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EFFECT OF HEAT TRANSFER ON THE PERFORMANCE OF AN AIR-STANDARD DIESEL CYCLE

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ABSTRACT

This paper presents thermodynamic analysis of an air-standard Diesel cycle. It presents the effect of heat transfer on the net work output and the indicated thermal efficiency of the cycle. The heat losses through cylinder walls are considered to be proportional to the average temperature during heat addition process. The effects of other parameters, in conjunction with heat transfer, such as cutoff ratio and intake air temperature were also reported. The results obtained from this work can be helpful in the design and evaluation of Diesel engines.

Introduction

In 1892 Rudolf Diesel proposed an engine that allows air alone to enter into the combustion chamber, instead of a combustible mixture, during the intake stroke. It is then compressed to a high pressure. Ultimately, raising its temperature to such a point that if a fuel is injected into the engine, it would self-ignite without a spark to initiate combustion. This type of engine is known as Diesel or compression-ignition engine [1-5]. Diesel engines sometimes are more attractive in internal combustion engine application over Otto engines, due the higher thermal efficiency. The higher thermal efficiency is a result of higher compression ratios that most Diesel engines operate with.

In general, real air-standard cycles are far from ideal cycles. Therefore, optimization and performance analysis can be applied using finite time thermodynamics technique [6-8]. A major assumption in ideal air-standard cycles is that all processes are reversible. However, during heat addition this assumption may not be valid for real cycles. The net work output and efficiency are affected considerably by irreversibility during combustion and the constant-pressure heat addition.

In this paper different parameters affecting indicated thermal efficiency and net work output in relation with heat transfer consideration in a Diesel cycle will be considered. The obtained results will be plotted as performance characteristic curves for the Diesel cycle using numerical examples.

Thermodynamic Analysis

Figure 1 presents a pressure-volume (*P-V*) diagram for the thermodynamic processes performed by an ideal air-standard Diesel cycle [2,5]. All four processes are reversible. Process 1-2 is adiabatic (isentropic) compression; process 2-3 is heat addition at a constant pressure; process 3-4 adiabatic (isentropic) expansion; finally, process 4-1 is heat rejection at a constant volume.

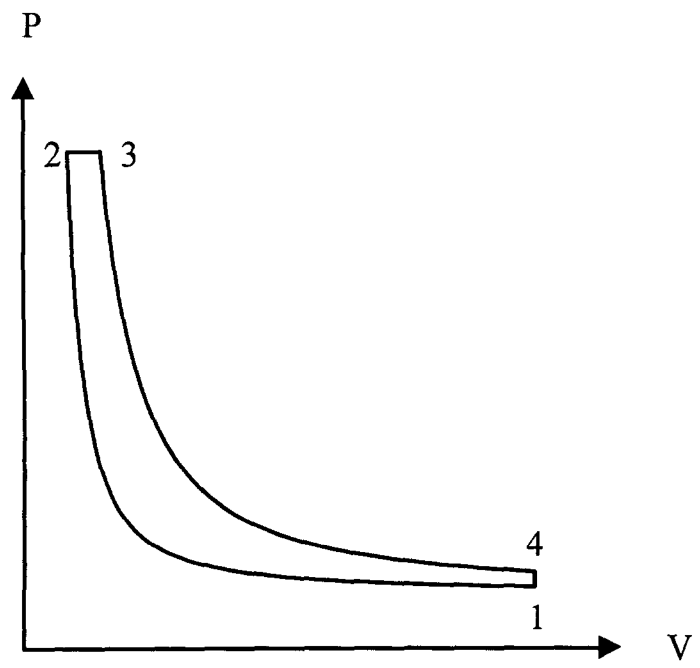

FIG. 1

Pressure-volume diagram of Diesel cycle.

Assuming constant specific heats, the net work output per unit mass of the working fluid is given by the following equation:

$$W = C_p(T_3 - T_2) - C_v(T_4 - T_1) \tag{1}$$

Where C_p and C_v is the constant-pressure and constant-volume specific heats, respectively; and T_1, T_2, T_3 and T_4 are the absolute temperatures at states 1, 2, 3, and 4, respectively. For the isentropic processes 1-2 and 3-4 relationships can be obtained for T_2 and T_4 in terms of T_1 and T_3, respectively, by the following equations:

$$T_2 = T_1 r_c^{\,k-1} \tag{2}$$

$$T_4 = T_3\left(\frac{r}{r_c}\right)^{k-1} \tag{3}$$

where k is the specific heat ratio (C_p/C_v), while r_c and r are the compression and cutoff ratios, respectively.

The heat added per unit mass of the working fluid during the constant-pressure process 2-3 (${}_2Q_3$) is written as:

$${}_2Q_3 = C_p(T_3 - T_2) \tag{4}$$

For an ideal Diesel cycle, no irreversible losses take place and the process is totally reversible. However, in a real Diesel cycle the process is more likely to be irreversible. In other words some heat will be transferred or lost through cylinder walls during combustion. The heat losses are proportional to the temperatures T_3 and T_2, since the process is taking place between states 2-3. Therefore, the heat added to the working fluid during combustion process can be given in the following linear expression [6-8]:

$${}_2Q_3 = A - B(T_3 + T_2) \tag{5}$$

where, A and B are constants related combustion and heat transfer. Therefore, by equating equations (4) and (5) an expression can be found for T_3 in terms of T_2:

$$T_3 = \frac{[A + (C_p - B)T_2]}{(C_p + B)} \tag{6}$$

Eliminating T_2 by substituting the result obtained from equation (2) for T_2 in the previous expression we obtain the following expression for T_3:

$$T_3 = \frac{[A + (C_p - B)T_1 r_c^{k-1}]}{(C_p + B)} \tag{7}$$

Similarly, substituting the result obtained from equation (7) into equation (3). T_4 is written as a function of T_1

$$T_4 = \frac{[A + (C_p - B)T_1 r_c^{k-1}]}{(C_p + B)} \left(\frac{r}{r_c}\right)^{k-1} \tag{8}$$

Combining the results obtained from equations (2), (7), and (8) into equation (1), and in terms of temperature T_1 the following expression for the net work output, W, can be introduced:

$$W = \frac{\left[C_p\left\{A + T_1 r_c^{k-1}(C_p - B) - T_1 r_c^{k-1}(C_p + B)\right\} - C_v\left\{\left[A + T_1 r_c^{k-1}(C_p - B)\right]\left(\frac{r}{r_c}\right)^{k-1} - T_1(C_p + B)\right\}\right]}{\left[C_p + B\right]} \tag{9}$$

and also the indicated thermal efficiency, η is given as:

$$\eta = 1 - \frac{\left[A + \left\{(C_p - B)r^{k-1} - (Cp + B)\right\}T_1\right]}{k\left[A - 2BT_1 r_c^{k-1}\right]} \tag{10}$$

Combining equations (9) and (10), the following expression yields:

$$W = \frac{C_p \eta [A - 2T_1 r_c^{k-1} B]}{(C_p + B)} \tag{11}$$

The above equation shows that a maximum work output can be found with respect to efficiency. In other words, the net work output W is differentiated with respect to efficiency, η, and its derivative is set to zero. An optimum or maximum value for W_{max} is thus obtained, i.e.,

$$\frac{\partial W}{\partial \eta} = 0 \tag{12}$$

and hence,

$$W_{\max} = \frac{AC_p}{B} - C_v T_1 (r^{k-1} - 1) \tag{13}$$

Numerical Results and Discussion

The derived formulae above are used and plotted in Figures 2 through 9, the following constants and range of parameters are selected:

$C_p = 1.003$ kJ/kg-K $C_v = 0.716$ kJ/kg-K $k = 1.400$ $T_1 = 300 - 400$ K

$A = 2500 - 4000$ kJ/kg $B = 0.6 - 2.0$ kJ/kg-K $r = 1.5 - 10$

For example, Fig. 2 shows the effect of parameter B on work output and thermal efficiency performance. According to equation (5) increasing the value of B decreases the amount of heat added to the engine. Therefore, the maximum work and efficiency will have lower values by increasing B, accordingly. The effect of parameter A is shown in Fig. 3. Evidently, it has the opposite effect than B. Maximum work and efficiency increase with increasing A.

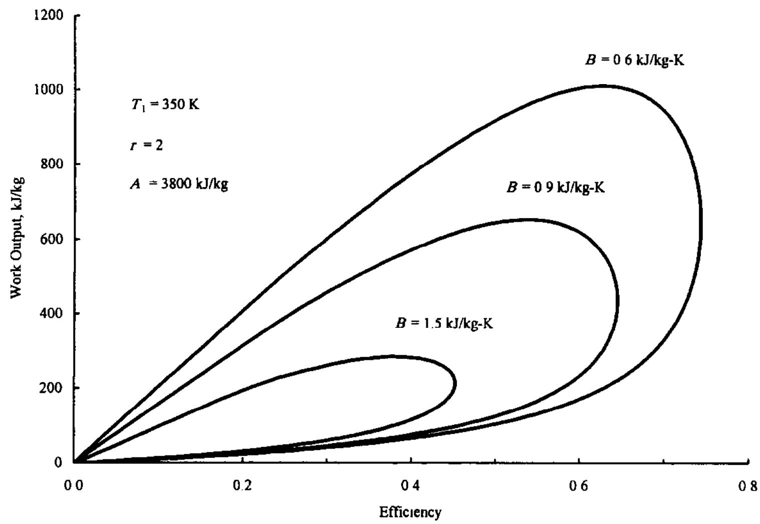

FIG. 2

Effect of B on net work output and thermal efficiency performance.

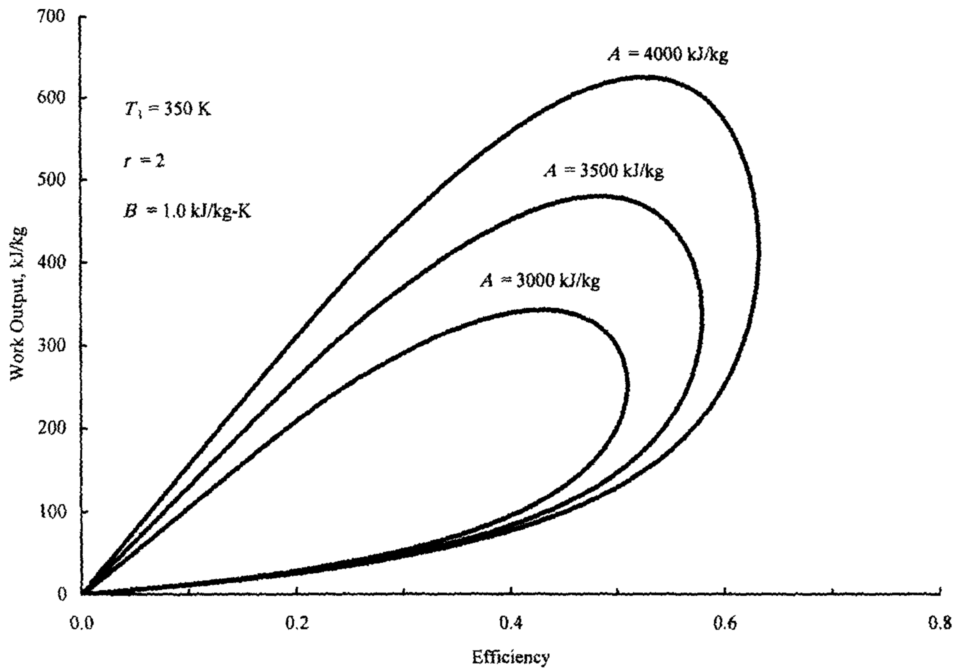

FIG. 3

Effect of A on net work output and thermal efficiency performance.

The effect of inlet intake temperature, T_1 is shown in Fig. 4. Increasing T_1 reduces maximum work and efficiency. Also, the cutoff ratio, r effects the engine performance. The maximum work and efficiency decrease by the increasing of cutoff ratio as shown in Fig. 5.

Figures 6 and 7 show the effect of varying B at different values of A and r, respectively. Maximum work output decreases by increasing B.

Conclusion

An expression for the net work output and the indicated thermal efficiency of a Diesel cycle is derived, which is effected by cylinder wall heat transfer. The effects of different parameters, such as cutoff ratio, intake air temperature, and heat transfer and combustion constants are investigated in this paper. The results of this paper are intended to provide some guidance for design improvement and effect of heat transfer on the performance of Diesel engines.

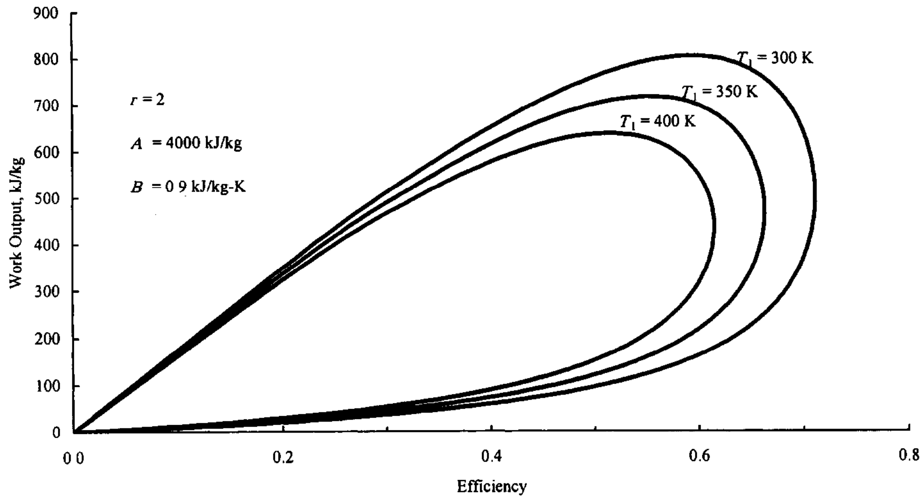

FIG. 4

Effect of intake air temperature on net work output and thermal efficiency performance.

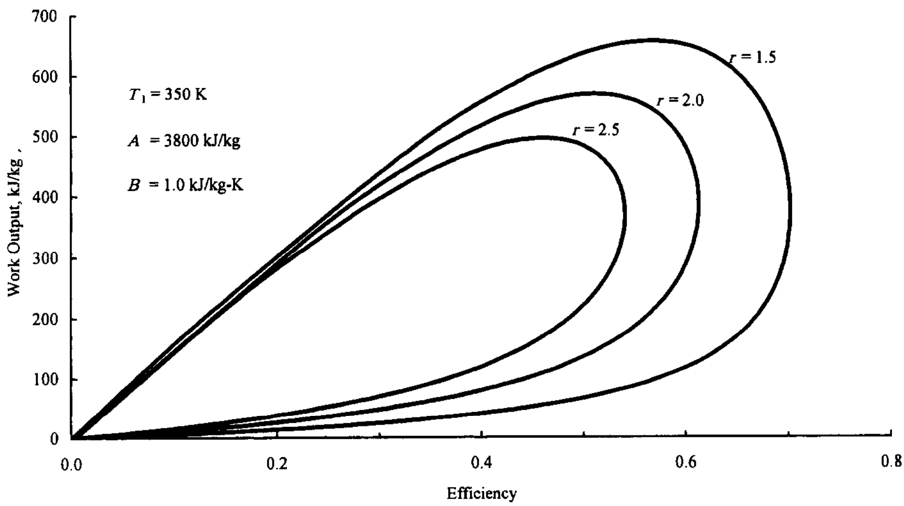

FIG. 5

Effect of cutoff ratio on net work output and thermal efficiency performance.

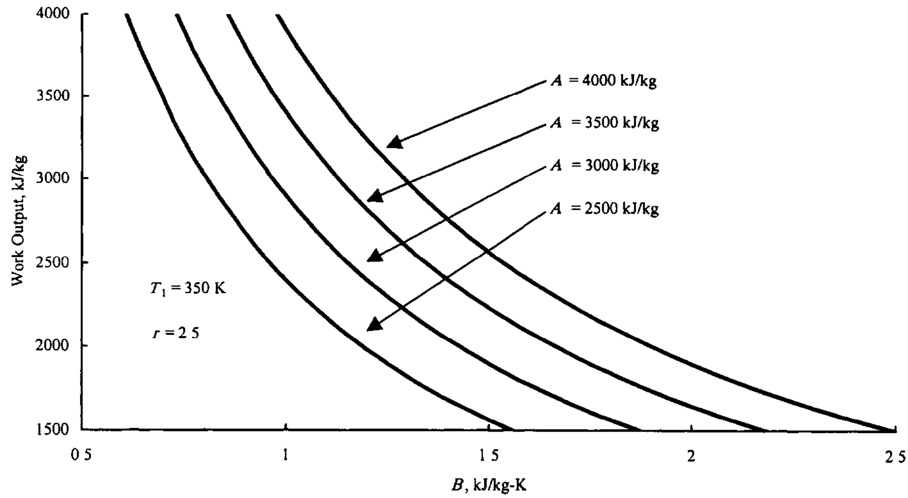

FIG. 6

Effect of A and B on maximum work output performance.

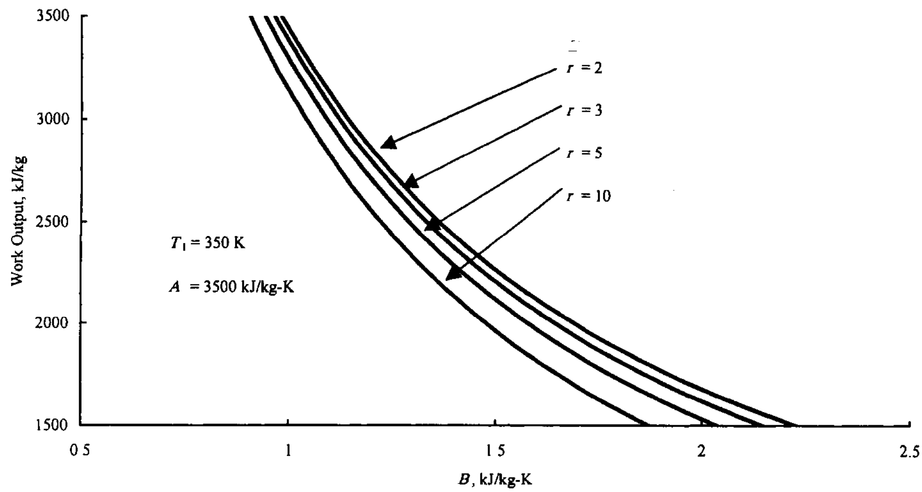

FIG. 7

Effect of B and cutoff ratio on maximum work output performance.

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