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# The Potential of Using Organic Fluid as Piston Diesel Engine Coolant

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***Abstract:*** *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. This work briefly analyzes the heat transfer potential of using organic fluid as a Diesel engine piston coolant.*
***Keywords:*** *Aircraft, Diesel Engine, Piston, Steel, Heat Transfer analysis, Structural analysis, Water, Boiling point, Two phase flow, Organic coolant, Dowtherm A*

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## I. Introduction

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 [1]. The common problem with conventional water based coolants. 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 [2]. The coolant bubbles don’t re-condense until they reach the Radiator, further limiting the effectiveness of the system (see figure 1) [3].

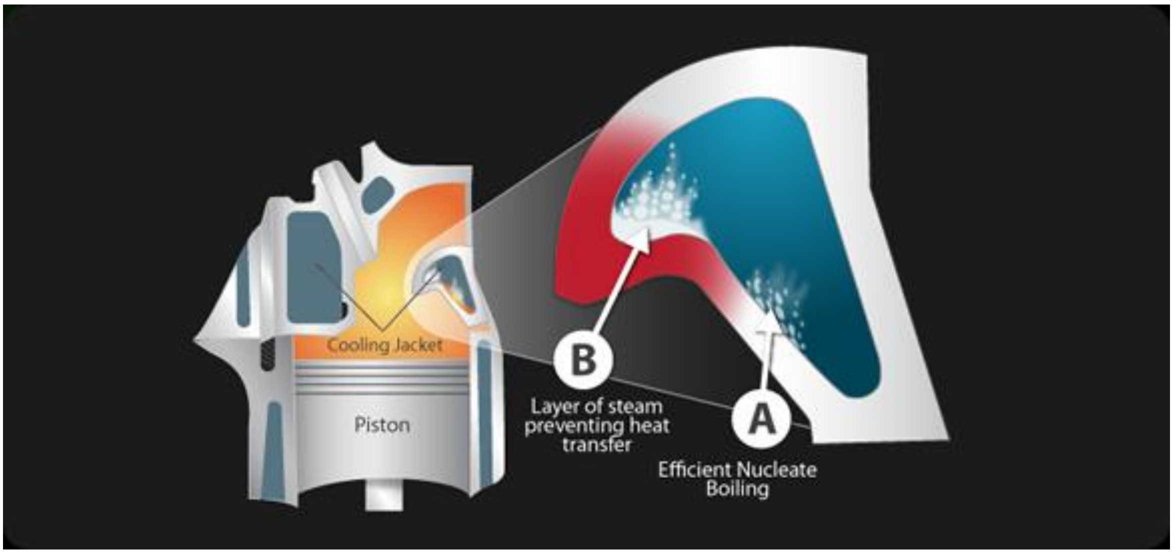


**Fig. 1: Engine coolant boiling [3].**

The typical failures in aircraft engine are caused by overheating due to insufficient cooling. Figure 2 shows failure in the engine piston [4].

**Fig. 2:** Severe overheat failure of engine piston [4].

Some of the key advantages of the organic coolant include [5]:
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.

**1.1 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 3 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). When designing a two-phase cooling system, CHF should generally be as large as possible to allow dissipation of high power density electronics; if the electronic power density if larger than the fluid CHF, overheating and potential failure are likely. Typically, critical heat flux can occur at wall superheats (temperature rise over the saturation temperature of the fluid) between 15 and 25 °C, placing a lower limit on the fluid boiling temperature for a given electronic device temperature before failure occurs [1].

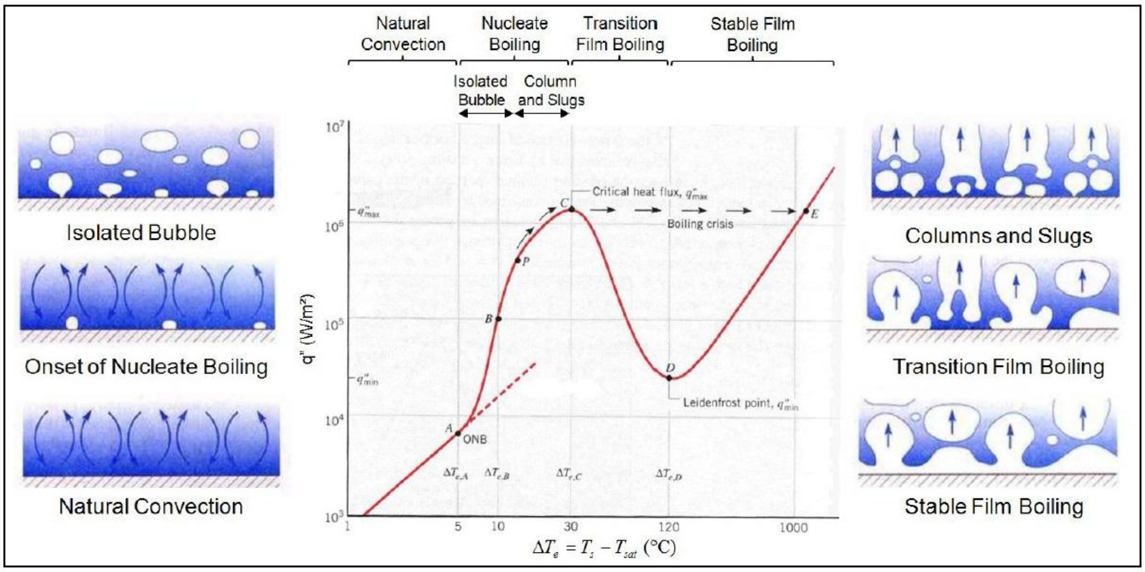


**Fig. 3:** Two Phase failure mechanism – CHF at 1 atm [1].

### 1.2 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 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 [1].

**Table 1:** Boiling points of various fluids at 1 atm [1].

| 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 |

### 1.3 Thermo-physical properties of Dowtherm coolant 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) [7]. 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 53.6ºF (12ºC) and can be used without steam tracing in installations protected from the weather. Figure 4 shows the thermo-physical properties of Dowtherm A [6].

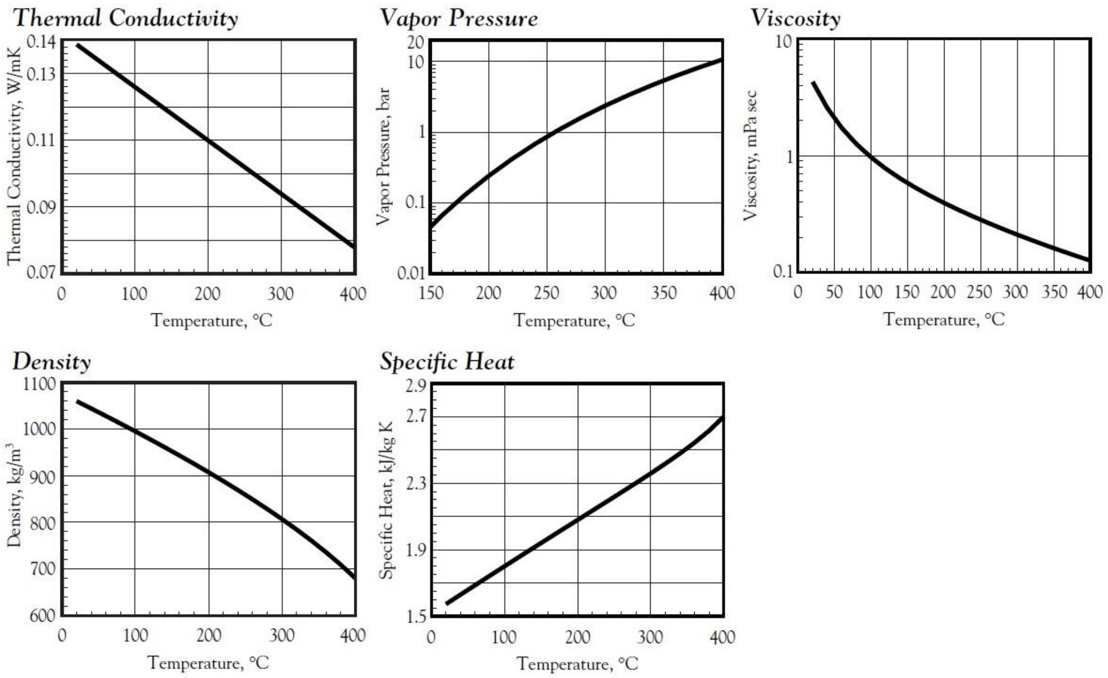


**Fig. 4:** Liquid properties of DOWTHERM A liquid [6].

## II. Theoretical MODEL

### 2.1 Coupled Finite element analysis model

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 [7]. The convective coefficient of the cooling fluid was calculated by Sieder & Tate equation empirical equation [8, 9]:

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

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.

**Table 2:** Thermo-physical and thermomechanical properties of steel AISI 4340

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

### 2.2 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:

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

Where r is the crank shaft radius (half of the engine stroke), ω is the angular frequency, r is the crankshaft radius and $\ell$ is the connecting rod length.

**Table 3:** Operating conditions of the Diesel engine [7]

| name | Expression | Description |
|---|---|---|
| n | 2,000 [1/min] | Revolutions per minute |
| omega | n*2*pi | Angular velocity |
| stroke | 0.144 [m] | Engine stroke |
| r | stroke/2 | Crankshaft radius |
| conrod_length | 0.26 [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 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 [10], 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 cooling water temperatures (see reference [10]). 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 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.

## III. Results

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

### 3.1 Thermal Results

Figure 5 shows the temperature field of the engine piston.

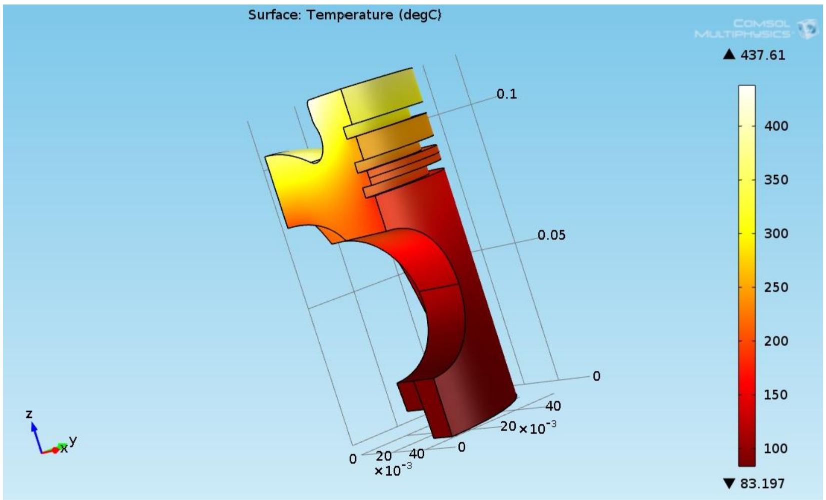


**Fig. 5:** Temperature field of the piston.

It can be seen from fig. 5 that the maximal temperature of the piston reaches to 438ºC.

**3.2 Thermomechanical results**

Figure 6 shows the Von Mises stress distribution in the engine piston.

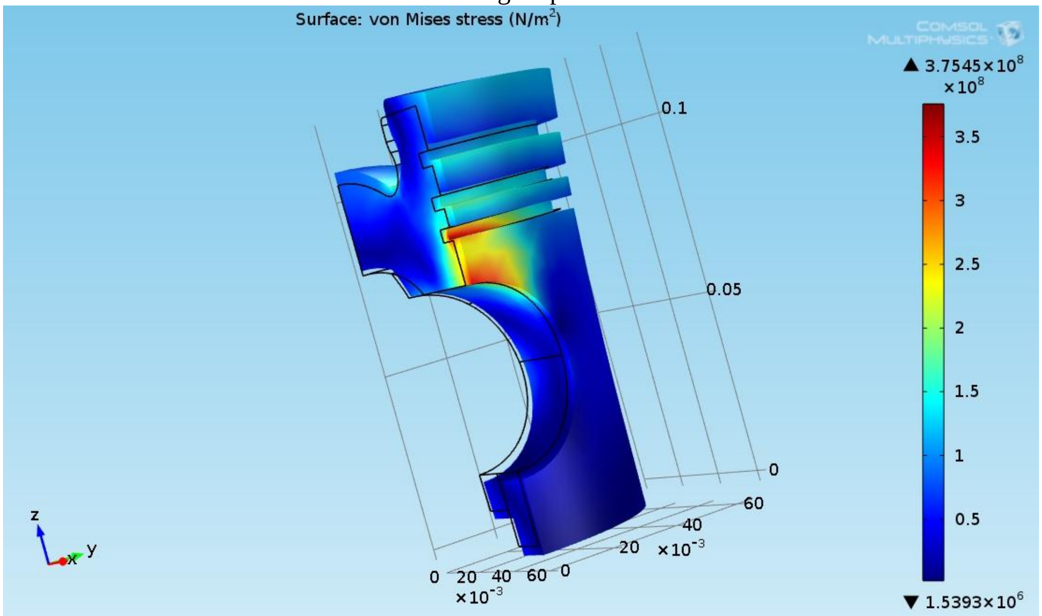


**Fig. 6:** Von Mises stress field of the piston.

As can be seen from fig. 6, the maximal stress reaches to 375 MPa. At this location the tempearture of the steel is less than 200ºC (392 ºF). Fig. 7 Shows the effect of temperature on the tensile ultimate stregth ($F_{tu}$) and tensile yield strength ($F_{ty}$) of AISI low alloy steels [11].

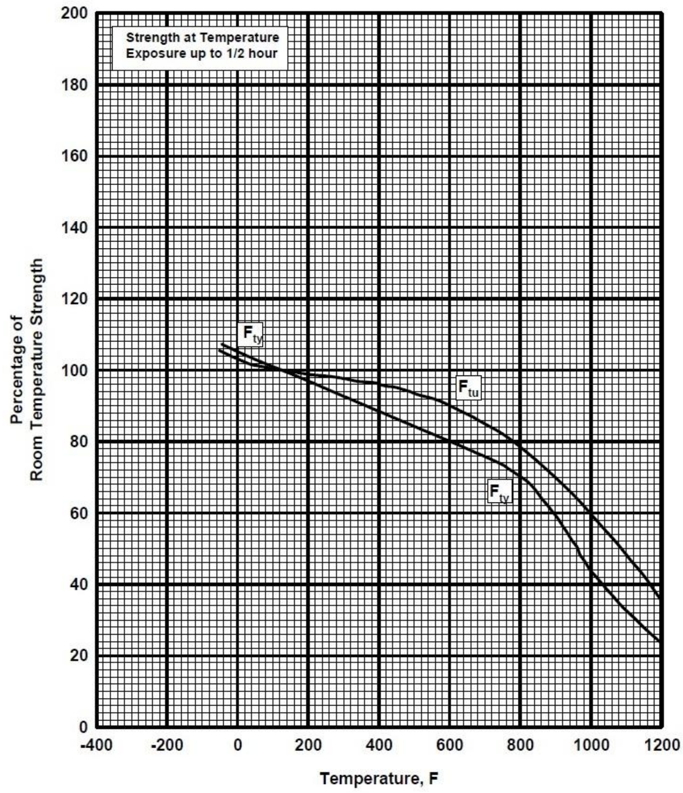


**Fig. 7:** The effect of temperature on the tensile ultimate stregth ($F_{tu}$) and tensile yield strength ($F_{ty}$) of AISI low alloy steels (all products) [11].

From figure 7, it can be seen that the decrease in the tensile yield strength is negligible at the position.

## IV. 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. 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 cooling fluid was calculated by Sieder & Tate equation empirical equation. It has been found that the maximal temperature of the piston reaches to 438ºC. The maximal stress reaches to 375 MPa. At this location the tempearture of the steel is less than 200ºC (392 ºF). the decrease in the tensile yield strength is negligible at the position.

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