Modeling the Effect of Engine Parameters on Engine Temperature during ICE Operations.

Motey Festus and Dr. Essel Ben Hagan

Nomenclature

ICE Internal Combustion Engine

Abstract

Increasing manufacturing of more complex internal combustion engines has led to application of sophisticated computer programs in automobile industry for production and servicing. Some of these applications are modeling and simulation. These applications are also applied to heat transfer with reference to the objective of this article. A maximum of thirty three percent (33%) of total heat generated is transferred to coolant to be blown away. During operations of internal combustion engines, factors such as age, fuel type, pressure, environmental temperature, road nature, among others influenced rate of heat transfer within ICEs. Internal combustion engines are design in such a way that waste combustible gases are channeled through exhaust system. Components of the exhaust system are designed in such a way to withstand failures, stresses, vibrations and impact. In these engines, convection, radiation and conduction are modes of heat transfer through metal components and fluid. Sources of heat generation in these engines are friction within piston – cylinder liner compact and internal fuel combustion. Thus, internal combustion is the process of burning fuel with oxidizer exclusively in combustion chamber to produce heat energy. The methodology adopted involves modeling and simulation of heat equations. Thus, objectively these equations are analyzed to predict how these parameters will decrease heat generations. Graphical results obtained from applying this methodology indicate temperature reduction.

Key words: Manifold; Exhaust; Heat transfer; Fuel; Combustion.

INTRODUCTION

Researching into operations of internal combustion engines is receiving much attention or is on the rise. More developed methods or approaches such as modeling and simulation have been applied for these researches. All components as well as processes of internal combustion engines needed to be analyzed through modeling and simulation for efficient functioning. Ferguson and Kirkpatrick, (2001). Thus, mathematical equations which represent models of such components, systems and processes were built for analysis such as state estimation, control strategies, predictive accuracy, control techniques, fault diagnostics, among others. Approximately twenty five percent (25%) to thirty three percent (33%) of the heat produced during internal combustion is transferred to coolant. Furthermore, heat generated from air – fuel mixture ignition and combustion is absorbed or transferred to cooling fluid in motion in the engine.

This absorption mode may vary with reference to operating conditions such as number of piston rings, number of pistons, piston diameter, weight of connecting rod and engine temperature. Agrawal, Singh, Sinha and Shukla, (2004); Hahn and Ozisik, (2012). Cylinder wall, cylinder bore, cylinder head, exhaust valve seals and lubricant characteristics are some of engine components through which generated heat is transferred. Normally, modes of heat transfer in internal combustion engines or vehicle engines are conduction, convection and radiation. Forced convection accompanied by minor radiation are modes of heat transfer from combustible gases to metal components within internal combustion engines. Conduction is the major mode of heat transfer from cylinder metal to lubricant channels within automobile engines.

Another source of heat generation within internal combustion engines is friction between inner liner of metal cylinder and piston as well as piston rings. This effect of heating due to friction is reduced by application of recommended lubricants. In internal combustion engines, transfer of heat may be unsteady, nonlinear and non-uniform which are problems that are still being researched into. Liquid fossil fuel or underground fractionally distilled crude oil used by ICEs are gasoline or petrol and diesel. Dembroski, (2002); Sumin, (2013). Vapourized fossil fuel burnt in combination with an oxidizer or oxygen exclusively in a tight combustion chamber is referred to as internal combustion. Processes of internal combustion occurs intermittently resulting to chemical energy content of fuel being converted to heat. This heat energy is subsequently converted to mechanical energy and transferred through powertrain for automobile motion. This energy provides force on automobile components such as shafts, bearings, nozzles, alternators, pistons, connecting rods, valves, among others. Fuel tanks, connecting lines, fuel pumps, filters, fuel meter and atomizer are some basic components of internal combustion engine fuel system. Devices such as carburetors and injectors are used to spray fine fuel droplets into filtered air or gas for combustion in internal combustion chamber. The objective of this research is to model and simulate the effect of some engine parameters on engine temperature when ICE is operating. Finol and Robinson, (2006); Chan, Ordys, Volkov and Duran, (2013); Mirko, (2014).

REVIEWED LITERATURE

Fresh air – fuel mixture enters the internal combustion chamber through inlet duct. At intake manifold, the mixture splits before being directed into cylinders. Port walls as well as upper surface of valve head form a convergent geometry for experiencing changing velocity of internal flows. Sanli, Ozsezen, Kilicasian and Canakci, (2007); Ge, Chen and Sun, (2008). Heat transfer properties are associated with fluid movement around cylinder as well as coefficient of heat transfer vibration. Designs of various cylinder heads are such that they must withstand failures, heat flux, velocity of gases impact as well as ensuring accurate temperature distribution. Shapes of exhaust and intake valves are practically the same but differ in type of flow that passes through them. Andrew, Robert and Jhon, (2002); Ebrahimi, (2009); Janowski, Shayler, Robinson and Goodman, (2011). Thus, exhaust valve heads experience push due to high velocity of exhaust gas leading to formation of stagnation region. Also, high turbulence, high impingement, high heat transfer, high thermal loads as well as high thermal stresses are experienced by exhaust system, particularly by exhaust valve. Increasing temperature gradients may occur and soot formation is removed by

oxidation. Turbulence as well as forced heat convection at exhaust valve due to exhaust gases also occurs. Carlos, (2008); Arcoumanis and Kamimoto, (2009).

METHODOLOGY

The models or equations consist of empirical and algebraic equations which are implemented. Combustion processes in cylinder are represented by mathematical models. Relatively, modeling and simulation are less time consuming as well as less expensive as compared to experimental and other computational methods. Thus, these models provide simulation as well as modeling tools. This methodology will serve as tools and diagnostic algorithms for control engineers. Arnold and Polking, (2004); Lan and Srinivasan, (2009). Also, this methodology can be used to analyze linkage between angular speed of crankshaft, net engine torque, inertial variation of crankshaft assembly as well as piston pin offset. Furthermore, this methodology is used for modeling and simulation of thermodynamic cyclic vibrations, exhaust manifold heat transfer, mass flow rate as well as friction. Coombes, Hunk, Lipsman, Osborn and Stuck, (2000). Air – fuel flow rates, engine load and vibrational crankshaft speed are also factors considered for this methodology. Modeling of these models or equations involved writing programs in matlab and using values from tables of previous literature. Simulation involves opening simulink window and picking relevant blocks from simulink libraries. Inputs from tables of values in previous literature are inserted into respective blocks and these blocks are linked by signal lines. Bayraktar and Durgun, (2003).

EQUATIONS FOR MODELING

The following equations given below account for characteristics of heat transfer through exhaust manifold in internal combustion engines.

Burn Fraction Variation

Equation 1

$$X_b(\theta) = 1 - \exp. \left[-a \left(\frac{\theta - \theta^n}{\theta^d} \right) \right]$$

Source: Karkamkar, (2013).

This equation above is a model for determining the fraction of unburnt fuel.

Where

$X_b(\theta)$ = burn fraction as a function of crank angle

θ = crank angle

θ_s = spark timing

θ^d = duration of heat release

n = Weibe form factor

a = Weibe efficiency factor

$$a = 5$$

$$n = 3$$

$$\theta_s = -p^i/a$$

Cylinder Average Heat Flow

Equation 2

$$Q_{g,w}^* = f(bmep, \eta_{ICE}, Tw)$$

Source: Kanne, 2000.

The equation (110) one hundred and ten above represents the sub – model for cycle average heat flow.

Where

$Q_{g,w}^*$ = Heat flow in cylinder wall

f = Fuel quantity

η_{ICE} = Rotational speed of internal combustion engine

Tw = Torque in cylinder wall

Density of Inlet

Engine External Cooling

Equation 3

$$Q_{r,in}^* = (T_{r,in} - T_{r,out}) \times C_{pc} \times m^*$$

Source: Kanne, 2000.

The equation (113) one hundred and thirteen above represents a sub – model for engine external cooling (flow of heat to heat exchanger).

Where:

$Q_{r,in}^*$ = Heat flow of coolant entering the radiator

$T_{r,in}$ = Temperature of coolant entering the radiator

$T_{r,out}$ = Temperature of coolant leaving the radiator

C_{pc} = Heat capacity of coolant

m_r^* = Mass flow rate of radiator

Equation 115

$$\frac{dX_b}{d\theta} = (1 - X_b) \frac{anQ_{in}}{\theta_d} \left(\frac{\theta - \theta_s}{\theta_d} \right)^{n-1}$$

Source: Kanne, (2000)

The equation (115) one hundred and fifteenth above is a model for change of fuel burnt fraction with respect to crank angle.

Where

Q_{in} =heat supplied by fuel

X_b = burnt fraction at crank angle

n = Weibe force factor

a = Weibe efficiency factor

θ =crank angle

θ_s =spark timing

θ_d =duration of speed

Heat Transfer

Equation 116

$$Q_{c,eb}^* = z_c \times \alpha_c \times A_{eb} \times (T_{eng,m} - T_{eb})$$

Source: Kanne, 2000.

The equation (116) one hundred and sixteen above is a sub – model for heat transfer between the coolant and engine block.

Where:

z_c = Number of cylinder

α_c = Coolant heat transfer coefficient

A_{eb} = Area of engine block

$T_{eng,m}$ = Temperature of engine cylinder wall

T_{eb} = Temperature of engine block.

Inlet or Intake Temperature of Fluid

Equation 118

$$T_{ip} = T_{of} \left[1 + \frac{1}{2} (\gamma - 1) M^2 \right]^{-1/\gamma-1}$$

Source: Clive (2007)

The above equation is a model for temperature of inlet liquid fuel.

Where

T_{ip} = Inlet temperature of fluid

T_{of} = Temperature of fluid at 293k or 0°C

M = mach number = 0.1

γ = 1.4

MODELING OF EQUATIONS

Matlab Program for Equation (109) One Hundred and Nine

```
%SPARK TIMING ON COMBUSTION
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```
%BURN FRACTION VARIATION
```

```
%
```

```
function burnfractionvariation(xb)
```

```
%xb = 1 - exp(-a*(theta - thetas/thetad)^n)
```

```

%xb1 = burn fraction variation with reference to H
%xb1 = 1 - exp(-a*(theta - thetas*T/thetad).^n)*H
%where
%theta = Crank angle
%thetad = Duration of heat release
%thetas = Spark timing
% Xb is the burn fraction as a function of crank angle,
% n is the Weibe form factor
% a is the Weibeeffeciency factor
% H is number of pistons
n = -5.9;
H = 7;
a = 5;
thetad = 30;
thetas = 13.7;
theta = 0:2:60;
xb = 1 - exp(-a*(theta - thetas/thetad).^n);
xb1 = 1 - exp(-a*(theta - thetas/thetad).^n)*H;
plot(theta,xb,theta,xb1)
xlabel('theta(deg)');%x-axis label
ylabel('xb/xb1');%y-axis label
legend(' (1-exp(-a*(theta-thetas/thetad).^n)', ' (1-exp(-a*(theta-
thetas/thetad).^n)*H) ')
title('BURN FRACTION VARIATION')
end

```

The equation for “xb” has been modified by this research to “xb1” by incorporating H.

Matlab Program for Equation (110) One Hundred and Ten.

```

%%MODELING FUEL CONSUMPTION REDUCTION hundred and ten. N TERMS OF THERMOMANGEMENT
%MODEL FOR AVERAGE HEAT FLOW
%
function cylinderaverageheatflow(Qgw)
%cylinder average heat at flow = Qgw
%Qgw=f.*(bmep*R*Tw)
%cylinder average heat flow with reference to P = Qgw1
%Qgw1=f.*(bmep*R*Tw).*P
%P=piston diameter
%f=fuel quantity
%R=rational speed of internal combustion engine
%Tw=Torque in cylinder wall
%C=Load
%bmep=brake mean effective pressure
f=0:5:1500;
bemp=350;
Tw=207;
P=0.3;
R=504;
Qgw=(f.*(bemp*R*Tw));
Qgw1=(f.*(bemp*R*Tw).*P);
plot(f,Qgw,f,Qgw1)
xlabel('f(m^3)');%x-axis label
ylabel('Qgw/Qgw1');%y-axis label
title('cylinder average heat flow')
legend(' (f.*(bemp*R*Tw)) ', ' (f.*(bemp*R*Tw).*P) ')
title('AVERAGE HEAT FLOW')

```

end

The equation for “Qgw” was analyzed by previous author and modified by incorporating P to form “Qgw”.

Matlab Program for Equation (113) One Hundred and Thirteen.

```
%%MODELING FUEL CONSUMPTION REDUCTION IN TERMS OF THERMOMANGEMENT
%MODEL FOR ENGINE EXTERNAL COOLING
%(FLOW OF HEAT TO HEAT EXCHANGE)
function engineexternalcooling
%Heat flow from coolant entering the radiator = Qin
%Qin=(Trin-Trout)*Cpc*mv
%Heat flow from coolant entering the radiator with reference to J = %Qin1
%Qin1=(Trin-Trout)*Cpc*mv/J
%where
%J=number of piston rings
%K=connecting rod velocity
%W=piston weight
%Trin=temperature of coolant entering the radiator
%Trout=temperature of coolant leaving the radiator
%Cpc=heat capacity of coolant
%mv=mass flow rate of radiator
J=6;
Trin=385;
Trout=10:5:250;
Cpc=105;
mv=45;
Qin=(Trin-Trout)*Cpc*mv;
Qin1=(Trin-Trout)*Cpc*mv/J;
plot(Trout,Qin,Trout,Qin1)
xlabel('Trout(deg)');%x axis label
ylabel('Qin/Qin1');%y axis label
legend('((Trin-Trout)*Cpc*mv)','((Trin-Trout)*Cpc*mv/J)')
title('ENGINE EXTERNAL COOLING')
```

This research has remodeled equation “Qin” to equation “Qin1” by incorporation of J into Qin.

Matlab Program for Equation (115) One Hundred and Fifteen

```
%%OPTIMIZATION OF CONTROL LAW FOR AIR FUEL RATIO
%HEAT RELEASE RATE
%-----
function heatreleaserate(HRR)
%heat release rate =(HRR)
%HRR= a*n*Qin/thetad(1-Xb)*(theta-thetas/thetad)^n-1
%heat release rate with reference to H = HRR1
%HRR1=a*n*Qin/thetad(1-Xb)*H*(theta-thetas/thetad)^n-1
% H is number of pistons
% Xb is the burn fraction as a function of crank angle
% Qin is heat supplied by the fuel
% n is the Weibe form factor
% a is the Weibeeffeciency factor
% theta = Crank angle
% thetas = Spark timing
%thetad = Duration of heat release
H=7;
n=3;
a=5;
thetas = 12.1;
```

```

thetad = 2;
theta = 4;
Qin = 6:0.5:120;
Xb = 7.3;
%HRR = a*n*Qin./15*(1.-Xb)*(theta-thetas/thetad)^.3-1;
HRR = a*n*Qin./thetad*(1.-Xb)*(theta-thetas/thetad)^n-1;
HRR1 = a*n*Qin./thetad*H*(1.-Xb)*(theta-thetas/thetad)^n-1
plot(Qin,HRR,Qin,HRR1)
xlabel('Qin') %x-axis label
ylabel('HRR/HRR1') %y-axis label
legend(' (a*n*Qin./thetad*(1.-Xb)*(theta-thetas/thetad)^n-1 ', ' (a*n*Qin./thetad*(1.-Xb)*H*(theta-thetas/thetad)^n-1 ')
title('HEAT RELEASE RATE')
end

```

Equation “HRR” has been modeled by the previous author but this research has also remodeled it to equation “HRR1” by incorporation of H.

Matlab Program for Equation (116) One Hundred and Sixteen

```

%%MODELING FUEL CONSUMPTION REDUCTION IN TERMS OF THERMOMANGEMENT
%MODEL FOR HEAT TRANSFER BETWEEN THE COOLANT AND ENGINE BLOCK
%
function heattransfer
%Heat transfer between the coolant and engine block = htb
%htb=zc*(xc*Aeb)*(Tengm-Teb)
%Heat transfer between coolant and engine block with reference to N = htb1
%htb1 = zc*(xc*Aeb)*(Tengm-Teb)*N
%N = connecting rod weight
%zc = number of cycle
%xc = coolant heat transfer coefficient
%Aeb = area of engine block
%Tengm = temperature of engine inlet
%Teb = temperature of engine block
N=3.5;
zc=1:0.5:160;
Aeb=32;
Tengm=76;
xc=38;
Teb=58;
htb=zc.*(xc*Aeb)*(Tengm-Teb);
htb1=zc.*(xc*Aeb)*(Tengm-Teb)*N;
plot(zc,htb,zc,htb1)
xlabel('zc') %x-axis label
ylabel('htb/htb1') %y-axis label
legend(' (zc.*(xc*Aeb)*(Tengm-Teb)) ', ' (zc.*(xc*Aeb)*(Tengm-Teb)*N')
title('HEAT TRANSFER BETWEEN THE COOLANT AND ENGINE BLOCK')
end

```

Analyzing equation “htb” was done by the previous author but has been remodeled by this research to form equation “htb1” through the incorporation of N.

Matlab Program for Equation (118) One Hundred and Eighteen

```

%FLUID CHARACTERISTICS MODELING IN ICE (MODELS)
% INLET TEMPERATURE OF FLUID
%
function inlettemperatureoffluid(ITOF)
% Inlet temperature of fluid = ITOF

```



```

%ITOF=Tof.*(1+0.5*(y-1)*M^2)^-1/y-1
%Inlet temperature of fluid under conditions of P = ITOF1
%ITOF1=Tof.*(1+0.5*(y-1)*M^2)^-1/y-1*P
%Where
%P = piston diameter
%M = U/a,U = M*a
%M = march number
%U = vehicle cruising speed
%Tof = temperature of fluid
%a = sonic velocity
%y = gamma or ideal constant
%density at inlet port diameter (pop)=d*(1+0.5*(y-1)*M^2)^-1/y-1
%d = density of fuel
%d=2:0.5:600
P=0.3
y=1.4;
a=98;
U=297;
M=U/a;
Tof=5:0.9:2500;
ITOF=Tof.*(1+0.5*(y-1)*M^2)^-1/y-1;
ITOF1=Tof.*(1+0.5*(y-1)*M^2)^-1/y-1*P;
plot(Tof,ITOF,Tof,ITOF1)
xlabel('Tof');%x-axis label
ylabel('ITOF/ITOF1');%y-axis label
legend(' (Tof.*(1+0.5*(y-1)*M^2)^-1/y-1 ', ' (Tof.*(1+0.5*(y-1)*M^2)^-1/y-1*P')
title('FLUID CHARATERISTICS MODELING IN ICE(MODELS)')
end

```

The equation “ITOF” has been remodeled by this research to form equation “ITOF1” by incorporating P.

SIMULATION OF EQUATIONS

Simulation scheme for equation 109

Simulation scheme for equation 110

Simulation scheme for equation 113

Simulation scheme for equation 115

Simulation scheme for equation 116

Simulation scheme for equation 118

MODELING RESULTS

Modeling results for equation 109

Modeling results for equation 110

Modeling results for equation 113

Modeling results for equation 115

Modeling results for equation 116

Modeling results for equation 118

SIMULATION RESULTS

Simulation result i for equation 109

Simulation result ii for equation 109

Simulation result i for equation 110

Simulation result ii for equation 110

Simulation result i for equation 113

Simulation result ii for equation 113

Simulation result i for equation 115

Simulation result ii for equation 115

Simulation result i for equation 116

Simulation result ii for equation 116

Simulation result i for equation 118

Simulation result ii for equation 118

ANALYSIS OF RESULTS

The modeling results contain two coloured lines. Thus, blue lines are for modeling results of previous authors' equations and green lines are for modeling results after incorporation of these parameters. These lines are analyzed to observe variations due to influence of ICE parameters that were incorporated into previous equations. Thus, graphically, these green lines will show relative reductions or increases with respect to variables to be measured. Also, simulation results ii indicates increases or decreases after incorporation of these parameters before simulation. Modeling results of equations 109 and 118 show reductions in fuel consumption and inlet temperature of fuel respectively. Thus, the parameters have reduction effects after their incorporations. Results for remodeling of equations 110, 113, 115 and 116 after incorporation of these parameters shown by the green lines that heat flow, heat released as well as heat transferred have increased. Simulation results ii for equations 109 and 118 indicated reductions in rate of fuel consumption and inlet fuel temperature. After incorporation of these parameters, simulation results ii of equations 110, 113, 115 and 116 show reductions in temperature.

CONCLUSION AND RECOMMENDATION.

All graphical results indicated that incorporation of these parameters show reduction effect on temperature and rate of fuel consumption. However, more research must be done using other methods and parameters.

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