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A Review on Simulation methodology of an Internal Combustion Working Cycles

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ABSTRACT

Mathematical model for the engine simulation is important for prediction of the engine operating characteristics like in-cylinder pressure, temperature, power output, cycle efficiency and heat release as well as combustion characteristics like burning speed and flame speed for developing a tool for engine pre-mapping. By mathematical modeling it becomes easy to predict engine performance and deduce the performance parameters analytically. More realistic techniques, like thermodynamic models, could be employed if it is desirable to investigate the impacts of every parameter on combustion and pollutant emissions. Predicting engine performance without having to go through the hassle of building, increasing, testing, and evaluating the outcome is undoubtedly advantageous. Time and money should be saved as a result of modelling.

Keywords : Mathematical Modeling; Thermodynamic Models; Combustion; Pollutant Emissions

I. INTRODUCTION

Fluctuations of pressure, temperature, volume inside the engine of the given shape are provided by a comprehensive modeling and simulation tool. It computes the power, torque, and fuel consumption. Along with the intake and exhaust levels, it also forecasts the majority of the exhaust. The purpose of simulation of an internal combustion engine working cycles are to forecast engine working characteristics without testing it first and secondly to determine the performance metrics that could be challenging to gauge during testing.

The models fall into one of three categories: multi-dimensional multi-zone, single-zone and zero-dimensional. Multi-dimensional and zero dimensional models are referred to as detailed models and phenomenological or thermodynamic models, respectively.

Multi-dimensional models of combustion chamber space are based on the numerical solution of a set of governing equations, coupled with partial-differential equations integrated in 2 or 3 dimensional geometric grids.[1]

Two categories of models of thermodynamic can be distinguished: quasi-dimensional multi-zone models (QDMZ) and zero-dimensional single-zone models (ZDSZ).[9]

The combustion process is addressed in ZDSZ models by treating it as a method of adding heat that is empirical.

Simplified quasi-steady equations that describe the various engine in cylinder processesareused to create the QDMZmodels [13] which show multiple zones of charge in the cylinder throughout different operations, particularly the combustion phase.

II. MODELS FOR SIMULATION

The main types of models for the simulation of an engine are:

1. Models for the flow through the valve,
2. Models for heat transfer,
3. Models for the combustion process.

A. MODELS FOR FLOW THROUGH VALVE

Fig. below depicts the throat area at any valving, A_pt., or minimum port aperture area. Any flow via the curtain sections of the valves which are the side regions of a frustum of cone, can be either inflow or outflow. The lowest flow area, however, could occur at the inner port as the diameter dips. at the maximum valve lifts.[3]

the area of the valve curtain, A_t, for this specific design is frequently oversimplified, represented as a cylinder's side surface area with diameter d_{is} and height L as,

$A_t = \pi d_{is} L$

In the event that the valve lift is L, the valve curtain at the throat is represented by the cone's frustum, which is determined by the valve lift L, the side length covered is X, the valve seat angle, $\emptyset$, the inner seat diameters, dis and outer seat diameters dos, and the radius r.The side surface area of a frustum of a cone, A_s is,

$A_s = \pi[(d_{major} + d_{minor}) /2] * X$

Where X is the valve seat sloping side and d_{major}istopdiameter$andd_{minor}$ is bottom diameter. The greatest region of geometric gas flow through a single valve's seat for flow to or from the port is represented by this area, A_s.

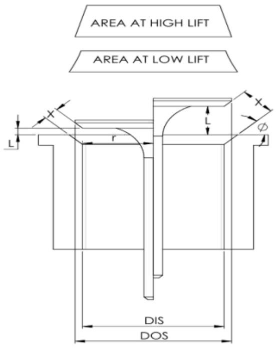

Fig. 1. valve curtain areas at low and high valve lifts

As seen in fig.1, the slanting length X has two values from which the gas flows. The valve lift has elevated over a lift limit L_{Limit} on the right and left, The value X is at right angle to the valve seat since the lift is so small.

Using basic geometry, this limiting value of lift is given by,

$$L_{Limit} = \frac{D_{OS} - D_{IS}}{2 sin\emptyset cos\emptyset}$$

Poppet valve lift for the beginning stage, where If $L_V \leq L_{Limit}$, A_t is

$$A_t = \pi * Lcos\emptyset * (D_{IS} + Lsin\emptyset cos\emptyset)$$

For the second stage of poppet valve lift, where If If $L_V > L_{Limit}$,

$$A_t = \pi * \frac{(D_{OS} + D_{IS})}{2} * \sqrt{[L_V - tan\emptyset * \left(\frac{D_{OS} - D_{IS}}{2}\right)]^2 + \left(\frac{D_{OS} - D_{IS}}{2}\right)^2}$$

The above equation becomes somewhat simpler if the seat angle is 45Υ, which is the conventional value. In this case, tan∅ equals unity.Heywood also provides the minimum flow area formula.He says that the instantaneous valve flow area is determined by the geometric properties of the valve head, seat, and stem in addition to the valve lift.

As lift increases, the flow area development happens in three stages.

Stage 1.

When $w/sin\beta cos\beta > L_V > 0$

The minimum area is $A_m = \pi L_V cos\beta(D_v - 2w + {L_V}/_2 \ sin2\beta)$

Stage 2.

At this moment, the smallest area of a right circular cone still consists of its slant surface, but it is no longer perpendicular to the valve seat.. The cone's base angle rises from (90-β)◦ to 90◦, the angle of a cylinder.

For this stage $(\frac{{D_p}^2 - {D_s}^2}{4D_m}^2 - w^2)^{0.5} + \text{w} \tan\beta \gg L_V > \frac{w}{sin\beta \ cos\beta}$

And $A_m = \pi D_m[(L_V - w \tan\beta)^2 + w^2]^{0.5}$

Stage 3.

when the lift is sufficiently large, the minimum flow area is no longer between the valve head and seat.

$$L_V > (\frac{{D_p}^2 - {D_s}^2}{4D_m}^2 - w^2)^{0.5} + \text{w} \tan\beta$$

Then, $A_m = \frac{\pi \ ({D_p}^2 - {D_s}^2)}{4}$

In actual use, valves are raised significantly, to a value of 0.35 to 0.4 times the inner diameter of seat.

B. GLOBAL HEAT TRANSFER MODELS

Under semi empirical models Woschni's correlationand Annand's correlationare involved.which predicated on the steady turbulent heat transfer's similarity law[13]. The formula is,

$h_t = 0.82 * B^{-0.2} * (P)^{0.8} * \omega_{mv}^{0.8} * T^{0.53} \ kw/m^2k$

Where, ω_{mv} is mean gas velocity affecting heat transfer[15]

$\omega_{mv} = [C_1 C_m + C_2 \left(\frac{V_s T_1}{pV_1}\right)(p - p_0)]$

For the gas exchange process, C_1= 6.18 &C_2 = 0,

For the compression process, C_1= 2.28 &C_2 = 0.

For the combustion and expansion process, C_1= 2.28 &C_2 = 3.24*10^{-3} ,

B is mean bore diameter,V_s is the displacement volume in cubic meters, p_0 is the pressure in MPa attained for the motoring, and C_m is the coefficient representing the mean piston speed. A specific moment in time where the temperature and pressure are known is indicated by the subscript 1. Despite being expressed in the manner of constant turbulent convective heat transfer, this formula incorporates all convection and radiation effects in lumped form.

Woschni's added a swirl term to the equation, gives the below value of the coefficient, for gas exchange process is $C_1 = 6.18 + 0.417 \ (\frac{C_1}{C_m})$, and for the remaining cycle,

Where, $C_m n_0$,n_0 is the paddle wheel rotational speed used in a steady swirl test rig.

Nusseltwas among the first to formulate an expression for the heat transfer coefficient in an I.C.Engine. The expression based on experimental observation, included radiation as well as convective effects[3], and it is

$$h = \underbrace{[0.99(P^2T)^{1/3} + (1 + 1.24W)]}_{\text{Convection}} + \underbrace{0.362[\{(T/100)^4 - (T_w/100)^4/\{T - T_w\}]}_{\text{Radiation}}$$

Where, h in kcal/m²hrK, w is average piston speed in m/sec

The radiation component of heat transfer is small compared to the convective component in I.C engine exhibiting normal combustion. Nusselt found that radiation heat transfer is about 5% or less of the total heat transfer.
accurate method for the calculation of heat transfer from the cylinder during the closed cycle is based on Anand's work.
The Anand approach's rationale sets it apart from other academics' heat transport theories by separating the terms for radiation and convection. Typical of the approach suggested for the heat transfer theory is Anand's expression for the Nusselt number Nu, which produces a standard derivation of the convection heat transfer coefficient C_h.Anandsuggested the below relation in terms of Reynolds and Nusselt No.
Nu = a $Re^{0.7}$
Where, a = 0.26, for the two stroke engines and is0.49, for the four stroke engines
The Reynolds no. is expressed as
Re $= \rho_{cy} C_p d_{cy} / \mu_{cy}$
$\mu_{cy}$is the viscosity of the cylinder gas is calculated as
$\mu_{Cylinder} = (7.457 * 10^{-6}) + (4.1547 * 10^{-6} * T_m) - (7.4793 * 10^{-12} {T_m}^2)$
T is in K, $C_p$is mean piston velocity,$C_p = 2\mathrm{N}L_{st}/60$,N is engine speed in rpm,L_{st}= cylinder stroke length
Reynold No. is obtained from Nu, Heat transfer coefficient by,
$C_h = C_k Nu / d_{cy}$
The thermal conductivity of the cylinder gas, denoted by parameter C_k, is assumed to be the same as that of air at the instantaneous cylinder temperature, T_{cy}
Thermal conductivity of gas
$C_k = 6.1944 * 10^{-3} + (7.3814 * 10^{-5} T_m) - (1.2491 * 10^{-8} * {T_m}^2)$
Anand also consider the radiation heat transfer coefficient, C_r
C_r= 4.25*$10^{-9}[T_{cy}^4 - T_{cw}^4]/(T_{cy} - T_{cw})$
Nonetheless, C_r value is substantially lower than C_h, to the point that C_r may be disregarded for the majority of engine cycle computations. The average temperature of the surfaces of the cylinder head, piston crown, and cylinder wall is T_{cw}

A. MODEL FOR COMBUSTION PROCESS

Modeling Approaches:
Richard Stone states that there are three methods for simulating combustion in order to add complexity, they are:
(i) Zero- dimension models or phenomenological models.[9][13]
[4]The best way to understand this method of combustion modeling is to utilize the specific model for spark ignition engines that Heywood et al. (1979) described. This model uses three zones among which two are of burnt gas:
(a) Unburnt gas
(b) Burnt gas
(c) Burnt gas that forms a thermal boundary layer, also known as a quench layer, next to the combustion chamber.
(ii) Quasi- dimensional models: [13]
By utilizing turbulence data as an input and Quasi-dimensional models make an effort to predict the burn rate data under the assumption of a spherical flame front geometry. This easy method provides the mass burning rate (dmb/dt) for spark ignition engines. There is room for improvement in this strategy, especially in terms of the turbulence. The flame front's magnitude in relation to the turbulence scale varies.
(iii) Multi- dimensional models: [12] [18]

Multi-dimensional model flow equations require the use of computational fluid dynamics (CFD) software or algorithms developed in other languages. It is necessary to incorporate submodels for processes like as ignition, combustion, flame-wall interactions, and emissions modeling.

Ducos et al.(1996) taken a number of combustion modeling methods and these include:

(a) Flamelet models, which on a micro scale considered the flame front as laminar,

(b) Eddy break-up models include a volumetric reaction rate; nonetheless, this leads to a flame speed that depends on the flame width. The position of the flame front is impractical if the flow is divergent since this leads to an unstable flame.

(c) Models of probability density functions rely on an assumed frequency distribution of turbulence intensity.

(d) The product of the local laminar burning velocity and the flame surface density is used in coherent flame models.

In this method, Woschni's (1967) correlation is used to forecast the heat transfer.

$Nu = a\, Re^{b}\, Pr^{c}$

The engine's shape and speed will determine the constants a, b, and c, however typical values are

a= 0.035, b= 0.8, c= 0.333

This kind of model is highly helpful in forecasting engine emissions in addition to engine efficiency.

Basic types of mass burning rate calculations:

Mass burning rates are calculated by using either the burning law or the phenomenological models.

Burning Law:[15]

there are two elements that determine how quickly reactants become products in S.I. engines. The flame front's transient surface area; and The flame front's velocity in relation to the nearby burned gas.[17] The position of the spark plug and the combustion chamber shape determine the factor.

The second factor is assumed to be dependent on engine operating conditions such as speed and is represented by mathematical functions of three different forms as follows: [15]

Model 1

The uniform rate formula given by,

$$X_b = {(\theta - \theta_s)}/{\Delta\theta_c}$$

$$dX_b = {d\theta}/{\Delta\theta_c} = 1/\mathrm{N}$$

Here, X_b is fraction of mass burned, dX_b is the mass fraction burned per crank angle, $\Delta\theta_c$ is the combustion duration in terms of crank angle, θ_s is at the start of ignition the crank angle and end of the combustion crank angle, θ_c is the combustion ends crank angle and N is the engine speed.

Analysis shows that in the very initial stages of combustion, the uniform rate formula is not practical. As a tiny sphere initially forms between the spark electrodes, 〖dX〗 _b⁄dθ ought to be extremely small for θ≈ θ_s. Model 1 does not meet this prerequisite. According to the uniform rate formula, the mass fraction burned curve in this model has a linear shape with respect to crank angle.

Model 2

The square law given by,

$$X_b = {(\theta - \theta_s)}/{\Delta\theta_c}$$

$$dX_b = 2[{(\theta - \theta_s)}/{\Delta\theta_c}]\, {d\theta}/{\Delta\theta_c}$$

$$= {2}/{N}\, {(\theta - \theta_s)}/{\Delta\theta_c}$$

Model 2 meets the requirement of a modest beginning expansion of the flame front, despite the fact that it predicts a continuous increase of the flame front, which is not the case in real engines. When the advancing flame front reaches the piston face or the cylinder head, the rate of change of the flame front area with time really stops abruptly. A square law describes the mass fraction burned plotted against crank angle in Model 2.

Model 3

An empirical law called the cosine burning law as follows:

$$X_b = \tfrac{1}{2}\{1 - \cos\{[{(\theta - \theta_s)}/{\Delta\theta_c}] * \pi\}\}$$

$$dX_b/d\theta = \sin\{[{(\theta - \theta_s)}/{\Delta\theta_c}] * \pi\} * ({\pi}/{2\Delta\theta_c})$$

Model 3 prooves the condition that $dX_b/d\theta$ is small both at the beginning and at the end of the combustion unlike Model 1 and Model 2, In this Model, the nature of the curve of mass fraction burned with crank angle follows " s " shape, i.e. it follows cosine law.

Blumberg states that the rate of combustion is typically determined by a functional relationship involving the present crank angle θ, the combustion duration Δθ_c, and the combustion start ▯_s. These include the Vibe function and the Cosine burning law. Vibe function reported by Blumberg is

$$dX_b/_{d\theta} = 1 - e^{-a_\theta(\frac{\theta-\theta_S}{\Delta\theta})^{m+1}}$$

Where, "a ", is parameter of efficiency and "m "is a parameter of slope.

The selection of any of the model is depends upon the extent of power to be influence by the model relationship.

B. Numerical Method [8] [11]

a set of linear equations numerically solved , these are:

1. Energy balance equation and mass balance equation.
2. First law of thermodynamics application for unburned and burned zones.
3. Equation of state.
4. Heat transfer correlation suggested by Woschni.
5. Chemical equilibrium correlation.

The computational method forecasting of the following parameters:

1. Unburned and burned zone pressure and temperature within the cylinder.
2. Chemical equilibrium considerations are used to determined concentrations of burned gases.

III. CONCLUSION FROM REVIEW

It is possible to estimate engine performance, emissions, and efficiency using any of the models. There is no direct correlation between the combustion chamber geometry and the Zero-dimensional and Quasi-dimensional models, they can be easily included into full engine models.As such, these models have utility in parametric research related to engine development. In situations when the combustion chamber geometry is crucial or highly variable, multi-dimensional models must be utilized. Instead of simulating the entire engine, combustion chambers are modelled using multi-dimensional models due to the high computing demands.

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