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THE TRANSFORMATION OF HEAT IN AN ENGINE

By Kurt Neumann

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1. The thermodynamic basis for rating heat engines.- The production of work by a heat engine rests on the operation of supplying heat, under favorable conditions, to a working fluid and then taking it away. The working fluid serves simply as a conveyor of energy: It must flow to the engine and then, after its energy has been transformed, it must be returned again to the surroundings. As working fluids, steam and gases are employed.

The goal of every technical process for obtaining work is to secure the maximum amount of work L_n from a given quantity of heat Q, and this by the simplest constructive device possible. The efficiency coefficient of the process is then,

$$\eta = \frac{L_n}{Q} .$$

The efficiency obtained at present in the utilization of heat energy in engines is about 40%.

*Translation in part of Chapter 9, Volume XI, of Geiger and Scheel's Handbuch der Physik, Julius Springer, Berlin, 1926. This chapter deals with the transformation of heat in the engine. The division of the chapter translated deals with those engines whose operation is based on the liberation of chemical energy - Internal Combustion Engines. The treatment of internal combustion engines from the standpoint of classical thermodynamics is important, for it forms a necessary introduction to their further investigation that must include a consideration of the thermodynamics (and kinetics) of the gaseous explosive reaction on which their operation depends.

The efficiency coefficient given above, sometimes called the economic coefficient, is not well adapted to express the rating of a heat engine. From the standpoint of thermodynamics this coefficient can never equal 1. In order to follow the heat transformation in the engine and obtain an expression for its efficiency coefficient based on the possibilities of a perfect (free from heat losses) engine, a rating known as the thermodynamic efficiency coefficient is used. This is the ratio between the work of the actual engine L_n and the work of the perfect (free from heat losses) engine L. That is,

$$\eta_{\text{thermodyn.}} = \frac{L_n}{L} \leqq 1,$$

and since the loss of work between the perfect and the actual engine is $L_v = L - L_n$, the proportion of the work lost is

$$\zeta = \frac{L_v}{L} = 1 - \eta_{\text{thermodyn.}}$$

That is, a knowledge of the thermodynamic efficiency coefficient gives at once the proportion of the work to be won, the differences between it and its limiting value 1, and the degree of work loss of the heat engine. The aim must be to make the value of ζ as small as possible. The smaller ζ, the better the mechanical construction of the engine, the more appropriate has been the means employed to meet in its construction the ends indicated by theoretical considerations to attain maximum efficiency.

The determination of the actual processes described of a heat engine whose work is expressed in terms of the thermodynamic efficiency coefficient may be carried out with any degree of precision depending only on the accuracy of the instruments employed and the precision with which the measurements are made. Of greater difficulty is it to determine the magnitude of the maximum work of the perfect engine - the value in the denominator in the ratio.

The second law of thermodynamics shows the way to proceed in order to obtain the greatest possible amount of work. At the same time it makes it possible to see that the losses that occur in the transformation processes are of two kinds, viz., avoidable and unavoidable losses.

As stated above, the working fluid is introduced into the engine. After its energy transformation into work the fluid is returned to the surroundings. The process is completed when the temperature T_0 and the pressure P_0 has become that of the surroundings. The surroundings are, therefore, one system with which we have to deal in calculating the output in work of the engine.

Indicate by P the pressure of the working fluid in the engine,
V its volume,
U its internal energy,
E its flow-energy,
S its entropy,

and indicate by one prime mark (') the initial condition of these factors and by two prime marks (") their final condition. If we designate by Q and Q_0 the quantity of heat reversibly absorbed or given out at the respective temperatures T_1 and T_0, then, from the first law

$$U' + PV' + E' + Q_1 = L_n + U'' + PV'' + E'' + Q_0.$$

Since

$$U + PV = J,$$

$$L_n = (J' + E') - (J'' + E'') + (Q_1 - Q_0)$$

According to the second law the total entropy of all substances involved in the transformation must increase. The increase in entropy is

$$\Delta S = S'' - S' - \frac{Q_1}{T_1} + \frac{Q_0}{T_0}.$$

From both these equations it follows

$$L_n = (J' + E') - (J'' + E'') + Q_1 \frac{T_1 - T_0}{T_1} + T_0 (S'' - S') - T_0 \Delta S.$$

If the initial and end condition of the working fluid is known, ΔS may be calculated. The increase of entropy ΔS is a measure of the nonreversibility of the energy transformation. The larger the value of ΔS the smaller the amount of work that may be obtained.

For the ideal case - the reversible transformation process - $\Delta S = 0$ and the greatest amount possible of work has been obtained from the quantity of heat transformed in the engine;

$$L_n = L_{max}.$$

The actual available work to be obtained from the engine is therefore

$$L_n = L_{max} - T_o \Delta S.$$

These equations lead to the following results:

In order to obtain a high work output of the perfect engine it is necessary that the flow energy E'' leaving the engine with the working fluid expelled be as small as possible. Likewise all nonreversible processes, since they involve an entropy increase, must be avoided. The loss in work

$$L_v = L_{max} - L_n$$
$$= T_o \Delta S,$$

which results in such cases from nonreversible processes is, measured in calories, equal to the product of the absolute temperature of the surroundings by the entropy increase due to that part of the transformation process not reversible.

If, from practical considerations, a heat exchange betweem the transforming system and the surroundings cannot be avoided, then it must suffice to bring the system reversibly to the same pressure as the surroundings.

For the special case of the reversible cycle process, where $\Delta S = 0$, $J'' = J'$, $S'' = S'$, and E'' and E' negligible, it follows from the general equation

$$L_n = Q_1 \frac{T_1 - T_o}{T_1} \quad \text{or,} \quad \eta = \frac{L_n}{Q_1} = 1 - \frac{T_o}{T_1},$$

that, from a given quantity of heat delivered to the engine Q_1

and at a given lower temperature T_o, the amount of work obtainable from the transformation is the greater the higher the temperature of the heat admitted to the engine.

This initial temperature, however, cannot be increased indefinitely, since at high temperatures dissociation of the molecules of the working fluid takes place. At this point, where dissociation intervenes, the upper limit of temperature for the working fluid is reached. At this temperature combustion is considered to take place reversibly.

If the working fluid conveyed to the engine receives no further addition of heat Q_1 at temperature T_1, as is usually the case, then, with E' and $E'' = 0$

$$L_n = J' - J'' + T_o (S'' - S') - T_o \Delta S;$$

or, for the ideal reversible process, $\Delta S = 0$,

$$L_{max} = J' - J'' + T_o (S'' - S').$$

For the difference between the initial and final heat content of the working fluid $J' - J''$, the heat value of the gaseous mixture for constant pressure may be substituted H_m; so that the maximum work obtainable from the combustion of the gaseous mixture may be expressed as

$$L_{max} = H_m + T_o (S'' - S').$$

It is to be seen also that besides the heat content of the gaseous mixture (heat of reaction) the entropy change of the working fluid corresponding to maximum work is also indicated. Only when

$S'' = S'$ and $S'' - S' = 0$ is $L_{max} = H_m$ and $\eta_{thermodyn.} = \frac{L_n}{H_m}$.
But since no substances are known where the entropy difference $S'' - S'$ is really different from 0, it is justifiable to write $L_{max} = H_m$.

Sometimes, as in the case of engine and boiler, it is possible to express the thermodynamic efficiency of the engine in terms of its indicated performance:

Let B represent the quantity of heat obtained from the combustion of coal,

h_u the heat value of 1 kg coal,

Q the quantity of heat delivered to the engine,

L_n the available work,

L_i the indicated work, and

L the work of the perfect engine.

$$\eta_{thermodyn.} = \frac{L_i}{L}$$

the following relation exists:

$$\frac{L_n}{Bh_u} = \frac{Q}{Bh_u}\,\frac{L}{Q}\,\frac{L_i}{L}\,\frac{L_n}{L_i}$$

or

$$\eta = \eta_k\,\eta_v\,\eta_{thermodyn.}\,\eta_m;$$

that is, the economic efficiency coefficient η, appears as the product of four factors: The efficiency factor of combustion, $\eta_k = \frac{Q}{Bh_u}$, the efficiency factor of the perfect engine $\eta = \frac{L}{Q}$, the thermodynamic efficiency factor $\eta_{thermodyn.} = \frac{L_i}{L}$ and the mechanical efficiency factor $\eta_m = \frac{L_n}{L_i}$.

Of the unavoidable sources of loss that occur as a result of nonreversible processes in the engine, the most important are friction, throttling, heat losses at containing walls, changes in the chemical and physical properties of the working fluid, nonreversible burning processes and in general a working process not completely united in a single closed system.

The requirement that the working fluid of the engine be returned to the surroundings under the same conditions of temperature and pressure of the surroundings would require the utilization of the heat energy still remaining in the working fluid as usually expelled from the engine. The utilization of this residual heat explains the high efficiency coefficient obtained by a direct union of power and heat systems.

Practical application and simplicity of construction often make it desirable to depart from what may be known to be theoretically the more economical practice. Progress in the use of heat energy to obtain work becomes the more difficult as the improvement in engine design and operation advances. By quantitative estimation of the various transformation processes of the actual engine it is possible to determine the magnitude of the separate loss sources that go to make up the total heat losses of the engine and in this way it may be determined in which direction efforts toward improvement may be easiest and most efficiently made.

Engines with Liberation of Chemical Energy - Internal Combustion Engines

(a) The Thermal Process

5. General remarks on Internal Combustion Engines.- The combustion in the engine cylinder is brought about by the ignition of combustible gas-air mixtures (gas engines) or oil vapor-air mixtures (Diesel engines). The ignition may take place by introducing into the combustion chamber a quantity of heat (ignition spark) or heat by adiabatic compression. In either case it is necessary that the rate at which the heat of reaction is liberated must exceed the rate at which it loses heat in order that the explosive form of the combustion process may continue itself within the combustible mixture.

The continuation of the ignition process within the combustible mixture depends on certain conditions of its composition. When the rate of propagation of the reaction zone becomes null the limit of the explosive form of transformation is reached. Between the lower and upper explosive limit (deficient air or excess of air) the velocity of propagation has its maximum.

Ignition temperature and explosive limits are not characteristic constants of any given fuel mixture. They depend intimately upon the external conditions imposed upon the reaction, as size and form of the container, (position and nature of ignition, pressure, initial temperature, etc.).

The propagation of the combustion (zone) within the combustible mixture may take place in two different ways:

1. Principally by heat conduction. (By this method the rate of propagation is relatively slow.)

2. By compression waves (in which case the rate of propagation is very high).

Only the first mode of propagation - that by heat conduction - will here be considered in reference to the thermodynamics of internal combustion engines.

The linear rate of flame propagation in a hydrogen-air mixture has been observed to be about 14 m/s by method 1.

The linear rate of flame movement by method 2 has been observed at 2800 m/s in an H_2, O_2 mixture.

Mixtures of the permanent gases are the hardest to ignite, requiring temperatures ranging from 600°C to 700°C. Vapors of liquid fuels with air ignite at lower temperatures, benzine and ether around 400°C, and some of the hydrocarbons as low as 300°C.

Of technical importance the combustion of carbon, hydrogen and sulphur are the principal elements that need be considered. These elements as fuels seldom exist as such in the pure state. As fuels they exist for the most part in highly complicated combinations. But when completely burned they yield as combustion products CO_2, H_2O and SO_2.

6. Air requirement. Composition of combustion products.- Equations for combustion reactions (the proportions are expressed in molecules or in cubic units).

1. Carbon

$$C + O_2 = CO_2 + 97700 \text{ cal/mol } C$$
$$C + \tfrac{1}{2}O_2 = CO + 29300 \quad "$$

2. Hydrogen

$$H_2 + \tfrac{1}{2}O_2 = H_2O_{\text{liquid}} + 68200 \text{ cal/mol } H_2$$
$$H_2 + \tfrac{1}{2}O_2 = H_2O_{\text{steam}} + 57400 \quad " \quad H_2$$

3. Carbon monoxide

$$CO + \tfrac{1}{2}O_2 = CO_2 + 68400 \text{ cal/mol } CO,$$

4. Hydrocarbons (m, n = 1, 2, 3,...):

$$C_m H_n + \left(m + \frac{n}{4}\right) O_2 = m\ CO_2 + \frac{n}{2} H_2O + Q \text{ cal/mol } C_m H_n.$$

For one part each, at 15°C and 1 atm., of a given gaseous mixture, $CO + H_2 + CH_4 + C_2H_4 + CO_2 + N_2 = 1\ m^3$, the theoretical amount of oxygen necessary for complete combustion will be

$$O_{2\,min} = \frac{CO + H_2}{2} + 2\ CH_4 + 3\ C_2H_4\ m^3$$

and the necessary amount of air required would be

$$Air_{min} = \frac{O_{2\,min}}{0.21}\ m^3$$

As combustion products from gas and air there will result:

$$CO_2'' = CO + CH_4 + 2\ C_2H_4 + CO_2\ m^3$$
$$H_2O'' = H_2 + 2\ CH_4 + 2\ C_2H_4\ m^3$$
$$N_2'' = 0.79\ air_{min} + N_2\ m^3$$

If the above mixture is burned with $air > air_{min}$ (excess of air $\lambda = \frac{air}{air_{min}}$, then free oxygen should occur in the combustion products,

$$O_2'' = 0.21\ air - O_{2_{min}}\ m^3$$
$$N_2'' = 0.79\ air + N_2\ m^3$$

If the combustion involves a solid or liquid fuel of given composition

$$c + h + o + w = 1\ kg$$

then according to its stoichiometric equation for complete oxidation with excess of air $(\lambda > 1)$

$$1\ kg\ fuel + \lambda\ air_{min}\ m^3\ air = 24.4\ \frac{c}{12}\ m^3\ CO'' + 24.4\left(\frac{h}{2} + \frac{w}{18}\right) m^3\ H_2O''$$
$$+ 0.21\ (\lambda - 1)\ air_{min}\ m^3\ O_2'' + 0.79\ \lambda\ air_{min}\ m^3\ N_2'',$$

from which $air_{min} = \frac{24.4}{12 \times 0.21} \left[c + 3\left(h - \frac{O}{8}\right)\right] m^3$ as the amount of air required.

In general combustion results in a change in the number of molecules. This changes the gas constant R and the volume constant. Besides the change produced in the number of molecules by the reaction, a physical contraction with energy transformation may also result in the case of the combustion of hydrogen or of a hydrocarbon if the gaseous products are cooled below the boiling point of water.

7. The generation of heat by combustion. Combustion temperature.- The amount of heat given out by the complete combustion of one kilogram or one cubic meter of substance whose combustion products are again brought to initial temperature is called the heat value of the substance. If in cooling the products of combustion to the initial temperature the water vapor present is condensed, the heat value obtained is called the higher heat value and is designated by H. If at the initial temperature the water vapor is not condensed, the heat value so obtained is called the lower heat value and is designated by H_u. Between the values of H and H_u the following relation exists:

$$H_u = H - 600 \text{ w cal},$$

where w signifies the weight in kilograms of water formed from the burning of one unit of the fuel.

When burned in an engine the change in condition of the working fluid takes place either at a constant volume or constant pressure. According to the first law (See Figure 1)

$$Q_{1\,2} = U_2 - U_1 + \int_1^2 P\,dV$$

or

$$Q_{1\,2} = J_2 - J_1 - \int_1^2 V\,dP.$$

The theoretical temperature of combustion may be calculated for the condition of constant volume V = const. and on the assumption of no heat losses during the transformation $Q_{1\,2} = 0$ and that $U_2 = U_1$. It may be calculated for the condition of constant pressure p = const., from $J_2 = J_1$. For the first condition, V = const., the internal energy of the system remains constant. For the second condition the heat content of the system is constant.

The process of combustion under conditions of constant volume is represented in the U,t-diagram (Figure 1).

From $dU = c_V' \, dt$ it follows that t_1 at initial temperature for the gaseous mixture before combustion is

$$U_1 = U_0' + c_V' \; t_1$$

after combustion and at temperature t_2

$$U_2 = U_0'' + \int_0^{t_2} c_V'' \; dt$$

Putting $U_0' - U_0'' = Q_V^0$, the quantity of heat liberated by the reaction, referred to 0^0, and because $U_1 = U_2$,

$$Q_V^0 + c_V' \; t_1 = \int_0^{t_2} c_V'' \; dt,$$

introducing the average specific heat $[c_V'']_{t_1}^{t_2}$.

$$t_2 = t_1 + \frac{Q_V^{t_1}}{[c_V'']_{t_1}^{t_2}} \; {}^oC.$$

This says that the temperature of combustion attainable by a gaseous transformation depends on the initial temperature of the mixture, its heat value (of combustion, Q), and on the mean value of the specific heats of the products of combustion.

Barring heat losses, the maximum pressure attained will be

$$p_2 = p_1 \frac{R''}{R'} \frac{T_2}{T_1} \text{ atm.}$$

(b) Work Process of the Internal Combustion Engine

8. <u>The work process in general</u>. The main difference between the internal combustion and the steam engine is that the combustion process is carried out within the cylinder of the internal combustion engine. The transfer of heat to the working fluid takes place in this case without the intervention of heated surfaces. A chemical transformation takes place. The high temperatures resulting from the combustion process make it necessary to cool the walls of the engine cylinder.

The upper temperature range in the combustion engine is of an order around 2000°. It is much higher than cylinder temperature reached in the steam engine. For this reason internal combustion engines have higher efficiency coefficients than steam engines. The work process - whether for four-stroke-cycle engines or two-stroke-cycle engines - is for both cases, in reference to the heat transformations within the cylinder, practically alike so that a mutual consideration in reference to the working process of the perfect engine is permissible.

Concerning the question of maximum work under the conditions stated below, the determination of the necessary factors for its expression in the case of internal combustion engines is unusually complicated. If the theoretical PV-diagram of the engine is to be developed with some approach to accuracy, the course of the combustion in the engine must be determined from laws governing fuel supply and the rate of molecular transformatiom of the explosive gases in the cylinder. During transformation of the gases and during expansion, the degree of dissociation of the working fluid is also to be taken into account. Our present knowledge of these processes as they run their course under the conditions of the working engine is not sufficient for an accurate determination of the necessary factors to construct an accurate PV-diagram showing the theoretical maximum work. Concerning the rate of molecular transformation within the cylinder and its relation to the various conditions imposed by the working engine there exists as yet no reliable basis for its determination.

Lacking more intimate and accurate data it has been the custom to substitute for the actual processes occurring in the engine an approximate estimation of the work process based on an imagined ideal process in which it is assumed that the engine cylinder encloses air. On this assumption a theoretical, reversible work cycle is carried out, heat being supplied from the surroundings or withdrawn by the surroundings. On these assumptions PV- and ST-diagrams of a working cycle are easily constructed to represent by analogy the work processes of an internal combustion engine.

9. Work processes of the perfect engine.- All engines operate with precompression of the working fluid, since the thermal efficiency coefficient η_t increases with increase of compression. In order to obtain maximum work under the conditions assumed the following procedure is described: (See Figures 2 & 3)

G kilograms of the mixture at P_0 and T_0 occupy a volume $V_1 = \frac{GRT_0}{P_0}$ m^3. Adiabatic compression 1 2 to p_2, combustion 2 3 corresponds to PV^n = const. Adiabatic expansion 3 4 to T_0 and isothermal compression 4 1 to P_0 then

$$L_{max} = Q - Q_0 = G \left\{ [c_n]_{T_2}^{T_3} (T_3 - T_2) - T_0 \int_{T_2}^{T_3} \frac{c_n}{T} dT \right\} \text{cal.}$$

It is not possible to carry out this process in a cylinder; expansion cannot take place to P_0 (point 4' on the figures), hence the loss indicated by the surface 4'41. So that

$$L_{max} = G \left\{ [c_n]_{T_2}^{T_3} (T_3 - T_2) - [c_p]_{T_0}^{T_4'} (T_4' - T_0) \right\} \text{cal.}$$

The difficulty in the computation lies in the determination of the combustion line 2 3. Accordingly, a step further in the assumptions already stated is made, viz., that the expansion proceeds only to pressure $p_4 > p_0$. That is, there is incomplete expansion.

For the special case, - combustion at constant volume - V = const. (the gas engine), Figure 4 shows a theoretical indicator card of the engine. For the points indicated on the diagram, the following relations may be given:

Point 1. $T_1 = T_0$ $\qquad p_1 = p_0$

" 2. $T_2 = T_1 \left|\frac{v_k + v_h}{v_k}\right|^{k-1}$, $\qquad p_2 = p_1 \left(\frac{v_k + v_h}{v_k}\right)^k$

" 3. $T_3 = T_2 + \frac{Q_v}{[c_v'']_{T_2}^{T_3}}$, $\qquad p_3 = p_2 \frac{R''}{R'} \frac{T_3}{T_2}$

$$\text{Point 4.}\quad T_4 = T_3 \left(\frac{v_k}{v_k + v_h}\right)^{k-1}, \quad p_4 = p_3 \left(\frac{v_k}{v_k + v_h}\right)^k,$$

In these expressions the heat of the mixture is $Q_v = \frac{}{1 -}$ cal/m³ and Hu the lower heat value of the gases, k is the ratio of specific heats.

For engines working at constant pressure (Diesel) the heat of combustion is delivered under condition of P = const. Figure 5 the characteristic points 3 and 4 are given

$$\text{Point 3.}\quad T_3 = T_2 + \frac{Q_p}{[c_p''] \frac{T_3}{T_2}} \qquad p_3 = p_2, \quad V_3 = \frac{GRT_3}{P_2},$$

$$\text{Point 4.}\quad T_4 = T_3 \left(\frac{v_3}{v_k + v_h}\right)^{k=1}, \quad p_4 = p_2 \left|\frac{v_3}{v_k + v_h}\right|^k$$

For both kinds of engine, it may be seen from Figures 4 and 5, the thermal efficiency coefficient is

$$\eta_t = 1 - \frac{\psi\, \varphi^k - 1}{\epsilon^{k-1}[(\psi - 1) + k\psi(\varphi - 1)]}$$

where $\epsilon = \frac{v_k + v_h}{v_k}$ is the compression ratio $\varphi = \frac{v_k + v_e}{v_k}$ the injection ratio and ψ the pressure ratio produced by the combustion.

For $v_e = 0$ or $\varphi = 1$, η_t for the gas engine is given by

$$\eta_t = 1 - \left|\frac{p_1}{p_2}\right|^{\frac{k-1}{k}}$$

and for $p_3 = p_2$ or $\psi = 1$ for the constant pressure engine (Diesel)

$$\eta_t = 1 - \left|\frac{p_1}{p_2}\right|^{\frac{k-1}{k}} \frac{1}{k} \frac{\varphi^k - 1}{\varphi - 1}.$$

In the second case the efficiency coefficient depends on the final compression pressure as well as on the injection ratio φ.

The following table gives examples of the efficiency coefficient of gas and Diesel engines:

Gas engine, $p_1 = 1$ atm.; $k = 1.36$

$p_2 =$	3	4	5	6	8	10 atm. abs.
$\eta_t =$	0.25	0.31	0.35	0.38	0.42	0.42

Diesel(const. press. engines) $p_1 = 1$ atm. $p_2 = 35$ atm. $k = 1.35$

$\varphi =$	4	3	2	1
$\eta_t =$	0.57	0.61	0.64	0.69

These values are only comparative and approximate since the assumptions underlying the processes on which they are based do not accurately correspond to the actual processes taking place. (Reversible transfer of heat from the surroundings is substituted for nonreversible combustion and condition changes of the working fluid at V = const. and P = const.)

In combustion turbine engines expansion proceeds to the point of initial pressure p_0. Combustion takes place beforehand in a special chamber. In its course through the rotors of the turbine the flow energy of the working fluid is transformed into mechanical work.

Figures 6 and 7 show PV-diagrams for combustion turbines working under conditions of combustion at constant volume, V = const. and constant-pressure. P = const. Both operate on precompression of the working fluid.

10. Work process of the actual engine.- The limit of precompression of the working fluid p_2 is determined by the ignition temperature of the charge. For engines that introduce the charge by suction, the compression temperature t_2 must lie below the ignition temperature of the explosive mixture. For engines that introduce the air by suction (Diesel engines) the temperature of compression must lie above the ignition temperature of the mixture. To the first class belong gas and light oil engines. To the second class engines using heavy oil where the fuel must be introduced into the cylinder by an injector.

The indicator diagram of the actual engine differs from that of the theoretical engine in consequence of various loss sources: Throttling at intake and outlet of the working fluid; heat losses at the cylinder walls; final combustion velocity; after-burning during expansion.

The economic efficiency coefficient $\eta = \frac{632\ N_e}{G\ H_u}$ may also for this case be resolved into partial efficiency coefficients: The thermal efficiency coefficient of an ideal engine working without heat losses η_v, the indicated efficiency coefficient η_i and the mechanical efficiency coefficient η_m.

$$\eta = \eta_v\ \eta_i\ \eta_m = \frac{632\ N_i^o}{G\ H_u}\ \frac{N_i}{N_i^o}\ \frac{N_e}{N_i}\ .$$

The maximum value of η attained, according to the work process and the precision with which it is carried out, is 38%.

Heat transformation in the internal combustion engine is primarily influenced by

a) The formation of the combustible mixture (mechanical process);

b) Combustion (thermodynamic process).

An intake valve is provided at the head of the cylinder. The determination of the mixture ratios, mixture of fuel and air and the expulsion of combustion products is effected by the pistons' action in four-stroke-cycle operations. Special pumps for gas and air are provided for two-stroke-cycle operations.

Four-stroke-cycle engines are more efficient than two-stroke-cycle because the time for charging and for removing combustion products is shorter for two-stroke-cycle engines. But for large engines the two-stroke-cycle procedure is preferable, especially for oil injection forms since charging losses are in this case smaller.

Intake and outlet are controlled by valves in four-stroke-cycle engines; in two-stroke-cycle engines the outlet is controlled by valves; the intake sometimes by the piston which opens slits in the cylinder walls.

Figures 8 and 9 show PV-diagrams of simple four-stroke-cycle and two-stroke-cycle engines.

Engines that introduce the charge by suction.- The compressed charge is fired at or near the dead center of the piston. The heat value of the combustion mixture, $\frac{H_u}{1-\lambda\ air_{min}}$ influences the rate of combustion, the maximum temperature attained, and the resulting pressure (gas engines).

With increasing excess of air, $\frac{H_u}{1-\lambda\ air_{min}}$ becomes smaller, the rate of combustion slower, maximum temperature and pressure decrease. For $\lambda = 1$, the heat value of the mixture is greatest also pressure and temperature, if the combustion is assumed to be complete. The indicator diagram shows a characteristic pointed apex.

Thermal efficiency is the best for highly compressed lean mixtures. For these conditions pressure increase is moderate and heat losses are small owing to lower combustion temperatures.

Values of combustion heats for mixtures of gases and air for theoretical complete combustion ($\lambda = 1$), lie between 500 and 700 cal/m³. The richer the gas, the greater the amount of air necessary.

For the more important kinds of gas the values in the following table are given.

Combustion Heats of a Few Gases

Gas	Combustion heat cal/m³	Theoretical air amount m³	Combustion heat of mixture cal/m³
Illuminating gas	4590	5.21	736
Water gas	2300	2.15	730
Generator gas	1095	1.00	550
Furnace gas	883	0.76	500

The heat value of the burned mixture is usually between 450 and 550 cal/m³.

The value of the combustion heat may be varied between wide limits by varying the mixture ratio of gas and air. With light oils the limits are narrower. The limits between which it is possible to maintain a zone of explosive reaction within the gaseous mixture limits the range of possible mixture ratios.

Table showing the Heat Distribution for
a 50 HP. Generator Gas Engine.
H_u = 1198 cal/m³) Experimental Values.

Thermal efficiency coefficient relative to	Indicated performance	η_{t_i}	= 0.322
	Effective work	η_{t_e}	= 0.222
Heat losses by	Cooling water	Q_k	= 0.374
	Exhaust gas	Q_z	= 0.239
	Radiation, etc.	Q_r	= 0.065
		$\eta_{t_i} + Q_k + Q_z + Q_r$	= 1.000

According to the experimental results shown in the above table, 22.2% of the heat energy delivered to the engine was returned in effective work.

For the resulting performance:

1. The weight of the charge introduced by suction is

$$G_1 = \frac{P_1 \ (v_k + v_h)}{R' \ T_1} \text{ kg}$$

2. The heat value of the mixture,

$$Q = \frac{H_u}{1 - \lambda \ air_{min}} \text{ cal/m}^3$$

The control of those engines charged by suction is effected either by

1. Changing the quantity of the charge taken in G_1: The heat of combustion of the mixture remains constant. Due to throttling the final compression falls with decreasing charge and with it the efficiency coefficient.

2. By changing the ratio of fuel and air: The heat of combustion is variable; the end compression constant, since the cylinder operates on a full charge of the gaseous mixture. As the limit is approached, the energy content of the charge becomes poorer and the ability to support a reaction zone less. This procedure is therefore limited. Therefore, for control

		By filling	By mixture ratio
Quantity of fuel	G	variable	variable
" " air	air	"	constant
Mixture ratio	$\frac{air}{G}$	constant	variable

For large engines, control by change of mixture ratio is the most favorable method. Small engines are controlled most generally by control of the amount of the fuel mixture taken into the cylinder.

Figures 10 and 11 show PV-diagrams for both partial filling and mixture ratio control.

Supercharging of the engine may be accomplished by increasing the weight of air taken into the cylinder. This is done by providing a compressor in connection with the cylinder head and increasing with the increase in the amount of air, the weight of fuel also admitted. The increase of heat attending this practice soon places a limit to the procedure.

Engines based upon oil injection draw in and compress pure air. The oil is injected at or near the dead center of the piston. This procedure is especially applicable to the use of heavy oils.

At moderate compression pressures, the fuel is injected into a hot bulb. At higher compression pressures the temperature of the compressed air is sufficient to ignite the oil vapor-air mixture.

Hot bulb engines are mostly constructed for two-stroke-cycle processes; Diesel engines for four-stroke-cycle processes.

The combustion of heavy oils in the engine cylinder is not yet well understood. The liquid fuel must be atomized, vaporized, mixed with air and burned. These operations overlap to some extent. The ignition perhaps continues itself through the production of lighter hydrocarbons from a liquid hydrocarbon.

Table Giving Properties of Principal Heavy Oils Used in Internal Combustion Engines.

Gas oil from		p e a t	bituminous coal
Composition:	C =	85.4 to 87.6 G.T.	87.1 to 91.4 G.T.
	H =	9.8 " 12.6	6.0 " 7.8
	O + N =	0.8 " 3.2	1.4 " 4.9
	S =	0.4 " 1.6	0.4 " 0.9
Heat value:		9400 " 10100 cal/kg	8800 " 9100 cal/kg
Specific weight:		0.82 " 0.96	1.00 " 1.10

The time for effecting the mixture and combustion of the oil is very short in Diesel engines. For an engine running at $n = 180$ R.P.M., and for a spray angle $\alpha = 30^\circ$, the time of burning is only $z = \frac{60}{n} \frac{\alpha}{360} = 0.028$ sec. The high compression pressure (30 to 40 atmospheres) and large air excess ($\lambda > 1.5$) tend to decrease unfavorable counter effects and give a good heat efficiency (η_{te}, as high as 0.35).

Simple injection of the fuel into the combustion chamber of the engine is not enough in general to insure complete combustion within the necessarily short time limit.

The classical Diesel engine operated with fuel-air injection. As a consequence there followed good atomization of the fuel and a violent turbulence of the gases in the combustion chamber by which means favorable combustion resulted.

Recent practice injects the fuel without air admixture and at high pressure (200 to 300 atmospheres) directly into the cylinder (radiating injection) or, into an ignition chamber (ignition chamber engine). In this chamber a partial combustion of the fuel takes place and owing to the excess of pressure so produced forces the fuel in the form of spray into the cylinder. Sometimes the fuel is mixed at lower pressures with the air strongly agitated by a device in connection with the working piston.

By eliminating the cooling effect of air injection engines employing radiating injection and those operating without compressor may function at low initial compression (20 to 25 atmospheres). They are characterized by sure firing and also by reduced speed, greater ignition space and higher supercharging. (Consult Figure 12.)

Smokeless combustion and low fuel consumption result from correct design of hot-bulb engines. They are specially adapted to the use of heavy oils, since the injection process is not sensitive.

Table Showing Heat Distribution of a Four-Stroke-Cycle Diesel Engine with Air Injection (N_e = 50 HP., n = 216 R.P.M.)

Transformed into indicated work		η_{ti}	= 0.41	in fractions of heat delivered to engine
Heat loss through	cooling water	Q_k	= 0.26	
	exhaust gases	Q_z	= 0.32	
	radiation, etc.	Q_r	= 0.01	

Heat exchange between working fluid and cylinder walls during a working process was found to be

For fuel intake	Q_{51}	= +0.013	in fractions of total heat loss
" compression	Q_{12}	= +0.046	
" combustion and expansion	Q_{24}	= −0.676	
" exhaust	Q_{45}	= −0.383	

It is seen that the principal heat loss occurs during burning and expansion.

Heat distribution for a compressorless four-stroke-cycle Diesel engine with ignition chamber (N_e = 300 HP., n = 160 R.P.M.)

Transformed into indicated work		ηt_i	= 0.44	in fractions of total heat delivered to engine
Heat losses	cooling water	Q_k	= 0.28	
	exhaust gases	Q_z	= 0.25	
	radiation, etc.	Q_r	= 0.03	

The control of solid injection engines is effected by changing the amount of oil introduced. The maximum charge is determined by the weight of air enclosed in the cylinder, by which the quantity of oil introduced is completely oxidized. Supercharging may also be effected by previously compressing the air introduced into the cylinder.

Progress in the utilization of energy by heat engines is probably best shown by a diagrammatic balance sheet from which comparison of the heat balance of steam, gas and Diesel engines may be made (See Figure 13).

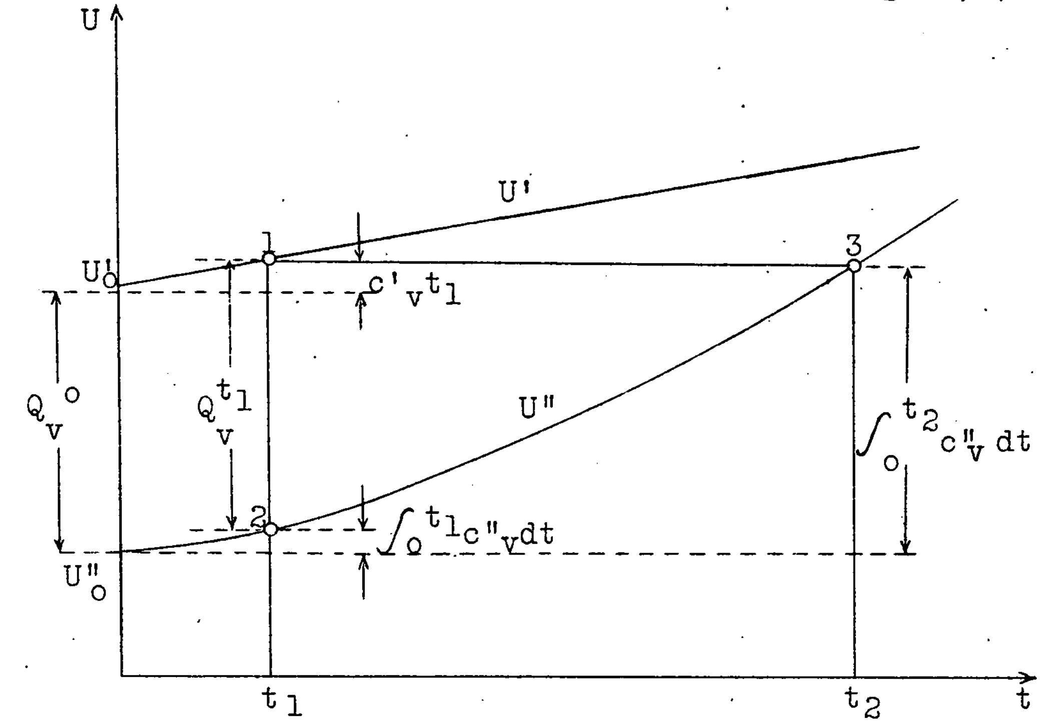
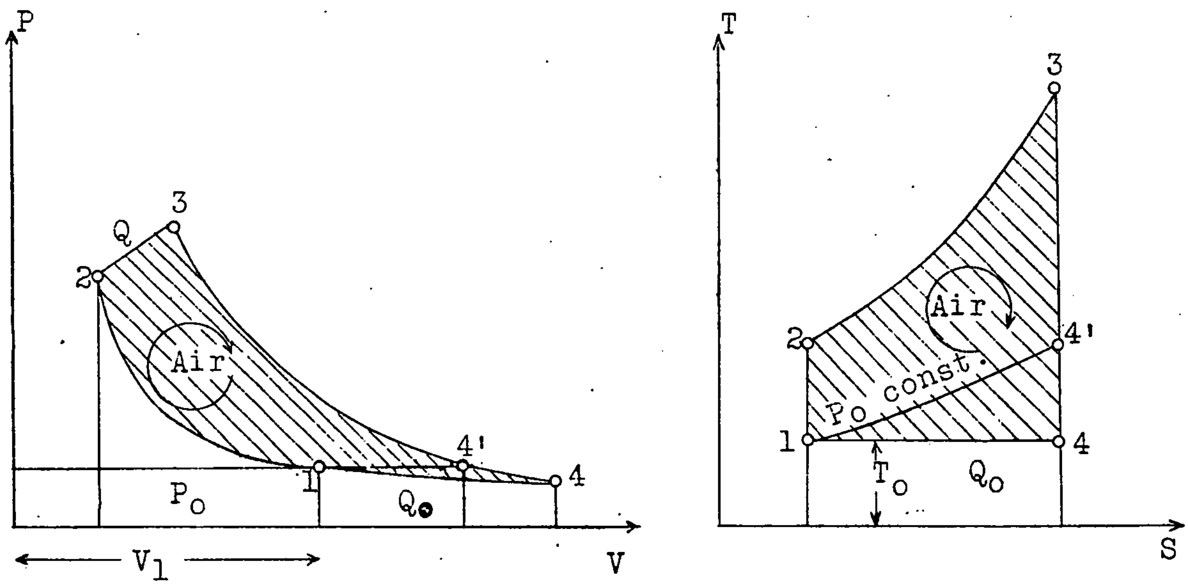


Fig.1 Determination of combustion temperature for combustion at constant volume.



Figures 2 & 3 showing working cycle of ideal engine in PV - ST - diagrams.

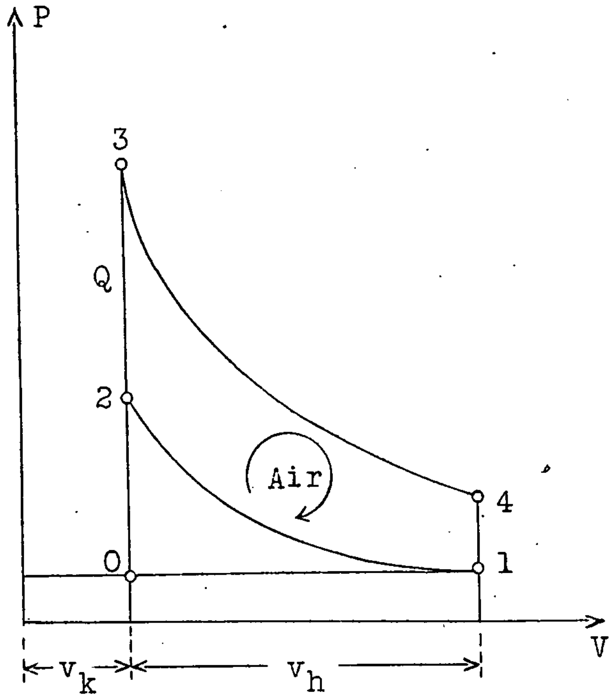


Fig.4 PV - Graph of constant volume engine.

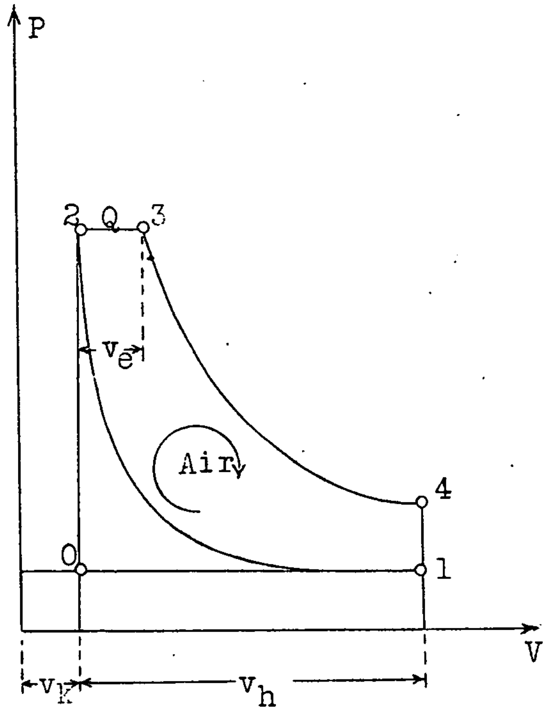


Fig.5 PV - Graph of constant pressure engine.

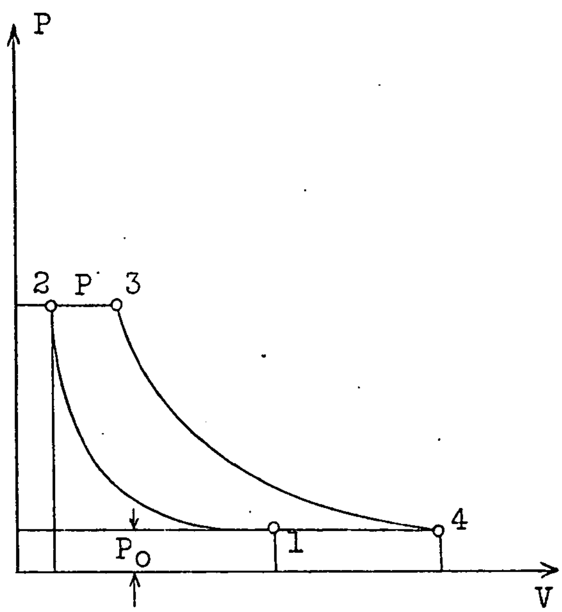


Fig.6 Pressure-volume diagram constant pressure turbine.

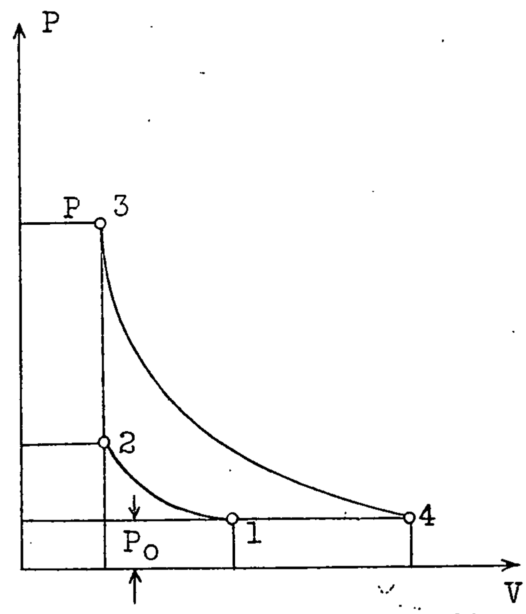


Fig.7 Pressure-volume diagram explosion turbine.

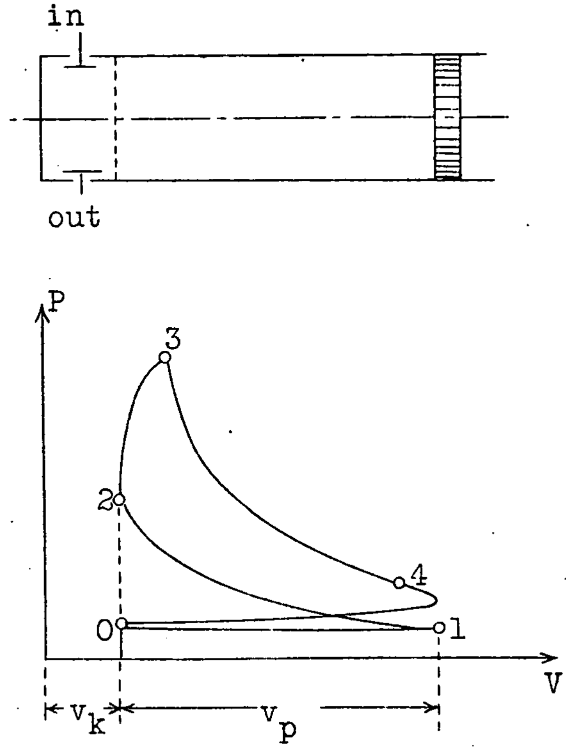


Fig.8 Four-stroke cycle.

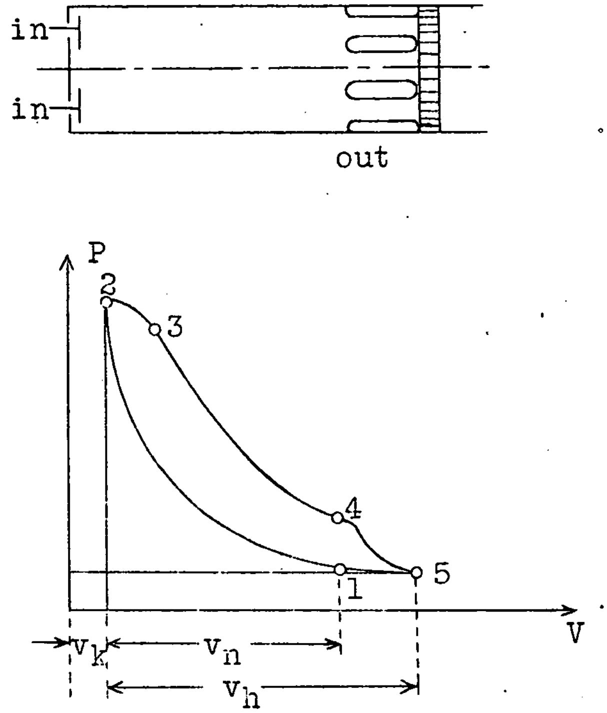


Fig.9 Two-stroke cycle.

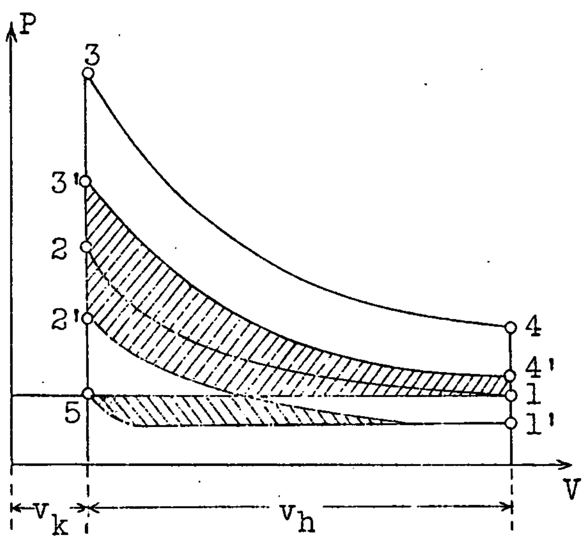


Fig.10 PV-Graph control
by filling charge.

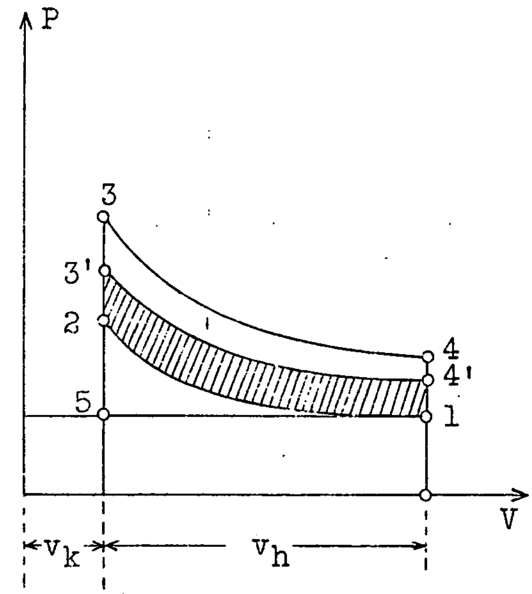


Fig.11 PV-Graph control
by mixture ratio.

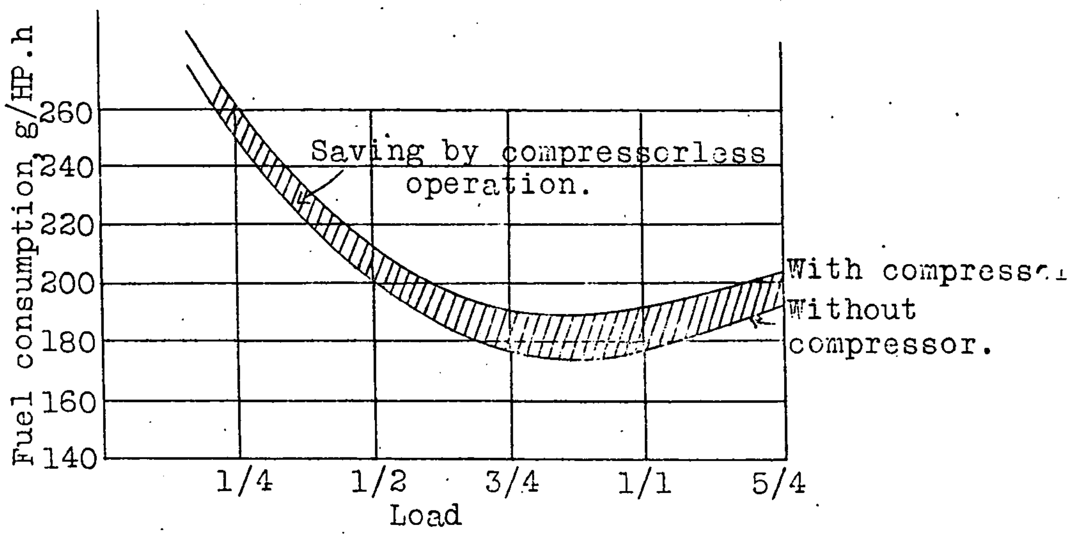


Fig.12 Fuel consume for Diesel engines with and without air injection.

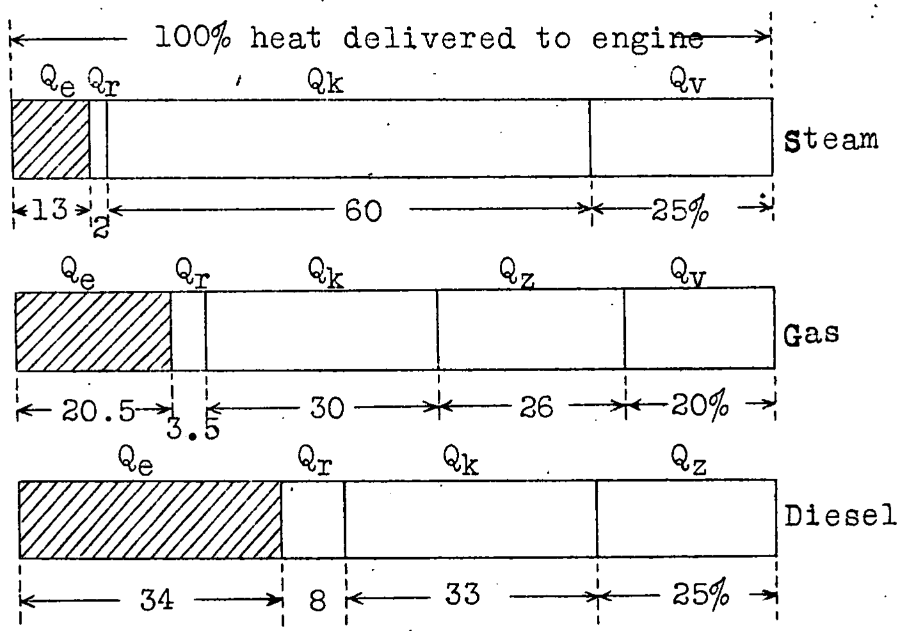


Fig.13 Heat balance sheet for a steam-, gas-, and Diesel-engine. Q_e = heat transformed into work. Q_r, Q_k, Q_z and Q_v signify heat losses from friction, cooling water, exhaust gas and gas intake respectively.