State-Space Formulation for a CNG Converted Diesel Engine Variables'

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Abstract: ***The conversion of diesel engine into compressed natural gas converted engine provides short term measure* in *mitigating the rising fuel cost, minimizing the emission and maintaining the current engine performance based on the market demand. The compressed natural gas converted engine is* done *successfully from various types of diesel engines with various engine displacements and variables.* The *conversion kits are commercially off the shelf items which include sensors and pressure regulators. These components are integrated into a working system. This is the most practical approach as the components come from various sources which are limited in their availability. Although the system works but it does not functioning at the optimum condition. Therefore, a robust control algorithm with the consideration of inputs from each engine variables and external disturbances are designed and introduced to control an ignition compression engine which is converted from diesel engine. This converted diesel engine is fuelled with compressed natural gas (CNG), and this engine is to be called CNG Converted Diesel Engine (CCDE). The proposed robust control algorithm is developed based on State-Space Equations from each engine variables. This paper presents on how state-space equations are formulated.***

Keywords*:* engine variables, system modelling, time variant, linearization, state-space equation

1. Introduction

The engine conversion technology from compression-ignition (CI) engine to Spark Ignition (SI) engine has to be commercially viable. The developed technology has to double the life cycle and reliability of the engine. The trade-off between product quality, production cost and conversion flexibility are decisive for the success of the commercialization program and subsequent mass production. The engine performance variability is quite noticeable in the fleet operation. **Fig-1** shows the first working prototype powered by a CCDE. A robust control system must be able to cope with all the modification on the system parameter variations and external disturbances which affects the engine behaviour. This behaviour of the engine sub-systems are described by mathematical model from its state-space formulation for each engine variables. The overall engine process are segregated into three important sub-systems' dynamics, which are the fuelling dynamics, manifold filling dynamics and crank shaft speed dynamics. Then, each process is described as state-space model which are formulated in designing the engine control algorithm.

Figure 1: The First Working Prototype which is powered by a CNG Conversion Diesel Engine – Engine Type: HICOM PERKASA MTB150 Engine, Model 4HF1, 4,300cc.

The mathematical model is not an accurate representation of the actual system. Each of the sub-system models is built on estimation and approximation of nonlinear time dependent variables. However the established and widely used mathematical tools are developed for linear systems. Non linear system is represented by partial differential equations which are difficult to handle. For this reason, the CCDE is approximated as time invariant linear systems.

2. Objective

This paper establishes the methodology to develop the mathematical models of the CCDE components and identify the state variables of the engine for the control model then to establish the state-space equations for the whole system. The control loop shaping processes using closed-loop interconnection flow modelling is also shown.

3. System Modelling

State Variables Identification - The state variables for the system modelling are identified as parameter-dependent systems and two-dimensional nonlinear functions, which maps the model parameters over the operating envelope. The dynamic systems have linear state inputs/outputs but the system states are parameters dependent and exogenous variables that are tabulated as a map. The construction of a dynamics model for each state variable is formulated base on the schematic diagram as shown in Fig-2. For example, the dynamics of the crankshaft are not modelled explicitly as it is considered to be quasi-stationary due to the large inertia, in comparison to the air and CNG flow processes. Therefore engine speed is considered to be a slow-varying exogenous variable. Even though the dynamics of the crankshaft are not part of the model, the model is still too large and too complex to be identified as a whole. In spite of the complex interactions between the components of the model, they are therefore treated separately for the purposes of system identification. The way to move forward is to use a simple approach to identify the individual variables using standard system identification techniques for MIMO engine systems from each sub-system variables.

Engine Elements - The internal engine system variables and engine dynamics is divided into five (5) basic sub-systems for the purpose of modelling the state variables (Guzzella & Onder, 2010, p. 25).

1) The internal energy in the inlet and outlet manifold
2) The air flow through the throttle valve and intake manifold,
3) The injection of fuel through the injector and wall wetting,
4) The exhaust emission which includes the vehicle and combustion delays, the gas mixing and the emission sensor, and
5) The torque output which includes the induction to the power stroke delays and several nonlinearities.

A CCDE requires all these sub-systems. Sub-systems (ii) and (iii) take into consideration the injection of fuel directly in a gaseous state before entering the combustion chamber. The injection of gas has no storage effects at the manifold walls and the fresh mixed air and gas has no atomization phenomena. So, the gaseous state in the manifold is important in modelling the engine controller. The elements and variables for the CCDE are shown in a schematic diagram as in **Fig-2**. There are two new important sub-systems in the CCDE as stated below (Dyntar, Onder, & Guzzella, 2002).

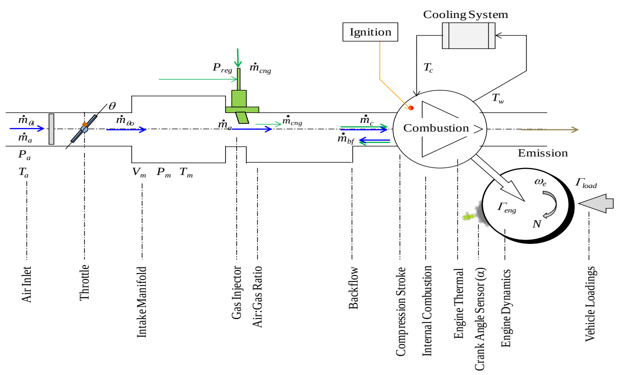

Figure 2: The Schematic Diagram for the whole CCDE system and sub-systems

1) A backflow which is a substantial transport of fresh charge back to the intake manifold, and
2) An injection timing dependent torque characteristic in which by injecting the gas when the intake valves are opened then stratification takes place in the cylinder.

a. **Throttle -** The changes of the mass flow rate of the air depend on the position of the throttle angle θ. The rate of the air mass flow into the manifold can be expressed as, <u>first</u> - An empirical function of the throttle angle with the function of air mass flow rate, and <u>second</u> - A function of the atmospheric and manifold pressure when the flow is chocked. These two functions are related to the mass flow rate across the throttle inlet valve that can be calculated from the standard orifice equation for compressible fluid flow as a function of θ as shown in **Fig-3**. The stagnation temperature T_0 and pressure P_0 are governed by the steady flow energy equation and for isentropic flow (Heywood J. B., 1988).

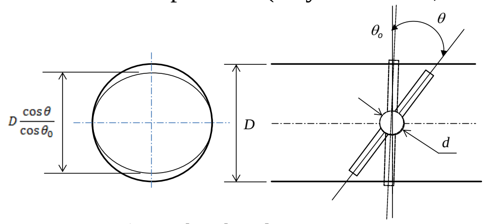

Fig-3: The Throttle Geometry

Throttle Plate Opened Area A_θ is defined as,

$$A_\theta = \frac{\pi D^2}{4}\left(1-\frac{\cos\theta}{\cos\theta_0}\right)+\frac{D^2}{2}\left[\frac{a}{\cos\theta}(\cos^2\theta -$$
a2cos2θ012−cosθcosθ0sin−1acosθ0cosθ−a 1−a212+sin−1a (1)

Where $a = \frac{d}{D}$ and summarized as below,

$$A_\theta(r,R) = 2R^2\left(\sin^{-1}\sqrt{1-\frac{r^2}{R^2}}-\frac{r}{R}\sqrt{1-\frac{r^2}{R^2}}\right) \quad (2)$$

And, the relationship for isentropic flow is given as,

$$\frac{T_c}{T_0} = \left(\frac{P_{Tc}}{P_0}\right)^{(\gamma-1)/\gamma} \quad (3)$$

and, stationery isentropic flow through an orifice is characterized by the flow function as,

$$\mathcal{K}_\gamma = \left\{\frac{2\gamma}{\gamma-1}\left[\left(\frac{P_{Tc}}{P_0}\right)^{2/\gamma}-\left(\frac{P_{Tc}}{P_o}\right)^{(\gamma+1)/\gamma}\right]\right\}^{1/2} ;\ \text{IF}\ \frac{P_{Tc}}{P_0} \geq \left(\frac{P_{Tc}}{P_0}\right)_{chocked} \quad (4)$$

$$\mathcal{K}_\gamma = \left(\mathcal{K}_\gamma\right)_{chocked} ;\ \text{IF}\ \frac{p_{Tc}}{p_0} < \left(\frac{P_{Tc}}{P_0}\right)_{chocked} \quad (5)$$

If the flow function is chocked, then **Equation 4** is written as the chocked flow function as below,

$$\left(\mathcal{K}_\gamma\right)_{chocked} = \left\{\gamma\left(\frac{2\gamma}{\gamma-1}\right)^{(\gamma+1)/(\gamma-1)}\right\}^{1/2} \quad (6)$$

Therefore, this air mass flow is defined by adapting 2D function using discharge coefficient (C_D) as a free parameter as shown below,

$$\dot{m}_{\theta 0} = \frac{C_D A_\theta P_o}{\sqrt{RT_o}} \cdot \left[\mathcal{K}_\gamma(P_R)\right] \text{ or} \quad (8)$$

$$\dot{m}_{\theta 0} = \left[\mathcal{K}_\theta(A_\theta)\right].\left[\mathcal{K}_\gamma(P_R)\right] \quad (9)$$

And, the air out mass flow rate ($\dot{m}_{\theta 0}$) is given as follow (Heywood J. B., 1988),

$$\dot{m}_{\theta o} = \frac{C_D A_\theta P_o}{\sqrt{\mathbf{R}T_o}} \left\{ \frac{2\gamma}{\gamma - 1} \left[\left(\frac{P_{T_c}}{P_0} \right)^{2/\gamma} - \left(\frac{P_{T_c}}{P_o} \right)^{(\gamma+1)/\gamma} \right] \right\}^{1/2} \quad (10)$$

and then the air mass flow rate is given as follow,

$$\dot{m}_{\theta i} = \frac{C_D A_\theta P_o}{\sqrt{\mathbf{R}T_o}} \left\{ \gamma \left(\frac{2\gamma}{\gamma - 1} \right)^{(\gamma+1)/(\gamma-1)} \right\}^{1/2} \quad (11)$$

These are part of the parameters for the development of the robust control modelling in which can be determined by referring to the steady-state air mass flow. This phenomenon is included in the modelling and calculating the air mass flow disturbances with respect to the current operating point.

b. **Intake Manifold -** The intake manifold is modelled in the control system design based on the manifold pressure and ratio of air-gas mass flow rate. It is modelled as a differential equation on the net changes between the incoming and outgoing of mass flow rate. The intake manifold mass flow is modelled up-to the intake valve. The air/gas ratio (AGR) is considered as discrete event for each combustion cylinder, in which the AGR for a cylinder is fixed till the next engine cycle. A model of the control system has to synchronous the air flow with the engine cycle. The air flow dynamic that goes into the intake manifold is time-based phenomenon synchronously with the crank shaft motion (α). This dynamic motion is converted into discrete model with the respect to the engine speed (*N*) (Powell, Fekete, & Chen, 1998). The time domain air flow dynamic is transformed into the crank angle as below,

$$N = \left(\frac{1}{6}\right) \frac{d\alpha}{dt}, \quad (12)$$

A continuity equation based on ideal gas law can be applied to the intake manifold provided that the values of the manifold pressure P_m and manifold temperature T_m are constant. The filling and emptying of the manifold are expressed as follow (Chang, 1993),

$$\dot{P}_m = \frac{RT_m}{V_m} \dot{m}_{\theta i} - \frac{\eta_v N V_d}{120 V_m} P_m \quad (13)$$

The air mass flow rate is discharged out from the intake manifold into the combustion cylinder is written as (Heywood J. B., 1988),

$$\dot{m}_{\theta o} = \frac{\eta_v V_d N}{120\, RT_m} P_m \quad (14)$$

By neglecting the inertial effects in the manifold within time t_c, the air mass flow rate $\dot{m}_a$ is determined by,

$$\dot{m}_a = -\frac{1}{t_c}\left(m_c - m_{ss}(\theta, N)\right) \quad (15)$$

The total air mass flow rate into the combustion cylinder $\dot{m}_c$ as a function of throttle angle θ, engine speed *N* within time Δt is determined by multiplying it with a constant as follow,

$$\dot{m}_c = K(\theta, N, \Delta t, m_a) \quad (16)$$

where, *K* is a constant value which can be adjusted to provide an appropriate air flow into each combustion cylinder as a function of Δt as determined by experiment. And, the value of air mass flow rate into the combustion cylinder is determined by,

$$\dot{m}_c = \frac{\eta_v \rho_a V_d N}{2} \quad (17)$$

The intake manifold dynamics is modelled as in **Equation 3.24** (Khan, Spurgeon, & Puleston, 2001)

$$\dot{P}_m = \frac{RT_m}{V_m}(\dot{m}_{\theta i} - \dot{m}_{\theta o}) \quad (18)$$

The output mass flow rate from the manifold into the combustion chamber is determined by **Equation (19)** (Crossley & Cook, 1991).

$$\dot{m}_{\theta o} = k_{mo} + k_{m1}\,.NP_m + k_{m2}\,NP_m^2 + k_{m3}.\,N^2 P_m \quad (19)$$

Where, $k_{mi}, i = 0, 1, 2, 3$: constant coefficient

c. **Air : CNG Ratio -** λ is an important variable in controlling the fuel of the CCDE for different engine model and manufacturer. There are several factors that affect the amount of the air mass flow and the CNG mass flow into the combustion cylinder: throttling, aerodynamics resistance and resonances in the intake manifold, mixture mass backflow due to closing of inlet valve, and the reverse flow of the burned gasses from the combustion into the inlet manifold. This also affects the amount of *mixture* that occupy the swept volume subjected to the regulator and manifold pressure as well the ambient pressure which is given via standard equation as below,

$$\boldsymbol{m_a = \rho_o\, V_d}. \quad (20)$$

Where, P_o=1.013 bar and ρ_o = 1.29 kg/m^3 (air density)

The value of *air:fuel* ratio λ influences the engine's work effectiveness W_e and thermal efficiency effectiveness η_e which is based on two variables - on the values of air λ_a and fuel ratio λ_f as shown in **Fig-4**, (Kiencke & Nielsen, Automotive Control Systems for Engine, Driveline and Vehicle, 2005).

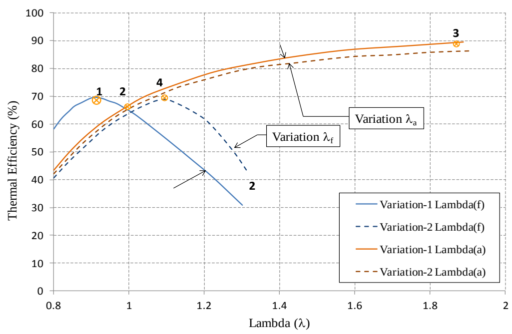

Figure 4: The Affect of the Thermal Efficiency (η_e %) due to the Variation Values of λ_f and λ_a 1. Maximum Power Input, 2. Stoichiometric Ration, 3. Diesel and Lean – Burn Engines, and 4. Moderately Lean Operation (after, *Kiencke & Nielsen, 2005*)

In this study, the value of CNG ratio (λ_{cng}) is evaluated under two (2) conditions; First – variation of λ_f and λ_a is determined by the driver. If λ >1 (Lean Operation), it means that the fuel injected into the manifold mixer is less than Stoichiometric Ratio (L_{SR}). If λ < 1 (Rich Operation), it means that the fuel injected into the manifold mixer is more than L_{SR}. Second – variation of λ_a and λ_f is determined by driver. If λ >1 (Lean Operation), it means that the air injected into the manifold mixer is more than the L_{SR}. If λ < 1 (Rich Operation), it means that the air injected into the manifold mixer is less than L_{SR}.

The ratio of the air and CNG (as the fuel) in which they are respectively the ratio between the real mass and the theoretical mass values as below,

$$\lambda_a = \frac{m_{a(real)}}{m_{a(theory)}} \text{ and } \lambda_{cng} = \frac{m_{cng\,(real)}}{m_{cng\,(theory)}} \qquad (21)$$

The graph in **Fig-4** shows the affect of the thermal efficiency (η_e%) due to the variation values of λ_f and λ_a as the basic operation of an engine, in which the fuel injection and ignition timing are assumed to be under an optimal control. The variations of air/fuel ratio affects exhaust emission gasses are formulated as shown in **Fig-5**.

$$\lambda = \frac{m_{a+cng}}{m_{a+cng\,.th}} \qquad (22)$$

Where, m_{a+cng} = mass of mixture between air & fuel at real value, and $m_{a+cng\,.th}$ = mass of mixture between air & fuel at theoretical value, and the value of λ is the ratio between the relative value of air and *CNG* as below,

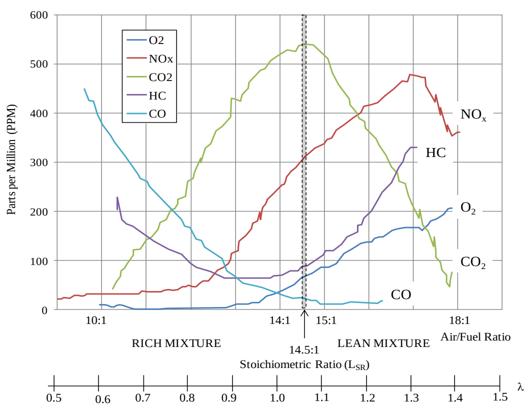

Fig-5: The Emission Values in Parts per Million (PPM) based on the Air/Fuel Ratio and the Required Stoichiometric Ratio (L_{SR}) (Heywood J. B., 1988).

d. **CNG Injection Timing** – the injected CNG is controlled by the injection timing (t_c) which is in the range of inlet valve opening period in second [s]. The value of CNG mass injected (m_{cng}) is determined as below,

$$\dot{m}_{cng} = \frac{1}{L_{SR}\,\lambda} \cdot \frac{\dot{m}_a}{N} \cdot \frac{2}{CC} \qquad (23)$$

where, *CC* = the number of 6 combustion cylinder, #2 = the air is combusted every alternate engine cycle. Noted that the amount of m_{cng} is proportionate to t_c and square root of the pressure difference ΔP of the intake manifold as shown below in **Fig-6**, (Kracke, 1992, p. 274) { This paper is **translated** by an industrial colleague of mine, Mr. Wojciech Stawecki, Export Manager of Auto-GAZ (Malaysia Branch), Head Quarter's Office based in Centrum, Poland.},

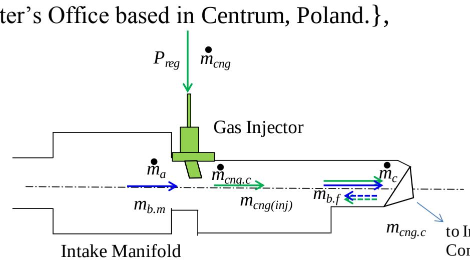

Figure 6: Gas Injector at the Intake Manifold

$$m_{cng} = \rho_{cng} \,.\, A_{noz} \,.\, C_D \,.\, \sqrt{\frac{2\Delta P}{\rho_{cng}}} \,.\, t_c \qquad (24)$$

Or,

$$\dot{m}_{cng} = A_{noz}\, C_D \left(2\rho_{cng}\,\Delta P\right)^{\frac{1}{2}} \qquad (25)$$

where, $\Delta P = P_{inj} - P_m$; $P_{inj} = P_{reg}$

IF the sample time of the *CNG* injection timing t_c is based on the function of the throttle angle $\Delta\alpha$ and engine speed N

$$t_c \cong \frac{1}{\lambda} \cdot \frac{\dot{m}_a}{N} \cdot \frac{2}{CC} \qquad (27)$$

IF the sample time of t_c for a stationary engine is based on the function of the air : *CNG* ratio (λ), N, $\dot{m}_a$ and the number of cylinder (CC) then,

$$t_c \cong \frac{\lambda_o}{\lambda} \,.\, t_o \qquad (28)$$

IF the sample time of the t_c is based on the function of an arbitrary air:fuel ratio $\lambda = 1$, reference air:fuel ratio λ_o from the Conversion Table – Lambda to Air:Fuel Ratio as shown in **Table 1**.

e. **State Space Equations for Throttle and Intake Manifold** – as stated in Clauses (a) and (b) as above, it makes available the values of air mass flow rate $\dot{m}_{\theta i}$ that enters into the throttle which is controlled by A_θ and θ. Then, the values of $\dot{m}_\theta$ can be determined with two conditions as shown in **Fig-7**.

Table 1: The Conversion Table from Lambda (λ) to AFR

λ	Gasoline	Alcohol	Gas	Diesel
0.70	10.3	4.5	10.9	10.2
0.75	11.0	4.8	11.6	10.9
0.80	11.8	5.1	12.4	11.6
0.85	12.5	5.4	13.2	12.3
0.90	13.2	5.8	14.0	13.1
0.95	14.0	6.1	14.7	13.8
1.00	14.7	6.4	15.5	14.5
1.05	15.4	6.7	16.3	15.2
1.10	16.2	7.0	17.1	16.0
1.15	16.9	7.4 7	17.8	16.7
1.20	17.6	7.7	18.6	17.4
1.25	18.4	8.0	19.4	18.1
1.30	19.1	8.3	20.2	18.9
1.35	19.8	8.6	20.9	19.6
1.40	20.6	9.0	21.7	20.3
1.45	21.3	9.3	22.5	21.0

1.50	22.1	9.6	23.3	21.8
1.55	22.8	9.9	24.0	22.5
1.60	23.5	10.2	24.8	23.2

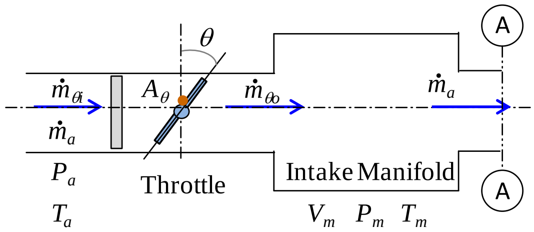

Figure 7: The Schematic Flow Diagram at Throttle & Intake Manifold

$$\dot{m}_{\theta o} = \frac{C_D A_\theta P_o}{\sqrt{\mathbf{R}T_o}}(\kappa_\gamma) \quad (29)$$

Let's κ_θ and κ_γ represent,

$$\kappa_\gamma = \left\{\frac{2\gamma}{\gamma-1}\left[\left(\frac{P_{Tc}}{P_0}\right)^{2/\gamma} - \left(\frac{P_{Tc}}{P_o}\right)^{(\gamma+1)/\gamma}\right]\right\}^{1/2}; \quad \text{IF } \frac{P_{Tc}}{P_0} \geq \left(\frac{P_{Tc}}{P_0}\right)_{chocked} \quad (30)$$

$$\kappa_\gamma = \kappa_{\gamma\,chocked} \;; IF \frac{p_{Tc}}{p_0} < \left(\frac{P_{Tc}}{P_0}\right)_{chocked} \quad (31)$$

Noted that,

$$\kappa_{\gamma\,chocked} = \left\{\gamma\left(\frac{2\gamma}{\gamma-1}\right)^{(\gamma+1)/(\gamma-1)}\right\}^{1/2} \quad (32)$$

And,

$$\kappa_\theta = \frac{C_D P_o}{\sqrt{RT_o}} A_\theta \quad (33)$$

Where, κ_θ and κ_γ are the constant multiplying factor for the throttle dynamical system. As the systems above is non-linear functions of two independent variables (A_θ and θ) and (γ and T_c) for each respective, applying Taylor Series expansion of $\mathrm{F}(\kappa_\theta)$ and $\mathrm{F}(\kappa_\gamma)$ around some nominal operating value which linearizes the system.

As the non-linear functions of $f_{\kappa_\gamma}(\gamma, T_c)$ and $f_{\kappa_\theta}(A_\theta, \theta)$ are approximated sufficiently, it is assumed that their higher-order derivatives are small and can be neglected. The expansions for the above equations are divided by increasingly large factorials which further wanes the magnitude of the higher-order. Then, this linearization is used to obtain a linear system approximation to determine the air/mass flow rate at the throttle outlet. The vector equation for the above formulation is written as the state-space equation for the throttle dynamics system,

$$\dot{m}_{\theta o} = \kappa_\theta(\theta) + \kappa_\gamma(T_c) \quad (34)$$

From **Clause (b)** above, the relation between both feed-back and feed-forward conditions can be rewritten as,

$$f_1(N, P_m) = \frac{\dot{m}_a}{\dot{m}_c} = \frac{T_m}{T_a} \quad (35)$$

The pressure change at the outlet manifold due to the regulator which affects the air mass flow rate as shown in **Fig-8**, so

$$\dot{P}_m = k\frac{RT_m}{V_m}(\dot{m}_{\theta i} - \dot{m}_{\theta o}) \quad (36)$$

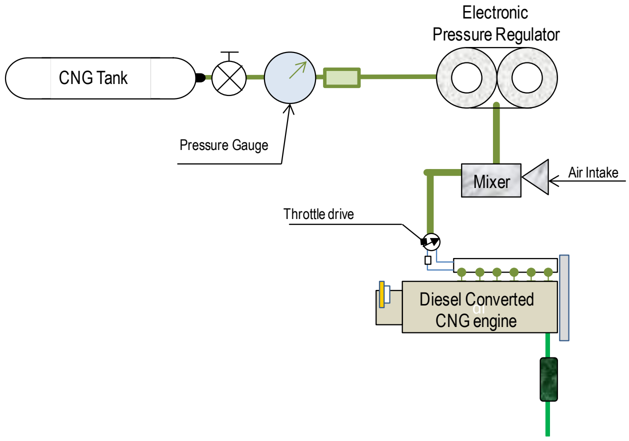

Figure 8: The Combination System of the CCDE

Then the equations are written as,

$$\dot{P}_m = k\frac{RT_m}{V_m}(\dot{m}_{\theta o} - f_1(N, P_m)\dot{m}_{\theta o}) \quad (37)$$

The first dynamic equation that has all these functions as the state variable,

$$\dot{P}_m = \frac{RT_m}{V_m}\dot{m}_{\theta o} - \frac{\eta_v N V_d}{120 V_m} P_m \;; dt = \frac{d\alpha}{6N} \quad (38)$$

The above equations can be derived as State-Space Equation for the intake manifold system as,

$$\dot{P}_m = \left(\frac{RT_m}{V_m}\right)\dot{m}_{\theta o} - \left(\frac{\eta_v V_d}{120 V_m}\right) N P_m \;; \text{ if and only if, } T_m \leq T_C \quad (39)$$

The throttle and intake manifold are only functioning if and only if $T_m \leq T_C$, then, the equation above can be rewritten as,

$$\dot{P}_m = \left(\frac{RT_c}{V_m}\right)\dot{m}_{\theta o} - \left(\frac{\eta_v V_d}{120 V_m}\right) N P_m \quad (40)$$

By introducing constant multiplying factors to the system, the above is formulated in the form of state vector equation as the state-space dynamics equation as below,

$$\dot{P}_m = \kappa_{m2}(\dot{m}_{\theta o}) - \kappa_{m1}(N)(P_m) \quad (41)$$

Noted that,

$$\kappa_{m1} = \frac{\eta_v V_d}{120 V_m} \; and \; \kappa_{m2} = \frac{RT_c}{V_m}; \quad (42)$$

Where, κ_{m1} and κ_{m2} are the constant multiplying factor for the dynamics system for the intake manifold. From the second state variable for one of the output with the function of engine speed (N), Temperature ($T_m \leq T_c$), and pressure manifold (P_m), it can be formulated by introducing constant multiplying factors of the intake manifold as,

$$\dot{m}_a = \left(\frac{\eta_v V_d}{120 V_m}\right)\left(\frac{V_m}{RT_c}\right) N P_m \quad (43)$$

And, the state vector equation as the state space equation can be written as follows,

$$\dot{m}_a = \frac{\kappa_{m1}}{\kappa_{m2}}(N)(P_m) \quad (44)$$

therefore $\dot{P}_m$ is determined as,

$$\dot{P}_m = \kappa_{m1}(N)\left(\frac{\dot{m}_{\theta o}}{\dot{m}_a} - 1\right)(P_m) \;; \text{if and only if, } \dot{m}_{\theta o} \neq \dot{m}_a \quad (45)$$

Therefore, if, $\dot{m}_{\theta o} = \dot{m}_a$, then, the pressure in the manifold $\dot{P}_m$ will be zero and the engine stop working.

Then, by introducing a constant multiplying factor κ_{m3}, the state vector for air mass flow that goes into the combustion cylinder is one of the state-space equation,

$$m_a = \kappa_{m3}\left(\frac{1}{T_c}\right)(\Delta\alpha)(P_m) \quad (46)$$

$$\text{where, } \kappa_{m3} = \frac{\eta_v V_d}{720R} \quad (46)$$

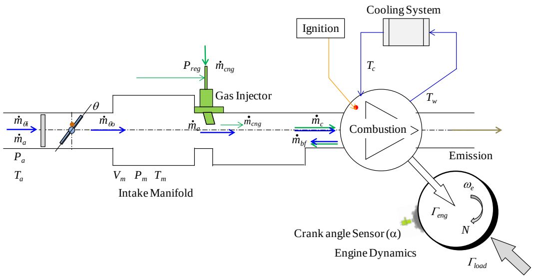

Figure 9: The Schematic Flow Diagram from the Manifold Outlet to the Engine Dynamics

f. **CNG injection** - Since it is a four-stroke engine, the air is usually combusted every the alternate engine cycle, then **Equation 24** as shown in **Fig-6** is written with the respect to the nozzle area for the CNG injector and the square root is for the pressure difference as mentioned in the same equation. Noted that the amount of m_{cng} is proportionate to t_c and square root of the pressure difference (ΔP) of the intake manifold as shown in **Fig-9**, (Kracke, 1992, p. 274) [This paper is **translated** by an industrial comrade of mine, Mr. Wojciech Stawecki, Export Manager of Auto-GAZ (Malaysia Branch), Head Quarter's Office based in Centrum, Poland] as below,

$$\dot{m}_{cng} = A_{noz}\, C_D \left(2\rho_{cng}\, \Delta P\right)^{\frac{1}{2}} \quad (47)$$

g. **Backflow -** It causes incomplete flame propagation. The intake manifold absolute pressure (*imap*) drops quickly as engine starts to accelerate. The low *imap* causes unnecessary backflow that is also known as an internal exhaust gas recirculation during valve overlapping period. Flame speed and flame propagation front slows down in diluted intake charge.

The backflow transport substantial fresh charge back to the intake manifold and a stratification process takes place in the cylinder after an injection of the gas as the fuel when the intake valve is opened. This situation is crucial in modelling a controller for the CCDE as shown in **Fig-9**.

Other variables to be considered in formulating the amount of CNG in the cylinder during the period when inlet valve is closing are as follows,

a. The maximum mass of CNG entering the combustion area ($m_{cng.c}$)
b. The mass of CNG flowing from the combustion area back to the intake manifold ($m_{b.f}$)
c. The balance mass of CNG in the combustion area after inlet valve is closed ($m_{cng.c}$)
d. The balance mass of CNG in the manifold because of the injection delay ($m_{b.m}$)
e. The mass of CNG injected ($m_{cng.inj}$)
f. The ratio of the CNG that back flows into the manifold and the maximum volume of CNG that is assumed as homogeneous charge (φ), where $\varphi = 0$ thus if all the CNG flows into the combustion area with no backflow
g. The stratification factor (ς_f)
h. Injection overlapping (I_o), where $I_o = 0$ thus if all CNG is injected into the combustion; $I_o = 1 \rightarrow$ if all CNG is not injected into the combustion area, but it will be injected during the induction stroke.

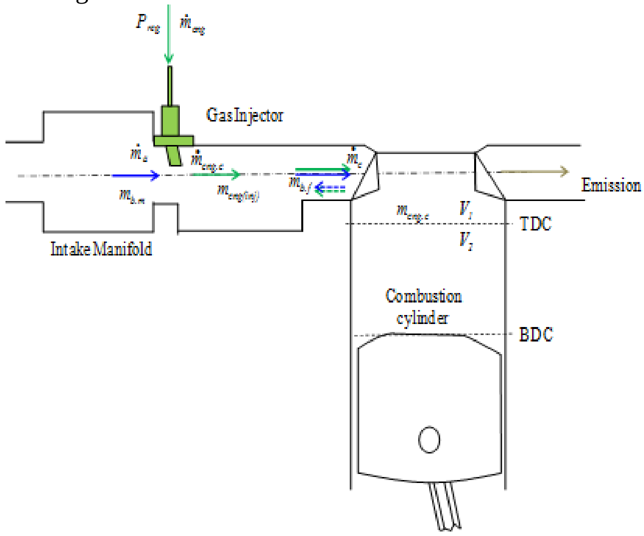

Figure 10: The Schematic Diagram for the CCDE with Backlash

If the piston is at the BDC, the volume above the piston can be divided into two volumes of engine cylinder as shown in **Fig-10**:

(i) $V_1 = V_a$: only cylinder volume occupied by air (V_a) and not by gas,

(ii) $V_2 = V_a + V_{cng}$: the CNG and air mixture is assumed homogeneous, where V_{cng} = cylinder volume occupied by CNG that goes into the cylinder,

(iii) $\varphi = \frac{V_{b.f}}{V_a + V_{cng}}$: the fraction between the CNG volume backflows into the manifold and the air and CNG mixture volume that goes into the cylinder assumed as homogeneous charge,

(iv) $V_{b.f} = \varphi.\varsigma_f.(V_a + V_{cng})$: the stratification process that occurs which is considered in the backflow calculation,
where,
V_{BDC} = cylinder volume during piston at BDC
$\varsigma_f = 1$ for *homogeneous* charge where $V_a = 0$

Considering items (i) to (iv) above and noting that the stratification factor is the ratio between the cylinder volume and second volume (V_2) when the piston is at the BDC,

$$\varsigma_f = \frac{V_{BDC}}{V_2} \text{ OR, } \varsigma_f = \frac{V_1 + V_2}{V_2} \text{ OR, } \varsigma_f = \frac{2V_a + V_{cng}}{V_a + V_{cng}} \quad (48)$$

$$\rho_{cng} = \frac{m_{b.f}}{V_{b.f}};\ \rho_a = \frac{m_a}{V_a};\ \rho_{cng} = \frac{m_{cng}}{V_{cng}} \quad (49)$$

Assumed that, $\rho_a = \rho_{cng}$, mixtures of air and CNG,

$$\text{So, } \frac{m_a}{m_{cng}} = \frac{V_a}{V_{cng}} \text{ and, } \frac{m_{b.f}}{m_{cng}} = \frac{V_{b.f}}{V_{cng}} \quad (50)$$

$$m_{b.f} = \varphi.\varsigma_f.m_a \quad (51)$$

From the assumption made in **Equation 51,** the stratification factor in **Equation 48** can be rewritten as,

$$\varsigma_f = \frac{2m_a + m_{cng}}{m_a + m_{cng}} \quad (52)$$

Then, the mass of backflow is given as below,

$$m_{b.f} = \varphi.m_c.\left(\frac{2m_a + m_{cng}}{m_a + m_{cng}}\right) \quad (53)$$

Due to the mass backflow $m_{b.f}$, some of the mixture of air and CNG returns to the manifold. Therefore the total mass into the cylinder m_c is given below,

$$m_c = m_a + m_{cng} - m_{b.f} \quad (54)$$

Backflow State Space Equation – The backflow of the CNG within the air flow back into the manifold is one of the state variables or disturbances that is very crucial to be considered in designing the controller of the CCDE.

The mixed air and CNG volume that goes into the cylinder $\dot{\varphi}$, the air mass flow from the manifold m_a that mixed with the injected *CNG* (m_{cng}) from the regulator pressure P_{reg}, and the engine speed *N*. So, if the function is with the respect to time,

$$\dot{m}_{b.f} = \dot{\varphi}.\varsigma_f.m_a \quad (55)$$

Noted that the variables λ in **Clause c.** above, the value of mixed gas mass flow is determined by the function of time constant (t_c),

$$m_c = \rho_g.A_{eff}.\left(\frac{2\Delta P}{\rho_g}\right)^{1/2}.t_c \quad (56)$$

Where, $\Delta P = \left(P_{reg} - P_m\right)$; $\rightarrow P_{reg}$ = Regulator Pressure (57)

Therefore the value for the backflow mass with the respect to the crank angle (α) is obtained by substituting the above and the state space equation for the backflow is given below,

$$\frac{dm_{bf}}{d\alpha} = \frac{\dot{\varphi}.\varsigma_f.\rho_g.A_{eff}.\Delta\alpha.}{6N}\left[\frac{2(P_{reg} - P_m)}{\rho_g}\right]^{1/2} \quad (58)$$

h. **Combustion and Engine Dynamics**

Compression Stroke - A compression stroke starts when both inlet and outlet valves are closed; the air/gas mixture in the combustion area is a small fraction of its initial volume and combustion is initiated towards the end of the compression stroke as the piston approaches the TC; then the cylinder pressure rises more rapidly. **Fig-11** shows the standard geometry of reciprocating engine parameters and the relevant equations for the development of the control model (Heywood J. B., 1988).

$$s = r\cos\alpha + (l^2 - r^2 sin^2\alpha)^{1/2} \quad (59)$$

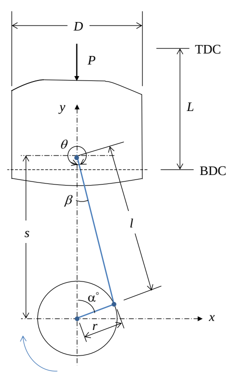

Figure 11: The Geometrical Diagram for the Engine Variables such as Piston, connecting Rod (*l*), Crank Radius (*r*), Crank Angle (α), Bore (*D*), Stroke (s_{st}) and Distance between the Crank Axis and the Piston Pin (*s*)

The ratio between the instantaneous piston velocity and the mean speed can be summarized as follow,

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2} sin\,\alpha\left[1 + \frac{\cos\alpha}{(R^2 - \sin^2\alpha)^{1/2}}\right] \quad (60)$$

The piston stroke (s) can be written with respect to the crank angle (α),

$$s(\alpha) = r\left[1 - \cos\alpha + \frac{1}{r}\left(1 - \left(1 - \right.\right.\right.$$
$r2l2$sin2$\alpha12$ (61)

THEN the derivatives for the piston stroke (ds_{st}) over the crank angle ($d\alpha$) as,

$$\dot{s}_{st} = r.\dot{\alpha}.\left(\sin\alpha + \frac{r}{l}.\frac{\sin\alpha\cos\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}}\right) \quad (62)$$

$$\ddot{s}_{st} = r.\dot{\alpha}^2.\left(\cos\alpha + \frac{\frac{r}{l}(\cos^2\alpha - \sin^2\alpha) + \frac{r^2}{l^2}\sin^4\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}}\right) + r.\ddot{\alpha}.\left(\sin\alpha + \frac{r}{l}.\frac{\sin\alpha\cos\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}}\right) \quad (63)$$

Introducing κ_{s1} and κ_{s2} as the two constant multiplying factors as below,

$$\kappa_{s1} = \left(\sin\alpha + \frac{r}{l}.\frac{\sin\alpha\cos\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}}\right); \quad (64)$$

$$\kappa_{s2} = \left(\cos\alpha + \frac{\frac{r}{l}(\cos^2\alpha - \sin^2\alpha) + \frac{r^2}{l^2}\sin^4\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}}\right) \quad (65)$$

then,

$$\dot{s}_{st} = r\kappa_{s1}\dot{\alpha} \quad (66)$$

Or,

$$\ddot{s}_{st} = r(\kappa_{s2}\dot{\alpha}^2 + \kappa_{s1}\ddot{\alpha}) \quad (67)$$

From the swept volume (V_d) can also be determined by,

$$V_d = C_c(V_a - V_b) \quad (68)$$

Combustion dynamics – the work done by the force *F* after combustion and Kinetic energy K_e which is converted and exerted into the combustion chamber with respect to time is,

$$K_e = \frac{1}{2}mv^2 \Rightarrow \frac{dK_e}{dt} = \frac{1}{2}m\left(\frac{dv}{dt}v\right) + \frac{1}{2}m\left(v\frac{dv}{dt}\right) \quad (69)$$

And, the scalar of the formulation product is commutative to each other, then,

$$\mathbf{P} = \frac{dK_e}{dt} \text{ OR } \mathbf{P} = \dot{K}_e \quad (70)$$

Internal Rigid Body Dynamics - The internal combustion engine is very complex non-linear mechanical system which is difficult to characterize. Usually, a torque or intake manifold pressure is given with engine speed. It is possible to characterize and model as steady-state condition through the dynamics behaviour of the internal rigid body of the engine subsystem. The power exerted due to combustion pressure generates force and torque relative to the crankshaft angle.

The free rigid body diagram of the piston, con-rod and crank shaft as shown in **Fig-12**, the stroke is determined as differentiating and merging that gives,

$$\dot{\beta} = \frac{r\cos\alpha}{l\cos\beta}\dot{\alpha} \text{ and,} \quad (71)$$

$$\dot{s} = r\frac{\sin(\alpha+\beta)}{\cos\beta}\dot{\alpha} \quad (72)$$

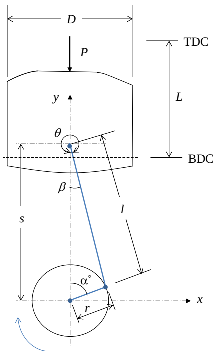

Fig-12: The Free Rigid Body Diagram

Kinetic Energy - Referring to **Fig-13** considers the formulation of the kinetic energy and moment of force on the piston, crankshaft and connecting rod. It is assumed that all the rigid body are symmetric and the centre of mass for each is at their geometric centre.

The formulation for the total kinetic energy of the slider mechanism is as follows,

$$\sum K_e = \frac{1}{2}(\dot{\alpha})^2 \underbrace{\left[m_P r^2\left(\frac{\sin(\alpha+\beta)}{\cos\beta}\right)^2 + m_Q r^2\right]}_{I_c} \quad (73)$$

$$\sum K_e = \frac{1}{2}I_c(\dot{\alpha})^2 \quad (74)$$

Where,

$$I_c = \underbrace{m_P r^2\left(\frac{\sin(\alpha+\beta)}{\cos\beta}\right)^2}_{1st\ part} + \underbrace{m_Q r^2}_{2nd\ part}\ ] \quad (75)$$

:= Moment of Inertia at crankshaft [kg.m^2]

Noted that the 2nd part of the **equation above** is a moment of inertia I_c generated from m_P and m_Q with the distance r which affects the crankshaft.

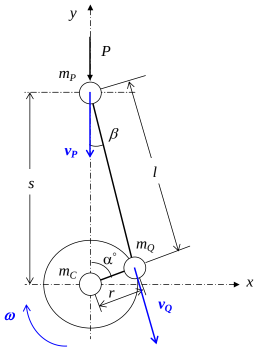

Figure 13: The Assumption of Total Mass for the Mechanism of the Dynamics System Model

Exerted Power - Referring to **Fig-14**, the power is exerted onto the dynamics mechanism system as,

$$\mathbf{P} = M_t\dot{\alpha} + (P_{atm} - P)A_c\dot{s} \quad (76)$$

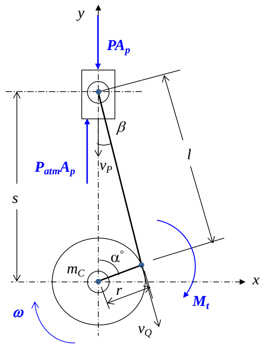

Figure 14: The Exerted Power and Moment of Force Modelling on the Dynamics Mechanism System

And it is emerged as,

$$\mathbf{P} = \frac{1}{2}\left[\frac{dI_c}{dt}(\dot{\alpha})^2 + I_c(2\dot{\alpha}\ddot{\alpha})\right] \quad (77)$$

It gives,

$$M_t\dot{\alpha} + (P_{atm} - P)A_c\dot{s} = \frac{1}{2}\left[\frac{dI_c}{dt}(\dot{\alpha})^2 + \right. \quad (78)$$

$Ic2aazero\ with\ constant\ speed$

Assuming that the engine is running at a constant speed, then the approximation is valid because the angular acceleration is zero, so **the equation above** can be simplified as,

$$M_t\dot{\alpha} + (P_{atm} - P)A_c\dot{s} = \frac{1}{2}\frac{d}{dt}I_c(\dot{\alpha})^2 \quad (79)$$

Therefore, the turning moment with respect to the angular velocity of the crankshaft is determined as,

$$M_t\dot{\alpha} = (P - P_{atm})A_c\dot{s} + \frac{1}{2}\frac{d}{dt}I_c(\dot{\alpha})^2 \quad (80)$$

then the above equation can be expressed as,

$$M_t\dot{\alpha} = (P - P_{atm})A_c\dot{s} + \frac{1}{2}\frac{d}{dt}\left[\underbrace{m_Q r^2}_{neglected} + \right. \quad (81)$$

$mPr2\sin\alpha+\beta\cos\beta22nd\ part\alpha2$

and solve the 2nd part of the differential equation to determine the turning moment as below,

$$M_t = (P - P_{atm})A_c . r\frac{\sin(\alpha+\beta)}{\cos\beta}(\dot{\alpha}) + m_P.r^2\left(\frac{\sin(\alpha+\beta)}{\cos^3\beta}\right)\left[\frac{r}{l}\frac{\cos^2\alpha}{\cos\beta} + \cos\beta\cos(\alpha + \right. \quad (82)$$

$\beta\alpha2$

Therefore, the turning moment M_t is equal to the pressure moment in the combustion cylinder M_p.

$$M_p = M_t = (P - P_{atm})A_c . r\frac{\sin(\alpha+\beta)}{\cos\beta}(\dot{\alpha}) \quad (83)$$

And, the moments due to the inertia M_i of the moving parts of the mechanism is summarized as,

$$M_i = m_P.r^2\left(\frac{\sin(\alpha+\beta)}{\cos^3\beta}\right)\left[\frac{r}{l}\frac{\cos^2\alpha}{\cos\beta} + \right. \quad (84)$$

$\cos\beta\cos\alpha+\beta\alpha2$

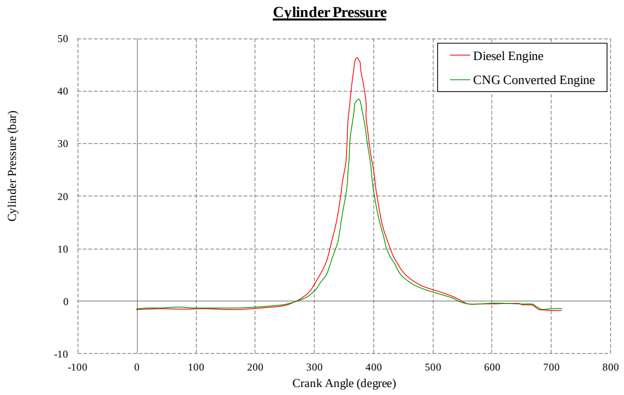

Figure 15: The values of Cylinder Pressure (*P*, bar) is determined via testing result between diesel engine and CNG converted engine vs Crank Angle (degree), P_{max} (CNG converted engine) is approximately 39bar at TDC (720 degree)

Engine Dynamics - one of the main objectives in designing an engine is to generate a maximum torque. It can be done by controlling the quantity of the mixture and air:CNG ratio. The engine torque is a nonlinear function with respect to the engine variables, such as fuel mass in the cylinder, air:fuel ratio, engine speed, and ignition or injection timing.

The engine torque Γ_{eng} is impressed by the function of engine speed *N*, total truck load Γ_{load} and mass flow rate into the combustion area m_m which are influenced by the values of *air:fuel* ratio, fuel thermal/heating value, volumetric efficiency, fuel consumption efficiency and air density inlet (Crossley & Cook, 1991). The variables that are due to the function of engine torque are as follows,

$$\Gamma_{eng} := \begin{cases} Q_{HV} \\ \eta_f \\ \eta_v \\ \rho_{ai} \\ \lambda \end{cases}$$

where,

λ = air:gas ratio → aims to keep the stoichiometric relative air:gas ratio,

λ = 1 for the best of conversion efficiency, throttle control and ignition timing.

In which the relationship of those functions is shown below,

$$\Gamma_{eng} = \frac{Q_{HV} . \eta_f . \eta_v . \rho_{ai} . V_d . \lambda}{4\pi} \quad (85)$$

Power from the engine can be determined as,

$$P = 2\pi N \Gamma_{eng} \quad (86)$$

Notes that, $\lambda = \frac{\dot{m}_a}{\dot{m}_g}$ (87)

where the ratio is estimated from 0.056 to 0.083

$$\eta_f = \frac{1}{(sfc)Q_{HV}} \quad (88)$$

where, $sfc = \frac{\dot{m}_g}{P}$ is specific fuel consumption and sometimes it is called brake specific fuel consumption (*bsfc*). It is a measure of the fuel efficiency that measure the rate of fuel consumption over the power produced with the resulting units are $^{g}/_{kW.h}$. Where, $\dot{m}_g$ is the fuel consumption rate in grams per hour, and P is the power produced in kW, and, $P = \mathcal{T}\omega$ where, $\mathcal{T}$ is the engine torque in *Nm* and ω is the engine speed in *rad/s*. The best *sfc* value for spark ignition engine is 270 g/kWh, and Q_{HV} is the heating value which is accepted by the commercial is between 42 ~ 44 MJ/kg

The torque can also be determined by analytical curve fitting technique using test data obtained from engine test and the steady-state brake torque can be estimated as follows, (Khan, Spurgeon, & Puleston, 2001) (Crossley & Cook, 1991),

$$\begin{aligned} \Gamma_{eng} = {} & k_{e0} + k_{e2}(m_c) + k_{e3}(\lambda) + k_{e4}(\lambda)^2 \\ & + k_{e5}(\sigma) + k_{e5}\sigma^2 + k_{e6}(N) \\ & + k_{e7}(N)^2 + k_{e8}(N\sigma) \\ & + k_{e9}(\sigma m_c) + k_{e10}(\sigma^2 m_c) \end{aligned} \quad (89)$$

And, the crankshaft speed-state equation can be written as,

$$I_c \dot{N} = \Gamma_{eng} - \Gamma_{load} \quad (90)$$

Where, $k_{ei}, i = 1, 2, 3, \ldots, 10$ = are constant coefficients

The torque can also be determined by taking into consideration on **Equations 86** and **89**,

$$P = 2\pi N \Gamma_{eng} = M_t \dot{\alpha} + (P_{atm} - P) A_c \dot{s} \quad (91)$$

$$\Gamma_{eng} = \frac{M_t \dot{\alpha} + (P_{atm} - P) A_c \dot{s}}{2\pi N} \quad (92)$$

Where, the engine torque, Γ_{eng} for CCDE is determined from the experiment of the engine speed *N* as plotted in **Fig-16**.

On the Road Power Load - The truck has to be capable of moving on any road levels at a steady speed. The power requirements called "On the Road Power Load" are useful reference point. The Power is used to overcome the rolling resistance which is consumed by the friction of the tires, and the aerodynamics drag resistance against the wind loading. For the given Rolling Resistance (C_r) and the drag coefficient (C_D), the approximate value for road power load (P_r) is given by, (Heywood J. B., 1988, pp. 48-50),

$$P_r = \left(C_r M_v g + \frac{1}{2}\rho_a C_D A_v S_v^2\right) S_v \quad (93)$$

From the those constant values as stated below, **Equation 93** can be summarized as

$$P_r = [2.73 C_r W_V + 0.0126 C_D A_V S_V] S_V \times 10^{-3} \quad (94)$$

The torque produced by the *CNG* engine is comparable to the diesel engine before the conversion is shown in **Fig-16**.

Mean Effective Pressure - The work done by the combustion in each cylinder produces torque. The torque generated by any particular CCDE depends on the engine used. The useful relative engine performance measurement is called Mean Effective Pressure (*mep*) which is the work per cycle divided by the cylinder volume displaced per cycle [N/m]. Another approach is to be used the engine dynamics model to estimate the *mep* is by extracting the instantaneous torque from flywheel and convert it to *mep*. The power per cylinder is related to the indicated work per cycle is by (Heywood J. B., 1988),

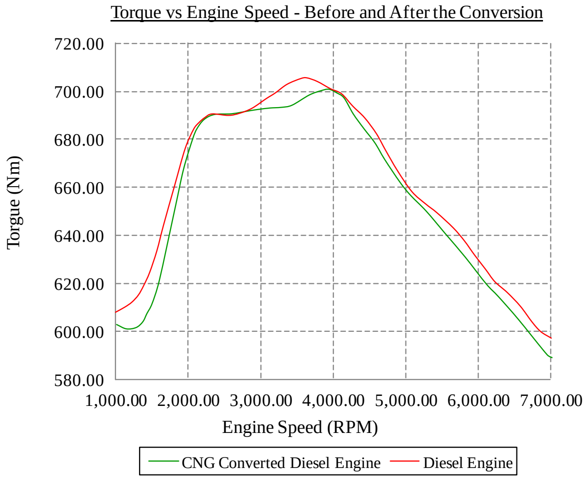

Figure 16: The Performance Curve (Torque vs Speed), Before and After the Conversion (Source: the Result from this Project)

$$\mathbf{P} = \frac{WN}{n_R} \quad (95)$$

Where, n_R = 2; for four-stroke engine. Therefore, the work per cycle is determined by,

$$W = \frac{\mathrm{P}n_R}{N}; \; W \text{ is the work done per unit cycle} \quad (96)$$

The mean effective pressure (*mep*) is determined by dividing **Equation 96** with swept volume of the cylinder as shown in **Equation 68**,

$$mep = \frac{Pn_R \times 10^3}{V_d N}; \text{mep is in } [kPa] \quad (97)$$

Then, *mep* can also be expressed in terms of torque,

$$mep = \frac{6.28\, n_R T}{V_d}; \text{mep is in } [kPa] \quad (98)$$

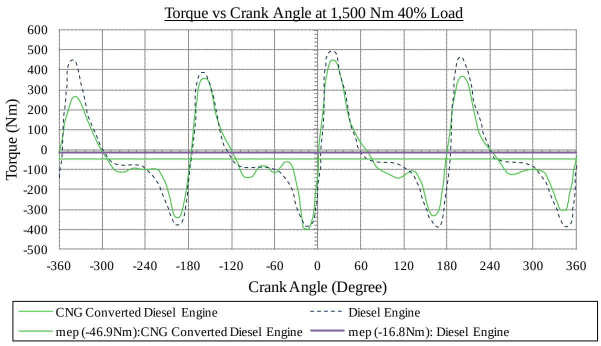

Figure 17: The Comparison Engine Torque Values between CNG Converted Engine and Diesel Engine at 1,500RPM with Additional 40% Total Truck Load vs Crank Angle

4. State Space Equation

State Space Equations for the Combustion and Engine Dynamics - The conservation of energy that generates the internal works done on the system from the internal combustion. The total work done is expressed via the Otto cycle at constant volume is,

$$W = m_a c_v (\Delta T_{3-4}) \quad (99)$$

The work done due to the conservation of energy as above is comparable provided with certain periodic time to the power that is transformed by the combustion as,

$$P = \frac{dW}{dt} = \dot{m}_a c_v (\delta \Delta T) \quad (100)$$

The power exerted on the system particles is equivalent with the value of kinetic energy with respect to the time change in the system particles as below,

$$P = 2\pi N \Gamma_{eng} = \dot{m}_a c_v (\delta \Delta T) \quad (101)$$

$$\Gamma_{eng} = \frac{\dot{m}_a c_v (\delta \Delta T)}{2\pi N} \quad (102)$$

And, to determine the piston stroke with respect to time is as written below (Arifin & Hassan, 2015),

$$\dot{s}_{st} = r\kappa_{s1}\dot{\alpha} \quad (103)$$

$$\ddot{s}_{st} = r(\kappa_{s2}\dot{\alpha}^2 + \kappa_{s1}\ddot{\alpha}) \quad (104)$$

Where, κ_{s1} and κ_{s2} both are the constant multiplying factor for piston dynamical system as given below (Arifin & Hassan, 2015),

$$\kappa_{s1} = \left(\sin\alpha + \frac{r}{l} \cdot \frac{\sin\alpha \cos\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}} \right) \quad (105)$$

$$\kappa_{s2} = \left(\cos\alpha + \frac{\frac{r}{l}(\cos^2\alpha - \sin^2\alpha) + \frac{r^2}{l^2}\sin^4\alpha}{\left(1 - \frac{r^2}{l^2}\sin^2\alpha\right)^{1/2}} \right) \quad (106)$$

The power exerted in the combustion system is actually the kinetic energy with respect to a time derivative and directly provide the moment of force on the piston, and the state-space equations for the importance variables in the whole system are as follows,

$$P = \dot{K}_e = \frac{1}{2}\frac{d}{dt}[I_c(\dot{\alpha})^2] \quad (107)$$

Where, I_c = an effective inertia

$$\frac{1}{2}[I_c(\dot{\alpha})^2] = m_a c_v (\delta \Delta T) \quad (108)$$

The overall schematic diagram and processing flow for the CCDE are shown in **Fig-18 and Fig-19**.

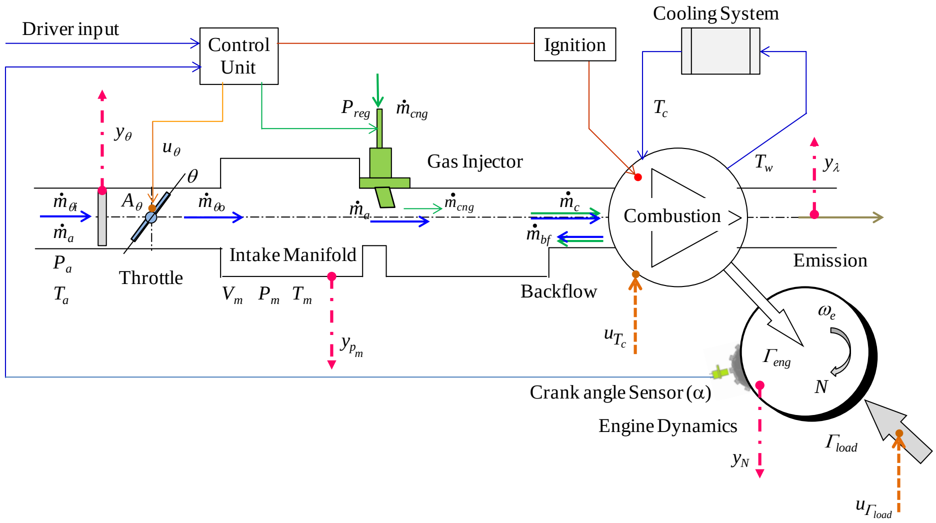

Figure 18: The overall Schematic Flow Diagram for the CCDE

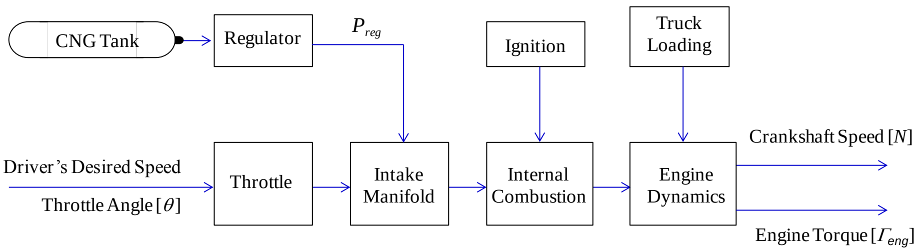

Figure 19: The Opened-loop Schematic Processing Flow for the CCDE System

Engine Variables Interconnection - The development of the interconnection between the state space equations is to establish a validity of the whole system. The above mathematical model and state variables on the CCDE elements (Arifin & Hassan, 2015) develop the relevant state space equations of the engine variables as follows:

$$\dot{m}_{\theta o} = \kappa_\theta(\theta) + \kappa_\gamma(T_c) \quad (109)$$

$$\dot{P}_m = \kappa_{m2}(\dot{m}_{\theta o}) - \kappa_{m1}(N)(P_m) \quad (110)$$

$$\kappa_{m1} = \frac{\eta_v V_d}{120 V_m} \text{ and } \kappa_{m2} = \frac{RT_c}{V_m}; \quad (111)$$

$$m_a = \kappa_{m3}\left(\frac{1}{T_c}\right)(\Delta\alpha)(P_m) \quad (112)$$

$$\kappa_\gamma = \left\{\frac{2\gamma}{\gamma - 1}\left[\left(\frac{P_{T_c}}{P_0}\right)^{2/\gamma} - \left(\frac{P_{T_c}}{P_o}\right)^{(\gamma+1)/\gamma}\right]\right\}^{1/2} \quad (113)$$

$$; IF \frac{P_{T_c}}{P_0} \geq \left(\frac{P_{T_c}}{P_0}\right)_{chocked}$$

$$\kappa_{\gamma\, chocked} = \left\{\gamma\left(\frac{2\gamma}{\gamma-1}\right)^{(\gamma+1)/(\gamma-1)}\right\}^{1/2} \quad (114)$$

$$\dot{m}_{cng} = \frac{1}{L_{SR}\lambda}\frac{\dot{m}_a}{N}\frac{2}{CC} \quad (115)$$

$$\dot{m}_{b.f} = \dot{\varphi}.\varsigma_f.m_a \quad (116)$$

$$P = \frac{dK_e}{dt} \text{ OR } P = \dot{K}_e = \frac{1}{2}\frac{d}{dt}[I_c\dot{\alpha}] \quad (117)$$

All the relevant state-space equations of the engine variables are as follows (Arifin & Hassan, 2015):

Variable #1: The rate of air mass flow at the throttle outlet $\dot{m}_{\theta o}$

$$\dot{m}_{\theta o} = \frac{C_D A_\theta P_o}{\sqrt{\mathbf{R}T_o}}(\kappa_\gamma) \quad (118)$$

Where, $\dot{m}_{\theta o}$ is the value of air mass flow rate at the throttle outlet. This equation is generalized by introducing a constant multiplying factor κ_θ with the function of throttle angle θ as follows,

$$\dot{m}_{\theta o} = \kappa_\theta(\theta).\frac{C_D P_o}{\sqrt{\mathbf{R}T_o}}.\kappa_\gamma \quad (119)$$

Where, $A_\theta = \kappa_\theta(\theta)$ in which κ_θ is a constant multiplying factor.

Variable #2: The rate of filling and emptying manifold pressure $\dot{P}_m$

$$\dot{P}_m = \frac{RT_m}{V_m}\dot{m}_{\theta i} - \frac{\eta_v N V_d}{120 V_m}P_m \quad (120)$$

The above equation is reconstructed as below,

$$\dot{P}_m = \frac{RT_m}{V_m}(\dot{m}_{\theta i} - \dot{m}_{\theta o}) \quad (121)$$

where, $\dot{m}_{\theta o}$ is the rate of air mass flow at the manifold outlet as below,

$$\dot{m}_{\theta o} = \frac{\eta_v V_d N}{120RT_m} P_m \quad (122)$$

The equation above is the second state variable which is one of the outputs with the functions of engine speed N, manifold temperature T_m and manifold pressure P_m . If the outlet manifold temperature T_c is more than T_m ($T_c \geq T_m$), then the rate of air mass flow represented by $\dot{m}_a$ is given below,

$$\dot{m}_a = \left(\frac{\eta_v V_d N}{120RT_c}\right) P_m \quad (123)$$

Variable #3: The rate of CNG mass flow at the injector $\dot{m}_{cng}$

The CNG mass flow rate from the injector into the outlet of the manifold is one of the requirement variables. The rate of CNG mass flow as shown below is based on the factors of air/CNG ratio with stiochiometric environment,

$$\dot{m}_{cng} = \frac{1}{L_{SR}\lambda} \cdot \frac{1}{N} \cdot \frac{2}{CC} \cdot \dot{m}_a \quad (124)$$

Variable #4: The rate of mixture mass backflow $\dot{m}_{b.f}$

The rate of the mixture mass backflow as is due to the closing of the inlet valve which is given as below,

$$\dot{m}_{b.f} = \dot{\varphi}.\varsigma_f.m_a \quad (125)$$

Variable #5: The air and CNG mixture mass flow m_c

The mixture of air and CNG mass balance m_c flow as one of the important variables that enter into the swept volume V_d as given below,

$$m_c = m_a + m_{cng} - m_{bf} \quad (126)$$

Variable #6: The torque Γ_{eng} generated from the engine combustion

The torque Γ_{eng} is generated from the engine combustion with the functions of air mass flow rate $\dot{m}_a$, specific heat at constant volume c_v and engine speed N within a certain periodic time period $\delta\Delta T$.

$$\Gamma_{eng} = \frac{\dot{m}_a c_v(\delta\Delta T)}{2\pi N} \quad (127)$$

Variable #7: The crankshaft acceleration or deceleration $\dot{N}$ from engine dynamics

The crankshaft acceleration or deceleration is determined which due to the engine inertia I_c , engine torque Γ_{eng} and truck load Γ_{load} as shown in the equations below,

$$I_c\dot{N} = \Gamma_{eng} - \Gamma_{load} \quad (128)$$

$$or, \dot{N} = \frac{1}{I_c}\left(\Gamma_{eng} - \Gamma_{load}\right) \quad (129)$$

The design and development of the interconnected process flow has to justify that all the relevant variables in the whole system are connected according to the flow steps as shown in **Fig-20**. The state-space equation determines the engine torque Γ_{eng} variable from the engine dynamics as shown in the above equation is restructured as below,

$$\Gamma_{eng} = \left(\frac{\eta_v V_d}{120RT_c}\right)\left(\frac{c_v(\Delta T)}{2\pi}\right) P_m \quad (130)$$

Due to the disturbances from the friction and unpredictable behaviour of the truck's driver on the truck loading, the variable $\dot{N}$ as given in the equation above is restructured as follow,

$$\dot{N} = \frac{1}{I_c}\left(\Gamma_{eng} - K_f N + K_l.\Gamma_{load}\right) \quad (131)$$

Where, K_f and K_l are the dynamics friction coefficient and loading coefficient respectively. Then, it gives,

$$\dot{N} = \frac{1}{I_c}\left(\frac{\eta_v V_d}{120RT_c}\right)\left(\frac{c_v(\Delta T)}{2\pi}\right) P_m - \frac{K_f}{I_c} N + \frac{K_l}{I_c}.\Gamma_{load} \quad (132)$$

From the Variable #4, the state equation for the rate of mixture mass backflow from the combustion has to be regulated with respect to time. Therefore, the mass backflow rate is differentiate with respect to time and multiplied by the total mixed mass of air and CNG m_c that enter into the combustion cylinder. Therefore the equations above are differentiated and written as,

$$\dot{m}_{b.f} = \dot{\varphi}.\varsigma_f.m_c \quad (133)$$

$$\dot{m}_{cng} = \frac{1}{L_{SR}\lambda}\frac{1}{N}\frac{2}{CC}\dot{m}_a \quad (134)$$

From the Variable #5, the state-space equation for the total mass flow rate $\dot{m}_c$ is referred to loop model is written as,

$$\dot{m}_c = \dot{m}_a + \dot{m}_{cng} - \dot{m}_{bf} \quad (135)$$

then,

$$\dot{m}_c = \left(\frac{\eta_v V_d}{120RT_c} \cdot \frac{2}{L_{SR}.\lambda.CC} + \frac{\eta_v V_d}{120RT_c}\right) P_m - \dot{\varphi}.\varsigma_f.m_c \quad (136)$$

From the Variable #2, the state-space equation for the rate of filling and emptying manifold pressure is reconstructed and written as,

$$\dot{P}_m = \left(\frac{\eta_v V_d N}{120V_m}\right) P_m + \left(\frac{C_D P_o}{(RT_o)^{\frac{1}{2}}}.K_\gamma.K_{th}\right)\theta \quad (137)$$

From the Variable #6, the state-space equation for the output torque at the crankshaft can be determined from the input of the mixture air/gas flow rate $\dot{m}_c$ together with the input of air mass flow rate $\dot{m}_a$. So, the torque output generated from the engine combustion can be written as below,

$$\Gamma_{eng} = \frac{\eta_v C_v V_d \Delta T}{240\pi RT_c}\left(1 + \frac{2}{L_{SR} \cdot \lambda \cdot N \cdot CC}\right) P_m - \frac{C_v N \vartheta \varsigma_f}{2\pi} m_c \quad (138)$$

From the above formulations and revisions, the finally interconnected on the engine operational processing flow is shown in **Fig-20**.

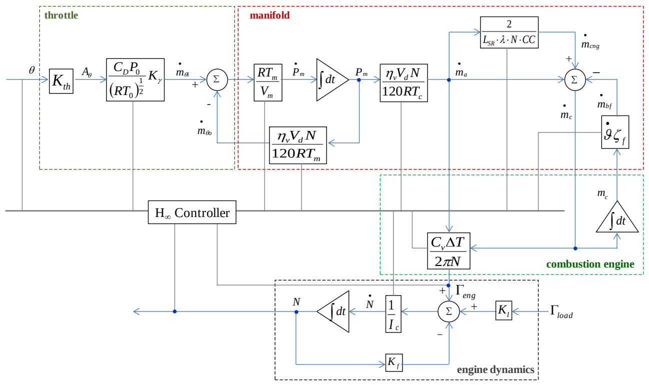

Figure 20: The Closed -Loop Control Model for the CCDE

From the interconnected closed-loop control flow as shown in the above figure, the overall state-space equations to represent the dynamical systems of the CCDE are developed as follows,

$$\dot{P}_m = \left(\frac{\eta_v V_d N}{120 V_m}\right) P_m + \left(\frac{C_D P_o}{(RT_o)^{\frac{1}{2}}} . K_\gamma . K_{th}\right) \theta \qquad (139)$$

$$\dot{N} = \frac{1}{I_c}\left(\frac{\eta_v V_d}{120 RT_c}\right)\left(\frac{c_v(\Delta T)}{2\pi}\right) P_m - \frac{K_f}{I_c} N + \frac{K_l}{I_c} . \Gamma_{load} \qquad (140)$$

$$\dot{m}_c = \left(\frac{\eta_v V_d}{120 RT_c} \cdot \frac{2}{L_{SR} .\lambda .CC} + \frac{\eta_v V_d}{120 RT_c}\right) P_m - \dot{\varphi} . \varsigma_f . m_c \qquad (141)$$

$$\Gamma_{eng} = \frac{\eta_v C_v V_d \Delta T}{240 \pi R T_c}\left(1 + \frac{2}{L_{SR} \cdot \lambda \cdot N \cdot CC}\right) P_m - \frac{C_v N \vartheta \varsigma_f}{2\pi} m_c \qquad (142)$$

Simulation Results - From the data gathered after few experiments as recorded in Appendix A, the power and torque against the speed of the diesel engine before and after the conversion are simulated, thus plotting the power and torque curve as shown in **Fig-21**.

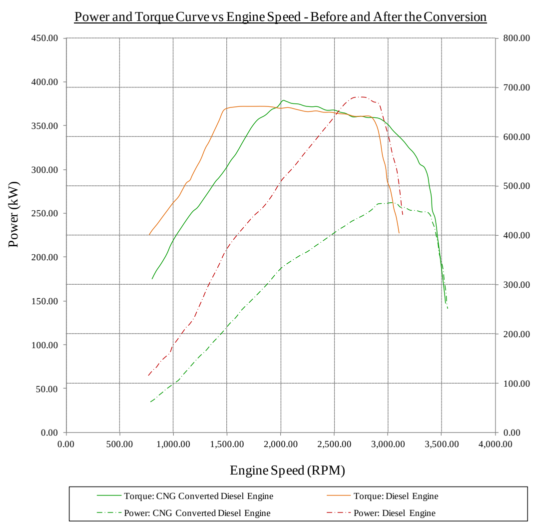

Figure 21: The simulation results of the power and torque versus the engine speed before and after the conversion

5. Conclusion

The opportunity for advance technology application is never been greater than to improve the truck's overall efficiency and performance thus reduce overall transportation cost. Numbers of applications are already established such as combustion systems, suspension systems, anti-lock braking systems and many more. Modern technology in automotive is becoming more and more important for the analysis and design of effective engine mechanism which uses various type of fuel or gasses into new engine or new engine modification systems or for whatever reason, the basis is for the purpose of improving overall efficiency, performance and fuel consumption then reducing overall cost.

These issues of flexibility in the conversion diesel engine are being solved by completely different approaches. As an example, what is the most effective mechanism to engage or disengage control systems in different operating modes with different type of engines? Should the engine and power-train management be centralized or divided into several decentralization units? The limitation of various systems within the diesel engines which have been developed by the manufacturer(s) is necessary to reconsider the underlying hardware and software control architecture. They have to be flexible in terms of the methods of development strategy for converting it into CCDE.

The application of robust design for the engine control system is motivated by the practical experience and "*trial-and-error*" efforts. It formulates the engine variables which are useful for the next step which is on the development of a control modelling and algorithm using H_∞ control method. It is based on the set of non-linear models for each variables of the engine system which then be linearized for state-space equations.

The solution deals with engine variables, unstructured uncertainties and employs the H_∞ closed loop shaping as suggested by the past various researches since 1988 done by the previous scientist. In this design, as the CNG is fed to the injector at the pressure between 3 to 5 bar above the intake manifold pressure and the pressure in the CNG tank is about 200 to 220 bar. The regulator is used to reduce the pressure just before the intake manifold.

Since the natural gas is decompressed into the tank with the pressure between 3 to 5 bars causes the CNG temperature to decrease. The further formulation of the state-space equations of the engine variables into the matrix form are demonstrated in the coming for the development of the robust control system using H_∞ control method. The development towards the justification of stability, controllability and observability of the system are also provided.

6. Symbols

γ = Specific Heat Ratio (c_p/c_v); [γ = 1.4 if CPR = 0.528; [γ = 1.3 if CPR = 0.546]

α = crank angle [deg]

$\frac{P_{T_c}}{P_O}$ = Critical Pressure Ratio (CPR)

$\dot{P}_m$ = rate of change of manifold pressure (bar/s)
$\dot{m}_{\theta i}$ = mass flow rate that goes into the intake manifold [g/s]
$\dot{m}_{\theta i}$ = mass air flow rate into the manifold (g/s)
$\dot{m}_{\theta o}$ = mass air flow rate from the manifold and goes into the cylinder for combustion (g/s) = $\dot{m}_a$
$\dot{m}_{\theta o}$ = mass air flow rate that goes into manifold, or air mass flow rate at the throttle outlet [g/s]
$\dot{m}_\theta$ = mass air flow rate at intake throttle
$\dot{m}_a$ = air mass flow rate (g/h)
$\dot{m}_g$ = gas mass flow rate (g/h)
m_m = mass flow rate into the combustion area

A_θ = throttle plate opened area, [mm^2]
C_D = discharge coefficient (is determined through experimental)
P_T = downstream pressure of throttle (it is assumed that the pressure is the same value at the minimum area or no pressure recovery occurs), [bar]
P_{atm} = ambient pressure
P_o & T_o = upstream pressure [bar] and temperature, [K]
m_c = mass of air inducted per intake stroke
m_{ss} = the steady-state value of m_a which can be determined by table look-up based on prototype testing
t_c = sample time
$\mathcal{K}_\gamma$ = Constant coefficient in which its value relies on Specific Heat Ratio γ and Mark Number M.
η_v = volumetric efficiency (measures the pumping performance of the cylinder and inlet valve which relies on the value of P_m.
θ_0 = Throttle Angle when the plate is at the closed position
AGR = air/gas ratio
c_p = Specific Heat at constant pressure [J/kg K]
c_v = Specific Heat at constant volume [J/kg K]
d = Throttle Shaft Diameter
D = Throttle Bore Diameter
N = engine speed [rad/s]
P_m = intake manifold pressure with respect to the
P_R = pressure ratio in which
R = Specific Gas Constant
T_m = manifold temperature (K)
V_d = engine swept volume [m^3]
V_m = manifold volume [m^3]
Υ = stationery isentropic flow function
R = Gas Constant
θ = throttle angle, [°]
λ = *air:fuel* ratio
λ_a = air ratio
λ_f = fuel ratio
λ_{cng} = CNG ratio
λ_o = reference air:fuel ratio
W_e = engine's work effectiveness
η_e = thermal efficiency effectiveness which is based on two variables - on the values of air λ_a and fuel ratio λ_f
L_{SR} = Stoichiometric Ratio
P_{inj} = injector pressure
P_{reg} = regulator pressure
P_m = manifold pressure
A_{noz} = injector nozzle minimum area
C_D = discharge coefficient
ρ_{cng} = density for CNG
C_R = Compression Ratio (10 ~ 12 is considered in the study)
R = Connecting Rod Crank Radius Ratio (3.5 ~ 4.0 is considered)
S = distance between the piston pin and crank rotation axis.
s_{st} = piston stroke.
A_p = piston surface area
D = bore
N = rotational speed of the crankshaft
$\bar{S}_p$ = Mean piston speed is more appropriate parameter as a function of speed compared to the crank rotational speed for correlating engine behaviour (The gas flow velocities in the intake and the cylinder are all scale with the mean speed.
C_c = number of combustion cylinder
V_d = engine total swept volume
M_t = turning moment acting on the crankshaft [N.m]
P = power [kW] or [N.m/s] or [J/s]
P_{atm} = atmosphere pressure [N/m^2]
P = pressure acts on piston [N/m^2]
A_c = piston area = $\frac{\pi D^2}{4}$ [m^2]
M_t = turning moment
M_p = pressure moment in the combustion cylinder
Γ_{eng} = engine torque [Nm]
Γ_{load} = total truck load [Nm]
Q_{HV} = fuel thermal /heating value
η_f = fuel consumption efficiency
η_v = volumetric efficiency
ρ_{ai} = air density inlet
λ = air:gas ratio → aims to keep the stoichiometric relative air:gas ratio, λ = 1 for the best of conversion efficiency, throttle control and ignition timing
m_c = mass of air that goes into the combustion area [g]
σ = spark advance degree before TDC [σ=30°]
I_c = effective engine inertia [kgm^2]
P_r = Power [kW]
M_v = mass of the vehicle [kg]
W_v = weight of the vehicle, [kN]
ρ_a = ambient air density [1.225kg/m3]
C_r = coefficient of Rolling Resistance, [$0.012 < C_r < 0.015$]
C_D = Drag Coefficient, [$0.3 < C_D \leq 0.5$]
A_V = Frontal area of the vehicle [m2]
S_V = Speed of the vehicle [km/h]

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