A challenging future virtualization technology in IC engine development focusing on HEV application

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Abstract- This paper involves the brief literature review on virtualization technology in IC engine development. The first part of the paper covers the various previous research carried on the virtual volume, pressure,torque, and emission sensor technology for understanding and analysis purpose. This part of research uses ADVISOR package as pre-requisite for understanding the parameters effect such as compression ratio, pressure, & temperature on thermal NO_x production in diesel engine configuration. In second part, a thermodynamic model for virtual NO_x sensor which is capable of predicting nitrogen oxides (NO_x) emissions in diesel engines is carried out. In-cylinder volume and pressure are the inputs from estimator. The proposed method is based on a thermodynamic model. The contribution of proposed method arises from: In-cylinder volume and pressure are the inputs from estimator using slider crank mechanism andthe thermodynamic modelling framework is applied in a way that the thermodynamic equations can be solved in a closed form, which provides the basis for achieving high level of predictiveness using MATLAB. The model is validated using real data values. Themodel is suitable for two different direct injection diesel engines, i.e. a light and a heavy-duty engine.

Keywords –Virtualization, Compression Ratio, Thermodynamics, NO_x (Nitrogen Oxide), Thermal NO_x, MATLAB, ADVISOR (Advanced Vehicle Simulator), Estimator, Two-Zones (Burned and Unburned Gas Zone).

I. INTRODUCTION

Diesel engine powered automobiles are increasing in recent years, due to high thermal efficiency of diesel engines. These engines have become an environmental issue since it represents a major source of NO_x emissions [1]. Advanced Vehicle Simulator (ADVISOR) can be used for Vehicle system problem analysis. It was developed at national renewable energy laboratory in November 1994. It was designed to assist the US department of energy (DOE) for analysis purpose of hybrid electric vehicles (HEVs) through the HEV propulsion system contracts with General motors, Daimler Chrysler and Ford motor. ADVISOR package acts as pre-requisite for understanding the exhaust gas temperature, exhaust catalytic efficiency and emissions for a selected vehicle configuration. ADVISOR package runs in background of MATLAB environment, which provides a backbone for simulation and its analysis for the defined drivetrain configuration [2,11].Automobiles equipped with diesel powered engines have been increasingly used and highlighting from last decades. The reason for the higher usage of these engines is because of its higher thermal efficiency. Also, the increasingly burning of fuel taking place in these automobiles increases the engine out emission such as CO, NO_x, HC, soot and particulate matter. According to world attention of environmental protection has allowed a strict law for these emission pollutants. Focusing on this well-known deficiency area it is therefore defined development in technology as prior.In this sense, the European norms (EURO-2) restrict the pollutants for NO_x and PM for union mobile source of application. 20% of emissions in light-duty diesel engines must reduce with EURO-5 and Euro-6 focusses on 50% reduction in comparison with EURO-4[5]. The modern engines use the specially designed device called as exhaust gas recirculation (EGR) to minimize the pollutants.The estimation of in cylinder engine $NO_x$with respect to crank angle adds huge value. The NO_x data can be used for direct control strategies. The RT based models of NO_x sensor can be used in EURO-6 which restricts the amount of harmful pollutants formed while burning fuel.[6,8] The important parameter required for thermal NO_x prediction are rate of heat release (ROHR) and relevant temperature with species concentration in the combustion chamber or interested zones of combustion chamber. Emission norms are always request for development in the area of emission for engine management and after-treatment system. Since these systems are complex and more expensive the virtual sensors technology finds a best opportunity for the development of such systems of diesel engines. Onboard diagnostic and future legislation of emission require optimized feature with respect to engine control module and after-treatment systems [9].

The diesel engine has higher formation of NO_x which adds attention for the development of NO_x estimation. During calibrating engines, the combustion optimization plays important role which leads to low engine out emissions, also it acts as pre-requisites for a robust working of exhaust after-treatment system. The NO_x estimation at upstream and downstream also have advantage for estimating SCR efficiency which involves urea dosing. Current technology in vehicles uses NO_x sensor mounted before the NO_x after-treatment system. Further, with the development of model based mathematical equations can be implemented with ECU which allows the diagnosis physically the engine out NO_x and considering cost reason the upstream of SCR is not mounted with NO_x sensor. The physical sensors have disadvantages like they lose their accurate measurement after aging and they should depend on the calibration effort throughout their complete lifetime [10]. Therefore, to overcome these difficulties of physical NO_x sensor robust model should be developed with good prediction accuracy for NO_x estimation which can replace physical NO_x sensor. The requirement for such development with aspect of monitoring emissions requires the use of sensors, model-based concepts or both. In case of dynamic response instance, the sensors with onboard characteristics are limited and thus there is necessity models. The research area undergoing in this paper that is after-treatment system has a major topic for several decades and this is also key feature for next generation IC engines [12].

Virtualization is basically abstraction of one or more computer resources to achieve the behavior of the desired system. The simulation is the desired system here or testing platform for a different architecture etc. In the beginning, the term emulation was usually used for virtualization using hardware. However, the term emulation in the software context is commonly used. The hardware emulation refers to the virtualization of the required hardware on the host machine. Here, we considered the case of NO_x sensor virtualization. Such a 'Virtual NO_x sensor' can be used to move development tasks to PC, where they can often perform, faster, cheaper or better. To overcome these increasingly stringent emissions in diesel engine adoption of control strategies based on models is required. A real time-based NO_x model can be able to estimate engine out NO_x in diesel engines.

Application of fast emission model is for engines fitted with a NO_x after treatment system that adds a reduction agent to the exhaust stream. In that case it is essential to know engine out NO_x level in order to control the reduction agent flow into the catalytic converter [3]. Some of the modern diesel engines are equipped with pressure sensors (Transducer), which gives input to combustion related data [5]. The importance of measuring in-cylinder pressure used for obtaining information related to aging of engine components as well as adaption of the injection strategy with respect to the variation in the fuel quality [6]. Thus, focusing on this importance of pressure estimator, it can be useful for emerging a pressure estimator model. The research uses input for the combustion modelling and input for calculation of thermodynamic parameters inside the cylinder which enables virtual NO_x sensor with higher level of predictiveness. Over the past decade, diesel engine technology has increasing rapidly and evolved to the point where control solutions for obtaining high fuel efficiency and low NO_x emissions are possible. The strategies are being implemented to manage fueling system and air handling system. Also, this will be relying on sensor replacement and limitations of costs. A space of special interest is considered for this type of analysis inside the cylinder volume. The laws of thermodynamics are being an application for solving this kind of equations [7]. According to piston-cylinder geometry it is seen that straight-line motion can be converted into rotary motion. Hence the slider crank model has led to derive in-cylinder volume. The available piezoelectric transducer will lose its sensibility due to aging factor. Hence this allows for development in grey area of an automobile [6]. The model accuracy largely depends on tuning parameter. Hence, advanced virtual NO_x sensor models have general applicability for NO_x estimation [3]. The estimation of NO_x at upstream and downstream opens a capability for getting SCR efficiency for further urea dosing strategy. As per the research, the main thermodynamic parameters require for NO_x prediction are rate of heat release (ROHR) and consequently the temperatures and concentrations in the combustion chamber. The properties of thermodynamic model can be assessed by physical depth in models which also reflects computational complexity of models [3]. The cylinder out NO_x emissions can be done with state estimation which reduces the sensor cost for the after-treatment NO_x control systems based on SCR [1].

II. Proposed workflow and methodology

A. *Proposed workflow –*

The workflow starts with the identification of area of interest for research and development under after-treatment system of an automobile. The grey area is identified for further development and then analysis is carried on to understanding the exhaust gas temperature, exhaust catalytic efficiency and emissions for a selected vehicle configuration using ADVISOR package. ADVISOR package runs in background of MATLAB environment, which provides a backbone for simulation and its analysis for the defined drivetrain configurationthe modelling framework starts with estimation of in-cylinder volume which is obtained using slider crank mechanism. The instantaneous volume can be obtained by applying derivative. The estimation of in-cylinder volume is further used for instantaneous pressure estimation. The instantaneous values are inputs from estimator. The measurement of temperature for considered ideal and non-perfect gas is followed. The main concentration is given for deducting temperature out of mass of fuel, heat addition terms using thermodynamic modeling for thermal NO_x calculation. After the combustion begins and burns the charge it led to formation of NO_x. Thus, applying control volume concept on the combustion zone. Figure 1 represents the workflow of the model.

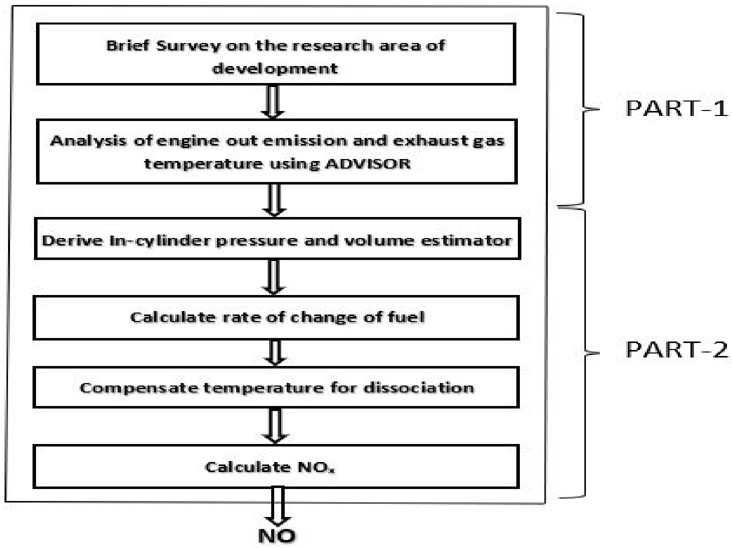


Fig 1: Work flow for calculating NO_x

B. Proposed Methodology –

The proposed methodology is shown in figure 2, which indicates the Overall view of models used for modelling the NO_x estimator. The following are the models which are modelled as per the engine specification: Volume Estimator model, Pressure estimator model, Fuel rate model, Heat release model, Heat transfer model, Temperature model for thermal NO_x calculation, and NO_x estimator model.

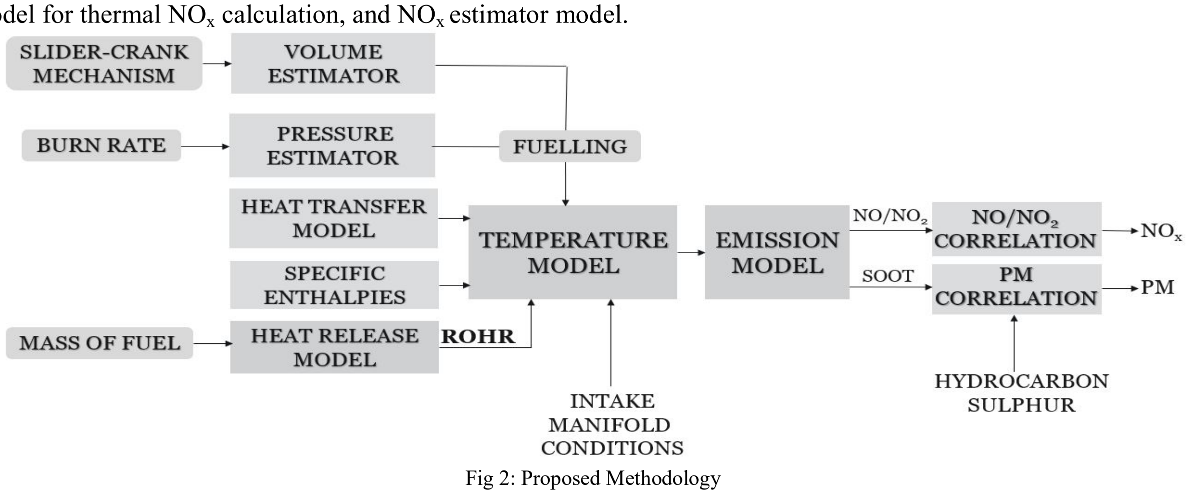


Fig 2: Proposed Methodology

III. Simulation of NO_X sensor

1. *Volume Estimator model*

It is assumed that volume can be derived using slider crank model from the compression ratio, bore, connecting rod and stroke. The energy equations are applied to the volume and crank-angle domain and the volume estimator obtained as given in equation 1 [17]:

$$\frac{dV}{d\theta} = A_p\left[(-r\sin\theta)\left(1 + \frac{\cos\theta}{\sqrt{\left(\frac{L_c}{r}\right)^2 - \sin^2\theta}}\right)\right] \quad (1)$$

2. *Pressure estimator model*

The energy equations are applied to the pressure and crank-angle domain and the pressure estimator derived as follows [7]:

$$\frac{dP}{d\theta} - -\frac{\gamma}{V_{cyl}}\frac{dV}{d\theta}P_{cyl} + \frac{\gamma - 1}{V_{cyl}}\left[(1-\alpha)m_{fuel}Q_{LHV}\frac{dx_b}{d\theta}\right] \quad (2)$$

3. Model of conserved mass in combustion chamber

The Conservation of mass is applied to the control volume and conservation of mass of burnt is equal to the mass transfer from unburnt to burnt zone, accord into this assumption it is given as[3]:

$$\frac{dm_{u\to b}}{d\theta} = \frac{dm_{fuel}}{d\theta}\frac{(\lambda_{comb}L_{st} + 1)(1 + \lambda_u L_{st})}{L_{st}(\lambda_u - 1)} \quad (3)$$

Thus, the conservation of mass of burnt is obtained as[3]:

$$\frac{dm_b}{d\theta} = \frac{dm_{u\to b}}{d\theta} \quad (4)$$

4. Heat release model

It is assumed that the rate of heat release occurs and can be obtained by heat addition parameter with lower heating value for diesel fuel as follows [12]:

$$\frac{dQ_{b,b}}{d\theta} = \frac{dm_{fuel}}{d\theta}Q_{LHV} \quad (5)$$

5. Heat transfer model

The total heat transfer between the in-cylinder charge and the combustion chamber walls is estimated by means of convective formulation [13,16]:

$$\frac{dQ_{ht}}{d\theta} = \frac{1}{6n}\alpha_{ht}[A_p(T_p - T_{cyl}) + A_l(T_l - T_{cyl}) + A_h(T_h - T_{cyl})] \quad (6)$$

The mean temperature in the cylinder (T_{cyl}) is for the purpose of heat transfer sub-model obtained from the ideal gas law. In the analysed case, the woschni correlation was applied to determine the heat transfer coefficient [13,12]:

$$\alpha_{ht} = 130\, D^{-0.2}p^{0.8}T_{cyl}^{-0.53}\left[\left(2.28 + 0.308\frac{v_u}{v_m}\right)v_m + 0.00324\frac{V\,T_{IVC}}{p_{IVC}V_{IVC}}(p - p_{IVC})\right]^{0.8} \quad (7)$$

5. Temperature model for thermal NO_x calculation

The temperature responsible for thermal NO_x is calculated using following equation and it takes variation of pressure, mass of fuel, heat transfer and ideal gas parameters [3,12]:

$$\frac{dT_b}{d\theta} = \frac{\frac{dQ_{b,B}}{d\theta} + \frac{dQ_{ht,b}}{d\theta} + h_u\frac{dm_b}{d\theta} - h_b\frac{dm_b}{d\theta} + \frac{m_b R_b T_b}{p}\frac{dp}{d\theta}}{m_b R_b} \quad (8)$$

6. NO_x estimator model

The equilibrium species calculation is being performed using Gibbs free energy equation. To reduce the complexity for computation. The pre-calculated values are being used as stored look-up table. The calculation of equilibrium species can be performed for many sets of reactions using extended zeldovich mechanism. Here the case of one reaction is considered for modelling basic NO_x estimator[16].

$$N_2 + O \rightarrow NO + N$$

The extended zeldovich mechanism is obtained for the above reaction and initial NO formation rate is given as[16]:

$$\frac{d[NO]}{d\theta} = \frac{6 * 10^{16}}{T_b^{1/2}} e^{\frac{-69{,}090}{T_b}} [O]_e^{0.5} [N_2]_e \tag{9}$$

Now The formation of NO_x was modelled for six reactions [1,3,14] according to the extended Zeldovich mechanism described by the following steps:

R1: $N_2 + O \leftrightarrow NO + N$
R2: $O_2 + N \leftrightarrow NO + O$
R3: $OH + N \leftrightarrow NO + H$
R4: $N_2O + O \leftrightarrow NO + NO$
R5: $O_2 + N_2 \leftrightarrow N_2O + O$
R6: $OH + N_2 \leftrightarrow N_2O + H$

Even though the NO_2 formation in compression ignition engines cannot be neglected in the described model, approximation on NO_x formation was made only on calculation of NO formation. Similar as in NO formation and NO_x emissions are assumed to be correlated and one constant calibrating factor was applied on the final cycle NO_x mass to match the experimental tailpipe NO_x emissions. The rate of formation of NO in the burned zone is therefore derived by the following equation [3,12,15]:

$$\frac{d[NO]}{d\theta} = \frac{2}{6n} C_{NO,multi} (1 - \alpha_{Nc}^2)[\frac{k_1 C_{e,N_2} C_{e,O}}{1 + \alpha_{NO} V_2} + \frac{V_4 C_{e,N_2O} C_{e,O}}{1 + V_4}] \tag{10}$$

Where is the ratio of actual NO concentration and its equilibrium concentration calculated according to equation58 and n is the engine speed and $C_{NO,multi}$ is multiplicative parameter that also takes into account difference between predicted NO formation and actual NO_x formation [3].

$$\alpha_{NO} = \frac{C_{NO}}{C_{e,NO}} \tag{11}$$

Furthermore, C_e represents equilibrium species concentration read from the look-up table. V_2 and V_4 are help variables calculated according to Equation (12&13):

$$V_2 = \frac{k_1 C_{e,N_2} C_{e,O}}{k_2 C_{e,O_2} C_{e,N} + k_3 C_{e,OH} C_{e,N}} \tag{12}$$

$$V_4 = \frac{k_4 C_{e,N_2O} C_{e,O}}{k_5 C_{e,O_2} C_{e,N_2} + k_6 C_{e,OH} C_{e,N_2}} \tag{13}$$

IV. SPECIFICATION OF WORK

The specifications used for modelling is given below. The simulation is carried on light duty vehicle and heavy-duty vehicle. Table 1 provides specification of light-duty vehicle. Similarly, Table 2 provides specification of heavy-duty vehicle.

Table 1: Light duty vehicle specificationTable 2: Heavy duty vehicle specification

Cylinders	**4 inline**
Displacement	1560Cms
Bore x Stroke	75mm x 88.3mm
Compression	18:1
Fuel injection system	CRDI
Connecting rod length	136.8mm
Maximum power	66.2kW @ 4000 rpm
Maximum Torque	215Nm @ 1750 rpm

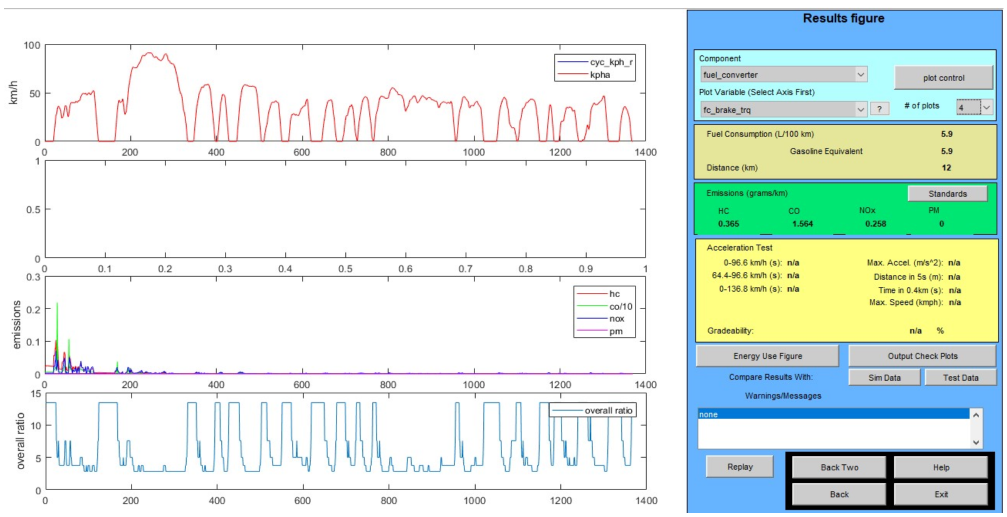
Cylinders	6 inline
Displacement	6870cm^3
Bore x Stroke	108mm x 125mm
Compression	18:1
Fuel injection system	DI
Connecting rod length	182.5mm
Maximum power	162kW @ 2400 rpm
Maximum Torque	825Nm @ 1400-1700 rpm

V. RESULT AND DISCUSSION

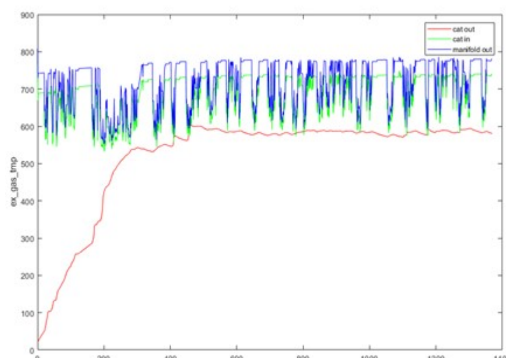
I. Analysis outcome using ADVISOR package (PART-1)

A. Result of conventional vehicle

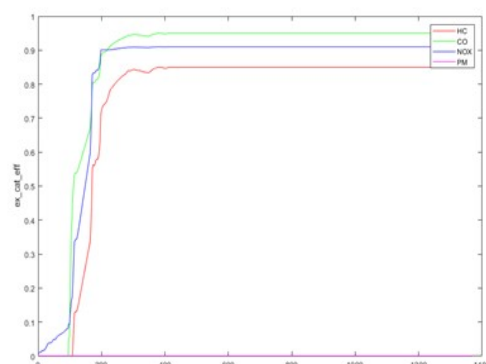
The result of speed versus time is shown in figure 3(a)plot 1, indicates the UDDS drive cycle. The figure 3(a) plot 3shows emissions for hydrocarbon (HC=0.965g/km), carbon monoxide (CO=1.564g/km), nitrogen dioxide (NO_x=0.258g/km), and particulate matter (PM=0). Figure 3(a)plot 4, shows the overall ratio of emissions and drive cycle, which is average value of UDDC and gas emissions. Figure 3(b) shows the exhaust gas temperature, Figure 3(c) exhaust catalyst efficiency.



(a)



(b)

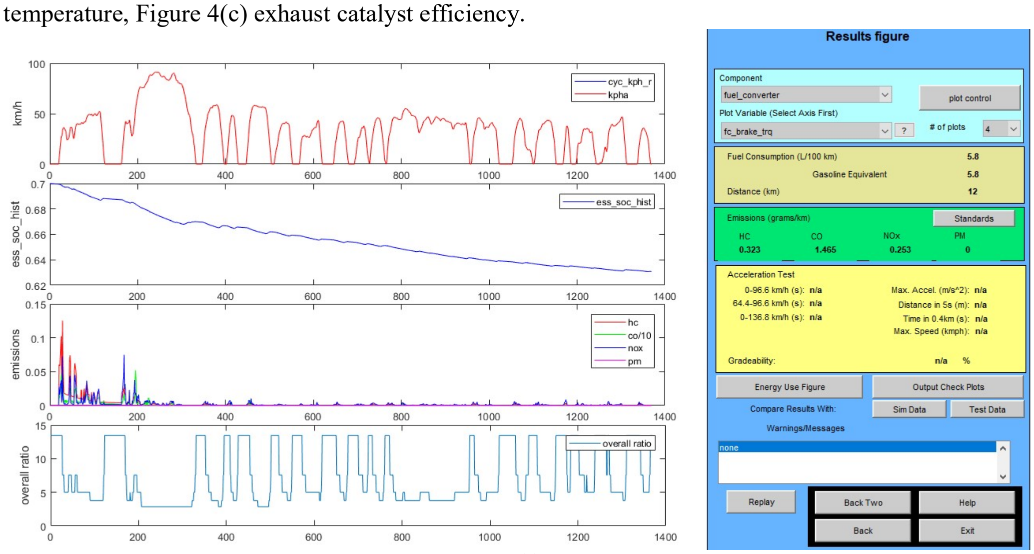


(c)

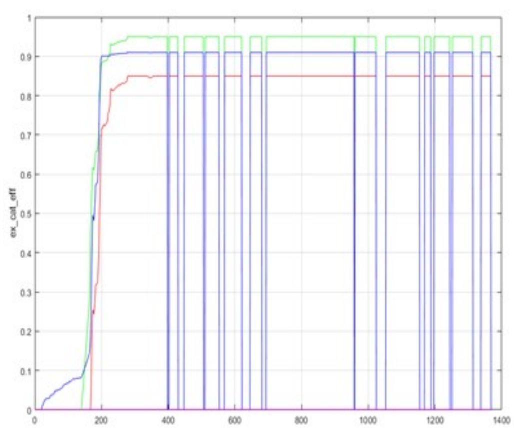
Fig 3: Conventional vehicle result (a)ADVISOR result window (b) ExhaustGas Temperature (c) Exhaust Catalyst Efficiency

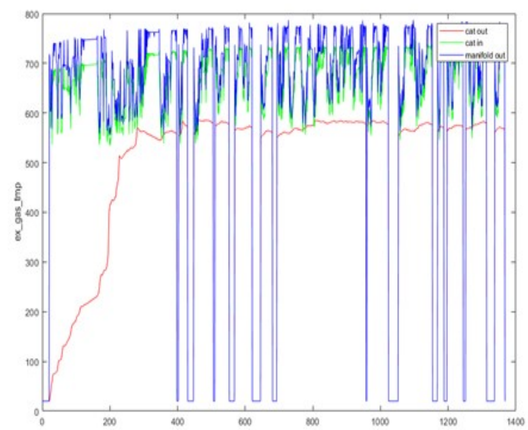
B. *Result of hybrid electric vehicle (HEV)*

The result of speed versus time is shown in figure 4(a)plot 1, indicates the UDDS drive cycle. Figure 4(a) plot 2 shows, battery state of charge (SOC) since it's a combination of IC engine and battery configured. The figure 4(a) plot 3 shows emissions for hydrocarbon (HC=0.323g/km), carbon monoxide (CO=1.465g/km), nitrogen dioxide (NO_x =0.253g/km), and particulate matter (PM=0). Figure 4(a)plot 4, shows the overall ratio of emissions and drive cycle, which is average value of UDDC and gas emissions. Figure 4(b) shows the exhaust gas temperature, Figure 4(c) exhaust catalyst efficiency.



(a)





(b) (c)

Fig 4: Hybrid electric vehicle result (a)ADVISOR result window (b) ExhaustGas Temperature (c) Exhaust Catalyst Efficiency

II. SIMULATION RESULT OF NO_X SENSOR (PART-2)

The simulation of NO_x estimator is done for light-duty vehicle and heavy-duty vehicle. The mass of air, mass of fuel, engine specifications are different to simulate on two different engines. Considering these factors, the simulation is carried in MATLAB platform.

A - Simulation result of light-duty vehicle

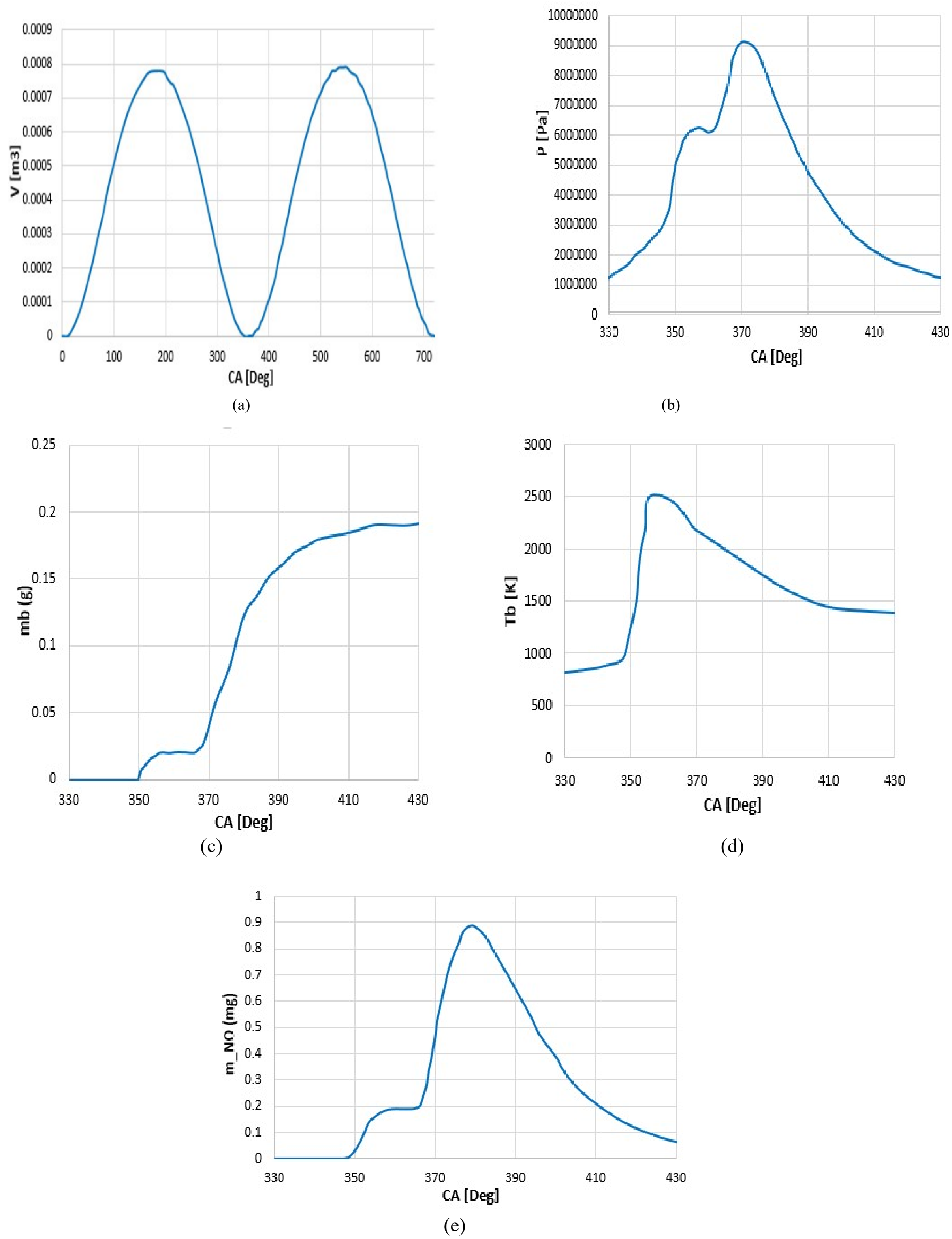


Fig 4: Simulation result of Light-duty engine operating point at N=3000rpm
(a) Volume estimator, (b) Pressure estimator, (c) Mass of burnt, (d) Burnt zone temperature, (e) NO_x estimator.

LD ENGINE: The light-duty engine is equipped with common rail direct injection system. The build in pressure for the light-duty engine is less comparative to heavy-duty engine. Sensible check is carried and analyzed the plots of mass of burnt which is getting increased as combustion begins with respect to provided combustion angle. The combustion temperature starts increasing rapidly at same instant, it also considers heat transfer. Thus, the NO formation starts at same angle for the cycle.

B - Simulation result of Heavy-duty vehicle

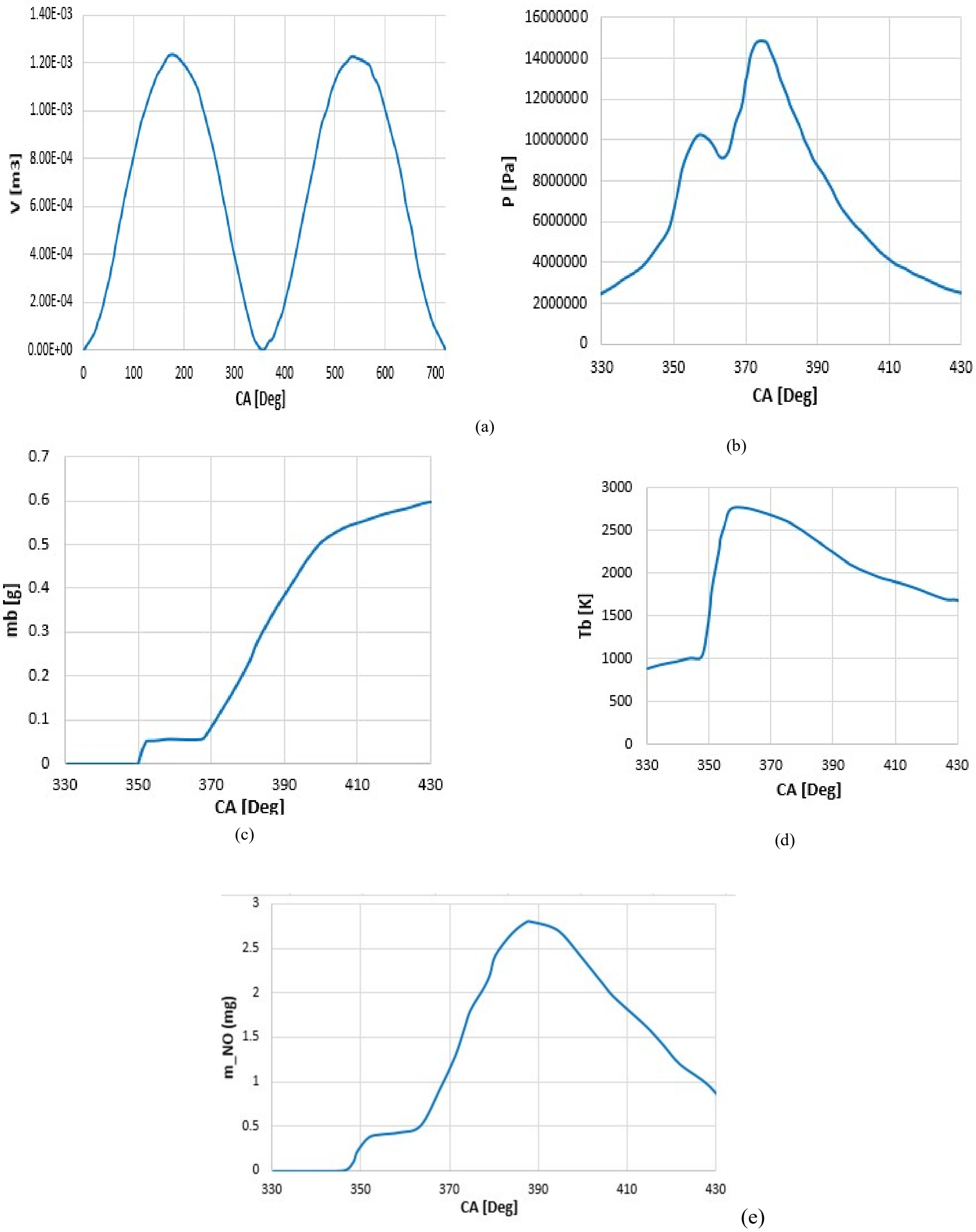


Fig 4: Simulation result of Heavy-duty engine operating point at N=3000rpm

(a) Volume estimator, (b) Pressure estimator, (c) Mass of burnt, (d) Burnt zone temperature, (e) NO_x estimator.

HD ENGINE: The heavy-duty engine is equipped with direct injection fuel injection system for validation purpose which influences the air and fuel preparation, also the HD-engine operates with leaner mixture compare to LD-engine. Thus, it impacts on start of combustion temperature and concentration. The build in pressure for the heavy-duty engine is very high. Sensible check is carried and analyzed the plots of mass of burnt which is getting increased as combustion begins with respect to provided combustion angle. The combustion temperature starts increasing rapidly at same instant, also it considers heat transfer. Thus, the NO formation starts at same angle for the cycle.

VICONCLUSION

The literature review carried in the area of emission and analysis is carried using ADVISOR package. Then NO_x sensor estimator is modelled for diesel engine which is able to measure engine out NO_x using thermodynamic modelling framework which is usually measured to maintain low tail pipe emissions. In-cylinder pressure and volume are obtained successfully using slider crank mechanism, and input from pressure and volume estimator is used for calculating fuel rate. Finally, the temperature is obtained which estimates engine out thermal NO_x. A thermodynamic modelling is carried in such a way that, it gives the mass of burnt and its temperature which signifies the amount of thermal NO_x produced in diesel engine using MATLAB. The application of model on two different injection engines is validated, which allows to obtain NO_x in heavy-duty engine which produces slighter higher amount of NO_x emission compare to light-duty engine as mentioned in equation (9& 10). The reason for NO_x production is due to higher inlet temperature for this purpose equation (8) is modelled. The pressure estimator is modelled to correct the inlet pressure. Thus, by increasing inlet pressure the lower NO_x can be obtained. It also seen that increase in compression ratio increases pressure which effects the combustion temperature to a higher value which produces higher NO_x. The future work needs to be carried out to maintain low emissions while minimizing fuel consumption. Thus, it limits harmful gases from internal combustion engine.

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ABBREVIATIONS

Q_{lhv}	Lower heating value in J /kg
T	Temperature in K
V	Volume in m^3
x	Mass fraction
N_e	Engine speed in RPM
ω	Rotational speed Rad /s
θ	Crank angle in degrees
γ	Ratio of specific heats
m_{air}	Mass of air in g
m_{fuel}	Mass of fuel in g
λ	Air-Fuel ratio
b	Bore in m
s	Stroke in m
l_c	Connecting rod length in m
r	Crank radius (= ½ s) in m
A_p	Area of the piston in $m^2 = D_p^2/4$,
n	Polytropic coefficient
Q_g	Apparent gross heat release
Q_{ht}	Heat transferred to the cylinder wall
V_{cyl}	Cylinder volume in m^3
P_{cyl}	Cylinder volume in bars/pascal
α	Heat transfer scaling factor
$\frac{dx_b}{d\theta}$	Burn rate
m	Mass in g
m_b	Mass of burned charge in g
V_b	Volume of burned charge in m^3
or	or burnt
λ_u	Air – Fuel ratio of unburnt
L_{st}	Stoichiometric ratio
p_{air}	Air pressure in bars / pascals
V_{cyl}	Cylinder volume in m^3
R_b	Characteristic gas constant of burnt gas in J/g-K
Q	Total energy crossing boundary as heat in J
W	Total energy crossing boundary as work in J
A_p	Surface area of the piston in m^2
A_l	Surface area of the liner in m^2
A_h	Surface area of the cylinder head in m^2
α_{h1}	Heat transfer coefficient

D	Cylinder bore in m
V	Actual cylinder volume in m^3
V_u	Circumferential velocity in m^3
V_m	Mean piston speed in m/s
T_{cyl}	Cylinder temperature in K
T_l	Liner temperature in K
T_p	Piston temperature in K
T_h	Cylinder
T_{IVC}	Temperature at intake valve closing (IVC) in K
P_{IVC}	Pressure at intake valve closing (IVC) in Pa/Bars
V_{IVC}	Cylinder Volume at intake valve closing (IVC) in m^3
N	Nitrogen
N_2	Dinitrogen
O	Oxygen
O_2	Dioxygen
H	Hydrogen
NO	Nitric oxide or Nitrogen monoxide
NO_2	Nitrogen dioxide
N_2O	Nitrous oxide or dinitrogen oxide or dinitrogen monoxide
OH	Hydroxide

SUBSCRIPTS & SUPERSCRIPTS

b	burned zone
comb	combustion
cyl	cylinder
e	engine
IVC	Inlet valve closing
EVO	Exhaust valve opening
c	Connecting rod
p	Piston
l	Liner
h	Cylinder head
Tot	total
Pilot	Pilot injection
Main	Main injection
ht	Heat transfer
lhv	Lower heating value
fuel	Fuel
st	Stoichiometric
2	Two atoms
SOC	Start of Combustion

lam	Lambda

ACRONYMS

LD	Light duty vehicle
HD	Heavy duty vehicle
TDC	Top dead centre
BDC	Bottom dead centre
AFR	Air-Fuel Ratio
WLTC	Worldwide harmonized Light-duty Test Cycle test
SCR	Selective Catalytic Reduction
HEV	Hybrid Electric Vehicle
NEDC	New European Driving Cycle
ROHR	Rate of Heat Release
ECU	Engine control unit
ID	Ignition delay
SOI	Start of Injection
SOC	Start of Combustion
IVC	Intake valve closing
EVO	Exhaust valve opening
ADVISOR	Advanced VehIcle SimultOR
SCR	Selective catalytic reduction
EGR	Exhaust gas recirculation