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Modelling of the engine cooling system of a locomotive engine with the aid of a unidimensional calculating program

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Abstract*– Nowadays the thermal load of components for alternating internal combustion engines becomes higher and higher. Therefore, not only the materials and components have to be enhanced, but also the way in which we design cooling systems acquires a larger importance. Optimization of cooling systems design is necessary because of several reasons. On the one hand the remit and requirement to the components of the present must be accomplished and on the other hand, a more important point has to be respected, which is reducing fuel consumption, also closely related to the emissions of exhaust gases. The best way to design and model these systems is to create and use simulation tools. The optimal designing of cooling systems permits saving time during the construction process, reducing the amount of required experimental work, thus improving the economic aspect. This paper is about illustrating the computer program "Flowmaster" (a thermal, hydraulic simulation software) to represent a real circuit of a locomotive into the so called "simulation software". It is built the virtual model of the cooling system of the EMD 12N-710G3B-MUI engine installed in a locomotive, using the FlowMaster thermofluid-dynamic circuit simulation program. The models generated can be used to predict the performance of the cooling system under different conditions such as cold start and working under a singular drive cycle.

Keywords—Cooling systems, simulation tool, virtual model, FlowMaster.

I. INTRODUCTION

Optimization of engine thermo-hydraulic systems is of paramount importance in the improvement of combustion engine performance and the reduction of fuel consumption. Due to the low production volume, in locomotive design it is costly to perform extensive testing of prototypes like in the automotive industry, making strategic to base the engine systems development on computer simulation methods and tools, coupled with experimental methodologies.

Nowadays the design and construction processes of new thermal-hydraulic cooling and lubricating systems can be based on system level 1D thermo-fluid simulations, which allow the analysis of complex piping systems to be investigated [1-2]. The availability of system level tools such as FlowMaster allows for complete descriptions of entire arrangements to be considered, provides the components of the systems to be designed are properly and reliable characterized, making use of the catalogue of components representing items such as pipes, bends, valves, reservoirs, pumps etc. The flow characteristics of most of these components are based on theoretical correlations and functions experimentally validated by the British Association for Fluid Dynamics Research, as summarized in [3]. The biggest advantage of these tools is to easily change and add components without the need to redesign them [4]. CAE tools help to create but cannot displace the execution of experiments which creation is time-consuming and costly. So, the best way to save time is by reducing the amount of required experimental work.

The fuel consumption and the corresponding emission of exhaust gases are directly proportional to the demanded power of locomotive engines. This translates in the increasing thermal cargo of alternating internal combustion engines, and therefore the remit and requirement of designing cooling systems acquire a large importance. Besides several options to reduce the consumption, it is of great importance, related to the cooling systems of the engines, to optimize their combustion by regulating the temperature to its optimum. In this respect, the best way now to design and model these systems is to create and use simulation tools. The optimal designing of cooling systems permits reducing the fuel consumption related to the exhaust gases by knowing the several characteristics of the cooling system while designing it. That reduces the amount of required experimental work, thus improving the economical side. On the other hand, in modern locomotives the engine power, the cooling system of the fresh intake air after the compression stages in the superchargers, and its exhaust gas control system determine the architecture of the hydraulic part of the cooling system (topology of the cooling system with different designs for cooling the recirculating exhaust gases and intake air) [5], while the end use of the locomotive generally determines the architecture of the air part of the cooling system (ventilation system).

Flowmaster is a thermal hydraulic simulating tool that can be integrated with, for instance, a heat exchanger design tool, and a CFD software to optimize the election and construction of the components of cooling circuits [1]. Flowmaster enables the evaluation of the mass and heat flow exchanges in components, and allows for the evaluation of thermal fluid variables, such as pressure and temperature at any network node of a cooling circuit [6].

[7] relates the hydraulic and thermal study of a commercial original cooling system circuit. Established the specifications of the engine, the performance characteristic and the operating point of the pump, the heat to be evacuated is estimated, the fan and radiator suitable for the thermal capacity demanded and adjusted to the space and location restrictions given by the

particular application are selected, the system thus assembled is modelled and compliance with the coolant and air temperatures for the engine full load conditions is ensured.

The objective of the work is to predict the behavior of the thermohydraulic system under conditions of maximum load for the traction system. The program can be automated to simulate different engines and component operating conditions. The first section of the paper addresses the model of a diesel locomotive cooling system, the second focuses on the particularities of the component models available in the computational tool, and the third section presents some elements of solutions of the simulation.

II. Heat transfer basics and engine cooling system

Heat transfer is thermal energy in transit due to a temperature difference. Whenever a temperature difference exists in a media or between media, heat transfer must occur. Part of the heat caused by the combustion of fuel in internal combustion engines diverts to the engine parts, while other part exits over the exhaust gases [8]. Three heat transfer mechanisms are present in the components of an engine cooling system: *conduction* due to a temperature gradient across a stationary medium, *convection* due to a temperature gradient between a surface and a moving fluid, and *thermal radiation* between media at different temperatures [9].

Heat produced in the engine is transferred through the walls of the cylinder Block and cylinder heat by conduction. The convection heat transfer process, comprised of conduction and advection (energy transferred by the bulk, or macroscopic, motion of the fluid) is present between the engine fluids in motion and the bounding walls at different temperature gradients, according to Newton’s law of cooling, and the convection heat transfer coefficient, dependent on conditions in the boundary layer, influenced by surface geometry, the nature of fluid motion, and an assortment of fluid thermodynamic and transport properties. Convected from the in-cylinder gases to the working chamber walls, part of the heat is stored in the surrounding masses, and part transported through the solid walls and convected to the moving cooling fluids circulating in the engine. From the cooling fluids the heat is convected to the air by heat transfer in the heat exchangers, radiators and aftercoolers. To some extend the engine solid parts are cooled by convection when the airflow passes under the hood. Radiation in combustion engines occurs mainly in presence of soot during combustion, particularly in diesel engines [10].

Flowmaster® outline. Flowmaster® is a 1D fluid flow modelling software which performs hydraulic calculations in networks, thermal and transient calculations, and is used to analyse fluid flow conditions in simple and complex piping systems. Each component of the cooling loop can be easily modelled by means of standard components of Flowmaster® library. It provides a graphical virtual environment where it is possible to design, refine and test the entire fluid flow system. The Flowmaster® single-phase steady state and transient modules have been specifically designed for modelling the effects of heat transfer in many application areas. The transient modules also allow transient events to be analyzed.

Each Flowmaster® component represents a mathematical model of an engineering component. Selected components are connected via nodes to form a network, which forms the actual computer model of the flow system. Data tables, curves and surfaces can be used to define the operation and performance of each component. When an analysis is run, pressure, flow rates and temperature are calculated throughout the network. The possible results obtained in each connection are volumetric flow, velocity, pressure, mass flow rate, temperature, fluid density and viscosity.

The governing equations used by Flowmaster are the conservation of mass, energy, and momentum (*Navier-Stokes* equations). These are used to calculate the flow into and out of the components as a function of pressure. The flow is then eliminated from the equation using continuity at each node. This leaves a set of equations, solved simultaneously during each run process. The resulting pressures are substituted back into the component equations to calculate new estimates of the flows. Any pressure specifying components impose a pressure at its connecting node. This iterative process is repeated until the achievement of stable values. When solving a transient heat transfer problem, an iterative solution is adopted because of the dependence of the specific heat on the solution. Flowmaster treats the circuits like a network with lots of nodes linking the components like pressure and flow sources, pumps and fans, solid components, heat exchangers, different valves, discrete loss components, pipe components -straight pipes, bends and junctions with their pipe length diameter and roughness-, controllers, gauges, thermostats, among others, dependent on the version of the software under progressive development.

Locomotive cooling systems. Modern cooling systems are based on complex coolant networks with different temperature levels in different hydraulic systems. State-of-the-art systems consist of two water circuits. The first circuit handles, at a high temperature level, heat management of the diesel engine. The second low temperature level circuit indirectly handles the heat regulation of the charged air for the diesel engine. Due to the design of locomotives, the ram pressure created by the vehicle cannot be used so that cooling system needs fans to create an active forced cooling airflow [1]. These fan drives consume considerable energy. The cooling unit as considered in this study includes the hydraulic and fan-driven cooling air network.

The cooling systems applied to transport engines are very diverse. However, their “structure” is based on the traditional scheme of heat transfer from the engine to a cooling device. Diesel engine cooling systems generally can be subdivided into the ventilation cooling system, diesel engine water cooling system, charge air cooling system and all kinds of fluids (oil, hydraulic oil, etc.) cooling systems. The cooling system of the diesel engine itself is a combination of cooling airtight water, engine oil and supercharged air-cooling equipment, which includes surface heat exchangers (radiators), fan units, air passages, shutters, and heat sinks.

III. Modelled engine and cooling system features

The engine is a turbocharged aftercooled two-stroke uniflow scavenged 45° V diesel engine, an EMD 12N-710G3B-MUI model with four poppet exhaust valves in the cylinder head. Cylinder heads, cylinder liners, pistons, piston carriers, and piston rods can be individually replaced. The block is made from flat, formed, and rolled structural steel members and steel forgings welded into a single structure. Each bank of cylinders has a camshaft which operates the exhaust valves and the unit injectors. The Mechanically actuated and controlled Unit Injectors (MUI) use the engine camshaft and push rods to generate fuel injection pressure, and a mechanical linkage system to control the amount of fuel injected into the cylinders. Placed the powertrain compartment in the locomotive, it is organized that the cooling air of the cooling system enter sideways of the locomotive though grills, passes the radiator and exits the vehicle at the top supported by the suction of the Fan. It is a habitual system that is used in a lot of trains.

The heat sink consists of two radiators sitting vertically in the locomotive with horizontal flow [11]. Each bank will receive half the water flow from the coolant pump. Each bank is a two-pass crossflow configuration with the inlet and outlet connections on the same tank. Two fans are mounted above the radiator panels and draw air through the radiators. The fan performance is bases on a a normal EMD 9 blade 48” diameter fan.

The main data of the system are: 3300 hp engine power at 926 rpm, total engine heat rejection of 108000 Btu/min; 1000 gpm pump water flow with 20 psi maximum pressure; 5500 lbs/min air mass flow of two in-parallel cooling fans rated at 1840 rpm; air pressure losses of 1 water inch for the duct and 1,2 water inches for the radiator.

A schematic of the liquid cooling circuit of the locomotive is shown in Fig. 1, where there are represented the engine, the radiators, expansion water tank, and oil cooler. The system consists of the engine cooling jacket and the aftercooler encompassed in the same set, two radiators and, at the outlet of these, two branches, one through the air compressor and the other through the oil cooler that converge towards the pump inlet [12]. At the engine outlet a small amount of coolant is bled into an expansion-deaeration tank. The two radiators are traversed by air currents drawn by two vertical axial fans.

The information provided for the construction of the locomotive cooling system model is: general diagram of the complete cooling circuit, radiator data sheets for the aftercooler and engine subcircuits, refrigerant used, P-Q pump and fan performance data and the external load available, engine data sheet, heat dissipated by the sources (engine, aftercooler, air compressor, oil cooler), geometrical and thermos-hydraulic information of the radiators and expansion tank, operating temperatures of coolant and air at radiator inlet and outlet. This information is presented in the next section devoted to the characterization of the components.

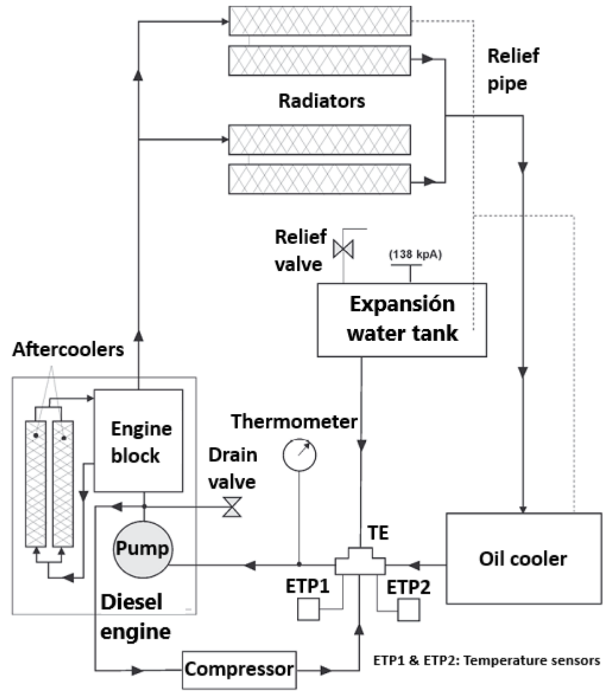

Fig. 1 Overview of the locomotive cooling system [12]

IV. Modelling methodology

Basic recommendations for modeling circuits with Flowmaster. Initially, the plans and technical specifications of the real locomotive cooling circuit are analyzed. To build a model that simulates the variations in air flow as it passes through the engine path, engine room, radiators, backwater, fan, atmosphere, it is desirable to have results of fluidic computational dynamics (CFD) calculations. The flow and resistance characteristics of all the components that participate in the air circuit must be studied.

The modeling process in Flowmaster starts by creating a catalog of system components and a catalog of performance curves for each project created; then, it is configured a pre-design of the circuit on FlowMaster space, the characteristics of the components are added into the database: pump curves, both torque and head-versus-flow, pressure loss curves for discrete components such as exchangers, radiators, piping, joints, heat dissipation characteristic surfaces of exchangers, and radiators.

By comparing the models developed with the help of the 1D program with the experimental tests, it will be possible to conclude about the suitability of the models. If the models obtained are close to the real circuits, the use of the characteristic parameters will generate the same results as the physical model. The characteristic parameters of the engine are the revolutions, load and speed of the locomotive.

Configuring the locomotive cooling system Model in FlowMaster.

Using the information provided by the manufacturer and observing the actual system layout in the locomotive powertrain room, the circuit illustrated in Fig. 2 has been built in FlowMaster, considering the components: water pump, engine block (left and right banks), turbocharger, cylinder liners, water jumpers, cylinder heads, water raisers, water return header, pipes, vent pipes, water manifold, radiators, aftercooler, radiator fan, expansion tanks, expansion tank equalizing pipe, suction pipe, among others [13].

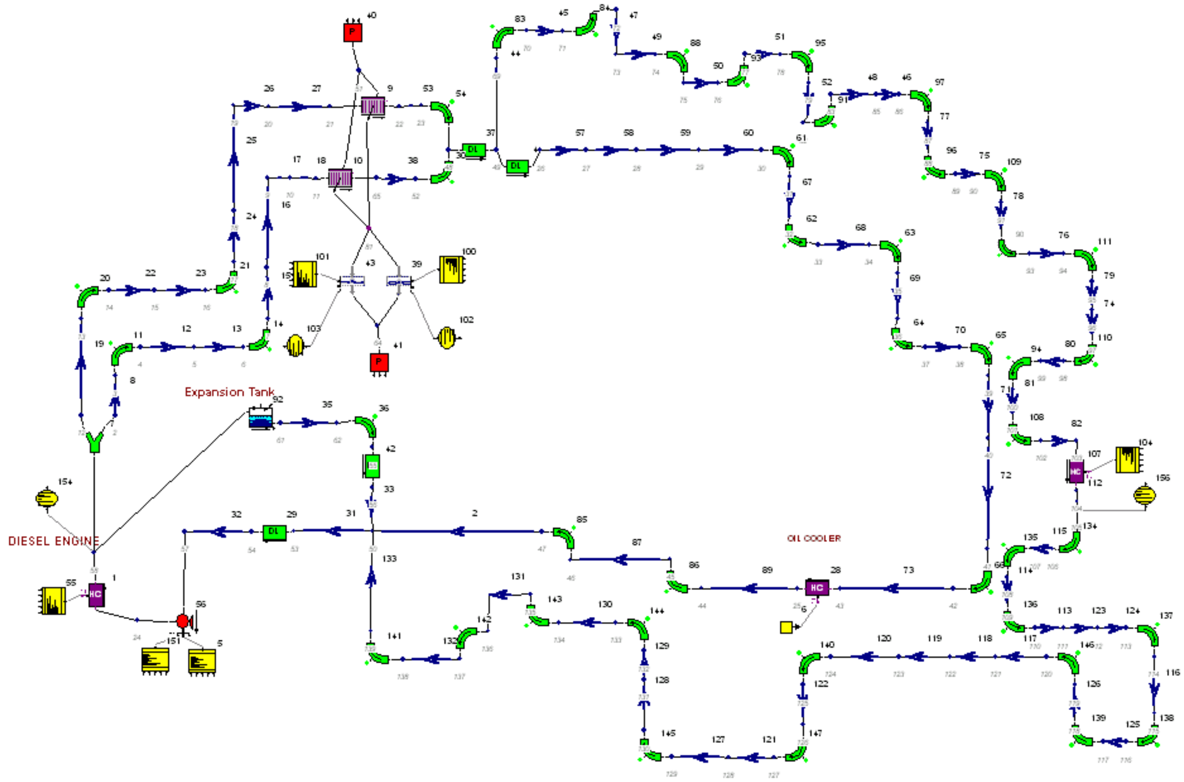

Fig. 2 Model of the cooling circuit

A) *Characterization of heat sources*

In this model, the diesel engine is characterized as a heat exchanger, a thermal power input controller to be dissipated and a discrete loss of pressure. Through a master controller it can be assigned the power that the engine will dissipate depending on the speed, according to an initialization script (this script accesses the dissipated power according to the locomotive's operating cycle, provided in an Excel sheet; the initialization script and other scripts used are edited in the appendix), according to the information provided by manufacturer, or after estimating the cooling power as a percentage of the engine's power. This value of the heat to be dissipated depending on the effective power of the motor is transferred to a "Heater-Cooler" type heat exchanger element. The hydraulic resistance to the passage through the engine is calibrated with the passage section and the loss coefficient. A "Gauge" type element is used to monitor the temperature at the engine output. Fig. 3 illustrates the combination of these components.

The rejected heat rate of the engine is shown in the graph of Fig. 4, with a maximum value of 1678 kW.

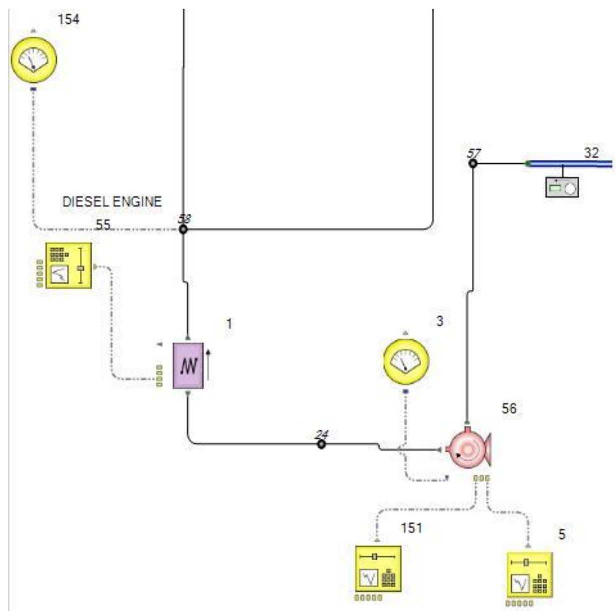

Fig. 3 Detail of the modelling of the motor as a heat source and the pump.

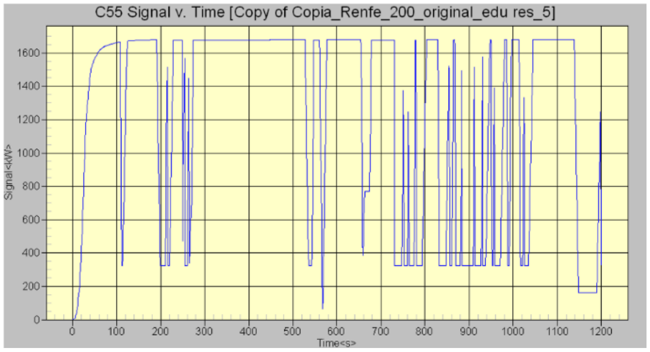

Fig. 4 Thermal power dissipated by the engine as a function of the locomotive's operating cycle.

Since the pressure losses per engine block are not known, the determination of the equivalent passage area and the loss coefficient was made by an approximate procedure that considers the availability of pressure losses in the engine after subtracting the available head from the known pump. The pressure losses due to the hydraulic resistance of pipes and radiators are added. The validation of this procedure was carried out in an Excel sheet.

Both the oil cooler and the compressor cooler have been modeled as "Heater-Cooler" elements as well. The values of its pressure loss and area coefficients have been reasonably tuned to adjust the flows and follow the proper behavior of the pressures.

B) Characterization of the pump.

The pump of the system (16G pump) has been characterized with its H vs. Q curves, according to the information given by the manufacturer. Indicating the total estimated flow rates given (1090 gpm) in the characteristics of the pumps, the pressures of the pumps were determined for the engine speed of 926 rpm. For a stable operating point of the system, hydraulic resistance (pressure losses) must be equal to that pressure availability of the pumping system and with this information the general characteristics of pressure losses were determined as $\Delta p = K \cdot Q_v^2$.

In addition to the pumps, two controllers are provided in the circuit to configure the operating status and the pump revolutions. A view of the engine pump configuration is shown in Fig. 3. In addition, a speed sensor has been configured to monitor the pump operation during the simulation. The performance characteristic of the 16G pump is illustrated in Fig. 5.

To validate the characteristics and pressure loss coefficients in the components, trend lines have been added to the pressure loss curves of the 16G pump and the K_{res} in an Excel spreadsheet, Fig. 6, obtaining the following relationships:

$$\Delta P_{pump16G} = -210963Q_v^4 + 16064Q_v^3 - 504.85Q_v^2 + 1.1033Q_v + 5.3751 \qquad (1)$$

$$\Delta P_{Kres} = 962.97Q_v^2 \qquad (2)$$

The operating point is defined as:

$$\Delta P_{pump16G} = \Delta P_{Kres} \qquad (3)$$

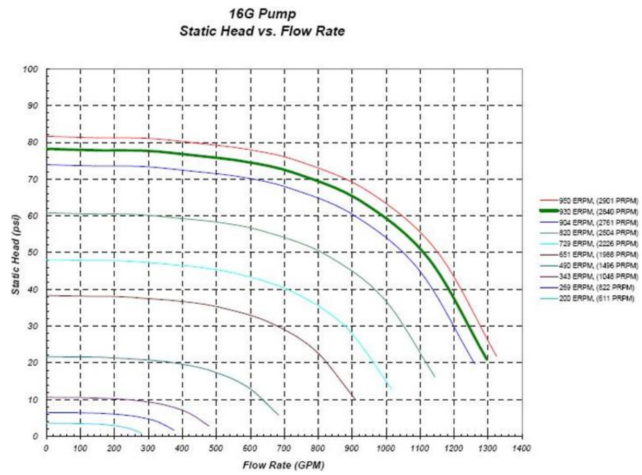

Fig. 5 Feature of 16G pump [11]

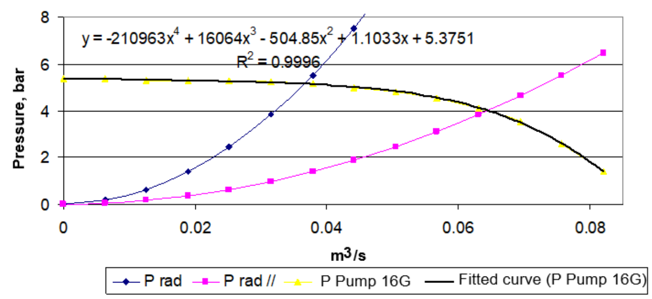

Fig. 6 Matching the pump performance curve and the system resistance.

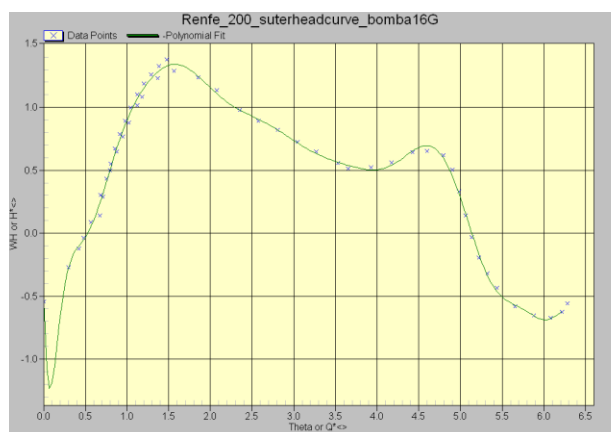

Fig. 7 Feature of the centrifugal pump in the FlowMaster program.

When solving the system of equations, the operating point of the hydraulic system is found Qv = 0.0655m^3/s. When entering the pump characteristic in FlowMaster, a procedure is followed that consists on the construction of the head and torque characteristic against flow from pre-existing centrifugal pump templates in the database, for specific determined speeds. The presentation of the dimensionless H vs. Q characteristic of the 16G pump is illustrated in Fig. 7.

C) Characterization of the radiator.

The radiator is modelled with the "Radiator" type heat exchanger element. A radiator calculation program has been used to, after a sweep of air and refrigerant flows, determine the maximum transferred heats and construct the characteristics Q/ITD·Area vs. Mflw1 & Mflw2/Area. Table I references the inlet and outlet temperatures of the coolant and air as well as the radiator area used in the calculation of the characteristics.

TABLE I
TEMPERATURES AND AREAS IN CHARACTERISTIC DETERMINATION Q/(ITD·AREA) VS. MFLW1 & MFLW2/AREA

Engine Radiator	°F	°C		
The Air (Tamb)	113	45	Area (m^2)	2.979
Cooling Tea	205	96.11	ITD. A (°C.m^2)	152.257
DT=(Ter-Tea)	92	51.11		

The thermal characteristic of the radiator has been entered in the form Q/(ITD·Area vs. Mflw1 & Mflw2/Area), along with the sections of passage of the fluids and the characteristics of pressure losses to the passage of coolant and air, with the particularity that in the latter it has been decided to enter the total characteristic of pressure losses to the passage of air throughout the system from the atmosphere through the grille and radiators to the exit of the fans (the latter work as vacuum cleaners). The characteristic Q/(ITD·Area vs. Mflw1 & Mflw2/Area) of the radiator can be seen in Fig. 8.

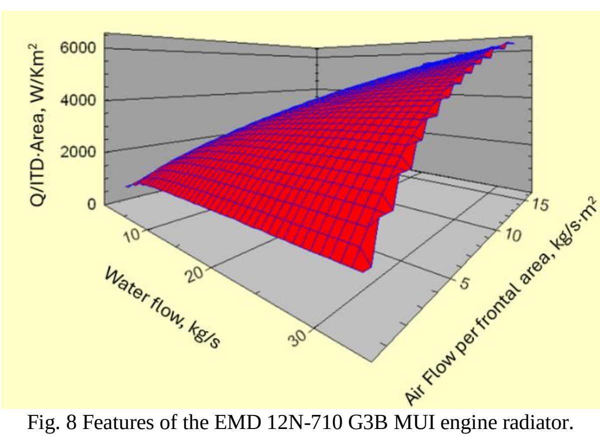

Fig. 8 Features of the EMD 12N-710 G3B MUI engine radiator.

The pressure losses on the water side of the radiators were also obtained during the sweep leading to the definition of the thermal characteristics. The radiator calculation program only gives pressure losses on the air side corresponding to the radiator core, so experimental values must be used, either by analogues or, in the case of direct design, using the results of CFD calculations for the actual modeled geometry. Fig. 9 illustrates the pressure loss characteristics of the coolant side of radiators

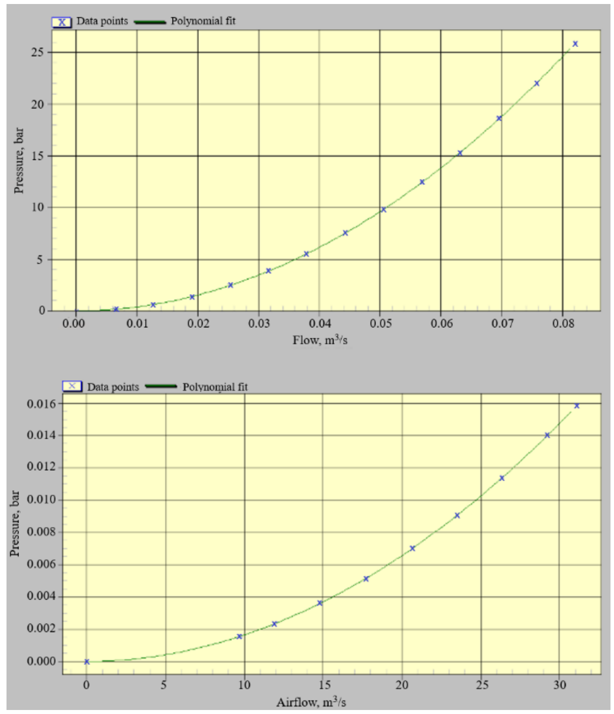

Fig. 9 Pressure losses as coolant (upper) and air (lower) pass through the engine radiator

As experimental information on the pressure losses on the air side was available, it has been used to characterize the engine radiator, assuming symmetrical distribution of the air flows drawn by the fans. The characteristics of pressure losses as air passes through the radiators is illustrated in the right graph of Fig. 9.

The other parameters entered in the information required by the program were the coolant and air passage areas and the hydraulic diameter. For both radiators, the representative area in the coolant passage has been taken to be equal to the area of the inlet, $A_1 = 0.006207$ m^2; the representative area in the air passage has been taken equal to the effective area of the exchanger panel, $A_2 = 2.98$ m^2, and the hydraulic diameter has been taken equal to the diameter of the pipe, $d_h = 0.0889$ m.

At the exit of the radiators, both refrigerant and air flows converge and are directed towards the suction of the pumps (although the refrigerant flow here is divided: a greater amount will flow to the oil cooler and the remaining part will flow to the compressor), as illustrated in Fig. 10.

D. Characterization of pipes and fittings.

In the hydraulic part, it remains to be commented that, after characterizing the main components, the discrete pressure losses 4 and 37 of Fig. 10 have been adjusted to achieve the distribution of the flows according to the experimental values that were known. Wherever the junction goes, FlowMaster's help recommends facilitating the running of the program to replace the "T" connections with discrete losses connected through a geometric node and this is what was done, just where there were originally some diameter transition elements. It is further clarified that adjustments can be made directly on the heat exchanger elements of the oil and compressor themselves.

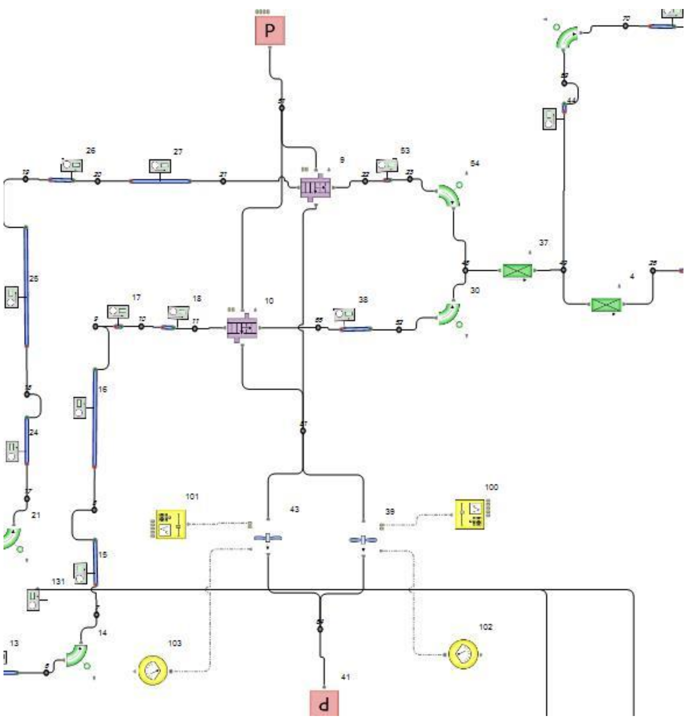

Fig. 10 Detail of the connection of the two radiators in parallel on the engine side and the two fans.

Prior to the introduction of the characteristics of the components and accessories of the circuit, systematized calculations were made in an Excel sheet. To ensure the fairness of the magnitudes that were going to be introduced of the components. For validation, apart from the model in FlowMaster, the load characteristics of the system and the pressure of the pump and fans have been built in the Excel sheet. Fig. 11 shows the matching of the fan characteristics and the system resistance. Again, this is done to ensure that the information that will be fed into the model in FlowMaster makes sense. In Fig. 6, the same was done with the characteristics of the water side.

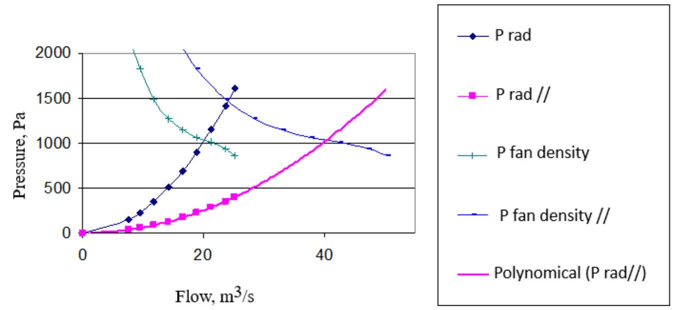

Fig. 11 Graphical solution of the fan system – pneumatic loads.

When configuring the information required by FlowMaster for the pipes, in addition to the diameter and length, a roughness of 0.025 mm has been specified for rolled steel pipe and the program has been told to calculate the friction losses by the Colebrook-White equation, although the coefficients λ found could have been used if the "fixed friction factor" option had been configured. The dialog box looks as it appears in Fig. 12.

Thermal inertia can be configured for pipes, but we have not considered this in the model. An external convection coefficient value of h = 4 W/m2 ºC has been predicted, although we do not have the information regarding the temperature surrounding the motor and the pipes (temperature in the engine chamber), so the value of T_a = 50 ºC has been assumed. All the pipes have been left with the same configuration, that is, the same thermal and friction models.

In the hydraulic part, the modeling of the expansion tank remains to be commented on. For the expansion tank corresponding to the engine, the information is provided as illustrated in Fig. 13.

E. Characterization of the fans.

On the air side, it was already commented that the load characteristic to be overcome by the fans was included in the passage losses through the radiators. It must be noted that in FlowMaster the performance data supplied by the company have been introduced to calculate the difference in pressures handled by the fan (the characteristic reported by the manufacturer has been given for a density of 0.070 lb/ft^3 or 1.121 kg/m^3).

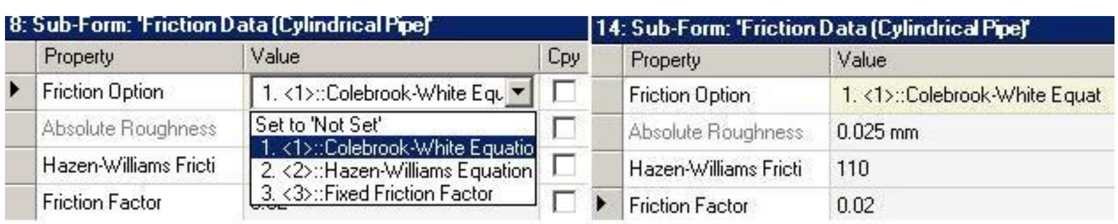

Fig. 12 Configuration of pipe friction information.

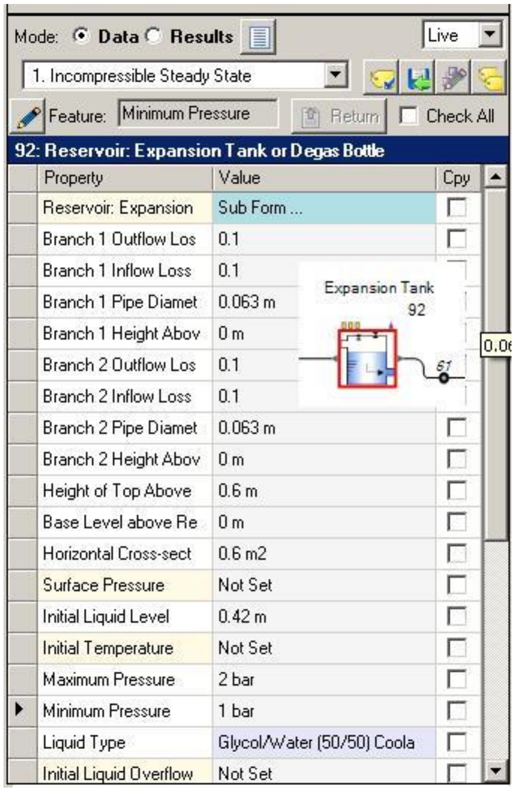

Fig. 13 Engine expansion tank dialog box.

Fig. 10 illustrates the feature of the parallel fan coupling in the FlowMaster model [14]. In the same figure it is given one of the "Pressure Source" type elements, used to define the environment. This component is only defined by the type of fluid with which it operates, the pressure and the temperature. Fig. 14 shows the fan characterization in the FlowMaster dialog box.

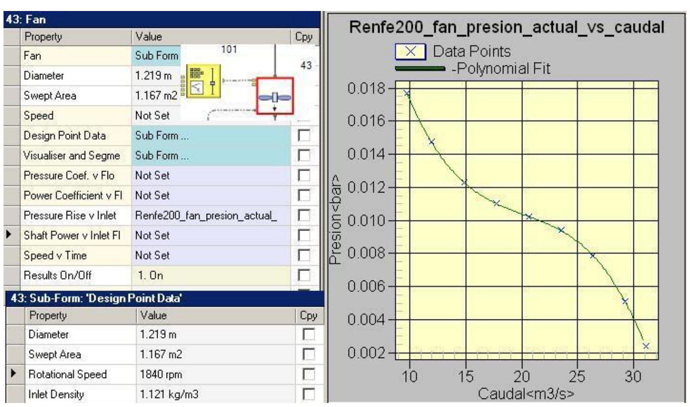

Fig. 14 Characterization of the fan used in the cooling system.

The parameters external diameter of the blades, suction area (value that was calculated from the plan), revolutions and the pressure against flow characteristic (volumetric flow) are entered. The speed regime has been imposed on it with a "Master Control" type element.

F. Conditions of experiment

The actual data of the engine cooling circuit has been measured by experimental work under special but realistic conditions. The tests have been run until the total stabilization of temperature of the liquid circuits (water and oil). The minimum time of the test was 30 minutes depending on the previous warming state of the tested locomotive. The experimental data given for the radiator is presented in Table II.

TABLE II
RADIATOR DESIGN DATA

Heat Rejection (total)	108000 Btu/min	1897 kW
Inlet temp	205 F	96,1 C
Ambient temp	113 F	45 C
Airflow	7202 lb/min	54,50 kg/s
Coolant mass flow (per radiator)	4675 lb/min	35,37 kg/s
Coolant volume flow (per radiator)	545 gpm	123878 l/h

The temperatures of the fluids through the radiator after the experimental test are presented in Table III, under conditions: Max Load, Thr 8, Fan 1 & Fan 2 ON (max heat rejection).

TABLE III
EXPERIMENTAL TEMPERATURES OF WORKING FLUIDS THROUGH RADIATOR

	SI Units			
	Core 1	Core 2	Core 3	
Test Data				
Inlet water temp	78 C	78,3 C	78,15 C	
Outlet water temp	64,4 C	64,7 C	64,55 C	
Coolant flow	86760 l/hr	86760 l/hr	173520 l/hr	
Air in	22,1 C	23, 3 C	22,7 C	
Air out	61,7 C	61,7 C	61,7 C	

The coolant fluid in the circuit is 35/65 % Glycol/ water, with the calculated fluid properties presented in Table IV.

TABLE IV
COOLANT PROPERTIES

Fluid Properties (under test conditions)		
Coolant	Type	Water/Glycol
	Mixture	65/35 %
	Density	1028 kg/m^3
	Cp	3,75 kJ/kg K
Air	Density	1,015 kg/m^3
	Cp	1,18 kJ/kg K

G. Experimental Dissipated Heat (real data)

The heat that is transferred to the air in the experiment measured for radiator1 and radiator2 are, respectively 1086,8 kW and 1053,9 kW.

V. SIMULATION RESULTS

The heats delivered to the cooling system are shown in the graph in Fig. 15. These heats were provided by the manufacturer; the heat supply has been left according to the operating cycle of the locomotive, even though the tests were done for constant loading. This has been left for illustration, but for the evaluation of the model it is sufficient to consider that all the charges are constant. In the same figure there appear the graphs of the heat dissipated by the individual radiators.

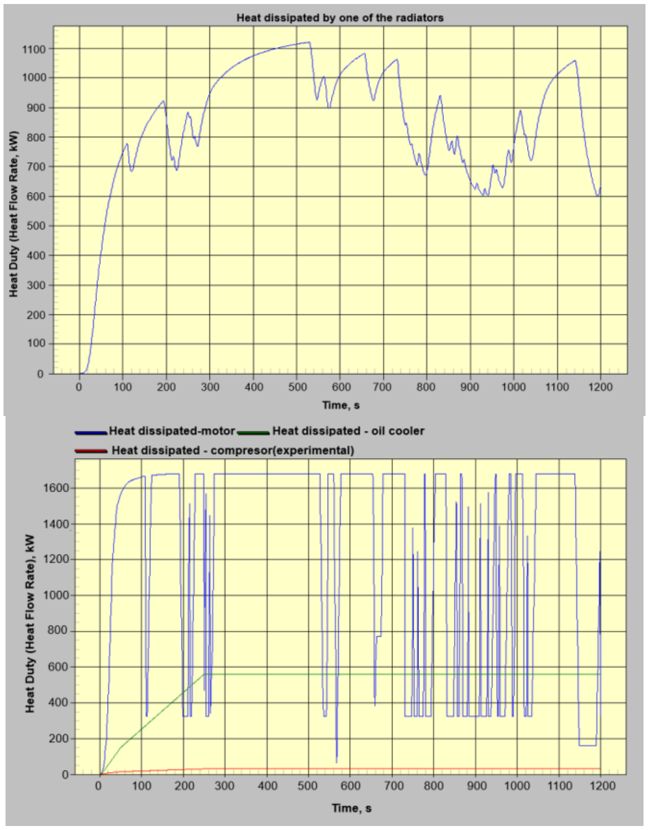

Fig. 15 Heat contributed by the engine, compressor and oil cooler to the cooling system. On the lower it is presented the graph of the variation of heat dissipated by one of the radiators.

The flow rates obtained at the branches of the engine radiator are shown in Fig. 16.

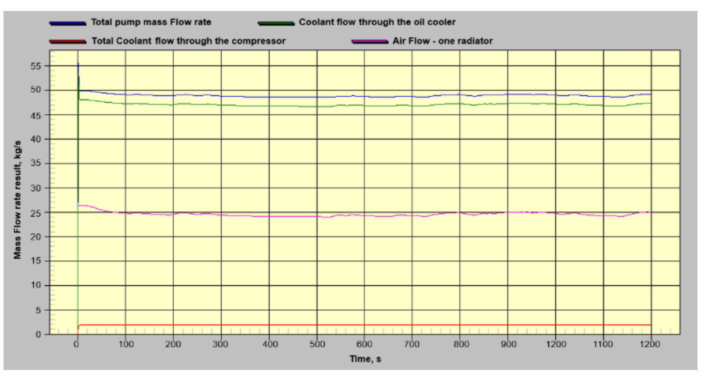

Fig. 16. Air and coolant flow through the engine cooling system.

The coolant inlet and outlet temperatures through one of the radiators are shown in Fig. 17.

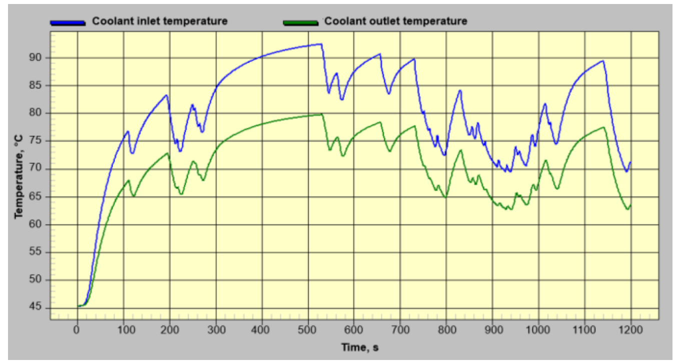

Fig. 17. Coolant inlet and outlet temperatures in one of the radiators.

The experimental results during the operation with constant load of the locomotive engine are those shown in Fig. 18.

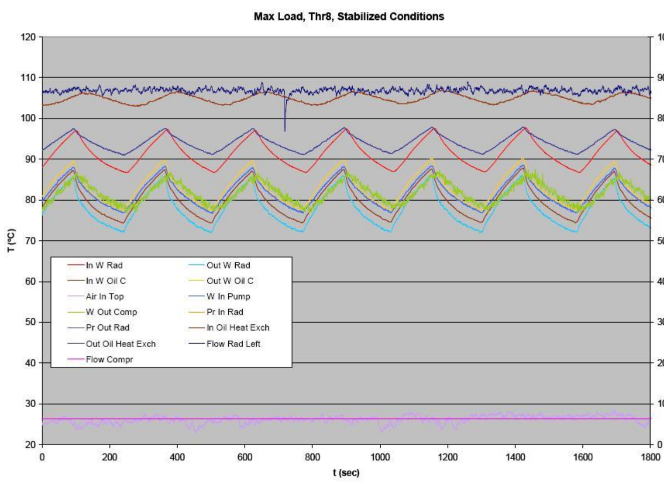

Fig. 18. Fluid temperatures and flows of the locomotive cooling system after simulation.

The results of the model during cyclically loaded operation of the locomotive engine are shown in Fig. 19.

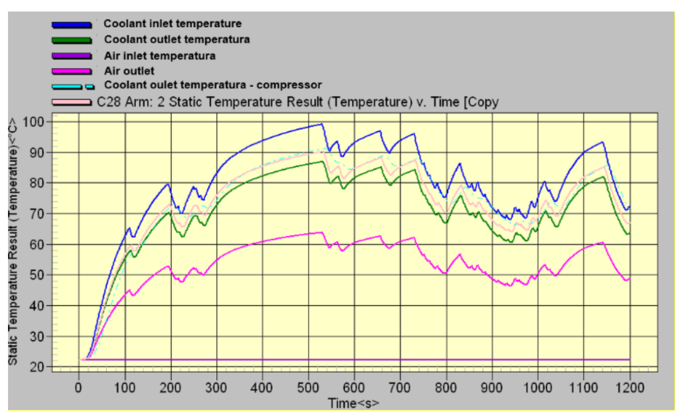

Fig. 19. Variation of the coolant and air temperatures as they pass through the components of the locomotive cooling system.

Variations in signals are due to cycle transients and to fan connections and disconnections.

CONCLUSIONS

It has been the purpose of this paper to illustrate the benefits of the thermal-fluid tools during the design and study of the thermal fluid systems, with the emphasis placed in an engine cooling system. With appropriate real data for each component, an analysis can be carried out during the design stage of a project, and during academic activities.

Provided that the input data are reliable, the resulting model can be used as a predictive tool once validated against experimental results. This enables anticipating the performance of the cooling system under possible changes, modifications, or operating conditions, and facilitates the evaluation of alternative components such as engines, radiators, pumps, fans, thermostats, among others.

The virtual model developed is specific to the locomotive cooling system being studied and can be used to predict its performance under different configurations, as long as the thermo-fluid-dynamic characteristics of the components are known.

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