Condensation within a Charge Air Cooler

Tomás Roseiro Murcela

tomas.murcela@tecnico.ulisboa.pt

Instituto Superior Técnico, Universidade de Lisboa, Portugal

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Abstract

The objective of this collaboration with the JDEUS company was to solve the problem of water accumulation due to condensation in charge air coolers, which causes engine damage in vehicles when large amounts of water are suddenly expelled to the combustion chamber.

It has been found that the condensation in the intercooler can occur globally or locally. A model of calculation of the average temperature at the exit of the intercooler was developed to predict condensation, and this model was validated with experimental tests.

Taking advantage of the previous experimental information, the calculation model to study the temperature profile at the exit of the intercooler was refined.

These models were applied to a standard intercooler, under various operating regimes. The results were compared with the previous experimental tests conducted in JDEUS. Problematic operating regimes and intercooler regions most susceptible to condensation have been identified.

A device for reducing or eliminating condensation was proposed and experimentally demonstrated. For this, the modified series intercooler was subjected to a battery of tests in wind tunnel.

The results of the new prototype were compared with the previous results. It was concluded that the developed solution effectively avoids condensation, without impairing the thermal efficiency nor increasing the pressure drop or compromise structural integrity of the intercooler.

Keywords: Intercooler, Heat exchangers, Non-uniformity of temperature in intercoolers, EGR

1 Introduction

Competition drives intercooler manufacturers to seek distinctive technological innovations. One of the challenges is to reduce the accumulation of condensed water in the intercoolers, which disrupts the operation of the engine when water is aspired in large quantities.

Nowadays, intercoolers efficiency is increasing and the outlet temperature of the compressed gases is lower and lower, with greater danger of reaching dew temperature.

To lower the cost and facilitate the manufacturing process, car brands often choose to use the same intercooler body in different cars. Thus, the appropriate dimensioning of an intercooler for higher displacement engines at high power regimes leads to overcooling of the intercooler in smaller engines, especially in situations where the engine operates at low power. Even in the same car, the intercooler sized for maximum power can cause condensation when the engine runs at low power.

This research, ought to identify under which operating conditions condensation takes place in an intercooler and then search for plausible solutions to reduce the probability of water condensation.

2 The Intercooler

A typical intercooler consists of

- Core
- External fin
- Internal fin
- Core tubes
- Header plate
- Side plate
- Upper tank
- Lower tank
Charge air from the compressor is called internal air and the cooling air is called external air. Internal fins are those inside intercooler tubes for hot air and the external fins are those for cold air.

Intercooler core is the heat exchanger itself. The charge air (hot air) from the compressor passes inside the tubes through the internal fins, while the cold air from the atmosphere passes through the external fins. Internal and external fins form small individual channels to promote heat exchange.

Generally, intercoolers are cross-flow heat exchangers.

Two header plates delimit the heat exchanger in the direction of length, while two side plates delimit intercooler width.

Core corresponds to the heat exchanger body, including the inner tubes and cold air channels.

The area of the core, exposed to external air, is commonly known as frontal area.

2.1 Condensation problem

Global condensation means that average outlet temperature of the charge air (internal flow) is lower than the dew temperature under the conditions of operation under study.

Local condensation occurs when outlet temperature of the charge air (internal flow) is lower than the dew temperature only at some regions of the intercooler.

Exhaust gas recirculation (EGR) reintroduces combustion products in intake air flow. Currently, car manufacturers use two EGR types: high pressure loop (HPL) and low pressure loop (LPL). HPL system introduces recirculated gases downstream the intercooler while LPL system reintroduces combustion products upstream the intercooler. This case has to be taken into account for water condensation study since combustion products include water, increasing absolute humidity at intake circuit upstream the intercooler.

Current solutions for documented cases of water condensation involve reducing intercooler efficiency or dealing with water accumulation instead of trying to avoid it.

One example is the 2016 Ford Focus RS [1], an example of poor sizing of an intercooler. During the test phase, excessive condensation was diagnosed inside the intercooler. Ford technical solution was to insert a plate in the front of the intercooler (figure 2, right side of the intercooler) to stop heat exchange in that area, reducing the efficiency of the intercooler.

3 Intercooler Heat Exchange Model

3.1 Heat transfer coefficient

At intercooler inlet, the flow (internal or external), presents high turbulence, although Reynolds number is relatively low, in a typical range of flat plate laminar flows aligned with the flow.

Air properties were calculated as mean of the temperatures between the entrance and exit of intercooler core.

In the following expressions, perimeters, areas and hydraulic diameters were calculated for intercoolers with similar geometry to those manufactured by JDEUS.

Reynolds number based on the hydraulic diameter is expressed by

\[ D_h = 4 \times \frac{A_t}{P}, \]

where \( A_t \) is the cross section of each channel and \( P \) is the perimeter.

Nusselt number calculation is the one that presents greater difficulties because both flows (internal and external) are turbulent flows with reduced Reynolds number.

We compared the most widely used correlations in engineering [2] [3]. Dittus-Boelter proposes for the calculation of the Nusselt number (\( \text{Nu}_D \)) a correlation involving Reynolds number and Prandtl number (\( \text{Pr} \)),

\[ \text{Nu}_D = 0.023 \, \text{Re}^{4/5} \, \text{Pr}^{1/3}. \]

The exponent \( n \) assumes \( n = 0.3 \) when air is being cooled and \( n = 0.4 \) when air is receiving heat. According to Gnielinsky, the Nusselt number can be calculated using the expression
The Nusselt number for this type of intercooler often assumes values below the applicability limit of Gnielinsky and Dittus-Boelter correlations, reaching $Re_D = 1000$. It is necessary to evaluate the behavior of the correlations described above for values of $Re_D$ below the limits given by the authors.

The following figure shows the evolution of the Nusselt number obtained with both correlations as a function of the Reynolds number.

![Figure 3 Nusselt Number](image)

Heat transfer coefficient $h$ is given by

$$ h = \frac{Nu_D k}{D}, $$

where $k$ is the thermal conductivity of air.

Compute the overall heat transfer coefficient, $U$, requires knowledge of the total heat transfer area $A_T$ between the hot fluid and the cold fluid.

Overall coefficient of heat transfer, should account for the difference in heat transfer area between the two fluids

$$ U = \frac{1}{A_T(\frac{1}{h_1A_T} + \frac{1}{h_2A_T})}. $$

Indexes 1 and 2 represent, respectively, the hot air zone and the cold air zone.

Offset and louvered fins (often seen in external fins) increase the flow turbulence and heat exchange relative to the plain fin. Therefore, it is expected that the correlations used will predict the coefficients $h$ by default and, consequently, the estimated value $U$ is expected to be inferior to the actual value, especially in the case of offset or louvered fins.

To compute local temperatures intercooler core was divides into smaller elements as shown in figure 4.

![Figure 4 intercooler core divided into elements](image)

In each element, internal and external air temperatures were estimated according to the methodology used in the previously for the overall heat transfer coefficient. The inlet temperatures in each element correspond to the outlet temperatures of the immediately upstream element.

### 3.2 Relative humidity quantification for condensation prediction

In order to predict condensation, an Excel calculation routine has been developed where, given the ambient conditions and the engine operating regime, it is possible to predict the occurrence of condensation in the intercooler and to quantify the mass of condensate for each engine operating point [4] [5].

The user provides the following input variables:

- Relative humidity of ambient air,
- Ambient temperature,
- External air velocity
- Intercooler inlet pressure
- EGR percentage
- Engine excess air

Given a certain ambient temperature and relative humidity, it is possible to obtain absolute humidity $W_{H_2O}$,

$$ W_{H_2O} = 0.622 \frac{P_w}{P - P_{ws}}, $$

where,

$$ P_w = \phi \times P_{ws}, $$

$$ P_{ws} = P_{sat@Tamb}.$$
$P_n$ is the partial vapor pressure of the flow at pressure $P$.

Internal air flow, $M_{g}$, varies in time within a known range of values. For certain conditions, it is compute water mass flow rate, $\dot{M}_{H_2O}$, intake air,

$$\dot{M}_{H_2O} = \dot{M}_{g} \times W_{H_2O} \cdot 10$$

Given the ambient conditions and the compression ratio, the efficiency, $\eta$, of the compressor and the internal air temperature at the inlet of the intercooler can be calculated. The efficiency $\eta$ is obtained from a compressor map supplied by a turbocharger manufacturer, for instance, Garrett [6].

$$T_{ar2} = T_{ar1} \times \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} \cdot 11$$

Where the point 1 relates to compressor inlet and point 2 to the compressor outlet or intercooler inlet.

The pressure $P_2$ is considered constant along the intercooler, that is, the internal pressure drop is neglected. $\gamma$ is approximated by $\gamma = 1.4$

EGR and excess air are used to compute the mass of water introduced by the EGR system into the intercooler.

Knowing petrol and Diesel chemical formulas [7], stoichiometric chemical equation state that for each gram of Diesel burned, 1.24 g of water is produced in the diesel engine and 1.42 g of water in the gasoline engine.

With the amount of excess air and stoichiometric chemical equation, it is possible to calculate the flow rate of fuel introduced into the combustion chamber and the amount of water resulting from the combustion products. Diesel engines excess air values range from 8 to 20 [7]. Small values are usual when high power is required. In that case EGR is reduced or neglectable. Modern petrol engines with direct injection systems and fuel stratification are able to work with excess air values similar to diesel engines.

If EGR is known, it is possible to calculate the amount of water introduced into the intake system from the combustion products. Since the reutilization of exhaust gases is a continuous process, the water flow introduced into the intake system from the EGR system at the end of $n$ engine cycles is a series as follows:

$$\dot{m}_{H_2O_{out}} = \sum_{i=1}^{\infty} egr_{x_i} (\dot{m}_{H_2O_{formed}} + x_{i-1}), \quad x_0 = 0$$

The series is convergent, stating that, in steady state and at the end of $n \rightarrow \infty$ engine cycles, the mass of water from the exhaust to the intercooler tends to a constant value.

In addition to the absolute humidity in the intake system, to calculate the relative humidity and predict condensation at the intercooler, it is necessary to compute dew temperatures for each point under investigation and charge air outlet temperature.

Intercollider efficiency $\varepsilon$ depends on the ratio between charge air mass flow and cooling air mass flow ($\dot{M}_{g}/\dot{M}_{a}$).

$$\varepsilon = \frac{T_{h,out} - T_{amb}}{T_{h,in} - T_{amb}} \cdot 13$$

if outlet temperature of the intercooler is higher than dew temperature condensation exists.

4 Testing Procedures

4.1 Compute $U$

The overall heat transfer coefficient for each operating point of the intercooler can be determined experimentally in a wind tunnel.

Figure 5 Typical efficiency plot for an intercooler

Figure 6 Wind tunnel scheme used to test intercoolers at JDEUS facilities (image from JDEUS)

In laboratory tests, the internal air is conducted to the intercooler through ducts connected to the factory-installed compressed air system. The geometry of the external air inlet duct is adaptable.
so that the area coincides with the front area of the intercooler. The intercooler is installed with the frontal zone perpendicular to the inlet duct of the external air.

In the tests we completed, the incoming flow of hot air were in the interval [300, 700] kg/h and, for each internal flow rate, external air velocity of the was 4, 8 and 12 m/s. The inlet temperature of the gases, the ambient temperature and the pressure were constant.

In order to obtain global heat transfer coefficient it is necessary to know the inlet and outlet temperatures of cold air and hot air at the points of operation under investigation.

According to the logarithmic mean temperature difference method [2],

\[ U = \frac{\dot{Q}}{A \Delta T_{LN} F} \]

Logarithmic mean temperature difference, \( \Delta T_{LN} \), is

\[ \Delta T_{LN} = \frac{T_2 - T_1}{\ln \frac{T_2}{T_1}} \]

and heat transfer rate, \( \dot{Q} \),

\[ \dot{Q} = \dot{m} c_p \Delta T \]

Indexes 1 and 2 correspond to the inlet and outlet section of the heat exchanger. Mean log temperatures difference was calculated for a countercurrent flow heat exchanger, applying the correction factor \( F \) for cross flow, obtained from the following graph [2].

In the particular case of the intercooler tested, the nozzles in the inlet and outlet tanks are directly facing core, approximately at half height, so, we considered that the distribution of flow through the 9 tubes of the nest is approximately symmetrical.

4.2 Compute intercooler outlet section temperature differences

To obtain the temperature distribution at core outlet temperature sensors (thermocouples) were added at specific points immediately downstream core exit section.

Some recommendations were followed:
- Thermocouples were located at some distance from outlet section (= 5mm) to prevent the probe from blocking the channel and shifting the flow.
- Sensor wires were placed on header plate so that they do not pass in front of the internal air tubes, to cause the least possible disturbance in the flow (figure 19).

In the figure below, the locations where the thermocouples are placed are shown on the left side and the heat exchanger section with the thermocouples properly installed on the right side. The brown tape that appears on the picture was removed after the paste glue dried.

Inlet charge air temperature is measured in the inlet duct of immediately upstream of the intercooler while outlet temperature is measured in the outlet air duct downstream of the intercooler.

In the particular case of the intercooler tested, the nozzles in the inlet and outlet tanks are directly facing core, approximately at half height, so, we considered that the distribution of flow through the 9 tubes of the nest is approximately symmetrical.
This allowed us to install sensors only at half of intercooler core outlet section, reducing complexity and possible flow perturbations due to wires crossing the flow.

4.3 Relative humidity test for model validation

The company JDEUS made available for this testing procedure a machine capable of measure relative humidity at intercooler outlet.

Charge air at a certain pressure and temperature enter in the intercooler with absolute humidity approximately zero is supplied to the intercooler. The internal air outlet temperature shall be specified in the test. The external airflow is adjusted automatically by the equipment so that the internal air temperature is reached.

Water is vaporized upstream the intercooler according to pre-defined test points.

Results were registered after steady state regime reached for each test point.

The machine has the following sensors:
- Inlet charge air temperature,
- Outlet charge air temperature,
- Charge air inlet pressure,
- Charge air flow meter,
- Water flow meter,
- Relative humidity sensor at intercooler outlet.

Water is vaporized upstream the intercooler according to pre-defined test points.

Results were registered after steady state regime reached for each test point.

The machine has the following sensors:
- Inlet charge air temperature,
- Outlet charge air temperature,
- Charge air inlet pressure,
- Charge air flow meter,
- Water flow meter,
- Relative humidity sensor at intercooler outlet.

Figure 10 Intercooler ready for testing.

5 Case study: Intercooler from a 3.0L, V6 engine

<table>
<thead>
<tr>
<th>Cilindrada</th>
<th>≈3000 cm³</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>160 kW / 215 cv</td>
</tr>
<tr>
<td>Torque</td>
<td>400 N.m</td>
</tr>
<tr>
<td>European Normative</td>
<td>Euro 6</td>
</tr>
<tr>
<td>Type of EGR</td>
<td>HPL+ LPL</td>
</tr>
</tbody>
</table>

Table 1 Engine specifications

<table>
<thead>
<tr>
<th>H [m]</th>
<th>0,720</th>
</tr>
</thead>
<tbody>
<tr>
<td>E [m]</td>
<td>0,08</td>
</tr>
<tr>
<td>W [m]</td>
<td>0,144</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>N_{tubes}</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>N_{external layers}</td>
<td>10</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Internal fin type</th>
<th>Plain</th>
</tr>
</thead>
<tbody>
<tr>
<td>External fin type</td>
<td>Louvered</td>
</tr>
</tbody>
</table>

Table 2 Intercooler specifications (JDEUS data)

5.1 $U_{\text{estimated}}$ vs $U_{\text{experimental}}$

Table 3 shows the values of the global heat transfer coefficient for the various operating points calculated according to the method described in chapter 3.1 and the values obtained in the wind tunnel according to the procedure described in chapter 4.1.

Charge air inlet temperature is 180 °C, ambient temperature 25 °C, and the absolute pressure inside the intercooler 2.5 bar.

<table>
<thead>
<tr>
<th>$M_g$ [kg/h]</th>
<th>$V_{\text{air}}$ [m³/h]</th>
<th>$U_{\text{estimated}}$ [W/m²K]</th>
<th>$U_{\text{measured}}$ [W/m²K]</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>4</td>
<td>15,03</td>
<td>20,24</td>
<td>26%</td>
</tr>
<tr>
<td>300</td>
<td>8</td>
<td>18,01</td>
<td>27,30</td>
<td>34%</td>
</tr>
<tr>
<td>300</td>
<td>12</td>
<td>19,42</td>
<td>33,78</td>
<td>43%</td>
</tr>
<tr>
<td>500</td>
<td>4</td>
<td>18,83</td>
<td>27,91</td>
<td>33%</td>
</tr>
<tr>
<td>500</td>
<td>8</td>
<td>23,88</td>
<td>36,82</td>
<td>35%</td>
</tr>
<tr>
<td>500</td>
<td>12</td>
<td>26,48</td>
<td>43,38</td>
<td>39%</td>
</tr>
<tr>
<td>700</td>
<td>4</td>
<td>21,30</td>
<td>33,49</td>
<td>36%</td>
</tr>
<tr>
<td>700</td>
<td>8</td>
<td>28,14</td>
<td>45,18</td>
<td>38%</td>
</tr>
<tr>
<td>700</td>
<td>12</td>
<td>31,87</td>
<td>53,02</td>
<td>40%</td>
</tr>
</tbody>
</table>

Table 3 Global heat transfer coefficient

The experimental value is greater than the theoretical value, as expected. In the theoretical estimation is neglected the increase of the heat exchange caused by the windows and roughness of the louvered type of the external fins. The flow, although turbulent, presents low Reynolds number, usually in the range of the laminar if they were flat plates. Thus, the correlations used for the Nusselt number are outside the main range of validity. All this explains a default estimate of the overall heat transfer coefficient.

Figure 24 shows evidence of a linear relationship between the theoretically calculated global heat transfer coefficient value and the experimental value.
5.2 Temperature profile at intercooler outlet

Figures 12 and 13 show the temperature difference between the end of tube 5 (central tube) and 9 (tube at one end of the intercooler) for a charge airmass flow of 500 kg/h.

The numbers 1 and 3 on the abscissa axis represent, respectively, the front zone of the intercooler and the rear zone.

The temperature difference between tube 5 and the tube 9 in the front zone of the intercooler is lower than the temperature difference between the tube 5 and the tube 9 in the rear zone of the intercooler.

The tubes located at the tip of the intercooler are more efficient because they have a larger heat transfer area of external fin available (there are 10 layers of external fins and 9 of internal fins). Thus, in the high efficiency regimes (high external air velocity), the temperature difference at tubes 1 and 9 is reduced compared to the central tubes.

The higher non-uniformity of temperature occurs in the central tubes, particularly in the tube 5.

5.3 Validation of relative humidity estimation

Several test points were investigated according to plan described on section 4.3.

Charge air inlet temperature (150°C), and charge air outlet temperature (35°C) were fixed.

Absolute pressure assumed values 1,5 bar, 2,0 bar and 2,5 bar. For each pressure value water volume flow assumed 4 values between [1,2;2,0] [l/h].

<table>
<thead>
<tr>
<th>Average error</th>
<th>Maximum error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,4%</td>
<td>7,9%</td>
</tr>
</tbody>
</table>

Table 4 Errors between $\varnothing$ estimated with method described on section 3.2 and measured $\varnothing$

5.4 Relative humidity at real operating conditions

The calculation model described in chapter 3, already modified with the experimental results obtained in the wind tunnel, allows estimation of relative humidity under real operating conditions, introducing in the calculation, the efficiency of the compressor and the operation of the EGR system.

The following figures show the predictions for relative humidity in the outlet section of the intercooler under study in two ranges of operating points:

<table>
<thead>
<tr>
<th>Cooling flow velocity [m/s]:</th>
<th>2.00</th>
<th>4.00</th>
<th>6.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Cooling flow velocity [m/s]:</th>
<th>8.00</th>
<th>10.00</th>
<th>12.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| Mg [kh/h] | 300-600 | 600-900 |
| Va [m/s]  | 6       | 10      |
| $P_{\text{absolute}}$ [bar] | 1.8 | 2.5 |
| $\varnothing$ | 75% |     |

Table 5 Operating conditions

5.4.1 Average temperature vs minimum temperature

Results will be compared with low pressure loop EGR =20. In the case of exhaust gas recirculation, *EGR = 20. Let us consider the excess air $\lambda = 18$. 
5.4.1.1 Low speed

Figures 14 and 15 for EGR = 20 show that condensation evaluated at minimum temperature is significant when compared to condensation evaluated at average temperature.

![Figure 14 Relative humidity at average temperature](image1)

![Figure 15 Relative humidity at minimum temperature](image2)

The previous graphs clearly show that core temperature inhomogeneity is one of the main factors for the occurrence of condensation.

5.4.1.2 High speed

The plots in figures 16 and 17 for EGR = 20 show the same results as the low speed condition.

The average temperature at the outlet of the intercooler does not have severe condensation, but looking at figure 17, condensation occurs at the front of the intercooler where the minimum temperatures are reached at all times.

![Figure 16 Relative humidity at average temperature](image3)

![Figure 17 Relative humidity at minimum temperature](image4)

With humidity close to 100% the droplets would be very small, where the ratio between the aerodynamic resistance and the drop weight would be high. At flow rates in the order of 10 m/s, considering a length of 10 cm, the residence time of the droplet would be approximately 0.01 s, insufficient for the accumulation of large amounts of water in the intercooler. The uniformity of temperatures at the intercooler may become a way of solving condensation problem.

6 Innovation proposal

In this chapter, sections 6.1 to 6.2.1 are kept confidential until public release of the patent related to this work.

6.1 Concept
6.2 Experimental tests on new prototype

6.2.1 Efficiency test

6.2.2 Average temperature vs minimum temperature of the prototype

Temperature difference in the central tube (tube with the greatest temperature difference) reduces about 40% in all operating modes.

Increasing minimum temperature prevents condensation in several zones of the intercooler, which in the standard version would be below dew temperature.

The following figures show differences in relative humidity evaluated at minimum temperature between at the exit of the intercooler for the standard intercooler and for the prototype.

6.2.2.1 Low speed regime

7 Conclusions

In situations with EGR, the number of operation regimes with relative humidity above 100% in the new prototype is always inferior compared to the series prototype. Even when condensation is predicted, relative humidity value is slightly larger than 100%, thus water accumulation is unlikely.
despite the formation of small droplets near the outlet of the prototype.
The temperature difference between the hottest zone and the cooler zone of each intercooler tube is reduced by about 40%.

8 Future work

Improve prediction of intercooler outlet temperatures in the design phase (method described in Chapter 3), parameterizing all types of fins according to the effect they produce on the convection coefficients.

9 References


