113 Development Mean Value Engine Model for SI Engine Simulation

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Simple and accurate engine models are required for control strategy development applications. The Mean Value Engine Model is one of the models that have been developed for these purposes. This model predicts engine behavior with good accuracy. In this paper the model equations are illustrated and a method for calibrating the model parameters with minimum experimental tests is proposed. The accuracy of the developed model is investigated through experimental works carried out on the engine of a locally manufactured vehicle called Samand.

Keywords: **Engine Modeling, Mean Value Engine Model, Intake Manifold Dynamics**

1. INTRODUCTION

Nowadays various engine models are used in control strategy development for engine and power train systems. For this purpose the models with good accuracy and low computational demands are suitable. The Mean Value Engine Model (MVEM) is developed with these criteria [1-4]. This model predicts engine behavior with acceptable accuracy, while governing equations are simple and easily calibrated. The model has three sub models including: Intake manifold, Fuel delivery dynamics and Rotational dynamics [1]. Because these sub models are used in practical applications, various investigations have been done for increasing the accuracy of the intake manifold and fuel delivery sub models [5-7].

For the engine model calibration, various experimental tests are needed. Decreasing the number of calibration tests for the intake manifold and rotational dynamics sub models are the main goals of this paper. First the governing equations at each sub model are illustrated and then a method is proposed for the intake manifold parameters calibration. Table (1) shows the test engine specifications.

Table1: Test engine specifications

No. of Cylinders	4
Displacement volume(cc)	1761
Compression ratio(r_c *)*	9.3
B×S(mm)	83×81.5

2. INTAKE MANIFOLD SUB MODEL

In this sub model the cylinder air mass flow and manifold pressure are calculated. By implementing a control volume for the intake manifold and using laws of conservation of mass and energy, the following equations are derived for the intake manifold pressure and temperature [6,8,9]:

$$\dot{P}_m = \frac{kRT_{amb}}{V_m}\left(\dot{m}_{at} - \frac{T_m}{T_{amb}}\dot{m}_{ac}\right)$$
$$\dot{T}_m = \frac{kRT_{amb}T_m}{V_m P_m}\times \left(\dot{m}_{at}(1-\frac{1}{k}\frac{T_m}{T_{amb}}) - \frac{T_m}{T_{amb}}\dot{m}_{ac}(1-\frac{1}{k})\right) \quad (1)$$

$\dot{m}_{at}$ and $\dot{m}_{ac}$ are the throttle and cylinder air mass flow respectively. If the variation of the manifold temperature is eliminated $(\dot{T}_m = 0)$, the manifold pressure equation is simplified. But in transient conditions the accuracy of the model is decreased [6,8,9]. The throttle air mass flow is calculated from the following equation [5,9]:

$$\dot{m}_{at} = c_d \frac{P_{amb}}{\sqrt{T_{amb}}}\beta_1(\theta)\beta_2(P_r) \quad (2)$$

c_d is discharge coefficient and depends on the throttle angle (θ) and the manifold pressure. For more simplification c_d could be taken constant $(c_d = 0.552)$ [8,9]. $\beta_2(P_r)$ and $\beta_1(\theta)$ show the manifold pressure and the throttle angle effects on the air mass flow as follows:

$$\beta_2(P_r) = \begin{cases} \frac{1}{0.74}\sqrt{P_r^{0.4404} - P_r^{2.3086}} & P_r \geq 0.4125 \\ 1 & P_r < 0.4125 \end{cases} \quad (3)$$

$$P_r = \frac{P_m}{P_{amb}} \tag{4}$$

$$\beta_1(\theta) = -\frac{d \times D}{2} \times A + \frac{d \times D}{2} \times \alpha + \frac{D^2}{2}\sin^{-1}(A) - \frac{D^2 \cos(\theta + 5^\circ)}{1.992} \times \sin^{-1}(\alpha) \quad for\, \theta \le \theta_{max} \tag{5}$$

$$\beta_1(\theta) = \beta_1(\theta_{max}) \qquad for\, \theta > \theta_{max} \tag{6}$$

$$A = \left[1 - \left(\frac{d}{D}\right)^2\right]^{\frac{1}{2}} \tag{7}$$

$$\alpha = \left[1 - \left(\frac{0.996 \times d}{D\cos(\theta + 5^\circ)}\right)^2\right]^{\frac{1}{2}} \tag{8}$$

$$\theta_{max} = \cos^{-1}\left[0.996 \times \frac{d}{D}\right] - 5^\circ \tag{9}$$

D and d are the bore diameter and the throttle plate rode diameter in millimeter respectively. The cylinder air mass flow equation is [5,7,9]:

$$\dot{m}_{ac}(n, P_m) = \frac{V_d \times n}{2RT_m}\left(s(n) \times P_m - y(n)\right) \tag{10}$$

Parameters s and y show engine breathing characteristics and are functions of the engine speed. These parameters are obtained from experimental data. Table 2 shows the average value of s and y for different engines. The value of these parameters are close to each other for various engines, therefore we can use these values for approximate modeling of any conventional SI engine. In section 5 of this paper a method is proposed for the calculation of s and y.

Table2: Average values of s, y for different engines. *From [5].

Engine	s	y(bar)
Samand 1.8L	0.933	0.0631
Paykan 1.6L	0.907	0.0909
Ford* 1.1L	0.95	0.087
Ford* 4.9L	0.953	0.0857

3. FUEL DELIVERY DYNAMICS SUB MODEL

Some of the injected fuel is deposited on the intake port walls and evaporates gradually. Therefore in transient conditions the injected fuel is not equal to the fuel that entered the cylinder. For considering this phenomenon the $x - \tau$ model could be used [1,10]:

$$\dot{m}_{fc} = (1 - x)\dot{m}_{fi} + \dot{m}_{ff}$$
$$\dot{m}_{ff} = \frac{1}{\tau}\left(x\dot{m}_{fi} - \dot{m}_{ff}\right) \tag{11}$$

$\dot{m}_{fi}, \dot{m}_{fc}, \dot{m}_{ff}$ are the injected fuel flow, cylinder fuel flow and evaporated fuel flow from the port walls that entered the cylinder respectively. x is the fraction of the injected fuel that deposited on the port walls and τ is the evaporation time constant. x and τ are dependent on coolant temperature, engine speed and manifold pressure[10]. If these parameters are not available, the following assumption could be considered:

$$\dot{m}_{fc} = \dot{m}_{fi} \tag{12}$$

4. ROTATIONAL DYNAMICS SUB MODEL

For calculation engine speed, the brake torque equation should be known. For simulation purposes, we need a general equation for brake torque calculation that could be calibrated with minimum number of tests. The brake torque could be obtained from following equation [11]:

$$T_b = T_i - T_{f/p} \tag{13}$$

T_i and $T_{f/p}$ are the indicated and friction/pumping torques respectively. The indicated torque is calculated as follows [8,9,12,13]:

$$T_i = 8027.5\frac{\dot{m}_{ac}}{n} \times \eta_{fc} \times SI \times AFI \tag{14}$$

η_{fc} is the fuel conversion efficiency[12,14]:

$$100\eta_{fc} = -7.44 - 70.13r_c^{-0.372} - 0.217r_c + 94.27\left(\frac{B}{L}\right)^{-0.02} - 14.31V_d^{-0.083} - 3.72\left(\frac{P_m}{101.3}\right)^{-0.258} - 9.77n^{-0.088} \tag{15}$$

AFI and SI show the air fuel ratio and spark advance effects on the indicated torque respectively. The friction/pumping torque could be obtained from equations that are developed by Sandovel and Heywood [15](see the appendix). For the test engine $T_{f/p}$ is [8,9]:

$$T_{f/p} = \frac{V_d}{12560} \times tf_{mep} \tag{16}$$

$$tf_{mep} = 50.26 + 0.0197n + 1.087E - 6n^2 + \frac{17689}{n} + 0.0193\sqrt{n} + 8.23\frac{P_m}{P_{am}} + 2.632\frac{P_m}{P_{am}} \times 9.3^{(1.33 - 0.00152n)} + 1.31E - 6\left(\frac{P_m n}{P_{am}}\right)^2 \tag{17}$$

The $T_{f/p}$ equations have three parameters that should be obtained from experimental tests. There are some suggestions for these parameters in [15]. These values could be used with few modifications. Fig. 1 shows the brake torque for the test engine that is obtained with proposed equations and experimental tests. The accuracy of calculated values is good in most of the engine operating conditions. Just in low manifold pressure the error is slightly increased. The inaccurate estimation of pumping losses in these conditions may be the cause of this error.

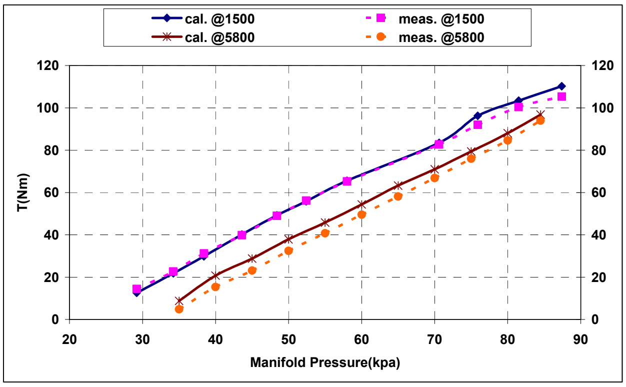

Fig. 1: Calculated and measured brake torque in two different engine speeds vs. manifold pressure

5. A METHOD FOR S AND Y CALCULATION

In this chapter a method for estimating s and y are proposed. For this purpose, we need engine brake torque vs. engine speed plot. By substituting $\dot{m}_{ac}$ from equation (10) in the equation (14) and assuming AFI and SI are unit, the following equation is obtained:

$$T_i = C \times M \times (sP_m - y)$$
$$M = 83.91 \frac{V_d}{T_m} \times \eta_{fc} \qquad (18)$$

C coefficient is obtained experimentally from two different engines and is:

$$C = \begin{cases} 1 & n < 2500rpm \\ 1.04 & n \geq 2500 \end{cases} \qquad (19)$$

The indicated torque is derived form the following equation:

$$T_i = T_b + T_{f/p} \qquad (20)$$

T_b is the maximum brake torque in each engine speed that is available for the most engines. $T_{f/p}$ is obtained from the equations that are available in the appendix .

For calculating s parameter P_m and y should be known. $P_m(bar)$ at full throttle can be estimated as:

$$P_m = P_{amb} - 7 \times 10^{-6} n \qquad (21)$$

By ignoring variation of $y(bar)$ with engine speed[5,9]:

$$y = 0.68 \left(\frac{g_e P_{amb}}{(r_c - 1)} \right) \qquad (22)$$

r_c is the compression ratio and $g_e \approx 1$. Now the s parameter for any engine speed is calculated as:

$$s = \frac{T_i / CM + y}{P_m} \qquad (23)$$

Fig. 2 shows the variation of s parameter vs. engine speed for the test engine. The error of calculated values is within 5%. If we have two plots for brake torque at two different conditions, then we can calculate s and y for each engine speed more accurately.

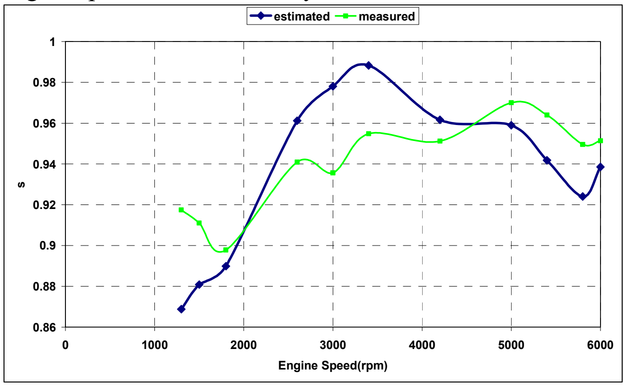

Fig. 2: Variation of s parameter for test engine

5. MODEL VALIDATION

In validation tests the 1.8L Samand engine was used. The test was done in two modes: constant engine speed and constant load torque. To investigate the accuracy of the model in all operating conditions, the validation tests were done at various engine speeds. Figs. 3 and 4 show the simulated and test results in two conditions. Fig. 3 relates to constant engine speed test and fig. 4 shows obtained results at constant load torque. Figs. 3.1 and 4.1 show the input throttle angle. The rise/fall time for the throttle angle variations is one second. The manifold pressures are shown in figs. 3.2 and 4.2. The maximum error of the calculated manifold pressure is 11%. Fig. 3.3 shows the engine brake torque at constant engine speed. The calculated brake torque is near to measured values in the most of regions. Fig. 4.3 shows the engine speed variations at constant load torque. The calculated result has good agreement with experimental data, where the maximum error is 5%.

Fig. 5 shows the engine behavior in idle conditions. The input throttle angle and manifold pressure are shown in fig. 5.1and Fig. 5.2 shows the engine speed. In the region that throttle angle is zero the calculated engine speed is higher than experimental values. Because pumping losses equations are not accurate and should be modified. This error caused the calculated manifold pressure to be lower than the measured data at zero throttle angles. If the engine speed is considered as an input, the manifold pressure can be calculated more accurately.

6. CONCLUSION

In this paper an engine model was developed that needs few experimental tests for calibration. Even it could be calibrated just with maximum brake torque plot. The capability of the model in predicting engine behavior was shown with various experimental tests. The error of the developed model is about 15%. This model can be used for various applications such as: control strategy development, obtaining fuel consumption in driving cycles, hybrid vehicle simulation and so on.

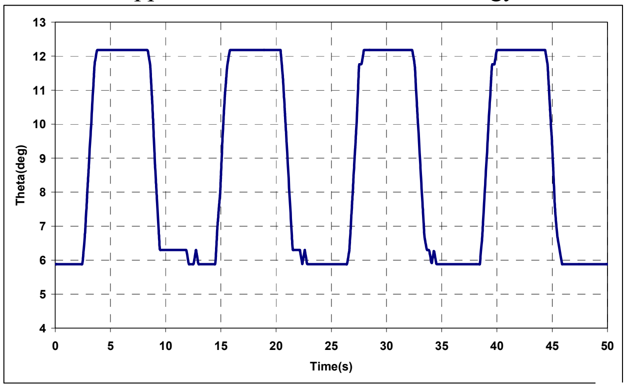

Fig.3.1: Input throttle angle at constant engine speed $n \approx 2500$ rpm

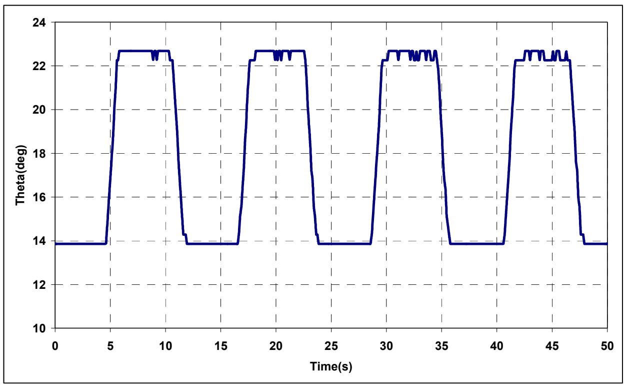

Fig. 4.1: Input throttle angle at constant load torque $T_L \approx 30$ Nm

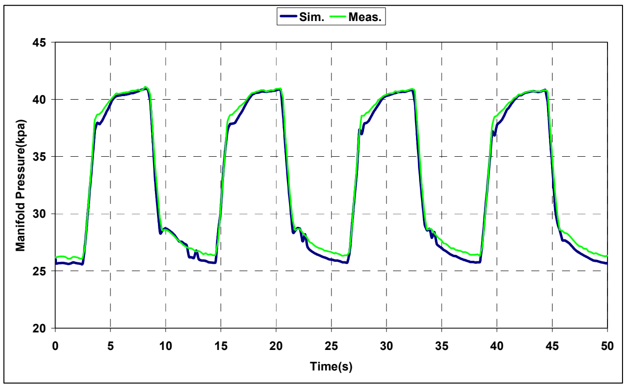

Fig.3.2: Manifold pressure variations at constant engine speed

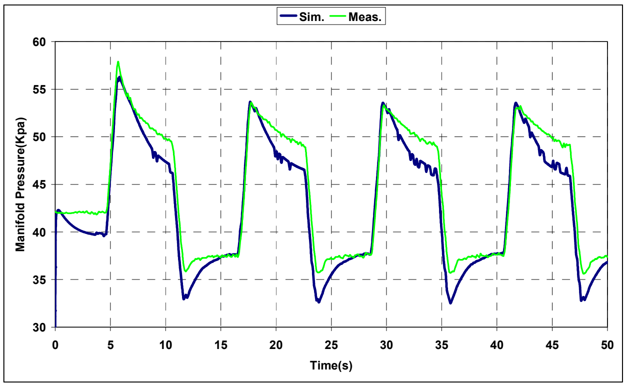

Fig.4.2: Manifold pressure variations at constant load torque

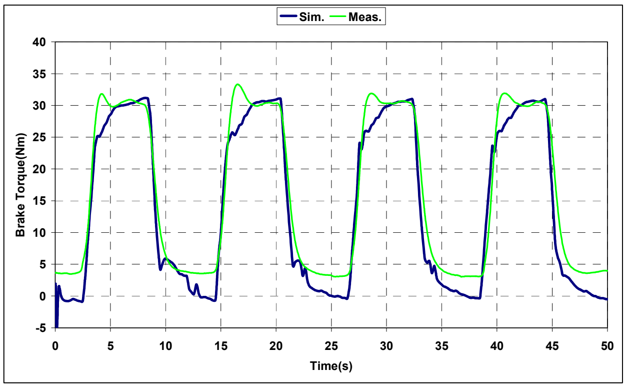

Fig.3.3: Engine brake torque at constant engine speed

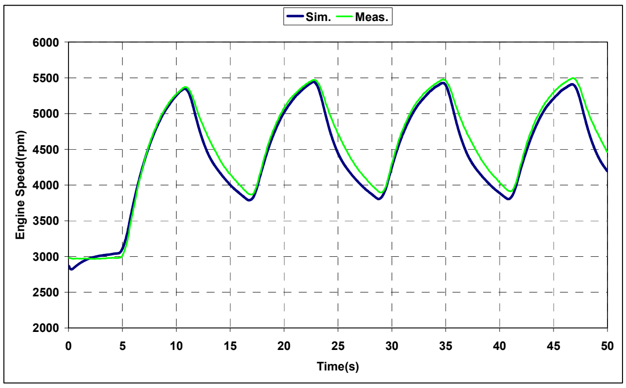

Fig.4.3: Engine speed variations at constant load torque

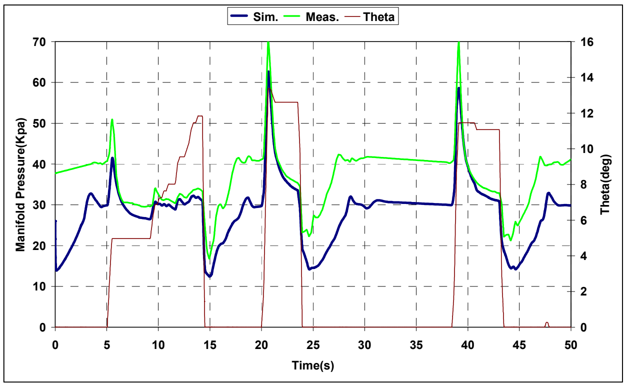

Fig.5.1: Manifold pressure and input throttle angle at idle conditions

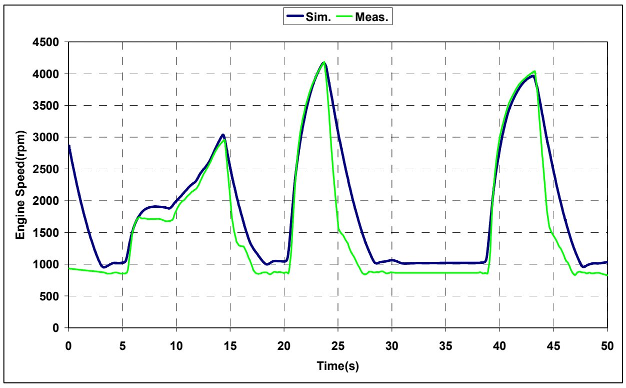

Fig.5.2: Engine Speed responses at idle conditions

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8. APPENDIX

Sandovel and Heywood proposed the following friction model for SI engines [15]:

$$T_{f/p} = \frac{V_d}{12560} \times tf_{mep} \tag{A.1}$$

V_d is cylinder displacement volume (cm^3) and tf_{mep} is total friction mean effective pressure (kpa), where:

$$tf_{mep} = f_{mep} + p_{mep} + a_{mep} \tag{A.2}$$

f_{mep} , p_{mep} and a_{mep} are friction, pumping and auxiliary losses respectively.

To consider oil viscosity variation with temperature, the following coefficient is defined:

$$\mu_{sc} = \sqrt{\frac{\upsilon(T)}{10.6}} \tag{A.3}$$

$$\upsilon(T) = k \times \mu_{\infty/o} \times \exp\left(\frac{\theta_1}{\theta_2 + T}\right) \tag{A.4}$$

T is oil temperature($^\circ C$) and other constants are given in table A.1 for two standard oil grades[15]:

Table A.1: viscosity data for two standard oil grades [15]

Oil Grade	$k(cst)$	$\mu_{\infty/o}$	θ_1	θ_2
10W30	0.1403	0.76	869.72	104.4
20W50	0.0639	0.84	1255.46	117.7

Friction losses

Friction losses consist of three terms: crank shaft (cf_{mep}), reciprocating (rf_{mep}) and valve train friction (vf_{mep}):

$$cf_{mep} = 1.22 \times 10^5 \left(\frac{D_b}{B^2 S n_c}\right) + 3.03 \times 10^{-4} \mu_{sc} \left(\frac{N D_b^3 L_b n_b}{B^2 S n_c}\right) + 1.35 \times 10^{-10} \left(\frac{D_b^2 N^2 n_b}{n_c}\right) \tag{A.5}$$

$D_b(mm), L_b(mm)$ and n_b are diameter, length and number of crank shaft bearing respectively. $B(mm)$, $S(mm)$ and n_c are bore, stroke and number of cylinders respectively. $N(rpm)$ is the engine speed.

Reciprocating losses consist of two terms: friction without gas pressure loading ($r_r f_{mep}$) and the increase in piston ring friction caused by gas pressure loading (gas_{mep}):

$$rf_{mep} = r_r f_{mep} + gas_{mep} \tag{A.6}$$

Where:

$$r_r f_{mep} = 294\mu_{sc}\left(\frac{S_p}{B}\right) + 4.06\times10^4 t_f\left(1+\frac{500}{N}\right)\left(\frac{1}{B^2}\right) + 3.03\times10^{-4}\mu_{sc}\left(\frac{ND_b^3L_bn_b}{B^2Sn_c}\right) \quad (A.7)$$

And:

$$gas_{mep} = 6.89\frac{P_m}{P_{amb}}\left(0.088\mu_{sc}r_c + 0.182t_{fp}\times r_c^{(1.33-0.056S_p)}\right) \quad (A.8)$$

$S_p\,(m/s)$ is mean piston speed. $D_b\,(mm), L_b\,(mm)$ and n_b are diameter, length and number of connecting rod bearings respectively. t_f and t_{fp} account for the effect of cylinder - piston ring improvement in friction reduction.

Valvetrain friction is:

$$vf_{mep} = 244\mu_{sc}\left(\frac{Nn_b}{B^2Sn_c}\right) + C_{ff}\left(1+\frac{500}{N}\right)\frac{n_v}{Sn_c} + C_{rf}\frac{Nn_v}{Sn_c} + C_{oh}\mu_{sc}\left(\frac{L_v^{1.5}N^{0.5}n_v}{BSn_c}\right) + C_{om}\left(1+\frac{500}{N}\right)\frac{L_vn_v}{Sn_c} + 4.12 \quad (A.9)$$

n_b, n_v and $L_v\,(mm)$ are the number of camshaft bearings, number of valves and maximum valve lift. The constants for the valvetrain terms ($C_{ff}, C_{rf}, C_{oh}, C_{om}$) depend on the valvetrain configuration being modeled [15]. Table A.2 shows these constants for Samand with SOHC direct acting configuration and Paykan with SOHC flat finger follower configuration. By adjusting the constants in the valvetrain equation, the friction losses and consequently the brake torque can be predicted appropriately

Table A.2: constants of valve train terms for two various configurations, parenthesis value are taken from [8,15].

	C_{ff}	C_{rf}	C_{oh}	C_{om}
Samand SOHC direct acting	600 (200)	0	0.5 (0.5)	50 (11)
Paykan SOHC flat finger follower	700 (600)	0	0.2 (0.2)	60 (43)

Pumping losses are the sum of intake and exhaust terms:

$$p_{mep} = \text{int}_{mep} + exh_{mep} \quad (A.10)$$

Where:

$$\text{int}_{mep} = (p_{amb} - p_m) + 3.0\times10^{-3}\left(\frac{p_m}{p_{amb}}\right)^2\frac{S_p^2}{n_v^2r_{in}^4} \quad (A.11)$$

And :

$$exh_{mep} = 0.178\left(\frac{p_mS_p}{p_{amb}}\right)^2 + 3.0\times10^{-3}\left(\frac{p_m}{p_{amb}}\right)^2\frac{S_p^2}{n_v^2r_{ex}^4} \quad (A.12)$$

$$r_{in} = \frac{D_{in}}{B} \quad (A.13)$$

$$r_{ex} = \frac{D_{ex}}{B} \quad (A.14)$$

D_{in}, D_{ex} are the inlet and exhaust valves diameters respectively.

Auxiliary losses indicate the oil pump, water pump and non –charging alternator friction:

$$a_{mep} = 8.32 + 1.86\times10^{-3}N + 7.45\times10^{-7}N^2 \quad (A.15)$$