

Possibility to Determine Diesel Engine Condition and Tuning from the Application of a Diagnostic Technique at a Single Operating Point

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ABSTRACT

A difficulty which exists when applying diagnostic techniques on large-scale diesel engines operating on the field, is that usually it is not possible to obtain measurement data at a wide engine operating range due to a number of constraints. In the present work is investigated the possibility to overcome this practical difficulty originating from the test procedure for engines operating on the field (i.e. marine or stationary applications). The main objective is to examine if a diagnosis procedure provides similar results when applied at various engine operating conditions. For this purpose an existing diagnostic technique, developed by the authors, is applied at different operating conditions on a large-scale two-stroke diesel engine used for power generation in a Greek island. From the diagnosis analysis, several engine operating parameters (i.e. peak firing pressure, cylinder fuel flow, brake power output, injection advance etc.) are estimated at each operating point, as well as the current operational condition of the engine and its subsystems (i.e. compression condition, compressor and turbine condition etc.). From the comparison of the diagnosis results (i.e. operating parameters, engine condition and tuning) at all operating points examined, rather similar results are obtained, regardless of the operating point at which the diagnosis is applied. However, at one operating point, the diagnosis results differ, which as revealed is attributed to load variation during measurement. For this reason, the diagnostic technique has been modified to account for load variation during a measurement. From the present analysis, it is shown that it is possible to estimate engine current condition and tuning from the application of the proposed diagnostic technique at a single operating point, since most of diagnosis results are similar,

regardless of the operating point at which diagnosis is applied. This is a serious advantage of the proposed technique, especially for large-scale diesel engines (i.e. marine), where it is not easy to perform measurements at a wide engine operating range, as required.

INTRODUCTION

The diesel engine remains the most efficient liquid fuel burning prime mover yet devised and therefore it holds a dominant position in many applications, i.e. marine propulsion, land transport – both road and rail –, power stations etc. The proper and efficient operation of a diesel engine is a major objective, especially for marine or stationary applications (i.e. power generation etc.). For this reason, condition monitoring and fault diagnosis techniques hold an important position in the area of diesel engine development and application, especially for the large-scale one. Up to now, various diagnostic methods and techniques have been proposed from researchers and manufacturers [1-4]. The basic purpose of condition monitoring and fault diagnosis techniques is to provide the current operating condition and tuning of the engine and its subsystems. Moreover, such techniques are used to detect the actual cause of a malfunction or fault. The specific procedure (i.e. detection of the actual cause of a fault) is usually very complicated, considering that the engine operational behavior is affected from a large number of parameters, which are usually very difficult to measure or estimate. Therefore the actual cause of a malfunction cannot easily be determined, using conventional methods. Furthermore, in most applications (i.e. marine) it is not possible to apply trial and error techniques to detect the actual cause of a fault [5].

Most of the proposed diagnostic techniques developed up to now, are based on the processing of measurement data obtained from the engine during actual operation [5-9]. This data are mainly thermo-dynamic parameters, i.e. temperatures, pressures etc. or dynamic ones, i.e. torsional or axial stresses on the engine crankshaft etc. Then, the measurement data are processed from advanced computational tools and techniques [10-12] to provide general or detailed indications for the present engine condition. However, concerning engine measurements several problems exist. For example it is usually not possible to obtain measurement data at the desired operating point (i.e. rpm or load) or at a wide engine operating range. For this reason it is necessary to examine, if the results obtained from the application of a diagnostic technique at a random operating point are of general validity. The aforementioned consideration is a main objective of the present work. For this purpose, an advanced diagnostic technique developed by the authors, is applied at various operating conditions on a large-scale two-stroke diesel engine installed on a Greek island for electrical power production. This is an extremely difficult task considering engine size and use. Cylinder pressure measurements have been obtained from the engine together with various peripheral engine parameters at 4 different operating points (i.e. at 6.2, 8.0, 10.2 and 10.4 MW which correspond to 50%, 65%, 80% and 82% of engine load, respectively). Then, the diagnostic technique is applied at each operating point to estimate the current engine condition and tuning.

The proposed diagnostic technique has been applied up to now on various engines and its reliability to estimate the engine operating condition and detect a malfunction has been verified [5-6], [13-16]. The basic idea of the diagnosis method is based on a mathematical procedure where the current operational condition of the engine is compared to the reference one, which is derived using the shop/sea trials data, referred from now on as “new” engine. The operational behavior of the diesel engine – current or “new” – is described from various operating parameters which are estimated from the diagnosis procedure. The calculations for the engine operating parameters are based on thermodynamics, modeling of the engine and its components and processing of measured engine data. Before the application of the method, the simulation model is calibrated. The calibration uses as basis the engine shop test data, to determine model constants. Having calibrated the model, it is then used to predict the measured data from the shop/sea trials. The comparison between calculated parameters and measured ones verifies the validity of the simulation model.

From the diagnosis analysis several operating parameters are estimated (i.e. brake power output, cylinder fuel flow rate, peak firing pressure, compression condition, ignition and injection timing, turbine and compressor efficiency etc). Thus, at all operating points examined, it is derived a detailed picture for engine condition and tuning and additionally for the condition of engine subsystems. From the diagnosis results at all

operating cases it is revealed that the overall engine condition is acceptable, even though deviations between individual cylinders do exist, which have a negative impact on engine operation. Furthermore, specific irregularities are detected for individual cylinders referring to compression condition, injector condition, injection advance, cylinder fuel flow etc. For this reason, adjustments for engine tuning and other inspections are proposed, to improve current engine condition. On the other hand, it is revealed that the condition of engine subsystems (i.e. compressor, turbine and air cooler) is rather good and they operate properly.

From the comparison of the diagnosis results provided at different operating points, it is observed that in most cases they are quite similar (i.e. an indication for lower compression condition of an engine cylinder is obtained at all operating points). Furthermore, the engine operating parameters estimated from the diagnostic technique (i.e. brake power output, fuelling, compression condition, injection timing etc.) follow a similar fluctuation trend between engine cylinders at the different operating points examined. However, at one operating point (8.0MW) it is observed that the diagnosis results differ and do not follow the general trend of the other operating points (see Figs. 6 & 8). This is possibly attributed to an engine load variation during measurement. To investigate a potential load variation during measurement, a new methodology is introduced, based on the estimation of the initial pressure value at compression initiation. Using the aforementioned methodology, it is verified that engine load variation occurred during the specific measurement. Thus, it is revealed that the proposed technique accounts properly for engine load variation. The diagnosis results obtained for the specific operating point (8.0MW) with the new methodology are similar with the results for the other three points (i.e. 6.2, 10.2 and 10.4 MW) for both tuning and cylinder condition. However, the specific operating point cannot be used to propose adjustments for fuel rack position to achieve uniform cylinder operation as far as power output is concerned.

From the present investigation, it is revealed that the diagnosis results are practically similar, regardless of the operating point at which the diagnosis methodology is applied. Therefore, it is possible to estimate engine condition and detect a malfunction from the application of the proposed diagnostic technique at a single operating point. On the other hand, in order to make proposals for uniform cylinder operation (i.e. power output) it is still necessary to avoid load fluctuation during measurement.

THE ENGINE SIMULATION MODEL – BRIEF DESCRIPTION

The “heart” of the present diagnosis technique is a multi-zone combustion model [17]. As known, simpler simulation models (i.e. single or two-zone) are adequate to describe engine performance for diagnostic purposes. However, in the proposed diagnostic technique the

existing two-zone model used in previous investigations [14-16], is replaced by a multi-zone one. There are two main reasons for this modification. The first is that a multi-zone approach describes more accurately the air-fuel mixing mechanism allowing for the effect of engine geometry, fuel injection characteristics, load variation etc. This allows the use of fairly constant values of the model constants which is extremely useful for the philosophy of the proposed diagnostic technique. The second is that such a model is more suitable for NO_x emissions prediction, which is currently becoming important for both stationary and marine diesel engine applications.

An analytical description of the multi-zone simulation model has been published in the past [17-18]. Since the purpose of the present work is not to describe the simulation code in detail, but its application for diagnostic purposes, it is only given a general description, with special emphasis on the mechanisms related to the diagnosis procedure. The correlations used are based on thermodynamics and the following paragraphs give a brief description of the sub-models used.

SIMULATION OF IN-CYLINDER PROCESSES

Heat Transfer Model – A turbulent kinetic energy viscous dissipation rate $k\sim\varepsilon_t$ model [19] is used to determine the characteristic velocity for the heat transfer calculations as proposed by Assanis and Heywood [21] and described in detail in previous publications [5,19]. The heat transfer coefficient is estimated using the following relation [3,19],

$$h_c = c \operatorname{Re}^{0.8} \operatorname{Pr}^{0.33} \frac{\lambda}{l_{\text{car}}} \quad (1)$$

while the instantaneous heat transfer rate is obtained from:

$$\dot{Q} = A [h_c (T_g - T_w) + c_r (T_g^4 - T_w^4)] \quad (2)$$

The mean gas temperature is determined from the following relation:

$$T_g = \frac{\sum_{i=1}^{n_z} m_i c_{vi} T_i}{\sum_{i=1}^{n_z} m_i c_{vi}} \quad (3)$$

where index i denotes the ' i_{th} ' zone of a total number n_z (air zone included).

In eq. 3 the local temperature of each zone (T_i) is estimated using the ideal gas equation and the 1st thermodynamic law. As known, at high pressure conditions gas behavior can deviate from the ideal state. However for pressures in the range of ~130 bar (i.e.

peak cylinder pressure for the specific engine) the error introduced in the present calculations from the ideal gas consideration is rather small and acceptable for diagnosis applications.

The overall estimated heat exchange rate obtained from eq. (2) is distributed among the surrounding air and the jet zones according to their mass, temperature and specific heat capacity as follows,

$$\dot{Q}_j = \frac{\dot{Q} (m_j c_{vj} T_j)}{\sum_{i=1}^{n_z} m_i c_{vi} T_i} \quad (4)$$

Cylinder Blow-by – An important parameter for engine diagnosis is blow-by which affects both the compression and combustion expansion stroke [22,23]. In the present work a simplified model is used instead of a detailed one developed in the past [24] since the latter requires knowledge of detailed engine data which in most cases are not available for field applications. For this reason blow-by rate is estimated using an equivalent blow-by area between the cylinder rings and the cylinder bore [5]. The mass flow is calculated using isentropic compressible flow relations. The equivalent blow-by area A , is equal to:

$$A = \pi D \delta r \quad (5)$$

where δr is the equivalent cylinder-ring clearance.

As long as the fresh air zone exists the blow-by rate is abstracted from it. When the burning zone (i.e. fuel jet) occupies the entire combustion chamber, then the blow-by rate is attributed to each zone as follows:

$$dm_{bl,i} = dm_{bl,tot} \times \frac{m_i}{m_{tot}} \quad (6)$$

Where $dm_{bl,tot}$ is the total blow-by rate, m_i the zone mass and m_{tot} the total instantaneous cylinder charge mass.

The Jet Model – A multi-zone combustion model is used to simulate the combustion mechanism.

The overall fuel jet is assumed to have the shape of a cone penetrating inside the engine cylinder consuming the surrounding air. Its penetration length inside the cylinder is estimated from empirical correlations that provide the velocity along the spray axis [17,23,25],

$$u = u_{inj} = C_d \left(\frac{2\Delta P}{\rho_l} \right)^{0.5} \quad \text{for } x < L$$

$$u = u_{inj} \left(\frac{L}{x} \right)^n \quad \text{for } x > L \quad (7)$$

where exponent “n” is in the range of 0.7-0.8 and the break-up length “L” is given from the following expression [25,26],

$$L = u_{inj} t_{break} \cong c_1 \left(\frac{\rho_l}{\rho_a} \right)^{0.5} d_{inj} \quad (8)$$

c_1 is a constant and ρ_a , ρ_l the densities of air and fuel, respectively.

The jet angle is considered variable and obtained from the following relation [17,23,25,26],

$$\alpha = 0.05 \left(\frac{d_{inj}^2 \rho_a \Delta P}{\mu_a^2} \right)^{0.25} \quad (9)$$

Furthermore, the effect of air swirl upon jet penetration and cone angle is considered for using the following well tested correlations [26]:

$$x_s = x \left(1 + \frac{c_s}{1 + c_s} \right)^{-1} \quad (10)$$

$$\alpha_s = \alpha \left(1 + \frac{c_s}{1 + c_s} \right) \quad (11)$$

where c_s is given by,

$$c_s = \frac{u_a}{30\pi N r} \quad (12)$$

and r is the radial distance of the inlet ports from the cylinder axis.

Air swirl results to an increase of the air entrainment rate inside the jet. The instantaneous swirl ratio c_s is obtained from the conservation of angular momentum inside the engine cylinder as described in [17].

After impingement on the cylinder walls, the wall jet theory of Glauert [27] is used to determine the jet history upon the cylinder walls. Details concerning this mechanism together with the analytical expressions can be found in [17,19].

Air Entrainment Rate – The air entrainment rate into each zone is estimated from momentum conservation as follows:

$$m_f u_{inj} = (m_a + m_f) u_p \Rightarrow m_a = \left(m_f \frac{u_{inj}}{u_p} - m_f \right) \quad (13)$$

$$m_{a,cor} = c_a m_a$$

The air-entrainment rate coefficient c_a is used to adjust the overall air entrainment rate to account for

combustion chamber geometry and injector dispersion quality. The value is used to define injector condition.

Combustion Rate – The injected fuel is divided into groups having the same Sauter Mean Diameter depending on their time of entrance into the burning zone. The Sauter Mean Diameter of each group (DSM) is obtained from the following semi-empirical expressions used by various researchers in multi-zone codes [17,25] derived by analyzing experimental data:

$$D_{SM,1} = 0.38 Re_{inj}^{0.25} We_{inj}^{-0.32} \left(\frac{v_l}{v_a} \right)^{0.37} \left(\frac{\rho_l}{\rho_a} \right)^{-0.47} d_{inj} \quad (14a)$$

$$D_{SM,2} = 4.12 Re_{inj}^{0.12} We_{inj}^{-0.75} \left(\frac{v_l}{v_a} \right)^{0.54} \left(\frac{\rho_l}{\rho_a} \right)^{0.18} d_{inj} \quad (14b)$$

where subscripts “1” and “2” denote complete and incomplete sprays, respectively. The Reynolds and Weber numbers are given by,

$$Re_{inj} = \frac{u_{inj} d_{inj}}{v_l}, \quad We_{inj} = \frac{u_{inj}^2 d_{inj} \rho_l}{\sigma} \quad (15)$$

The Sauter Mean Diameter is taken to be the maximum of these two values calculated in Eqs (14a) and (14b). The fuel droplets evaporate and the resulting vapor mixes with the surrounding air to form a combustible mixture. For the evaporation process, the model of Borman and Johnson [20] is used, described in [17].

Ignition Delay – Ignition delay of each spray element (i.e. zone) is determined from the relation [22,23,28],

$$S_{pr} = \int_0^t \frac{1}{a_{del} P_g^{-2.5} \Phi_{eq}^{-1.04} \exp\left(\frac{5000}{T_g}\right)} dt = 1 \quad (16)$$

where “ Φ_{eq} ” is the local equivalence ratio of the fuel air mixture inside the zone and T_g , P_g the cylinder temperature and pressure respectively expressed in K and bar.

The combustion rate of fuel is modeled using an Arrhenius type equation of the form,

$$\dot{m}_{fb} = K_b C_f^{af} C_o^{ao} e^{\frac{-E_c}{T}} \quad (17)$$

where K_b is a constant, E_c the reduced activation energy (K) and C_f , C_o are the mass concentrations of fuel and oxygen respectively.

THE FUEL INJECTION SYSTEM

A detailed simulation model has been developed in the past for the fuel injection system [29] requiring a great number of geometrical data that are extremely difficult to obtain for commercial engines. For this reason a simpler model is used for diagnostic purposes [30]. In this model the following control volumes are considered: High pressure pump chamber (1), Delivery valve chamber (2), Delivery pipe from pump to injector (3), Injector (4)

The simulation of each control volume is accomplished by considering the incoming and outgoing volume flow rates and the fuel compressibility, obtaining the following relation [30],

$$\frac{dP_j}{dt} = \frac{K_f}{V_j} \left(\frac{dV_j}{dt} - \dot{V}_{tj} \right) \quad (18)$$

where $\dot{V}_{tj} = \sum \dot{V}_j$ is the total net volume flow rate into the control volume and dV_j/dt is the rate of its volume change. The volumetric flow rate through orifices, various openings or ports is given by the formula,

$$\dot{V}_j = A_j C_{dj} \left(\frac{2\Delta P_j}{\rho_j} \right)^{0.5} \quad (19)$$

where "j" is the corresponding volume. The delivery valve is modeled in a rather simple way that has been found to give relatively accurate results for engine performance prediction. It is considered as a check valve allowing the fuel to flow only from the delivery chamber to the fuel pipe. The injector is also modeled, as a check valve, allowing fuel to flow towards the combustion chamber only when the pressure exceeds its opening pressure. The pressure history inside the fuel pipe is obtained using the method of characteristics to solve unsteady flow equations [30].

From the application of this model the injection rate is derived and the spray characteristics (i.e. initial zone velocity and mass) for the spray model are estimated.

SIMULATION OF INLET AND EXHAUST SYSTEM

In the present work the filling-emptying method [21,23,31,32] is used to estimate the pressure-temperature versus time history in the two manifolds (intake-exhaust). The mass flow rate through the turbine nozzle is calculated using isentropic flow relations and the effective area A_{eff} of the turbine nozzle. This is adequate for the specific engine which is equipped with a constant pressure turbocharging system.

SCAVENGING MODEL

An important mechanism for two-stroke turbocharged engines is scavenging [22]. For this reason a two-zone

scavenging model has been developed while in the past simple correlations were used. During gas exchange, the cylinder contents are divided into two parts: one consisting only of fresh entrained air, and a second consisting of combustion products from the previous cycle and fresh entrained air. During scavenging, part of the intake air escapes directly to the exhaust manifold (short-circuiting) from the fresh air-zone reducing the exhaust gas temperature considerably [23]. If $d_{ma,inl}$ is the total amount of air entering the cylinder at a certain time instant during intake, one part enters the fresh air and the remaining one the combustion products zone. These amounts are given by the following two relations,

$$dm_{a,fz} = dm_{a,inl} (1 - C_{1scav}) \quad (20a)$$

$$dm_{a,cz} = dm_{a,inl} \times C_{1scav} \quad (20b)$$

If $d_{mg,exh}$ is the total amount of exhausted cylinder mass to the exhaust manifold; this is taken partially from the fresh air zone and the combustion products one. The gas masses taken from the two zones are given from the following relations,

$$dm_{g,fz} = dm_{g,exh} \times C_{2scav} \quad (21a)$$

$$dm_{g,cz} = dm_{g,exh} (1 - C_{2scav}) \quad (21b)$$

where C_{1scav} and C_{2scav} are the constants of the scavenging model. At the end of scavenging (start of compression stroke), perfect mixing between the two zones is assumed resulting in a single zone that is a mixture of both fresh entrained air and combustion products from the previous cycle. The previous constant values are estimated using an iterative procedure to match the measured exhaust gas temperature before the turbine, using the engine shop test data.

TURBOCHARGER MODELING

It is usually difficult to obtain characteristic charts for the compressor and the turbine especially for existing engines. To overcome this problem, the method of operation similarity [6,33] is used from which are reproduced the charts from existing experimental data. The method is efficient for engine loads in the range of 40% to 100% and is as follows: We calculate, using a least squares method and data from the shop tests, the constants of polynomial curves fitting the following functions,

$$\eta_{isc} = f_1(\phi) \quad (22)$$

$$\eta_{isT} = f_2(\phi) \quad (23)$$

$$k_{is} = f_3(\phi) = \Delta h_{is}/U^2 \quad (24)$$

where $\phi = \dot{m}/(\rho AU)$ is the flow coefficient.

The data required for the calculation of the turbine and compressor characteristic maps are:

- (a) pressure before and after the compressor,
- (b) pressure before and after the turbine,
- (c) air temperature before and after the compressor,
- (d) exhaust gas temperature before and after the turbine,
- (e) rotational speed of the turbocharger,
- (f) air and exhaust gas mass flow rate obtained from the simulation using the previous data.

Then using the previous relations T/C performance is estimated from an iterative procedure. The mass flow rate through the turbine nozzle is estimated using compressible flow relations [30,31] and the equivalent nozzle flow area “ A_{eff} ”.

AIR COOLER

The air cooler is modeled in a rather simple way compared to the turbocharger, based on experimental data from the engine shop tests. The pressure drop and $\phi = A/C$ effectiveness are considered as functions of the air mass flow rate through A/C [5,31] as follows,

$$\varepsilon = 1 - b \dot{m}^2 \quad (25)$$

$$\Delta P_{ac} = a_{ac} \dot{m}^2 \quad (26)$$

where effectiveness “ ε ” is defined as,

$$\varepsilon = \frac{T_{a,in} - T_{a,out}}{T_{a,in} - T_{c,in}} \quad (27)$$

and subscripts “a, c, in, out” denote: air, cooling medium, inlet and outlet from air cooler respectively. Constants “ a_{ac} ” and “ b ” are determined from the constants determination procedure, so that the values match the ones obtained from the engine shop tests. The mass flow rate of air is calculated from the simulation model using engine data from the shop tests or the measurements.

THE PROPOSED DIAGNOSTIC TECHNIQUE

From the aforementioned brief description of the simulation model it is shown that a number of constants are used in various correlations (Table 1). As observed, the model constants are parameters related to geometrical characteristics of the engine or other physical processes inside the cylinder. For this reason, their values should remain practically constant for a specific engine, regardless of the operating conditions. Thus, model's constants are related to the condition of various engine subsystems and therefore their values are considered as an index for the current condition of engine components.

Table 1 Constant values determined from calibration procedure

Constant	Description	Eq.(-)
TDC	Top Dead Center Position	(-)
CR	Compression Ratio	(-)
c	Heat transfer constant	(1)
δr	Equivalent piston ring clearance	(5)
T_w	Cylinder wall temperature	(2)
a_{del}	Ignition delay constant	(16)
c_a	Air-entrainment rate coefficient	(13)
K_b	Constant for combustion rate	(17)
A_{eff}	Equivalent nozzle effective area	(-)
a_{ac}	Constant for A/C pressure drop	(26)
b	Constant for A/C effectiveness	(25)
h_c	Heat transfer coefficient	(32c)

The reference constant values are estimated using measurement data from the shop tests (calibration procedure). The overall procedure is based on the optimum fitting between calculated and measured operating parameters given in Table 2. Then, the same procedure is followed and the model constants are estimated for the current engine condition using the measured data (i.e. measured cylinder pressure traces and peripheral data). From the comparison between current and reference values of model constants, the condition of various engine components and subsystems is obtained. This is the basic concept of the proposed diagnostic technique applied in the present work.

Table 2. Shop Test Data Used for Model Calibration

Main Engine Data	Inlet/Exhaust System Data
Engine Speed	Gas pressure before/after the turbine
Fuel flow rate	Turbine speed
Brake power output	Air pressure before/after compressor
Peak firing pressure	Gas temperature before/after the turbine
Peak compression pressure	Compressor air inlet temperature
Fuel rack setting	Air temperature before/ after the air cooler
Injection advance	Water temp. before/after A/C
	Air cooler pressure drop

In Figure 1 it is given a schematic depiction of the calculation procedure for the constant determination procedure described:

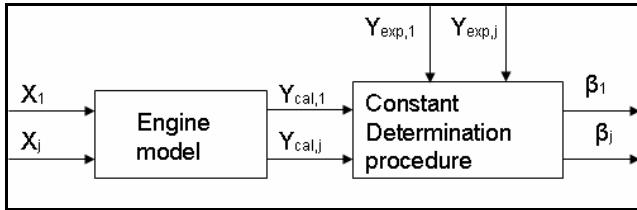


Fig. 1 Constant determination philosophy

In Fig. 1, variables $X_1, \dots, X_j$ correspond to the input data of the simulation model (i.e. measured pressures, temperatures etc), variables $Y_{cal,1}, \dots, Y_{cal,j}$ correspond to the calculated output data from the simulation (i.e. power output, fueling, bmep, bsfc etc) and variables $Y_{exp,1}, \dots, Y_{exp,j}$ to the respective measured ones. Variables $\beta_1, \dots, \beta_j$ are the model constants estimated from the calculation procedure.

A fault or malfunction exists if the following criterion is satisfied:

$$\left| \frac{\beta - \beta_o}{\beta_o} \right| \times 100\% \geq 3\% \quad (28a)$$

where β are the constants determined at the present state and β_o their corresponding values at the reference state i.e. new engine condition. The three percent error limit is used to account for measurement and calibration error.

The condition of an engine component or subsystem is now derived from the following relation:

$$\frac{\beta}{\beta_o} \times 100\% \quad (28b)$$

where β_o refers to the constants of a "new" engine estimated from the shop test data.

ESTIMATION OF MODEL CONSTANTS

In the following, it is described in detail the mathematical procedure to estimate the values of the model constants (i.e. for both reference and current engine operating condition).

Assuming that the number of constants to be determined is "j", and the available number of measured parameters is "i" a least squares method is used, aiming to the minimization of the following function,

$$f_{err}(\beta_{1,2,\dots,j}) = \sum_i (Y_{exp,i} - Y_{cal,i})^2 \quad (29)$$

where Y_{exp} and Y_{cal} are the matrices of the experimental and calculated values respectively. The problem is then reduced to the minimization of the sum of squares

function f_{err} which is non-linear in parameters [1,2]. The algorithm for the solution is as follows,

$$\beta^{k+1} = \beta^k + Z^k X^{T(k)} (Y_{exp} - Y_{cal}^{(k)}) \quad \text{where} \quad (30)$$

$$Z^{-1(k)} \equiv X^{T(k)} W X^{(k)}$$

In this relation β represents the constants or parameters vector to be determined, k is the iteration number, X is the sensitivity matrix defined by,

$$X = \begin{bmatrix} X_{11} & \dots & X_{1j} \\ \vdots & \ddots & \vdots \\ X_{i1} & \dots & X_{ij} \end{bmatrix} = \begin{bmatrix} \frac{\partial Y_{1,cal}}{\partial \beta_1} & \dots & \frac{\partial Y_{1,cal}}{\partial \beta_j} \\ \vdots & \ddots & \vdots \\ \frac{\partial Y_{i,cal}}{\partial \beta_1} & \dots & \frac{\partial Y_{i,cal}}{\partial \beta_j} \end{bmatrix} \quad (31)$$

and W is a weighting matrix defined to consider the importance of each parameter by either including or excluding it from the current calculation step (values 1 and 0 respectively).

A more detailed description of the constant determination procedure is given in [16]. However, to give a general idea, it is mentioned that the constants determination procedure is divided into two parts. The first involves the closed cycle calculations for constant values related to compression, injection, combustion and expansion. The second is related to the gas exchange mechanism and the involved subsystems.

VALIDATION OF CONSTANT DETERMINATION PROCEDURE

Having determined model constant values at the reference and the current state, the simulation model can be used to predict engine performance characteristics at both operating conditions. To verify the accuracy of the constant determination procedure, it is given in Figs.2 & 3, the comparison between calculated and measured cylinder pressure traces for cylinder No.1 at 6.2MW and at 10.2 MW. From the figures is observed a good agreement between measured and calculated cylinder pressure traces. Thus, the accuracy of model constants estimation procedure and model's predictive ability are both verified. Similar results are obtained for other engine parameters (i.e. power output etc.).