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Effects of Different Frictional Losses Parameters on Performance of an Air Standard Miller Cycle

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Abstract: The effects of different frictional losses parameters on the performance of a Miller cycle during the finite time thermodynamic are investigated. In the cycle model, parameters frictional losses computed from the empirical correlation, the specific heat ratio are varied with temperature, the internal irreversibility described by using the heat transfer loss are considered. The relations between brake power and frictional losses parameters and between thermal efficiency and frictional losses parameters are derived. Moreover, the effects of frictional losses parameters on the cycle performance are analyzed. The results show that the engine RPM and inlet valve diameter are more effective on performance of a Miller engine. Also, the point of maximum brake power of Miller cycle is at compression ratio of 8. The results are of importance to provide good guidance for the performance evaluation, improvement and design of practical a Miller engines.

Key words: Frictional losses, miller cycle, internal irreversibility, finite-time processes, parameters

INTRODUCTION

A series of achievements have been made since finite-time thermodynamics was used to simulate, analyze and optimize the performance of ideal thermodynamic processes, devices and cycles (Jesudason, 2009; Abbassi *et al*., 2012; Parlak, 2005; Chen *et al*., 2008). Miller (1947) proposed a different Otto cycle with an unequal compression and expansion stroke called the air standard Miller cycle in the 1940's. An air standard Miller cycle has been given attention recently (Kesgin, 2005) and some researchers have examined the finite-time thermodynamic performance of the Miller cycle. Hatamura *et al*. (1997) reported that the Miller cycle has advantages such as a higher mean effective pressure and power output than the Otto cycle with lower nominal compression ratio. Fukuzawa *et al*. (2001) described and characterized the main technologies and performance specifications for a high efficiency Miller cycle gas engine as well as for the series of engines planned in the future. Al-Sarkhi *et al*. (2002) compared the performance characteristic curves of the Atkinson cycle with those of the Miller and Joule-Brayton cycles by using numerical simulations and outlined the effect of maximizing power density on the performance of the cycle efficiency. They found that efficiency of Atkinson and Joule-Brayton cycles at the maximum power density is greater than that of Miller cycle. Sasaki *et al*. (2002) related an efficient Miller cycle with a high performance capacitor system for hybrid busses. Wu *et al*. (2003) investigated a performance analysis and optimization of a supercharged Miller cycle Otto engine. This study deals with the analysis of a supercharged Otto engine adopted for Miller cycle operation. The Miller cycle shows no efficiency advantage and suffers a penalty in power output in the normally aspirated version. Ge *et al*. (2005a-c) derived the performance characteristics of the Miller cycle with heat transfer loss (Ge *et al*., 2005c) and with heat transfer and friction losses (Ge *et al*., 2005b), respectively. These works were done without considering the variable specific heats of the working fluid. Ge *et al*. (2005a) also studied the effect of variable specific heats of working fluid on the performance of an air standard Miller cycle. In this study using finite-time thermodynamics, the relations between thermal efficiency, compression and expansion ratios for an ideal naturally-aspirated (air-standard) Miller cycle have been derived. Al-Sarkhi *et al*. (2006) and Zhao and Chen (2007) presented theoretical investigations into the Miller cycle's engine performance, studying the influence of the main engine design parameters and system irreversibility. Al-Sarkhi *et al*. (2007) evaluated the performance of the Miller engine by taking into consideration the different specific heat models. It was found that an accurate model such as the fourth order polynomial is essential for accurate prediction of cycle performance. Lin and Hou (2008) examined the effects of

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heat losses characterized by a percentage of fuel energy, friction losses and variable specific heats of working fluid on the performance of an air standard Miller cycle. They found that the power output as well as the efficiency where maximum power output occurs will increase with the increase of maximum cycle temperature. The temperature-dependent specific heats of working fluid have a significant influence on the performance. The power output and the working range of the cycle increase while the efficiency decreases with the increase of specific heats of working fluid. The friction loss has a negative effect on the performance. Therefore, the power output and efficiency of the cycle decrease with increasing friction loss. Comparison of the performance of air-standard Miller and Otto cycles show that the Miller cycle has larger power output and efficiency than Otto cycle does, i.e., Miller cycle is more efficient than Otto cycle.

In all of the earlier mentioned researches, the specific heats at a constant pressure and volume of working fluid are assumed to be constants or functions of temperature alone and have linear, nonlinear and polynomial forms. But when calculating the chemical heat released in combustion process at each instant of time for ideal internal combustion engine, the specific heat ratio is generally modeled as a linear function of mean charge temperature (Gatowski *et al*., 1984). The model has been used and the phenomena that it takes into account are well known (Klein, 2004). However, since the specific heat ratio has a great influence on the heat release peak and on the shape of the heat release curve (Brunt *et al*., 1998), some investigators have elaborated different mathematical equations to describe the dependence of specific heat ratio on temperature (Gatowski *et al*., 1984; Klein, 2004; Brunt *et al*., 1998; Ebrahimi, 2011a; Ceviz and Kaymaz, 2005). Ebrahimi (2011c) considered specific heat ratio as linear with temperature. It should be mentioned here that the most important thermodynamic property used in the heat release calculations for engines is the specific heat ratio (Ceviz and Kaymaz, 2005). Also the investigation of the effects of frictional losses on the performance of the Miller cycle does not appear to have been published. Therefore the objective of this study is to examine the effect of variable specific heat ratio and different frictional losses on the brake power and the brake thermal efficiency of the air standard Miller cycle.

CYCLE ANALYSIS AND PERFORMANCE ANALYSIS

The Pressure-Volume (P-V) and the Temperature-entropy (T-S) diagrams of an irreversible Miller cycle is shown in Fig. 1 where, T_1, T_2, T_3, T_4 and T_5 are the temperatures of the working fluid in state points 1, 2, 3, 4 and 5. Process 1→2 is an isentropic compression. The heat addition occurs in the constant volume process 2→3. The process 3→4 is an isentropic expansion process. Heat rejection occurs in two steps: processes 4→5 and 5→1 are constant volume and constant pressure heat rejections, respectively.

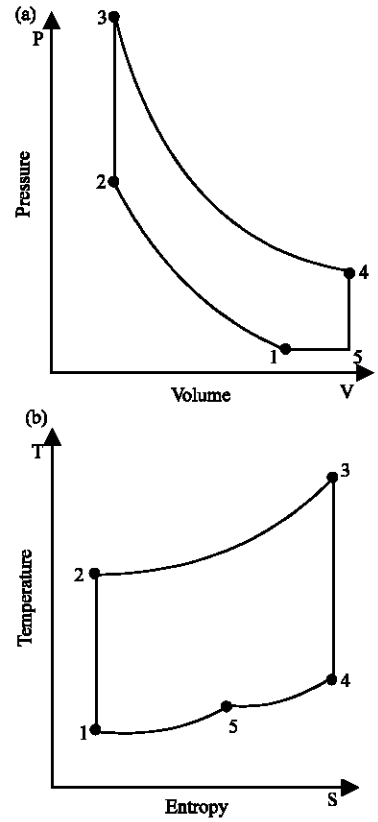

Fig. 1: a) P-V diagram; b) T-S diagram for the air standard Miller cycle

As already mentioned in the earlier study, it can be supposed that the specific heat ratio of the working fluid is a function of temperature alone and has the linear forms:

$$\gamma = \gamma_0 - k_1 T \qquad (1)$$

Where:

γ = The specific heat ratio ($\gamma = c_p/c_v$)
T = The absolute temperature
γ_0 and k_1 = Constants

Assuming that the heat engine is operated at the rate of N revolutions per second, the heat added per second in the isochoric 2→3 heat addition process may be written as:

$$\dot{Q}_{in} = NM\int_{T_2}^{T_3} c_v dT = NM\int_{T_2}^{T_3}\left(\frac{R}{\gamma_0\text{-}k_1T\text{-}1}\right)dT = \frac{NMR}{k_1}\ln\left(\frac{\gamma_0\text{-}k_1T_2\text{-}1}{\gamma_0\text{-}k_1T_3\text{-}1}\right) \quad (2)$$

Where:

M = The molar number of the working fluid

R and c_v = Molar gas constant and molar specific heat at constant volume for the working fluid, respectively

The heat rejected per second by the working fluid during processes 4→5 and 5→1 is:

$$\begin{aligned}\dot{Q}_{out} &= NM\left(\int_{T_5}^{T_4} c_v dT + \int_{T_1}^{T_5} c_p dT\right)\\ &= NM\left[\int_{T_5}^{T_4}\left(\frac{R}{\gamma_0\text{-}k_1T\text{-}1}\right)dT + \int_{T_1}^{T_5}\left(\frac{(\gamma_0\text{-}k_1T)R}{\gamma_0\text{-}k_1T\text{-}1}\right)dT\right]\\ &= NMR\left[T_5\text{-}T_1 + \frac{1}{k1}\ln\left(\frac{\gamma_0\text{-}k_1T_1\text{-}1}{\gamma_0\text{-}k_1T_4\text{-}1}\right)\right]\end{aligned} \quad (3)$$

where, c_p is the molar specific heat at constant pressure for the working fluid. According to Ge *et al.* (2005a) and Ebrahimi (2010) the equation for a reversible adiabatic process with variable specific heat ratio can be written as follows:

$$TV^{\gamma-1} = (T+dT)\,(V+dV)^{\gamma-1} \quad (4)$$

Re-arranging Eq. 1 and 4, researchers get the following equation:

$$T_i\,(\gamma_0-k_1T_j-1) = T_j\,(\gamma_0-k_1T_j-1)\,(V_j/V_i)^{\gamma 0-1} \quad (5)$$

The volume ratio of heat rejection process, ψ, the effective compression ratio, r_c^* and the compression ratio, r_c are defined as:

$$\psi = \frac{V_5}{V_1} = \frac{T_5}{T_1} \quad (6)$$

$$r_c^* = \frac{V_1}{V_2} \quad (7)$$

And:

$$r_c = \frac{V_5}{V_2} = \psi r_c^* \quad (8)$$

Therefore, the equations for processes (1→2) and (3→4) are shown, respectively by the following:

$$T_1\,(\gamma_0-k_1T_2-1)(r_c{*})^{\gamma_0-1} = T_2\,(\gamma_0-k_1T_1-1) \quad (9)$$

And:

$$T_3\,(\gamma_0-k_1T_4-1) = T_4(\gamma_0-k_1T_3-1)\,(\psi r_c{*})^{\gamma_0-1} \quad (10)$$

For an ideal Miller Cycle Model, there are no losses. However, for a real internal combustion engine cycle, the heat transfer irreversibility between the working fluid, the cylinder wall and friction like term loss are not negligible. The heat loss through the cylinder wall is assumed to be proportional to the average temperature of both the working fluid and the cylinder wall and the wall temperature is constant. The energy transferred to the working fluid during combustion is given by the following linear relation (Al-Sarkhi *et al.*, 2002; Ge *et al.*, 2005c; Ebrahimi, 2010b):

$$\dot{Q}_{in} = NM\left[A\text{-}B(T_2+T_3)\right] \quad (11)$$

where, A and B are two constants related to combustion and heat transfer which are functions of engine speed.

The mean effective losses of power due to friction in different moving parts are calculated by using the following empirical relations (Gogoi and Baruah, 2010).

Mean Effective Pressure (FMEP) lost due to friction in the piston and piston rings:

$$FMEP1 = 12.85\frac{P_{sl}}{DL}\times\frac{100\times V_p}{1000} \quad (12)$$

Where:

P_{sl} = The piston skirt length (mm)

D = The cylinder bore (m)

L = The stroke length (m)

V_p = mean piston speed (m sec^{-1})

MEP lost in bearing friction:

$$FMEP2 = 0.0564\frac{D}{L}\times\frac{N}{1000} \quad (13)$$

Where:

N = The engine RPM

MEP lost in friction in the valve gear:

$$FMEP3 = 0.226\left(30\text{-}\frac{4N}{1000}\right)\times\frac{nD_{vi}^{1.75}}{D^2L} \quad (14)$$

Where:

n = The number of intake valve/cylinder

D_{vi} = The inlet valve diameter

MEP lost in overcoming inlet and throttling losses:

$$\text{FMEP4} = \frac{\text{Pe}}{2.75} + P_{\text{imf}} \tag{15}$$

Where:
p_e = The exhaust gas back pressure (bar)
p_{imf} = The inlet manifold pressure (bar)

MEP lost in pumping:

$$\text{FMEP5} = 0.0275 \times \left(\frac{N}{1000}\right)^{1.5} \tag{16}$$

MEP lost in friction due to gas pressure behind rings:

$$\text{FMEP6} = 4.2\frac{L}{D^2}\left(P_a - P_{\text{imf}}\right) \times \left(0.0888 r_c + 0.182 r_c^{1.33-0.39V_p/100}\right) \tag{17}$$

Where:
p_a = The atmospheric pressure (bar)

MEP lost in friction due to wall tension in rings:

$$\text{FMEP7} = 10 \times \frac{0.377 L n_{pr}}{D^2} \tag{18}$$

Where:
n_{pr} = The number of piston rings

Blow by losses:

$$\text{FMEP8} = \sqrt{P_a - P_{\text{imf}}} \times \left[0.121 r_c^{0.4} - \left(0.0345 + 0.001055 r_c\right)\left(\frac{N}{100}\right)^{1.185}\right] \tag{19}$$

Finally, total mean effective pressure is:

$$\text{fmep} = \sum_{1}^{8} \text{FMEPi} \tag{20}$$

Therefore, the lost power due to friction is:

$$P_{fn} = 1500 \times \frac{f_{mep} V_d N}{\pi} \tag{21}$$

where, V_d is cylinder volume. The unit of f_{mep} is bar. The net actual brake power of the Miller cycle engine can be written as:

$$P_{b,\text{Miller}} = \dot{Q}_{in} - \dot{Q}_{out} - P_{fn} \tag{22}$$

The brake thermal efficiency of the Miller cycle engine is expressed by:

$$\eta_{b,\text{Miller}} = P_{b,\text{Miller}} / \dot{Q}_{in} \tag{23}$$

When r_c^*, ψ and T_1 are given, T_2 can be obtained from Eq. 9 then substituting Eq. 2 into Eq. 11 yields T_3 and T_4 can be found from Eq. 10; at last, T_5 can be found by Eq. 6. Substituting T_1, T_2, T_3, T_4 and T_5 into Eq. 22 and 23 yields the brake power and brake thermal efficiency. Therefore, the relations between the brake power, the brake thermal efficiency and the compression ratio can be derived.

NUMERICAL EXAMPLES AND DISCUSSION

The following constants and parameters have been used in this exercise: ψ = 1.3, T_1 = 300 K, B = 25 J mol^{-1} K^{-1}, A = 60000 J mol^{-1}, M = 1.57× 10^{-5} kmol, k_1 = 0.00008 K^{-1}, γ_0 = 1.4 and N = 1000→3000 rpm (Hatamura *et al*., 1997; Al-Sarkhi *et al*., 2002; Ge *et al*., 2005a; Ebrahimi, 2010a, b; Chen *et al*., 2010). Table 1 shows the engine configuration and the engine operating conditions.

Using the above constants and parameters range of Table 1, the brake power versus compression ratio characteristic, the brake thermal efficiency versus compression ratio characteristic and the brake power versus brake thermal efficiency characteristic can be plotted as in Fig. 2-8.

Figure 2-8 show the effects of the variable specific heat ratio of the working fluid on the performance of the cycle with heat resistance and different frictional losses parameters. From these figures, it can be found that engine RPM and inlet valve diameter play a key role on the brake power and the brake thermal efficiency. It is clearly seen that the effects of N, D_{vi} and V_p on the brake power and brake thermal efficiency are related to the compression ratio. They reflect the performance characteristics of an irreversible Miller cycle engine.

Figure 2-4 show the influence of the parameter N on the Miller cycle performance. It can be seen that the engine speed plays a significant role on the Miller cycle performance. The maximum brake power increases with increasing engine speed up to about 3000 rpm. The range of r_c in which the cycle can work normally remains constant with increase of engine speed. It can also be

Table 1: Engine design and operating design (Gogoi and Baruah, 2010)

Parameters	Values
Bore (mm (D))	87.5
Stroke (mm (L))	110
Piston skirt length (P_{sl})	80
Connecting rod length mm	230
Inlet valve diameter (mm (D_{vi}))	20-30
Exhaust valve diameter (mm)	34
Inlet manifold pressure (atm (p_{imf}))	1
Exhaust manifold pressure (atm)	1
Mean piston speed (m sec^{-1})	5-15
Number of piston rings (n_{pr})	3
Exhaust gas back pressure (atm (p_e))	1.1

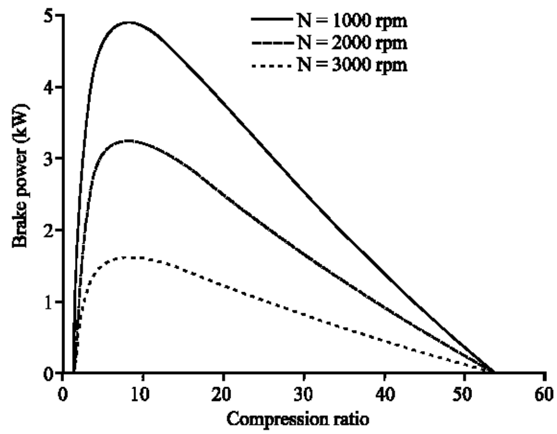

Fig. 2: Effect of engine RPM on $P_{b,\ Miller}$-rc characteristic for D_{vi} = 25 mm, P_{sl} = 80 mm, V_p = 10 m sec^{-1}

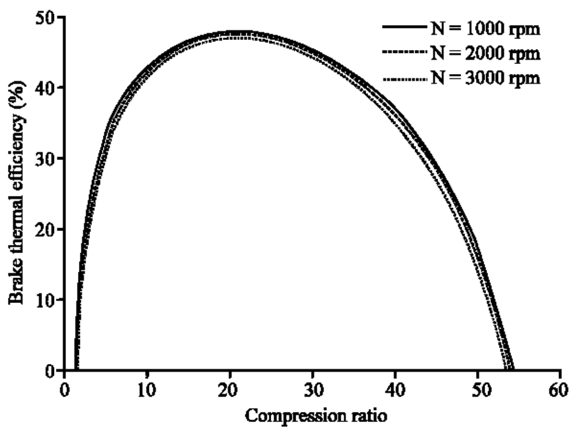

Fig. 3: Effect of engine RPM on $\eta_{b,Miller}$−rc characteristic for D_{vi} = 24 mm, P_{sl} = 80 mm, V_p = 10 m sec^{-1}

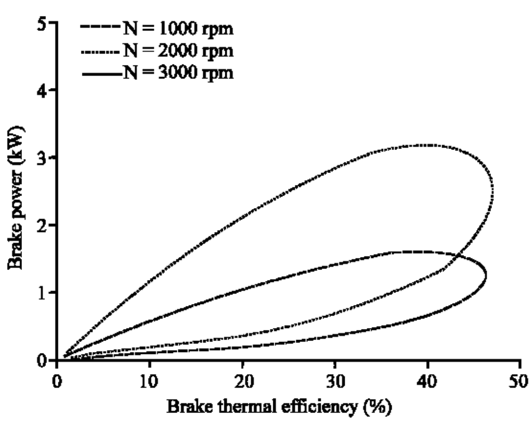

Fig. 4: Effect of engine RPM on $P_{b,Miller}$−$\eta_{b,Miller}$ characteristic for D_{vi} = 24 mm, P_{sl} = 80 mm, V_p = 10 m sec^{-1}

concluded that, e.g., at the compression ratio of 8, the brake power increases with the increase of engine speed. The maximum brake thermal efficiency of Miller cycle decreases smoothly with increase of N: When N increases from 1000-3000 rpm, the maximum brake thermal efficiency of Miller cycle will decreases from 47.46-46.42% (about 2.24%). Also the maximum brake power of Miller cycle increases sharply with increase of N: When N increases from 1000-3000 rpm, the maximum brake power of Miller cycle will increases from 1.60-4.89 KW (about 205.63%). The influence of the engine speed on the brake power versus brake thermal efficiency is displayed in Fig. 4. As can be seen from this figure, the brake power versus brake thermal efficiency is a loop shaped one. It can be seen that the brake power at maximum brake thermal efficiency improves with increasing engine speed from 1000 to around 3000 rpm.

Figure 5 and 6 show the effects of inlet valve diameter (D_{vi}) on the performance of the Miller cycle. Figure 5

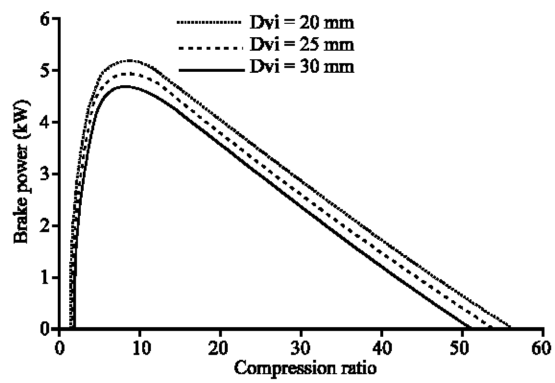

Fig. 5: Effect of inlet valve diameter on $P_{b,Miller}$−rc characteristic for V_p = 10 m sec^{-1}, P_{sl} = 80 mm, N = 3000 rpm

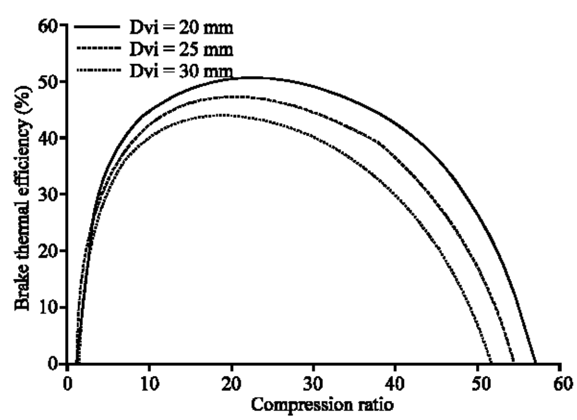

Fig. 6: Effect of inlet valve diameter on $\eta_{b,Miller}$−rc characteristic for V_p = 10 m sec^{-1}, P_{sl} = 80 mm, N = 3000 rpm

and 6 shows that the points of maximum brake thermal efficiency and brake power of cycle will decreases with an increase of D_{vi}. This happens due to the increase of the friction losses (Eq. 14). The maximum brake thermal efficiency and the maximum brake power of Miller cycle decreases sharply with increase of D_{vi}: When D_{vi} increases from 20-30 mm (50%), the maximum brake thermal efficiency of Miller cycle will decrease from 50.5-44.12% (about 12.63%) and maximum brake power of Miller cycle will decrease from 5.13-4.62 KW (about 9.94%).

Figure 7 and 8 show the effects of mean piston speed (V_p) on the performance of the Miller cycle. Figure 7 and 8 shows that the points of maximum brake thermal efficiency and brake power of cycle will decreases with an increase of V_p. This happens due to the increase of the efficiency and the maximum brake power of Miller cycle

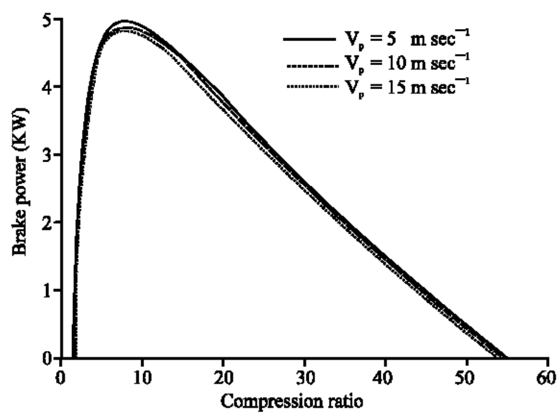

Fig. 7: Effect of mean piston speed on $P_{b,\ Miller-rc}$ characteristic for D_{vi} = 25 mm, P_{sl} = 80 mm, N = 3000 rpm

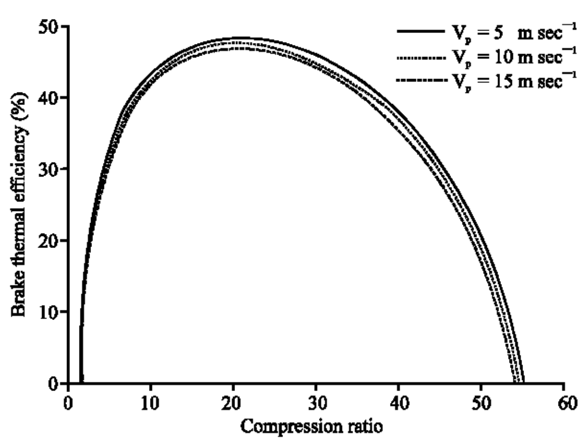

Fig. 8: Effect of mean piston speed on $\eta_{b,Miller-rc}$ characteristic for D_{vi} = 25 mm, P_{sl} = 80 mm, N = 3000 rpm

friction losses (Eq. 12 and 17). The maximum brake thermal decreases smoothly with increase of V_p: When V_p increases from 5-30 m sec^{-1} (200%), the maximum brake thermal efficiency of Miller cycle will decrease from 48.09-46.83% (about 2.62%) and maximum brake power of Miller cycle will decrease from 4.95-4.84 KW (about 2.22%).

CONCLUSION

This study is aimed at investigating the effects of different frictional losses and variable specific heat ratio of the working fluid on the Miller cycle's performance. By using finite time thermodynamics theory, the characteristic curves of the brake power versus compression ratio, brake thermal efficiency versus compression ratio and the brake power versus brake thermal efficiency are obtained. In the model, the linear relation between the specific heats ratio of working fluid and its temperature, the frictional loss computed according to the empirical correlation and heat transfer loss are considered. The general conclusions drawn from the results of this study are as follows:

- The point of maximum brake power of Miller cycle is at compression ratio of 8
- The engine speed plays a significant role on the Miller cycle performance
- The points of maximum brake power and brake thermal efficiency of Miller cycle will decreases with an increase of D_{vi}, V_p
- The points of maximum brake power of Miller cycle will increases with an increase of N
- The results of this investigation are of importance when considering the designs of Miller engines

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