 doi:10.20944/preprints201806.0048.v1

Article

Proposal of an Alternative UAV Engine Organic Coolants

Alon Davidy [1]*

[1] IMI Systems, POB. 1044/66 Ramat-Hasharon, 47100, Israel; alon.davidy@gmail.com
* Correspondence: alon.davidy@gmail.com;

Abstract: The demands for high power and low emissions put focus on energy efficiency when developing new engines. An important part is the UAV engine cooling system, which cools the engine structure below damaging temperatures. For high temperatures in the cooling system there is a risk that boiling occurs and while the initial nucleate boiling enhances the cooling effect, the subsequent Film boiling decreases the heat transfer drastically. It is important to understand the benefits and limitations of various fluids when designing a two-phase cooling system. This paper proposes an alternative organic coolant. Its normal application range is 15°C to 400°C, and its pressure range is from atmospheric to 10.6 bar. DOWTHERM A fluid possesses unsurpassed thermal stability at temperatures of 400°C. This fluid, in both the liquid and vapor form, is noncorrosive toward common metals and alloys. Even at the high temperatures involved, the equipment usually exhibits excellent service life. Original equipment in many systems is still being used after 30 years of continuous service. The numerical results, described in this paper, show the potential of DOWTHERM A as a good alternative for cooling a UAV engine. It will extend the working temperature range of the Diesel engine and will maintain the structural integrity of the piston, cylinder and the piston rings.

Keywords: UAV; Diesel engine; Piston; Two phase flow, fatigue cracking , Heat Transfer Analysis; Structural Analysis; Water; Boiling point; Two-phase flow, Organic coolant, DOWTHERM A organic fluid

1. Introduction

1.1 Description of Unmanned Aerial Vehicle (UAV) Engine Cooling System

The cooling system of the engine, shown in Figure 1, is designed for liquid cooling of the cylinder heads and ram-air cooling of the cylinders. The cooling system of the cylinder heads is a closed circuit with an expansion tank. The coolant flow is forced by a water pump driven from the camshaft, from the radiator, to the cylinder heads. From the top of the cylinder heads, the coolant passes on to the expansion tank (1). Since the standard location of the radiator (2) is below engine level, the expansion tank located on top of the engine allows for coolant expansion. The expansion tank is closed by a pressure cap (3) (with excess pressure valve and return valve). As the temperature of the coolant rises, the excess pressure valve opens and the coolant flows via a hose at atmospheric pressure to the transparent overflow bottle (4). When cooling down, the coolant is sucked back into the cooling circuit [1].

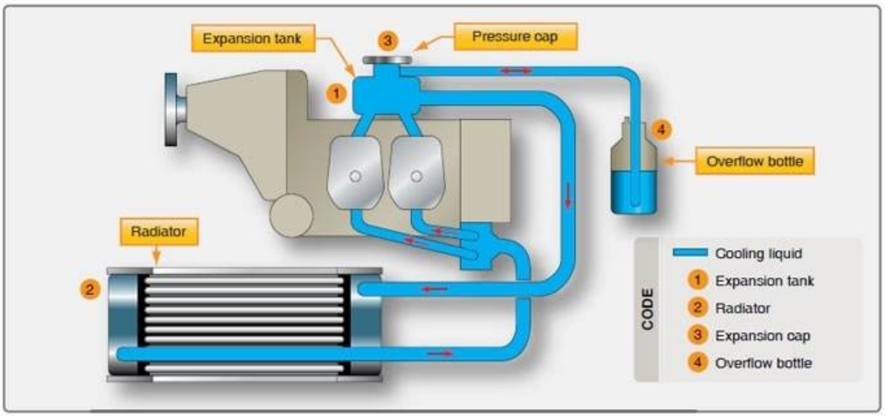

Figure 1: Rotax cooling system [1]

1.2 Engine Heat Transfer

The peak burned gas temperature in the cylinder of an internal combustion engine is of order 2500 K. Maximum metal temperatures for the inside of the combustion chamber space are limited to much lower values by a number of considerations, and cooling for the cylinder head, cylinder, and piston must be therefore be provided. These conditions lead to heat fluxes to the chamber walls that can reach as high as 10 MW/m² during the combustion period. However, during other parts of the operating cycle, the heat flux is essentially zero. In regions of high heat flux, thermal stresses must be kept below levels that would cause fatigue cracking (so temperatures must be less than about 400ºC for cast iron and 300ºC for aluminum alloys). The gas side surface of the cylinder wall must be kept below 180 ºC to prevent deterioration of the lubricating oil film [2]. Engine can be cooled by air [3] or by water [4]. In general, two-phase cooling has emerged as an attractive solution to meeting the high-temperature. However, it is important to understand the benefits and limitations of various fluids when designing a two-phase cooling system [5]. The common problem with conventional water based coolants is illustrated in Figure 1 [7-8].

(A) As the conventional coolant flows over a hot spot, it "boils off". In this "boiling off" area the heat is not being carried away as it is essentially being surrounded by air bubbles. The hot spot remains hot possibly leading to detonation or pre-ignition [7-8]. The coolant bubbles don't re-condense until they reach the Radiator (see Figure 1), further limiting the effectiveness of the system (see Figure 2).

(B) Demonstrates high engine load sustained "Nucleate boiling", causes vapor blanketing. Thus causes overheating of the engine parts.

Preprints (www.preprints.org) | NOT PEER-REVIEWED | Posted: 5 June 2018 doi:10.20944/preprints201806.0048.v1

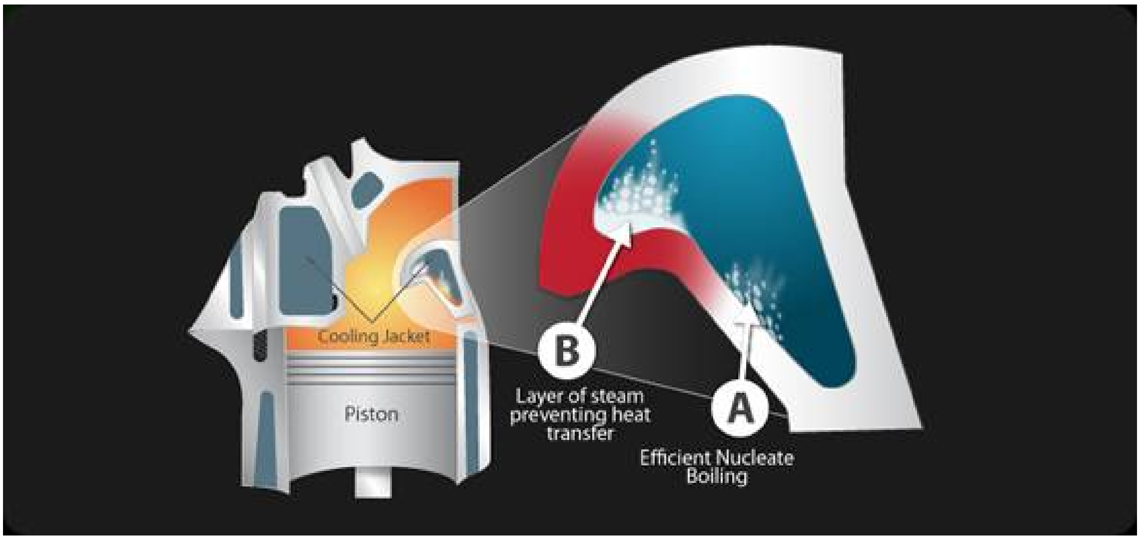

Figure 2: Engine coolant boiling [7-8].

The typical failures in Diesel engine are caused by overheating due to insufficient cooling. Figure 3 shows failure in the engine piston [9].

Figure 3: Severe overheat failure of engine piston [9].

Some of the key advantages of the organic coolant include [10]:

1) Low vapor pressure of the organic coolant, enabling high temperature operation near atmospheric pressure.

2) Coolant compatibility with low-cost materials and virtually no corrosion potential enabling use of plain carbon steel and aluminum.

The aim of this work is to seek a good alternative for cooling a UAV engine. This coolant will extend the working temperature range of the Diesel engine and will maintain the structural integrity of the piston, cylinder and the piston rings.

1.3 Critical Heat Flux

There are inherent benefits of two-phase cooling. A poorly designed two-phase cooling system, however, can fail catastrophically due to critical heat flux. Figure 4 shows the transition from natural convection to nucleate boiling (A), column and slug boiling (P), and critical heat flux (C) for water at 1 atm [1]. Critical heat flux (C) is marked by an excessive rise in device temperature that can result in overheating or physical melting of system components (E).

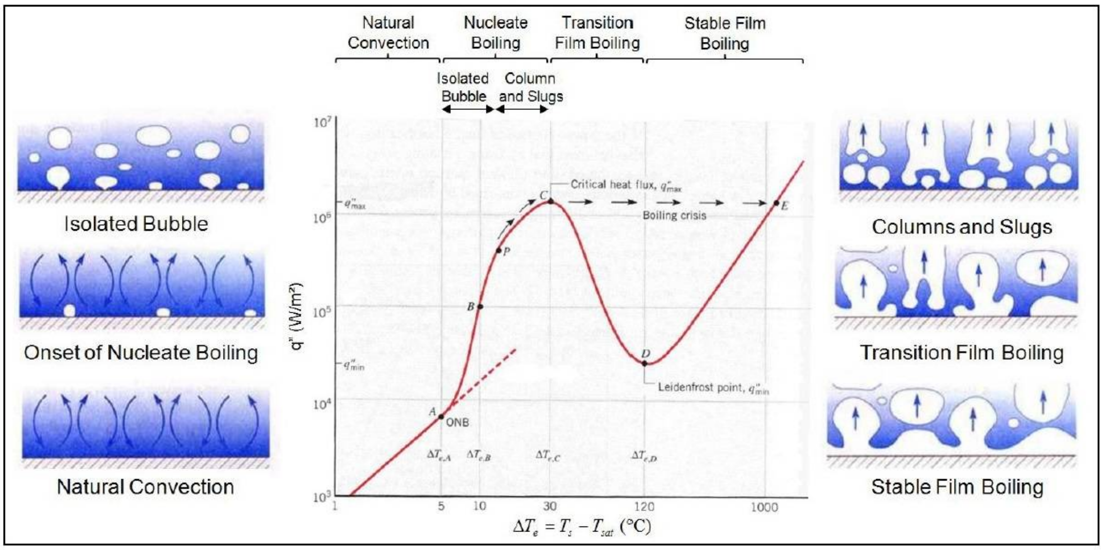

Figure 4: Two Phase failure mechanism – CHF at 1 atm [5].

At the point of dry-out the depletion of the liquid layer leads to a significant decrease in heat transfer at the surface which will result in an increase in piston wall temperature. But a part from the dry-out condition, a similar decrease in heat transfer can also occur at lower vapor qualities during the subcooled or saturated nucleate boiling regimes. This condition is achieved at high heat fluxes when rapid nucleation of bubbles result in formation of dry areas that start to cover more and more of the surface. Finally, a vapor film separates the liquid from the piston surface and film boiling has been initiated. This cause a drastic reduction in heat transfer since the vapor has a much lower thermal conductivity than the liquid. The condition is referred to as Departure from Nucleate Boiling (DNB) where the point of maximum heat flux is called the Critical Heat Flux (CHF). When designing a two-phase cooling system, CHF should generally be as large as possible to allow dissipation of engine high heat power density, if the heat power density if larger than the fluid CHF, overheating and potential failure of the piston are likely to occur[5].

1.4 Film Boiling

The heat transfer during film boiling can be estimated from the following expression from the study of Film boiling on a vertical surface by Tong & Tang [6]:

$$h = 0.62\left[\frac{k_v^3 \rho_v (\rho_\ell - \rho_v) g \left(h_{fg} + 0.5 c_{pv} \Delta T\right)}{\mu_v (T_w - T_{sat}) D}\right]^{\frac{1}{4}} \tag{1}$$

Where k_v is the thermal conductivity of the vapor $\left[w/(m \cdot K)\right]$, ρ_v is the vapor density $\left[kg/m^3\right]$, ρ_ℓ is the liquid density $\left[kg/m^3\right]$, g is the acceleration due to gravity $\left[m/s^2\right]$, h_{fg} is the latent heat of

 doi:10.20944/preprints201806.0048.v1

evaporation [J/kg], c_{pv} is the specific heat at constant pressure of the vapor [J/(kg ·K)], ΔT is the temperature difference between the wall and the coolant [°C] , μ_v is the viscosity of the vapor [Pa·s]. D is the diameter [m].

2. Materials and Methods

2.1 High Temperature Fluids

Table 1 shows a list of notable fluids that boil at temperatures exceeding 100 °C. While most of these fluids are flammable, extremely corrosive, explosive, or harmful (or all four), three of these fluids may be suitable as a high-temperature coolant—DOWTHERM (boiling point 258 °C), ethylene glycol (boiling point 197 °C), and propylene glycol (boiling point 187 °C). DOWTHERM A is a heat transfer fluid developed by Dow Chemical Company. It is a eutectic mixture of two stable compounds, biphenyl and diphenyl. These compounds have practically the same vapor pressure, so the mixture can be handled as a single compound rather than a binary mixture. Propylene Glycol and Ethylene Glycol are organic compounds that are widely used as automotive antifreezes [5].

Table 1: boiling points of various fluids at 1 atm [5].

Fluid	Boiling point (°C)
Acetic acid anhydride	139
Alcohol	97-117
Aniline	184
Butyric acid n	162
Carbonic acid	182
Dowtherm	258
Glycerin	290
Ethylene bromide	131
Ethylene glycol	197
Iodine	184
Jet fuel	163
Kerosine	150-300
Mercury	359
Napthalene	218
Nitric acid	120
Nitrobenzene	210
Nonane-n	150
Octane-n	125
Olive oil	300
Petroleum	210
Propionic acid	141
Propylene Glycol	187
Toluene	110
Turpentine	160
Xylene-o	142

2.2 Properties of DOWTHERM A Organic Fluid

DOWTHERM A is a heat transfer fluid is a eutectic mixture of two very stable organic compounds, biphenyl ($C_{12}H_{10}$) and diphenyl oxide ($C_{12}H_{10}O$). These compounds have practically the same vapor pressures, so the mixture can be handled as if it were a single compound. DOWTHERM A fluid may be used in systems employing either liquid phase or vapor phase heating. Its normal application range is 60°F to 750°F (15°C to 400°C), and its pressure range is from atmospheric to 152.5 psig (10.6 bar) [10]. DOWTHERM A fluid possesses unsurpassed thermal stability at temperatures of 750°F (400°C). The maximum recommended film temperature is 800°F (425°C). DOWTHERM A heat transfer fluid, in both the liquid and vapor form, is noncorrosive toward common metals and alloys. Even at the high temperatures involved, the equipment usually exhibits excellent service life. Original equipment in many systems is still being used after 30 years of continuous service. Steel is used predominantly, although low alloy steels, stainless steels, Monel alloy, etc. are also used in miscellaneous pieces of equipment and instruments. DOWTHERM A fluid has a freezing point of 53.6°F (12°C) and can be used without steam tracing in installations protected from the weather. Figure 5 shows the thermo-physical properties of DOWTHERM A.

The viscosity of DOWTHERM A fluid is low and changes only slightly between the melting point of the product and its top operating temperature. As a result, start-up problems are minimized. DOWTHERM A heat transfer fluid is a combustible material but has a relatively high flash point of 236°F (113°C) (SETA), a fire point of 245°F (118°C) (C.O.C.), and an auto-ignition temperature of 1110°F (599°C) (ASTM, E659-78). The lower flammable limit is 0.6% (volume) at 175°C, while the upper limit is 6.8% at 190°C. Vapors of DOWTHERM A fluid do not pose a serious flammability hazard at room temperature, because the saturation concentration is so far below the lower flammability limit [10]. It shall be mentioned that the coolant flows on the engine wall tunnels and it is not in direct contact with the combustion gases.

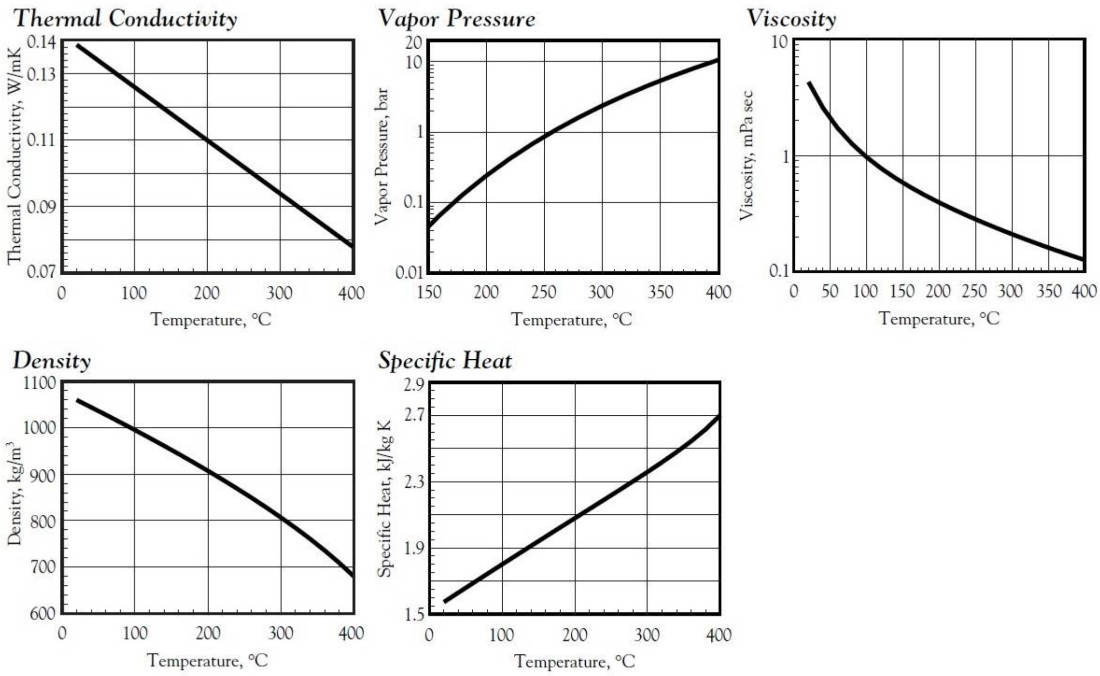

Figure 5: Liquid properties of DOWTHERM A [11].

2.3 Coupled Finite element analysis model

A finite element multi-physics COMSOL code was applied to generate 3D temperature and stress distribution in the piston. The diesel engine piston is studied at steady-state conditions, i.e. at a continuous engine speed and load. The combustion process at steady state produces cyclic pressure loads and a high constant temperature. These load conditions could yield a piston failure due to fatigue cracking, so-called high cycle fatigue cracks [12]. The convective coefficient of the cooling fluid was taken from Sieder & Tate correlation [13-14]:

$$Nu = 0.027\, Re^{0.8}\, Pr^{1/3} \left(\mu/\mu_w\right)^{0.14} \tag{2}$$

doi:10.20944/preprints201806.0048.v1

Where Nu is the Nusselt number, Re is the Reynolds number, Pr is the Prandtl number and μ is the viscosity of the coolant. It was assumed that the piston is made of Steel AISI 4340. The thermo-physical and thermomechanical properties of the steel are listed in Table 2 [12].

Table 2: Thermo-physical and thermomechanical properties of steel AISI 4340 [12]

Material Property	value
E	205E9 [Pa]
nu	0.28
rho	7,850 [kg/m³]
alpha	12.3e-6 [1/°c]
Cp	475 [J/(kg*°c)]
k	44.5 [w/(m*°c)]

2.3.1 Mechanical loads

The applied mechanical loads consist of the following two parts:

1) The peak combustion pressure which is applied on the combustion bowl, crown and top area is 13 MPa

2) The maximal inertia load at top dead center, TDC, that is, at the top of the stroke. The acceleration at TDC is calculated from [15]:

$$a = r\omega^2\left(1 + \frac{r}{\ell}\right)$$

Where r is the crank shaft radius (half of the engine stroke), ω is the angular frequency, r is the crankshaft radius and ℓ is the connecting rod length. The engine used in the investigation was a 2.0 liter turbocharged Ford diesel engine (Puma CD132 130PS HPCR) with four cylinders, high pressure common rail fuel system and four valves per cylinder. Table 3 contains the main characteristics of the engine [16].

Table 3: Operating conditions of the Diesel engine [16]

name	Expression	Description
n	4,000 [1/min]	Revolutions per minute
omega	n*2*pi	Angular velocity
stroke	0.086 [m]	Engine stroke
r	stroke/2	Crankshaft radius
conrod_length	0.160 [m]	Connecting rod length
lda	r/conrod_length	Radius-length ratio
pistonacc	omega^2*r*(1+lda)	Piston acceleration
P	130e5 [Pa]	Face load
tn	5e5 [Pa]	Input estimate of contact force
en	1.0e14 [N/m^3]	Penalty stiffness

2.3.2 Thermal Boundary conditions

The effects of the cyclic swing in surface temperature during the combustion cycle are small compared to the time-averaged temperatures. The major effect of the heat transfer on thermal stresses is therefore taken into account by time-averaged boundary conditions [12], that is, through constant convective boundary conditions.

The heat transfer coefficients on all boundaries are some typical values for a high speed diesel engine, as well as the bulk combustion gas temperature, engine oil temperature, and coolant temperatures (see reference [14]). The following thermal boundary conditions are applied:

1) The combustion gas temperature (900 °C) is applied to the combustion bowl and piston crown areas as an external temperature. The convective heat transfer coefficient is set to 500 W/ (m^2·°C) in these areas.

2) The outside of the piston is cooled by DOWTHERM coolant, whereas the inside is cooled by the engine oil, both at a temperature of 80 °C. Different heat transfer coefficients are applied on different boundaries and thereby reflecting the different cooling rates at each boundary. For example, a high heat transfer coefficient is applied to the bottom of the piston inside as this is the area where the piston oil cooling jet is directed. It was assumed that convective heat coefficient is 500 W/(m^2·K).

 doi:10.20944/preprints201806.0048.v1

3. Results

This section divided into two parts. In section 3.1 the thermal results are shown. In section 3.2 structural analysis results are presented.

3.1 Thermal Results

This section contains the validation of the numerical model (section 3.1.1). Section 3.1.2 deals with dry-out the depletion of the liquid layer.

3.1.1 Validation of the Numerical Model under Actual Operating Conditions of the Engine

The numerical model was evaluated on actual operating conditions of the Puma engine described in Section 2.3.1. The geometry of the model chosen for the investigation is illustrated in Figure 6. It shows the temperature field of the engine piston.

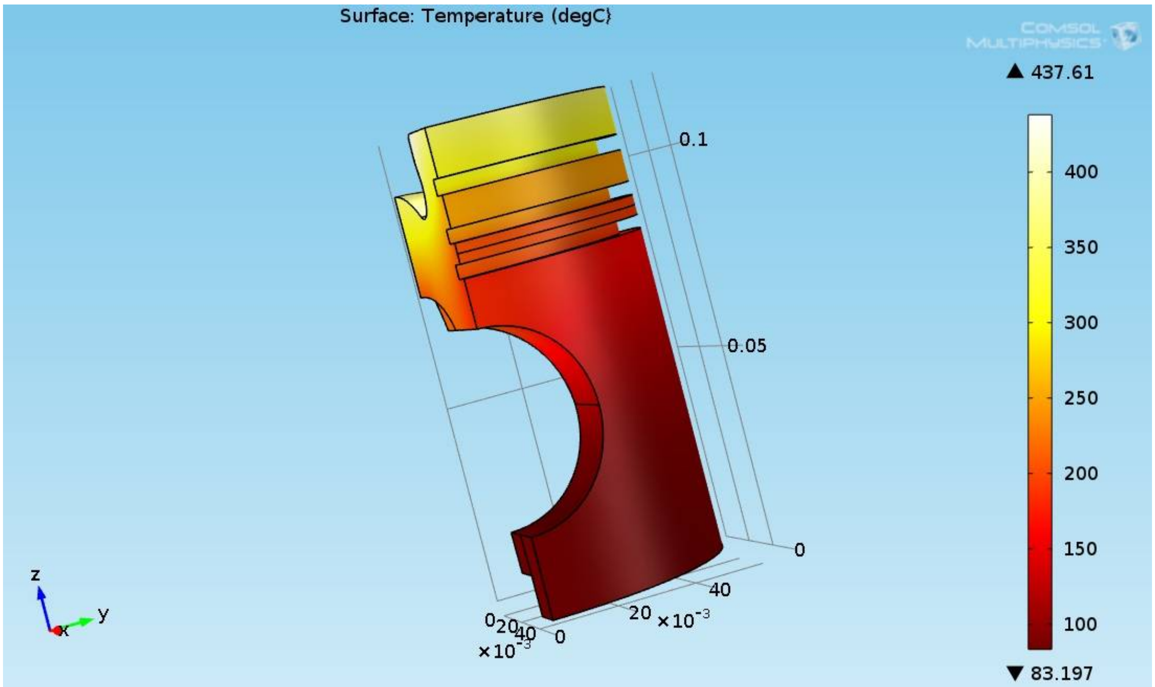

Figure 6: Calculated temperature field of the piston.

From figure 6, it can be seen that the maximal temperarture of the piston reaches to 438°C. This is because the piston bowl is exposed to intense heat transfer from the gaseous and soot mixture. The heat is transfered by convection and radiative heat transfer. In internal combustion engines the fluid motions are turbulent. Heat is transferred by forced convection between the combustion gases and piston bowl during induction, compression, expansion and exhaust processes. There are two sources of radiative heat transfer within the cylinder: the high temperature burned gases and the soot particles in the diesel engine flames. In the compression-ignition engine, most if the fuel burns in turbulent diffusion flame as fuel and air mix together. The flame is highly luminous, and soot particles are formed at an intermediate step in the combustion process. The radiation from soot particles in the diesel engine flame is about five times the radiation from the gaseous combustion products. The radiative heat transfer in diesel engine is not negligible. It contributes 20 to 35 percent of the total heat transfer and a higher fraction of the maximum heat transfer rate [1]. The calculated temperature of piston wall was compared to measurement results reported in [16]. The resulting temperature profile is shown in Figure 7. This contour plot shows the temperature distribution on the center plane of the cylindrical section.

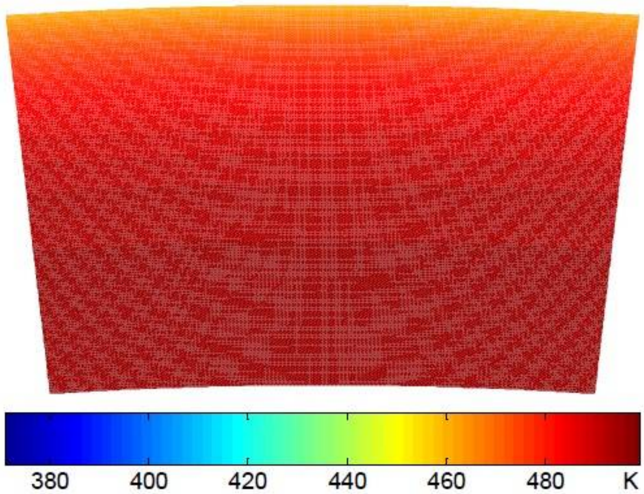

Figure 7: Temperature distribution in a cylindrical section of the cylinder wall [16].

From figure 6 it can be seen that that the temperature of cylinder section of the piston is 451 K (178°C). From figure 7 it can be seen that the measured tempearture at the upper section is 460 K. Thus, there is good agreement between the calculated tempearture obtained here and obtaineded by [16]

3.1.2 Thermal Analysis results obtained for the case of Water Film Boiling

The convective heat transfer coefficient for this case of water film boiling, has been calculated according to Eq. (1). Figure 8 shows the calculated temperature field of the piston assuming that water film boiling occurs. Under these circumtances liquid layer is depleted.

 doi:10.20944/preprints201806.0048.v1

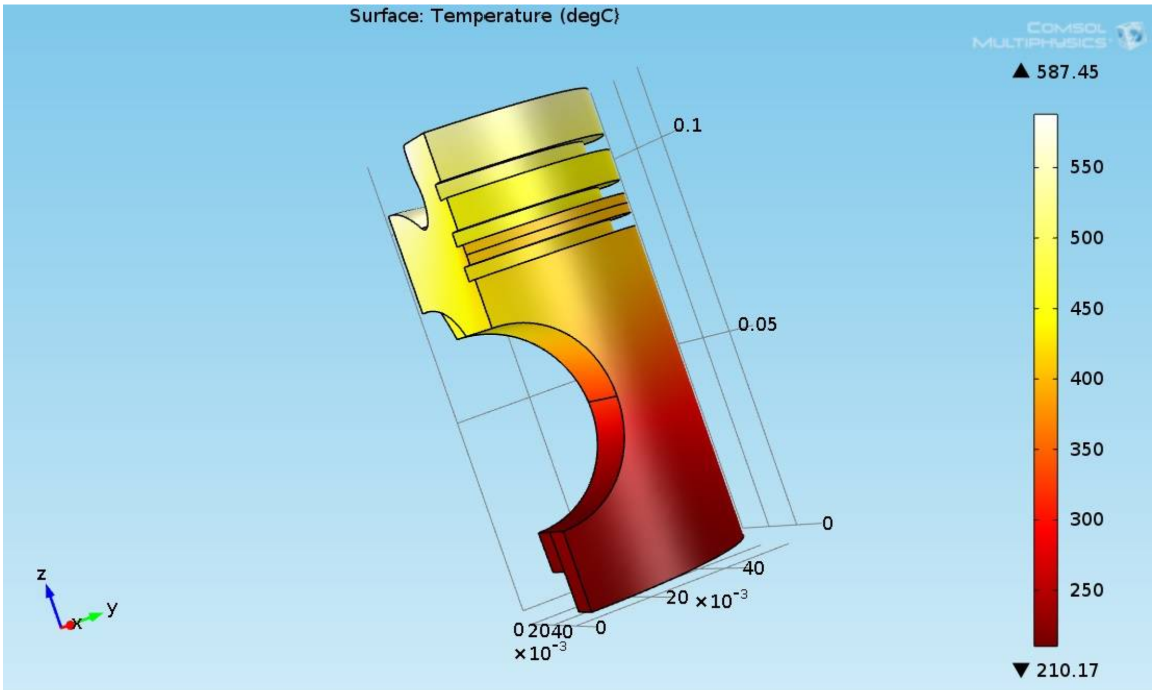

Figure 8: Calculated temperature field of the piston at dry out conditions.

It can be seen from Figure 8 that at the point of dry-out the depletion of the liquid layer leads to a significant decrease in heat transfer at the surface which will result in an increase in piston wall temperature. Finally, a vapor film separates the liquid from the piston surface and film boiling has been initiated. This cause a drastic reduction in heat transfer since the vapor has a much lower thermal conductivity than the liquid. The maximal temperarture of the piston reaches to 587°C.

DOWTHERM A fluid possesses unsurpassed thermal stability at temperatures of 750°F (400°C). The maximum recommended film temperature is 800°F (425°C). Since "DOWTHERM A" has higher boiling temperature than water, it will not boil under these conditions. The thermal results show the potential of DOWTHERM A as a good alternative for cooling a UAV engine. It will extend the working temperature range of the Diesel engine and will maintain the structural integrity of the piston, cylinder and the piston rings. It will prevent overheating of this piston region.

3.2 Thermomechanical results

Figure 9 shows the Von Mises stress distribution in the engine piston under actual engine operating conditions.

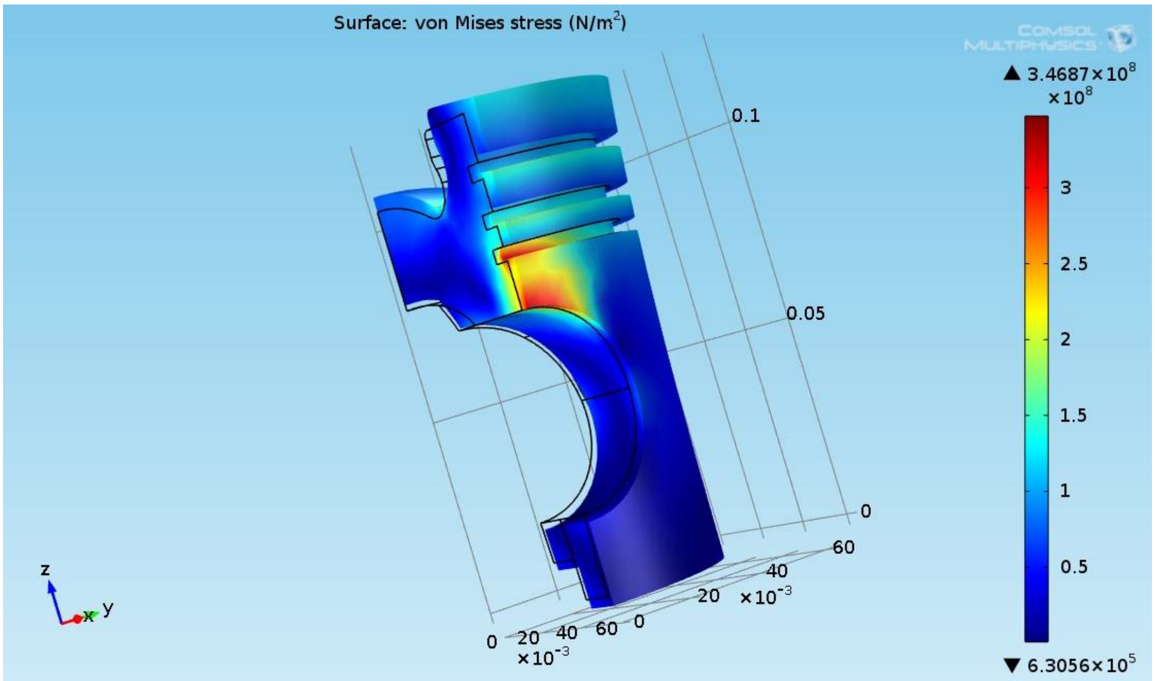

Figure 9: Von Mises stress field of the piston.

As can be seen from figure 9, the maximal stress reaches to 347 MPa. The tensile ultimate strength and yield strength of Stainless Steel 4340 are 745 MPa and 470 MPa respectively. At this location the temperature of the steel is less than 200ºC (392 ºF). Figure 10 shows the effect of temperature on the tensile ultimate strength (Ftu) and tensile yield strength (Fty) of AISI low alloy steels [17].

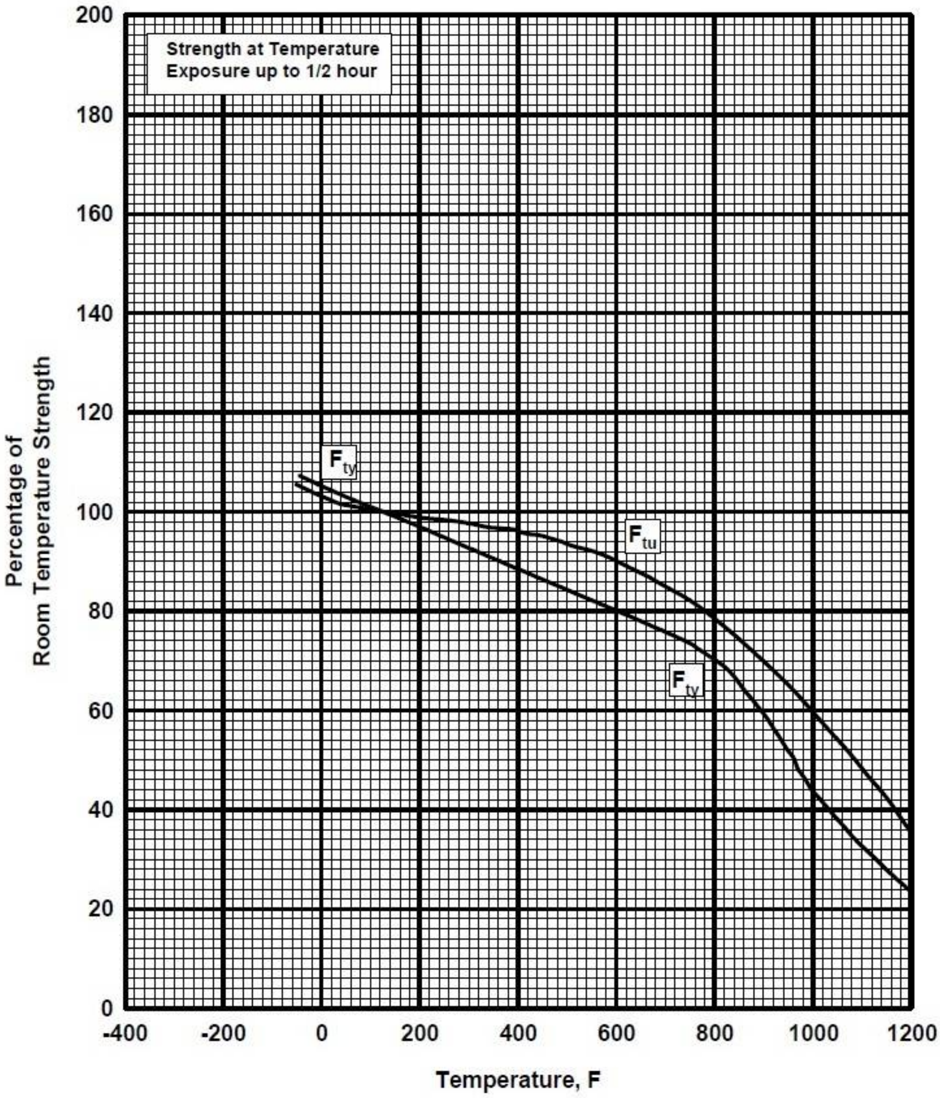

Figure 10: The effects of the temperature on the tensile ultimate strength (Ftu) and tensile yield strength (Fty) of AISI low alloy steels (all products) [17].

From figure 10, it can be seen that under actual operating of the engine, the decrease in the tensile yield strength is negligible at the position, where the maximal temperature has been found. At dry out conditions (Section 3.1.2), the decrease in tensile yield strength is about 40% at that point. The temperature of piston is 450 °C (or 842 °F).

The piston rings were also considered in this work. Piston rings are circular elastic elements with high expansive force. They have following main functions: to provide the sealing of the gases in the combustion chamber, to control the lubricating oil film of the cylinder walls and to be a transmitting element for the heat, from the piston to the cylinder. They are made from fluorocarbon. Figure 11 shows the plot the thermal stability of fluorocarbon piston rings [18].

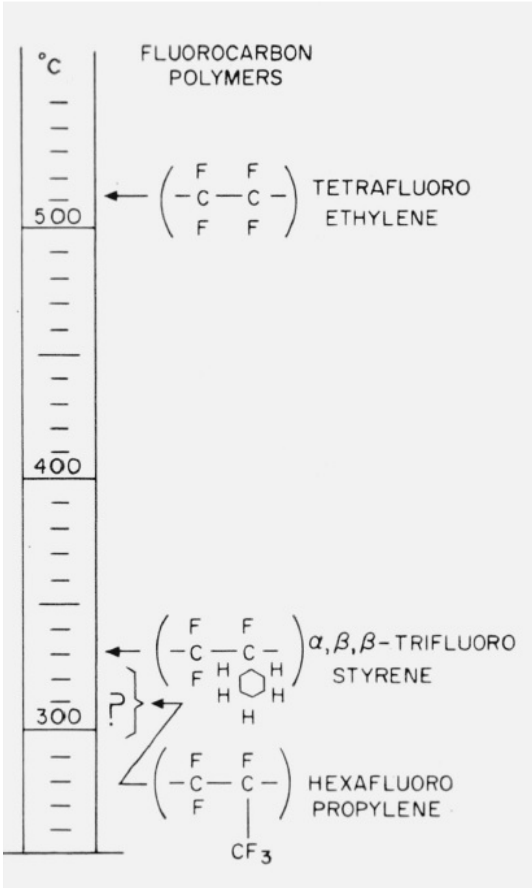

Figure 11: Thermal stability of fluorocarbon polymers at which rate of volatilization is 1 %/min [18].

From figure 11 it can be seen that at the maximal temperature of the piston, there isn't going to occur a thermal decomposition of the fluorocarbon piston rings.

4. Discussion

The demands for high power and low emissions put focus on energy efficiency when developing new engines. An important part is the engine cooling system, which cools the engine structure below damaging temperatures. For high temperatures in the cooling system there is a risk that boiling occurs and while the initial nucleate boiling enhances the cooling effect the subsequent Film boiling decreases the heat transfer drastically.

A finite element multi-physics COMSOL code was applied to generate 3D temperature and stress distribution in the piston. The diesel engine piston is studied at steady-state conditions, i.e. at a continuous engine speed and load. The combustion process at steady state produces cyclic pressure loads and a high constant temperature. These load conditions could yield a piston failure due to fatigue cracking, so-called high cycle fatigue cracks. The thermal simulation has been validated on actual diesel engine. The engine used in the investigation was a 2.0 liter turbocharged Ford diesel engine (Puma CD132 130PS HPCR) with four cylinders, high pressure common rail fuel system and four valves per cylinder. There is good agreement between the model results and the results taken from the dissertation. Additional Thermal Analysis has been performed for the case of water film boiling. It was found out that at the point of dry-out the depletion of the liquid layer leads to a significant decrease in heat transfer at the surface which will result in an increase in piston wall temperature. Finally, a vapor film separates the liquid from the piston surface and film boiling has been initiated. This cause a drastic reduction in heat transfer since the vapor has a much lower thermal conductivity than the liquid.

"DOWTHERM A" fluid possesses unsurpassed thermal stability at temperatures of 750°F (400°C). The maximum recommended film temperature is 800°F (425°C). Since "DOWTHERM A" has higher boiling temperature than water, it will not boil under these conditions. The results, described in the previous section, show the potential of DOWTHERM A as a good alternative for cooling a UAV engine. It will extend the working temperature range of the Diesel engine and will maintain the structural integrity of the piston, cylinder and the piston rings. It will prevent overheating of this piston region, plus the abrasion caused by the carbon materials, do result in excessive groove wear, and consequently could cause ring flutter. The proposed cooling method may be applied also in lightweight piston (i.e. made by Aluminum alloy).

5. Conclusion

High-temperature environments and device self-heating are pushing the thermal limits of automotive applications. In general, two-phase cooling has emerged as an attractive solution to meeting the high-temperature. However, it is important to understand the benefits and limitations of various fluids when designing a two-phase cooling system. According to DOW Company, the coolant "Dowtherm A" is applied in solar energy power plants, Oil and Gas industries and Heat Recovery installations. "Dowtherm A" have been discussed as one of the alternatives for cooling Internal Combustion Engine and electronic equipment. "Dowtherm A" has been analyzed as a working fluid for potential use in Rankine cycle power systems. In this work, the diesel engine piston is studied at steady-state conditions, i.e. at a continuous engine speed and load. The combustion process at steady state produces cyclic pressure loads and a high constant temperature. These load conditions could yield a piston failure due to fatigue cracking, so-called high cycle fatigue cracks. The convective coefficient of the water was calculated by empirical equation. The numerical tools developed in this work may be applied for other coolants in order to verify their cooling capability.

Conflicts of Interest: The author declares no conflict of interest. No funding sponsors had any role in the numerical analyses, or interpretation of data; in the writing of the manuscript, and in the decision to publish the results.

References

1. Aviation Maintenance Technician Power Plant Handbook Chapter 11. Light and Sport Aircraft Engine, http://content.aviation-safety-bureau.com/allmembers/faa-h-8083-32-amt-powerplant/sections/chapter-11.php
2. Heywood, J.B., Internal Combustion Engine Fundamentals, McGraw- Hill Book Company, 1988.
3. Poran, A. & Tartakovsky, L., Performance and emissions of a direct injection internal combustion engine devised for joint operation with a high-pressure thermochemical recuperation system, Energy, **2017**, 124, 214-226.
4. Tartakovsky, L., Gutman, M. & Mosyak, A., Energy Efficiency of Road Vehicles Trends and Challenges, In Energy Efficiency, Editors E.F. Santos Cavalcanti and M. Ribeiro Barbosa, 2012 Nova Science Publisher, Inc.
5. Sharar D., Jankowski, N. & Bar Cohen, A., *Two Phase Fluid Selection for High Temperature Automotive Platforms*, ARL-TR-6171, September 2012.
6. Tong, L.S. & Tang, Y.S., Boiling Heat Transfer and Two Phase Flow, Tayor & Francis, 1997.
7. Rotax Owner Assistance Network (ROAN), Rotax 912/914 Cooling System.

 doi:10.20944/preprints201806.0048.v1

8. MX-5 Miata Forum, https://forum.miata.net/vb/showthread.php?t=496912.
9. Piancastelli, L., The Different Failure Modes of Automotive and Aircraft Engine, 3rd Conference on Propulsion Technologies for Unmanned Aerial Vehicles, Samuel Neaman Institute Technion, Haifa, January 30, 2014.
10. Makens R.F., Organic Coolant Summary Report, IDO-11401, Idaho Operations Office, US Atomic Energy Commission, October, 1964.
11. DOWTHERM A, Heat Transfer Fluid, Product Technical Tool, http://www.dow.com/heattrans.
12. COMSOL 3.5a, Structural Mechanics Module – Model Library, 2009.
13. Sieder E.N., & Tate, G.E., *Ind. Eng. Chem.*, **1936**, 28, 1429-1436.
14. Kern, D.Q., Process Heat Transfer, McGraw Hill Book Company, 1950.
15. Borman G., Internal-Combustion Engine Heat Transfer, Mechanical Engineering Department, University of Wisconsin-Madison, 1987, USA.
16. Parra Finol C.A., Heat Transfer Investigations in a Modern Diesel Engine, A thesis submitted for the degree of Doctor of Philosophy University of Bath Department of Mechanical Engineering, February 2008.
17. Department of Defense, Metallic Materials and Elements for Aerospace Vehicle Structures, MIL-HDBK-5H, 1998.
18. Wall, L.A. & Straus, S., Pyrolysis of Fluorocarbons Polymers, Journal of Research of the National Bureau of Standards – A Physical and Chemistry, 65, 3, May-June 1961.