普通高等教育汽车类专业“十二五”规划教材

UTO MOBILE

Automobile Engine in Theory

汽车发动机原理

主编 吴建华 王军

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前　　言

汽车工业是我国的支柱产业，至 2011 年我国的汽车保有量已逾一亿辆，随着汽车工业的高速发展，急需大量的设计和制造方面的人才，这也带动了各高等院校车辆工程专业的急速发展。另外，在目前经济全球化的大背景下，对外交流日益增多，需要越来越多的既掌握专业知识又具有相当英语水平的人才，基于这种需求趋势，教育部于 2001 年颁布了《本科教学工作提高教学质量的若干意见》，要求各高校积极推动使用英语教学，其中“本科教育要创造条件使用英语等外语进行公共课和专业课教学”。自教育部明确提出该项要求以来，国内大部分高校都先后展开本科课程的双语教学工作，但适合双语教学的教材十分缺乏，基于此，由淮阴工学院、常熟理工学院、江苏技术师范学院和福建工程学院的相关教师共同编写了这本发动机原理英文教材。

本书沿用了吴建华主编《汽车发动机原理》的基本构架，讲述了发动机的工作过程。主要内容包括发动机的性能指标、发动机的换气过程、发动机废气涡轮增压、燃料与燃烧化学、柴油机混合气的形成与燃烧、汽油机混合气的形成与燃烧、发动机特性、发动机排放与噪声等。

本书由淮阴工学院吴建华和王军主编；淮阴工学院常绿，江苏技术师范学院张兰春，淮阴工学院王程，常熟理工学院孟杰任副主编；福建工程学院张庆永，淮阴工学院陈勇、常熟理工学院陈庆樟参与了本书编写。

本书按 30～50 学时编写，可作为车辆工程专业本科发动机原理双语教材，也可供从事发动机设计、制造和运用的工程技术人员参考。

本书引用了有关教科书、内燃机专业期刊的许多资料，在此一并表示感谢。

由于编者水平有限，错误在所难免，欢迎读者批评指正。

CONTENTS

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Chapter 1 Engine Thermodynamics and Performance Parameters

This chapter provides criteria by which to judge the performance of internal combustion engines. Most important are the thermodynamic cycles based on ideal gases undergoing ideal processes. However, internal combustion engines follow a mechanical cycle, not a thermodynamic cycle. The start and end points are mechanically the same in the cycle for an internal combustion engine, whether it is a two-stroke or four-stroke mechanical cycle.

By studying the operating processes of mechanical cycle and thermodynamic cycle, we can find general rules to improve the performances of internal combustion engines.

1.1 Ideal air standard cycles

The internal combustion engine is a non-cyclic, open-circuit, quasi steady-flow, work-producing device. None the less it is very convenient to compare internal combustion engines with the ideal air standard cycles, as they are a simple basis for comparison. This can be justified by arguing that the main constituent of the working fluid, nitrogen, remains virtually unchanged in the processes. Air standard cycles have limitations as air and, in particular, air/fuel mixtures do not behave as ideal gases. Despite this, the simple air standard cycles are very useful, as they indicate trends. Most important is the trend that as compression ratio increases cycle efficiency should also increase.

Whether an internal combustion engine operates on a two-stroke or four-stroke cycle and whether it uses spark ignition or compression ignition, it follows a mechanical cycle not a thermodynamic cycle. However, the thermal efficiency of such an engine is assessed by comparison with the thermal efficiency of air standard cycles, because of the similarity between the engine indicator diagram and the state diagram of the corresponding hypothetical cycle. The engine indicator diagram is the record of pressure against cylinder volume, recorded from an actual engine. Pressure/volume diagrams are very useful, as the enclosed area equates to the work in the cycle.

1.1.1 The ideal air standard Otto cycle

The Otto cycle is usually used as a basis of comparison for spark ignition and high-speed compression ignition engines. The cycle consists of four non-flow processes, as shown in Fig. 1.1 (a). The compression and expansion processes are assumed to be adiabatic (no heat transfer) and reversible, and thus isentropic. The processes are as follows:

a-c isentropic compression of air through a volume ratio V_a/V_c, the compression ratioε.

c-z addition of heat Q_1 at constant volume.

z-b isentropic expansion of air to the original volume.

b-a rejection of heat Q_2 at constant volume to complete the cycle.

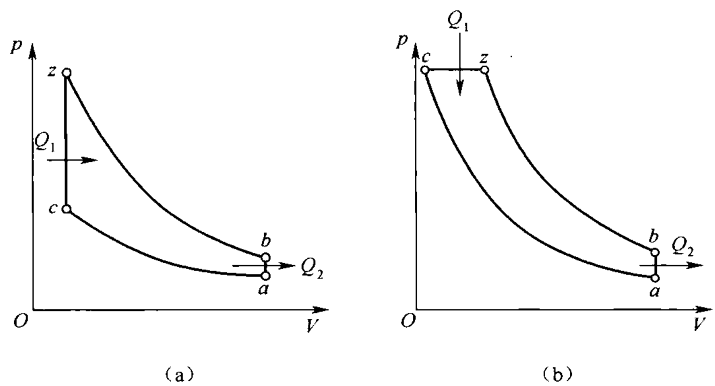

Fig. 1.1 *p-V* indicator diagram of ideal air standard Otto cycle and Diesel cycle
(a) Otto cycle; (b) Diesel cycle.

1.1.2 The ideal air standard Diesel cycle

The Diesel cycle has heat addition at constant pressure, instead of heat addition at constant volume as in the Otto cycle. With the combination of high compression ratio, to cause selfignition of the fuel, and constant - volume combustion the peak pressures can be very high. In large compression ignition engines, such as marine engines, fuel injection is arranged so that combustion occurs at approximately constant pressure in order to limit the peak pressures.

The four non-flow processes constituting the cycle are shown in the state diagram (Fig. 1.1 (b)). The processes are all reversible, and are as follows:

a-c isentropic compression of air through a volume ratio V_a/V_b, the compression ratio ε.

c-z addition of heat Q_1 at constant pressure while the volume expands through a ratio V_c/V_z, the load or cut-off ration.

z-b isentropic expansion of air to the original volume.

b-a rejection of heat Q_2 at constant volume to complete the cycle.

1.1.3 The ideal air standard Dual cycle

In practice, combustion occurs neither at constant volume nor at constant pressure. This leads to the Dual, Limited Pressure, or Mixed cycle which has heat addition in two stages, firstly at constant volume, and secondly at constant pressure. The state diagram is shown in Fig. 1.2, again all processes are assumed to be reversible. As might be expected, the efficiency lies between that of the Otto cycle and the Diesel cycle.

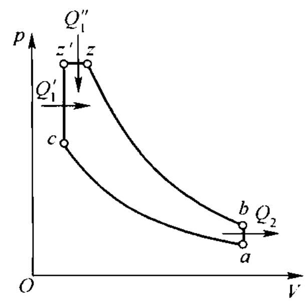

Fig. 1.2 p-V indicator diagram of ideal air standard Dual cycles

1.1.4 Cycle fuel conversion efficiency

The cycle fuel conversion efficiency η_t is used to evaluate the cycle economy, which is defined as:

$$\eta_t = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \tag{1.1}$$

Where W = work output;

Q_1 = cycle heat addition;

Q_2 = cycle heat rejection.

The cycle thermal efficiency of standard Dual cycle can be derived by thermodynamic equation:

$$\eta_{dual} = 1 - \frac{1}{\varepsilon_c^{\kappa-1}} \frac{\lambda_p \rho_0^{\kappa} - 1}{(\lambda_p - 1) + \kappa \lambda_p (\rho_0 - 1)} \tag{1.2}$$

Where ε_c = compression ratio, $\varepsilon_c = V_a/V_c = (V_s + V_c)/V_c = 1 + V_s/V_c$, in which V_a = cylinder displacement, V_c = combustion chamber clearance, V_s = swept volume; ρ_0 = volumetric expansion ratio during constant-pressure heat addition, $\rho_0 = V_z/V'_z$; λ_p = pressure ratio during constant-volume heat addition, $\lambda_p = p_z/p_c$; κ = isentropic exponent.

For $\rho_0 = 1$, the thermal efficiency of Otto cycle can be derived:

$$\eta_{otto} = 1 - \frac{1}{\varepsilon_c^{\kappa-1}}$$

For $\lambda_p = 1$, the thermal efficiency of Diesel cycle can be derived:

$$\eta_{diesel} = 1 - \frac{1}{\varepsilon_c^{\kappa-1}} \frac{\rho_0^{\kappa-1}}{\kappa(\rho_0 - 1)} \tag{1.3}$$

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We can draw a conclusion by eq (1.2) that η_t increases with the increasing of compression ratio ε_c, pressure ratio λ_p, isentropic exponent K and decreases with the increasing of expansion ratio ρ_0.

1.1.5 Mean effective pressure

Mean effective pressure p_t (MPa) is cycle work done by unit cylinder volume. It is used to evaluate the power capability of a cylinder and is defined as follow:

$$p_t = \frac{W}{V_s}$$

Where W (kJ) = cycle work done by the cylinder;

V_s(L) = swept volume.

We can also deduce the mean effective pressure of Dual cycle by thermodynamic equation:

$$p_t = \frac{\varepsilon_c^\kappa}{\varepsilon_c - 1} \frac{p_{de}}{\kappa - 1}[(\lambda_p - 1) + \kappa\lambda_p(\rho_0 - 1)]\eta_t \tag{1.4}$$

Where, p_{de} = is cylinder pressure when inlet valve is closed.

In Eq. (1.4), if $\rho_0 = 1$, the mean effective pressure of Otto cycle can be obtained:

$$p_t = \frac{\varepsilon_c^\kappa}{\varepsilon_c} \frac{p_{de}}{\kappa - 1}(\lambda_p - 1)\eta_t$$

If $\lambda_p = 1$, the mean effective pressure of Diesel cycle can be obtained:

$$p_t = \frac{\varepsilon_c^\kappa}{\varepsilon_c} \frac{p_{de}}{\kappa - 1}\kappa(\rho_0 - 1)\eta_t$$

From Eq. (1.4), a conclusion can be drawn that p_t increases with increasing of pressure p_{de}, compression ratio ε_c, expansion ratio ρ_0, isentropic exponent K and cycle thermal efficiency η_t.

1.1.6 Comparison of three basic ideal cycles

The important variable factors which are used as a basis for comparison of the cycles are as following: compression ratio, maximum pressure, heat supplied and rejected, net work.

Some of the above mentioned variables are fixed when the performance of Otto, Diesel and Dual combustion cycles is to be compared.

1. Efficiency versus compression ratio

Fig. 1.3 shows the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles at various compression ratios and with given cut off ratio for the Diesel and Dual combustion cycles. It is evident from the Fig. 1.3 that the air standard efficiencies increase with the increase in the compression ratio. For a given compression ratio Otto cycle is the most efficient while the Diesel cycle is the least efficient. ($\eta_{otto} > \eta_{dtal} > \eta_{diesel}$).

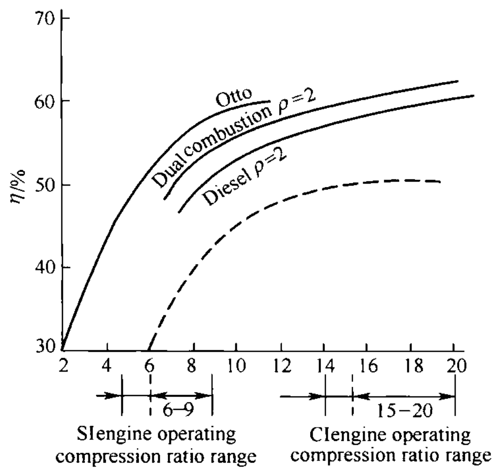


Fig. 1. 3 Comparison of efficiency at various compression ratio

2. For the same compression ratio and the same heat input

A compression of the cycles (Otto, Dual and Diesel) on the *T-S* diagram for the same compression ratio and heat supplied is shown in Fig. 1. 4.

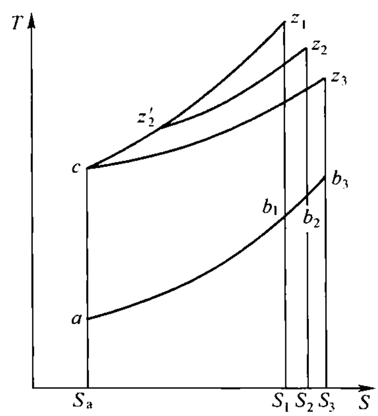


Fig. 1. 4 *T-S* diagram of the three cycles for the same compression ratio

We know that, $\eta_t = 1 - \dfrac{Q_2}{Q_1}$, where Q_2 is the heat rejected and Q_1 is the heat supplied. Since all the cycles reject their heat at the same specific volume, process line from state 4 to 1, the quantity of heat rejected from each cycle is represented by the appropriate area under the line 4 tol on the *T-S* diagram. As is evident from the above equation, the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and the Diesel cycle is the least efficient of the three cycles, i. e. $\eta_{otto} > \eta_{dual} > \eta_{\text{diesel}}$.

3. For constant maximum pressure and heat supplied

Fig. 1. 5 shows the three cycles *T-S* diagram for constant maximum pressure and heat input respectively.

For the constant maximum pressure, the points z_1, z_2, z_3 must lie on a constant pressure

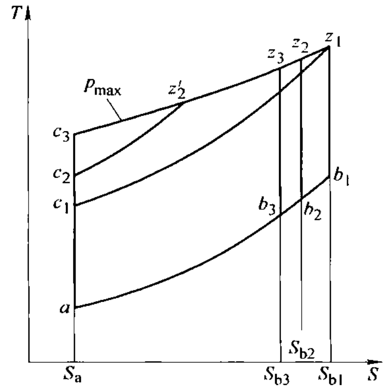


Fig. 1.5 *T-S* diagram of the three cycles for constant maximum pressure

line. On *T-S* diagram the heat rejected from the Diesel cycle is represented by the area under the line b_3-a, and the heat rejected from the Otto cycle is the area under the line b_1-a. Hence the Diesel cycle is the most efficient and the Otto cycle is the least efficient, i. e., $\eta_{otto} < \eta_{dual} < \eta_{diesel}$.

1.2 Real cycles of four-stroke engine

1.2.1 Real cycle *p-V* diagram of four-stroke engine

The preceding discussion covered the theoretical combustion cycles which serve as the basis for modern engines. In actual operation, modern engines operate on modifications of the theoretical cycles. However, some characteristics of the true cycles are incorporated in the actual cycles of modern engines, as you will see in the following discussion of the actual cycles of operation in gasoline and diesel engines. A mechanical four-stroke cycle consists of four strokes as follow:

1. Intake stroke

The first stroke in the sequence is the intake stroke. During this stroke, the piston is moving downward and the intake valve is open. This downward movement of the piston produces a partial vacuum in the cylinder, and air and fuel rush into the cylinder past the open intake valve. This action produces a result similar to that which occurs when you drink through a straw. You produce a partial vacuum in your mouth, and the liquid moves up through the straw to fill the vacuum.

2. Compression stroke

When the piston reaches bottom dead center at the end of the intake stroke (and is therefore at the bottom of the cylinder) the intake valve closes and seals the upper end of the cylinder. As the crankshaft continues to rotate, it pushes the connecting rod up against the piston. The piston then moves upward and compresses the combustible mixture in the cylinder. This action is known as the compression stroke. In gasoline engines, the mixture is compressed to about one-eighth of its original volume. (In a diesel engine the mixture may be compressed to as little as one-sixteenth of its original volume.) This compression of the air-fuel mixture increases the pressure within the cylinder. Compressing the mixture in this way makes it more combustible; not only does the pressure in the cylinder go up, but the temperature of the mixture also increases.

3. Power stroke

As the piston reaches top dead center at the end of the compression stroke (and is therefore at the top of the cylinder), the ignition system produces an electric spark. The spark sets fire to the fuel-air mixture. In burning, the mixture gets very hot and expands in all directions. The pressure rises to about 600 to 700 pounds per square inch. Since the piston is the only part that can move, the force produced by the expanding gases forces the piston down. This force, or thrust, is carried through the connecting rod to the crankpin on the crankshaft. The crankshaft is given a powerful twist. This is known as the power stroke.

4. Exhaust stroke

After the fuel-air mixture has burned, it must be cleared from the cylinder. Therefore, the exhaust valve opens as the power stroke is finished and the piston starts back up on the exhaust stroke. The piston forces the burned gases of the cylinder past the open exhaust valve. The four strokes (intake, compression, power, and exhaust) are continuously repeated as the engine runs.

An example of a pressure-volume diagram for a modified four-stroke engine is shown in Fig. 1.6. Some of the events are exaggerated to show more clearly the changes that take place.

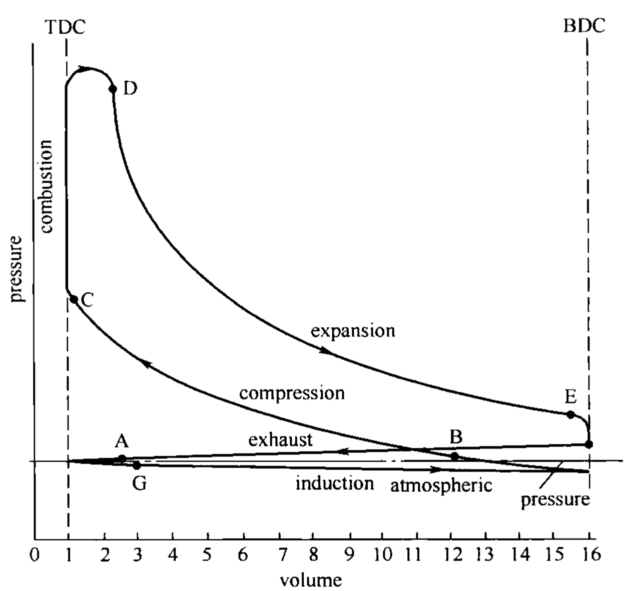
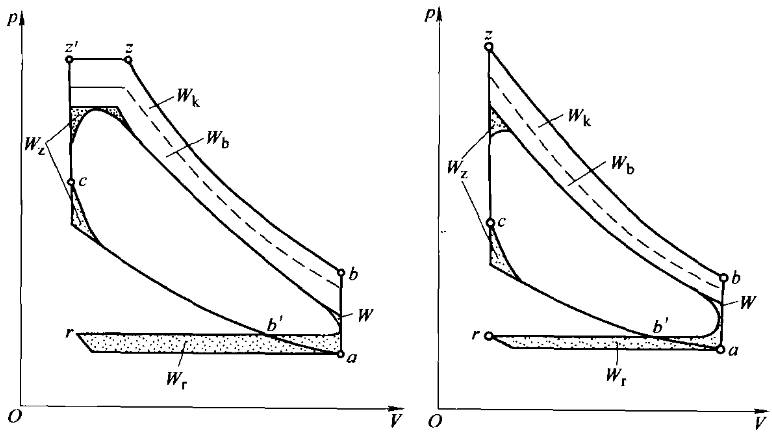


Fig. 1.6 p-V indicator diagram for a real four-stroke cycle

The inlet valve is open at point A, fresh charge is inhaled into cylinder with the piston downward movement. Compression stroke begins when pistons go upwards. Notice that the volume line (bottom of figure) is divided into 16 units. These units indicate a 16:1 compression ratio. The higher compression ratio accounts for the increased temperature necessary for ignition of the charge. Fuel is injected (diesel engine) or a spark produced (gasoline engine) at point C and combustion is represented by line CD. During this period of time, there is a sharp increase in pressure until the piston reaches a point slightly past TDC. Then, combustion continues at a relatively constant pressure which drops slightly as combustion ends at point D. The burning gas pushes the piston down complete the expansion stroke. The exhaust valves open at point E and the burnt gas are pushed out the cylinder until the exhaust valves closed at point G.

1.2.2 Comparison with ideal cycles

A comparison of a real engine p-V diagram over the four strokes with an equivalent fuel-air cycle analysis is shown in Fig. 1.7.



(a) (b)

Fig. 1.7 Comparison of real cycle with ideal cycles

(a) Diesel engine; (b) Gasoline engine.

The real engine and the fuel-air cycle have the same geometric compression ratio, fuel chemical composition and equivalence ratio, residual fraction and mixture density before compression. Modest differences in pressure during intake and the early part of the compression process result from the pressure drop across the intake valve during the intake process and the closing of the intake valve 40 to 60°CA after BDC in the real engine. The expansion stroke pressures for the engine fall below the fuel-air cycle pressures for the following reasons: (1) heat transfer from the burned gases to the walls; (2) finite time required to burn the charge; (3) incomplete combustion of the charge; (4) exhaust blowdown loss due to opening the exhaust valve before BDC; (5) gas flow into crevice regions and leakage past the piston rings; (6) pumping loss.

These differences, in decreasing order of importance, are described below. Together, they contribute to the enclosed area on the p-V diagram for a properly adjusted engine with optimum timing being about 80 percent of the enclosed area of an equivalent fuel-air cycle p-V diagram. The indicated fuel conversion or availability conversion efficiency of the actual engine is therefore about 0.8 times the efficiency calculated for the ideal cycle. Use is often made of this ratio to estimate the performance of actual engines from ideal cycle results.

1. Heat transfer (W_b)

Heat transfer from the unburned mixture to the cylinder walls has a negligible effect on the p-V line for the compression process. Heat transfer from the burned gases is much more important. Due to heat transfer during combustion, the pressure at the end of combustion in the real cycle will be lower. During expansion, heat transfer will cause the gas pressure in the real cycle to fall below an isentropic expansion line as the volume increases. A decrease in efficiency results from this heat loss.

2. Finite combustion time (W_z)

In an SI engine with spark-timing adjusted for optimum efficiency, combustion typically starts 10 to 40 crank angle degrees before TDC, is half complete at about 10°CA after TDC, and is essentially complete 30 to 40°CA after TDC. Peak pressure occurs at about 15°CA after TDC. In a diesel engine, the burning process starts shortly before TDC. The pressure rises rapidly to a peak some 5 to 10°CA after TDC since the initial rate of burning is fast. However, the final stages of burning are much slower, and combustion continues until 40 to 50°CA after TDC. Thus, the peak pressure in the engine is substantially below the ideal cycle peak pressure value, because combustion continues until well after TDC, when the cylinder volume is much greater than the clearance volume. After peak pressure, expansion stroke pressures in the engine are higher than fuel-air cycle values in the absence of other loss mechanisms, because less work has been extracted from the cylinder gases. A comparison of the constant-volume and Dual cycles in Fig. 1.4 demonstrates this.

For spark or fuel-injection timing which is retarded from the optimum for maximum efficiency, the peak pressure in the real cycle will be lower, and expansion stroke pressures after the peak pressure will be higher than in the optimum timing cycle.

3. Incomplete combustion (W_z)

Combustion of the cylinder charge is incomplete; the exhaust gases contain combustible species. For example, in spark-ignition engines the hydrocarbon emissions from a warmed-up engine (which come largely from the crevice regions) are 2 to 3 percent of the fuel mass under normal operating conditions; carbon monoxide and hydrogen in the exhaust contain an additional 1 to 2 percent or more of the fuel energy, even with excess air present. Hence, the chemical energy of the fuel which is released in the actual engine is about 5 percent less than the chemi-

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