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MULTIPORT APPROACH APPLIED TO THERMOFLUID SYSTEMS

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Abstract. *The purpose of this paper is to present how a numerical thermofluid library was developed. This library comprises a set of basic components from which it is easy to model large thermal-hydraulic networks and modify, if required, the design of the system studied. These basic components were specified based on bond graphs and multiport methods and developed by using lumped parameter mathematical models approach.*
To complement the Thermofluid library, the Cooling system library is presented. This is an empirical library composed of a set of specific components, which are constructed based on experimental data. This paper ends up with an overview of the validations that have been made and the extensions planned to obtain better results for engine cooling systems.

Keywords: *Engine cooling, Thermal system, Bond graphs, Multiport, Lumped parameters*

1. INRODUCTION

In the analysis of phenomena in engineering, the Computational Fluid Dynamics methods are unquestionable useful. Nevertheless, this approach shows its limitations when we are interested in the behavior of multidisciplinary components and systems. The lumped parameter approach fits these kinds of systems. This awareness has led to the development of methods for structuring these kinds of problems.

The Bond Graphs, Karnopp *et al.* (1990), constitutes the most developed of these methods, but it has remained a tool used by the modeling specialist only. By applying the Bond Graphs method, we can extract some fundamental concepts, for example: choice of variables on ports, causality and multiport system.

Places where an element may be connected or *bonded* to another element to form part of a system are called ports. At these places, pairs of complementary variables are simultaneously constrained to be equal for the two elements. The choice of these variables is made taking into account some practical aspects: variables that are commonly used by experimenters and variables that are easy to manipulate.

The multiport method is a complement to the single or signal port approach and it enables constructing models of circuits based on physical connections of elementary components contained in a library.

Applying these fundamentals concepts, the Thermofluid and Cooling system libraries were created. These libraries comprise a set of basic and specific components from which it is easy to model large thermofluid and engine cooling networks. These basic and specific elements facilitate the study of phenomena whose knowledge is indispensable to analyze a whole system.

2. THERMOFLUID SYSTEM APPROACH

The description of a thermofluid system in terms of Bond Graph approach leads to identifying components like: thermodynamic accumulators, resistances and conductances, shown in Karnopp *et al.* (1990). By applying the multiport method to these Bond Graphs components, it is possible to specify a physical thermofluid library composed of multiport capacity, resistance and conductance elements. Also, this method allows us to divide the thermofluid library in three categories of components distinguished by their ports, they are: thermal, thermal-hydraulic and thermal-pneumatic categories. In this article only the thermal and the thermal-hydraulic categories are presented, however, it is important to note that a thermal-pneumatic category is already in development.

2.1 Thermal category

In this category all the components have thermal and signal ports only. The thermal ports are defined by the following variables: temperature[°C] and heat flow rate[W].

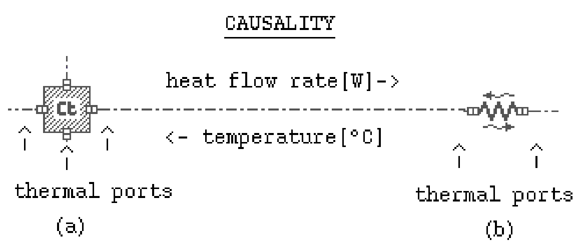
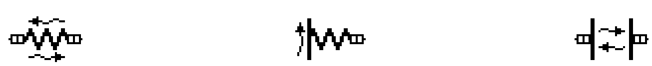


Figure 2- (a) Thermal capacity, (b) thermal conductance.

The thermal capacity component, Fig. 2a, contains a lumped parameter differential equation to predict transient variations of temperature in a solid mass as a function of heat flow computed by the thermal conductances, Fig 2b. These last components can be divided in physical elements as shown in Fig. 3.



(a) (b) (c)

Figure 3 - (a) Conduction, (b) free and forced convection with the environment and (c) radiation heat transfer.

A complete information about the thermal category components and their equations can be found in Da Silva (1999).

2.2 Thermal-hydraulic category

In this category, the components have thermal, signal and thermal-hydraulic ports. The variables used on thermal-hydraulic ports are: absolute pressure[bar], mass flow rate[kg/s], temperature [°C] and enthalpy flow rate[W].

Some components found in the thermal-hydraulic category are shown in Fig. 4.

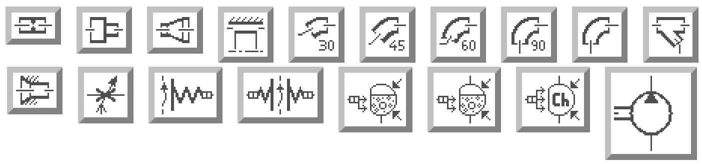


Figure 4 - Thermal-hydraulic components.

The thermal-hydraulic capacity, Fig. 5a, is an example of a multiport component having thermal-hydraulic and thermal ports. In terms of Bond Graphs, it is represented by the C-element in which pressure and temperature (intensive variables) are calculated.

In Fig. 5b is shown the thermal-hydraulic resistance, which is the Bond Graph R-element. This component computes both the mass flow rate through hydraulic resistances and the increase of enthalpy flow rate due to the fluid pressure drop, assuming the transformation is adiabatic.

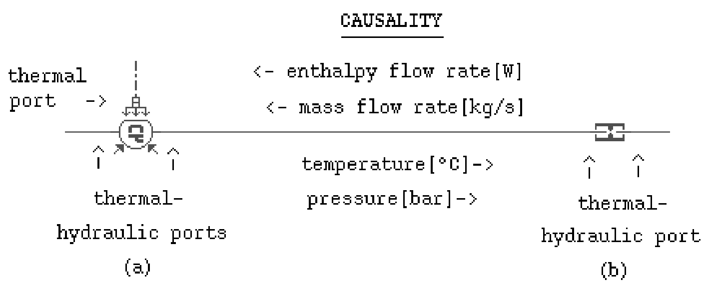


Figure 5 - (a) Thermal-hydraulic capacity, (b) thermal-hydraulic resistance.

Causality considerations. It is important to note that the flow variables presented in Fig. 5 are not defined as a true Bond Graph, because, the product of temperature and enthalpy flow rate is not a power. Enthalpy flow rate itself has the dimension of a power. This kind of causality is often called *Pseudo Bond Graph*, Rosenberg *et al.* (1983).

The true Bond Graph for thermal-hydraulic systems is represented by temperature and entropy flow rate. However, in this kind of systems, it is neither common nor comfortable to deal with entropy variables. For this reason, the following have been used; heat flow rate - on thermal ports - and enthalpy flow rate - on thermal-hydraulic ports, instead of entropy flow rate as a flow variable.

Choice of variables on ports. To enable an easier analysis of the system and a direct comparison between simulation and test bench results, the variables chosen on ports must be usual and readily measurable. Therefore, the thermal-hydraulic category aim is to compute the following main variables: pressure, temperature, mass flow rate and heat exchanges with the environment.

Lumped parameter approach. The thermal-hydraulic category is composed of lumped parameter components in which properties such as pressure are represented by a single representative value. A lumped parameter approach is used in a C-element to obtain a set of differential equations based on conservation of mass and energy to compute pressure and temperature respectively. In a R-element, this approach is based on conservation of momentum to calculate mass flow rate variable.

Also, some thermodynamics properties of the fluid are required to solve these conservation equations. They are: density, bulk modulus, viscosity, specific heat, enthalpy, thermal conductivity and cubic expansion coefficient. These properties are expressed as analytical functions of pressure and temperature.

It is important to notice that, as a first approach, some assumptions for the conservation equations were made taking into account a traditional engine cooling system.

Conservation of fluid mass (pressure is a state variable). In their work, Sidders *et al*. (1996) has showed that the continuity equation can be formulated in terms of the pressure derivative as follows:

$$\frac{dP}{dt} = \frac{1}{\left.\frac{\partial\rho}{\partial P}\right|_T}\left[\frac{d\rho}{dt} - \left.\frac{\partial\rho}{\partial T}\right|_P \cdot \frac{dT}{dt}\right] \tag{1}$$

Equation (1) can be reformulated using the thermodynamics relations for the isothermal bulk modulus, defined as $B_m = \rho / \left.(\partial\rho/\partial P)\right|_T$ and for the cubic expansion coefficient, defined as $\alpha = -1/\rho \cdot \left.(\partial\rho/\partial T)\right|_P$, (Wylen & Sonntag, 1985). It leads to:

$$\frac{dP}{dt} = B_m(P,T)\cdot\left[\frac{1}{\rho}\frac{d\rho}{dt} + \alpha(P,T)\cdot\frac{dT}{dt}\right] \tag{2}$$

Assuming a C-element with a constant volume, the density derivative term, $d\rho/dt$, is computed as follows:

$$\frac{d\rho}{dt} = \frac{\dot{m}_e - \dot{m}_l}{V} \tag{3}$$

where, V is the volume and $\dot{m}_e$ and $\dot{m}_l$ are respectively the volume entering and leaving mass flow rates. Combining Eq. (2) and Eq.(3) leads to the continuity equation formulated in terms of pressure used in the thermal-hydraulic capacity:

$$\frac{dP}{dt} = B_m(P,T)\cdot\left[\frac{1}{\rho}\frac{\dot{m}_e - \dot{m}_l}{V} + \alpha(P,T)\cdot\frac{dT}{dt}\right] \tag{4}$$

The temperature derivative term in Eq. (4) is often neglected in conventional simulation of fluid power system. However, for engine cooling systems simulations, it becomes a significant term because of the fast variations of coolant temperature.

Conservation of energy (temperature is a state variable). The first assumption used for the energy equation is kinetic and potential energy can be neglected, which gives

$$\frac{d(mu)}{dt} = (\dot{m}h)_e - (\dot{m}h)_l + \dot{Q} - \dot{W}_{cv} \tag{5}$$

where, u and h are respectively the specific internal energy and the specific enthalpy of the fluid and $\dot{Q}$ is the heat exchanged between the fluid and the environment.

Assuming that no work is done on or by the fluid, the work term is defined as $W_{cv} = PdV$. Noting that the C-element presented in this paper has a constant volume, this term can be neglected. However, the thermal-hydraulic category contains some capacities components such as expansion vases, which have variable volumes.

To rewrite the energy equation, Eq. (5), in terms of the temperature derivative, the following thermodynamic relationship is used:

$$\frac{dh}{dt} = c_p \cdot \frac{dT}{dt} + (1 - T \cdot \alpha) \cdot \frac{1}{\rho} \frac{dP}{dt} \tag{6}$$

where, c_p is the specific heat at constant pressure. Thus, by using the well known thermodynamic relation: $u = h - P/\rho$, it makes it possible to combine Eq. (5) with Eq. (6). After some algebraic manipulation, it leads to:

$$\frac{dT}{dt} = \frac{(\dot{m}h)_i - (\dot{m}h)_o - (dm/dt) \cdot h + \dot{Q}}{\rho \cdot V \cdot c_p} + \frac{T \cdot \alpha}{\rho \cdot c_p} \frac{dP}{dt} \tag{7}$$

The second term in Eq. (7) shows a cross-coupling effect with the continuity equation, Eq. (4). However, in the first version of the thermal-hydraulic category, this pressure derivative term is neglected. This assumption is supported by two aspects concerning an engine cooling system: the great heat transfers applied to the coolant and the slow pressure variation in this kind of system. Nevertheless, this cross-coupling effect between energy and continuity equation has been studied in fuel injection systems taking into account its influence over the simulation period.

Conservation of momentum considerations. The evaluations of pressure drops in the thermal-hydraulic components is totally based on the philosophy of the already existing Hydraulic Resistance library, (AMESim, 1999). Hence, it is possible to compute pressures and mass flow rates in orifices, pipes, hoses taking into account the wall compliance, expansions/ contractions components, diffusers, annular pipes, bends and T-junctions.

The mass flow rate is calculated from a steady state momentum consideration, hence the mass flow rate is an algebraic variable. It means that the mass flow rate varies instantaneously with the pressure drop variation. It is a great approach for engine cooling systems, which have small pressure spikes in the coolant circuit. Also, this assumption simplifies the system model by reducing the number of state variable.

Adiabatic resistance. To compute the increase of enthalpy flow rate when a fluid mass flow rate crosses a hydraulic resistance, the transformation is considered to be adiabatic. Hence, the enthalpy flow rate leaving the R-element is given by:

$$(\dot{m}h)_l = (\dot{m}h)_e + \frac{\dot{m}}{\rho}(P_d - P_u) \qquad (8)$$

where, $\dot{m}$ is the mass flow rate crossing the R-element and P_d and P_u are respectively the downstream and the upstream pressure.

3. ENGINE COOLING SYSTEM

From a thermodynamic point of view, engine cooling is an undesirable system, since it reduces power ratio produced by fuel flow rate. However, the engine materials, used in conventional vehicles have some metallurgical constraints, which limit the engine operation to a certain maximum temperature level. For this reason, as shown in Araci *et al.* (1999), an engine cooling system must provide some important aspects such as:

- Increase in coolant, oil and engine metals temperatures during cold start or low ambient temperatures and light route load conditions.
- Sufficient cooling of the oil and the engine metal temperatures under uphill and full load conditions.

By improving these aspects, it is possible to ensure an optimum compromise between important engine characteristics such as fuel consumption, engine life and gas emission with a good engine performance. Thus, following the increasing demands for engine cooling system efficiency and performance, the Cooling system library was created. Some components found in this library are shown in Fig. 6.

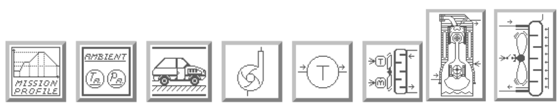


Figure 6 - Cooling system library.

This library is a set of components fully compatible with AMESim Thermofluid library. By combining these two libraries, it is possible to model dynamics behavior of an engine cooling system. For example:

- The time spent before an engine reaches its operating temperature after a cold start;
- The performance of a cooling system with a complex thermostat in which hysteresis and the thermal inertia of wax are taken into account.
- Prediction of the temperature and the pressure at the pump inlet allowing you to investigate pump cavitation.

3.1 Cooling system library components

The Cooling system library contains two kinds of elements: data components (without ports) and system components (with ports).

Data components. From these components, it is possible to define controlling variables such as car velocity, gear box ratio, road inclination percentage and ambient temperature. These are global variables, which means that they can be recovered by the system components.

System components. These are specific components such as centrifugal pump, thermostat, heater core, immersion heater, oil/coolant exchanger, expansion vase, exhaust gas recirculation (EGR) exchanger, engine, radiator and condenser-compressor from the air conditioning system. They are empirical elements based on experimental data.

Taking the radiator model as an example, it was suggested by (Michea, 1997) to use experimental data for the air and coolant heat flow exchanged for a constant inlet temperature difference between both. Hence, the heat flow exchanged during the simulation is:

$$\dot{Q} = \dot{Q}_{exp} \cdot \frac{\left(T_{coolant\_inlet} - T_{air\_inlet}\right)}{\ddot{A}T_{exp}} \tag{9}$$

where, $\dot{Q}_{exp}$ is the experimental heat flow given as a function of the coolant volumetric flow rate and the air velocity. Concerning this last variable, the model uses semi-empirical equations to compute the air velocity through the radiator and the condenser. It is computed as a function of the car velocity by using the radiator and condenser aerodynamics coefficients and the fan characteristic curve (volumetric flow rate as a function of pressure drop). If the fan is operating, a new air velocity through the ventilated area is calculated. This is the area in which the air velocity through the radiator is directly influenced by the fan. Hence, as shown in (Michea, 1997), a new heat flow exchange can be calculated as follows:

$$\dot{Q} = \frac{\dot{Q}_{exp\_nv} \cdot S_{nv} + \dot{Q}_{exp\_v} \cdot S_{v}}{S_{total}} \tag{10}$$

where, S_{nv} and S_v are respectively the non-ventilated and the ventilated area and $\dot{Q}_{exp\_nv}$ and $\dot{Q}_{exp\_v}$ are calculated as a function of the air velocity through the non-ventilated and ventilated area respectively.

3.2 Steady state models

The Cooling system library contains some components which use steady state experimental data. For example, the engine model uses a table of heat flow rate between engine and coolant as a function of the engine speed and the power developed by the engine shaft. Noting that the "*thermal inertia*" of the engine metal masses can not be neglected, it is not possible to obtain good results in transient simulations using this kind of model. For this reason, the thermal category has been used to model the cylinder block and heat transfer exchanged between the cylinder surface and the combustion gas.

4. VALIDATING A COOLING SYSTEM MODEL

The following sections present how the Thermofluid and Cooling system libraries have been validated using a RENAULT engine and describe the extensions planned to improve these libraries.

4.1 Which data for the validation

Before a car enters production, a prototype must undergo a series of tests to check if all the technical specifications are correct. Concerning the cooling system, you must make sure that the engine and all components within the engine compartment are sufficiently cooled, even under extreme external conditions.

For that purpose, cars are tested in wind tunnels through out the designing process, in order to simulate critical working conditions. Three main mission profiles are simulated: a high speed drive on a highway at high outside temperature, a drive on a road at the same temperature and a drive on a mountain road with a full loaded vehicle.

Several temperatures are measured to serve as criteria for the validation of the car. RENAULT uses the water temperature at engine outlet, the oil temperature in the engine and the oil temperature in the gearbox. With a classical validation test, it is not possible to get any other data about the cooling system. Only special tests made during the design of the car can provide extra data. Unfortunately these results are less easily available.

On the other hand, data from tests carried out to validate the heating of the vehicle cabin are available.

The wind tunnel enables us to simulate a cold start at very low outside temperature. In that case the results concern the water temperature at heater inlet and outlet, and the temperature of the air blown towards the driver.

4.2 The validation of the model

As previously said, the heat exchanged between engine and coolant in the simulation comes directly from steady state experimental data. Consequently only the steady state results can be considered as reliable. Then, as a first approach, the model will be validated for steady state simulations only.

Thus, the steady state temperatures provided by simulations and the ones provided by experiments are compared. For this validation, the cooling system of a gasoline engine has been simulated. The circuit is shown in Fig. 7:

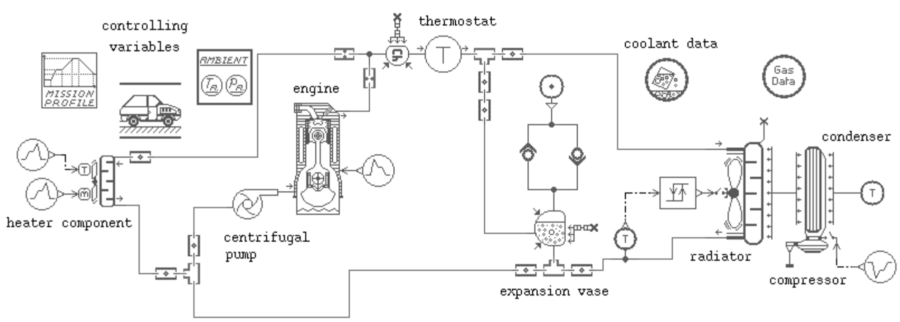


Figure 7 - Complete engine cooling system.

The comparison between experimental and simulation results is described in the following array for three working conditions: 200 km/h, 165 km/h and 130 km/h on a flat road under high outside temperature.

Table 1. Water temperature at engine outlet

Working speed	**Tests**	**Simulations**
200 km/h	104 °C	104.7 °C
165 km/h	99 °C	97.5 °C
130 km/h	94 °C	96 °C

The maximum error made on the water temperature at engine outlet for these working conditions is only 2 °C. This is quite good results, considering that the mean accuracy of the probes used for the tests is approximately +/- 1 °C. The model agrees very well with the measurement. Consequently, it is possible to assume that this calculation tool can be used to anticipate the validation tests in order to detect possible mistakes before the creation of a prototype.

Moreover at 165 km/h and 200 km/h, the fan is on and the calculated temperatures are correct. That means that for high speed driving, modeling the combined effects of the fan and of the flow due to the car displacement led to good results.

Thus, the model is validated for speed above 100 km/h. However, problems still remain for driving at low speed with high engine load: indeed at these working points the fan is on and the dynamic flow is weak. Modeling the effect of the fan on the flow and consequently calculating the heat exchanged in the radiator was not successful. The water temperatures calculated in these cases are 10 °C too low.

It seems that the model of the fan is too efficient at low vehicle speeds compared to the real fan. A way to improve the simulation results for this kind of working conditions is being studied.

4.3 Planned extensions

Currently, the simulation of transient phenomena is not reliable in a quantitative analysis way because the engine metal masses is not taken into account.

However, the qualitative results are consistent. The following curve, Fig. 8, shows the simulated evolution of the coolant temperature at the engine outlet as a function of time.

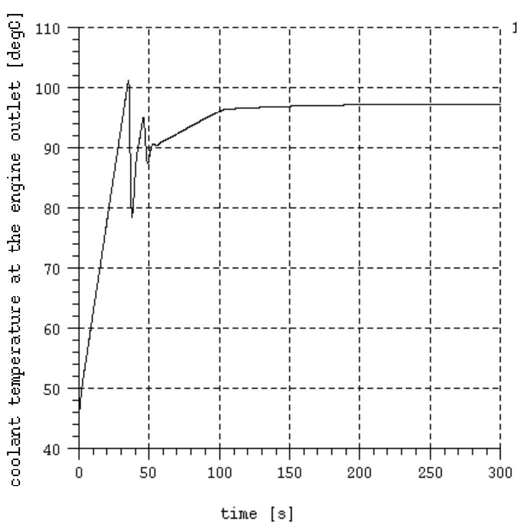


Figure 8 - Coolant temperature at the engine outlet.

Time t = 40 [s] shows the thermostat opening phenomena and the resulting temperature oscillation due to a cold coolant flow coming from the radiator part of the circuit.

Furthermore, when time t = 100 [s], we notice the change in slope which is due to the fan operating start. Indeed when the fan switches on, the power evacuated by the radiator is increased. The temperature increase is then slower and therefore the slope of the curve is weaker.

This analysis leads to the conclusion that, in a qualitative way, the transient phenomena is correctly reproduced.

Thus, what is needed now is to take into account the engine mass "*thermal inertia*". As shown in (Yu, 1998) a higher level of detail can be reached by dividing the engine mass in nodal components to model more accurately its thermal behavior. This method has been applied by using the thermal category components. These components enable us to predict the transient engine mass temperature, taking into account the heat transfer between combustion gas and cylinder, combustion gas and piston, the heat transfer through the cylinder and piston and the convective heat transfer between the engine mass and coolant.

5. CONCLUSION

The libraries presented in this paper use the AMESim solver, which allows to model the transient and steady state behavior of engineering systems.

The solver used in AMESim is based on variable-step methods in which, discontinuities in the circuit, such as the thermostat hysteresis, are automatically taken into account. Thus, complex thermofluid systems have been calculated with low simulation times, usually between 2 and 4 minutes.

The multiport approach provides an easier comprehension of the system designed, and the AMESim graphic interface, as shown in Fig. 7, allows easy changes to the system sketch and parameters. These characteristics are extremely important when a complete engine system is being analyzed.

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