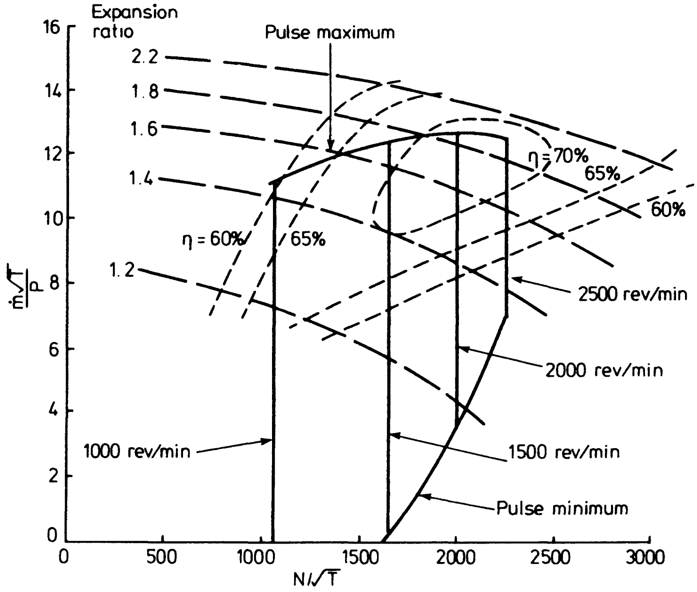

Figure 10.5 *Turbine performance map with pulse flow operating lines superimposed*

minimum corresponds to a mass flow parameter value of zero. In figure 10.5, optimum turbine efficiency coincides with a high pressure ratio point on the 2000 rev/min line. Since the point of maximum mass flow is also the point of highest pressure ratio, optimum turbine efficiency coincides with a high instantaneous mass flow rate, at 2000 rev/min.

Maximum instantaneous turbine power will be developed when the combination of mass flow rate, pressure ratio and efficiency are optimum, since

$$\dot{W}_{t_i} = \dot{m} c_p T_p \; [1 - (P_a/P_p)^{(\gamma_e - 1)/\gamma_e}] \, \eta_t$$

Maximum total turbine power must be calculated by integrating instantaneous power, over an engine cycle, since

$$\dot{W}_t = \frac{1}{720} \int_0^{720} \dot{W}_{t_i} \, d\theta$$

This will occur when peak efficiency falls at a point associated with just less than maximum pressure ratio or mass flow rate. Thus the turbine match shown in figure 10.5 is optimum for the engine speed range of 1700 to 2500 rev/min, at full load. If peak turbine efficiency is required at low speed, then a turbine having a high efficiency area at lower rotational speeds, will be optimum.

The compressor may be matched initially by choosing the best combination of impeller and diffuser such that the engine operating characteristics lie within the guide lines given above. Final matching will depend on the type of power

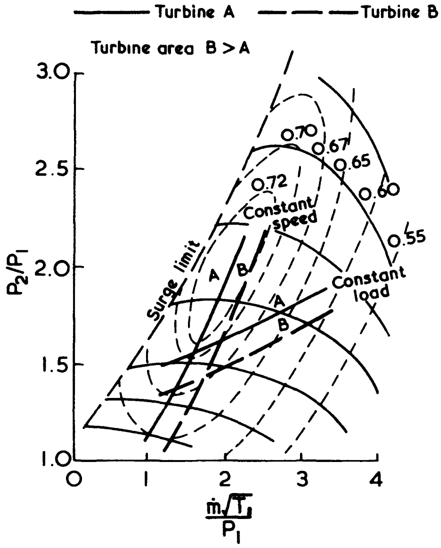

Figure 10.6 *Superimposition of engine running lines on compressor characteristics: effect of turbine area*

or torque curve required from the engine for a certain application, and will be discussed later. However, it has been pre-supposed that the turbine is able to provide sufficient power to drive the compressor and produce the air flow conditions discussed. Since the useful flow range at the turbine is wider than that at the compressor, the turbine supplied by the turbocharger manufacturer will inevitably be able to cope with the necessary mass flow. Whether it produces sufficient power depends on its efficiency and the turbine area (since this dominates the energy available for useful expansion). The turbine nozzle ring or volute controls its effective area. Thus the effective area at the turbine will be adjusted (by changing the components mentioned above) to achieve the desired boost level from the compressor. If turbine area is reduced, then the compressor boost pressure and the mass flow rate will go up, the former by a larger amount than the latter since charge temperature will also rise. The effect, in terms of the engine operating lines superimposed on the compressor characteristic is shown in figure 10.6.

10.2.2 Two-stroke Engines

The air flow characteristics of a two-stroke engine will depend on whether the engine is fitted with a turbocharger alone or has an auxiliary scavenge pump or blower. Consider first the situation when a turbocharger is used on its own. During the period when the inlet ports are open, the exhaust ports or valves will also be open and the air flow rate will depend on the pressure drop between intake and exhaust manifolds. The physical arrangement is analogous to flow

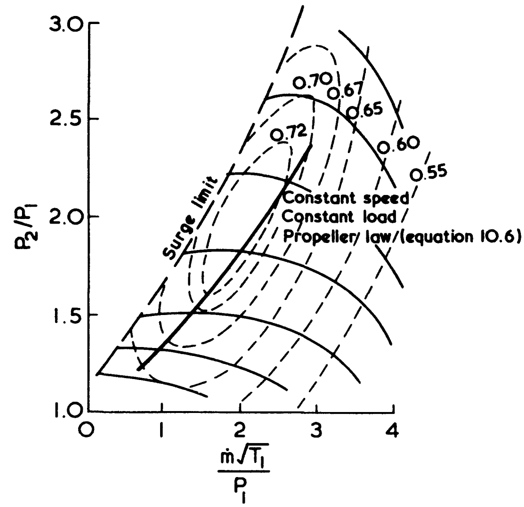

Figure 10.7 *Superimposition of engine running lines on compressor characteristics: propeller law, two-stroke engine*

through two orifices placed in series. The mass flow against pressure ratio characteristic for steady flow through two orifices in series is a unique curve and it follows that the two-stroke engine will exhibit a similar characteristic. Thus the engine operating line, when superimposed on the compressor characteristic will be a unique curve, almost regardless of engine load or speed (figure 10.7). Fortunately the compressor characteristics are well suited to this type of demand, as shown in figure 10.7, and it is relatively easy to match the turbocharger. If an aftercooler is fitted (which it is on most engines of this type) the mass flow/pressure ratio curve will have a different slope (smaller gradient) which suits the compressor characteristic even more.

If scavenge pumps or positive displacement compressors are placed in series with the turbocharger compressor on a two-stroke engine, then the air flow characteristic will be dominated by the scavenging device. For example, a reciprocating scavenge pump will exhibit much the same flow characteristics as a four-stroke engine, hence when constant load and speed operation lines are plotted on the compressor characteristic, the result is little different from that shown in figure 10.4. Somewhat similar characteristics are obtained if a rotary compressor (for example, a Roots blower) is used, although small differences will result from different compressor characteristics. In general, however, the matching problem will not be significantly different from that of a four-stroke engine and the comments previously made apply. The additional difficulty that does arise is that the scavenge blower must also be matched to the engine at the same time, since a compromise must be reached between scavenge blower and turbocharger work.

Generally the turbocharger will be expected to do as much of the compression work as possible. Depending on the quality of the scavenging system, turbocharger efficiency, use of pulse or constant pressure turbocharging, etc., the turbocharger may well be able to provide sufficient boost pressure at full power but not at low load. The subject has been discussed in detail in chapters 6 and 7. It follows that the work division between turbocharger and auxiliary compressor may well be governed by the requirement for adequate scavenging (and hence engine performance) for part load operation. The final balance will vary from engine to engine since many different factors are involved. An example, involving an application where a wide speed and load requirement exists, is discussed in section 10.8.

10.3 Matching for Constant Speed Operation

The most common application requiring a constant speed diesel power source is electricity generation, either for steady base load or stand-by units in the event of failure of an external supply. It is a relatively simple requirement from the point of view of turbocharger matching, as mentioned in section 10.2. The difficulties are associated more with the tight speed-governing requirements (since the frequency of alternating current generation will be governed by engine speed) rather than turbocharger matching, during sudden load changes. This latter situation is discussed in chapter 12.

The basic compressor characteristics resulting from this varying load at constant speed application is shown in figure 10.2. Small adjustments in turbine area will affect both the boost and exhaust pressures, hence pumping work and the resultant fuel consumption of the engine. Within the limits of acceptable mechanical and thermal loading of the engine, turbine matching will be used to achieve optimum performance and fuel consumption. Figure 10.8 shows the effect of turbine area changes on the specific fuel consumption with varying load at constant speed. Matching will be a compromise between performance at low and high load. If the diesel generator is required for base load operation, then the turbocharger will be matched at the rated load (full line in figure 10.8). Otherwise a larger nozzle ring or volute will be fitted to improve part load efficiency. The small nozzle increases piston pumping work and hence increases fuel consumption, except at high load, when the engine benefits from the extra air which results from more turbine and compressor work.

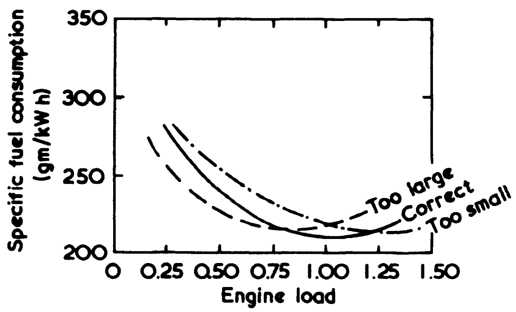

Figure 10.8 *Optimum turbine matching for fixed load, constant speed*

Fine tuning of the turbine match produces the smaller changes in engine performance shown in figure 10.9. This is a more highly rated engine (17 bar BMEP) with large valve overlap. The flow characteristics of the three turbine trims used are given in figure 10.10. In this case, the smaller turbine, although benefitting from increased available energy, has a lower efficiency than the larger trims (being cropped from a larger design). In addition the high exhaust pressure level developed by the small effective flow area of the turbine, has an adverse effect on the scavenge air flow during valve overlap. These effects combine to impair engine performance with the smaller turbine trim.

In figure 10.11 the engine air flow rates are superimposed on the compressor map, with each of the three turbine trims. Clearly the choice of compressor

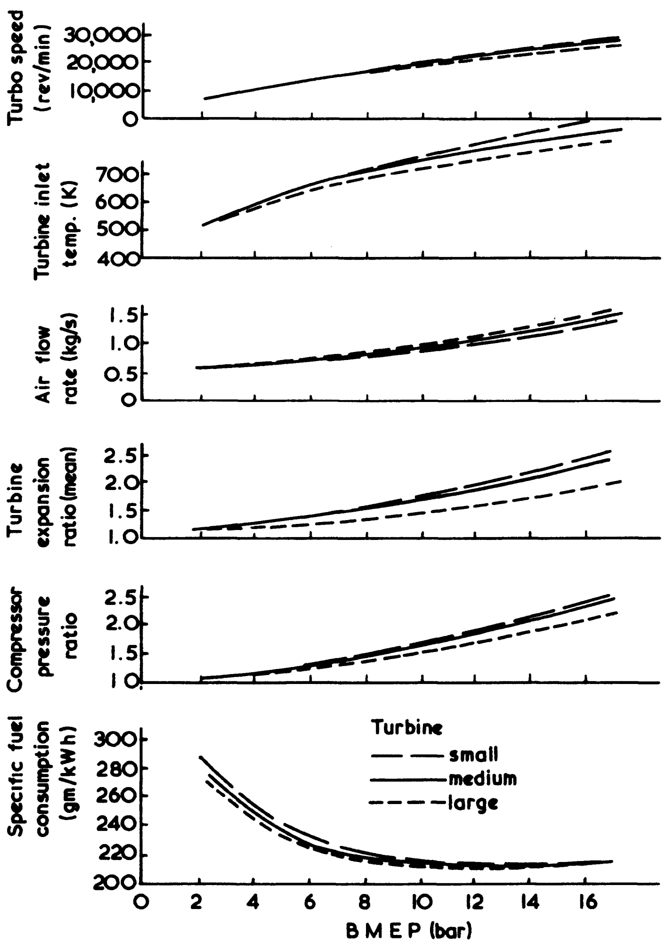

Figure 10.9 *The effect of small turbine area change on engine performance at constant speed*

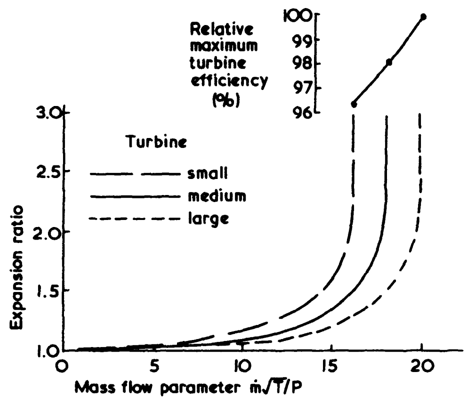

Figure 10.10 *Turbine trims used in figure 10.9*

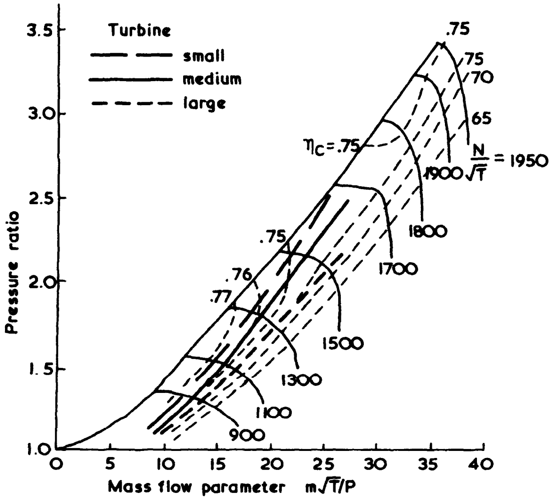

Figure 10.11 *The effect of turbine matching on compressor requirement for constant engine speed operation*

is correct with the medium turbine, but a smaller compressor trim would be needed to avoid surge with the smallest turbine, and a slightly larger variant to achieve optimum compressor efficiency with the largest turbine.

10.4 Matching the Marine Engine

The required power/speed characteristics of the marine engine are governed by the performance of the propeller, and will therefore depend on whether a fixed or variable pitch propeller is used.

The characteristics of the fixed pitch propeller are such that the power requirement increases with the cube of the speed (the well-known 'propeller law')

$$\dot{W} \propto N^3 \tag{10.6}$$

Thus BMEP increases with speed squared (figure 10.12). It happens that the output characteristics of the turbocharged engine are ideal for this application, hence matching is a case of optimisation rather than compromise. Compressor pressure ratio rises with engine speed as well as with load. The reason may be found in the flow characteristics of the turbine. As engine speed increases, so does the air flow through it (equation 10.1, for a four-stroke engine). The flow characteristic

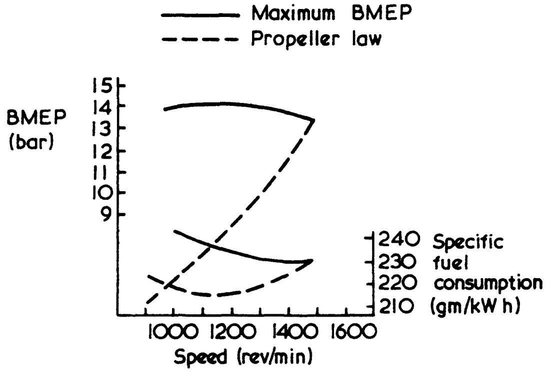

Figure 10.12 *Engine performance comparison at maximum BMEP and propeller law*

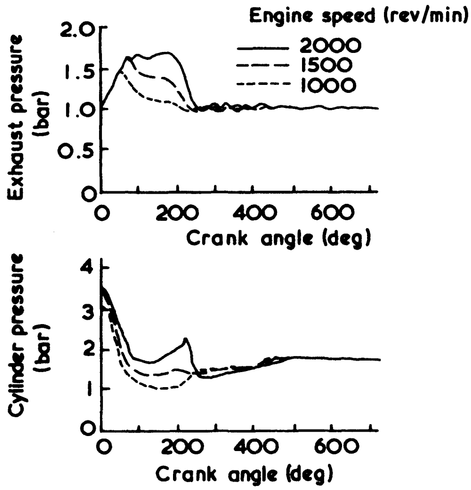

Figure 10.13 *The effect of engine speed on the exhaust process of a single-cylinder engine [2]*

of the turbine (figure 10.10) is such that the expansion ratio is forced to increase with air flow and therefore engine speed (figure 10.13). The torque developed by the turbine is largely a function of expansion ratio and turbine inlet temperature, if changes in efficiency are small, since

$$TQ = \dot{W}/2\pi N = [1 - (P_a/P_p)^{(\gamma_e - 1)/\gamma_e}]\, T_t c_{p_e} \eta_t\, \dot{m}/2\pi N \tag{10.7}$$

Thus turbine, and hence compressor, torque rises with engine speed. The boost pressure developed by the compressor is largely a function of torque input from the turbine, hence speed and load. However, optimum specific fuel consumption will occur at lower speeds, where mechanical friction losses are lower.

10.4.1 The Four-stroke Engine with Fixed Pitch Propeller

Figure 10.14 illustrates a typical operating line on the compressor characteristic for fixed pitch propeller operation of a four-stroke engine. Two important facts are clear. Firstly, if the turbocharger is correctly matched, the compressor is working in an area of reasonably high efficiency at all engine speeds and loads, but if highly rated, the surge margin will be governed by mid-speed performance, due to the 'waist' shown in the surge line.

When matching at full speed the turbine will be matched to produce maximum engine power output, subject to thermal and mechanical limits. Generally the result will also be minimum specific fuel consumption. If different turbine areas are tried, various boost pressures will be developed and the propeller law working line will move across the compressor map. The compressor diffuser (or the complete assembly) will be changed to ensure that the operating line falls through the optimum efficiency area with a sufficient surge margin to allow for intake pul-

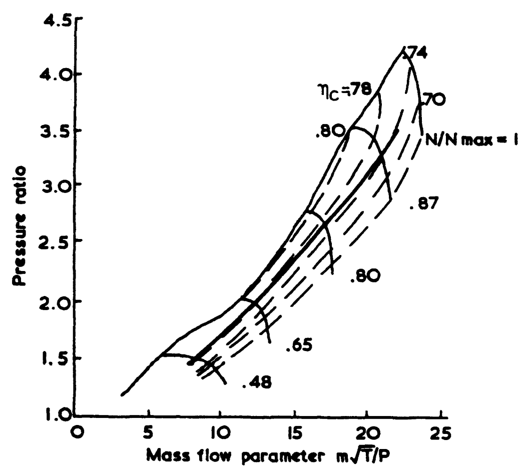

Figure 10.14 *Compressor match for a four-stroke engine with fixed pitch propeller [3]*

sations, and compressor fouling in service. When matching, the power output, specific fuel consumption and exhaust temperature will be monitored, but generally there will be no major conflict between variables.

It is common practice among the builders of medium and slow-speed diesels to offer a large range of power outputs by offering engines with increasing numbers of cylinders, each cylinder being virtually a self-contained unit fixed to a common crankcase. For convenience the engine may be rated at X kW (or BHP) per cylinder producing, for example, $12X$ kW in V12 form. A problem then sometimes arises in achieving the same power output per cylinder on those engines which have favourable numbers of cylinders for pulse operation (for example, 12) and those that do not (for example, 8, see chapter 7). In the latter case poor turbine efficiency due to the highly unsteady entry flow must be compensated for by reducing turbine area to increase the energy available at the turbine. The result will be a higher exhaust temperature and possibly a small deterioration in specific fuel consumption due to increased pumping work.

Frequently, several medium-speed engines will be used in a single vessel, with perhaps two engines geared down and geared together driving each propeller. The requirement can arise, if one engine fails, of driving the propeller with one engine only. If the engine-turbocharger combination has been matched for optimum performance when producing half of the maximum power required by the propeller, then the match will be unsuitable for operation as one engine. Naturally this engine cannot drive the propeller at full speed (it will not have the power), but it is theoretically capable of turning it at nearly three-quarters of its maximum speed due to the power requirement being proportional to the cube of the speed (propeller law). Consider what happens if the engine runs at

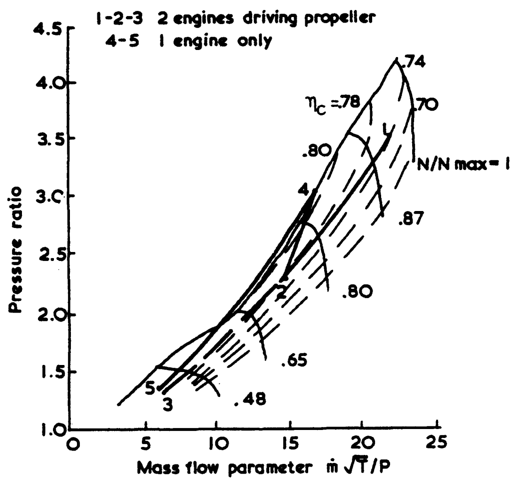

Figure 10.15 *Engine mass flow characteristic superimposed on compressor map when one of two engines is running, with a fixed pitch propeller [3]*

70 per cent of full speed, remembering that the power required is double what would be required from each engine if both were running at that speed. If both engines were running, the operating point would be that denoted by point 2 in figure 10.15, but point 4 with one engine. Unless the compressor is matched with a very large surge margin, surge is inevitable. Thus the optimum match with both engines working correctly is compromised.

10.4.2 The Four-stroke Engine with Variable Pitch Propeller

A much broader power/speed requirement is obtained from a variable pitch propeller. At each pitch setting, the propeller law will apply and hence the characteristic will be an envelope of propeller law curves. Figure 10.16 illustrates how the power curve of the engine will cut across an 'extreme pitch' curve if the propeller is matched to require full engine power at its rated speed at optimum

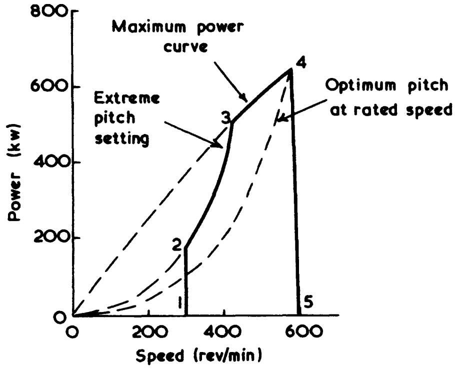

Figure 10.16 *Propeller law power curves with a variable pitch propeller [3]*

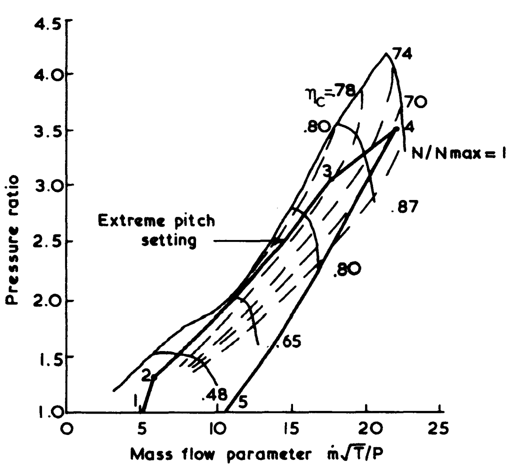

Figure 10.17 *Compressor matching with a variable pitch propeller [3]*

pitch. Lines 1-2 and 5-4 denote minimum and maximum engine speeds. When plotting the resultant operating regime on the compressor characteristic (figure 10.17) it can be seen that it is the 'extreme pitch' curve that determines the surge margin, not the maximum speed point (4). The engine can still be matched for optimum performance at full speed, but the compressor will have to be chosen to allow line 2-3 to be well clear of surge. This may result in a small penalty in performance at point 4.

10.4.3 The Two-stroke Engine with Fixed Pitch Propeller

The different types of scavenging system used by two-stroke engines have been described in earlier chapters. In general, only pulse turbocharged uniflow scavenged engines can operate without a scavenge pump or fan in addition to the turbocharger over the complete operating range required. This class includes the very large opposed piston and 'valve in head' marine two-stroke engines. The power required by the propeller will naturally be governed by the propeller law and it follows that, if the engine is matched at its rated speed, the equilibrium running line will fall on the compressor map in a similar manner to the four-stroke engine. However, the air flow characteristics will be governed by the scavenge period, and may be simulated by two orifices in series (that is, to represent the intake and exhaust of the engine). This characteristic suits the compressor well, and matching is usually a reasonably straightforward exercise similar to that described for four-stroke engines.

If scavenge pumps are used, the division of work between them and the turbocharger will influence the matching process. Generally though, the capacity of the scavenge pumps will have been estimated and the turbocharger is matched for optimum performance in conjunction with those pumps. A change in pump capacity might follow, depending on the success of the matching exercise, in which case the whole procedure must be repeated. To ensure that the engine will not stall at low speeds, the scavenge pumps must supply sufficient air to raise the inlet manifold pressure above that in the exhaust. If the turbocharger compressor precedes the scavenge pump, then the air flow characteristics obtained on the compressor map at low speeds, will be highly dependant on the capacity of the scavenge pumps. Thus a range of air flow requirement curves could be plotted on the compressor map. At low speeds the air flow will depend principally on the pump speed, but at higher speeds, the influence of the turbocharger compressor will significantly affect air flow by increasing the charge density at entry to the scavenge pumps.

10.5 Matching for Diesel-Electric Traction

Diesel generators are frequently used for rail traction since the characteristics of the basic diesel engine are not ideally suited for direct drive. In particular, the locomotive engine requires high torque at zero speed, to accelerate a heavy train from rest. Sufficient voltage can be generated (by a diesel generator) to produce the excitation necessary at the electric drive motors. The power needed to accelerate the electric motor from zero speed is less than the capability of the diesel engine at higher speeds. If the turbocharger is matched to the engine at its rated speed and load, the equilibrium running lines will usually be quite well positioned on the compressor map at other speeds and loads. A possible area of trouble is surge at low speed and high load, and this may mean that the compressor build finally chosen leaves a rather large surge margin at full power.

Since, like the automotive engine, the diesel-electric locomotive unit is mobile, it is possible that it may be required to operate at altitude. The turbocharger must be matched to allow sufficient surge margin, and, if it is known that the engine will run at a particularly high altitude, the fuel injection system and turbocharger match will need to be adjusted to suit. The engine will effectively need derating to prevent over-loading at altitude (see section 10.9).

10.6 Matching for Other Industrial Duties

The power/speed requirements for other industrial duties will generally fall between those of an automotive engine and the other applications described above. A typical industrial application might be the drive to a reciprocating compressor. The compressor can be operated over a reasonably wide speed range, from zero to full load as the control valves open or close. The resultant operating area on the engine and compressor maps is similar to the requirements for an automotive engine (section 10.7). The surge margin will be governed by full load operation at low speeds and this will force the full speed and load point well away from surge and possibly into a low efficiency area of compressor operation.

10.7 Matching the Four-stroke Automotive Engine

Turbocharger matching for many industrial and marine duties is relatively straightforward due to the limited speed and load ranges required. Matching the turbocharger to an automotive engine is considerably more difficult due to the wide speed and load variations encountered. Although the power required to propel a vehicle increases rapidly with speed, it is inappropriate to design an engine with this type of output. Consider the power and torque curves of two different engines as shown in figure 10.18.

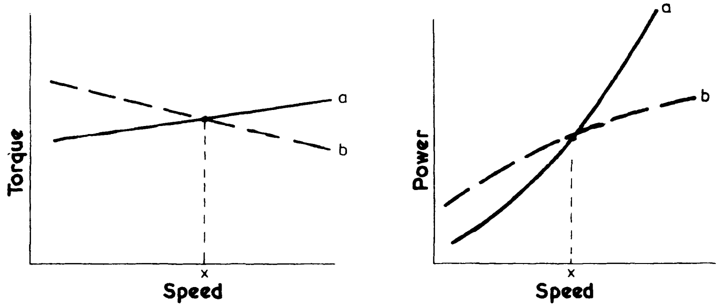

Figure 10.18 *Power and torque curves*

If two identical vehicles (with engines 'a' and 'b') are running along a flat road at speed x, both could be using the same maximum torque output from their engines. When the vehicles arrive at a small hill, the engine load will increase but, since both engines are already producing their maximum torque at that speed, the speed must fall. As engine speed falls, the engine in truck 'b' will produce more torque to meet the increased load applied, enabling the engine to propel the vehicle up the hill at a somewhat lower speed. In contrast, the engine in truck 'a' will produce less torque as its speed drops, compounding the deceleration of the vehicle. Vehicle speed will fall rapidly forcing the driver to change to a lower gear. Although the situation has been over-simplified, it illustrates the requirement for a torque curve that rises as speed falls, otherwise driving the vehicle will be tiring due to the need for frequent gear changes. Such a torque curve is said to have good 'torque back-up'. Pulse turbocharging is essential in obtaining a good torque back-up at low engine speed.

With good torque back-up, the vehicle will benefit from high torque at low speeds to provide a margin for acceleration and to allow the vehicle to lug up very steep hills with a limited number of ratios in the gearbox. Figure 10.19 shows the ability of a truck to climb hills if constant power is available over the whole speed range. Superimposed is the vehicle's hill-climbing ability in each gear ratio (five-speed gearbox) with a more typical engine fitted. The shaded portions represent running conditions falling below the maximum power output of the engine but are not achievable in practice because the engine cannot develop sufficient torque at that speed. These areas are also shown, but more clearly, in the power availability envelopes of figure 10.20. By increasing engine torque at less than full speed, the left-hand part of the power envelope in each gear will rise, possibly at the expense of maximum power. However, the result will probably be a more useful power availability for a fixed number of gear ratios. Otherwise the only method of reducing the shaded areas is to use a gearbox having more ratios.

Clearly the turbocharged automotive diesel engine should not be matched at full power and a compromise must be reached between power and a suitably

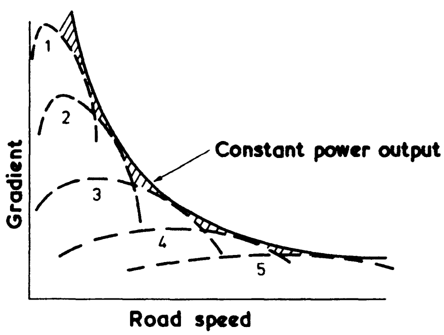

Figure 10.19 *Gradient against speed ability of a truck with a five-speed gearbox [4]*

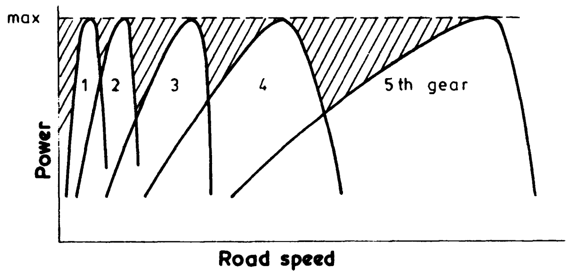

Figure 10.20 *Power/speed envelope for a five-speed gearbox truck*

shaped torque curve. Figure 10.21 shows a typical torque curve for a naturally aspirated automotive diesel engine (curve 'a'). Torque back-up may be defined as

$$\frac{\text{maximum torque} - \text{torque at maximum speed}}{\text{torque at maximum speed}}$$

Maximum torque of engine 'a' occurs at 54 per cent of its maximum speed, and the torque back-up is 16 per cent. Since the engine will not normally be required to work below about 40 per cent of the maximum speed, torque rises with reducing speed over half the useful speed range. This characteristic is, fortunately, a reasonable compromise between power and low-speed torque, enabling trucks to use five or six-speed gearboxes.

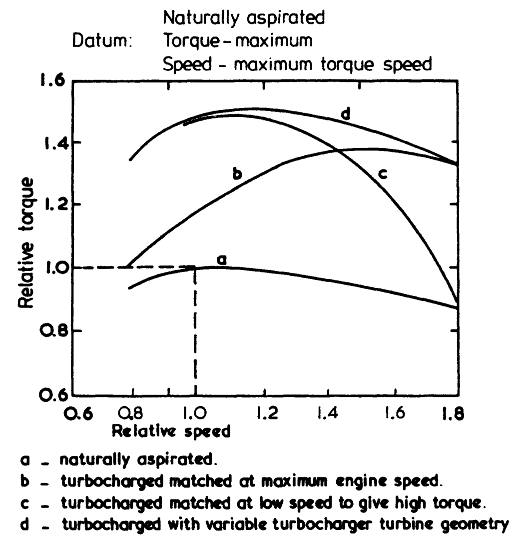

Figure 10.21 *Torque/speed relationships: naturally aspirated and with quite different turbocharger matches*

Although it has been made clear that it is not sensible to match the turbocharger at the maximum speed of the engine for this application, the torque curve that might result is shown in figure 10.21 (curve b). The result is a totally inappropriate torque curve. An alternative is to match the turbocharger to produce maximum torque at the same speed as might be expected on a naturally aspirated engine (curve c), if possible. Excellent torque back-up is obtained but the maximum power output (torque x speed) is considerably reduced.

Before discussing the merits of compromise solutions between the extremes shown in figure 10.21, it is worthwhile considering what engine or turbocharger limitations might prevent such curves being obtained in practice. Engine performance will typically be limited (for reliability) by a maximum value of cylinder pressure and possibly exhaust temperature (since the latter is an approximate guide to thermal loading, for example at the exhaust valve). In addition, a limit will be imposed by the acceptable (or legislated) exhaust smoke level, which will be determined largely by the quantity of air delivered by the turbocharger. The turbocharger will be limited by a maximum safe rotational speed and turbine inlet temperature (governed by the creep and scaling properties of the turbine wheel and housing). Depending on engine rating, some or all of these factors will limit engine performance. These limits can be superimposed on the torque (or BMEP, since torque ≈ BMEP) curve as shown in figure 10.22, where the maximum possible torque curve is shown. However, the position of most of these limiting lines will move if the turbocharger match is changed.

It is clear from figure 10.22 that the factor that is most restrictive when trying to achieve a desirable torque characteristic is the low-speed smoke limit. This comes as no surprise since, in section 10.4, it was explained that it is normal for boost pressure to rise with engine speed, as a direct result of the air flow characteristics of the turbocharger turbine. For example, the flow characteristics of

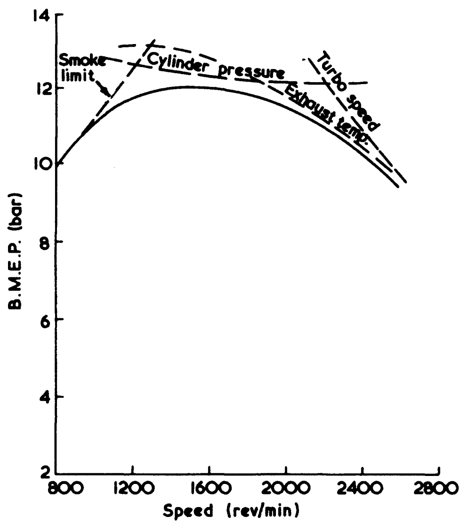

Figure 10.22 *Turbocharger and engine limitations [5]*

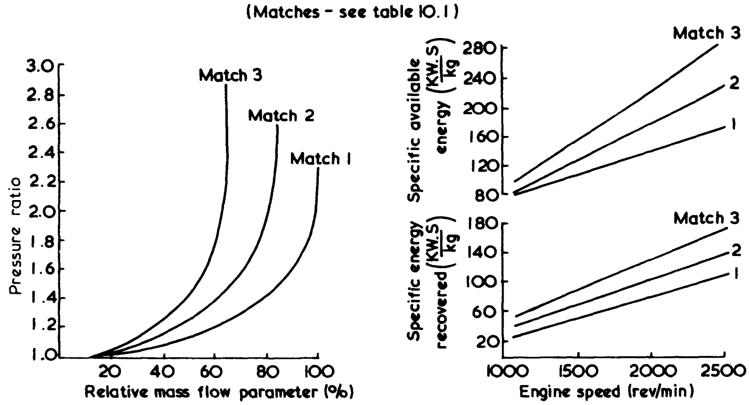

Figure 10.23 *Turbine flow characteristics and resultant energy utilisation of the turbine [6]*

three radial turbine trims (volutes) are shown in figure 10.23. The smallest area volute will generate the highest turbine inlet pressure and therefore the highest specific available energy at the turbine. Allowing for the variation of turbine efficiency over the pulsating and mean flow range, specific energy recovered by the turbine also increases substantially with engine speed. This characteristic is a consequence of the almost constant effective flow area of a fixed geometry turbine. Thus the natural characteristic of the turbocharged truck engine will be

TABLE 10.1 *Turbocharger and Fuelling Changes*

Match	Comment	Peak Compressor Efficiency (%)	Peak Turbine Efficiency (%)	Relative Turbine Area	Maximum Fuelling (relative)
1	Baseline	70	68	100	100
2	Smaller turbine	75	73	82.8	100
3	Smallest turbine	75	67	66.0	100
4	Match 2 plus increased fuelling	75	73	82.8	114
5	Match 4 plus waste gate	75	73	82.8	114
6	Match 4 with tailored fuelling	75	73	82.8	114
7	Match 2 increased fuel and limited speed range	75	73	82.8	118

compressor work (and therefore boost pressure) rising with engine speed. The smoke limit is caused by insufficient boost pressure, and hence air flow, at low engine speeds.

If the fuel delivered to the cylinders is allowed to increase with speed, to match the increase in air flow, then torque curve b (figure 10.21) will result. In order to achieve a more acceptable torque curve, the fuel delivery (per cycle) is held relatively constant over the speed range and efforts are made to raise boost pressure at low speed. Two techniques are available to achieve this. Either the turbocharger efficiency must be raised at this operating point, or the thermodynamic availability of energy delivered to the turbine (that is, specific available energy, figure 10.23) must be increased. Both techniques are usually adopted, and will be discussed under the headings of turbine and compressor matching.

10.7.1 Turbine Matching

Figure 10.23 shows that by reducing turbine area, for example from match 1 to match 3, specific available energy at the turbine increases at all speeds. If the fuel delivery schedule is unchanged, then boost pressure increases as shown in figure 10.24, weakening the air/fuel ratio and reducing low-speed smoke. Match 2 is based on a small turbine area reduction, hence the increase in available energy at low speed is small (figure 10.23). However, turbine operation is more efficient with this build, hence the benefit in actual turbine work is more substantial.

The benefit of reduced smoke at low speeds does not come without accompanying disadvantages. Figure 10.23 shows that with the smallest turbine (match 3), the expansion ratio across the turbine will be very high at the maximum engine speed, when air flow is greatest.* Thus the piston must pump the exhaust

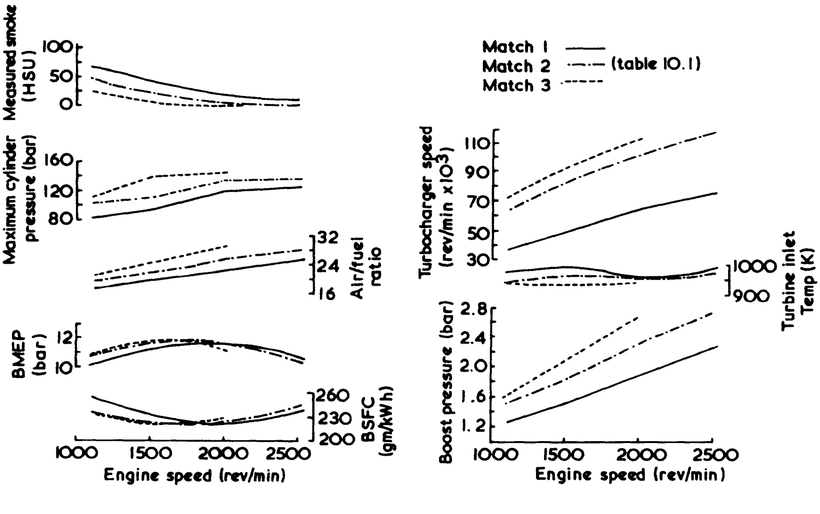

Figure 10.24 *The effect of turbine match on engine performance with no change in fuelling [6]*

gases out against a high pressure, resulting in poor net power output and fuel consumption. This is seen as low BMEP and high BSFC at high engine speed (2000 rev/min) with match 3, figure 10.24. In addition, the engine exceeds the allowable limits of maximum cylinder pressure and turbocharger speed. Thus match 2 is a reasonable compromise, except that maximum power (BMEP at maximum speed) is marginally less than that achieved with match 1.

Ideally a variable geometry turbine is required. This would have the effective turbine area of match 1 at full engine speed, but the effective area of match 2 at mid-speed and match 3 at low speed. This would offer the low exhaust smoke, low fuel consumption and high BMEP of match 3 at 1000 rev/min, with the low cylinder pressure and low fuel consumption of match 1 at 2500 rev/min. This is an ideal that has attracted turbocharger and engine manufacturers for many years, since it reduces the speed dependence of turbine specific available energy shown in figure 10.23. The problem is one of engineering a cheap, reliable and effective system, and has not been solved to date. Prototypes exist, but no system is currently available in mass production. Experimental results with a prototype variable geometry system are presented in chapter 11.

By taking advantage of the extra air flow generated by match 2, fuelling may be increased over the complete speed range in order to raise torque and restore the maximum power output of match 1. Figure 10.25, match 4, is the extreme case, in which fuelling is increased (with turbine 2) until the original maximum air/fuel ratio is achieved. Naturally, low-speed smoke has reverted to its former value, but a substantial increase in power output has been obtained, without loss of torque back-up or a substantially higher turbine inlet temperature. A small deficit in fuel consumption at high speeds is countered by a substantial improvement at low speeds. The disadvantages are high mechanical loading (maximum

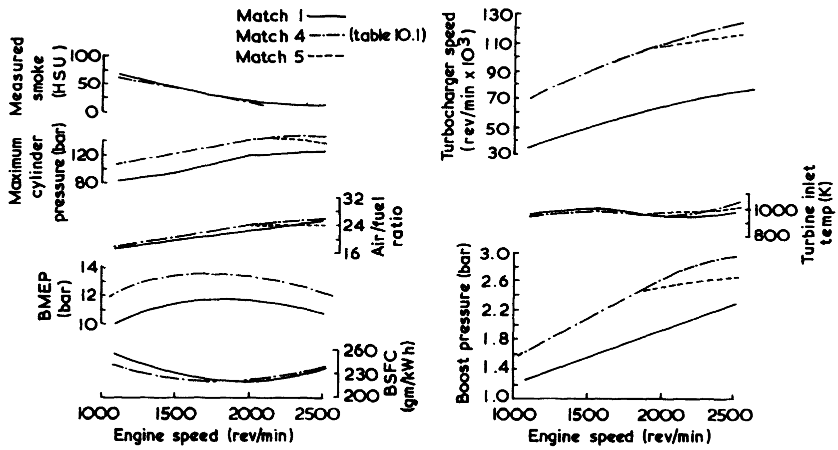

Figure 10.25 *The effect of turbine match on engine performance at constant minimum trapped air/fuel ratio [6]*

* Turbocharger specifications are given in table 10.1.

cylinder pressure and turbocharger speed). In practice, a complete envelope of performance options between matches 1 and 4 is available by suitable choice of turbine trim and fuel setting.

Further 'fine-tuning' of the turbine may be achieved by selecting from different components having approximately the same effective area at the mid engine speed condition, but whose flow and efficiency variations differ over the working range of the engine (see chapter 4 and Watson [1]).

10.7.2 Exhaust Waste Gates and Automotive Engines

The problem of over-speeding the turbocharger and coping with high cylinder pressures becomes prominent when engines which operate over a very wide speed range are turbocharged and matched for good torque back-up. The small turbocharged passenger car diesel engine falls into this category. A method of avoiding this problem is to by-pass some of the exhaust gas around the turbine (figure 10.26) at high speed and load. Thus, when a small turbine is fitted to achieve good low-speed boost, the massive increase in specific available energy at the turbine at high speed is alleviated by increasing the effective flow area out of the exhaust manifold. This has two effects. Firstly, only part of the exhaust gas flow goes through the turbine. Secondly, the increase in flow area reduces the exhaust pressure that would otherwise build up. Both measures reduce turbine work and hence boost pressure. In addition, the second factor reduces pumping work during the exhaust stroke and would, for example, moderate the loss in BMEP and deterioration in fuel consumption shown in figure 10.24, match 3, at high speeds.

The by-pass valve will usually be built into the turbine casing, and will consist of a spring-loaded valve acting in response to the inlet manifold pressure acting on a controlling diaphragm. Different combinations of spring load, diaphragm area and valve area can be used to achieve a wide variety of boost pressure variations with engine speed. Disadvantages are increased cost, potential unreliability and the restriction to a single-entry turbine housing. Thus some of the

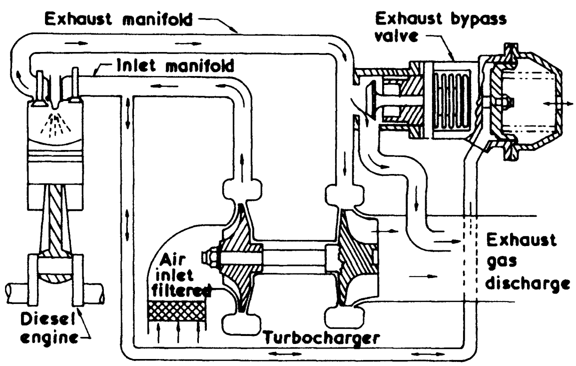

Figure 10.26 *Turbocharger exhaust by-pass valve [7]*

benefit of good pulse energy utilisation at low speed (chapter 7) is lost, particularly on six and eight-cylinder engines.

Figure 10.25, match 5, shows how an exhaust waste gate may be used to reduce the high boost pressure developed by a small turbine (match 4) and hence control turbocharger speed and, to a lesser extent, maximum cylinder pressure.

10.7.3 Fuel System Matching

Development of fuel injection pumps and associated equipment has introduced additional freedom to vary fuel delivery over the speed range of an engine. The turbocharger matching process must be closely linked with fuel system matching even after optimum injection rates, pressures, nozzle sizes and swirl have been achieved.

Tailoring of the fuel delivery characteristic is a method of achieving good torque back-up within the framework of engine and turbocharger limitations (figure 10.22). For example, match 6, figure 10.27, is identical to match 4, except that maximum fuelling is restricted at high speeds in order to limit the maximum turbocharger speed. Fuelling is gradually restricted from 2000 rev/min upwards, but a reduction from 1500 rev/min upwards would reduce not only turbocharger speed but maximum cylinder pressure. Thus impressive torque back-up would be achieved, but at the expense of a low maximum power output.

At the other end of the speed range, excessive smoke can be reduced by restricting fuelling until sufficient boost is available to generate a reasonable air/fuel ratio. Thus a spring-loaded diaphragm senses boost pressure and allows the maximum fuel stop to open as engine speed increases. Since fuelling is restricted only when the boost pressure is zero or low, torque is only reduced at

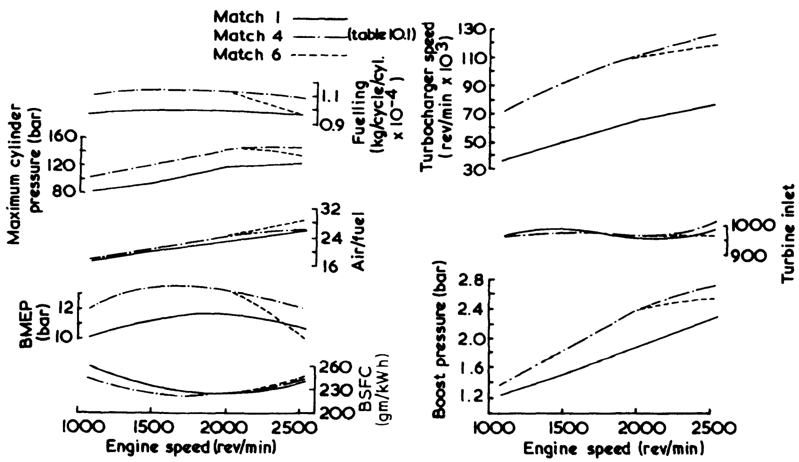

Figure 10.27 *The effect of turbine match and fuel control on engine performance [6]*

very low speeds, that is, below that at which maximum torque is achieved. The device is commonly called an 'aneroid' (fuel controller).

10.7.4 Engine Speed Range

The difficulty of achieving a satisfactory match over a wide speed range has been explained. In certain circumstances it may be advantageous to reduce the rated speed of an engine while increasing BMEP to achieve the same maximum power output.

By reducing the turbine area and increasing fuelling (figure 10.25, match 4), high BMEP is obtained. If maximum rated speed is reduced from 2500 rev/min to 2000 rev/min (an extreme case), excessive turbocharger speed is avoided. Naturally the final drive gear ratio of the truck must be raised to compensate, hence the engine is working at a higher load than would normally be the case at the same vehicle speed and load. Since specific fuel consumption reduces with load, fuel savings are possible.

Figure 10.28 illustrates the fuel consumption of a truck with a conventional turbocharger (match 1), and a reduced speed (match 7), with the same maximum power output. The four diagrams denote operation at maximum speed (top), 80, 60 and 40 per cent of maximum speed (bottom). In all cases the speed of the match 7 engine is 84 per cent of the match 1 engine, with the final drive of the vehicle adjusted accordingly. The BMEP with match 7 is higher than with match 1, in order for both systems to develop the same power at their respective maxi-

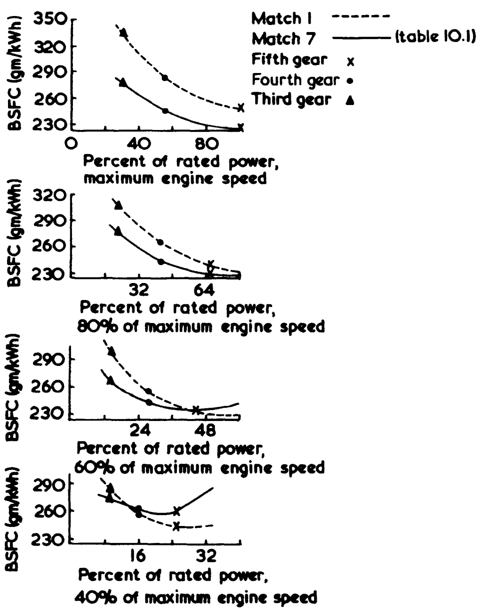

Figure 10.28 *The effect of reducing rated speed and increasing BMEP on steady speed fuel consumption [6]*

mum speeds. The right-hand points on the top diagram (denoted x) represent operation with the vehicle running at full speed along a flat road in fifth gear. Each engine is at full load and maximum speed. The specific fuel consumption is lower with match 7, since this engine is operating at lower speed and higher load, where engine efficiency is superior. The other points on the top diagram (denoted by ● and Δ) represent full engine speed in fourth and third gear, again with the vehicle driving steadily along a flat road. In these cases, vehicle speed is lower, hence the engine is operating at less than rated power (or BMEP) but its maximum speed. Match 7 remains more efficient than match 1 because BMEP is higher and speed lower.

At lower engine speeds the same basic trends are shown but the difference between match 1 and match 7 reduces as engine speed reduces and changes sign at very low speeds. Thus limiting speed range is seen to result in a significant improvement in fuel consumption over most of the operating range of the vehicle, except in high gear at low speeds. This advantage primarily results from the fact that running at lower engine speeds substantially reduces mechanical friction in the engine while higher load operation only causes a marginal increase.

10.7.5 Compressor Matching

Since the truck engine operates over a wide speed and load range, the air flow requirements cover large areas of the compressor map. A typical superimposition

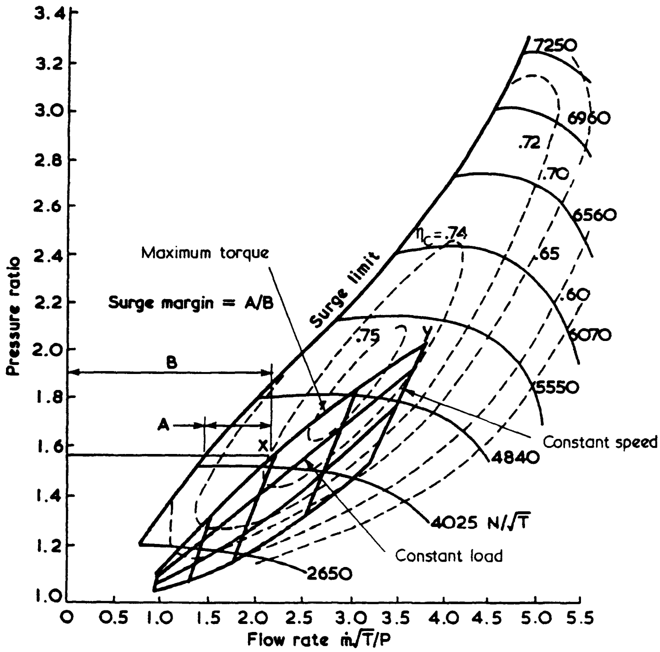

Figure 10.29 *Engine operation area superimposed on compressor map, showing surge margin*

of engine air flow on the compressor map is given in figure 10.29, showing lines of constant engine speed (1000, 1500, 1900, 2400 and 2800 rev/min) and load (3.85, 6.17 and 8.48 bar), and the maximum torque curve.

Selection of the correct compressor is largely a matter of ensuring a sufficient surge margin (A/B, figure 10.29) and that the operating points at maximum torque and power (points X and Y, figure 10.29) occur at reasonable compressor speed and efficiencies. Thus the compressor shown in figure 10.29 is a satisfactory choice, since the operational area is clear from the surge line and lies in an area of high efficiency. However, a slightly larger compressor might result in more of the operating regime experiencing higher compressor efficiency, with a less generous surge margin. A small improvement in low speed boost could also be obtained if maximum compressor efficiency occurred at a lower pressure ratio (for example, 1.6 compared with 1.8 in figure 10.29).

Reducing turbine area will raise the boost pressure, reducing the surge margin (compare figures 10.29 and 10.30) and therefore a smaller compressor trim may be required in some cases. However, the surge margin shown in figure 10.30 is adequate, the compressor being well matched.

Figure 10.31 shows a poor compressor match, using too small a compressor trim on an intercooled engine. At maximum speed and load (point Y) the compressor efficiency is low. Boost pressure actually falls as the engine speed increases from 2400 and 2800 rev/min as a result. It may occasionally be convenient to

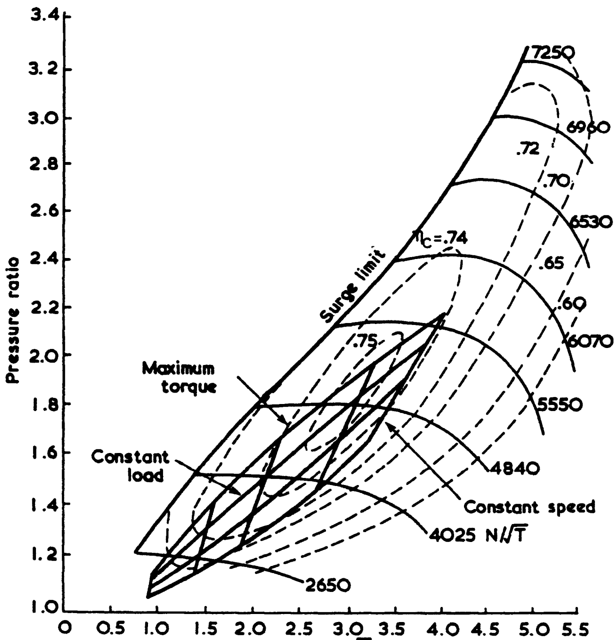

Figure 10.30 *Engine operation area superimposed on compressor map, showing surge margin with reduced turbine area*

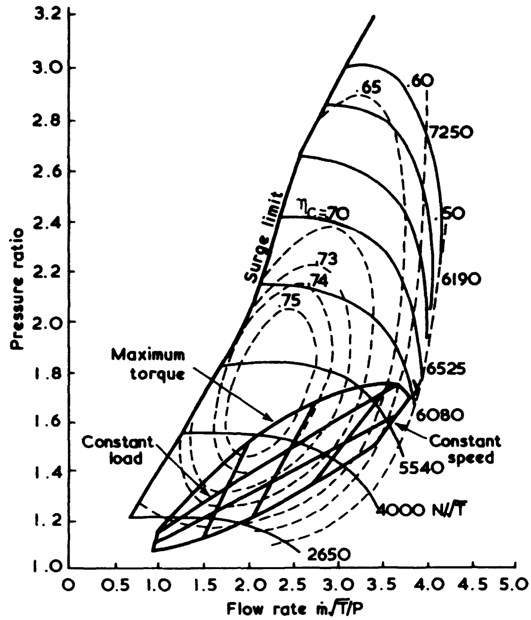

Figure 10.31 *An incorrectly matched compressor on a charged cooled engine*

deliberately match into an inefficient region at maximum engine speed and load, in order to hold the boost pressure and therefore maximum cylinder pressure down. However, an excessive reduction in efficiency occurs in the extreme case shown, and piston pumping work during gas exchange will suffer.

10.8 Matching the Two-stroke Automotive Engine

Few manufacturers now produce automotive two-stroke engines, General Motors (Detroit Diesel) being the only large company involved. Turbocharging the two-stroke automotive diesel engine involves some additional concepts to those already discussed above. Since the engine must work over a wide speed and load range, and must be capable of starting from a battery, some form of assistance will be required to produce the pressure drop from intake to exhaust manifolds so essential for scavenging at cranking speeds. Normally a Roots blower is used, placed in series after the turbocharger compressor. As explained in section 10.2, the Roots blower will ensure that the air flow characteristics are similar to those of a four-stroke engine. However, the balance of work between Roots blower and turbocharger will affect the power output, torque curve and specific fuel consumption.

Similarly to a four-stroke engine, the boost pressure will rise with speed and load. It follows that the Roots blower will be expected to provide most of the boost pressure at low speeds and loads, yet a reducing proportion of the total as speed and load rises. The capacity of the Roots blower will be governed by the need for acceptable performance at low speeds. The roots blower absorbs power from the crankshaft of the engine, hence it reduces the power available at the flywheel. The larger the capacity of the Roots blower, the greater the loss in engine power output. Specific fuel consumption deteriorates. If the Roots blower is too small, scavenging at low speeds will be poor. If it is too large, then power will be wasted in developing an excessive boost pressure at high speeds and loads, where the turbocharger alone could provide sufficient boost.

Both the Roots blower and turbocharger must be matched to the engine together. Figures 10.32 to 10.34 show the performance obtained from an automotive two-stroke engine at three speeds, during turbocharger and Roots blower matching tests. At low speeds (figure 10.32), the larger Roots blower (85) substantially reduces specific fuel consumption and turbine inlet temperature. To achieve anything approaching the same performance with the smaller Roots blower (75), the turbocharger must be encouraged to do more work, by reducing turbine area (that is, increasing nozzle stator angle). At medium speeds, the differences in specific fuel consumption are smaller (figure 10.33). However, the

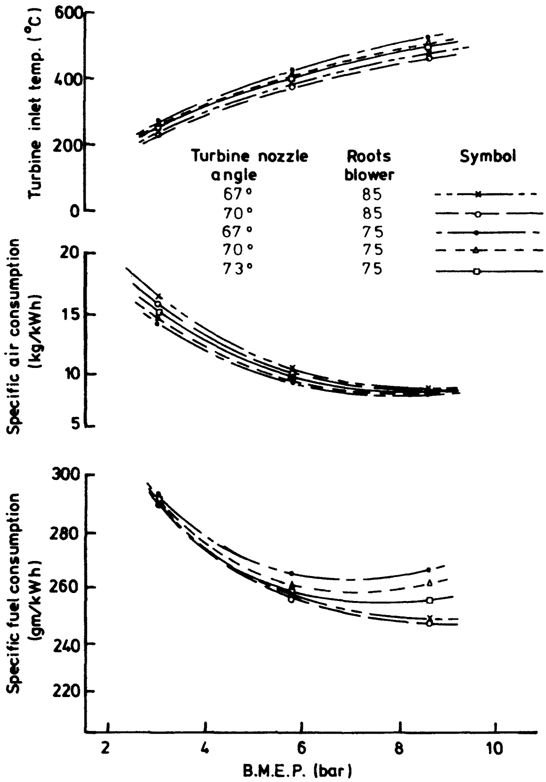

Figure 10.32 *Two-stroke turbocharger and Roots blower matching curves at 1000 rev/min*

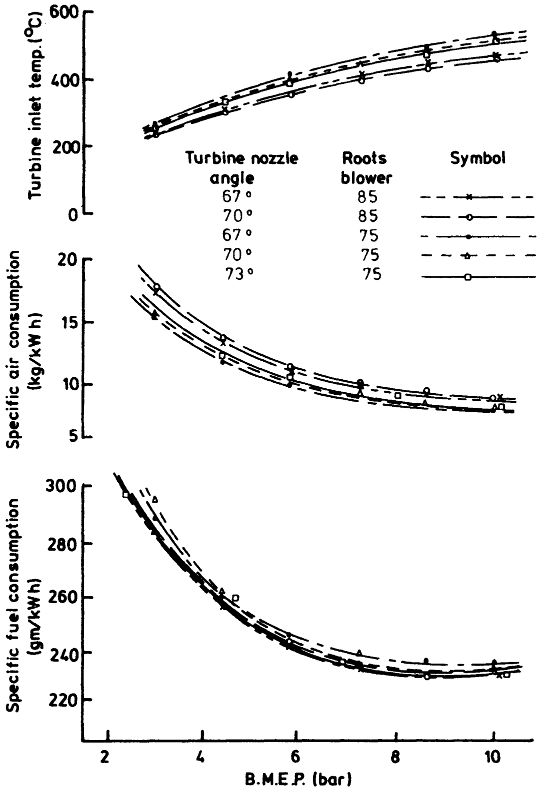

Figure 10.33 *Two-stroke turbocharger and Roots blower matching curves at 1500 rev/min*

air flow (or specific air consumption) is significantly greater with the larger Roots blower. Scavenging improves, but the gain is offset by the additional power absorbed by the Roots blower. At high speed (figure 10.34), the engine benefits little from the additional air flow from the larger blower, since the turbocharger is providing enough boost for good scavenging when the smaller blower is fitted. Thus a saving in blower power is reflected in superior fuel consumption. Any scavenging benefit obtained by fitting the larger blower is marginal and is not offset by its extra power requirement. Thus these figures illustrate the conflict between performance at high and low speed. A sensible compromise is to fit the larger Roots blower and the smallest turbine housing shown.

10.9 Changes in Ambient Conditions

Changes in inlet air density may be caused by ambient temperature and pressure changes at sea level, or operation of the engine at altitude. The change of air

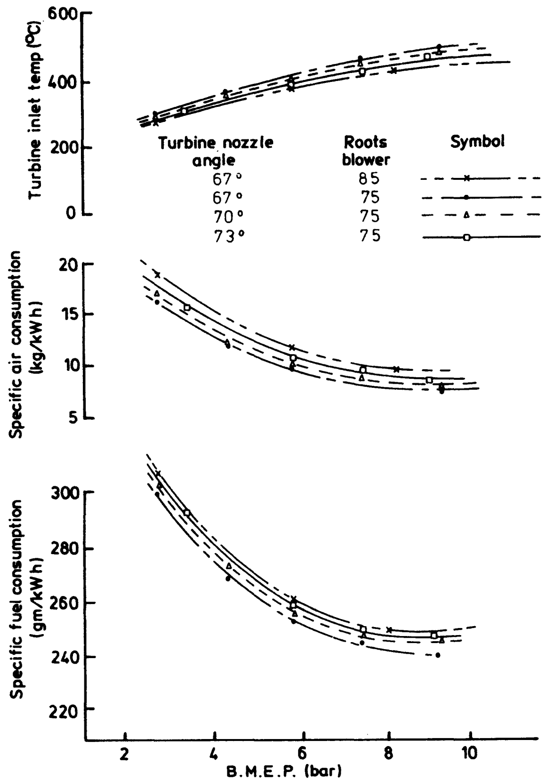

Figure 10.34 *Two-stroke turbocharger and Roots blower matching curves at 2000 rev/min*

flow may readily be predicted if the engine is naturally aspirated, but this is more difficult if the engine is turbocharged.

In mobile applications, such as a truck, the engine may be required to operate at sea level in a cold winter climate and perhaps at altitude during a hot summer. Thus the final compressor and turbine matches selected will be something of a compromise, particularly if the pressure ratio is high. The match selected will suit the normal operating environment of the engine, but with sufficient margins on surge, turbine inlet temperature and turbocharger speed to cover other conditions.

If the engine is designed for stationary applications, its operating altitude will be known, hence the manufacturer will have the option of rematching the turbocharger to suit the environment. The alternative will be to derate the engine for operation at altitude.

Although operation under changing ambient conditions introduces additional complications for the manufacturer of a turbocharged engine (such as reduction of the surge margin), the turbocharging system does offer partial compensation for reducing air inlet density at altitude. If variations in turbocharger efficiency are ignored for simplicity, the energy equation for constant pressure turbocharging (equation 6.1) may be written as

$$[(P_2/P_1)^{(\gamma_a-1)/\gamma_a} - 1] = K(T_3/T_1)\,[1 - (P_1/P_3)^{(\gamma_e-1)/\gamma_e}] \tag{10.8}$$

where K is a constant. As air density and therefore air mass flow rate reduce, so the turbine inlet temperature (T_3) will rise due to the richer air/fuel ratio. This means that the ratio of compressor to turbine pressure ratios will be altered in favour of the compressor. Its pressure ratio will increase, partially offsetting the reduction in air inlet density. It is also apparent from equation 10.8 that as ambient pressure (P_1) falls, so the expansion ratio of the turbine increases, raising compressor pressure ratio, provided that the turbine inlet pressure (P_3) does not fall at the same rate as ambient pressure.

An increase in ambient temperature (T_1) however, has an undesirable effect on the turbine to compressor energy balance, hence the turbocharger will tend to amplify the effect of such a change on air flow rate.

10.9.1 Operation under Changing Ambient Conditions, without Rematching

Large variations in ambient conditions can lead to problems due to compressor surge, excessive cylinder pressure, turbine inlet temperature, turbocharger speed or smoke emission. The actual performance of an engine under varying ambient conditions will depend on several factors which, for convenience of explanation, were assumed to be constant in the simplified analysis given above. For example, if air mass flow rate and compressor pressure ratio change, movement across the operating map of the compressor will be accompanied by an efficiency change. It follows that engines of similar performance at sea level will not necessarily perform comparatively at altitude. Techniques have been developed for accurately predicting the effect of varying ambient conditions (see chapter 15), but these require detailed turbine and compressor maps. A simpler, but less rigorous approach, is to correlate the performance of existing engines obtained when operating at altitude, or in very hot and very cold climates.

The parameter that limits engine performance will depend on the design of individual engines. At high ambient temperatures (figure 10.35), the limits are

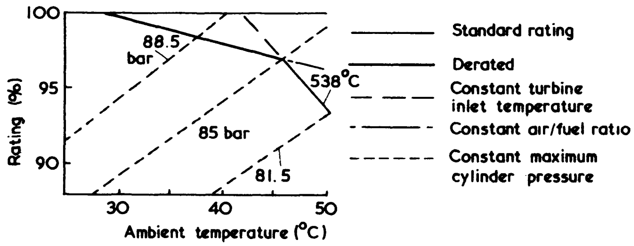

Figure 10.35 *Engine limitations at high ambient temperatures [8]*

likely to be smoke, due to reducing air flow, then turbine inlet or exhaust valve temperature or thermal loading of the engine. At low ambient temperature, maximum cylinder pressure or compressor surge (due to high pressure ratio, see equation 6.1) may be a limiting factor. It will be the limitations of a particular engine and turbocharger combination that will govern to what extent fuelling must be reduced, derating the engine for acceptable reliability or smoke emission. Low temperature, through its influence on compressor pressure ratio, can lead to compressor surge if the temperature change is very great.

Operation at altitude is usually, but not always, accompanied by a reduction in temperature. Through the combined effects of the rich fuel/air ratio on turbine inlet temperature and low turbine exit pressure, the turbocharging system offers partial compensation of the inlet air density reduction at altitude. Thus an engine may have to be derated, but not by as much as a naturally aspirated engine.

The effect of altitude on a turbocharged truck engine at full power, with and without intercooling, is shown in figure 10.36. Although the absolute inlet manifold pressure reduces with altitude, the reduction is smaller than that of ambient pressure (figure 10.37). Turbocharger speed increases due to the increase in turbine inlet temperature and expansion ratio. It can be seen that thermal limits and the maximum permissible turbocharger speed will be the limiting factors, particularly the latter. Movement towards the surge on the compressor map is shown in figure 10.38 for the non-intercooled engine, operating at part speed and load.

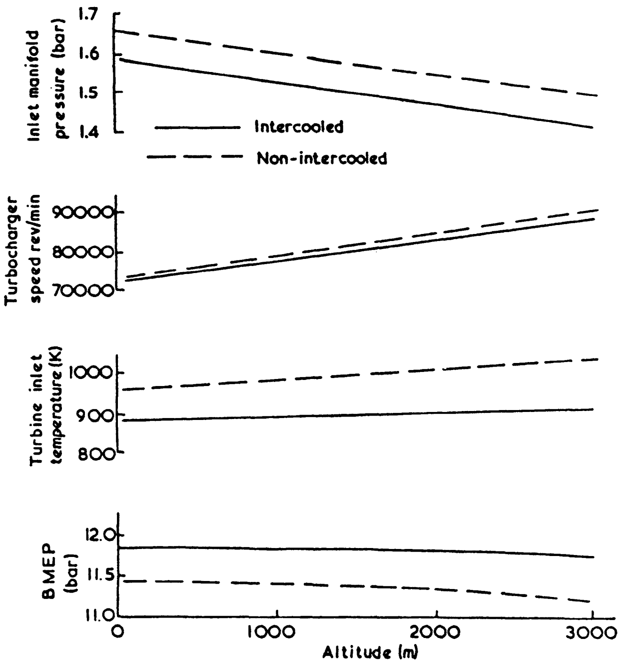

Figure 10.36 *The effect of altitude on truck engine performance*

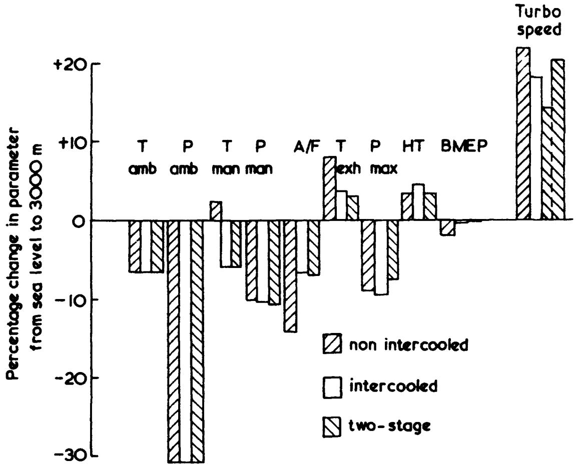

Figure 10.37 *Relative change in performance parameters at altitude*

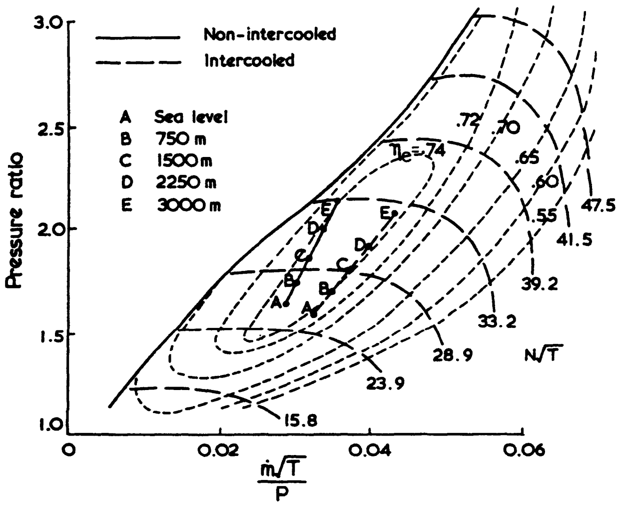

Figure 10.38 *Reduction in surge margin at altitude (part speed and load)*

If a stationary engine is not rematched for operation at altitude, then initially smoke emission, followed by turbocharger speed or inlet temperature will be the factors governing the reduction in fuelling and therefore rated power output, required. CIMAC (Conseil International des Machines à Combustion) recommend an empirical formula for derating at altitude, based on a limitation of constant turbine inlet temperature. The formula is

$$\dot{W}_{alt} = \dot{W}_{ref}\left[K - 0.7(1-K)\left(\frac{1}{\eta_{mech}} - 1\right)\right] \tag{10.9}$$

TABLE 10.2 *Exponents in CIMAC formula for derating at altitude*

Four-stroke Turbocharged Diesel Engines

	m	n	q
Without charge cooling	0.7	2.0	–
With charge cooling	0.7	1.2	1.0

where

$$K = \frac{\dot{W}_{\mathrm{alt(ind)}}}{\dot{W}_{\mathrm{ref(ind)}}} = \left(\frac{P_{\mathrm{a(alt)}} - hf\phi_{\mathrm{alt}}P_{\mathrm{svp}}}{P_{\mathrm{a(ref)}} - hf\phi_{\mathrm{ref}}P_{\mathrm{svp(ref)}}}\right)^m$$

$$\times \left(\frac{T_{\mathrm{ref}}}{T_{\mathrm{alt}}}\right)^n \left(\frac{T_{\mathrm{cool(ref)}}}{T_{\mathrm{cool(alt)}}}\right)^q$$

Values of the indices m, n and q are given in table 10.2.

10.9.2 Rematching to Suit Local Ambient Conditions

If the turbocharger is selected to suit the local ambient condition, additional density compensation can be provided in some cases. By reducing the turbine trim, more work can be extracted from the turbine enabling boost pressure to be raised, offsetting the loss in ambient density. This will, for example, delay the smoke limit or turbine inlet temperature limit to higher altitudes or ambient temperatures (compare figures 10.35 and 10.39). Thus derating may be avoided altogether or reduced. However, the influence of a more restrictive turbine on exhaust manifold pressure levels must be considered. Some engine performance deficit may occur.

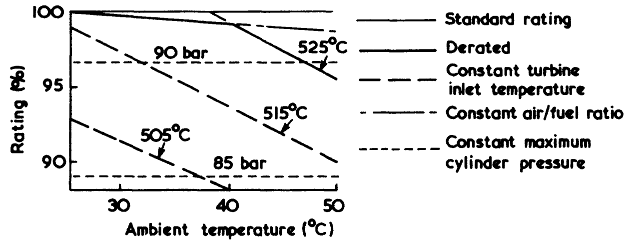

Figure 10.39 *Engine limitations at high ambient temperatures with appropriately matched turbocharger [8]*

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