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Performance analysis of air-standard Diesel cycle using an alternative irreversible heat transfer approach

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ABSTRACT

This study presents the investigation of air-standard Diesel cycle under irreversible heat transfer conditions. The effects of various engine parameters are presented. An alternative approach is used to evaluate net power output and cycle thermal efficiency from more realistic parameters such as air–fuel ratio, fuel mass flow rate, intake temperature, engine design parameters, etc. It is shown that for a given fuel flow rate, thermal efficiency and maximum power output increase with decreasing air–fuel ratio. Also, for a given air–fuel ratio, the maximum power output increases with increasing fuel rate. However, the effect of the thermal efficiency is limited.

1. Introduction

In ideal air-standard power cycle analyses an assumption is often used, and that is heat transfer rate in the cycle is reversible [1,2]. However, it is a well known fact that in a real Diesel engine this is not true. Therefore, some previous studies assume a somewhat more realistic simplified heat transfer model than reversible heat transfer processes. The approximation used states that the heat added to the working fluid during the combustion process has a sort of linear expression with some constant parameters [3–7]. Performance analysis of an irreversible Diesel cycle was presented by Parlak [8,9]. It was concluded that such analysis can provide guidelines for improving design of Diesel engines. In a recent study a relationship between fuel's heating value and heat loss through cylinder walls was reported by Ozsoysal [10]. Recently, Karamangil et al. presented parametric study on the variations of the coefficients of convection heat transfer of the cylinder and coolant side of the engine [11]. Other studies concerning various effects of heat transfer in internal combustion engines are found in literature [12,13].

In general, real air-standard cycles are far from ideal cycles. Therefore, optimisation and thermodynamic performance analysis may be used to study the effect of various parameters on the performance of heat engines for comparative reasons. The net power output and cycle thermal efficiency are affected considerably by irreversibilities during combustion and heat addition process. In this study a more realistic approach is implemented to study the performance of Diesel cycle. The evaluation of maximum net power output and cycle thermal efficiency were studied as functions of more realistic parameters such as air–fuel ratio, mass flow rate of the fuel, overall heat transfer coefficient, and other engine design parameters.

2. Thermodynamic analysis

As shown in Fig. 1, the ideal air-standard Diesel cycle is made of four reversible processes; process 1–2 is reversible adiabatic (isentropic), which represents the compression stroke in a compression–ignition engine. The combustion process is modelled as a constant pressure heat addition process (i.e., process 2–3). It is followed by isentropic expansion process (i.e., process 3–4), which is representing the power stroke in the heat engine. Finally, process 4–1, which completes the cycle, is a constant volume heat rejection process.

For the isentropic processes 1–2 and 3–4, the following relationships can be obtained for T_2 and T_4 in terms of T_1 and T_3, respectively [14]:

$$T_2 = T_1 (r_c)^{\gamma-1} \tag{1}$$

$$T_4 = T_3 \left(\frac{\beta}{r_c}\right)^{\gamma-1} \tag{2}$$

where, γ is the specific heat ratio (c_p/c_v), r_c is the compression ratio, and β is the cutoff ratio.

For the constant pressure heat addition process 2–3, the rate of heat added to the air–fuel mixture during the process can be given as

$$\dot{Q}_{2-3} = \dot{m}_m c_p (T_3 - T_2) \tag{3}$$

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Nomenclature

A_p	piston crown area, m^2
AFR	air–fuel ratio
B	cylinder bore, m
c_p	constant pressure specific heat, kJ/kg K
c_v	constant volume specific heat, kJ/kg K
h_o	overall heat transfer coefficient, kW/m^2 K
k	thermal conductivity, kW/m K
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
$\dot{Q}$	heat transfer rate, kW
Q_{LHV}	lower heating value of the fuel, kJ/kg
r_c	compression ratio
Re	Reynolds number
T	gas temperature, K
T_c	coolant temperature, K
$\dot{W}$	power, kW

Greek symbols

β	cutoff ratio
γ	specific heat ratio
η	thermal efficiency
η_c	combustion efficiency
μ	dynamic viscosity, kg/m s

Subscripts

f	fuel
m	air–fuel mixture
avg	average

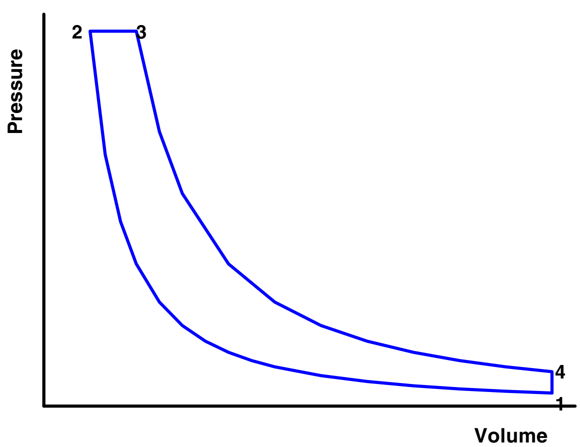

Fig. 1. Pressure–volume diagram of an air-standard Diesel cycle.

Although in an ideal Diesel cycle this process is modelled as a reversible process, irreversible heat losses from the gas mixture through the cylinder walls occur in a real cycle. In fact, such irreversibility is encountered throughout the cycle. However, it becomes particularly significant during this process due to the very high temperature differential across the cylinder walls. In order to account for these irreversible losses, the rate of heat transferred to the gas can also be represented in terms of the combustion efficiency, the fuel mass flow rate and its lower heating value, in addition to the rate of heat loss from the mixture in the cylinder

$$\dot{Q}_{2-3} = \eta_c \dot{m}_f Q_{LHV} - \dot{Q}_{loss} \tag{4}$$

The rate of heat loss can in turn be evaluated using the Taylor model which is based on evaluating the heat transfer rate between the cylinder gas and the coolant as follows [15]:

$$\dot{Q}_{loss} = h_o A_p (T_{avg} - T_c) \tag{5}$$

The reference area in this model is the piston crown area:

$$A_p = \frac{\pi B^2}{4} \tag{6}$$

The overall heat transfer coefficient can be evaluated from the Nusselt number defined on the basis of the piston diameter:

$$Nu = \frac{h_o B}{k} \tag{7}$$

This Nusselt number implicitly includes the conduction and radiation heat transfer components and is correlated to Reynolds number as follows [15]:

$$Nu = 10.4 Re^{0.75} \tag{8}$$

Reynolds number in this correlation is based on the mass flow rate of the air–fuel mixture [14]:

$$Re = \frac{\dot{m}_m B}{A_p \mu} \tag{9}$$

The average temperature of the cylinder gases during the heat addition process can be approximated as

$$T_{avg} = \frac{(T_2 + T_3)}{2} \tag{10}$$

Utilising the definition of the cutoff ratio and substituting with Eq. (1), the average temperature can be written in terms of the inlet temperature T_1:

$$T_{avg} = T_1 (r_c)^{\gamma-1} \left(\frac{1+\beta}{2} \right) \tag{11}$$

Combining Eqs. (3)–(5), T_3 can be expressed in terms of T_2:

$$T_3 = T_2 + \frac{\eta_c \dot{m}_f Q_{LHV} - h_o A_p (T_{avg} - T_c)}{\dot{m}_m c_p} \tag{12}$$

Substituting with Eqs. (1) and (11), T_2 can be eliminated and T_3 can be expressed in terms of the inlet temperature T_1:

$$T_3 = T_1 (r_c)^{\gamma-1} + \frac{\eta_c \dot{m}_f Q_{LHV} - h_o A_p (T_1 (r_c)^{\gamma-1} (1+\beta)/2 - T_c)}{\dot{m}_f (1 + \mathrm{AFR}) c_p} \tag{13}$$

Similarly, T_4 can also be expressed in terms of T_1 by substituting Eq. (13) into Eq. (2):

$$T_4 = T_1 (\beta)^{\gamma-1} + \frac{(\beta/r_c)^{\gamma-1}}{\dot{m}_f (1 + \mathrm{AFR}) c_p} [\eta_c \dot{m}_f Q_{LHV} - h_o A_p (T_1 (r_c)^{\gamma-1} \times (1+\beta)/2 - T_c)] \tag{14}$$

The rate of heat rejected in the constant volume heat rejection process 4-1 is given as

$$\dot{Q}_{4-1} = \dot{m}_m c_v (T_4 - T_1) \tag{15}$$

The net power produced in the cycle can be written as

$$\dot{W}_{net} = \dot{Q}_{2-3} - \dot{Q}_{4-1} \tag{16}$$

Therefore, its efficiency is

$$\eta = 1 - \frac{\dot{Q}_{41}}{\dot{Q}_{23}} \tag{17}$$

Eqs. (16) and (17) can be rewritten in terms of the inlet temperature T_1:

$$\dot{W}_{net} = \frac{\gamma - (\beta/r_c)^{\gamma-1}}{\gamma}[\eta_c \dot{m}_f Q_{LHV} - h_o A_p (T_1 (r_c)^{\gamma-1}(1+\beta)/2 - T_c)] - \dot{m}_f(\mathrm{AFR}+1)c_v(\beta^{\gamma-1}-1)T_1 \tag{18}$$

$$\eta = \frac{\gamma - (\beta/r_c)^{\gamma-1}}{\gamma} - \frac{\dot{m}_f(\mathrm{AFR}+1)c_v(\beta^{\gamma-1}-1)}{[\eta_c \dot{m}_f Q_{LHV} - h_o A_p (T_1 (r_c)^{\gamma-1}(1+\beta)/2 - T_c)]} T_1 \tag{19}$$

3. Numerical results and discussion

As it can be clearly seen from Eqs. (18) and (19), the efficiency and the net power output of the Diesel cycle are dependent on the air inlet temperature, the cutoff ratio, the fuel mass flow rate and the air–fuel ratio. In order to illustrate the effect of these parameters, the net power output was plotted against the efficiency for selected cases presented in Figs. 2–5. The values of the constants and the parameters used in this example are summarised in Table 1. It should be noted that all gas properties were taken at a temperature of 1000 K which is normally considered as a good approximation of the average in-cylinder temperature during the cycle. The lower heating value is that of light Diesel fuel and all other parameters were selected to represent typical values usually encountered in practical operation of Diesel engines.

Fig. 2 shows the effect of the cutoff ratio on the net power output and the thermal efficiency of the cycle. By decreasing the cutoff ratio an increase of the maximum engine's net power will result. Also for a given power output, decreasing the cutoff ratio increases the efficiency of the engine at which this output is produced. This result is expected from a thermodynamic point of view, since the decrease of the cutoff ratio of a Diesel cycle to approach unity brings it closer to the efficiency of an Otto cycle. For a given compression ratio, an Otto cycle is known to have a higher efficiency in comparison with a Diesel cycle.

The effect of the air inlet temperature is shown in Fig. 3. Increased air inlet temperatures would result in significant reduction in both the efficiency and the net power output of the engine. As the inlet temperature increases, the in-cylinder temperatures increase accordingly with magnitudes that become amplified with

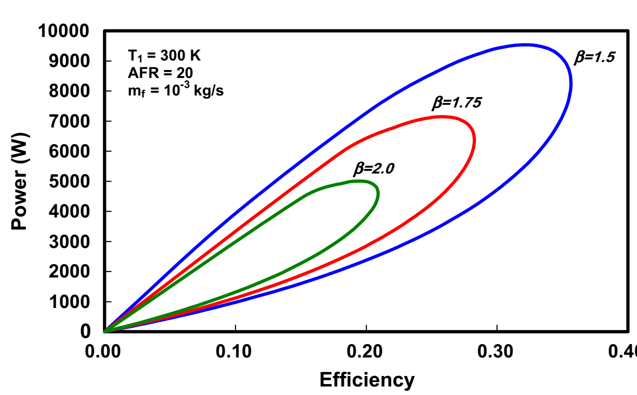

Fig. 2. Effect of the cutoff ratio on the power output versus efficiency curve.

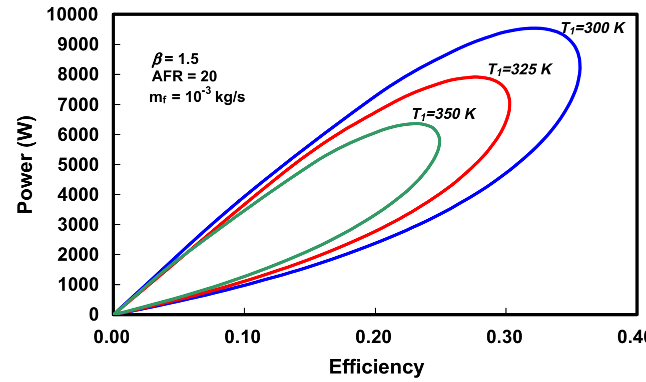

Fig. 3. Effect of the air inlet temperature on the power output versus efficiency curve.

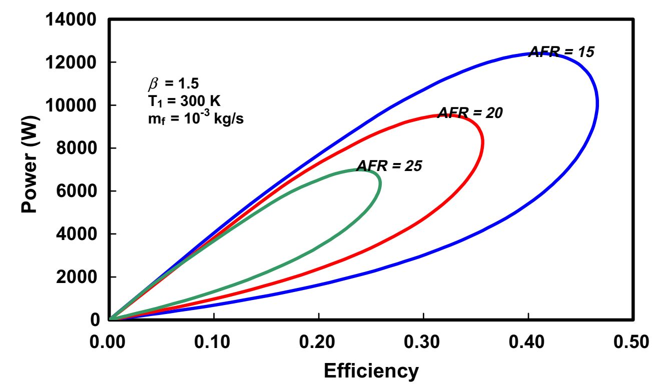

Fig. 4. Effect of the air–fuel ratio on the power output versus efficiency curve.

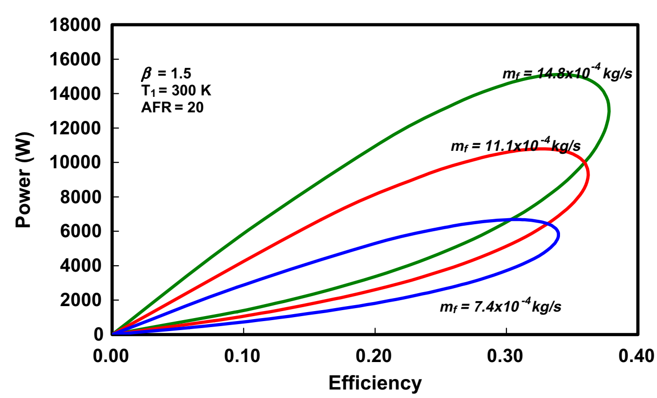

Fig. 5. Effect of the fuel mass flow rate on the power output versus efficiency curve.

higher compression ratios. Such high temperatures induce increased heat losses from the cylinder, which can significantly bring down the efficiency and the net power output of the engine.

Fig. 4 illustrates the effect of the air–fuel ratio. Decreasing the air–fuel ratio can significantly improve the efficiency of the engine and increase its maximum power output. This can be mainly attributed to increased heat losses at higher air–fuel ratios. For a given fuel rate and engine geometry, increasing the air–fuel ratio results in higher values of in-cylinder Reynolds number, and consequently higher values of Nusselt number and overall heat transfer coefficient.

The effect of the fuel mass flow rate, demonstrated in Fig. 5, is particularly interesting. As expected, increasing the fuel rate

Table 1
Constants and parameters used in the numerical example

Cylinder bore	0.1 m
Combustion efficiency	100%
Lower heating value of the fuel	42,500 kJ/kg
Constant pressure specific heat	1.141 kJ/kg K
Constant volume specific heat	0.853 kJ/kg K
Specific heat ratio	1.338
Thermal conductivity	6.67×10^{-5} W/m K
Dynamic viscosity	4.24×10^{-5} kg/m s
Air inlet temperature	300–350 K
Coolant temperature	350 K
Air–fuel ratio	15–25
Cutoff ratio	1.5–2.0
Fuel mass flow rate	$(7.4–14.8) \times 10^{-4}$ kg/s

increases the maximum power that can be produced under given operational conditions. However, the effect of increasing the fuel rate on the efficiency at which maximum power is produced is minimal. This is due to the fact that there are two factors acting against each other here: As the fuel rate increases, the amount of heat added to the gas mixture increases too. On the other hand, heat losses from the cylinder increase as well due to the increase in the temperature and the overall heat transfer rate. In this case, it is clear that the increase in the added heat is faster than the increase in the lost heat. This can however change depending on the fuel used and the parameters controlling the overall heat transfer coefficient.

4. Conclusions

The investigation of air-standard Diesel cycle under irreversible heat transfer conditions was presented in this study. The effects of various engine parameters were presented by using an alternative approach to evaluate net power output and cycle thermal efficiency from more realistic parameters such as air–fuel ratio, fuel mass flow rate, intake air temperature, engine design parameters, etc.

A numerical example showed that an increase of the maximum engine's net power and of the efficiency of the engine will result by decreasing the cutoff ratio. Also, increasing air inlet temperatures would result in reduction in both the efficiency and the net power output of the engine, significantly.

The effect of air–fuel ratio was presented. It was found that the efficiency of the engine can be significantly improved by decreasing the air–fuel ratio; it causes its maximum power output to increase. Finally, increasing the fuel flow rate increases the maximum power that can be produced under given operational conditions. However, increasing the fuel rate has little effect on the efficiency at which maximum power is produced.

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