Heat Leak Term - a Signature of Irreversibility in Analysing the Actual SI Engine Cycle Using Finite-Time Thermodynamics

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Due to the headway in innovations, it ends up noticeably unavoidable to do a complex thermodynamic analysis of thermal systems. The major problem faced during the development of a thermodynamic model for the analysis is to check the feasibility of the proposed model contrasted with the current options. Finite Time Thermodynamics can be utilized adequately to meet this without going for a mind-boggling investigation. Nonetheless, if the process is divided into infinitesimally small processes with the irreversibilities accounted at all equilibrium conditions; a reversible thermodynamic approach could be utilized. This paper shows the efficacy of such an approach and the errors accompanied with while analysing the Otto cycle. The Finite Time model created in this work is investigated and contrasted with the actual values. For the thermal efficiency calculated, Finite Time Thermodynamic model was observed to be more accurate than other similar techniques that do not involve complex thermodynamic analysis. Finite Time Thermodynamic model created accounts for the entire complex phenomenon occurring in an SI engine with an introduction of only a heat leak term into it.

Keywords: *SI engine, Finite Time Thermodynamics, Quasi-static process, Endoreversible.*

1. INTRODUCTION

Over the span of advancement, the unrealistic application of reversible thermodynamics turned out to be horrendous to clarify the real phenomenon [1]. This prompts the advancement of theories portraying the realistic nature of the system. Finite Time Thermodynamics (FTT) is an augmentation of conventional thermodynamics, thus developed, that deals with the processes having explicit time or rate dependencies [2]. FTT combines the most fundamental ideas of heat transfer, fluid mechanics and thermodynamics [3]. In FTT, the real concern is about the effect of constraints on time or rate on the performance of the system [4].

FTT was utilized as a part of many works, for optimising the irreversible process that was subject to Finite time constraints. Chambadal [5] developed an FTT model, assessing the irreversibilities as heat leak on both the high and low-temperature sides of the system. Novikov [6] modified the Carnot cycle by introducing external irreversibility as a finite temperature gradient between both the high-temperature source and the working fluid. He also considered internal irreversibility during expansion with a parameter, which is the ratio of actual and ideal entropy drop during the isothermal heat rejection [5]. Later, Curzon and Ahlborn [7] broadened Chambadal's [5] work by assessing the low-temperature side, and it was presumed that the finite time heat fluxes between both the reservoirs and the working fluid were proportionate to their temperature differences. They used the overall heat transfer coefficient as the proportionality constant. Another exceptional work done during this period was by Wu and Kiang [8]. They presented internal irreversibilities in both the compression and expansion process apart from the internal irreversibilities considered during the expansion process alone by Novikov [6].

Aside from the works done on the Carnot cycle to provide it with a realistic nature, Kaushik and Kumar [9] and Yaqi *et al*. [10] had done FTT investigation of an endoreversible Stirling heat engine. They utilized FTT to determine the conditions at which the Stirling heat engine produces maximum power and its corresponding efficiency. Comparative works were done by Sieniutycz [11] to optimise the power developed in thermal, chemical and electrochemical systems. Durmayaz *et al*. [5] had done detailed work on the optimisation of thermal systems in light of FTT and Thermoeconomics. They also provide a definite outline of the optimisation of a wide range of thermal systems, both endoreversible and irreversible. In an endoreversible system, the working fluid is supposed to be in internally consistent equilibrium [12].

Abu-Nada *et al*. [13] in their work used air-standard relations with temperature-dependent specific heats. The results demonstrate that at higher engine speeds, there was a significant variation between the results obtained using constant specific heats and the temperature-dependent relations. This may be because, at higher engine speeds, the cycle temperature will be higher than

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that at lower engine speeds for the same air-fuel ratio [14], due to the reduced time for heat loss from the engine at higher speeds. Additionally, the specific heat of air is found to be changing abruptly between the temperatures 1500K and 3000 K, which is supposed to be the temperature range of the working fluid during the combustion and expansion processes. Hence tempe–rature-dependent specific heat relations could be utilised to get results that are more accurate than the constant specific heat relations.

Similar to the work done by Curzon and Ahlborn [7], Chen *et al*. [15] proposed the relations for maximum work output and its corresponding efficiency with the heat transfer assumptions for the Otto cycle. They introduced two constants to accompany the effects of heat transfer, combustion and the maximum temperature, within a range of specified values. However, they had failed to locate a reasonable approach to set up the correct estimations of these constants and the maximum temperature, instead, had provided a range of values. Similarly, Osman [16] intro–duced heat transfer and combustion parameters and also provided the means to find these parameters as well as the maximum temperature. From these relations, it is seen that the maximum temperature achieved increases from lean to rich mixtures [16]. However, in reality, the peak tempe–rature increases as the leanness of the mixture get reduced and accomplish a most extreme value at slightly rich mixture and then starts to reduce with the increase in the richness of the mixture. Curto-Risso *et al*. [17], had consi–dered three irreversibility sources, namely internal irre–versibilities (fluid friction, effects of viscosity, combus–tion etc.), mechanical losses and heat transferred to the coolant. They additionally identified engine speed-depen–dent parameters for temperature, the mass of working fluid admitted, internal irreversibility factor and heat transfer loses. These quantities used as a part of the FTT model created, produced results closer to the real values [17]. Apart from these, there are many zero-dimensional ther–modynamic models available for analysing the per–formance of SI engines, which are much more accurate than the above mentioned FTT models [18, 19]. How–ever, FTT can be used as a simple and powerful tool for optimising thermal systems rather than going for complex thermodynamic analysis. Hence, FTT finds applications in the wider areas of thermal analysis and optimisation.

In addition to the early works done for modelling and optimizing the thermal systems, FTT had found applications in modelling the chemical process at cellu–lar levels in biological systems [20, 21] and even in economics [22]. FTT had evolved a lot and is capable of modelling quantum systems that involve the calculation of energy interactions or heat losses with a higher level of accuracy [23–25]. The fluctuations of thermo–dynamic quantities could not be neglected and play a vital role as the system size is very small [26]. Various models and modifications have evolved based on quan–tum FTT to analyse the power developed and efficiency of the microscopic thermal systems and also to optimise them [27, 28]. These models involve the applications of thermodynamic theories and quantum theory along with a lot of mathematical computations.

In the current work, rather than finding the irreversibilities and operational conditions in connection with the engine speed, or complex FTT modelling as seen in the latest works, the whole process is divided into small time-steps. This technique is used to repro–duce the impact of irreversibilities, the effect of various physical/chemical phenomena and other operating parameters on the SI engine performance, with the introduction of a heat leak term alone at the end of every time-step.

In this work, various processes occurring in an Otto cycle are assumed to be quasi-static with a heat leak from the system occurring at every equilibrium point; accounting for the irreversibility of the system. The more the number of equilibrium points, the more intently it will tend to behave as a reversible process, getting the results close to the actual values. More information about the developed FTT model is provided in the sections that follow.

2. FINITE TIME THERMODYNAMIC MODELLING

Generally, the thermodynamic analysis of an SI engine incorporates, mass flow rate and stoichiometric calculations along with the models for knock and dissociation. This makes the analysis more convoluted and tedious. In the current work, a simple model similar to that of the air standard cycle analysis, but could predict the results much more accurately and close to the real value is introduced. Figure 1 shows a schematic illustration of the newly created model.

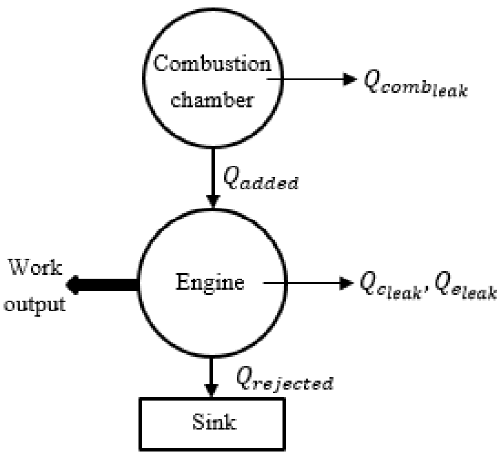

Figure 1. The schematic diagram of the Finite Time Ther–modynamic Model developed and associated energy interactions.

In the present model, it is presumed that the compression and expansion process is occurring inside the engine cylinder while the combustion of fuel is occurring inside a separate combustion chamber. Heat energy produced during the combustion is added into the engine cylinder instantaneously at the end of the compression process as in the case of the air standard cycle. The loss of heat during the combustion process is calculated using equation (11) mentioned in section 2.1. The maximum temperature obtained during the com–bustion process is calculated using the air standard assumption. However, the combustion process is not instantaneous and also it depends on several factors like engine speed, stoichiometric ratio, spark timing, in-cylinder temperature, shape and size of the combustion chamber etc. All these factors affect the combustion efficiency of the cycle leading to lower energy release resulting in a reduced in-cylinder temperature than that estimated using the air standard assumption. In addition to this, the heat lost from the combustion chamber to the surrounding further reduces the in-cylinder temperature. For considering all these parameters, in the current work, the maximum temperature achieved is determined by considering the adiabatic burning of fuel and the energy losses from the combustion chamber during the combustion process are taken into account by intro–ducing a heat leak term. Figure 1 exemplifies the thermodynamic model's schematic diagram and various energy interactions associated with it. The compression and expansion processes are divided into small ele–ments/time steps and the irreversibilities are accounted as a heat leak from the system at the end of each time step. While compression and expansion are taking place, the number of time step is varied during the ana–lysis to study its dependency on the accuracy of results obtained. As the number of time-steps during each process is increased, the number of equilibrium point's in between the process also gets increased and the process tends to become quasi-static. In this manner, the processes could be dealt with as reversible processes.

The major assumption made for the analysis is that there is no mass interaction between the systems. The air inside the engine cylinder is compressed and the energy available with the compressed air is calculated after accounting for the irreversibilities at the end of each time step. At the end of the compression process, heat energy equivalent to the combustion of fuel is supplied to the engine cylinder from the combustion chamber. Later, upon receiving all these energies, the working fluid (air) inside the engine expands to deliver work. The remaining energy is expelled to complete the cycle. From the literature, it is seen that, about 12% of the energy is lost as heat loss during the combustion and expansion process [28]. However, in the current work, all the irreversibilities including the heat loss are accounted for only by using the heat leak terms. As a result, in the current study, the only reason that the thermal efficiency deviates from unity is due to the heat leak that occurs throughout vari–ous processes and the heat rejection that occurs at the end of the expansion process. The amount of heat leak during the combustion process and the heat rejected at the end of the expansion process is approximately 52% at sto–ichiometric mixture and compression ratio 8. The rema–ining amount of heat energy is lost at the end of every time step during the compression and expansion process, calculated using equations (4) and (5). Further details about the time step and its dependency on the accuracy of the results are discussed the sections 2.1 and 3.1.

The following sections provide further information on the thermodynamic relations used to create the model.

2.1 Thermodynamic modelling for calculating the heat interactions

The temperature and pressure variations for each time step during the compression/expansion process are cal–culated using the isentropic relations. Since the irreversibilities are accounted for after each time step, the isentropic relations hold good for the remaining process.

$$\frac{P_{i+1}}{P_i} = \left(\frac{V_i}{V_{i+1}}\right)^{\gamma} \tag{1}$$

$$\frac{T_{i+1_{initial}}}{T_i} = \left(\frac{P_{i+1}}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \tag{2}$$

where, i denotes the instantaneous values at each time step and $i+1_{initial}$ denotes the temperature/pressure cor–responding to the next time-step, before accounting for the heat leak. The ratio of specific heat is evaluated using equations (6) and (7).

The temperature of the air in the next time-step is calculated after accounting for the heat leak term using the following relation.

$$T_{i+1_{final}} = T_{i+1_{initial}} - \frac{Q_{leak_i}}{m_a C_p} \tag{3}$$

While the heat leak corresponding to each time-step during compression and expansion processes is cal–culated as,

$$\left(Q_{c_{leak}}\right)_i = h_i A_i \left(T_{c_i} - T_a\right) t^*_{c_i} \tag{4}$$

$$\left(Q_{e_{leak}}\right)_i = h_i A_i \left(T_{e_i} - T_a\right) t^*_{e_i} \tag{5}$$

The specific heat of air at constant pressure during each time step is obtained from the polynomial equations developed by NASA [29].

For temperatures ranging from 200 K to 1000 K, the relation for specific heat is [29],

$$\begin{aligned}\frac{C_p}{R_g} &= 3.08793 + 12.4597\times10^{-4} T_i \\ &- 0.42372\times10^{-6} T_i^2 + 67.4775\times10^{-12} T_i^3 \\ &- 3.97077\times10^{-15} T_i^4\end{aligned} \tag{6}$$

However, for the temperature ranging from 1000 K to 6000 K, the relation for specific heat becomes [29],

$$\begin{aligned}\frac{C_p}{R_g} &= 3.08793 + 12.4597\times10^{-4} T_i \\ &- 0.42372\times10^{-6} T_i^2 + 67.4775\times10^{-12} T_i^3 \\ &- 3.97077\times10^{-15} T_i^4\end{aligned} \tag{7}$$

The following relationships are used to estimate the volume of the working fluid within the engine cylinder at any instant as well as the combustion chamber's ins–tantaneous surface area as a function of crank angle [14],

$$\begin{aligned}V_i(\theta) &= \frac{V_d}{r-1} \\ &+ \frac{V_d}{2}\left[1 + \frac{l}{r_c} - \cos(\theta)\left(\left(\frac{l}{r_c}\right)^2 - \sin^2(\theta)\right)^{\frac{1}{2}}\right]\end{aligned} \tag{8}$$

Hohenberg's correlation is used to obtain the heat transfer coefficient to estimate the heat leak. [30].

$$h_i = C_1 V_i^{-0.06} P_i^{0.8} T_i^{-0.4} \left(C_2 + \overline{S}_p\right)^{0.8} \tag{10}$$

The constants C_1 and C_2 are 130 and 1.4 respec–tively [30, 31].

During the transfer of heat produced during the combustion of fuel from the combustion chamber into the engine cylinder, heat leak occurring is accounted as,

$$Q_{comb_{leak}} = m_a C_p \left(T_{max} - T_{e_{initial}}\right) \tag{11}$$

T_{max} is the peak temperature obtained using the air standard cycle analysis. However, $T_{e_{initial}}$ is the tempe–rature calculated at adiabatic conditions by keeping the volume constant during the combustion of fuel inside the combustion chamber [17]. The heat of combustion is evaluated at the temperature $T_{c_{final}}$ for calculating the temperature T_{max} and $T_{e_{initial}}$.

The fuel-air equivalence ratio is calculated using the relation:

$$\phi = \frac{\left(F/A\right)_a}{\left(F/A\right)_s} \tag{12}$$

$\eta_{th} = \dfrac{W}{Q_{comb}}$ is the actual fuel-air ratio and $\left(F/A\right)_s$ is the stoichiometric fuel-air ratio.

The heat liberated during the combustion of fuel (Q_{comb}) and the heat rejected to sink from the engine cylinder ($Q_{rejected}$) is obtained using the relations,

$$Q_{comb} = m_f CV \tag{13}$$

$$Q_{rejected} = m_a C_v \left(T_{exhaust} - T_a\right) \tag{14}$$

The mass of fuel is varied in the combustion cham–ber to get the heat of combustion at various air-fuel ratios. The results thus obtained using FTT are com–pared with that of the actual cycle analysis or the experimental values at various equivalence ratios. The internal irreversibilities such as dissociation of the products of combustion at higher temperatures, engine knocking, incomplete combustion of the fuels, frictional losses, and the changes in volumetric efficiency are not considered in this work. Instead, it is assumed that all these irreversibilities are accounted for through the heat leak term incorporated in the FTT model.

2.2 Relations for calculating the Power developed and cycle thermal efficiency

The First Law of Thermodynamics serves as the foun–dation for the Finite Time Thermodynamic model that was created to analyze the Otto cycle. Work done during the cycle is calculated as the algebraic sum of the heat interactions between the systems.

$$W = Q_{added} - Q_{rejected} \tag{15}$$

The combustion chamber delivers heat to the engine at the end of the compression process, elevating the working fluid's temperature from $T_{c_{final}}$ to $T_{e_{initial}}$. Where, $Q_{rejected}$ is the amount of heat rejected from the engine cylinder at the end of the expansion process. After taking the heat loss into account, the quantity of heat given to the engine cylinder from the combustion chamber is calculated as,

$$Q_{added} = Q_{comb} - Q_{comb_{leak}} \tag{16}$$

The thermal efficiency of the cycle is found using the relation,

$$\eta_{th} = \frac{W}{Q_{comb}} \tag{17}$$

Table 1. Engine geometry parameters used for the analysis.

Item	Content
Bore	70mm
Stroke length	90mm
Connecting rod length	197.3 mm
Compression ratio	1:12
Ambient pressure	1 bar
Ambient temperature	303 K
Fuel	Isooctane
Stoichiometric Air Fuel ratio	14.5

3. RESULT AND DISCUSSION

Numerical analysis based on the FTT model discussed in the previous section is performed and is compared to check the accuracy. The dependency of the model on the engine operating parameters like engine speed, compression ratio and air-fuel ratio are also analysed. The engine geometry parameters and the fuel properties considered for the analysis are shown in Table 1.

3.1 Dependency of FTT model developed on the number of time steps

Evaluation of thermal efficiency

Thermal efficiency is calculated using the FTT model created by dividing each $100°CA$ into 10, 100, 1000, 10000, and 100000 equilibrium points or time-steps or elements. As each process takes $180°CA$, the number of time steps obtained for each process was 18, 180, 1800, 18000, & 180000 respectively. Figure 2, shows the change in thermal efficiency of the cycle with the number of time-steps/equilibrium points at 2400 RPM (the engine speed chosen for the analysis is 2400 RPM, since these results are used to compare with the results obtained by Lavoie *et al.* [32], which was at 2400 RPM) for various fuel-air equivalence ratios using the FTT model. The results indicate that the engine's estimated thermal efficiency changes as the number of time-steps increases and finally shows convergence at time steps higher than 1,800 divisions with a variation of less than 0.05%. This is because, as the number of time-steps increases, the number of equilibrium states in the process also gets increased and thus achieving close proximity to the quasi-static process. From Fig. 2 it is clear that, at lower equivalence ratios, the thermal efficiency obtained is less. However, with the increase in the equivalence ratio, thermal efficiency starts to increase up to a particular extent and then starts to reduce further. This result shows the same trend as that of the actual cycle analysis [29].

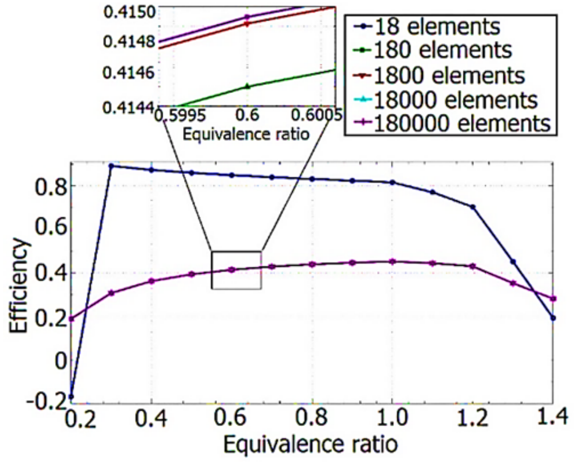

Figure 2. Variation in thermal efficiency with the number of equilibrium states provided using FTT.

Evaluation of power developed

Figure 3 shows the variation in power calculated using the FTT model at 2400 RPM for various fuel-air equivalence ratios. The power calculated using the FTT model gets converged at a time-step of 1800 divisions with a variation of about 0.02%. After convergence, it is evident that the power generated first increases with the equivalence ratio and subsequently starts to decrease at rich mixtures. In the actual case, it is observed that for the lean mixture, the heat released during the combustion process is less leading to lower flame temperature and in turn lower flame speed.

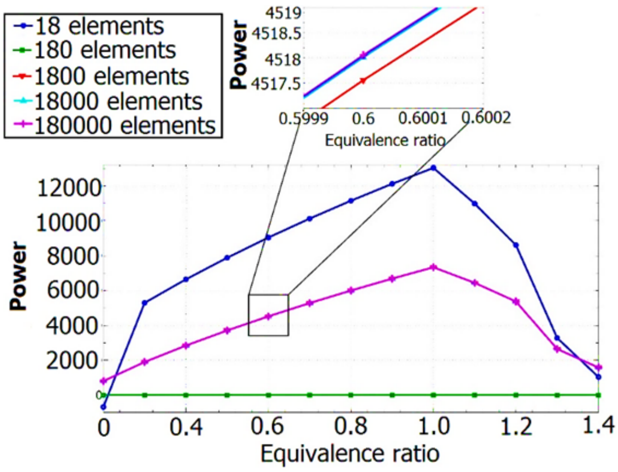

Figure 3. Variation in the power with the number of equilibrium states provided using the FTT.

The heat released will be again low at very rich mixtures also. This is due to incomplete combustion, resulting in a lower flame temperature and lower flame speed. Theoretically, complete combustion and maxi–mum heat release should be observed in the stoichi–ometric mixture, while in the actual case, complete combustion will not occur in stoichiometric mixtures, leaving unutilized oxygen. Therefore, complete utili–zation of the available oxygen will be practically pos–sible at a slightly rich mixture. Hence, maximum flame speed, maximum temperature and pressure will be observed at a slightly rich mixture. This in turn affects the maximum power developed by the engine. Another reason for the deviation in the peak values at the sto–ichiometric mixture is dissociation. Theoretically, the maximum temperature would have been produced at the stoichiometric mixture, leading to the dissociation of gases, thus reducing the temperature and power developed. In the actual case, as mentioned previously, the maximum temperature is observed at 6% richness [28], where the dissociation of gases will be maximum, leading to the reduction in the power developed. As the richness of the fuel increases further, the in-cylinder temperature gets reduced; this in turn reduces the effect of dissociation leading to the increase in the pressure and power developed. During the combustion process, the combustion chamber wall temperature and residual gas temperature rise with the fuel-air equivalence ratio [33]. As a result, the heat loss during the combustion stage rises, affecting the power generated at higher fuel-air equivalence ratios. [33]. Combining all these effects, the maximum power developed will be at rich mixtures, but will get reduced with the increase in richness beyond a certain value. From the literature, the maximum power developed is observed around the fuel-air equivalence ratio of 1.2 for conventional SI engines [28]. However, in the current work, the maximum power developed is found to be maximum at the stoichiometric mixture. This is because; the model proposed had introduced a heat leak term alone to account for all the irreversibilities occurring in an SI engine. There are a lot of factors affecting the power developed by an SI engine, including the time loss factor and heat loss factor and other factors discussed previously. Incorporating all these factors makes the model more complex. Current work aims to develop a simple model similar to that of the air standard cycle analysis but could predict the power developed and the thermal efficiency of an SI engine much close to the real values.

Effect of Thermal efficiency on compression ratio

Figure 4 demonstrates the variation in thermal effici–ency with compression ratio at different fuel-air equi–valence ratios for an engine speed of 2400 RPM. From the results, it is observed that the percentage of increase in thermal efficiency is getting reduced with the increase in compression ratio. At stoichiometric mix–tures, the thermal efficiency has increased by 1.98%, 1.7%, 1.5% and 1.33% at the compression ratios 9, 10, 11 and 12 respectively. The percentage deviation calculated shows the variation of thermal efficiency calculated at a compression ratio to that of its prede–cessor. This reducing trend may be due to the effect of dissociation at high temperatures (in actual cases). With an increase in compression ratio [34], the temperature and pressure inside the engine cylinder increase, acce–lerating the dissociation of gases. Consequently, the rate of increase in thermal efficiency with compression ratio is decreased. Even though the relations to find the dissociation of gases are not included in the model, still, the effects of dissociation are reflected in the results. This is due to the increased rate of heat leak at every equilibrium point with the increase of in-cylinder temperature at higher compression ratios. From the results obtained, it is also observed that the maximum thermal efficiency is at a slightly lean mixture, which is close to the actual measurements. From the literature, the maximum thermal efficiency is found to be occurring at slightly lean mixtures varying from 0.75 to 0.9 equivalence ratio [19, 28].

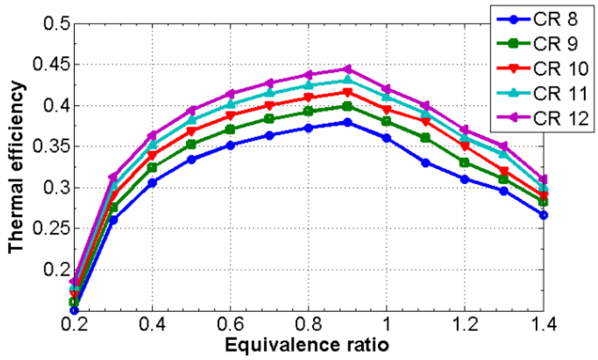

Figure 4. Variation of thermal efficiency with the fuel-air equivalence ratio at various compression ratios.

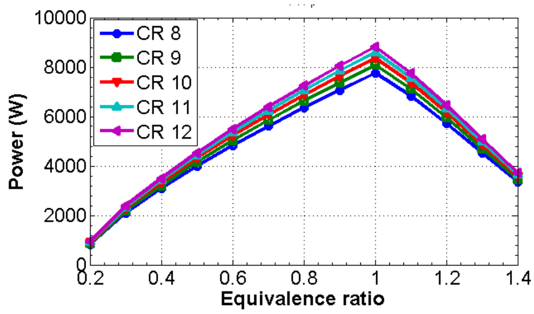

Figure 5. Variation of power developed with the fuel-air equivalence ratio at various compression ratios.

Effect of power developed on compression ratio

Figure 5 shows the power calculated at various comp–ression ratios and fuel-air equivalence ratios at an engine speed of 2400 RPM using FTT. The result shows that the power developed is getting increased with the compre–ssion ratio. This is due to several factors. The increase in compression ratio improves the pressure developed in the unburnt mixture at the end of the compression process, this in turn increases the heat of combustion [34] and hence the pressure developed during the combustion process. Furthermore, studies demonstrate that when the compression ratio increases, the exhaust gas temperature decreases, resulting in less unutilized energy and an improvement in thermal efficiency and power developed. Also, the heat loss through the cylinder walls gets reduced with the increase in the compression ratio.

Comparison of the thermal efficiency calculated using the FTT Model with the other models.

Figure 6 compares the thermal efficiency determined by the proposed FTT Model, the relations developed by Osman [16], Air standard cycle analysis and the experimental data obtained from Lavoie *et. al.* [32]. All these calculations are made at an engine speed of 2400 RPM. At lower fuel-air equivalence ratios, the thermal efficiency calculated using the FTT Model behaves si–milar to that of the brake thermal efficiency plot obtained by Lavoie *et al.* [32] with a deviation of 0-15%. However, at higher equivalence ratios, it shows a deviation of 0-10 % only. The thermal efficiency calculated using the air standard cycle analysis and the relations proposed by Osman [16] shows a deviation of 67-158% and 15-165% when compared with the brake thermal efficiency values obtained from Lavoie *et al.* [32]. The FTT Model proposed in this paper provides more accurate results than those obtained using the air standard cycle analysis or from the relations proposed by Osman [16]. However, the most interesting fact is that the physical or chemical phenomenon that occurs inside the engine is not considered in the FTT model. Instead, all the irreversibilities occurring are assumed to be equivalent to the heat leak from the system.

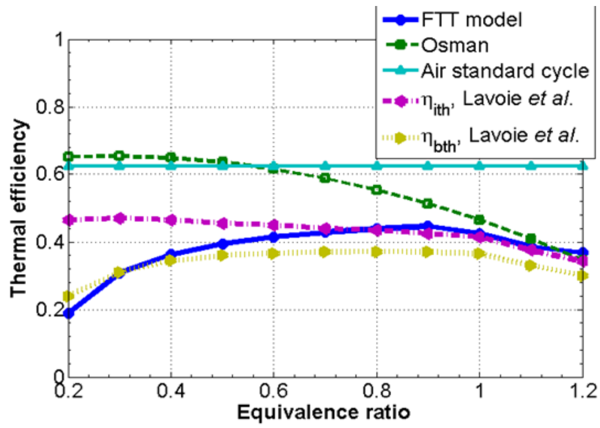

Figure 6. Variation of thermal efficiency with the fuel-air equivalence ratio and engine speed using various models.

Comparison of the results obtained for the power developed by the engine using the FTT Model and with the other models

Figure 7 shows the variation of power developed with the fuel-air equivalence ratio at an engine speed of 2400 RPM. Power calculated using the FTT Model, the relations proposed by Osman [16] and Air standard cycle analysis are presented in Fig. 7. The power calculated using the air-standard cycle analysis is seen to be increasing along with the equivalence ratio. This is because, during air standard cycle analysis, as the equivalence ratio increases, the amount of heat supplied during the constant volume heat addition increases. Thereby, increasing the peak pressure produced and hence the power developed by the cycle. The power calculated using the relations proposed by Osman [16] is almost close to the air standard cycle analysis at lean mixtures, while they are approaching towards the results obtained using the FTT Model at rich mixtures. Power calculated using the FTT Model is behaving similar to that of the actual cycle analysis. At lean mixtures, the power developed is less due to the lower density of the gaseous species. The power developed increases gradually with the decrease in the leanness of the mixture and will be maximum around the stoichiometric mixture. Then, with the further increase in the richness of the mixture, the power developed gets reduced due to the incomplete combustion of the fuel.

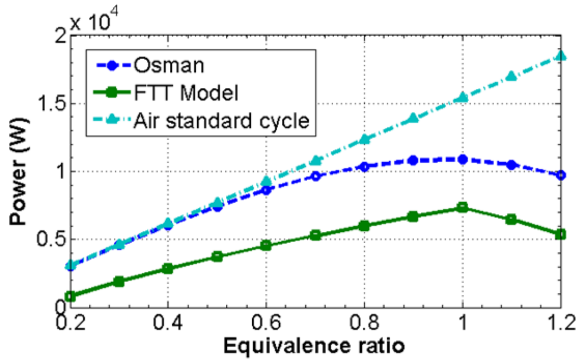

Figure 7. Variation of power with the fuel-air equivalence ratio and engine speed using various models.

4. CONCLUSION

The results produced by the FTT model show close proximity to the actual cycle analysis, which otherwise should have been modelled using complex relations. From the results obtained, it is evident that the maxi–mum thermal efficiency is observed at a slightly lean mixture. The same trend is observed in conventional SI engines also. However, some deviations were observed in the values obtained for the power developed. The power developed will be maximum at a slightly rich mixture having a fuel-air equivalence ratio of around 1.2. However, from the results obtained for the power developed using the FTT model, the maximum value is observed at stoichiometric mixtures. This deviation is due to the assumptions taken while developing the model to keep it very simple. The FTT model developed is having relations similar to that used in the Air standard cycle analysis. Thus, with less complexity, more realistic results could be obtained. This shows the importance of these models or similar techniques in thermal system analysis.

The thermal efficiency calculated using the FTT model shows a deviation of 0-15% from the values of brake thermal efficiency obtained by Lavoie et al. [32] at lean mixtures and a deviation of 0–10% at the rich mixtures. While air standard efficiency had shown a deviation of 67- 158% from the actual values. The results obtained using the relations proposed by Osman [16] behaves like an air standard cycle at lean mixtures and as fuel-air cycle at the rich mixtures.

In order to obtain more accurate results for the analysis of IC engines, considering the flammability limits, Osman [16] has established a valid range for the heat loss parameter through the relationship between fuel's chemical energy and heat leakage. In the current work, heat loss is considered to be a function of the temperature of the working fluid alone. From the results obtained using the FTT model, it is clear that, if the equivalence ratio is increased or decreased beyond the flammability limit, the thermal efficiency and power calculated tend to become zero. Thus, making sense that, it is impossible for the engine to work normally beyond the flammability limits. Therefore, it is not ne–cessary to have a fixed range for the heat loss parameter as in the work done by Osman to get realistic results [16].

The proposed Finite Time Thermodynamic model can be utilised to analyse the performance of SI engines with accuracy that is near to realistic values. A similar approach could be used to analyse the performance of other thermal systems also by modifying the current model suitably to equip the thermodynamic processes associated with it. Also, the model can be used to calculate the upper limit of the efficiency for a thermal system; beyond which a real system cannot achieve.

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NOMENCLATURE

P	Pressure of the working fluid during the compression and expansion process, [Pa]
V	In-cylinder volume, [m^3]
γ	Ratio of specific heats, [-]
Q_{leak}	Heat leak from the engine cylinder during the compression/expansion process, [kJ]
Q_{cleak}	Heat leak from the engine cylinder during the compression process, [kJ]
Q_{eleak}	Heat leak from the engine cylinder during the expansion process, [kJ]
h_i	Instantaneous heat transfer coefficient, [kJ/m^{2}K]
A_i	Instantaneous surface area of the cylinder, [m^2]
T_{c_i}	Instantaneous temperature during compression process, [K]
T_{e_i}	Instantaneous temperature during expansion process, [K]
T_a	Ambient temperature, [K]
$t^*_{c_i}$	Fraction of second taken at each time step during the compression process $\left(=\frac{t_{c_i}}{t}\right)$, [-]
t_{c_i}	Time taken for each time step during the compression process, [s]
$t^*_{e_i}$	Fraction of second taken at each time step during the expansion process $\left(=\frac{t_{e_i}}{t}\right)$, [-]
t_{e_i}	Time taken for each time step during the expansion process, [s]
t	Unit time (1 s), [s]
R_g	Characteristic gas constant of the working fluid, [KJ/kg K]
θ	Crank angle, [degree]
r	Compression ratio, [-]
V_d	Displacement volume, [m^3]
b	Bore diameter, [m]
t	Stroke length, [m]
l	Connecting rod length, [m]
r_c	Crank radius, [m]
$\bar{S}_p$	Mean piston speed, [m/s]

$Q_{combleak}$	Heat leak from the combustion chamber, [kJ]
T_{max}	Maximum temperature produced during the combustion process, [K]
$T_{e_{initial}}$	Temperature at the beginning of the expansion process, [K]
m_a	Mass of air in the cylinder, [kg]
m_f	Mass of fuel burned in the combustion chamber, [kg]
$T_{c_{initial}}$	Temperature of working fluid at the beginning of compression process, [K]
$T_{c_{final}}$	Temperature of working fluid at the end of compression process, [K]
CV	Calorific value of the fuel, [kJ/kg]
T_{exhaus}	Temperature of the working fluid at the end of the expansion process, [K]

ТЕРМИН ЦУРЕЊА ТОПЛОТЕ - ПОТПИС НЕПОВРАТНОСТИ У АНАЛИЗИ СТВАРНОГ ЦИКЛУСА СИ МОТОРА КОРИШЋЕЊЕМ ТЕРМОДИНАМИКЕ КОНАЧНОГ ВРЕМЕНА

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Због напретка у иновацијама, на крају је приметно неизбежно урадити сложену термодинамичку анализу термичких система. Главни проблем са којим се сусреће током развоја термодинамичког модела за анализу је да се провери изводљивост предложеног модела у супротности са тренутним опцијама. Термодинамика коначног времена може се адекватно искористити да би се ово испунило без одласка у запањујућу истрагу. Ипак, ако се процес подели на бесконачно мале процесе са неповрат–ностима које се узимају у обзир у свим условима равнотеже; могао би се користити реверзибилни термодинамички приступ. Овај рад показује ефи–касност оваквог приступа и грешке које су праћене анализом Ото циклуса. Модел коначног времена креиран у овом раду је истражен и супротстављен стварним вредностима. За израчунату термичку ефикасност, примећено је да је термодинамички модел коначног времена тачнији од других сличних техника које не укључују сложену термодинамичку анализу. Креирани термодинамички модел коначног времена објашњава читав комплексни феномен који се јавља у СИ мотору са увођењем само термина цурења топлоте у њега.