3.3 CHARACTERISTIC CURVES OF PISTON ENGINES

The characteristic curves of a piston engine are very different from those of turbomachinery, as the principles of operation are quite different here.

The main quantities used are the mean effective pressure (or the power factor), the engine torque, rotation speed and specific fuel consumption, whose presentations vary from country to country. However, even if the quantities used vary according to the authors, the transition from one mode of representation to another is usually quite easy, as we will see.

3.3.1 EFFECTIVE PERFORMANCE, MEP AND POWER FACTOR

The work produced by a displacement machine in a reversible cycle can be calculated from expression: $W = -\int_c PdV$

The calculation of the integral giving the cycle work is complex and relies on detailed dimensional quantities, difficult to compare from one engine to another. Therefore, to facilitate comparisons, we introduce more accessible quantities: the displacement V_s and the mean pressure MP.

For all displacement machines, the concept of swept volume or displacement is indeed important because it represents a fundamental geometric characteristic of the machine. The ratio of the cycle work W to engine displacement has the dimension of a pressure. It is called mean pressure MP, or sometimes equivalent mean pressure difference ΔPe.

$$MP = \Delta Pe = \frac{\int_c PdV}{V_s} \qquad (3.3.1)$$

It represents the pressure that should be applied to the piston to produce the same work as that provided by the engine. This definition clearly shows the proportional relationship between pressure and the average work done by the engine.

We can also use a dimensionless power factor W_0, ratio of the MP to the engine inlet pressure P_1:

$$W_0 = \frac{MP}{P_1}$$

We get this way a simple expression of the work done by the machine:

$$W = W_0 \, P_1 \, V_s$$

In practice, one must take into account cycle irreversibilities, which leads as we shall see later to identify two mean pressures: the indicated mean effective pressure IMEP, and the mean effective pressure MEP.

3.3.2 INFLUENCE OF THE ROTATION SPEED

Let us call P_e the output power, ω the rotation speed, W_c the work produced by a cylinder during a cycle, n the number of cylinders, and i the intermittency factor, number of cycles per revolution, i.e. 0.5 for a 4-stroke engine, and 1 for a 2-stroke.

With these notations, we have:

$$P_e = \frac{\omega}{2\pi} W_c \cdot i \cdot n_c \qquad (3.3.2)$$

As $\omega = \frac{2\pi N}{60}$, we get, K being a constant:

$P_e = K N$

The motor torque is in turn given by equation:

$$C = \frac{P}{\omega} = \frac{W_c}{2\pi} \cdot i \cdot n_c = K' \qquad (3.3.3)$$

In a cyclic displacement machine, the output power should be at first approximation proportional to the rotation speed, while the torque and efficiency of the machine should not depend on it. In practice, however, things are not so simple, because of interference phenomena that occur, particularly at very low and very high speeds.

At low speeds, heat losses within the machine (wall effect, conduction losses etc.) and leak losses take values disproportionate with the useful power. Therefore, the efficiency, the work provided and the torque all fall.

At high speeds, another category of losses sees its influence grow: mainly internal losses and admission and discharge losses. These losses are substantially proportional to the square of the fluid velocity, which explains why they become dominant.

Under these conditions, the laws of evolution of the engine power and torque relative to the rotation speed take the general appearance of Figures 3.3.1 and 3.3.2.

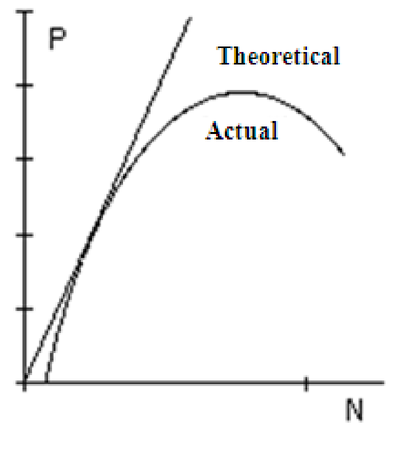

Figure 3.3.1: Power function of speed

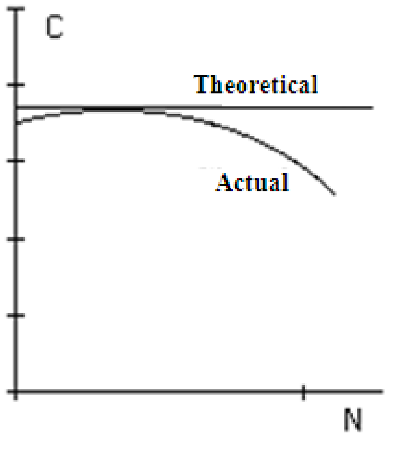

Figure 3.3.2: Torque function of speed

As a first approximation, and in most speeds encountered, we can consider that the relationship between power and speed is almost proportional. As power is also proportional to the MEP and the power factor, we can in practice use interchangeably as abscissa of the characteristic curves one of the four magnitudes P, MEP, W_0 or N, which explains that presentations may differ from one country to another.

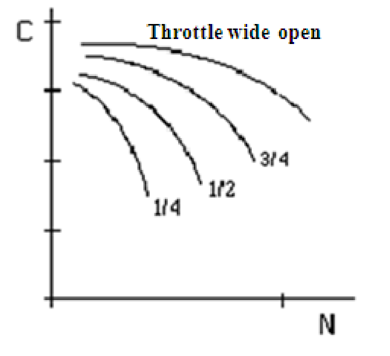

Figure 3.3.3: Gasoline part-load torque

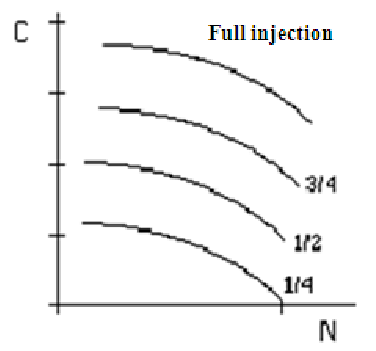

Figure 3.3.4: Diesel part-load torque

Moreover, these curves are only valid at full load. Part-load operation results in significant differences depending on whether it is a gasoline or a diesel engine. In the first case, we shall see that it is impossible to reduce richness below a lower limit R_i, which necessitates adjusting the engine by varying the pressure drop at the admission, by means of a valve which just strangles the gas stream admitted. It is customary in this case to give the "torque-speed" curves for different values of throttle opening, the corresponding pressure drop playing a major role.

For diesel engines, this type of problem does not arise, and we can control the power by varying the amount of fuel injected. It is not necessary to introduce an additional pressure drop, and torque characteristics are much less steep than in the previous case, as shown in Figures 3.3.3 and 3.3.4.

3.3.3 INDICATED PERFORMANCE, IMEP

We have so far neglected losses of mechanical origin, that are often taken into account, as shown in section 7.1.6 of Part 2, by means of a mechanical efficiency η_m. In section 3.3.4 we will show that to be accurate, it is preferable to operate differently.

We can therefore consider that the effective power P_e is the product of the internal or indicated power P_i by the mechanical efficiency η_m:

$$P_e = \eta_m P_i$$

We adopt thus the convention to identify the thermodynamic proper performance by index i, and the overall performance by index e.

Under these conditions, we define the indicated mean effective pressure IMEP:

$$\text{IMEP} = \frac{W_{ci}}{V_s} = \frac{\text{MEP}}{\eta_m} \qquad (3.3.4)$$

3.3.3.1 Volumetric efficiency

A number of factors are reducing the mass of fresh gas entering the cylinder during the intake phase: the residual presence of smoke, the high temperature of the walls, which has the effect of dilating the fresh gas, admission losses, which cause the pressure in the cylinder to be slightly less than atmospheric pressure, and finally the exhaust counter-pressure. Moreover, acoustic phenomena related to the charge pulsating flow may, depending on the case, either decrease it (known as filling holes), or instead increase it (this is called natural supercharging). This pulsating flow naturally stems from the reciprocation of the piston and the opening and closing of valves.

To characterize these losses, we define a volumetric efficiency C_r, such that the mass of the fresh gas charge admitted by cycle is equal to:

$$m_a = C_r \rho_a V_s = C_r V_s \frac{P_a}{r T_a} \qquad (3.3.5)$$

We can show that, in the absence of acoustic phenomena that we have just mentioned, if f is the residual gas rate, ρ the volumetric compression ratio, and if admission conditions are identified by index 1, the volumetric efficiency is given by:

$$C_r = (1 - f) \frac{\rho}{\rho - 1} \frac{r_a}{r_1} \frac{T_a}{T_1} \frac{P_1}{P_a} \qquad (3.3.6)$$

In practice, C_r is about 0.80 to 0.85 for a gasoline engine, and around 0.9 for a diesel engine.

3.3.3.2 Determination of the indicated mean effective pressure

Moreover, introducing richness R and the air fuel ratio AFR for stoichiometric combustion, we can write:

$m_c = m_a$ R/AFR, and the amount of heat released by combustion is equal to:

$Q_c = m_c$ LHV = R/AFR LHV m_a

In practice, for conventional fuels, we have, both for gasoline and diesel:

$$\frac{LHV}{AFR} \approx \frac{42}{14} = 3 \text{ MJ/kg.}$$

The indicated efficiency η_i is equal to the ratio of indicated work per cycle W_{ci} to Q_c:

$$\eta_i = \frac{AFR\ W_{ci}}{LHV\ R\ m_a} \qquad (3.3.7)$$

By replacing m_a by its value, we obtain for W_{ci}:

$$W_{ci} = \eta_i C_r V_s \frac{LHV}{AFR} R \frac{P_a}{r T_a} \qquad (3.3.8)$$

By definition of IMEP, we get : IMEP = $\frac{W_{ci}}{V_s}$

$$IMEP = \eta_i C_r \frac{LHV}{AFR} R \frac{P_a}{r T_a} \qquad (3.3.9)$$