Thermodynamic Consideration of Supairthermal Diesel Engine

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Abstract

In this paper, the performance of Supairthermal Diesel engine is calculated by the analytical method which has been developed by one of the authors. Further, the comparison between performances of turbocharged Supairthermal and conventional Diesel engine is discussed.

1. Introduction

Supairthermal engine has the following features[1]; higher specific engine output can be obtained without running into the excessive mean cycle temperatures that would normally be encountered. This is accomplished by increasing the density of the air charge to the cylinder by higher turbocharging pressures and by both external and internal cooling of the charge. External cooling takes place in an intercooler between turbocharger outlet and the inlets to the cylinders. The inlet valves close early in the intake stroke, allowing the air to expand, and thus cool during the balance of the stroke.

The point of inlet valve closing is varied automatically, controlled by intake air manifold pressure, so that the pressure at beginning of compression is kept to be constant. At low manifold pressure the time of valve closing approaches normal.

The present authors research thermodynamically the features of this system in consideration of equilibrium operating condition of the engine.

One of the authors has developed the method of evaluation for the performance of the conventional turbocharged Diesel engine[2]. By applying this method, the performance of the Supairthermal Diesel engine is calculated on the following assumptions:

1. The Diesel engine is operating on four-stroke cycle and a schematic p–V diagram of the Supairthermal Diesel engine is shown in Fig. 1.

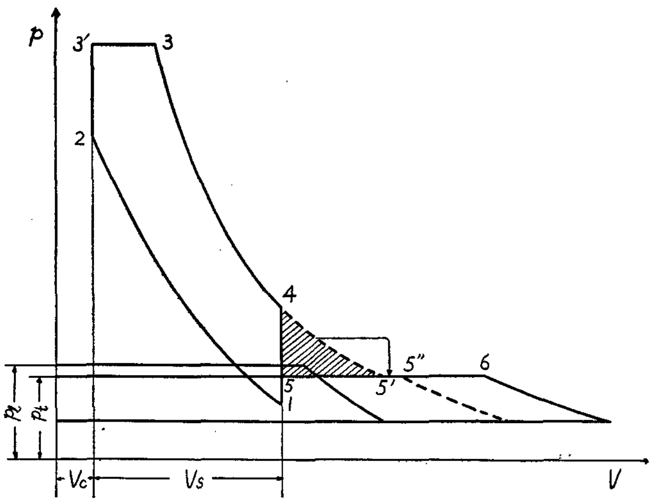

Fig. 1. A schematic p–V diagram of the Supairthermal Diesel engine.

2. Blow-down energy is

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completely converted into heat, producing the temperature rise of turbine inlet gas.

3. Turbine system is a constant pressure type.
4. Scavenging of residual gas is perfect.
5. Inlet air temperature is constant.

The obtained results are compared with those of the conventional Diesel engine.

2. Nomenclature

The following nomenclature is used in this paper:

a: Ratio of the short-circuited air to the total air delivered
c_m: Mean piston speed, m/s
c_p: Specific heat at constant pressure, kcal/kg°C
c_v: Specific heat at constant volume, kcal/kg°C
H_u: Lower calorific value, kcal/kg
L_0: Air quantity theoretically required for combustion, kg/kg
m: Exponent for compression and expansion of gas in cylinder
R: Gas constant, kgm/kg°K
S_r: Specific reduced time area of valve overlap=reduced time area of valve overlap in degree/piston area
T: Absolute temperature, °K
δ: Ratio of number of molecules before combustion to that after combustion
ε: Compression ratio
η_g: Diagram factor
η_T: Combined efficiency of turbocharger
η_{th}: Theoretical thermal efficiency
κ: Adiabatic exponent
λ: Excess air factor
ρ: Maximum pressure/compression pressure
σ: Cut-off ratio
φ: Pressure ratio

Subscripts

1: Condition at beginning of compression
g: Combustion gas condition
l: Engine inlet air condition
o: Ambient condition
t: Turbine inlet gas condition

3. Thermodynamic Relations in Supairthermal Diesel Engine[3]

From power balance of the turbine and compressor, the following equation is obtained:

$$\frac{\varphi_l^{\frac{\kappa_l-1}{\kappa_l}}-1}{1-\left(\frac{1}{\varphi_t}\right)^{\frac{\kappa_t-1}{\kappa_t}}}=\frac{T_t}{T_0}\eta_T\frac{c_{pt}}{c_{pl}}\left(1+\frac{1-a}{\lambda L_0}\right) \qquad \cdots\cdots(1)$$

The relation between the inlet air temperature and the temperature at beginning of compression is expressed by

$$T_1 = T_l\left(\frac{\varphi_1}{\varphi_l}\right)^{\frac{\kappa_l-1}{\kappa_l}}. \qquad \cdots\cdots(2)$$

On the assumption 1 and 2, the temperature of turbine inlet gas can be represented as follows:

$$T_t = \frac{T_l}{\frac{c_{pg}}{c_{pl}}+\frac{\lambda L_0}{1+\lambda L_0}\cdot\frac{a}{1-a}}\left[\frac{c_{vg}}{c_{pl}}\frac{\rho\sigma^m}{\delta}+\frac{\lambda L_0}{1+\lambda L_0}\left\{\frac{\kappa_l-1}{\kappa_l}\frac{\varphi_t}{\varphi_l}+\frac{T_1}{T_l}\frac{a}{1-a}\right\}\right] \qquad \cdots\cdots(3)$$

where

$$\sigma = \frac{c_{vg}}{c_{pg}}\left\{\frac{H_u}{c_{vg}(1+\lambda L_0)}\frac{\delta}{T_1\rho\varepsilon^{m-1}}+\frac{c_{pg}}{c_{vg}}-1+\frac{\delta}{\rho}\right\}. \qquad \cdots\cdots(4)$$

The amount of short-circuited air can be obtained by basic laws of flow as follows:

$$a = 1-\frac{\varepsilon}{(1+\varepsilon)\left(1+K\frac{\varphi_l}{\varphi_1}\frac{T_1}{T_l}\right)} \qquad \cdots\cdots(5)$$

where

$$K = \psi_1\sqrt{2gR_lT_l}\,\frac{S_r}{180c_m}$$

and ψ_1 is the Nusselt coefficient for ratio of the turbine inlet pressure p_t to the engine inlet pressure p_l.

Theoretical thermal efficiency of this cycle is given by

$$\eta_{th} = 1-\frac{1+\lambda L_0}{H_u}c_{vg}T_1\left(\frac{\rho\sigma^m}{\delta}-1\right). \qquad \cdots\cdots(6)$$

Indicated mean effective pressure is expressed as

$$p_i = 14.6\frac{H_u\eta_{th}\eta_g}{\lambda L_0}\frac{\varepsilon}{\varepsilon-1}\frac{p_1}{T_1}+\frac{L}{V_s}\ \frac{\text{kg}}{\text{cm}^2} \qquad \cdots\cdots(7)$$

where L/V_s is mean effective pressure of pumping work and is expressed by

$$\frac{L}{V_s} = \left[\frac{\kappa}{\kappa-1}\frac{\varepsilon}{\varepsilon-1}\left\{p_1\left(\frac{p_l}{p_1}\right)^{\frac{\kappa_l-1}{\kappa_l}}-p_1\right\}-\frac{p_l-p_1}{\varepsilon-1}\right]-(p_t-p_1). \qquad \cdots\cdots(8)$$

Maximum firing pressure and peak temperature in cylinder are expressed by

$$p_{\max} = p_1\rho\varepsilon^m \qquad \cdots\cdots(9)$$

$$T_{\max} = T_3 = T_1\rho\sigma\varepsilon^{m-1}/\delta \qquad \cdots\cdots(10)$$

respectively.

4. Computation and Its Results

When any two of eight operating variables $\cdots\lambda$, T_1, σ, T_t, φ_l, φ_t, c_m, $a\cdots$ are given, the relations between the rest can be obtained by solving simultaneously the equations

(1)~(5). Since it is difficult to solve algebraically the equations, the following graphical method is employed.

Method of Solution. The process of graphical solution is given by an example, where the following numerical values are chosen:

$$H_u = 10{,}000\ \text{kcal/kg},\ L_0 = 14.22\ \text{kg/kg},\ p_0 = 1\ \text{ata},\ T_0 = 288°\text{K},\ p_1 = 1.4\ \text{ata},$$
$$m = 1.3,\ \varepsilon = 12,\ \rho = 1.5,\ \eta_T = 0.5,\ \eta_g = 0.85,\ S_r = 38.18\ \text{deg.},\ c_m = 6\ \text{m/s}.$$

The method shown in Fig. 2 proceeds in the following ways: Eliminating the variable a from Eq. (1) by means of Eq. (2) and (5), the relations between φ_l, φ_t and T_t

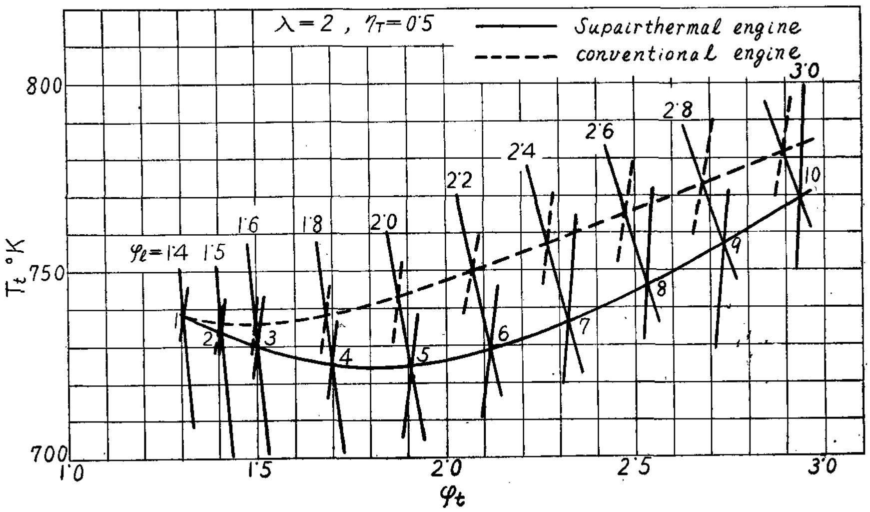


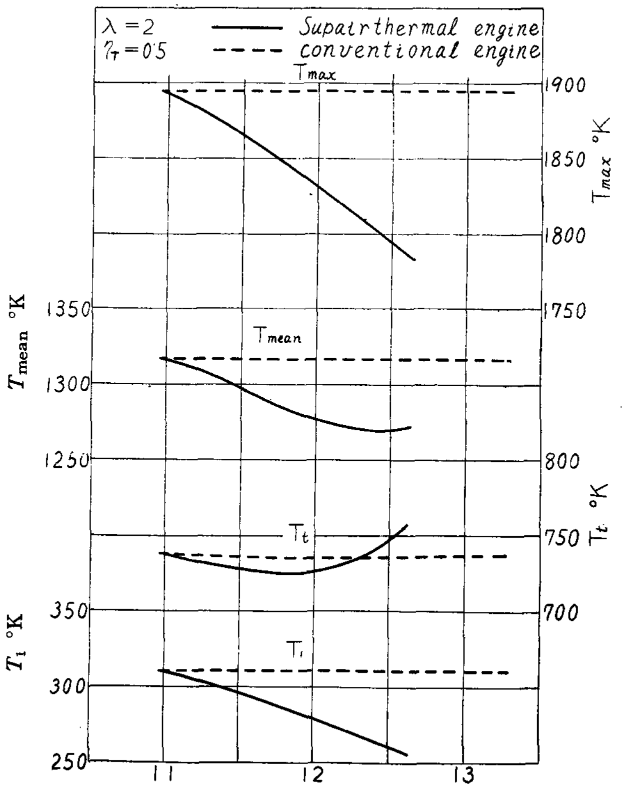
Fig. 2. An example of graphical solution.

existing in turbocharger can be obtained, and in the same manner the relations between them in the Diesel engine can be obtained by eliminating a, T_1 and λ from Eq. (3). They are shown in Fig. 2, in which φ_l is a parameter. The points 1, 2, 3,$\cdots$ denote the intersections of these curves having the same parameter. By substituting the values of φ_l at these points in Eq. (1), (4) and (5), the other variables can be calculated. Therefore, p_i, $T_{\max}$ and $p_{\max}$ can be obtained by means of Eq. (6)~(10) in corresponding to each point.

The results, computed by the above-mentioned method, are illustrated in Fig. 3, 4 and 5. In these figures, p_i is abscissa and dash-lines show the results of the conventional engine, calculated by the same method on assumption that T_l is constant and the numerical values are the same as those of the Supairthermal engine.

1. *Thermal Load.* The thermal load to which the piston is subjected may be assessed by the mean temperature in cylinder during the expansion stroke $=T_{\text{mean}}=$(peak temperature + exhaust temperature)/2$=(T_{\max}+T_t)/2$ and that to turbine blades is assessed by the turbine inlet temperature T_t.

For the same value of indicated mean effective pressure, as Miller persists[4], in the Supairthermal engine the mean temperature in cylinder is lower (Fig. 3). Considering



p_i kg/cm²

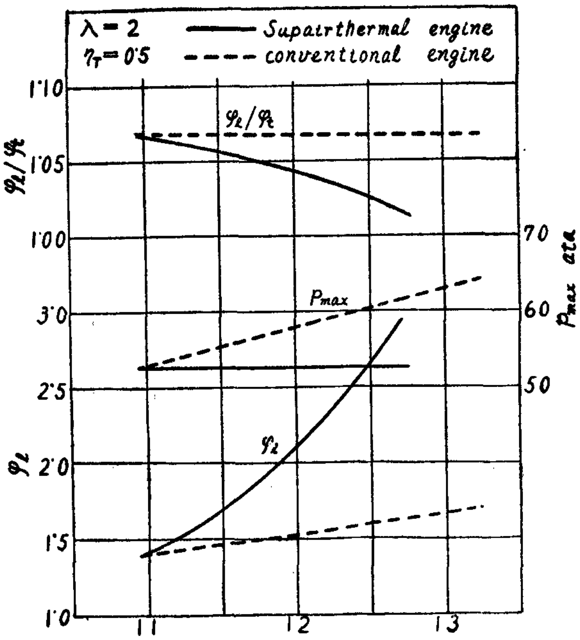
Fig. 3. Comparison of the temperatures of both engines.

the fact that engine output is limited by the thermal load of the piston, this is an important advantage.

On the other hand, as illustrated by T_t-curves in Fig. 3, the turbine inlet temperature of the Supairthermal engine is not always lower than that of the conventional engine, namely, in the higher range of p_i, the former is higher than the latter. From this point, it may be asserted that in the static pressure system the Supairthermal engine is not always more effective than the conventional engine to decrease their thermal load.

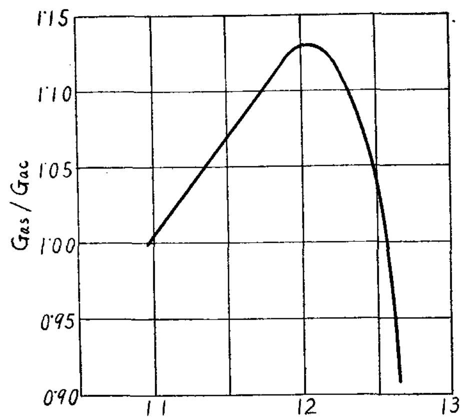
2. *Mechanical Load.* Maximum firing pressure is one of the most serious factors affecting the stress in all engine parts. p_{max}-curves in Fig. 4 afford the comparison of the maximum pressures in both engines. It will be noted that, in the conventional engine increase in p_i is accompanied with increase in p_{max}, but p_{max} in Supairthermal engine is constant independently of p_i.

3. *Flow Rate of the Short-circuited Air.* It is suggested[4)] that one of the features of Supairthermal engine is the more amount of cooled scavenging air being supplied by a higher intake air pressure, which promotes the cylinder cooling and decreases the turbine inlet gas temperature. It may be possible in the pulse system supercharging, but it is doubtful in the static pressure system. Because, in the static pressure system, the higher intake air pressure requires the higher turbine inlet gas pressure or temperature as shown



p_i kg/cm^2

Fig. 4. Comparison of the pressures of both engines.



p_i kg/cm^2

Fig. 5. Ratio of the amount of short-circuited air of Suparithermal engine to that of conventional engine.

in Eq. (1). The results, calculated with regard to the equilibrium operation of engine and turbine in this paper, show that the value of φ_l/φ_t in the Supairthermal engine is smaller than that in the conventional engine and, moreover, increase in φ_l, that is p_i, is accompanied with decrease in φ_l/φ_t, as shown in Fig. 4. Therefore, the ratio of the amount of short-circuited air in both engines, G_{as}/G_{ac}, which is calculated by Eq. (5), has the characteristics as shown in Fig. 5, having the maximum value 1.13 for a certain value of p_i. This value is too low to be a feature of Supairthermal engine. And, moreover, increase in p_i from this point is accompanied with sudden decrease in G_{as}/G_{ac} into 1, where the feature ceases to exist as a matter of course.

φ_l/φ_t curve shown in Fig. 4 supposes the existence of the point on which the value of φ_l/φ_t is equal to 1. In static pressure system, smooth operation is limited by $\varphi_l/\varphi_t=1$, therefore the maximum value of p_i in Supairthermal engine is also limited by $\varphi_l/\varphi_t=1$. Since the value of φ_l/φ_t at partial load is always smaller than that at full load, the value at full load should be greater than 1 in order to maintain safety operation in partial load. Therefore, when $\lambda=2$ at full load (this value is reasonable in turbocharged Diesel engine), this limitation of p_i should be lower.

By the example mentioned above, Supairthermal engine in static pressure system requires the higher turbocharger total efficiency in order to obtain the higher out-put. But the highest value of this efficiency is 50% to-day, so the higher output can not be expected.

5. Conclusion

1. It is possible to determine equilibrium performance of turbocharged Supairthermal Diesel engine similarly to that of conventional Diesel engine.

2. By this method comparison between performance of turbocharged Supairthermal and conventional Diesel engine is numerically computed. Obtained results show clearly the characteristic features of Supairthermal Diesel engine.

References

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3) Ditto : Ditto.
4) : Diesel Power, August, p. 42 (1951).