Numerical Assessment of Different Engine Model Levels in the View of Complex Hybrid Application

Giuseppe Di Luca[1,*], Massimiliano Muccillo[1], Giovanni Giardiello[1], Alfredo Gimelli[1] and Gabriele Di Blasio[2]

[1]*Unina, Dipartimento Ingegneria Industriale via Claudio, 80125 Napoli, Italy*

[2]*Istituto Motori – CNR, via Marconi 4 80125 Napoli, Italy*

Abstract: Despite the degree of railway electrification in many EU countries is higher than 50%, the diesel-driven railway vehicles continue to play an important role. As known, internal combustion engines, especially diesel engines, have also long been recognized as a significant source of pollutant emissions contributing to poor air quality, negative human health impacts and climate change. The future emissions regulatory control programs and the fuel-saving requirements for the new diesel engines for railways applications push worldwide OEMs, suppliers and scientific communities to investigate more advanced and alternative propulsion systems in which the diesel engines could still play an important role. Thus, the design of new power trains becomes more challenging considering the even more strict emission and efficiency targets. In this context, numerical simulation represents an essential tool in the entire development and optimization process of power trains. This study focuses on the numerical assessment of three different models of the same engine, characterized by different model accuracy, in order to evaluate the trade-off between model accuracy and computational time. The evaluation is carried out by performing the new emission standard Non-Road Transient Cycle (NRTC) applying the EU Non-Road Mobile Machinery (NRMM) directive to rail diesel vehicles. This work considers a 560 kW Heavy-Duty (HD) diesel engine. Regarding the models, the second and the third model are derived from the first one through an appropriate numerical procedure. The first, more accurate 1D model, is adequate when a deeper system analysis is required (*i.e.* wave dynamics, turbo-matching, etc.), while, for the evaluation of the global performance, the simplest model approach is more appropriate for complex systems, such as a hybrid powertrain. Indeed, the simplest model, despite its lower accuracy, shows good predictive results in terms of cumulative fuel consumption and cumulative NOx emissions over a transient homologation cycle. Moreover, for the lowest model accuracy, the *real time factor* is significantly lower compared to the more detailed one of about 250 times.

Keyword: Engine model accuracy, Heavy duty diesel, Engine numerical simulation, Real time factor.

1. INTRODUCTION

In general, the degree of railway electrification is, in many countries in Europe, higher than 50%. For example, in Sweden, Italy and Austria the degree of electrification ranges between 68% and 71% for the complete rail network [1]. However, the railway network is rarely 100% electrified due to the not convenient cost-benefit ratio or where electrification is difficult to realize (*i.e*. harbors, loading tracks, etc.) [2]. As stated in [1], at the age of the work, the average value of electrification in the EU (considering 27 members of countries) is around 52%. In later years, this value remains fairly constant as remarked in [3] and [4]. For this reason, in some countries, the regional traffic or the traffic on low utilization lines is covered with diesel trains. Generally, they are either single coaches or fixed coupled coaches consisting of 2 or 3 power cars. Each power car is driven by one medium power diesel engine, that ranges between 250 and 560 kW, and often derived from trucks or industrial engines [5]. In the case, the main concerns regards the harmful pollutants PM and NO_x and CO_2 [6] According to the European Environment Agency (EEA), nitric oxide (NOx) and particulate matter (PM) emissions from the rail sector account for only 1–1.5 % of the total emissions from all transport sectors. The impact of railway sector is a small contribution compared to that of the other sources of pollutant emissions [7]. However, the task to decrease the consumption of fossil fuel, the requirement of exhaust-free stations and the reduction of pollutant emissions remain significant. Additionally, the application of the actual NRMM Directive [8] to rail diesel vehicles and the upcoming new stricter emission standard named Stage V raise significantly the development of new powertrains in terms of design, performance, fuel consumption, emissions, etc. In order to achieve these goals, the numerical simulation can represent a valid support in the engine optimization or in the whole development of a new form of powertrain. This aspect becomes more relevant when considering alternative forms of propulsion, such as hybrid propulsion, where more components (*e.g*. electrical motors, batteries, etc.) are incorporated into. One of the greatest challenges in numerical simulation is to establish the right compromise in terms of model accuracy and computational time [9]. This challenge is particularly true for engine modelling, where different models, in terms of level of details, can be taken into account.

*Address correspondence to this author at the Unina, Dipartimento Ingegneria Industriale via Claudio, 80125 Napoli, Italy;
E-mail: giu.diluca@studenti.unina.it

E-ISSN: 2311-8741/20

In particular, the "level of detail of a model" is stated in [10] as "an assessment of the extent to which the observable system elements and the assumed system relationships are included in the model", where "level of detail" refers to the system that the model represents. Regarding the engine modelling, the different level of details depends on the phenomenon to be analyzed and on the x-D CFD simulation level approach.

The most accurate approach is the 3D-CFD generally used to investigate a specific phenomenon (*e.g*. mixture formation, combustion process, etc.) limited to an engine component or confined volumes, rather than the whole engine [11]. Contrary to 3D-CFD models, 1D models allow an engine system-level perspective. They can predict the distribution of gas properties in only direction, the axis of primary flow in the air-path. This numerical approach is suitable also to investigate some forms of combustion [12] or a new form of valvetrain [13]. A detailed 1D engine model offers information over hundreds of subs–volumes in engine performance simulation. But if computational speed is of priority, a detailed model can be converted into a simplified 1D model by lumping sub-volumes together, where possible. In this way, the computational time is reduced thanks to a larger time step and fewer sub-volumes to solve. The simplest approach to describe the entire engine in a data driven manner is through a map-based engine model which comprises the entire engine behavior into maps. With all given maps, vehicle performances and fuel consumption can be reasonably assessed for various legislation cycles and drive train configurations [14]. In this modelling approach engine components are not modelled, so the computational requirements are low. In this regard, there is a lack of information in literature in providing a trade-off in terms of model accuracy and computational effort considering a railway application when performing a transient emission homologation cycle. Related to this point, this paper provides an assessment on three different model levels of the same engine over NRTC transient homologation cycle. After an overview over the models presented, a section will be reserved to illustrate the implementation of the transient cycle into them. In the last part of the paper the main outcomes with indication on further future advances are discussed.

2. MATERIALS AND METHOD

2.1. Engine Characteristics and Modelling

The engine modelled in this study is a 560 kW HD Diesel Engine generally used in railcar or Diesel Multiple Units (DMUs) applications. Specifically, it is a V8 direct-injection four-stroke engine with a total displacement of 20 dm^3, turbocharged. The main engine characteristics are reported in Table **1**.

Table 1: Engine Characteristics [15]

Power	560 kW @ 2100 rpm
No. of cylinder and arrangement	8 – V90
Valves per cylinder	4
Turbocharging	Two Turbochargers equipped with aftercooler
Intake air cooling	Single common charge cooler
Firing order	1-3-7-2-6-5-4-8
Bore	145 mm
Stroke	152 mm
Compression Ratio	17.4:1
IVO / IVC	27.5° BTDC / 53.5° ABDC
EVO / EVC	60° BBDC / 22.5° ATDC
EGR	High – pressure circuit with 2 separate coolers
Injection System	Common Rail
Injection Management	Multi-injection

The engine is made up of two in-line four-cylinder banks with a bank angle of 90°. Each cylinder bank is fed by a turbocharger system.

In order to model the engine, the 1D modelling approach was used. In particular, the starting model was provided by a previous authors study [15], where a simulation model of a HD diesel engine, generally used in DMU, was developed and validated. The provided engine model is defined, in terms of "depth" of detail, as a "detailed 1D high fidelity model". In the case, the whole engine layout is discretized and modelled on the base of the real sub-components characteristics. In the figure below a scheme of the detailed model is depicted.

Then a Fast Running Model (FRM) is modeled. In the case, the difference in comparison to the detailed model consists of considering the intake and exhaust plenum as a unique element, as highlighted in Figure **2**. This permits theoretically a lower computational time at the expense of reducing the accuracy in the definition of the possible differences of in cylinder intake and exhaust characteristics.

The third level of engine model, in terms of "depth of details", taken into account in this work is the so called "map-based model". It models an engine just implementing performance maps describing the engine

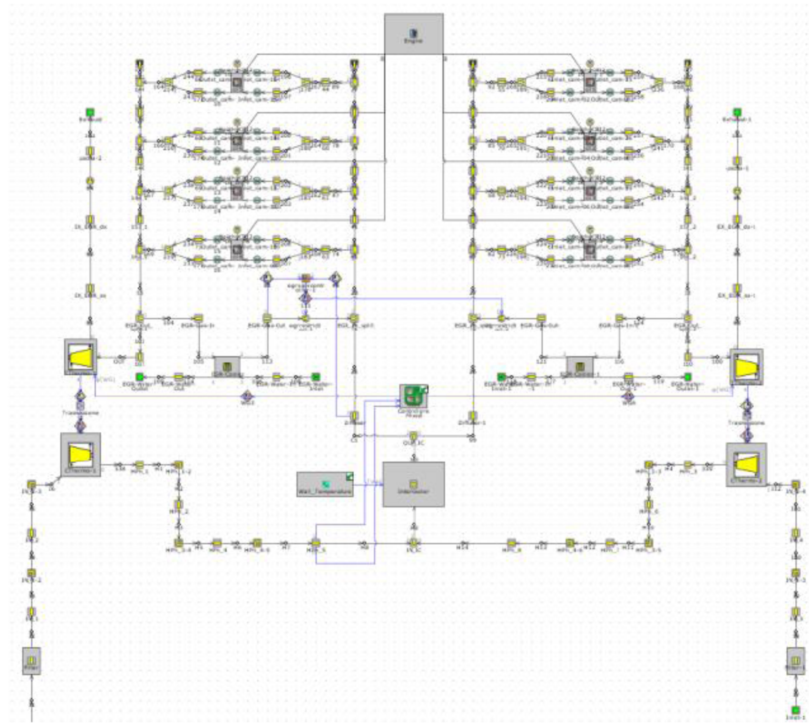


Figure 1: Detailed model of HD diesel engine.

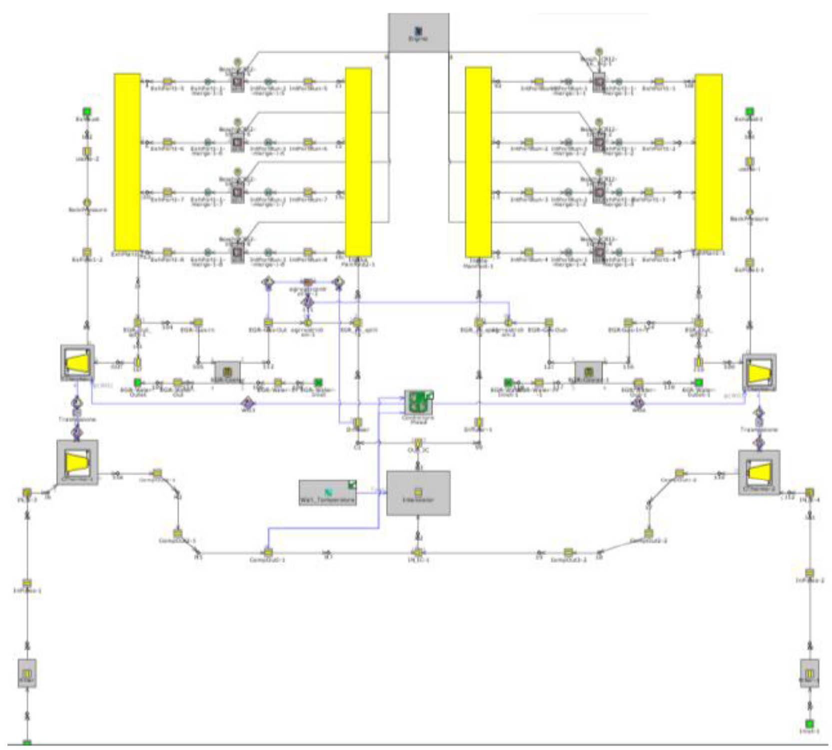


Figure 2: FRM of HD Diesel Engine.

behavior across the speed–load range. These maps are strictly correlated and provided by the OEMs, since they are obtained experimentally via dynamometer testing. But testing phase is both time consuming and expensive. For this study, since the physical engine at test bench, was not available, the performance maps were obtained via simulation approach starting from an experimentally validated "detailed model" previously introduced. Through a specific procedure, available in [16] and by means of a Full Factorial DoE, the engine performance maps were obtained.

Table 2: Weighting Factors of C1 ISO 8178 Test Cycle

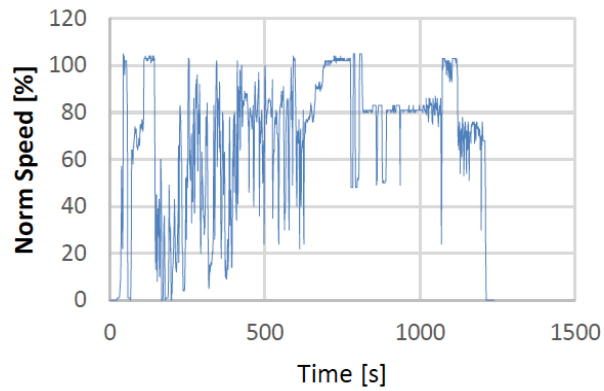
Torque[%]	100	75	50	10	100	75	50	0
Speed	Rated Speed				Intermediate Speed			Low Idle
Weight factor	0.15	0.15	0.15	0.1	0.10	0.1	0.1	0.15

2.2. Emission Test Cycle

In order to make the emission results more representative during real driving conditions, a new transient test procedure, named NRTC, was developed for the emission measurement. The new test, introduced in cooperation with US EPA (United States Environmental Protection Agency) and European Commission, needs to be used in parallel with the prior steady-state schedule, ISO 8178 C1, referred to as the Non-Road Steady Cycle (NRSC) [8]. The ISO 8178 is an international standard designed for non-road engine applications which includes a collection of different steady-state engine dynamometer test cycles (designated as C1, C2, etc.) for different classes of engines and equipment. Each of these cycles represents a sequence of several steady-state modes with different weighting factors. The C1 homologation test points are showed in terms of speed, torque and weighting factor in Table **2** [8].

The NRTC procedure represents the testing of the engine in transient conditions when engine speed and torque change dynamically. It is considered as one of the most stringent transient cycles since it has higher speed-load variations compared to other legislative transient cycles. The total length of the test is 1238 seconds, and the directive gives the program for the dynamometer which represents a sequence of so-called normalized speed and normalized torque of the engine.

Looking at the emission measurement test procedure, while the NRSC was used for stage I, II and IIIA testing, the NRTC, for non-road engines, it was still not mandatory for stage IIIA but obligatory for stages IIIB / IV and upcoming Stage V. Table **3** shows the emission targets for the past, current and upcoming regulations [8].



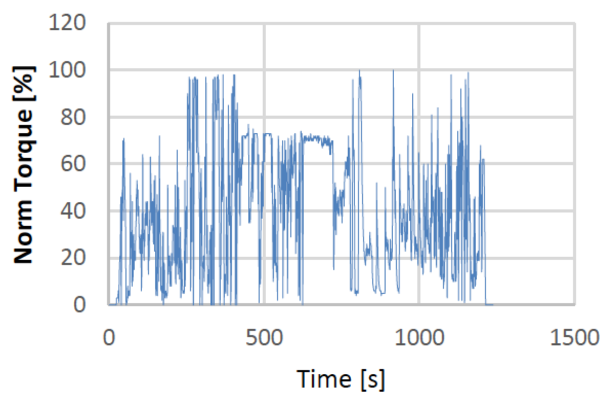


Figure 3: NRTC: Normalized Speed; Normalized Torque.

The simulations performed in this study are carried out considering the NRTC cycle and after the "denormalization procedure" of the engine speed and torque. The "denormalized" actual engine speed and torque sequences appear as:

Table 3: Evolution of the Emission Standard Limits for Non - Road Diesel Engines in the Power Range 130 - 560 kW

Stage	Date	Emissions [g/kWh]				
		CO	*HC*	*NOx*	*PM*	*PN*
IIIB	01.2011	3.50	0.19	2.0	0.025	-
IV	01.2014	3.50	0.19	2.0	0.025	-
V	2021	3.50	0.19	2.0	0.015	$1x10^{12}$

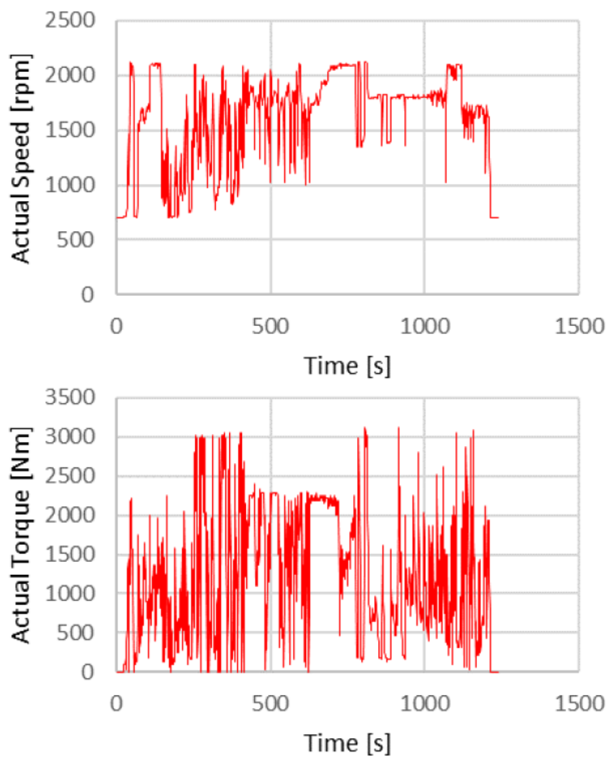


Figure 4: NRTC: Actual Speed; Actual Torque.

More information about denormalization procedure are reported in the appendix section [A.1]

The starting model, introduced previously, can perform the NRSC ISO 8178 C1 homologation cycle, in other words, it was able to run in pre–defined stationary speed-load points. In order to implement a transient cycle in this model, where the load and engine speed vary continuously, the initial model required some changes to perform the transient NRTC homologation cycle.

Indeed, the implementation of the transient operating cycle requires the knowledge of the real engine performance over the whole operating range of interest. Due to the lack of data, at partial load conditions, in order to build the whole engine maps, the data were obtained by means of simulation activities, starting from the validated model.

As known, the diesel engine power control is obtained by means of regulating the amount of fuel injected. Thus, the quantity of fuel to be injected at each operating point was defined as a function of the speed and load once defined the power target.

Indeed, to define the requested fuel mass to reach the power targets, an injection PID controller has been implemented into detailed model. At every time – step the controller calculates the desired amount of fuel considering several physical quantities of the engine (*e.g.* fuel energy, engine displacement, etc) in addition to the instantaneous values of engine rpm, power and the related airflow. With this simulation approach, seventy engine operating points were identified, and the operating maps of the engine were obtained by interpolating all points (Figure **5**).

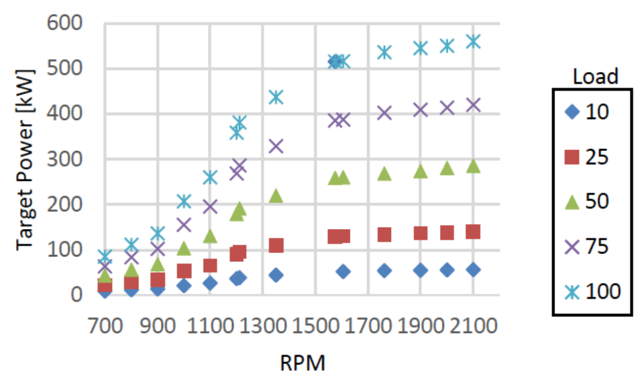


Figure 5: Speed-Power-Load [%] output map.

All the input maps (injection parameters, thermodynamic conditions, actuator positions, etc..) were derived and grouped into maps in order to complete the modelling of the "map-based" model able to run transient cycles too. Instant by instant, the simulation software via interpolation can calculate the desired input when performing the transient cycle.

The load level and the engine rpm are those imposed by the NRTC cycle after the denormalization procedure. For example, in Figure **6**, is schematically depicted how to implement the NRTC cycle to evaluate, via a look–up 2D table, the amount of fuel to be injected to achieve the load target.

3. RESULTS AND DISCUSSION

This paragraph is divided into three sections. The first one shows the FRM calibration results and its accuracy referred to the detailed model. Once the implementation of the transient NRTC cycle was performed into the models listed, simulations were carried out. The second and third section show the results that permit to assess the differences in terms of predictive accuracy and computational effort between the models.

3.1. Fast Running Calibration Results and Accuracy Check

The FRM was calibrated into the same steady points in which the detailed model was validated. The

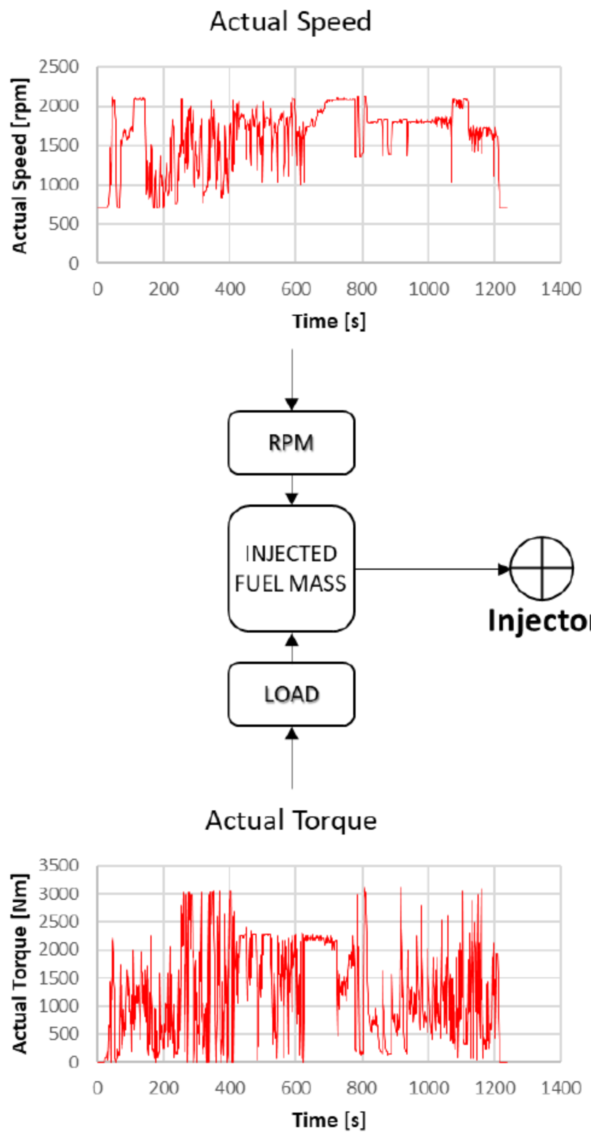


Figure 6: Schematic "Actual Speed" and "Actual Torque" implementation.

calibration process is composed of several steps. At each timestep, it is verified the matching with the original simulation result. The main results are shown below.

During the conversion process, especially where the volumes are joined (intake and exhaust plenums), it is necessary to identify the components which can restrict the timestep, *i.e*. especially the exhaust manifold due to the highest gas velocities occur into it. To ensure the right accuracy in comparison to the detailed model, both in terms of pressure and temperature, a recalibration of the duct diameters or heat transfer multiplier must be varied respectively. An example of the recalibrated heat transfer multiplier of the turbine inlet and outlet is reported in Figure **7**.

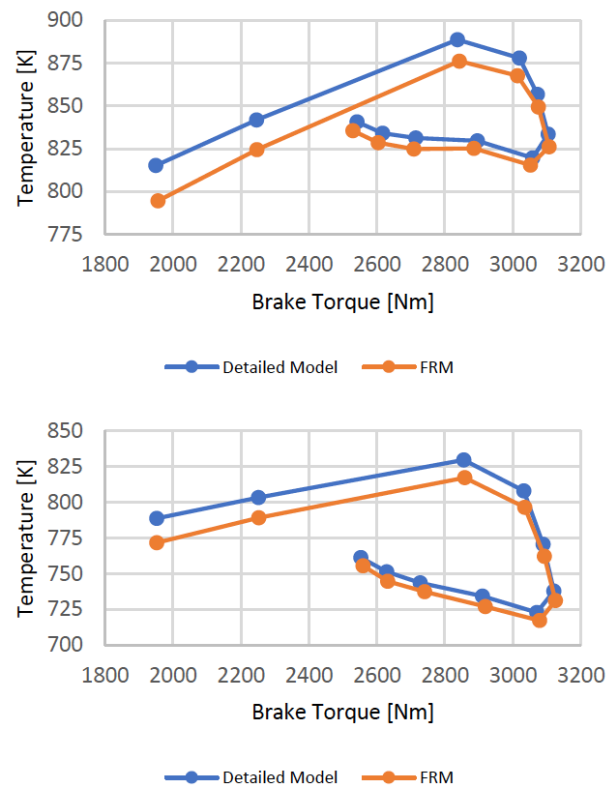


Figure 7: Exhaust manifold: Inlet and Outlet temperature comparison.

As can be seen, despite slight deviations between the trends, the results can be considered satisfactory. After the calibration process in order to compare the accuracy between the two models, some key parameters have been considered. Indeed, in terms of "performance" (Figure **8** Figure **9**), brake power, IMEP and fuel consumption prediction can be considered comparable.

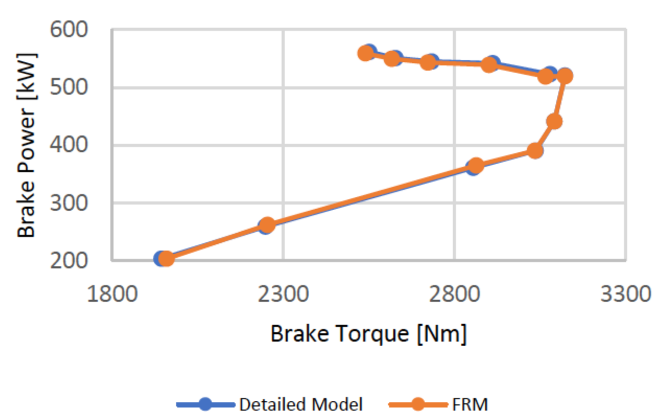


Figure 8: Brake Power Comparison.

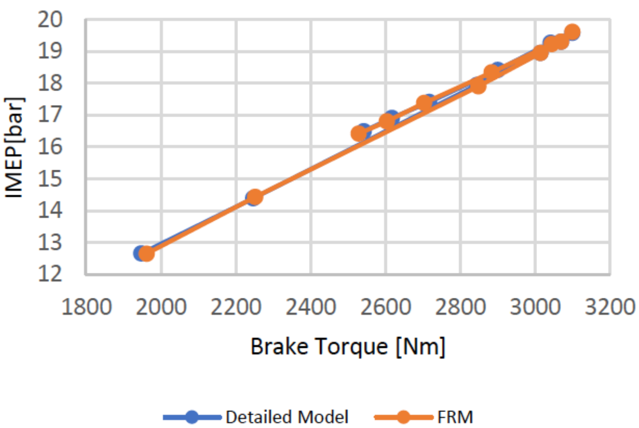


Figure 9: Imep Comparison.

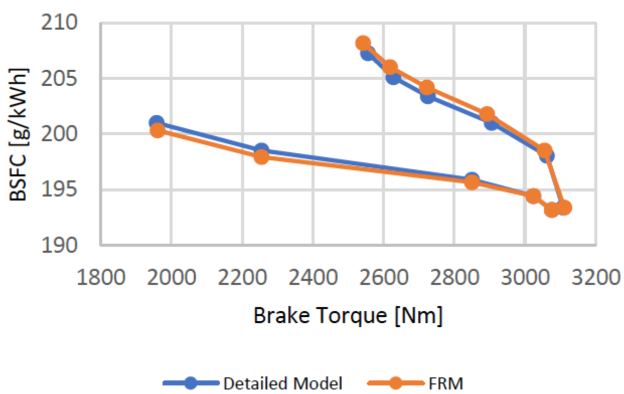


Figure 10: Brake Specific Fuel Consumption comparison.

Figure **11** shows the comparison in terms of timesteps in crank angle units. As expected, FRM provides an improvement in terms of simulation speed, compared to the detailed 1D model, due to an increased value of timestep. This trend is highlighted also by comparing the timestep in seconds (Figure **12**).

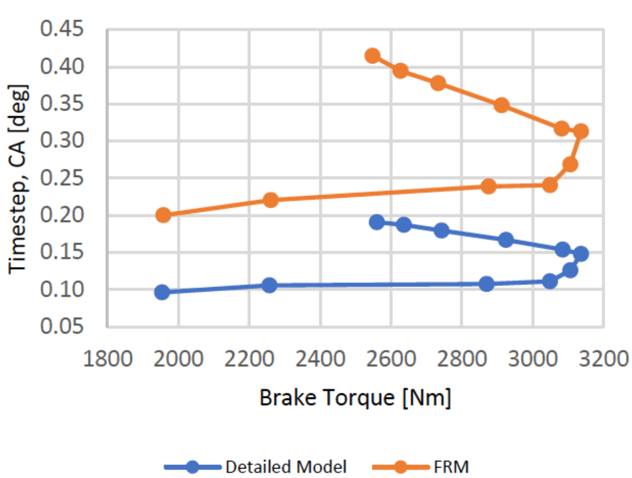


Figure 11: Timestep comparison in crank angle resolution.

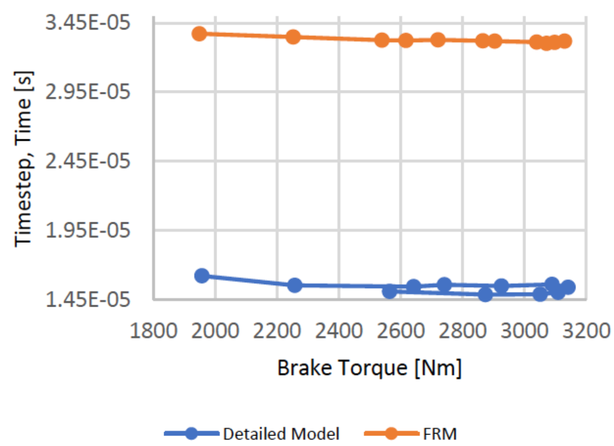


Figure 12: Timestep comparison in Time.

The user-imposed maximum timestep for the detailed model is 0.2 CAD. In any case, to have stable solver operations the timestep and discretization length must meet the Courant condition [17]:

$$\frac{\Delta t}{\Delta x}(|u| + c) \leq 0.8 \times m$$

For the FRM, the timestep value has been set to "def" value, which means, for the solver used (explicit method), that the maximum timestep is 1 CAD. As you can see in Figure **11**, the FRM timestep in each case is 50 % larger than for the detailed model. Related to this point, the FRM, is characterized by a lower computational requirement, ensuring a good predictability since the predictive "DiPulse" combustion model is unchanged. Further computational time reduction can be obtained, for example for real-time application, by superimposing the combustion evolution traces.

3.2. Predictive Comparison

From the emission point of view, for more accurate prediction of pollutant emissions, to employ detailed chemical kinetics, more complex models are necessary, but this is not the purpose of the present work. Since the detailed Zeldovich mechanism is already implemented, and the in-cylinder temperature with a predictive model is quite well predicted, the NOx emissions are considered instead.

First, a comparison between the detailed model and the fast-running model is made. As can be seen in Figure **13** the FRM overestimates the NOx emissions. The differences lay in the fact that the in-cylinder conditions are slightly different between the FRM and detailed model, because, in the first case the intake thermodynamic conditions are constant for all the cylinders while in the second case they are calculated for each cylinder.

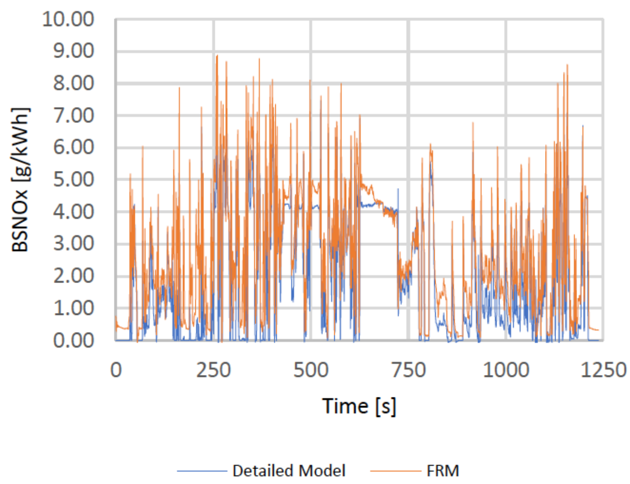


Figure 13: Brake Specific NOx Comparison.

This trend is confirmed by comparing cumulative NOx emission results over the NRTC (Figure **14**). This step is necessary to compare also the map-based model NOx emission results.

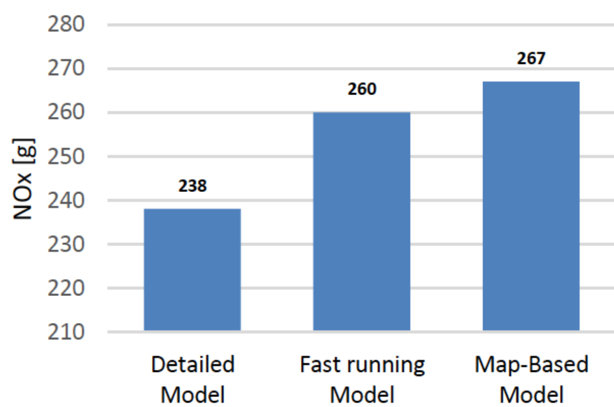


Figure 14: Cumulative NOx Emissions.

As can be seen, the map-based model, via interpolation, gives as cumulative NOx result 267 [g] that is about 10% higher than the value obtained via simulation with the detailed model. The result is satisfactory considering that a map-based model includes no chemical kinetics relationships. Figure **15** shows the comparison among the three models in terms of cumulative fuel consumption.

It can be noticed that the fuel consumption difference between FRM and detailed model is 0.7 %, the map-based and the detailed model one is 3.2%. On the base of these results, it can be stated that lower accuracy models are suitable for the trends analyses in terms of energy, NOx emissions, global and subsystems outputs, of complex powertrain systems.

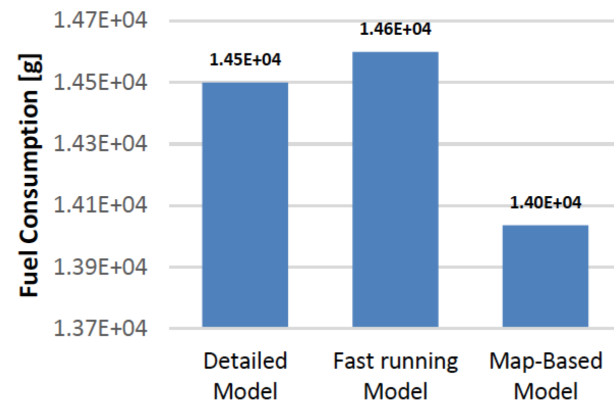


Figure 15: Cumulative Fuel Consumption.

3.3. Computational Comparison: Real Time Factor

In terms of computational effort of the three models the RT factor, defined as the ratio between the duration time of the simulation and the real-time of the physical event, is compared. In the case, the physical event is represented by the NRTC transient cycle with a total duration of 1238 seconds. The detailed model performed the NRTC cycle in 285522 seconds (≈79 hours) with a RT factor of about 230. The fast-running model performed the same cycle in 59312 seconds (≈17 hours) with a RT about of 48 while the map-based model in 1161 seconds with a RT of 0.9 (Figure **16**). It is worth to underline that the simulation ran on a workstation with an Intel® i7 processor @ 3.8 GHz.

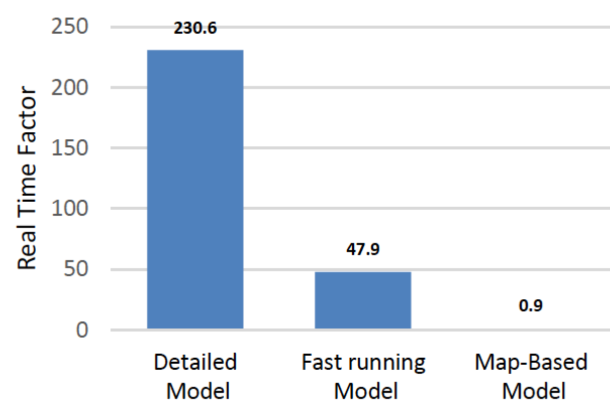


Figure 16: RT factor performing NRTC cycle.

In terms of simulation run time, the map-based model is roughly 250 times faster than detailed model and about 50 times faster than fast-running model performing NRTC transient cycle. In the case, the FRM characterized by a high predictivity in terms of simulation of the combustion process is not adequate for HiL applications where the speed simulation is a driving factor [18]. The map-based model is characterized by a RT factor near the unit, so the simulation duration is like the physical event. In this way the map-based model can be considered inexpensive in terms of computational requirements.

4. CONCLUSIONS

This work assesses the computational effort and simulation output differences among three different models, characterized by different level of detail of the same engine, over the NRTC transient homologation cycle. To achieve this goal, starting from a validated high fidelity 1D model, two simplified models were proposed, the FRM, a less detailed 1D model, and the map-based model, a non-dimensional model-built through operative maps to simulate engine behavior. In order to compare the three models, in terms of predictive accuracy and computational effort, the NRTC transient work cycle is implemented. Related to this point several simulations were carried out.

The simulation results of the study allowed to highlight several key aspects and conclusions:

- After the calibration procedure, FRM, despite losing some details due to the lumped volumes, maintains a good predictability with a lower computational effort compared to the detailed model;
- A qualitative analysis of the NO_x emission is carried out. It shows that the map-based model, with no chemical kinetic equations into it, achieves a good result in cumulative prediction compared to the detailed model; the difference ranges in the interval 0-10% with the high fidelity 1D model;
- In terms of cumulative fuel consumption prediction, map-based model shows comparable results with the other models listed, with a maximum difference of about 3% with the high fidelity 1D model;
- By comparing the three models in terms of computational speed, the map-based model is 250 times and 50 times faster compared to the detailed and FRM respectively.

The map-based model, despite its lower accuracy, shows good predictive results in terms of cumulative fuel consumption and cumulative NOx emissions over NRTC transient homologation cycle. Thanks to its low RT factor, it is suitable in applications where the computational speed is a driving factor, such as for complex hybrid architectures or HiL. In this regard, future activities are oriented in using the most adequate approach by evaluating the differences for Hybrid, Diesel-Electric, powertrain applications.

NOMENCLATURE

ABDC	After bottom dead center
ATDC	After top dead center
BBDC	Before bottom dead Center
BMEP	Brake-mean effective pressure
BSNOx	Brake-specific nitrogen oxides
BTDC	Before top dead center
CAD	Crank Angle Degree
CFD	Computational fluid dynamics
CO_2	Carbon dioxide
DMUs	Diesel Multiple Units
EEA	European Environmental Agency
EU	European Union
FRM	Fast Running Model
HD	Heavy duty
HiL	Hardware in the loop
IVC	Inlet valve closure
IVO	Inlet valve opening
NRMM	Non-Road Mobile Machinery
NRSC	Non-Road Steady Cycle
NRTC	Non-Road Transient Cycle
NOx	Nitric oxides
OEMs	Original equipment manufactures
PM	Particulate matter
RT	Real Time

APPENDIX

A.1 – Denormalization of NRTC Homologation Cycle

In the “denormalization” procedure the first necessary step is to evaluate the reference speed. This speed corresponds to the 100% normalized speed values specified in the engine dynamometer schedule, and it is close to the rated speed at which the engine delivers maximum power. As can be seen in Figure **3**, the maximum percentage in the engine dynamometer schedule of the normalised engine speed is not 100%, but 103%, in order to reach the rated speed. The reference engine speed is determined via following equation:

$$n_{ref} = n_{low} + 0.95(n_{high} - n_{low})$$

where:

n_{ref} is the reference engine speed $[1 \cdot \mathrm{min}^{-1}]$;
n_{high} is the high engine speed (the highest engine speed where 70% of rated power is delivered) $[1 \cdot \mathrm{min}^{-1}]$;
n_{low} is the low engine speed (the lowest engine speed where 50% of rated power is delivered) $[1 \cdot \mathrm{min}^{-1}]$.

Once the reference speed has been calculated, it is possible the denormalization of the engine speed with the following equation:

$$n_{sk} = \frac{\%Speed \cdot (n_{ref} - n_{idle})}{100} + n_{idle}$$

where:

n_{sk} is the actual engine speed at a given point of the test cycle $[1 \cdot \mathrm{min}^{-1}]$;
$\%speed$ is the normalised engine speed at a given point of the test cycle [%];
n_{idle} is the engine idle speed $[1 \cdot \mathrm{min}^{-1}]$.
Actual engine torque at different points within the test cycle is determined by means of equation:

$$T_{sk} = \frac{\%Torque \cdot T_{Max}}{100}$$

where:

T_{sk} is the actual engine torque at a given point of the test cycle $[Nm]$;
$\%torque$ is the normalised engine torque at a given point of the test cycle [%];
T_{max} is the maximal engine torque at a given engine speed $[Nm]$.

ACKNOWLEDGMENT

Authors would like to thank Ianora Stefano and Schiavo Mariano that contributed to this study with their thesis works.

REFERENCES

[1] A. p. Schiene, «Deutschland bei Bahn-Elektrifizierung nur Mittelmaß» [Online]. Available: https://www.allianz-pro-schiene.de/presse/pressemitteilungen/2012-019-elektromobilitaet-deutschland-bei-bahn-elektrifizierung-mittelmass/.

[2] Report: "Energy savings with hybrid locomotives on TEN-T corridors", 2015.

[3] N. Sonnichsen, «Statista» [Online]. Available: https://www.statista.com/statistics/451522/share-of-the-rail-network-which-is-electrified-in-europe/. [Consultato il giorno 30 Luglio 2019].

[4] European Commission, Mobility and Trasport: Electrified railway lines» [Online]. Available: https://ec.europa.eu/transport/facts-fundings/scoreboard/compare/energy-union-innovation/share-electrified-railway_en. [Consultato il giorno 30 Luglio 2019].

[5] D. Holger, H. Dirk e U. Jörg, «Reducing DMU fuel consumption by means of hybrid energy storage» European Transport Research Review, 2011.

[6] J. Norris, L. Ntziachristos, Z. Samaras e KH. Zieroch, «Air pollutant emission inventory guidebook: Railways» European Environment Agency (EEA), 2016.

[7] T. Dalmann e A. Menon, «Technology Pathways for Diesel used in Non-Road Vehicles» Internation Council on Clean Transportation, 2016.

[8] EU: Non Road Engines» [Online]. Available: https://www.dieselnet.com/standards/eu/nonroad.php.

[9] Z. Gao, JC. Conklin, CS. Daw e VK. Chakravarthy, «A proposed methodology for estimating transient engine - out temperature and emissions from steady - state maps» International Journal of Engine Research 2010; 11(2): 137-151. https://doi.org/10.1243/14680874JER05609

[10] RJ. Brooks e AM. Tobias, «Choosing the best model: Level of detail, complexity, and model perfomance,» Mathematical and Computer Modelling 1996; 24(4): 1-14. https://doi.org/10.1016/0895-7177(96)00103-3

[11] F. Millo, L. Rolando e M. Andreata, «Numerical Simulation for Vehicle Powertrain Development,» Numerical Analysis - Theory and Application 2014; pp. 519-540.

[12] D. Cieslar, A. Darlington, K. Glover e N. Collings, «Model based control for closed loop testing of 1-D engine simulation models» in Workshop on Engine and Powertrain Control, Simulation and Modeling, Rueil - Malmaison 2012. https://doi.org/10.3182/20121023-3-FR-4025.00005

[13] A. Gimelli, M. Muccillo e O. Pennacchia, «Study of a new mechanical variable valve actuation system: Part II-Estimation of the actual fuel consumption improvement through one-dimensional fluid dynamic analysis and valve train friction estimation» International Journal of Engine Research 2015; 16(6): 762-772. https://doi.org/10.1177/1468087415604095

[14] JC. Wurzenberger, R. Heinzle, MV. Deregnaucourt e T. Katrasnik, «A comprehensive study on different system level engine simulation models» 2013. https://doi.org/10.4271/2013-01-1116

[15] AG. Konstandopoulos, M. Kostoglou, C. Beatrice, G. Di Blasio, A. Imren e I. Denbratt, «Impact of Combination of EGR, SCR, and DPF Technologies for the Low - Emission

Rail Diesel Engines» Emission Control Science and Technology 2015; 1: 213-225. https://doi.org/10.1007/s40825-015-0020-0

[16] Gamma Technologies, Vehicle Driveline and HEV Application Manual, 2016.

[17] Gamma Technologies, Flow Theory Manual 2016.

[18] H. Wu e MF. Li. A Hardware in-the-loop (HIL) Bench Test of a GT - Power Fast Running Model for Rapid Control Prototyping (RCP) Verification» SAE Technical Paper 2016; 2016-01-0549. https://doi.org/10.4271/2016-01-0549

Received on 16-12-2019 Accepted on 14-1-2020 Published on 17-3-2020

DOI: https://doi.org/10.12974/2311-8741.2020.08.6