WATER-COOLED PETROL ENGINES: A REVIEW OF CONSIDERATIONS IN COOLING SYSTEMS CALCULATIONS WITH VARIABLE COOLANT DENSITY AND SPECIFIC HEAT

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ABSTRACT
A quick evaluation approach to internal combustion (IC) engine's radiator cooling system analysis is presented. A computer program in Microsoft Excel ™ is developed to assist in the calculations and analysis of engine cooling parameters such as fluid flow rate, effective cooling surface area, coolant passage tubes, and rate of heat dissipation when the density and specific heat at constant pressure vary due to the changing temperature. Derived curve-fitted correlations allow for proper estimates of fluid physical properties. Selection and application of a conservative heat transfer coefficient relationship based on the Nusselt relation, allows for determining an effective heat transfer area, taking into consideration the inter-relationships of all applicable parameters in the heat flow area. A method for estimating the number of tubes in the radiator for proper coolant circulation is shown. The positive side of using Water/ethylene-glycol mixture versus pure water used as coolant is discussed through a numerical example.

KEYWORDS: *Cooling system, Engine Cooling, Engineering with Microsoft Excel ™, Heat transfer in engines, Petrol Engine, Radiators, Spark-ignition engine, Water/Ethylene Glycol properties*

NOTATION
A_s = *Cross-sectional cooling surface area of radiator, m^2*
A_t = *Cross-sectional area of each tube in the radiator, m^2*
d = *Radiator tube diameter, m*
h_c = *Convective heat transfer coefficient, $J/s.m^2.K$ or $W/m^2.K$*
C_p = *Specific heat at constant pressure of pure water, J/kg.K*
C_{peg} = *Specific heat at constant pressure of water/ethylene-glycol mixture, J/kg.K*
m = *Mass flow rate of cooing water circulating through the system, kg/hr or kg/s*
n = *Number of tubes in radiator heat exchanger*
Q = *Heat flow rate or heat lost or dissipated to cooling water, J/hr*
v = *Velocity of flow, m/s*
T = *Temperature, K*
Y_{eg} = *Percentage of ethylene-glycol in Water/ethylene-glycol mixture solution*

GREEK LETTER
ρ = *Pure water density, kg/m^3*
ρ_{eg} = *Density of water/ethylene-glycol mixture, kg/m^3*
Δ = *Difference*

I. INTRODUCTION

The performance and efficiency of water-cooled spark-ignition or compression-ignition engines applied in motor vehicles or for stand-by power use relies on effective heat exchange between the engine and the surrounding medium. Performance here will require that there is proper carburetion, satisfactory oil viscosity and by implication correct clearances of the engine's static and moving parts [1]. Water-Cooling Systems consists of engine, (cooling jackets of the cylinder-block, cylinder head), radiator, fan, pump, engine temperature control devices, water distribution pipes and ducts and other elements [2, 3]. The engine parts of great concern are cylinder heads and wall liners, pistons, and valves. Carburetion problems could arise due to poorly vapourised petrol leading to some of the

combustion gases condensing on the cylinder walls causing a possible dilution of the oil in the pump and likely corrosion [4]. Proper moving engine parts will require that, lubrication of the engine parts is adequate allowing for the oil to flow freely at the right viscosity and temperature [4]. Engine combustion results in high temperature combustion gases. These high temperatures produced in the cylinders are transferred through the cylinder wall liners, cylinder heads, pistons and valves to the coolant by convection [5]. Rajput [4] in discussing such high temperatures, gives estimates as high as 1270 K – 1770 K and thus, exposing engine metal parts to such high temperatures will cause them to expand considerably, weaken them, result in high thermal stresses with reduced strength, safety concerns in overheated cylinders attaining flash temperature of the fuel thereby likely leading to pre-ignition, cause lubricating oils to evaporate rapidly leading to sticking pistons, piston rings, cylinders and eventual seizure and damage. A Cooling system is thus required to maintain stable operating temperature for the engine and prevent failure.

The radiator used to get rid of this heat, is a heat exchanger, which transfers the heat from the coolant to the air; the designs of which allow the coolant to flow through a bank of tubes exposed to the cross-flow of air, are of the two basic forms – *(a.)* cross-flow radiator, in which coolant flows from one side tank to the other, and *(b.)* down-flow radiator, in which coolant flows from a top tank to a bottom tank [6, 1].

The number of tubes is an important factor in the design of the ideal radiator in terms of the adequate surface cross-sectional area for effective cooling. The ideal radiator design should be compact, guided by minimum weight considerations, but, able to offer a large and effective cooling surface area; with coolant passages that should not be too small to avoid clogging by solid contaminants or scaling, with the attendant likely blockage restricting or limiting coolant flow leading to overheat of the engine, fouling and thermal corrosion, reduced endurance limit, and eventual stress corrosion cracking (SCC) [6, 1, 7]. In this paper a simple method for calculating the cooling system parameters that offers the cooling surface cross-sectional area for effective engine heat dissipation is presented. After, a discussion of methods of estimating the quantity of heat lost in the engine and radiator, methods of estimating the heat dissipation rate through application of appropriate heat transfer coefficient is evaluated. A limitation of engine and coolant temperature for safe operation is emphasized. A radiator sizing method is presented for effective heat transfer cooling area calculation. The behaviour of Coolant physical properties due to changing temperature is analysed, and derived appropriate mathematical relations presented for evaluation of coolant behaviour over a range of temperatures. The effects on engine performance, and by implication, thermal efficieny, are discussed.

II. APPLICABLE EQUATIONS

2.1 Heat Lost in Engine

Giri [1] expresses the total quantity of heat generated in the engine that is lost to the cooling water as in equation (1):

$$Q_{engine} = mC_p \Delta T \tag{1}$$

Since, there is a constant cooling air mass flow through the engine, radiator, fan and vehicle, from Energy Conservation or Continuity [2, 1]:

$$Q_{engine} = Q_{radiator} \tag{2}$$

2.2. Heat lost in Radiator – Determining the Heat Transfer Area

Rising [2] refers to the temperature drop through the radiator, as a temperature potential equal to the difference in temperature between the average water temperature and the inlet air temperature. Thus, the rate of heat dissipation from a radiator depends on the difference between the mean fluid or coolant (average coolant temperature, $T_{f\text{-}avg}$), and the surrounding air temperature, (ambient air temperature, T_a). The temperature drop across the radiator can be estimated by the relation:

$$\left(\frac{T_{f-out} + T_{f-in}}{2}\right) - T_a \tag{3}$$

Other considerations for effective heat dissipation are the number of tube rows and arrangement, coolant velocity and air velocity [6, 1]. Coolant velocity above 1.8 m/s can lead to waste of energy, [1]. The air velocity, V_a, can be estimated by the equation (3) [2]:

$$V_a = \frac{Radiator - Inlet - air - flow}{Surface - Area} \tag{3a}$$

Prockter [6] notes that the air velocity is more important than the coolant velocity for effective heat dissipation, since the purpose of a radiator is to transfer heat from the core fins to the air [8]. Since, air velocity is dependent on the vehicle speed, Meziere [8] further states that, vehicle speed should form a prime consideration in the design and sizing for an efficient radiator. Estimates for heat dissipation through the radiator are thus, dependent on the air velocity (equation 3a).

For tubular radiators, the following metric equivalent of the Prockter [6], heat dissipation in Kilowatts (per square metre per second per degree Kelvin) from a radiator as a function of the air velocity equation is:

$$Q_{radiator}[KJ/s]\left(1/m^2K\right) = 10^{-6}\left(6.379 - 0.473124R\right)\frac{V_a^{0.56}}{d^{0.63}} \tag{3a}$$

Where,

R = ratio of free air flowing surface area-to-water cooling surface area

V_a = air velocity in (m)

d = Tube inside diameter (m)

Radiator size and cooling surface heat transfer area can be estimated from the relation for the heat transmitted per unit time from a surface by convection [1, 4]:

$$Q_{radiator} = A_s h_c \left(T_f - T_s\right) \tag{3b}$$

This gives the cooling surface heat transfer area, A_s, as defined by equation (4):

$$A_s = \frac{Q_{radiator}}{h_c\left(T_f - T_s\right)} \tag{4}$$

Where, h_c = convective heat transfer coefficient, $W/m^2/K$

T_s = coolant side temperature or average wall surface temperature of radiator, K

$T_f = T_{f\text{-}avg}$ = mean fluid temperature between coolant inlet and outlet temperature, K

Giri [1], gives a range of values, (*353 K to 373 K*), for the operating temperatures of the *cylinder – block coolant fluid*. This implies a mean fluid or coolant temperature, of *363 K* .

Rajput [4], estimates, *25* percent – *to* - *35* percent of the heat supplied in the fuel is removed by the cooling medium, and that lost by lubricating oil and by radiation is 3-to-5 percent. This implies a total mean heat loss of *34* percent. This is a significant loss in engine power that could affect performance.

2.3. Effective Engine and Coolant Operating Temperatures

Proper engine performance is dependent on a certain safe and satisfactory operating temperature range. Some factors that can lead to adverse effects while operating outside this safe operating range can be separated into high and low. Consequences of high engine temperatures are reduced oil viscosity, leading to engine parts such as pistons not moving freely and likely sticking, causing loss of power, wear and eventual seizure. High temperatures can lead to burnt top cylinder gasket, and eventual metal-to-metal contact. Loss of oil lubricity can also lead to increased oil consumption.

Fuel vaporization is required for proper and complete fuel combustion, and at low engine temperatures, incomplete combustion can result, leading to excess fuel requirements for proper engine performance, due to improper vaporization. Improperly vaporized fuel can cool engine surfaces thereby causing condensation of combustion gases and water vapour formed during combustion, on cylinder walls, dilution of oil, soot formation, and the removal of oil film from cylinder wall surface – this can also cause wear of cylinder bore. Moisture from combustion can also mix with the unburnt hydrocarbon fuel forming acidic mixtures which can cause acidic corrosion. This can lead to engine damage.

At high coolant temperature, water may boil and evaporate, leading to likely oil film loss, and restricted parts movement since a certain lubricant temperature is required for proper oil flow. Damage to engine may also occur due to excessive coolant temperature, and by implication overheated engine as a result of detonation and pre-ignition. The maximum possible coolant temperature is limited by the coolants boiling point and the radiators heat transfer capacity, dependent on the number of fins, radiator surface area, and thickness, and the number of coolant tubes [4, 1].

Giri [1], has provided a range of recommended normal coolant temperatures, set at between, 345 K and 360 K, highlighting consequences for operating outside the recommended range; below 330 K, coolant temperature, metal degradation by rusting occurs rapidly, and below 320 K, likely water-oil mixing due to water from the combustion process accumulating in the oil. Furthermore, high rate of cylinder wall wear occur if coolant temperature is below 340 K.

III. ESTIMATING HEAT DISSIPATION RATE

3.1. Heat Transfer Coefficient

For water-cooled petrol engines, Nusselt investigated the heat transfer from the combustion gases to the combustion chamber walls, and expressed the overall heat transfer coefficient, h, in a formula made up of a convection heat transfer coefficient, h_c, and a radiation heat transfer coefficient, h_r. i.e., $h = h_c + h_r$, [9]. Stone [10], in reporting the work of Annand, notes that in spark-ignition engines, radiation may account for up to 20 % of the heat transfer, but it is usually subsumed into a convective heat transfer correlation. This view as corroborated by Rajput [4], in giving a figure of 95 % as the value of the amount of heat transfer by forced convection between the working fluid and engine components, and between the engine components and cooling fluids. Thus, the heat transfer between a working fluid and an inner surface in the engine is principally by forced convection; convection is also the mode of heat transfer between the engine and the outside environment [5]. Assessing for an appropriate value of heat transfer coefficient can pose some difficulty, [11].

A number of empirical mathematical models for estimating heat transfer coefficient, h_c, exist [11, 5]. These models are based on gas exchange velocity for estimating the value of the heat transfer coefficient. From experiments by Woschni, it has been posed that during intake, compression and exhaust operational conditions, mean values of the gas exchange velocity be taken as proportional to the mean piston speed, since the pressure changes inside the cylinder as a result of piston motion [5, 11]. Heywood [5], further notes that Woschni's additional efforts at estimating a value for the gas exchange velocity during pressure rise due to combustion, obtained by fitting a correlation integrated over the complete engine cycle, and time-average measurements of heat transfer to the coolant, resulted in a value of about 10 m/s. From the literature, four of such empirical models by Nusselt, Ashley-Campbell, Woschni and Eichelberg are:

Woschni:

$$h_c = 3.26 B^{-0.2} P^{0.8} T^{-0.55} Z^{0.8} \tag{5}$$

Where,
P = cylinder pressure, kPa

Ashley-Campbell:

$$h_c = 130 B^{0.12} P^{0.8} T^{-0.5} Z^{0.8} \tag{5a}$$

Where
B = cylinder bore diameter, m
P = cylinder pressure, atmosphere
Z = mean working velocity, m/s

Nusselt:

$$h_c = 1.155(1 + 1.24 C_m)(P^2 T)^{0.67} \tag{6}$$

Where, C_m = mean piston speed or gas exchange velocity ≈ 10 m/s, Heywood [5]
The Nusselt correlation is applied in the programmed example in this paper.

Eichelberg:

$$h_c = 2.1 (C_m)^{1/3} (PT)^{1/2} \tag{6a}$$

In equations (5, 5a, 6, 6a), $T = T_{f\text{-}avg}$ = mean fluid temperature. The mean fluid temperature is set at 350 K as per the work of Eichelberg, [10].

IV. COOLING AREA CONSIDERATION – ESTIMATING NUMBER OF TUBES

Giri [1], provides the following relation given in equation (7) for estimating the number of radiator tubes from the mass flow rate of circulating cooling water:

$$m = A_t v\rho = \left[\left(\frac{\pi d^2}{4}\right)n\right]v\rho \tag{7}$$

Resulting in number of tubes, *n*, given by, equation (8):

$$n = \frac{4m}{\pi d^2 v\rho} \tag{8}$$

V. PHYSICAL PROPERTIES OF WATER COOLANT

The physical properties of concern are the density, specific heat at constant pressure and the viscosity. Bosch [12] observes that the high specific heat of water providing for efficient thermal transitions of the radiator and engine materials, offer advantages in avoiding thermal overloads due to excessive component temperatures. Thus, the ease with which water flows due to its low viscosity and its ability to accept and release heat makes it an ideal coolant. Giri [1], however notes that its low boiling point of 373 K can cause loss of coolant and create voids or gas pockets in the water jackets which can lead to implosion and localized hot spots; on the other hand its high freezing point of 273 K can pose some problems to its efficient circulation as a coolant.

Water used for engine cooling is a mixture of drinking quality water and an antifreeze with additives, of which the recommended is ethylene-glycol. The additives act as inhibitors to protect against rust and corrosion [12]. Stone [10] gives typical concentrations of ethylene-glycol in engine coolant applications in the range of 25 %–to-60 % on a volumetric basis. Bosch [12] notes that, water-to-ethylene glycol ratios of 0.7: 0.3 and 0.5: 0.5 (30% - 50% ethylene-glycol concentrations) allows for raising the coolant mixture boiling point to allow for stable and higher engine operating temperatures of 120 ^oC at a pressure of 1.4 bar. Care is exercised in applying higher than the recommended ethylene-glycol concentrations because of likely increase of the freezing point leading to low coolant flow and less efficient heat transfer [1]. The table (1) compares some properties of pure water and water/ethylene-glycol mixture [10].

Table 1: Physical Properties of Water compare to Water/ethylene-glycol mixture

Property	Water	Ethylene-Glycol/Water mixture
Boiling Point at 1 bar	100 ^oC	111 ^oC
Specific Heat, kJ/kg K	4.25	3.74
Thermal conductivity W/mK	0.69	0.47
Density at 20 ^oC, kg/m^3	998	1057
Viscosity	0.89	4

Source: Stone [10]

5.1. Water Density

Water density varies as a function of temperature. For computer program applications, and design analysis purposes, the following modifications of Yaws, [13] relation can be used to estimate density of water given the temperature.

$$\rho = 1000\left\{anti\log\left(-0.4595 + 0.5622\left[1-\left(\frac{T}{647}\right)\right]^{0.2857}\right)\right\} \tag{9}$$

The results obtained from a simpler polynomial correlation, again useful for computer program codes, by Pramuditya, [14], applicable for the conditions - *P = 1 bar, 278.15 K≤T≤368.15 K,* and given as equation (9a), compares well with the Yaws [13] results.

$$\rho = 765.33 + 1.8142T - 0.0035T^2 \tag{9a}$$

The table (2) obtained from engineering toolbox [15], shows density of ethylene-glycol based water solutions at various temperatures:

Table 2: Density of Water/Ethylene-Glycol Solution in (kg/m^3)

	% by Volume of Ethylene-Glycol in Water/Ethylene-Glycol Solution						
Temperature, K	**25**	**30**	**40**	**50**	**60**	**65**	**100**
277.55	1048	1057	1070	1088	1100	1110	1145
299.85	1040	1048	1060	1077	1090	1095	1130
322.05	1030	1038	1050	1064	1077	1082	1115
344.25	1018	1025	1038	1050	1062	1068	1100
366.45	1005	1013	1026	1038	1049	1054	1084

Source: *Engineering toolbox* [15]

Within the bold highlighted window of interest, (*344.25≤T≤366.45*), the Density of Water/Ethylene-Glycol Solution for Y_{eg} percentage by volume of ethylene-glycol, is defined by the curve-fitted correlation:

$$\rho_{eg} = 0.00125Y_{eg} - 0.00054054T + 1.17358 \tag{10}$$

For the expanded range: (*277.55≤T≤366.45*), and percentage ethylene-glycol in solution for the range (*25 percent≤Yeg≤100 percent*), the following curve-fit of the density data in table (2) yields equation (10a):

$$\rho_{eg} = (-0.0000027T + 0.0020425)Y_{eg} - 0.0004162T + 1.13119 \tag{10a}$$

This curve fit has an error margin of about one percent.

5.2. Specific Heat at Constant Pressure of Water

A number of correlations have been proposed for specific heat at constant pressure, C_p, of water as a function of temperature, [13, 14]. The Pramuditya [14] polynomial correlation is as defined by the equation (11):

$$C_p = 28.07 - 0.2817T + 1.25\times10^{-3}T^2 - 2.48\times10^{-6}T^3 + 1.857\times10^{-9}T^4 \tag{11}$$

Applicable for *P = 1 bar, 278.15 K≤T≤368.15 K*

The table (3) obtained from engineering toolbox [15] shows specific heat of ethylene-glycol based water solutions at various temperatures:

Table 3: Water/Ethylene-Glycol Solution Specific heat at constant pressure, C_{peg} (KJ/kg.K)

	% by Volume of Ethylene-Glycol in Water/Ethylene-Glycol Solution						
Temperature, K	**25**	**30**	**40**	**50**	**60**	**65**	**100**
277.55	3.822	3.726	3.538	3.329	3.132	3.019	2.353
299.85	3.855	3.776	3.601	3.412	3.215	3.111	2.470
322.05	3.906	3.830	3.663	3.483	3.299	3.203	2.562
344.25	3.935	3.872	3.726	3.559	3.391	3.291	2.680
366.45	3.989	3.918	3.789	3.622	3.475	3.379	2.763

Source: Engineering toolbox[15]

Within the Bosch [10], and Giri [1], recommended range in the bold highlighted window of the table (3), i.e. 30 % – 50 % ethylene-glycol content of coolant mixture, the following correlation based on a curve fit of data within the window of interest, can be used to estimate the specific heat of water/ethylene-glycol mixture. For Y_{eg} percentage by volume of ethylene-glycol in water/ethylene-glycol coolant mixture within the temperature range (*344.25 K to 366.45 K*), which compares with the average operating temperatures of the cylinder – *block coolant fluid of* 363 K, the specific heat, C_{peg}, of water/ethylene-glycol coolant solution is:

$$C_{peg} = Y_{eg}[0.00004T - 0.028845] + 0.000872T + 4.02435 \tag{12}$$

And within the expanded temperature range (*277.55 K≤T≤366.45 K*):

For percentage ethylene-glycol in solution for the range (*25 percent≤Yeg≤50 percent*), the following curve-fit of the data in table (3) yields equation (12a):

$$C_{peg} = Y_{eg}[0.0000567T - 0.035457] + 0.000461T + 4.18705 \tag{12a}$$

For percentage ethylene-glycol in solution for the range (50 *percent≤Yeg≤100 percent*), the following curve-fit of the data in table (3) yields equation (12b):

$$C_{peg} = Y_{eg}\left[0.000026318T - 0.0277248\right] + 0.00198T + 3.8006 \qquad (12b)$$

VI. ENGINE THERMAL EFFICIENCY

Defined as the ratio of energy output to energy input, the engine thermal efficiency (which hovers within, 20-to-30 percent for petrol engines), is an indication of the amount of useful work produced compared to the total supplied fuel energy [1], and is given by the relation of equation (13). It is affected by the heat release rate [12]. Bosch [12] notes that the maximum heat release should occur at 5 – 10 degree crank angle, with too early a release likely to cause wall heat and mechanical losses, and too late a release, can cause a reduced thermal efficiency, and high exhaust gas temperatures. Stone [10] notes that, the predictions of heat transfer do not affect to a high degree, the engine output and efficiency.

$$\eta_T = \frac{Engine - Output - Energy}{Engine - Input - Energy} = \frac{Q_{out}}{Q_{in}} \qquad (13)$$

(Heat Lost) x (Quantity of Heat Supplied) = (Quantity of Heat Dissipated)
(Quantity of Heat Supplied) = Engine Energy Input = (Quantity of Heat Dissipated)/(Heat Lost)

A1	B	C	D	E	F
2	**Spreadsheet for Petrol Engine Cooling System Design Analysis Calculations**				
3	**Input**			**Output**	Pure water as
4				Water Temperature drop through Radiator (degK)	((C7+C8)/2)-C10
5				Water Temperature drop through Engine (degK)	C7-C8
6	Mass Flow rate through Water Jacket (kg/hr)			Specific Heat of Water (KJ/kg.K)	IF(AND(G7>344.25,G7<366.45),IF(AND(C17>=30,C17<=50),(28.07-(0.2817*F7)+(0.00125*(F7^2))-(0.000002*(F7^3))+(0.0000000018*(F7^4))),"out of range"))
7	Initial Engine Jacket Operating Temperature (degK)			Mean Coolant temperature (degK)	(C7+C8)/2
8	Final Engine Temperature due to Cooling Engine (degK)			Heat lost or Dissipated in Engine (KJ/hr)	C8*G6*F5
9	Average Wall Temperature of Radiator tube (degK)			Heat lost or Dissipated in Radiator (KJ/hr)	F8
10	Ambient Air Temperature (degK)			Heat lost or Dissipated in Engine (J/s)	(F8*1000)/3600
11	Radiator Tube Internal Diameter, D (mm)			Mass Flow rate of coolant water through radiator (kg/s)	F10/(F6*1000*F4)
12	Velocity of Water in Radiator tube (m/s)			Overall heat transfer coefficient, hc (kJ/s.m^2.K)	(1.155*(1+1.24*C16))*(((C15^2)*F7)^0.67)
13	Percent heat lost to coolant (%)			Heat transfer Area (m^2)	F10/(F12*(F7-C9))
14	Engine Thermal Efficiency (%)			Water density (kg/m^3)	1000*(10^(-0.4595+(0.5622*((1-(F7/647))^0.2857))))
15	Cylinder Pressure (atm)			Cross-sectional Area of each tube (m^2)	(PI()/4)*((C11/1000)^2)
16	Gas Exchange Velocity (m/s)			Number of radiator tubes	F11/(F15*C12*F14)
17	Percentage ethylene-Glycol concentration			Cross-sectional Area of flow (m^2)	F16*F15
18				Engine Power Input (kW)	(F10/(C13/100))/1000
19				Engine Power Output (kW)	(C14/100)*F18

Fig. (1): Microsoft Excel Computer Program with Cell Formulae for Pure Water as Coolant – note the out of range flagging

A1	B	C	D	E	G
2	**Spreadsheet for Petrol Engine Cooling System Design Analysis Calculations**				
3	**Input**			**Output**	Water/ethylene-glycol
4				Water Temperature drop through Radiator (degK)	((C7+C8)/2)-C10
5				Water Temperature drop through Engine (degK)	(C7-C8)
6	Mass Flow rate through Water Jacket (kg/hr)			Specific Heat of Water (KJ/kg.K)	IF(AND(G7>344.25,G7<366.45),IF(AND(C17>=30,C17<=50),((C17*((0.00004*(G7)-0.028845)))+(0.000872*G7)+4.02435),"out of range"))
7	Initial Engine Jacket Operating Temperature (degK)			Mean Coolant temperature (degK)	(C7+C8)/2
8	Final Engine Temperature due to Cooling Engine (degK)			Heat lost or Dissipated in Engine (KJ/hr)	C8*G6*G5
9	Average Wall Temperature of Radiator tube (degK)			Heat lost or Dissipated in Radiator (KJ/hr)	G8
10	Ambient Air Temperature (degK)			Heat lost or Dissipated in Engine (J/s)	(G8*1000)/3600
11	Radiator Tube Internal Diameter, D (mm)			Mass Flow rate of coolant water through radiator (kg/s)	G10/(G6*1000*G4)
12	Velocity of Water in Radiator tube (m/s)			Overall heat transfer coefficient, hc (kJ/s.m^2.K)	(1.155*(1+1.24*C16))*(((C15^2)*G7)^0.67)
13	Percent heat lost to coolant (%)			Heat transfer Area (m^2)	G10/(G12*(G7-C9))
14	Engine Thermal Efficiency (%)			Water density (kg/m^3)	1000*((0.00125*C17)-(0.00054054*G7)+1.17358)
15	Cylinder Pressure (atm)			Cross-sectional Area of each tube (m^2)	(PI()/4)*((C11/1000)^2)
16	Gas Exchange Velocity (m/s)			Number of radiator tubes	G11/(G15*C12*G14)
17	Percentage ethylene-Glycol concentration			Cross-sectional Area of flow (m^2)	G16*G15
18				Engine Power Input (kW)	(G10/(C13/100))/1000
19				Engine Power Output (kW)	(C14/100)*G18

Fig. (2): Microsoft Excel Computer Program with Cell Formulae with Water/Ethylene-Glycol Mixture as Coolant – note the out of range flagging

VII. NUMERICAL EXAMPLE

An investigator evaluating efficient heat dissipation in the cooling process of a car engine with better performance, argues that the variation of heat transfer coefficient as function of speed be based on the following empirical relationship:

$$h_c = 1.155\left(1 + 1.24C_m\right)\left(P^2T\right)^{0.67}$$

Use P = 1 atm, and 1.2 atm
The engine thermal efficiency is 25%.
The coolant water is to be cooled from 373 K –to-316 K in the engine block. Ambient inlet air temperature is maintained at 298 K, and the mass flow rate through the water jacket is 1250 Kg/hr. If the average wall temperature of the radiator is 310 K, with the mean velocity of water flow in the tube

being 1.35 m/s, and the resulting heat lost to the coolant equaling 25% of the heat supplied, write a small program to determine:

(a.) The required flow rate of the water in the radiator?

(b.) Assuming circular cross-section, the number of tubes to be used in the radiator core if tube internal diameter is 5.5 mm? (note, this simplified approach can be extended to models assuming elliptical cross-sections)

(c.) The engine power input and output?

7.1 Microsoft Excel Program Solution for Example

The solution to the numerical example is shown with the input and output in fig. (3) and fig. (4):

A1	B	C	D	E	F	G
2	Spreadsheet for Petrol Engine Cooling System Design Analysis Calculations					
3	Input			Output	Pure water	Water/ethylene-glycol
4				Water Temperature drop through Radiator (degK)	46.50	46.50
5				Water Temperature drop through Engine (degK)	57.00	57.00
6	Mass Flow rate through Water Jacket (kg/hr)	1250.00		Specific Heat of Water (KJ/kg.K)	4.13	3.87
7	Initial Engine Jacket Operating Temperature (degK)	373.00		Mean Coolant temperature (degK)	344.50	344.50
8	Final Engine Temperature due to Cooling Engine (degK)	316.00		Heat lost or Dissipated in Engine (kJ/hr)	294608.23	275937.29
9	Average Wall Temperature of Radiator tube (degK)	310.00		Heat lost or Dissipated in Radiator (KJ/hr)	294608.23	275937.29
10	Ambient Air Temperature (degK)	298.00		Heat lost or Dissipated in Engine (J/s)	81835.62	76649.25
11	Radiator Tube Internal Diameter, d (mm)	5.50		Mass Flow rate of coolant water through radiator (kg/s)	0.426	0.426
12	Velocity of Water in Radiator tube (m/s)	1.35		Overall heat transfer coefficient, hc (kJ/s.m^2 K)	775.54	775.54
13	Percent heat lost to coolant (%)	25.00		Heat transfer Area (m^2)	3.059	2.865
14	Engine Thermal Efficiency (%)	25.00		Water density (kg/m^3)	983.87	1024.86
15	Cylinder Pressure (atm)	1.00		Cross-sectional Area of each tube (m^2)	2.38E-05	2.38E-05
16	Gas Exchange Velocity (m/s)	10.00		Number of radiator tubes	13	13
17	Percentage ethylene-Glycol concentration	30		Cross-sectional Area of flow (m^2)	3.20E-04	3.08E-04
18				Engine Power Input (KW)	327.34	306.60
19				Engine Power Output (KW)	81.84	76.65

Fig. (3): Microsoft Excel Programmed Solution for Numerical Example

A1	B	C	D	E	F	G
2	Spreadsheet for Petrol Engine Cooling System Design Analysis Calculations					
3	Input			Output	Pure water	Water/ethylene-glycol
4				Water Temperature drop through Radiator (degK)	46.50	46.50
5				Water Temperature drop through Engine (degK)	57.00	57.00
6	Mass Flow rate through Water Jacket (kg/hr)	1250.00		Specific Heat of Water (KJ/kg.K)	4.13	3.87
7	Initial Engine Jacket Operating Temperature (degK)	373.00		Mean Coolant temperature (degK)	344.50	344.50
8	Final Engine Temperature due to Cooling Engine (degK)	316.00		Heat lost or Dissipated in Engine (kJ/hr)	294608.23	275937.29
9	Average Wall Temperature of Radiator tube (degK)	310.00		Heat lost or Dissipated in Radiator (KJ/hr)	294608.23	275937.29
10	Ambient Air Temperature (degK)	298.00		Heat lost or Dissipated in Engine (J/s)	81835.62	76649.25
11	Radiator Tube Internal Diameter, d (mm)	5.50		Mass Flow rate of coolant water through radiator (kg/s)	0.426	0.426
12	Velocity of Water in Radiator tube (m/s)	1.35		Overall heat transfer coefficient, hc (kJ/s.m^2 K)	990.16	990.16
13	Percent heat lost to coolant (%)	25.00		Heat transfer Area (m^2)	2.396	2.244
14	Engine Thermal Efficiency (%)	25.00		Water density (kg/m^3)	983.87	1024.86
15	Cylinder Pressure (atm)	1.20		Cross-sectional Area of each tube (m^2)	2.38E-05	2.38E-05
16	Gas Exchange Velocity (m/s)	10.00		Number of radiator tubes	13	13
17	Percentage ethylene-Glycol concentration	30		Cross-sectional Area of flow (m^2)	3.20E-04	3.08E-04
18				Engine Power Input (KW)	327.34	306.60
19				Engine Power Output (KW)	81.84	76.65

Fig. (4): Microsoft Excel Programmed Solution for Numerical Example with cylinder pressure increased to 1.2 atm.

VIII. CONCLUSION AND DISCUSSION FOR FUTURE WORK

This simplified approach to automobile engine cooling system analysis, accounts for the change in density and specific heat of the engine coolant due to variations in temperature. A comparison of pure water versus water/ethylene-glycol mixture solution shows that more heat is lost in engine and radiator when using pure water as coolant. There is not much change in number of radiator tubes. There is also not much difference in energy input and output as observed by [10]. Thus, there is only a little change in thermal efficiency. However, with increased cylinder pressure there is a reduction in the heat transfer area required (see Fig. (4). Since improved heat transfer to the air is attained, the hotter the radiator [1], it is more economical to operate at a higher pressure to achieve the highest temperature a radiator could operate at, which with pure water as coolant, it is the boiling point of water, 373 K.

In the analysis presented in this article, the assumption has been that nucleate boiling does not occur, and heat transfer is strictly by convection. However, Brace *et. al*. [16], notes that under high load engine conditions, nucleate boiling is likely to occur, and some modern engines have been fitted with

nucleate boiling sensing and control mechanism. Thus, for such engine types, the heat transfer mechanism is a combination of convective and nucleate boiling, and the heat transfer coefficient predicted by the Dittus-Boelter turbulent convection, and Chen boiling correlations [10]. This work can be extended to include heat transfer mechanism made up of effect of convective and of nucleate boiling. How this will affect the results for efficient cooling performance is not known. However, Robinson *et. al*. [17], observes that the Dittus–Boelter correlation is sensitive to variations in fluid physical properties and assumes that such variations with temperature are negligible, and estimates of fluid physical properties are evaluated at the bulk fluid temperature. This compares with this work, where fluid physical properties are based on the mean fluid coolant temperature. Furthermore, a limitation on the difference between the fluid coolant and surface wall temperature is pegged at 5.6^{o}C. It will be interesting to see how application of this compares with the earlier conclusion, based on the method adopted in this work.

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