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A new simple friction model for S. I. engine

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ABSTRACT

Internal combustion engine modeling is nowadays a widely employed tool for modern engine development. Zero and mono dimensional models of the intake and exhaust systems, combined with multi-zone combustion models, proved to be reliable enough for the accurate evaluation of in-cylinder pressure, which in turn allow the estimation of the engine performance in terms of indicated mean effective pressure (IMEP). In order to evaluate the net engine output, both the torque dissipation due to friction and the energy drawn by accessories must be taken into consideration, hence a model for the friction mean effective pressure (FMEP) evaluation is needed. One of the most used models accounts for engine speed dependent friction by means of a quadratic law, while the effect of engine load (i.e. the thrust that the gas exercises on the piston surface) is considered by means of a linear dependence from the maximum in-cylinder pressure: hence the model requires the calibration of four constants by means of experimental data. The author, on the basis of data acquired during an extensive experimental campaign carried out on the engine test bed, found this model to give an unsatisfying prediction, above all for retarded pressure cycles (i.e. with peak pressure positions higher than 20 crank angle degrees after top dead centre): hence, by means of analysis performed using these experimental data, the author arrived at a new formulation of the friction model, which substantially take into account the effect of engine load by means of the Location of Pressure Peak (LPP). The new model, once calibrated, proved to be effectively more accurate in the prediction of the FMEP than the Chen-Flynn model.

INTRODUCTION

Engine modeling is nowadays one of the most employed tools for internal combustion engines development. One of the most common output of an internal combustion engine model is the in-cylinder pressure, which allow the evaluation of the indicated mean effective pressure (IMEP). In order to obtain the brake mean effective pressure (BMEP), which is the real engine output, a sub-model for the friction mean effective pressure (FMEP) evaluation must be employed. Different approach to this problem can be followed, as proposed in literature: complex models, such as those from [1, 2, 3], estimate the instantaneous torque losses taking into account the different contribution to the total energy dissipated by friction or drawn from accessories (valve train, pumps, etc..); in these kinds of models, the friction losses at the piston-wall interface are evaluated separately from the losses at the bearings, and even the different kind of lubrication that may occur are considered (boundary, mixed or hydrodynamic lubrication). When calibrated, these models succeed in giving a precise prediction of the FMEP (obtained integrating the total torque losses), but this normally requires the estimation of several constants, possible only by means of accurate and detailed experimental data on instantaneous in-cylinder pressure and crankshaft speed.

Simpler models, instead, aim to estimate the overall FMEP, making use of few global variable, typically one related to the engine load and the other related to the engine speed, in order to separately account both the energy dissipated by friction due to gas thrust and the energy losses influenced by the speed (e.g. those related to inertia forces). In this second category, one of the most encountered in literature and employed in commercial software is known as the Chen & Flynn model [4], according to which the FMEP depends on in-cylinder maximum pressure and engine speed by means of the following law:

$$FMEP = A + B \cdot P_{\max} + C \cdot n + D \cdot n^2 \tag{1}$$

As shown, this model accounts for the engine speed effect by means of a quadratic law (through the constants *C* and *D*), while the load effect is represented by the maximum in-cylinder pressure through the constant *B*; the constant *A* instead accounts for the energy drawn by accessories and all the other invariable factors.

MAIN SECTION

During the set-up phase of an engine model for the prediction of engine performances, the author experienced the necessity to use an FMEP model. Lacking of detailed data on instantaneous speed and acceleration, the author decided to follow the common approach of the Chen-Flynn model, whose four constants *A*, *B*, *C* and *D* were determined by means of data fitting performed on experimental values obtained at the engine test bed. The in-cylinder pressure of a four cylinder series production spark ignition engine (whose characteristics are resumed in Table 1) fuelled with Compressed Natural Gas (CNG) was sampled under different operative conditions of engine speed, Manifold Absolute Pressure (MAP) and spark timing. The IMEP was estimated on the basis of the mean pressure cycle, evaluated over 50 consecutive pressure cycles sampled for each operative condition.

Displacement [cc]	1242
Bore [mm]	70.80
Stroke [mm]	78.86
Compression ratio ρ	9.8
Rod to crank ratio μ	3.27

Table 1 Engine characteristics

The in-cylinder pressure, together with all the other parameters (engine torque, MAP, engine speed) were acquired by means of a high speed National Instruments DAQ Board PCI-6133 using as trigger and scan clock the pulses generated by a 360ppr incremental encoder connected to the engine crankshaft. The BMEP was deduced by the mean engine torque measured in the same 50 cycles, spark timing and injection time were controlled by means of an Electronic Control Unit (ECU) from WALBRO TDD, while the air-fuel ratio was measured by the use of the UEGO sensor system AFR Recorder 2400 from ECM. A crucial aspect for the correct measurement of the IMEP is the precise determination of the TDC position [5]: as is known, in fact, a 1 degree error (which can be introduced setting the TDC at the peak pressure position of a motored pressure cycle) can cause up to a 10% error in the IMEP estimation, which is an inadmissible error for the FMEP evaluation. In the experimental campaign carried out the TDC position was determined by means of a Kistler capacitive sensor 2629B, whose precision is of 0.1 Crank Angle Degrees (CAD).

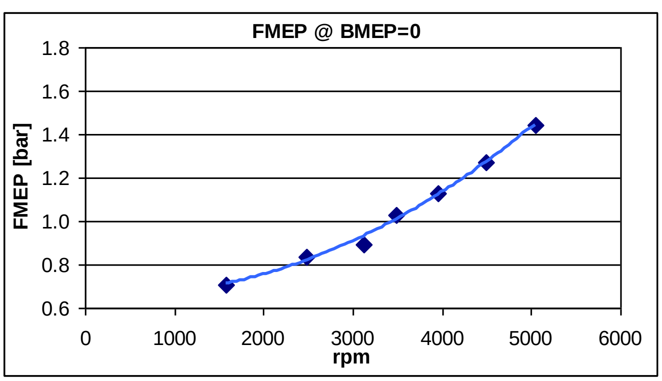

Fig. 1 FMEP as function of engine speed

In order to determine the engine speed related constants of the Chen-Flynn model, *C* and *D*, a first series of test have been carried out, running the engine without load (i.e. whit BMEP=0) at different speed (from 1500 to 5000 rpm). In this way the gas thrust effect is the minimum, and the FMEP depends mainly on the engine speed: as shown in Fig. 1, a good quadratic law was obtained.
A second series of test were performed to determine the entity of the constant *B*, running the engine on many different operative conditions of speed and load, as resumed in Table 2.

MAP [kPa]	50, 75, 100
Engine speed [rpm]	1500 to 5000 (steps of 500)
Spark advance [CAD with respect to MBT condition]	–10, –5, 0, +5, +10
Fuel	CNG
Air to Fuel ratio	Stoichiometric

Table 2 Operative condition tested

Hence, using an optimization procedure, the constant *A* and *B* were fixed minimizing the maximum percentage error of the Chen-Flynn model (see equation (1)) with respect to the experimental data. The results obtained, as can be seen in Fig. 2 and Fig. 3, were surprisingly inferior to what expected. It was found a bad matching between the model prediction and the experimental data, above all for the higher FMEP values. As resumed in Table 3, the maximum error could not be lower than 35%, and the maximum difference between model prediction and real FMEP reached 0.67 bar.

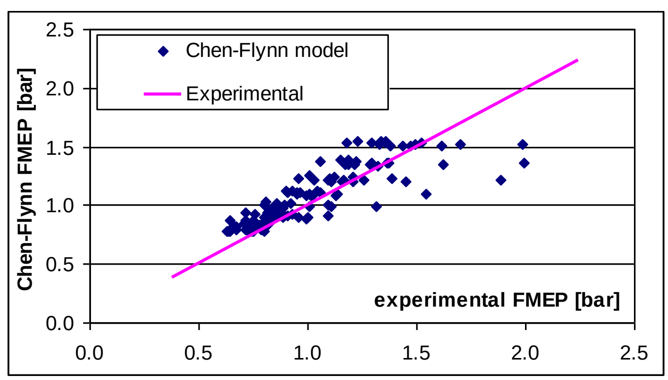

Fig. 2 Comparison between the Chen-Flynn model prediction and the experimental data

Model	Chen-Flynn
mean % error	13%
max % error	35%
max error [bar]	0.668

Table 3 Overall results of the Chen-Flynn model

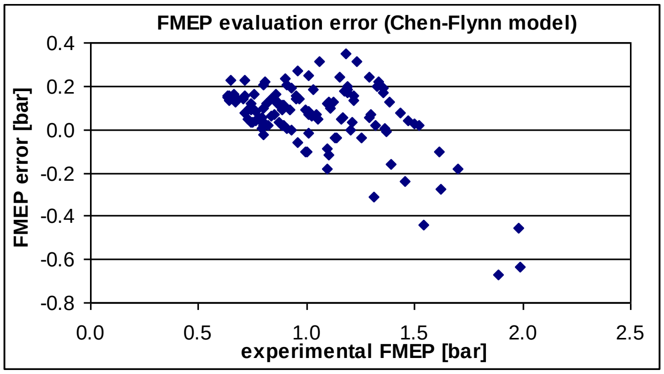

Fig. 3 FMEP evaluation error of the Chen-Flynn model

A further attempt has been made trying to use up to a 3rd order polynomial to account for the load factor, thus introducing the P_{max}^2 and P_{max}^3 terms; equation (1) then becomes:

$$FMEP = A + B \cdot P_{\max} + C \cdot P_{\max}^{2} + D \cdot P_{\max}^{3} + E \cdot n + F \cdot n^{2} \qquad (2)$$

The result of the fitting however still showed an unsatisfying prediction: as resumed by Table 4, equation (2) is unable to furnish a better FMEP evaluation than the original Chen-Flynn model of equation (1); the attempt to obtain a better correlation with the load factor P_{max} by introducing the higher order terms P_{max}^2 and P_{max}^3 revealed hence to be useless.

Model	Chen-Flynn with P_{max}^2 and P_{max}^3
mean % error	13%
max % error	33%
max error [bar]	0.619

Table 4 Overall results of the equation (2) model

The author also tried to adopt, in place of the maximum in-cylinder pressure, other load-linked variables, such as the manifold pressure MAP or the IMEP: both attempts revealed to be vane.

A deeper analysis of the experimental data was then carried out, aiming to a better understanding of the variables that really influence the FMEP. Fig. 4 shows the experimental FMEP values obtained for three different engine speed (1500, 3000 and 4500 rpm), three different loads (MAP=50, 75 and 100kPa) and with varying spark advance around the Maximum Brake Torque (MBT) value. As a first observation, the dependence from engine speed is confirmed, since for every MAP value, higher engine speed causes higher FMEP values.

The second important observation instead concerns the "load factor": according to the graph in Fig. 4, in fact, the FMEP increases when spark advance decreases, and this dependence seems to be emphasized by engine speed. Effectively, the graph in Fig. 5, realized using the same data used for Fig. 4, shows a clear connection between spark timing and FMEP for each of the engine speed tested. The same figure also points out the lack of any dependence from the manifold pressure, as Fig. 4 might lead to believe.

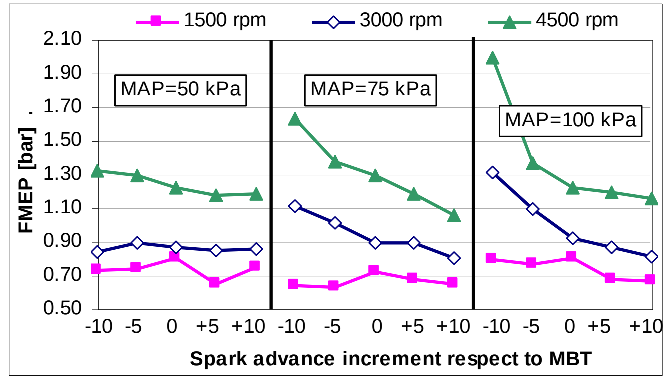

Fig. 4 Experimental FMEP values for different MAP, engine speed and spark timing

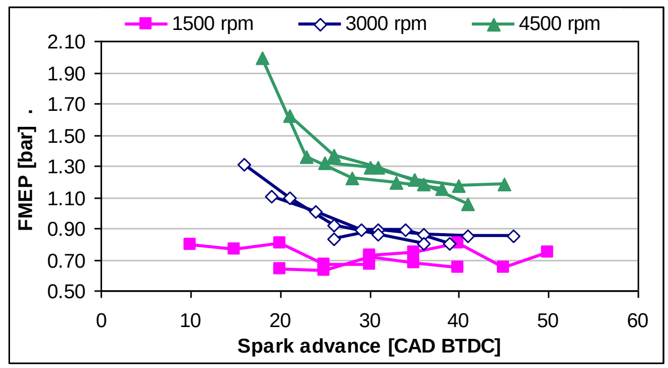

Fig. 5 Experimental FMEP vs. spark advance (same operative conditions of Fig. 4)

Since retarding the spark timing usually decreases the maximum in-cylinder pressure, as shown for example in Fig. 6, it can be further on concluded that, for the engine tested, the FMEP decreases when P_{max} increases; this fact is confirmed by Fig. 7 which reports the same FMEP values of Fig. 4 plotted against the maximum in-cylinder pressure. The same data in Fig. 7 shows that, for each engine speed, it is not possible to find a univocal relation between FMEP and P_{max}: this implies that, for the engine tested, the maximum in-cylinder pressure does not represent a valid "load variable" to be used in a simple FMEP model.

Spark advance however is only a control parameter and not a thermodynamic variable, hence it cannot be assumed as independent variable for the friction model; it acts on the phase of the combustion with respect to the piston motion, and hence it influences the phase of the pressure cycle, which can be easily represented by the Location of the Pressure Peak (LPP): this parameter, usually employed as combustion phase indicator, also expresses the crank position at which the gas thrust is maximum. As shown by the graph of Fig. 8, traced using the same experimental tests used for Fig. 4, a clear correlation exists between the LPP and the FMEP for each of the three engine speed.

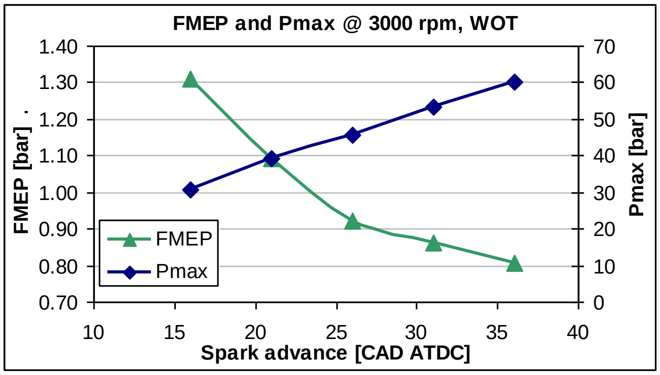

Fig. 6 FMEP and maximum in-cylinder pressure as function of spark advance (3000 rpm, WOT)

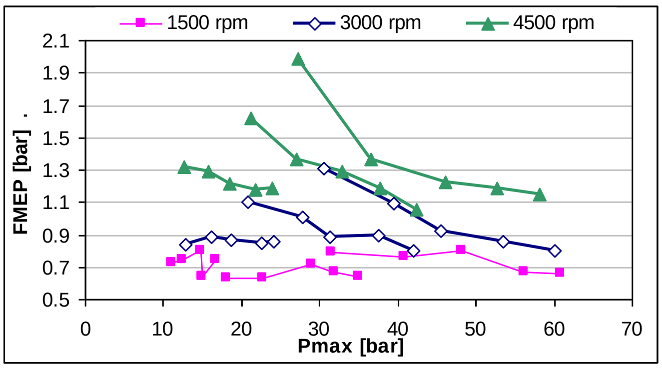

Fig. 7 Experimental FMEP vs. maximum in-cylinder pressure (same operative conditions of Fig. 4)

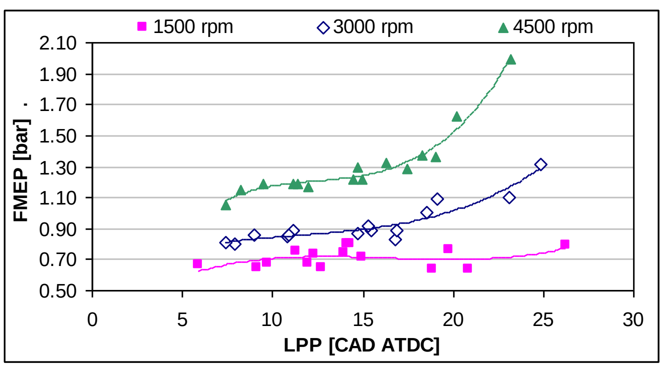

Fig. 8 Experimental FMEP vs. LPP (same operative conditions of Fig. 4)

The author hence tried to express the load term of the friction model using as "load variable" the LPP instead of the P_{max}, using a 3rd order polynomial (as suggested by the fitting curves in Fig. 8); moreover, observing that the LPP effect is amplified by the engine speed (see Fig. 4 and Fig. 8), it was decided to adopt the following formulation:

$$FMEP = A + FMEP_n + \frac{n}{1000}\left(a \cdot LPP + b \cdot LPP^2 + c \cdot LPP^3\right) \quad (3)$$

$$FMEP_n = C \cdot n + D \cdot n^2$$

where $FMEP_n$ represents the speed related contribution to the friction losses.

The calibration of the new friction model requires the determination of the constants *A*, *a*, *b* and *c*, (being *C* and *D* already fixed) which has been carried out minimizing the maximum percentage error with respect to the experimental data: the results obtained, resumed in the graphs of Fig. 9 and Fig. 10, revealed the new model to have a better consistency with the experimental data than the Chen-Flynn model (see Fig. 2 and Fig. 3). Moreover, as exposed in Table 5, all the evaluated errors reduced to the half of those obtained with the Chen-Flynn model.

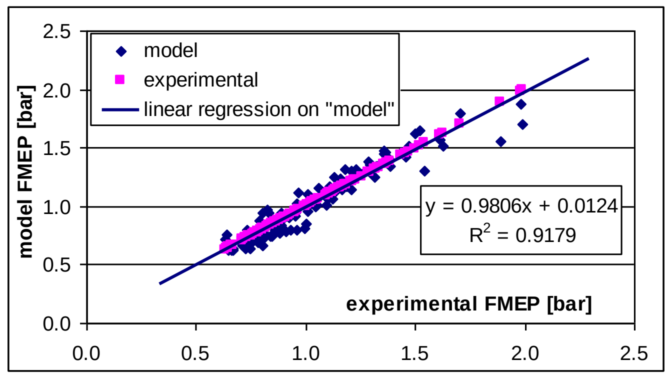

Fig. 9 Comparison between the new FMEP model prediction and the experimental data

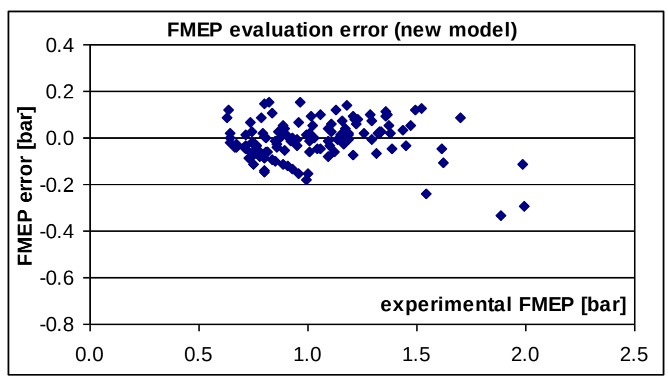

Fig. 10 New FMEP model error vs. experimental data

Model	based on LPP
mean % error	6%
max % error	18%
max error [bar]	0.336

Table 5 Overall results of the new FMEP model

With the aim to involve the maximum in-cylinder pressure in the FMEP evaluation model, a further attempt was made, introducing into equation (3) the term

$B \cdot P_{max}$. The new model constants (A, B, a, b and c) were determined once again minimizing the percentage FMEP evaluation error. The solution found was quite near the one obtained with the simple equation (3), and this means that, for the engine tested, the P_{max} has not a considerable significance on the FMEP. This fact is further on confirmed by the graph in Fig. 11, which reports the FMEP values as function of P_{max}, measured for all the engine speed and all the three loads when the LPP was 15° ATDC: as shown, large P_{max} rise caused small increments or decrements on the FMEP.

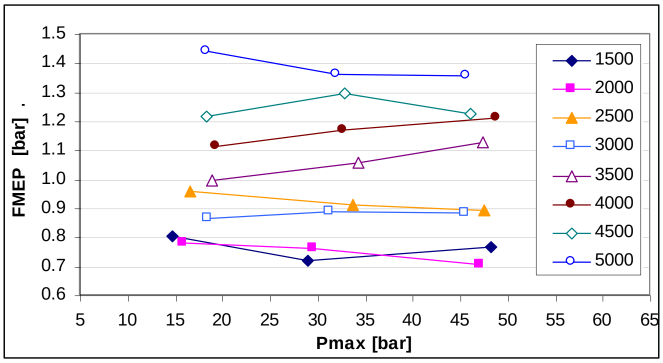

Fig. 11 FMEP function of P_{max} for pressure cycles with LPP≈15° ATDC

Same results can be traced for other fixed values of the LPP, and this explains why the term $B \cdot P_{max}$ tend to have a very little importance. It can be then concluded that, for the engine tested, the phase of the pressure cycle affects the friction losses more than the maximum pressure: the more the pressure cycle is retarded, the higher the friction loss is. This observation finds confirmation in literature, where it is often reported that, among the different contribution to the FMEP, the one which seems to have the stronger effect is the friction between piston and cylinder walls, which, depending on the lateral thrust, reduces if the pressure cycle is advanced.

It must be pointed out, however, that, even if the new model has been developed on the basis of experimental data involving also highly retarded pressure cycles (with LPP up to 26 CAD ATDC), it does not loose validity even if these cycles are excluded: considering, in fact, only pressure cycles with LPP lower than 20 CAD ATDC, the results, as shown in Fig. 12 and Fig. 13 and resumed in Table 6 and Table 7, confirm the new model to have a better consistency with the experimental data, giving a more accurate prediction of the FMEP.

Model	Chen-Flynn
mean % error	8%
max % error	22%
max error [bar]	0.358

Table 6 Overall results of the Chen-Flynn model (pressure cycles with LPP<=20)

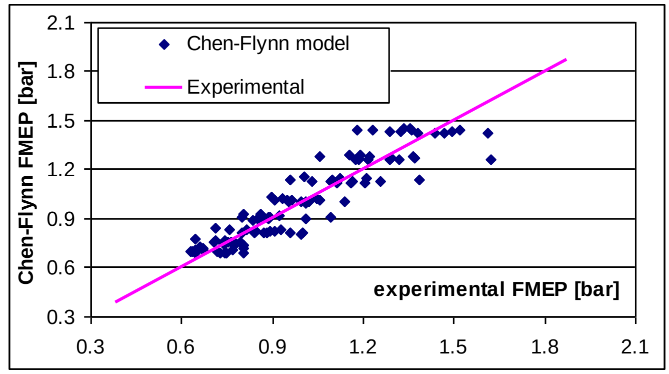

Fig. 12 Comparison between the Chen-Flynn model prediction and experimental data (pressure cycles with LPP<=20)

Model	based on LPP
mean % error	5%
max % error	15%
max error [bar]	0.243

Table 7 Overall results of the new FMEP model (pressure cycles with LPP<=20)

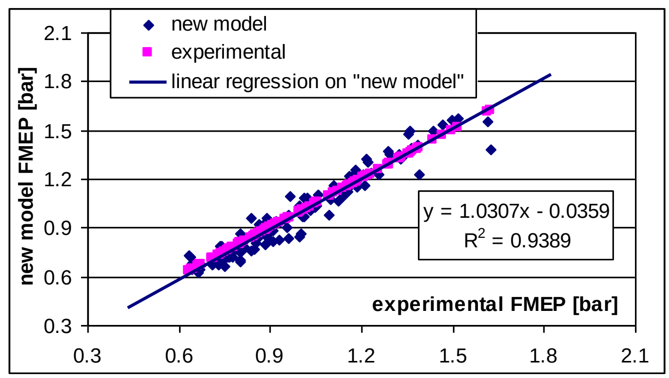

Fig. 13 Comparison between the new FMEP model prediction and experimental data (pressure cycles with LPP<=20)

CONCLUSION

This paper deals with the development of a new simple model for the prediction of the Friction Mean Effective Pressure (FMEP) in Spark Ignition engine. The work started from the observation that the classical adopted Chen-Flynn model may not find consistency with empirical data: experiments carried out on a S.I. fuelled with CNG showed in fact that the maximum in-cylinder pressure poorly influenced the FMEP, which instead revealed to be much more sensitive to the phase of the pressure cycle. On the basis of this observation, the author employed the experimental data to develop a new model, in which the “load variable” maximum pressure has been substituted by the Location of Pressure Peak (LPP): the new model, once calibrated, proved to be effectively more accurate in the prediction of the FMEP than the Chen-Flynn model.

It is worth to remark that this conclusion has been drawn on the basis of the analysis performed on data acquired on the engine test bed, using a series production S.I. engine (see Table 1) fuelled with CNG and may be valid for other engines; the author's intention is to suggest an alternative approach for the setting-up phase of a simple FMEP model, when the classical Chen-Flynn model results in an unsatisfying FMEP prediction.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

ATDC: After Top Dead Centre
BMEP: Brake Mean Effective Pressure
CAD: Crank Angle Degree
CNG: Compressed Natural gas
DAQ: Data Acquisition
ECU: Electronic Control Unit
FMEP: Friction Mean Effective Pressure
$FMEP_n$: Engine speed related term of the FMEP
IMEP: Indicated Mean Effective Pressure
LPP: Location of Pressure Peak
MAP: Manifold Absolute Pressure
MBT: Maximum Brake Torque
n: Engine speed
P_{max}: Maximum in-cylinder pressure
ppr: pulse per evolution
TDC: Top Dead Centre
UEGO: Universal Exhaust Gas Oxygen