

# Effect of Heat Transfer on the Output Performance Characteristics of an Air-Standard Otto Cycle

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## ABSTARCT:

This project presents an analysis of the thermodynamic engine spark ignition (SI). The implementation of the theoretical model of the Otto cycle heats the existence of specific heat temperature standard. It was comparable to the constant temperature heat set temperature employed. The study has a wide range of engine parameters. In most cases, there were no significant differences between the results obtained by using the temperature with the high specific temperature dependent constant obtained with specific classification, especially at high speeds. Therefore, it is more realistic to use temperature specific heat. This should be considered in the analysis mainly because the temperature difference in the actual sessions is a big cycle. Numerical experiments by writing computer code in C, shots, cylinders stories temperature of the expected pressure.

**INTRODUCTION:** In the thermal design of internal combustion engines Mostresearchers used standard cycle models air power to perform thermal analysis. These models are used for comparison reasons in order to show the effect of the engine and the conditions and characteristics of fluid parameters, etc. degrees in most of the previous studies on a power cycle standard air, and it was from the air is assumed as the working fluid as an ideal gas with specific fixed with high temperatures without temperature dependence taking into account high temperatures of specific resources. However, due to a large increase in the combustion temperature becomes this assumption less realistic. Although the power cycle analysis standard air gives only an approximation to the conditions and actual results, which will be very useful for comparing the performance of air power cycles with varying certain standard assumptions constantly heated. In a recent

study the effect of various parameters internal combustion engines in the SI engine was studied. Some studies have the effect of temperature. Made with high specific temperatures depend on different recording sessions the air like Otto, Diesel, and Miller. It was, however, use the temperature of the sample with a high specific temperature it depends on the linear model. It can be used as a specific heat set to temperature changes are very small models. Moreover, the linear model can be applied to moderate temperature changes. However, to make significant changes in temperature, and there is a need for more accurate models. In this study it will be a more realistic approach applied to the behavior of the variable specific classification of the SI engine performance. In most cycle models standard air supply is supposed to behave as an ideal gas air heated to a specified fixed. In general, use values specific classification of the cold. This assumption may be only slight differences in the correct temperature. However, the assumption that the production of the greatest error in all cycles standard air supply.

### **Two Stroke Spark Ignition (SI) engine:**

SI engine running two cycles was completed within two strokes of the piston or a

complete revolution (360 degrees) of the crankshaft. In this engine strokes ports have intake and exhaust are used instead of valves. In this session, it is mixed with gasoline and lubricating oil, resulting in a simpler system, but more damaging to the environment, as excess oil not burned and left as residue. With the continuation of the piston down to open another port, the intake port fuel / air. mixture / fuel / air oil comes from the carburetor, where it mixes with the remaining fuel in the neighboring room. When the piston moves down and the cylinder over not have any other gas, the fuel mixture starts flowing the combustion chamber and the process starts from the second fuel pressure. cumbersome design processes and carefully consider the extent that the mixture of fuel and air should not be mixed with the exhaust gas, so the fuel injection and are synchronized to avoid this problem. It should be noted that the piston has three functions in their work:

- combustion chamber when the piston with the cylinder and compresses the air / fuel mixture, receives back the energy released, and become the crankshaft.
- The movement of the piston creates a vacuum that sucks the fuel / air carburetor and paid from the crankcase (adjacent room) to the combustion chamber.



And • aspects of work, such as piston valves, cover and uncover the intake and exhaust ports and drilling on the other side of the cylinder wall.

## LITERATURE REVIEW:

Al-Sarkhi, B. Akash, J. Jaber, M. Mohsen, E. Abu-Nada,[2] Efficiency of Miller engine at maximum power density air was assumed as the working fluid as an ideal gas with constant specific heats without taking into consideration temperature dependence of the specific heats of the working fluid.

B. Akash,[1] Effect of heat transfer on the performance of an air-standard diesel cycle, air was assumed as the working fluid as an ideal gas with constant specific heats without taking into consideration temperature dependence of the specific heats of the working fluid.

M. Karamangil, O. Kaynakli, A. Surmen,[12] Parametric investigation of cylinder and jacket side convective heat transfer coefficients of gasoline engines effect of various internal combustion engines parameters in the SI engine were studied .

Al-Sarkhi, J. Jaber, M. Abu-Qudais, S. Probert,[13] Effects of friction and

temperature-dependent specific-heat of the working fluid on the performance of a diesel- engine, presented the effect of having temperature dependent specific heats on various air-standard cycles such as Otto, Diesel, and Miller .

A. Al-Sarkhi, J. Jaber, S. Probert,[14] Efficiency of a Miller engine, presented the effect of having temperature dependent specific heats on various air-standard cycles such as Otto, Diesel, and Miller .

Y. Ge, L. Chen, F. Sun, C. Wu,[15] Thermodynamic simulation of performance of Otto cycle with heat transfer and variable specific heats of working fluid presented the effect of having temperature dependent specific heats on various air-standard cycles such as Otto, Diesel, and Miller .

A. Jafari, S. Hannani, [8] Effect of fuel and engine operational characteristics on the heat loss from combustion chamber surfaces of SI engines, air was assumed as the working fluid as an ideal gas with constant specific heats without taking into consideration temperature dependence of the specific heats of the working fluid.

## METHODOLOGY:

Combustion may be defined as a relatively rapid chemical combination of



hydrogen and carbon in fuel with oxygen in air resulting in liberation of energy in the form of heat.

**Following conditions are necessary for combustion to take place:**

1. The presence of combustible mixture
2. Some means to initiate mixture
3. Stabilization and propagation of flame in Combustion Chamber

In S I Engines, carburetor supplies a combustible mixture of petrol and air and spark plug initiates combustion

**FACTORS AFFECTING THE FLAME PROPAGATION**

Rate of flame propagation affects the combustion process in SI engines. Higher combustion efficiency and fuel economy can be achieved by higher flame propagation velocities. Unfortunately flame velocities for most of fuel range between 10 to 30 m/second.

**The factors which affect the flame propagations are:**

1. Air fuel ratio
2. Compression ratio
3. Load on engine
4. Turbulence and engine speed
5. Other factors

**1. A: F ratio**

The mixture strength influences the rate of combustion and amount of heat generated. The maximum flame speed for all hydrocarbon fuels occurs at nearly 10% rich mixture. Flame speed is reduced both for lean and as well as for very rich mixture. Lean mixture releases less heat resulting lower flame temperature and lower flame speed. Very rich mixture results in incomplete combustion (C CO instead of C<sub>0</sub> and also results in production of less heat and flame speed remains low.

**2. Compression ratio**

The higher compression ratio increases the pressure and temperature of the mixture and also decreases the concentration of residual gases. All these factors reduce the ignition lag and help to speed up the second phase of combustion. The maximum pressure of the cycle as well as mean effective pressure of the cycle with increase in compression ratio. Higher compression ratio increases the surface to volume ratio and thereby increases the part of the mixture which after-burns in the third phase.

**3. Load on Engine**



With increase in load, the cycle pressures increase and the flamespeed also increases. In S.I. engine, the power developed by an engine is controlled by throttling. At lower load and higher throttle, the initial and final pressure of the mixture after compression decrease and mixture is also diluted by the more residual gases. This reduces the flame propagation and prolongs the ignition lag. This is the reason, the advance mechanism is also provided with change in load on the engine. This difficulty can be partly overcome by providing rich mixture at part loads but this definitely increases the chances of afterburning. The after burning is prolonged with richer mixture. In fact, poor combustion at part loads and necessity of providing richer mixture are the main disadvantages of S.I. engines which causes wastage of fuel and discharge of large amount of CO with exhaust gases.

#### 4. Turbulence

Turbulence plays very important role in combustion of fuel as the flamespeed is directly proportional to the turbulence of the mixture. This is because, the turbulence increases the mixing and heat transfer coefficient or heat transfer rate between the burned and unburned mixture. The turbulence of the mixture can be increased at

the end of compression by suitable design of the combustion chamber (geometry of cylinder head and piston crown). Insufficient turbulence provides low flame velocity and incomplete combustion and reduces the power output. But excessive turbulence is also not desirable as it increases the combustion rapidly and leads to detonation. Excessive turbulence causes to cool the flame generated and flame propagation is reduced. Moderate turbulence is always desirable as it accelerates the chemical reaction, reduces ignition lag, increases flame propagation and even allows weak mixture to burn efficiently.

#### Engine Speed:

The turbulence of the mixture increases with an increase in engine speed. For this reason the flame speed almost increases linearly with engine speed. If the engine speed is doubled, flame to traverse the combustion chamber is halved. Double the original speed and half the original time give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation, which is main phase of combustion will remain almost constant at all speeds. This is an important characteristic of all petrol engines.

## Engine Size:

Engines of similar design generally run at the same piston speed. This is achieved by using small engines having larger RPM and larger engines having smaller RPM. Due to same piston speed, the inlet velocity, degree of turbulence and flame speed are nearly same in similar engines regardless of the size. However, in small engines the flame travel is small and in large engines large. Therefore, if the engine size is doubled the time required for propagation of flame through combustion space is also doubled.

But with lower RPM of large engines the time for flame propagation in terms of crank would be nearly same as in small engines. In other words, the number of crank degrees required for flame travel will be about the same irrespective of engine size provided the engines are similar.

## 5. Other Factors

Among the other factors, the factors which increase the flame speed are supercharging of the engine, spark timing and residual gases left in the engine at the end of exhaust stroke. The air humidity also affects the flame velocity but its exact effect

is not known. Anyhow, its effect is not large compared with A: F ratio and turbulence.

## SPECIFICATIONS:

### Engine and operational specifications used in simulation:

Fuel	:	C <sub>8</sub> H <sub>18</sub>
Compression ratio	:	8.3
Cylinder bore (m)	:	0.0864
Stroke (m)	:	0.0674
Connecting rod length (m)	:	0.13
Crank radius (m)	:	0.0337
Clearance volume (m <sup>3</sup> )	:	5.41×10 <sup>-5</sup>
Swept volume (m <sup>3</sup> )	:	3.95×10 <sup>-4</sup>
Engine speed (rpm)	:	2000–5000
Inlet pressure (bar)	:	1
Equivalence ratio	:	1
Ignition timing	:	-25° BTDC
Duration of combustion	:	70°
Wall temperature (K)	:	400

## THERMODYNAMIC MODELLING:

For a closed system, the first law of thermodynamics is written as:

$$\delta Q - \delta W = dU \quad (5.1)$$

By using the definition of work, the first law can be expressed as:

$$\delta Q_{in} - \delta Q_{loss} - PdV = dU \quad (5.2)$$

For an ideal gas the equation of state is expressed as:

$$PV = mR_g T_g \quad (5.3)$$

By differentiating Eq. (5.3), the following equation is obtained:

$$PdV + VdP = mR_g dT_g \quad (5.4)$$

Also, for an ideal gas the change in internal energy is expressed as:

$$dU = d(mC_v T_g) \quad (5.5)$$

Using the chain rule of differentiation, Eq. (5.5) is rearranged as:

$$mR_g T_g = \left( \frac{R_g}{C_v} \right) (dU - mT_g dC_v) \quad (5.6)$$

By substituting Eq. (5.6) into Eq. (5.4) and solving for the change in Internal energy:

$$dU = \frac{C_v}{R_g} (pdV + Vdp) + mT_g dC_v \quad (5.7)$$

Also, by substituting Eq. (5.7) into Eq. (5.1), the first law is written as

$$\delta Q_{in} - \delta Q_{loss} - PdV = \frac{C_v}{R_g} (PdV + VdP) + mT_g dC_v \quad (5.8)$$

Dividing Eq. (5.8) by  $d\theta$

$$\frac{\delta Q_{in}}{d\theta} - \frac{\delta Q_{loss}}{d\theta} - P \frac{dV}{d\theta} = \frac{C_v}{R_g} \left( P \frac{dV}{d\theta} + V \frac{dP}{d\theta} \right) + mT_g \frac{dC_v}{d\theta} \quad (5.9)$$

Expressing the gradient of the specific heat as:

$$\frac{dC_v}{d\theta} = \frac{dC_v}{dk} \frac{dk}{d\theta} \quad (5.10)$$

Noting that:

$$\frac{R_g}{C_v} = k - 1 \quad (5.11)$$

Plugging Eq.(5.11) into Eq.(5.10) , then the gradient of the specific heat is expressed as:

$$\frac{dC_v}{d\theta} = - \frac{R_g}{(k-1)^2} \frac{dk}{d\theta} \quad (5.12)$$

Substituting Eq.(5.12) into Eq.(5.9) , the final form of the governing equation is:

$$\frac{dp}{d\theta} = \frac{k-1}{V} \left( \frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta} \right) - k \frac{P}{V} \frac{dV}{d\theta} + \frac{P}{k-1} \frac{dk}{d\theta} \quad (5.13)$$

In Eq.(5.13) , the rate of heat loss  $\frac{dQ_{loss}}{d\theta}$  is

expressed as :

$$\frac{dQ_{loss}}{d\theta} = hA(\theta)(T_g - T_w) \frac{1}{\omega}$$

(5.14)

The convective heat transfer coefficient is given by the Woschni model as [10,15,16]:

$$h = 3.26D^{-0.2} P^{0.8} T_g^{-0.55} w^{0.8} \quad (5.15)$$

The velocity of the burned gas and is given as:

$$w(\theta) = 2.28 \overline{U}_p + C_1 \frac{V_d T_{gr}}{p_r V_r} (p(\theta) - p_m)$$

(5.16)

The quantities  $V_r$ ,  $T_{gr}$ , and  $p_r$  are reference state properties at closing of inlet valve and  $p_m$  is the pressure at same position to obtain  $p$  without combustion (pressure values in cranking). The value of  $C_1$  is given as: for compression process:  $C_1=0$  and for combustion and expansion processes:  $C_1=0.00324$ . The  $\overline{U}_p$  average piston speed is calculated from:

$$\overline{U}_p = \frac{2NS}{60}$$

(5.17)

On the other hand, the rate of the heat input

$\frac{dQ_{in}}{d\theta}$  (heat release) can be modelled using a

simple Weibe function:

$$\frac{dQ_{in}}{d\theta} = a \left( \frac{Q_p}{\theta_p} \right) m_p \left( \frac{\theta}{\theta_p} \right)^{m_p-1} \exp \left( -a \left( \frac{\theta}{\theta_p} \right)^{m_p} \right)$$

(5.18)

Where  $p$  refers to premixed phase of combustion. The parameter  $\theta_p$  represent the duration of the premixed combustion phases. Also,  $Q_p$  represent the integrated energy release for premixed phase. For SI engine entire combustion takes place in premixed state. The constants  $a$ ,  $m_p$  are selected to match experimental data. For the current study, these values are selected as 6.9, 4 respectively. It is assumed that the total heat input to the cylinder by combustion for one cycle is:

$$Q_{in} = m_f LHV$$

(5.19)

Eq. (5.13) is discretized using a first order finite difference method to solve for the pressure at each crank angle ( $\theta$ ) [1–4]. Once



the pressure is calculated, the temperature of the gases in the cylinder can be calculated using the equation of state as:

$$T_g = \frac{P(\theta)V(\theta)}{mR_g} \quad (5.20)$$

The instantaneous cylinder volume, area, and displacement are given as [14]:

$$V(\theta) = V_c + \frac{\pi D^2}{4} x(\theta) \quad (5.21)$$

$$A_h(\theta) = \frac{\pi D^2}{4} + \frac{\pi D S}{2} (R + 1 - \cos(\theta) + (R^2 - \sin^2(\theta))^{1/2}) \quad (5.22)$$

$$x(\theta) = (\lambda + R) - (R \cos(\theta) + (\lambda^2 - \sin^2(\theta))^{1/2}) \quad (5.23)$$

An equation describing the variation of air specific heats for the temperature range 300–3500 K is adopted [17]. The equation is based on the assumption that air is an ideal gas mixture containing 78.1% N<sub>2</sub>, 20.95% O<sub>2</sub>, 0.92% Ar, and 0.03% Co<sub>2</sub> (on mole basis):

It is found from that specific heat at constant pressure increases with temperature from

about 1.0 kJ/kg K at 300 K to about 1.3 kJ/kg K at 3000 K and such difference should be taken into consideration. Similarly, the specific heat ratio, *k*, decreases from 1.40 to about 1.28 within the same temperature range.

### RESULTS:

Parametric studies have been performed based on the numerical solution of Eq. (16). The study covers wide range of dependent variables such as engine speed, air–fuel ratio and others, taking into consideration the variation of the specific heat with temperature.

#### Effect of engine speed on p vs. v diagram using constant-specific heats:

In order to examine the validity and sensitivity of the presented model, cylinder pressure is presented in Fig. 1. It shows the variation of cylinder pressure versus volume for SI engine using constant-average specific heats running at piston speeds of 2000 and 5000 rpm at a given air–fuel ratio of 15. It is obvious that cylinder pressure is higher at higher engine speeds.

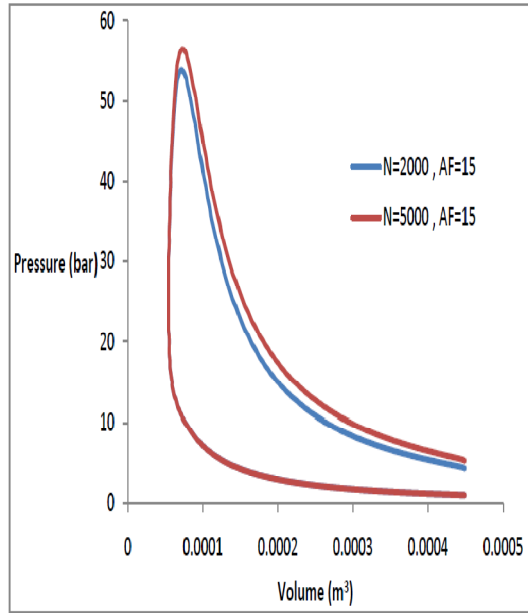


Fig. . Variation of cylinder pressure versus volume for SI engine using constant-specific heats running at 2000 rpm and 5000 rpm at air-fuel ratio of 15.

**Effect of engine air –fuel ratio on p vs v diagram using constant-specific heats:**

Fig. 5.2. Shows the variation of cylinder pressure versus volume for SI engine using constant-average specific heats running at air-fuel ratios of 14, 16 at a given engine speed of 2000 rpm. It is obvious that cylinder pressure is higher lower air-fuel ratios.

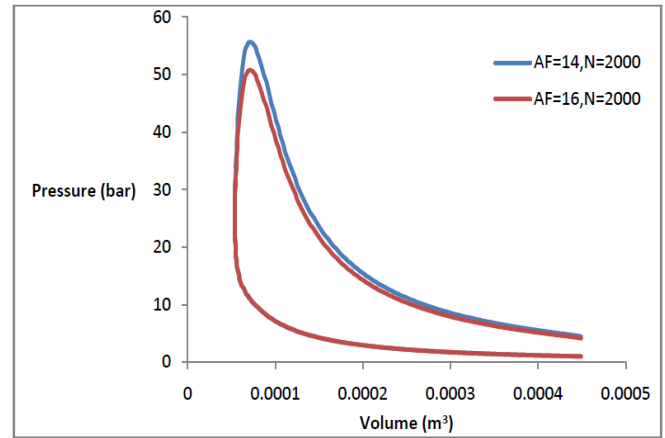


Fig. Variation of cylinder pressure versus volume for SI engine using constant-specific running with air-fuel ratios of 14, 16 and running at engine speed 2000 rpm.

**Effect of temperature dependent specific heats on cylinder pressure profile:**

In order to study the effect of temperature dependent specific heats, Fig. is presented. It shows variation of pressure versus crank angle using variable and constant-average specific heats running at engine speed of 5000 rpm and in-cylinder air–fuel ratio of 15. It is obvious that there is some difference when temperature dependent specific heat is used instead of constant specific heat. Although they have similar trends, the maximum pressure with constant specific heats is significantly over-estimated in comparison with results obtained with variable specific heats.

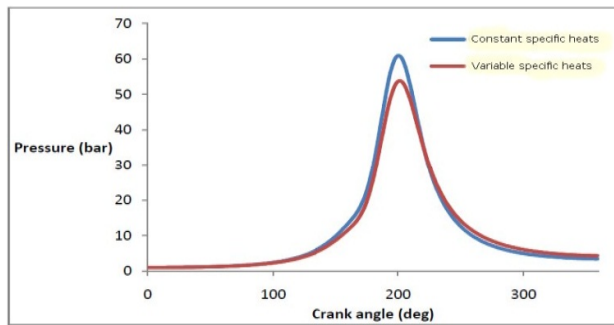


Fig. Variation of cylinder pressure versus crank angle for SI engine using variable and constant specific heats running at 5000 rpm and air-fuel ratio of 15.

**Effect of temperature dependent specific heats on gas temperature profile:**

In order to study the effect of temperature dependent specific heats, Fig. 5.4 is presented. It shows variation of gas temperature versus crank angle using variable and constant-average specific heats running at engine speed of 5000 rpm and in-cylinder air-fuel ratio of 15. It is obvious that there is some difference when temperature dependent specific heat is used instead of constant specific heat. Although they have similar trends, the maximum temperatures with constant specific heats are significantly over-estimated in comparison with results obtained with variable specific heats.

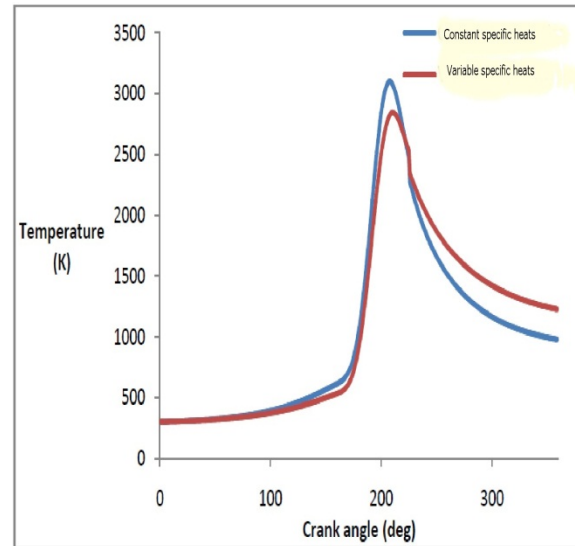


Fig. Variation of gas temperature versus crank angle for SI engine using variable and constant specific heats running at 5000 rpm and air-fuel ratio of 15.

**Effect of temperature dependent specific heats on peak gas temperature by varying the engine speed:**

Fig. presents the maximum gas temperature versus engine speed at air-fuel ratios of 14 (rich mixture) and 16 (lean mixture). Higher values of maximum temperatures are obtained at higher engine speeds and lower air-fuel ratios. Again, as noted previously, the effect of temperature dependent specific heat is very significant on the reported maximum gas temperatures.

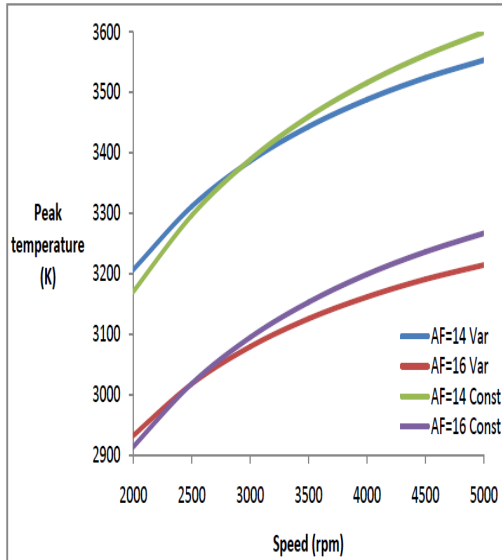


Fig: Maximum gas temperature versus engine speed at different air-fuel ratios using constant and variable specific heat models

## CONCLUSIONS

It follows that the motor parameters are affected by work specific heats variable, dramatically. The results show that there is a significant impact on the specific heat of the temperature depends on the working fluid in the cycle performance standard Otto. Therefore, it is more realistic to use the specified temperature heat during the investigation depends on the level of power cycles. This should be considered in the analysis of the practice session, particularly in the actual cycles of temperature changes are quite large It is expected to provide

important guidance for assessing performance and improving real engines SI results.

## REFERENCES

- [1] B. Akash, Effect of heat transfer on the performance of an air-standard diesel cycle, *International Communications in Heat and Mass Transfer* 28(1) (2001) 87–95.
- [2] A. Al-Sarkhi, B. Akash, J. Jaber, M. Mohsen, E. Abu-Nada, Efficiency of Miller engine at maximum power density, *International Communications in Heat and Mass Transfer* 29 (8) (2002) 1159–1167.
- [3] Y. Ge, L. Chen, F. Sun, C. Wu, Effect of heat transfer and friction on the performance of an irreversible air-standard miller cycle, *International Communications in Heat and Mass Transfer* 32 (2005) 1045–1056.
- [4] L. Chen, C. Wu, F. Sun, S. Cao, Heat transfer effects on the net work output and efficiency characteristics for an air-standard Otto cycle, *Energy Conversion and Management* 39 (1998) 643–648.
- [5] A. Parlak, Comparative performance analysis of irreversible dual and diesel cycles under maximum power conditions, *Energy Conversion and Management* 46 (3) (2005) 351–359.
- [6] A. Parlak, The effect of heat transfer on performance of the diesel cycle and exergy of the exhaust gas stream in a LHR Diesel engine at the optimum injection timing, *Energy Conversion and Management* 46 (2) (2005) 167–179.
- [7] S. Hou, Heat transfer effects on the performance of an air standard dual cycle,



Energy Conversion and Management 45 (2004) 3003–3015.

[8] A. Jafari, S. Hannani, Effect of fuel and engine operational characteristics on the heat loss from combustion chamber surfaces of SI engines, *International Communications in Heat and Mass Transfer* 33 (2006) 122–124.

[9] O. Ozsoysal, Heat loss as a percentage of fuel's energy in air standard Otto and Diesel cycles, *Energy Conversion and Management* 47 (2006) 1051–1062.

[10] P. Wang, S. Hou, Performance analysis and comparison of an Atkinson cycle coupled to variable temperature heat reservoirs under maximum power and maximum power density conditions, *Energy Conversion and Management* 46 (2005) 2637–2655.

[11] W. Pulkrabek, *Engineering Fundamentals of the Internal Combustion Engine*, second ed., Pearson Prentice-Hall, Upper Saddle River, New Jersey, 2004.

[12] M. Karamangil, O. Kaynakli, A. Surmen, Parametric investigation of cylinder and jacket side convective heat transfer coefficients of gasoline engines, *Energy Conversion and Management* 47 (2006) 800–816.

[13] A. Al-Sarkhi, J. Jaber, M. Abu-Qudais, S. Probert, Effects of friction and temperature-dependent specific-heat of the working fluid on the performance of a diesel-engine, *Applied Energy* 83 (2) (2006) 153–165.

[14] A. Al-Sarkhi, J. Jaber, S. Probert, Efficiency of a Miller engine, *Applied Energy* 83 (4) (2006) 343–351.

[15] Y. Ge, L. Chen, F. Sun, C. Wu, Thermodynamic simulation of performance of Otto cycle with heat transfer and variable specific heats of working fluid, *International Journal of Thermal Sciences* 44 (5) (2005) 506–511.