DIMENSIONING PROCEDURE OF COOLANT RADIATORS FOR TRUCKS AND BUSES

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Abstract. *The coolant radiators are fundamental components to the operation of internal combustion engines, and its sizing, particularly, frontal area, should be sufficient to allow the appropriate heat transfer from the engine. This work describes a procedure for calculating the frontal area of radiators to the liquid coolant, especially applicable to trucks and buses. The procedure consists on the determination of the heat rejected by the liquid coolant (water), which is based on the energy associated to the fuel burnt on combustion chambers, or technical information by radiator suppliers, and or from the power developed by the engines. A description of the cooling test for vehicle approval is contained in this work, as well as an equation of the air to boil (ATB) is used for the test validation. A flowchart to calculate the radiator frontal area is proposed. The calculation procedure developed proved to be very accurate when compared with the results of cooling tests (three tests were performed). In general, the temperature results of the air and water (in several points of the system) obtained on calculations, agreed with the values measured in the cooling tests. The worksheet allows evaluate the behavior of the coolant system studied, being very useful in the estimation of frontal area of the radiator's vehicle.*

Keywords*: liquid coolant system, cooling tests, air to boil (ATB), radiator, energetic balance.*

1. INTRODUCTION

Water radiators on vehicular applications are responsible by heat dissipation in mainly of internal combustion engines and also by other sources of heat, like water/oil heat exchanger of automatic transmissions.

The radiator must ensure the heat transfer, dissipating heat through the circulating air (Bosch, 2005). When engines need a higher performance or when there is a small space for installation, the better layout indicated is the radiator with thin tube and corrugated fins (SAE J631, 1988), and this type is the most currently used by vehicular assemblers. The cooling system should be compact, low weight and its installation should occupy the smallest space as possible (Basshuysen e Schafer, 2004).

In vehicle design like trucks and buses is important to meet all versions and variations that each vehicle may have, this is necessary for customizing a cooling package for all needs and avoid different developments for each application. Cooling system design need a long time for 3D components modeling for further evaluation in CFD software (Computational Fluid Dynamics). For this reason in some situations the design is based on previous application, and validation is done with a test in track run called cooling test, where is determined the ATB (air to boil) temperature, being this the main methodology used by engine manufactures with application in vehicles.

To minimize costs in development and ensure that the selected cooling package meets the system thermal rejection, this work shows a calculation methodology that helps engineers and designers in determining the frontal area of the water radiators. The calculation methodology is based on cooling air conditions, responsible by heat rejection in radiator.

To help the calculations an electronic worksheet with the EXCEL® application (by MICROSOFT) was developed. This worksheet has the intention to simulate different conditions and enable value iterations when necessary in some calculations.

2. COOLING SYSTEM CHARACTERISTICS

The calculation methodology needs data input in different stages, that will be described and presented on the sequence of this work: cooling system, environment air characteristics, thermal balance engines, thermal rejection and pressure drop on radiators, fan performance curve, system curve and operating point, water and air temperatures and ATB calculation.

2.1. Cooling system

A cooling system assembly for trucks and buses engines is shown in Fig. 1, which is composed by an aftercooler, a water radiator and a fan. In this case the main components for thermal capacity calculation of the cooling system can be found.

The Fig. 1 shows also the applications of cooling system in chassis with front and rear engines, as well as the flows of water and air crossing the package.

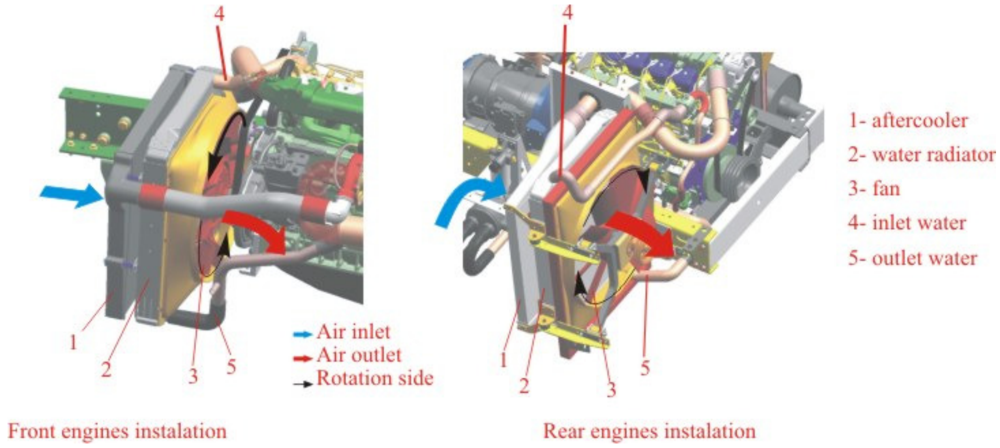


Figure 1. Cooling package – Application example

According to Bosch (2005), air flow crossing the radiator is a crucial factor and depends on the driving speed (chassis with front engines), of the flow resistance into the engine compartment, of the flow resistance in radiator and of the fan efficiency.

2.2. Environment air characteristics

From some equations described in ASHRAE (1997) is possible to determine the main air properties at various positions of the cooling system. These equations are the following:

The barometric pressure above sea level is according Eq. (1).

$$P_0 = 101.325(1 - 2.25577x10^{-5} Z)^{5.2559} \tag{1}$$

where P_0 is the barometric pressure, [kPa]; and Z is the height above the sea level.

The absolute humidity is determined by the Eq. (2).

$$\omega = 0.62198 \frac{(\varphi \cdot P_s)}{P_0 - (\varphi \cdot P_s)} \tag{2}$$

where ω is the absolute humidity, [kg (water) / kg (dry air)]; ϕ is the relative humidity of environment air or inlet air of radiator, [%]; and P_s is the partial pressure of saturated water [kPa] in the mixture temperature.

The partial pressure is determined by the Eq. (3), which is valid for a temperature air in the range from 0 to 200°C.

$$\ln(P_s) = \frac{C_8}{T} + C_9 + C_{10}T + C_{11}T^2 + C_{12}T^3 + C_{13}\ln(T) \tag{3}$$

where T is the air temperature in any position of cooling system, [K]; C_8=-5.8002206E+03; C_9=1.3914993E+00; C_{10}=-4.8640239E-02; C_{11}=4.1764768E-05; C_{12}=-1.4452093E-08; e C_{13}=6.5459673E+00.

The specific volume of air can be calculated by Eq. (4).

$$v = \frac{R_a \cdot T \cdot (1 + 1.6078 \cdot \omega)}{P_0} \tag{4}$$

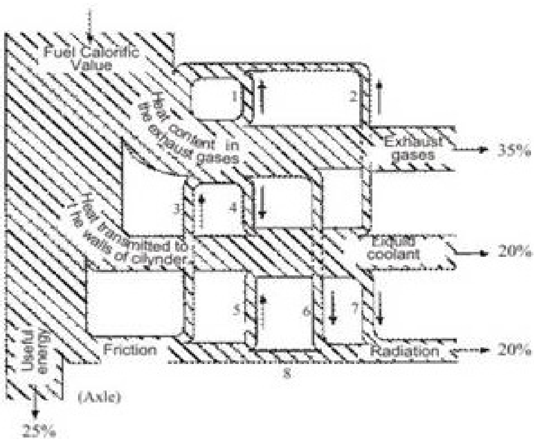
where v is the specific volume, [m³/kg of dry air]; e R_a is the gas constant of dry air, [287.055J/kg.K].

2.3. Thermal balance engine

According to Heywood (1988), the engine performance, its efficiency and the exhaust gases emissions reduction are directly related to the good operation of the cooling system. To Taylor (1971), is in the peak power condition that occurs the biggest thermal rejection of engines.

An usual way to show the thermal balance is by Sankey diagram. In this methodology becomes more evident the representation of the heat rejection percentages (Franceschini *apud* Da Campo, 2003).

In accordance with Giacosa (2004), the energy flow on engines occurs by four ways: axle power, exhaust gases, liquid coolant and radiation, and in Fig. 2 these flows are identified.



1- Heat from waste gases;
2- Heat transmitted through cylinder walls;
3- Friction heat to the exhaust gases;
4- Heat from the exhaust gases to the cooling;
5- Friction heat to the liquid coolant;
6- Radiation from the exhaust system;
7- Radiation from cooling system; and
8- Heat radiated from engine parts not cooled.

Figure 2. Heat rejection flows

According to Bussien (1965), Heywood (1988), Giacosa (2004) and Heisler (2002) the thermal rejection from engine to water is in the range from 16 to 35% of the total energy entering with the fuel. Table 1 shows an analysis of this rejection.

Table 1. Energy balance percentages

Author	Power	Liquid coolant	Exhasut gases	Radiation
Bussien	32	32	29	7
Heywood	34-38	16-35	22-35	2-6
Giacosa	25	20	35	20
Heisler	-	20-30	-	-
Working range	25-38	16-35	22-35	2-20
Mean values	33	26	30	11

The estimation of heat rejection from engine to water can be determined by three ways: by a percentage of energy contained in the inlet fuel, (option A); by a percentage of the axle power engine, (option B); and by performance tests developed by engine's supplier, (option C).

The option A is defined by Eq. (5).

$$Q_{engine} = \frac{FSC \cdot P_{engine} \cdot LHV_{fuel} \cdot \alpha}{3.6 \cdot 10^6} \tag{5}$$

where Q_{engine} is the thermal rejection to the liquid coolant, [kW]; *FSC* is the Fuel Specific Consumption, [g/kW.h]; P_{engine} is the shaft power, [kW]; LHV_{fuel} is the Lower Heating Value of the fuel, [kJ/kg]; and α is the percentage of heat transfer from engine to water which, in accordance to Table 1, lies between 16 and 35%.

The Eq. (6) defines the option B.

$$Q_{engine} = P_{engine} \cdot \beta \tag{6}$$

where β varies between 45 and 55%, in accordance with the characteristics of the internal combustion engine.

The values of thermal rejection to option C are obtained from performance tests, and this option is the preferential option to use in calculations. When option C values are not available, the Eq. (6) provides approximated values to the engine heat rejection.

2.4. Thermal rejection and pressure drop on radiators

Bussien (1965) establishes values for specific heat rejection (φ) to different thicknesses of radiators. The Fig. 3 shows these values in relation of air specific mass flow.

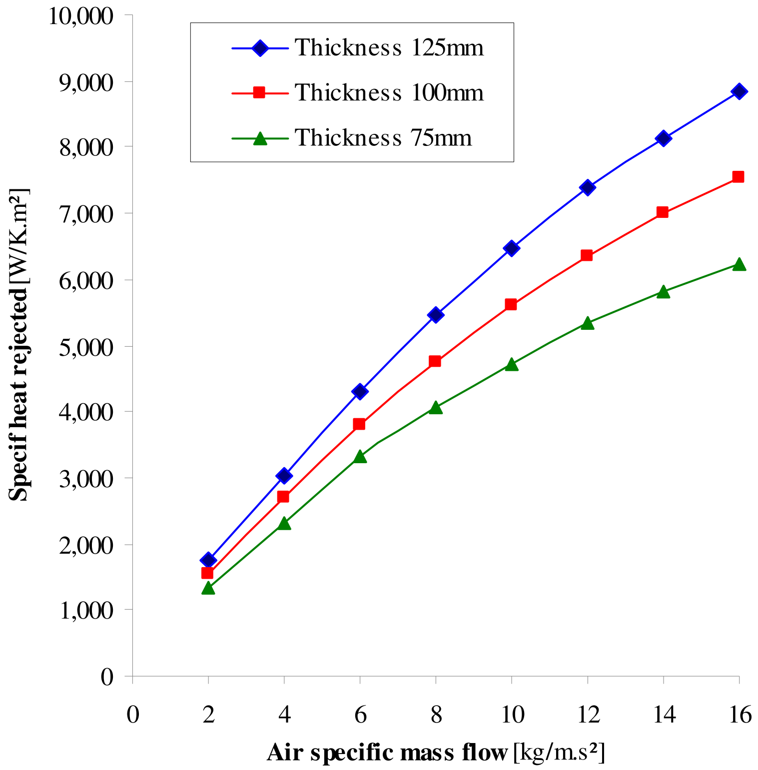


Figure 3. Heat rejected on vehicular radiators

With Fig. 3 is possible to calculate the heat rejection by Eq. (7).

$$Q = \varphi \cdot A_{f,EST} \cdot \Delta T \tag{7}$$

where Q is the heat rejected in the radiator, [W]; φ is the specific heat rejected, [W/m².K] obtained in Fig 3; $A_{f,EST}$ is the estimated frontal area of radiator, [m²]; and ΔT is the temperature difference between the inlet water temperature and the inlet air temperature in the radiator, [°C].

According to the Eq. (7) defined by Bussien (1965), is possible to calculate the frontal area of the radiator based on the thermal balance engines, item 2.3. The Eq. (8) describes this condition.

$$A_{f,EST} = \frac{Q}{\varphi \cdot \Delta T} \tag{8}$$

However, vehicular radiator manufacturers provide performance curves determined on wind tunnel. The Fig. 4 shows an example of performance curves of a radiator with 0.4962m² of frontal area, 3.25 kg/s of liquid coolant mass flow rate, inlet water temperature of 373.15 K, and inlet air temperature of 303.15 K.

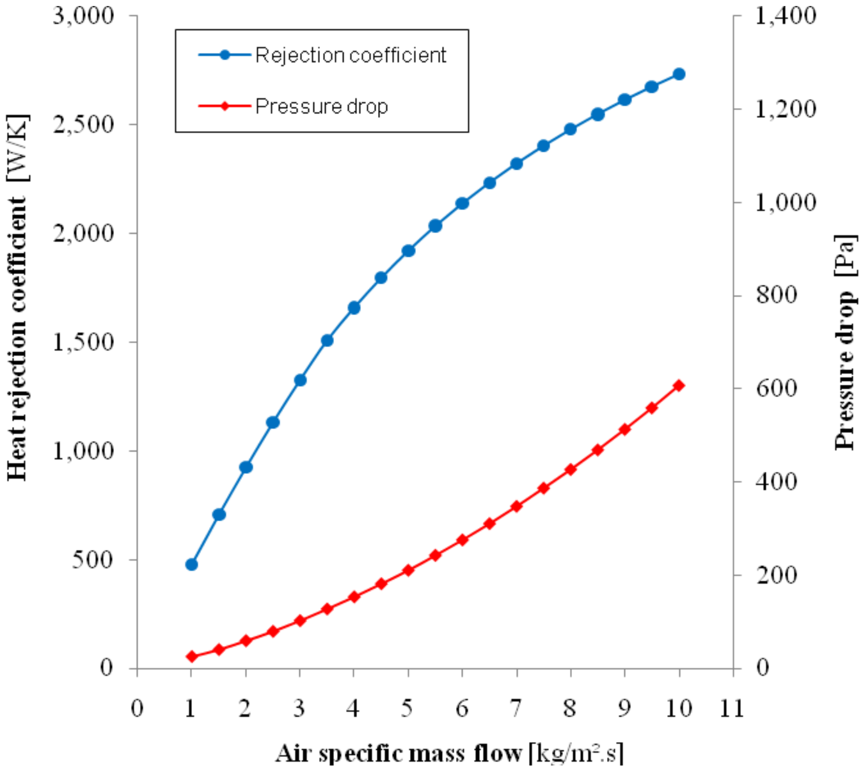


Figure 4. Performance curve of a radiator

According to Quim (2007), the vehicle radiator can be classified as a cross-flow heat exchanger. When only the inlet temperatures are known is preferable to use the effectiveness (ε) – Number of Transfer Units (NTU) method (Incropera and Dewitt, 1998), through which is possible to determine the overall heat transfer coefficient.

From the performance curves of the radiator, the heat flow rate is determined by Eq. (9).

$$Q = \gamma \cdot (T_{h,i} - T_{c,i}) \tag{9}$$

where Q is heat flow rate on radiator, [W]; γ is the heat rejection coefficient by air side selected from the performance curve, [W/K]; $T_{h,i}$ is the inlet water temperature (hot fluid), [°C]; e $T_{c,i}$ is the inlet air temperature (cold fluid), [°C].

From the definitions of Incropera and Dewitt (2003), Eq. (9) can be written in terms of ε and the heat capacity rate using Eq. (10).

$$Q = \varepsilon \cdot C_{\min} \cdot (T_{h,i} - T_{c,i}) \tag{10}$$

where ε is dimensionless; and C_{min} is the minimum heat capacity rate, [W/K].

The heat capacity rates on air side and water side, can be determined by equations (11) and (12). For vehicular radiators C_{min} is always related to air side and C_{max} refers to the liquid coolant side (water).

$$C_{\min} = C_c = m_c \cdot c_{p,c} \tag{11}$$

$$C_{\max} = C_q = m_q \cdot c_{p,q} \tag{12}$$

where m_c is the mass flow rate of the cold fluid (air), [kg/s]; $c_{p,c}$ is the specific heat of the cold fluid (air), [J/kg.K]; C_h is the heat capacity rate of the hot side, [W/K]; m_h is the mass flow rate of the hot fluid (water), [kg/s]; e $c_{p,h}$ is the specific heat of the hot fluid (water), [J/kg.K].

The cold fluid mass flow rate can be determined by Eq. (13).

$$m_c = m_{air,spec.} \cdot A_f \tag{13}$$

where $m_{air,spec}$ is the air specific mass flow, [kg/m².s]; and A_f is the frontal area of the radiator used, [m²].

From radiator performance curves is possible to determine the thermal rejection coefficient (γ) to different radiator areas and different liquid coolant flow rates. In order to simplify the procedure is important to use an iterative process through electronic worksheets for example, where the NTU and the overall heat transfer coefficient "U" values are determined. According to Quim (2007), the water radiator effectiveness is defined through the Eq. (14), and the NTU value can be obtained from numerical iteration.

$$\varepsilon = 1 - \exp[\frac{1}{C_r} \cdot NUT^{0.22} \cdot (\exp[-C_r \cdot NUT^{0.78}] - 1)] \tag{14}$$

where C_r is the ratio between the heat capacity rates, $C_r = C_{min}/C_{max}$.

The radiator thermal rejection coefficient may be obtained to the new area with the following steps: from equations (11) and (13) calculate the value ($C_{\min}$) to the new area; from Eq. (14) and an iterative process to obtain a new effectiveness value (ε), maintaining the same NTU value of the previous area; and from the relation between the equations (9) and (10), may be used to determine the value of the radiator thermal rejection coefficient, i.e., based on Eq. (15).

$$\gamma = \varepsilon \cdot C_{\min} \tag{15}$$

To find the value of (γ) when occurs a change in the hot fluid flow rate (m_h), may be used the Eq. (16) suggested by Kreith and Bohn (2003).

$$UA_2 = UA_1 \cdot (\frac{m_{h1}}{m_{h2}})^{0.8} \tag{16}$$

where UA is the overall conductance, [W/K]; and m_{h2} is the new hot fluid mass flow rate (water), [kg/s].

The overall conductance (UA) is defined by Eq. (17).

$$UA = NUT \cdot C_{\min} \tag{17}$$

In this way, with the new value (UA_2), is possible use Eq. (18) to find the NTU value and through the Eq. (14) calculate a new effectiveness (ε) value.

$$NUT = \frac{UA_2}{C_{\min}} \tag{18}$$

The Eq. (16) that considers the mass ratio elevated at 0.8 doesn't have good approximation with the results obtained from radiator performance curves. So, the Eq. (16) was replaced by Eq. (19) that had a better approximation in the results.

$$UA_2 = UA_1 \cdot (\frac{m_{h1}}{m_{h2}})^{0.33} \tag{19}$$

2.5. Fan performance curves

In general the automotive vehicles use axial fans to force the air flow through the cooling package. According to Henn (2006), the fan performance curves relate the total pressure or static pressure with the air mass flow rate for several speeds. In Fig. 5, this relation is shown.

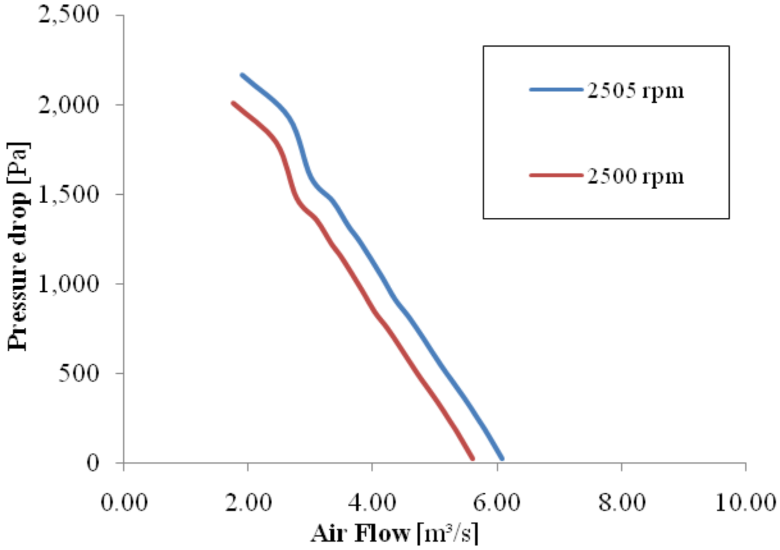


Figure 5. Fan performance curves

When other rotations are needed is possible to make corrections in flow and static pressure according to the similarity law and dimensionless quantities, equations (20) and (21), obtained from known fan laws.

$$\frac{P_{ST}}{P_{ST'}} = \left(\frac{n}{n'}\right)^2 \tag{20}$$

where P_{ST} is the fan static pressure, [Pa]; n is the fan rotation, [rpm]; n' is the new rotation desired, [rpm]; and P_{ST}' is the static pressure related to n', [Pa].

$$\frac{V}{V'} = \frac{n}{n'} \tag{21}$$

where V is the fan air volumetric flow rate, [m³/s]; and V' is the desired flow related to n', [m³/s].

2.6. System curve and operating point

The system curve is the sum of the different pressure drop by the components in the cooling system. In case of cooling system by buses and trucks the loss is related to the aftercooler, water radiator and frontal louvers.

According to Henn (2006), as a fan cannot work outside its characteristic curve and to attend a certain flow rate, the fan must answer the energy required by the pressure drop of the system. In the spreadsheet the operating point is determined by the intersection between the system curve and the fan characteristic curve, e.g., this intersection is illustrated in Fig. 6.

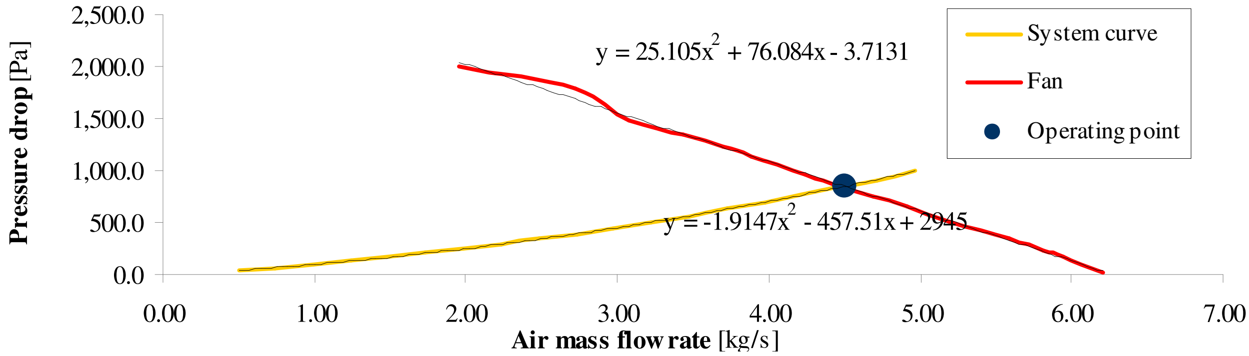


Figure 6. System curve and operating point

2.7. Water and air temperature

Through the operating point, shown in Fig. 6, is possible to determine the air mass flow rate, and from a radiator performance curve is possible to obtain the thermal rejection coefficient (γ). So, using equations (9) or (10) is possible to determine the radiator inlet temperature, using the heat rejected by the engine, and in this case using ($Q = Q_{engine}$).

The outlet temperature of hot and cold fluids (water and air, respectively) may be determined for equations (22) and (23) (Incropera and Dewitt, 2003).

$$T_{h,o} = T_{h,i} - \frac{Q}{m_h \cdot c_{p,h}} \tag{22}$$

$$T_{c,o} = T_{c,i} + \frac{Q}{m_c \cdot c_{p,c}} \tag{23}$$

where $T_{h,o}$ is the hot fluid outlet temperature, [°C]; and $T_{c,s}$ is the cold fluid outlet temperature, [°C].

2.8. ATB calculation

SAE J829 (1987) describes the Eq. (24) to calculate the ATB temperature, taking into account the environment temperature condition.

$$ATB = (T_{BOIL} - T_{h,i}) + T_{env} \tag{24}$$

where T_{BOIL} is the specified water temperature to boil, whose value is defined by supplier according to the engine application, [°C]; $T_{h,i}$ is the inlet water temperature in the radiator; and T_{env} is the environment temperature, [°C].

3. MATERIAL AND METHODS

The tests must be made under some conditions which demand special attention to the following: the thermostatic radiator valve must be locked on the fully open position to assure the maximum water flow rate through the radiator; the viscous clutch of the fan must receive a locking to ensure the maximum air flow rate; the liquid coolant must be water without additive to reduce the boiling point; a vehicle of weight and power higher than the vehicle tested must be connected behind this to serve as brake to ensure the maximum speed and power of the engine; the tests should be made on low-traffic road or on a specific highway; the environment conditions should be favorable, i.e., with temperatures above 15°C, low incidence of wind, weather without precipitation (rain) and avoid high relative humidity.

In this study three cases were analyzed: case 1 and case 2 refer to a city bus, and case 3 refers to a road bus. The city bus was equipped with 132 kW engine and the road bus with 162 kW. For cases 1 and 2 the tests were realized at environment temperature of 16°C and 35°C, respectively, and for case 3, the temperature was 29°C. The buses were tested in highways with duration of around three hours each test, and were made according to the above required conditions.

During the cooling test, the temperatures of water flow and air flow through radiator were made by a data acquisition system from HBM (Hottinger Baldwing Messtechnik) mark and MGCplus model. The temperatures were measured in six points, as shown in Fig. 7, with a frequency of 0.5 Hertz, and K type thermocouples were used, with accuracy of ±0.5°C. The data were saved into electronic worksheet (MICROSOFT EXCEL®).

The radiators used were of the thin tubes type with corrugated fins, according to SAE J631, and for the three cases, the radiators frontal areas were the same with the following dimensions: width of 615.8mm and height of 806mm. The fans used were of the axial type and all equals, i.e., 9 blades and 650mm of diameter. The air mass flow rate was determined by the operating point (item 2.6 – Fig. 6).

The test was made keeping the engine in maximum power rotation until the stabilization of the water temperature. In the ATB calculation by Eq. (24), the value can't be lower than the temperature provided by the manufacturer. For example, if the ATB temperature calculated has a value of 35°C, means that the vehicle can’t work in sites where the environment temperature is higher than 35°C, because the water of the cooling system will boil at this condition.

To validate the methodology, the same values of hot fluid temperature, air temperature on radiator face and environment temperature were used in the worksheet for each test. This way is possible to compare the calculated radiator outlet water temperatures with the tested temperatures, as showing Fig.8.

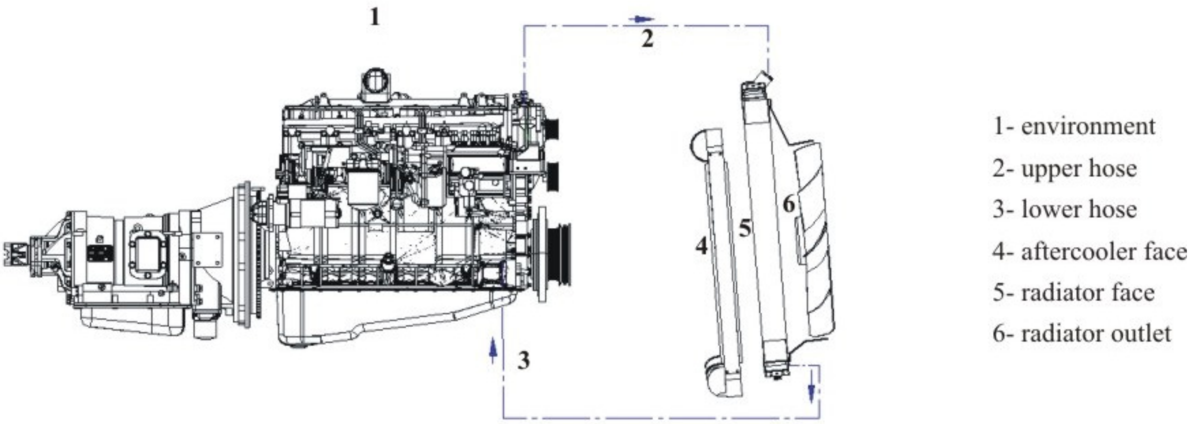


Figure 7. Thermocouple positions

4. RESULTS

In Fig. 8, a comparison between the outlet water radiator temperature obtained from the calculation methodology and obtained in cooling test is shown. With the purpose to prove the calculation methodology, the values of inlet water radiator temperatures are the same, and in this way the comparison is made in the outlet water temperature.

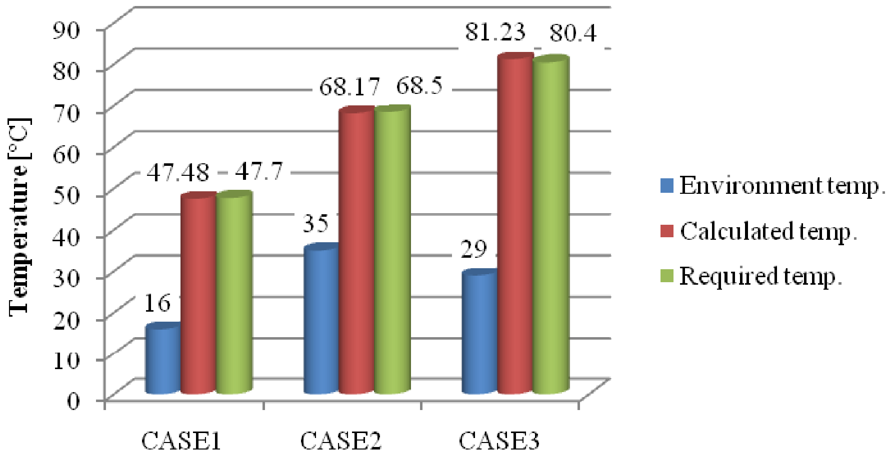


Figure 8. Comparative between temperatures calculated and tested

Is possible to show in Fig. 9 a comparative between the ATB temperatures required and calculated.

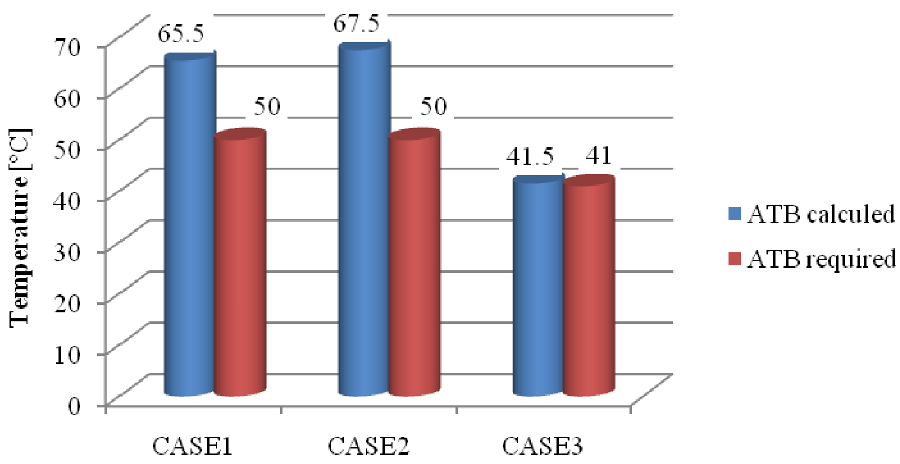


Figure 9. Comparative between the ATB temperature required and calculated

As can be seen in Fig. 9, the ATB temperatures calculated were 65.5ºC and 67.5°C, while the ATB required for this application is 50ºC. These results show that the radiator is oversized for cases 1 and 2 and its frontal area could be dimensioned between 0.4080 and 0.3570 m², respectively.

5. CONCLUSIONS

The work provided a more accurate knowledge about the cooling package behavior, allowing a better estimation of water radiator dimensions to meet certain requirements of ATB temperature. Currently, there are various software to calculate the thermal rejection of cooling systems, but in the majority of the cases the software license costs are high, preventing its acquisition. These software's use the methodology CFD as mathematical solution, such as KULI® (Magna), Flowmaster™, FLUENT®, StarCD®, UH3D®, CFX® and PAM FLOW®.

The calculation methodology allowed the evaluation of the cooling system behavior, as well as was very useful during the cooling test to check if the acquired temperatures were consistent, so avoiding possible instrumentation problems. This justified that the calculation methodology was implemented in a spreadsheet to enable the iteration of the input data.

The utilization of this methodology should also be tied with preliminary studies of each design, where should be provided the most possible number of versions that has this design. In this way is possible to obtain a radiator family more compact that meets this configuration avoiding an oversized system.

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