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Review of underhood aerothermal management: Towards vehicle simplified models

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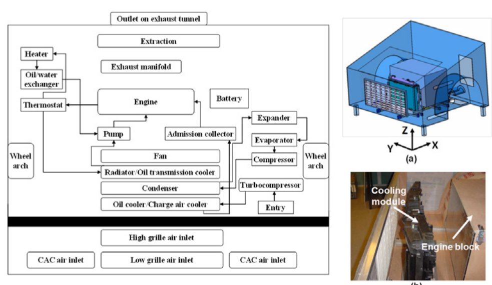
HIGHLIGHTS

- This paper focuses on the review of underhood aerothermal management.
- The different components in the underhood are classified with respect to aerothermal orders of impact.
- Two geometrical models (simple and advanced) are developed based on the impact orders.
- A simplified vehicle body structure model is designed and implemented based on the developed simple underhood models.

GRAPHICAL ABSTRACT



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ABSTRACT

This paper concentrates on the assessment of automobiles aerothermal management; namely, the consequence of the architectural arrangements of electrical and mechanical components on the aerothermal behavior in the underhood compartment. Consequently, it is natural to review the architectural arrangements of underhood components implemented by automotive companies such as Renault, Peugeot, Audi, BMW, Mercedes, Volvo, General Motors, Jaguar, Land Rover, Porsche, Nissan, Chrysler, Ford, Hyundai, Kia, and Toyota. Moreover, this study will evaluate the qualitative impact of each individual component on the aerothermal environment of the underhood. Furthermore, the study is determined to examine explicitly the components architectural arrangements' of underhood compartments and present simplified models of the compartments so that they can serve as aerothermal baseline models in the early stages of design. Hence, the different components in the underhood compartment are classified with respect to aerothermal orders of impact. Followed, two geometrical models, simple and advanced, are developed based on the impact orders. These models serve for future aerothermal studies (analytical, numerical and experimental). Finally, a simplified vehicle model is designed and implemented as underhood compartment model. The structurally simplified model is already utilized in experiments and can be used afterward in both numerical and experimental analyses.

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1. Introduction

In recent years, automotive manufactures have attempted persistently to deliver high engine performance and controlled climate systems. However, both attempts have encountered design hardships due to geometrical and space restrictions in the underhood compartments. Obviously, the need to answer the market requests of delivering sophisticated cooling systems requires additional electrical and mechanical components to be installed in already congested underhood compartments. In addition, noise reduction has augmented the use of insulation around the underhood compartment which compounded the problems of having limited space in underhood compartments.

The aforementioned facts formed components' congestions in vehicles underhood compartments, and therefore creating complex airflows and difficult air paths to take place within the compartments [1–3]. In Ref. [1], a numerical study of the underhood flow shows recirculation regions formed in relation with the complex underhood. In Ref. [2] and for simplification reason, the entire flow domain in underhood was broken into various airflow passages. Due to the underhood geometry complexity, a model radiator was included in the numerical simulations presented in Ref. [3] to take account of the pressure drop in the under-hood compartment since numerical representation of the radiator geometry is very difficult. The airflow entering from the front grille is influenced by many components obstructing its path. Even small geometry details affect the flow direction and can easily impact the underhood cooling situation [4–6]. As illustration in Ref. [5], the front end cooling module configuration (fan, fan shroud, heat exchangers) was shown to have a considerable impact on the underhood overall cooling airflow and fan torque. On the other hand in Ref. [6], it was shown that down-hill and up-hill slight inclinations (of 0.5–1.5 cm differences between rear and front elevations of the vehicle) increase temperatures of components, air zones and engine parameters in the underhood up to 20% at some locations. The complex underhood airflow along with the external airflow and the underhood thermal requirements, have raised a concern about the complex aerothermal phenomena which had to be studied [7,8]. These aerothermal phenomena involving airflow, heat transfer, and fluid dynamics are coupled. Consequently, the main objective is to review the behavioral aerothermal management systems taking aerothermal phenomena into consideration [9,10]. The aerothermal phenomena continue to present demanding aerothermal management challenges in the design of the air intakes and the layout of the front-end cooling module in combination with the complex geometry of the underhood [11,12].

The underhood aerothermal study covers both thermal [13–15] and aerodynamic [16–18] domains. Moreover, there are many components in the underhood compartment which are involved in the thermal and aerodynamic environments. These engagements are not simple to avoid because they engender different compromises to adopt between the two domains (thermal and aerodynamic). Additionally, these compromises are becoming more difficult to attain especially that the underhood compartment design is often subjected to different constraints as stated below:

- Permanent augmentation of the engine specific power, hence heat to be dissipated in the underhood compartment;
- Augmentation of the demanded reliability;
- Augmentation of the compactness needs of the underhood compartment related to increasingly demanding engines;
- Demand for reduction of the components cost limiting the requirement of thermally more resistive materials.

To satisfy the aforementioned requirements along with the underhood compartment thermal management and vehicle styling constraints, it is essential to perform experimental work on cars' prototypes [19–22]. Simultaneously, manufacturers seek increasingly fixed design and production milestone giving priority to competitive costs. Therefore, there is an increasing need for a fast, dependable, and reliable aerothermal management design and analysis procedures to be employed at an early stage of the vehicle development program. Numerical simulations appear to be an appropriate means to satisfy these requirements and have the potential to reduce the amount of prototypes [23]. These numerical simulations concern mainly the investigation of the thermal soak phase of a vehicle where the flow is buoyancy driven [24,25], the underhood compartment thermal management of passenger cars and trucks [26], the exhaust system during cold start emissions [27], diesel engine with a dual-loop Exhaust Gas Recirculation EGR system [28], the airflow over a radiator and convex and concave tubes [29,30] and vehicle suspension system [31]. However, the simulation means need to be validated by experimentally using prototypes [32,33] and real vehicles [34,35]. In addition, the numerical simulations procedure requires boundary conditions to be extracted from genuine experimental results [36–38].

MIRA [39,40] renounced the need for the experimental boundary conditions in the numerical simulations. They focused their studies on the interaction among different simulation tools. Results from each simulation package are fed into the other packages as boundary conditions.

Other manufacturers such as RENAULT and AUDI [41,42] have developed analytical models providing some useful recommendations concerning the design and the underhood compartment architecture, for example, they utilized vertical air outlets rather than horizontal outlets. However, due to complex airflows in the underhood compartment, the modeling techniques are difficult to implement.

In 1999, Volvo [43] followed a direction similar to MIRA when entering the sport utility vehicle (SUV) based on the S80 platform. Considering time constraints, no prototype was made available and only the numerical simulations have been applied in the project.

Porsche AG [44] in 2010 developed an integrated numerical and experimental approach to determine the cooling air mass flow in different vehicle development stages.

The aerothermal phenomena which take place in the underhood compartment are essentially related to the confinement and the large number of components aerothermally involved. Therefore, to understand the aerothermal phenomena occurring in the underhood compartment, it is necessary to study in a combination with the confinement the involvement of components in those phenomena. However, due to the large number of components in the underhood compartment (Table 1), it is proved difficult to include all components in analytical, experimental and numerical studies.

Table 1
Examples of components present in the vehicle underhood.

Category	Components
First order (high aerothermal implication)	Air inlets, air outlets, heat exchangers, fan, engine, turbo-compressor, exhaust manifold.
Second order (middle aerothermal implication)	Battery, compressor, evaporator, expander, admission collector, pump, thermostat, heater, oil/water exchanger.
Third order (low aerothermal implication)	Distribution belt, transmission, shield under the engine, cradle, apron, thermal shield, accessories, cables.

Moreover, from experimental point of view, the existence of such a large number of components in the underhood compartment complicates the use of the advanced aerodynamic and thermal measurement techniques such as, Particle Image Velocimetry PIV, Laser Doppler Velocimetry LDV, Thermocouples, and Fluxmeter, etc. Hence, simplified models of the underhood compartment, based on different components that are aerothermal active or involved, appear to be crucial and beneficial since they would reduce the number of parameters taken into consideration in the analytical and numerical studies. A component is defined as aerothermally involved or active when either its functioning or positioning contributes to the creation of the aerothermal phenomena or when it is influenced thermally or aerodynamically by these phenomena. Furthermore, simplified models facilitate the access of the advanced measurement techniques while performing experimental studies.

To construct these simplified models, all the components that are aerothermally involved or active are defined in subsequent sections, namely, Air inlet and outlet, heat exchangers, fan, engine, exhaust manifold, turbocompressor, thermostat, pump, air admission elements, air conditioning elements, cooling circuit elements, and engine batteries, etc. For each component, solutions adopted by different manufacturers are reviewed and associated critical analyzes are proposed. The influence of each component in the aerothermal phenomena is then deduced. According to these influences, the different components are classified with respect to three orders of impact (as described in Section 2). Each impact order as well as its corresponding components is detailed in Sections 3–5. Section 6 concentrates on the development two underhood compartment simplified models. These models are based on the impact orders as well as the design and implementation of the vehicle simplified body. Finally, Section 7 summarizes the conclusions of the work.

2. Criteria of impacts classification

Underhood aerothermal phenomena are summarized as follows:

- Thermal management (especially the cooling module packaging, convective heat transfer, radiative heat transfer);
- Complex internal airflows in the underhood compartment;
- Aerodynamic interaction between the internal and external airflows and its impact on the aerodynamic drag.

In order to obtain a good explanation of the aerothermal phenomena, it is deemed necessary to define the associated involved or active components. However, the underhood compartment has become more complex by introducing a large number of essential components that are needed to enhance customer requirements and to survive in a competitive automotive market. However, it is difficult, in the aerothermal studies, to take into account all the components in the compact underhood compartment. Subsequently, it is essential to devise simplified, but representative, models of the underhood compartment covering the main aerothermal phenomena related to the different components.

To proceed, all the essential components existing in the underhood compartment of actual vehicles will be listed later in Sections 3–5 and only those of interest from aerothermal point of view will be retained. This selection can be accomplished by fixing some criteria for the components' selection. Due to the packaging existing among different components in the underhood compartment (Fig. 1), most of the elements interacts aerothermally with each other (at least by thermal transfers). Therefore, the selection criteria must be built based on a classification of each component's

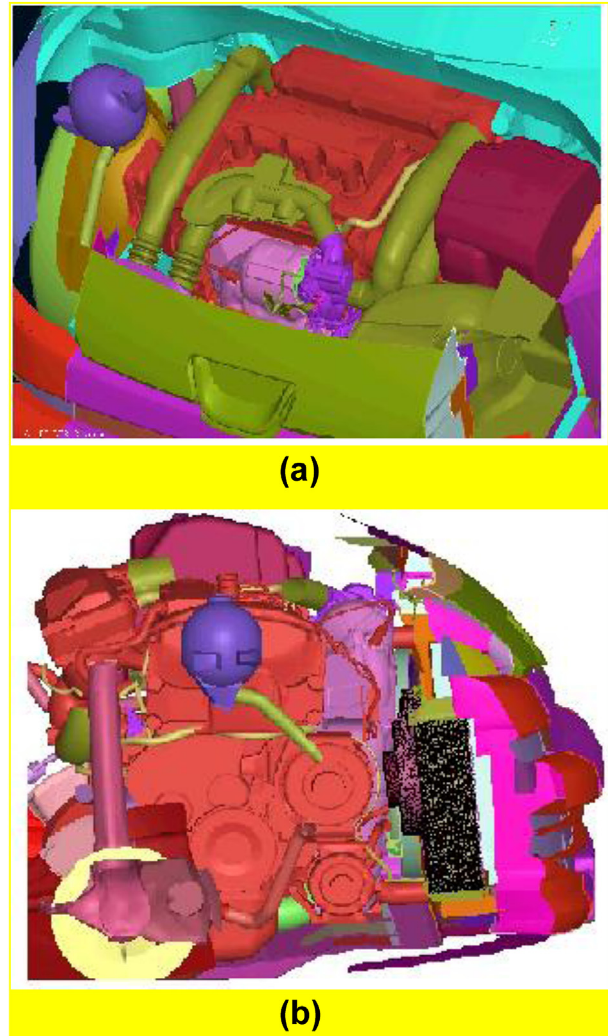

Fig. 1. (a) Top view and (b) side view of the underhood of a Peugeot.

impact. Components are thus classified according to three orders of impacts. For this classification, different terms are defined as follows:

Component of first order impact: a component whose functioning or positioning influences directly the aerothermal phenomena and its outcome cannot be replaced or canceled for the sake of simplification. For example, the engine is one of the most important heat sources and confining components in the underhood compartment; it impacts directly the aerothermal environment and it neither can be replaced nor canceled, therefore, it is considered as a component of first order impact.

Component of second order impact: a component of a second order impact is a component whose functioning or positioning influences indirectly the aerothermal phenomena and its contribution may be accounted for indirectly; that is using outside source for prototypes or simplified model for numerical simulations, but not canceled for a simplification purposes. For example, the car battery is essential to operate the engine starter; hence, it is a component which contributes to the functioning of another component which is of first order impact. Consequently, this component impacts indirectly the aerothermal phenomena and its effect can be replaced, as indicated above, but not canceled for a simplification aim. Therefore, the battery is a component of a second order impact.

Component of third order impact: a component of a third order impact is a component whose functioning or positioning influences indirectly the aerothermal phenomena and its consequence may be replaced and even neglected for modeling simplification. For example, the air filter cleans exterior air which will be distributed among different cylinders. Therefore, the air filter is a component that contributes indirectly to the engine's functioning, that is, it influences indirectly the aerothermal environment and hence, it can be replaced or canceled for a simplification mean. In that case, the air filter is a component of a third order impact.

Simple Model: a simplified model of the underhood compartment composed only of the components of a first order impact.

Advanced Model: a simplified model of the underhood composed of the components of first and second order impacts.

3. Components of first order impact

In this section, components of the first order impact and their implications in the aerothermal phenomena are presented. They are mainly the air inlets and outlets (Section 3.1), cooling module (Section 3.2) and heat sources (Section 3.3).

3.1. Air inlets and outlets

Air inlets and outlets play a significant role in cooling the underhood compartment. The underhood compartment is subjected to thermal loads released from running engines. From the thermal standpoint, the efficiency which is defined as the ratio of the net power restored at the crank shaft and the primary power provided to the thermal cycle is about 0.33 [45–47]. Therefore, the tiny value of the consumed efficiency explains the necessity to evacuate the excess of energy which forms about 70 percent and can be classified as lost energy. In the underhood compartment, the heat evacuation is generally provided by the airflow circulation arriving from the exterior. This airflow permits to cool sufficiently the neighboring region of the heat production and to extract the exchangers' heat. This airflow is provided by the air intakes at the front end which are placed in a positive pressure zone, and facilitates, for a given vehicle speed, the flow path of the air in the underhood compartment. The air circulation cools, by convection, all the components thermally active or involved. This chilling caused by air circulation ensures a satisfactory components' lifespan. Therefore, the air needs to be well oriented among the inlet and outlet openings of the underhood compartment. The design of air intakes depends heavily on the dimension, position, and profile of the heat exchangers and air outlets.

While designing air inlets, compromises between the aerodynamic and thermal requirements must take place so that components are soundly functioning. For example, an efficient air inlet favors the underhood thermal environment (improves the convective heat transfers) but at the same time it increases the aerodynamic drag due to cooling systems. Furthermore, it is worthy to be cognizant that Automobiles whose engines placed in front have two types of intakes (Fig. 2), namely: high grille air intake (HGAI), and low grille air intake (LGAI). The design of air intakes is based on the aforementioned types and can also be classified as:

- Two air intakes design: Air intake of high grille type (Superior); and Air intake of low grille type (Inferior)
- One air intake design: Air intake of high grille or low grille type

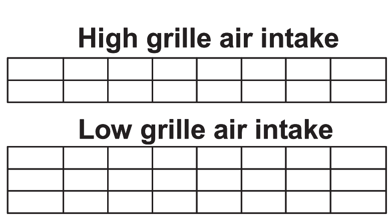

Fig. 2. Air intake grilles.

From the aerodynamic point of view, the air inlet has a heavy impact on the aerodynamic tensor of the vehicle, particularly on the cooling air drag in frontal or lateral wind conditions. Indeed, the cooling air drag is essentially a function of the underhood geometry, the design and layout of the air inlet and outlet, pressures at the inlet and outlet, and the direction and speed of the penetrating air [41,42,48].

In traditional automobiles design, the air inlets are placed on the front end and the air outlets on the exhaust tunnel. In this case, the contribution of the cooling air drag on the global aerodynamic drag is about 7–10% [49–51]. Moreover in 2010, Nissan Ashok Leyland Technologies in collaboration with Defiance Technologies [52] showed that acoustic advantages on automotive vehicles can be reached by air intake system optimization. The reduction of this contribution constitutes a challenge of high interest to the majority of the car manufacturers [53]. They search for reduced cooling air drag while respecting the underhood compartmental thermal requirements [54]. Chrysler LLC in 2009 [55] presented a parametric shape optimization study through numerical simulations which permits to avoid time consuming process such as experimental work on scaled models in order to get drag and cooling flow information. This numerical parametric study permits to provide a complementary path to get answers in a timely manner in the design cycles of new vehicles. General Motors company in 2011 [56] presented some of the challenges and successful outcomes in developing the aerodynamic characteristics of the Chevrolet Volt, an electric vehicle with an extended-range capability. The end result is a unique electric vehicle that, at start of production, has the lowest coefficient of drag of any Chevrolet sedan. Always in the same context of aerodynamic drag reduction, General Motors Company in collaboration with ANSYS Inc. in 2011 [57] developed an adjoint method for aerodynamic shape improvement in comparison with surface pressure gradient method. The adjoint method allows the computation of sensitivity information for a large number of shape parameters simultaneously.

Similar to air inlets, there are several types of air outlets. The mostly adopted by manufacturers are the outlets on the exhaust tunnel, extraction from the underneath of the engine, and front wheel arches. However, from the aerodynamic point of view, the air outlets should be placed in a negative pressure zones in order to facilitate the path air in the underhood compartment. In addition, air outlets should be designed such that the air exits from the underhood to be nearly horizontal. In fact, the air leaving the underhood interacts with the horizontal external flow. Consequently, neither does the air exit the underhood compartment horizontally, in the case of a vertical outlet (outlet on the exhaust tunnel), nor vertically in the case of a horizontal outlet (extraction under the engine). The aerodynamic interactions between the exiting airflow and the flow under the vehicle are not well studied; however, these interactions are heavily coupled with the aerodynamic drag. Therefore, it appears crucial to understand the nature of the airflow near the air outlets. Understanding the airflow nature is necessary to evaluate the drag forces emerged from cooling system and from underneath the vehicle [58].

From the thermal point of view, the air outlets are greatly dependent on the presence of heat exchangers and air inlets (dimension, position, profile) [59], which should allow good air circulations in the underhood compartment avoiding under-cooled zones [60]. In some cases, and due to the thermal constraints to some components, one has to extract air from the engine beneath. However, this solution may at the same time, have an impact on the aerodynamic situation. Hence, it is necessary to respect some aerothermal compromises in the process of an air outlet design.

Aside from the designed air inlets and outlets, there are several significant air leakages in the underhood compartment [61] and they are indentified as follows:

- At the junctions between the hood and front face, side face and windscreen which are considered as air inlets because of the existence of positive pressure zones.
- At the junctions between the hood and headlights which are considered as air outlets because they are situated in negative pressure zones.

While performing aerodynamic and aerothermal numerical simulations, the contribution of the aforementioned leakages are not considered assuming that their cooling contribution is not significant. On the contrary, studies have shown that they have effects on the aerodynamic drag and temperatures of the underhood compartment. Nevertheless, studies have shown that the effect of the leakages zones on the aerodynamic drag is negligible. Therefore, ignoring these zones in the aerodynamic simulations would not influence the results. However, from thermal point of view, it was shown [61,62] that the leakages zones impact the temperatures of underhood compartment during the operational mode of the car, namely, the phases of constant speed driving and in thermal soak which takes place after vehicle stopping. Therefore, leakages should be taken into consideration in aerothermal numerical simulations.

The UNI-CAR Concept [41] developed by Potthoff is based on dividing the air extraction of the underhood compartment into three parts Fig. 3, as follows:

- Two lateral branches permitting the extraction of 35–40% of the overall airflow rate.
- A center duct enclosing the exhaust system which permits the extraction of the remaining 60–65% of the total airflow rate.

Usually the cooling air drag is about 7%–10% of the overall air drag experienced by automobiles; however, the UNI-CAR approach reduces the aforementioned percentage to about 1.7%. However, as air exits through the underbody rear end outlet, it undoubtedly interacts with the exhaust gas of the vehicle. This technology is named "*base bleeding*". In addition, this air extraction system permits the air to exit the vehicle tangentially to the exterior airflow, hence conserving the maximum possible momentum in the direction of the vehicle. Therefore, the reduced exhaust gas assist in minimizing the aerodynamic cooling air drag. Even though the UNI-CAR concept appears extremely important from an aerodynamic point of view, it is surprising that it is not adopted by a large number of automobiles manufacturers when accounting for air drag cooling systems where they keep using 7–10% instead of the sound 1.7% of the global air drag.

The UNI-CAR approach should be validated experimentally so that its consequences on the underhood thermal environment are measured. The aerothermal validation should take place since the UNI-CAR approach; namely, duct solutions for air extraction, reduces the aerodynamic drag due to cooling systems. Simultaneously, these solutions for air extraction decrease the air speed traversing the cooling module (due to increased pressure losses) hence the heat exchanger performances decrease. Conversely, their negative effect on the underhood cooling, they are not far from some aerothermal compromises. In general, solutions based on air guided by ducts are rarely studied.

On the other hand, analytical models [41,42] are developed in order to estimate the cooling air drag. Analysis are based on a momentum balance through a control volume to demonstrate that the cooling air drag is a function of the geometry of the air inlets and outlets, underhood geometry, inlet and outlet pressures and velocities. The momentum balance was established through a control volume which starts at the infinity upstream of the car [42] or at the inlet opening surface [41] and ends at the outlet opening surface as shown in Fig. 4.

Results from analytical models were compared with their experimental wind tunnel counterparts where tests were performed on the "Ahmed Body" (simplified, ground vehicle geometry of a bluff body type) in the wind-tunnel [42]. Comparisons were completed for two exit types; one under the compartment and the other on the bottom of the rear face where only one exit was opened during the measurements. Comparing analytical results to their experimental counterparts, a model which takes into consideration the aerodynamic interactions, in which air does not enter or exit perpendicular to the surface, offers more agreeable results than a model that does not consider the aerodynamic interactions. Therefore, the similarity between the analytical and experimental results; when aerodynamic interactions are considered proves the strong relation between the aerodynamic drag due to the cooling system and the interactions between the internal and external airflows at the inlet and outlet openings.

However; most of the comparisons demonstrated that analytical models deliver cooling drag values that are largely different from their experimental counterparts; therefore, the aforementioned disagreement reveals the necessity to improve the estimation of the inlet and outlet velocity angles. Simultaneously and from the car aerodynamic point of view, it confirms that a good understanding of the interactions improves the cooling air drag quantification.

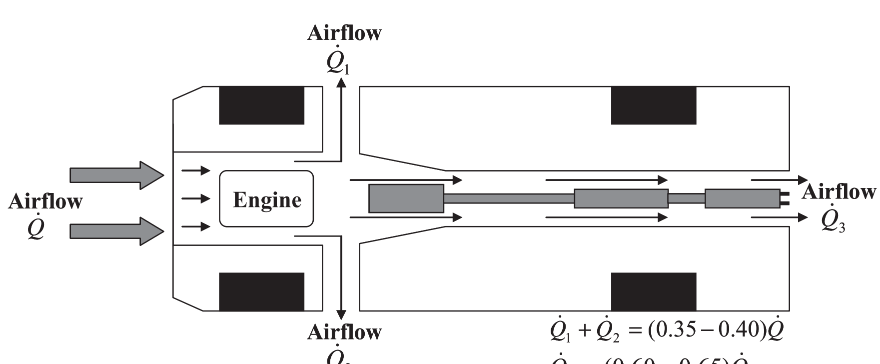

Fig. 3. Extraction airflow paths for the UNI-CAR concept [42].

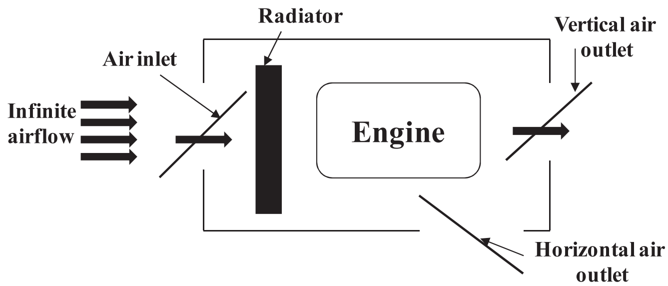

Fig. 4. Control volume for the momentum balance in analytical models for the cooling drag.

Experimental results illustrated that cooling air drag is smaller when the exit is placed on the bottom of the rear face rather than under the underhood compartment. Hence the aerodynamic drag due to the cooling system depends greatly on the exit type. Particularly, a vertical exit is more convenient than a horizontal exit for the drag.

The above confirms that aerodynamic interactions is an important factor to consider while assessing the overall management of vehicle aerodynamic which still needs advanced research efforts.

At the numerical level, Ford Motor Company in 2009 [63] presented a rapid meshing for CFD simulations of vehicle aerodynamics. The new technique presented can reduce the engineering time required for the CAD-to-Mesh process to just a few hours.

3.2. Cooling module

The cooling module is an assembly consisting of different components such as heat exchangers, radiator, condenser, engine oil cooler, transmission oil coolers, charge air cooler and the fan [64–66].

Due to the airflow arriving from the exterior to the underhood, the cooling module extracts a good portion of heat dissipated in the underhood. The radiator extracts about 25–30% of the heat produced by engine combustion; whereas the engine oil cooler extracts about 7–9% of the heat produced by engine combustion; in addition, the transmission oil cooler extracts the heat of the transmission oil; while the condenser extracts the heat absorbed by the refrigerant during the air conditioning cycle; furthermore, the charge air cooler decreases the temperature of the admission compressed air before it enters different cylinders. Lastly, to allow the heat exchanger to dissipate heat in difficult operating conditions, it has been situated backward in the engine compartment [67–69]. Therefore, from an aerodynamic point of view, the airflow in the underhood is largely conditioned by the cooling module.

In fact, in the case of blowing fan, the airflow is oriented by the grills (fins and tubes) constituting the module heat exchangers. This quasi-forced airflow orientation, in combination with the complex packaging of components in the underhood compartment can provoke re-circulations and dead zones. Being poorly cooled, these zones put the pertinent components in a thermally critical situation. However, in the case of suction fan, the airflow is no more conditioned by the cooling module, airflows (in swirl) must be generated and permit to attain poorly cooled zones.

From the aerodynamic point of view, the cooling module has an impact on the drag due to cooling system since it decreases the air pressure. On the contrary, from the thermal point of view, the cooling module is in its optimal functioning conditions when its heat exchangers dissipate heat in a regular manner. The heat extraction is heavily related to the airflow; hence, the performance of the cooling module depends strongly on the architecture of the inlet and outlet air openings; namely, position, size, and profile. Lastly, from the hydraulic point of view, the uniformity of both the airflow and flow of the heat exchangers fluid have a great influence on the performance of the cooling module.

Performance of a single-phase heat exchanger is generally described by its overall heat transfer coefficient defined by the following expression [36,70]:

$$\dot{Q} = UA\mathrm{ETD} \tag{1}$$

where $\dot{Q}$ is the heat transfer rate of the exchanger, A is the heat exchange surface and ETD is the extreme temperature difference and can be expressed as:

$$\mathrm{ETD} = \frac{T_{\mathrm{in,fluid}} + T_{\mathrm{out,fluid}}}{2} - T_{\mathrm{in,air}} \tag{2}$$

where $T_{\mathrm{in,fluid}}$ and $T_{\mathrm{out,fluid}}$ are the exchanger fluid temperature at the inlet and outlet of the exchanger, respectively, and $T_{\mathrm{in,air}}$ is the air temperature upstream of the exchanger.

Many studies [36,70,71] have shown that the exchanger overall heat transfer coefficient U is independent of the temperature difference (ETD). It is a function of the heat exchanger fluid flow rate $\dot{m}_{\mathrm{fluid}}$ and the cooling airflow rate$\dot{m}_{\mathrm{air}}$:

$$U = f\left(\dot{m}_{\mathrm{fluid}}; \dot{m}_{\mathrm{air}}\right) \tag{3}$$

Contrary to the majority of the underhood components, heat exchangers need to be simplified while modeling rather than representing them accurately. This is due to the difficulties encountered in representing precisely their complex small-scale geometries related to the heat exchanger fins and the airflow passing through them. Generally, the modeling of a heat exchanger is divided into two parts; aerodynamic and thermal modeling. In both modeling, one always needs data from experiments and experimental boundary conditions for numerical calculations.

In the thermal modeling of heat exchangers, the representation of the complex geometry of fins needs a very large number of cells which render the calculation very intensive. For this reason, the radiator matrix is usually defined as a porous core. In addition to the radiator matrix, the two fluids; namely, cooling and air need to be represented in the model in order to describe the heat transfer between the cooling and air fluids. The meshes for the two fluids should be superposed and be identical in each cell of the radiator matrix. Then, the link between the two fluids is obtained by applying the heat balance; conservation of energy, at each cell of the radiator matrix as portrayed in Fig. 5.

Assuming that the definition of the heat exchanger coefficient U given by equation (1) is valid at each cell of the heat exchanger matrix, one can then write:

$$\dot{Q}_{\mathrm{cell}} = U_{\mathrm{spec}} A_{\mathrm{cell}} \left(T_{\mathrm{mean,fluid}} - T_{\mathrm{air}}\right)_{\mathrm{cell}} \tag{4}$$

where $\dot{Q}_{\mathrm{cell}}$ is the heat transfer rate exchanged between the cooling and air fluids at the exchanger cell, A_{cell} is the cell area, $T_{\mathrm{mean,fluid,cell}}$ is the fluid average temperature through the cell and $T_{\mathrm{air,cell}}$ is the air temperature upstream of the cell. Meanwhile, by considering the energy balance between the entrance and the exit of a cell then $\dot{Q}_{\mathrm{cell}}$ can be written as:

$$\dot{Q}_{\mathrm{cell}} = \dot{m}_{\mathrm{cell,fluid}} C_{\mathrm{p,cell}} \left(T_{\mathrm{in,cell}} - T_{\mathrm{out,cell}}\right) \tag{5}$$

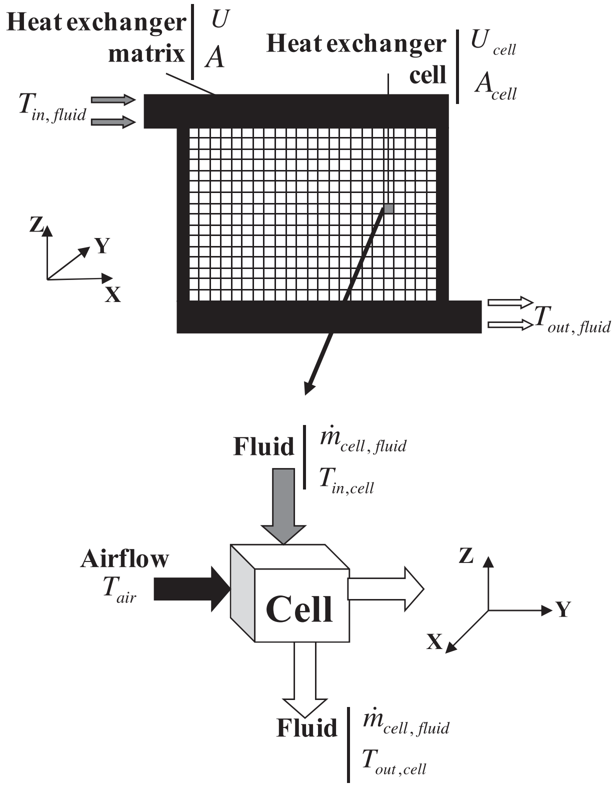

Fig. 5. Schematic diagram of a heat exchanger thermal modeling principle.

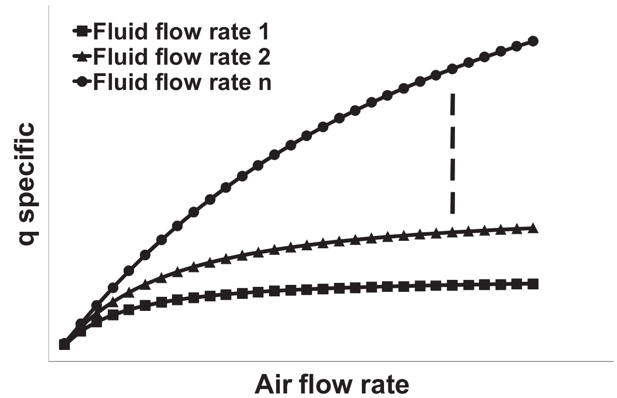

Fig. 6. Experimental characteristic curves of heat exchanger thermal performance.

where $\dot{m}_{\mathrm{cell,fluid}}$ is the fluid flow rate passing though the cell, $C_{\mathrm{p,cell}}$ is the fluid specific heat at the average temperature through the cell and $T_{\mathrm{in,cell}}$ and $T_{\mathrm{out,cell}}$ are the fluid temperatures respectively at the inlet and outlet of the cell. Hence, the total heat transferred between the two fluids can be rewritten in the two following forms:

$$\dot{Q} = \sum_{i=1}^{N} \dot{Q}_{\mathrm{cell}} = \sum_{i=1}^{N} U_{\mathrm{cell}} A_{\mathrm{cell}} \left(T_{\mathrm{mean,fluid}} - T_{\mathrm{air}}\right)_{\mathrm{cell}} = UA\left(T_{\mathrm{mean,fluid}} - T_{\mathrm{air}}\right) \quad (6)$$

$$\dot{Q} = \sum_{i=1}^{N} \dot{Q}_{\mathrm{cell}} = \sum_{i=1}^{N} \dot{m}_{\mathrm{cell,fluid}} C_{\mathrm{p,cell}} \left(T_{\mathrm{in,cell}} - T_{\mathrm{out,cell}}\right) = \dot{m} C_{\mathrm{p}} \left(T_{\mathrm{in,fluid}} - T_{\mathrm{out,fluid}}\right) \quad (7)$$

Finally, equations (6) and (7) are considered to be the basic equations for the heat exchanger thermal modeling, with U and U_{cell} always extracted from experimental curves $U = f(\dot{m}_{\mathrm{fluid}}; \dot{m}_{\mathrm{air}})$ as shown in Fig. 6.

A numerical study that investigated the influence of airflow uniformity and the radiator fluid flow on thermal characteristics had been carried out by Mercedes [72]. The thermal characteristics include the radiator performance, the temperature distribution in the radiator along with the temperature distribution of air on a downstream surface of the radiator. The investigation has shown that the velocity distribution of the airflow influences greatly the thermal characteristics/more than the coolant flow.

From the numerical point of view, a uniform air or coolant velocity distribution flow can be achieved. Nevertheless, this uniformity is not achievable practically due to the combination between the geometry of the air intakes and the position of the radiator. On the contrary, a defined uniform airflow through certain criteria of uniformity does not exist in the reality. Therefore, actual experimental work should seek geometries providing the maximum possible uniform airflow. Moreover, studies on uniformities are rare and still need more research, particularly; more work should be performed to capture a truthful relation between the airflow distribution and the air inlet geometry.

Using the basic equations (6) and (7) for modeling the heat transfer through heat exchangers, an analytical approach was developed in Ref. [70] to determine the thermal performance of cross-flow air-cooled heat exchangers as a function of the flow statistics of the upstream cooling air. The analytical model was validated against two-dimensional computational code results and showed satisfactory agreement. The mean relative error in the heat exchanger thermal performance determined by the numerical computations and analytical approach was about 0.5%. Based on the analytical model developed, it has been shown that an increase in the non-uniformity, reported as standard deviation, of the upstream velocity distribution increases the heat exchanger water outlet temperature and thus decreases its thermal performance. The decrease in thermal performance of a heat exchanger when subjected to non-uniform velocity distribution was also proved analytically by Ng. et al. [73].

In the aerodynamic modeling, the pressure difference of air through a heat exchanger is usually computed using the following expression:

$$\frac{dP}{dx} = C_1 \mu V + \frac{1}{2} \rho C_2 V^2 \quad (8)$$

where μ and ρ are the dynamic viscosity and density of the air, respectively, V is the air velocity and C_1 and C_2 are constant coefficients obtained from the experimental curves of air pressure difference of the heat exchanger as a function of the air velocity($\Delta P = h(V)$) (Fig. 7). These curves should be obtained at adiabatic conditions; if they are obtained at operation conditions, they should be transformed to curves at adiabatic conditions.

In fact, since the energy equation is solved, the density is calculated at the air temperature by the incompressible ideal gas law [36] or through the equation of state [37]; the molecular viscosity is obtained from the Sutherland's law [36]. As the air travels through the heat exchanger its temperature rises and, hence, decreases the density and increases in the local velocity. Once these consequences are taken into consideration in equation (8), the coefficients C_1 and C_2 should be defined, at adiabatic conditions, so that avoiding the contribution of temperature on thermo physical properties twice.

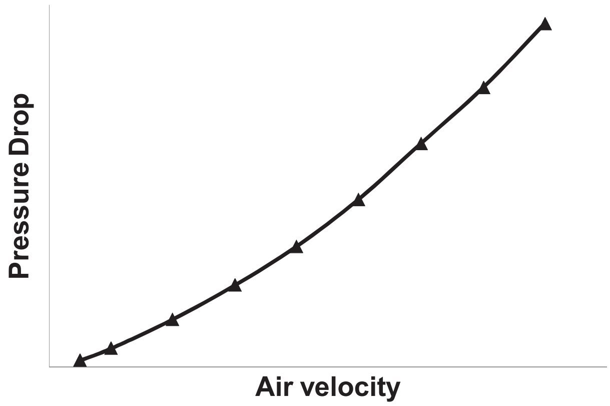

Fig. 7. Air pressure drop through a heat exchanger as a function of the air velocity.

The underhood airflow induced by the fan is usually modeled by two procedures as described below:

1 *The Body Force Fan model or the Lumped Fan Model* [34,36,38]: this modeling considers the fan blades in a cylindrical zone and adds a local forcing term to the momentum equation based on the experimental curves of the fan pressure rise over the blades as a function of the velocity. These curves are modeled by a polynomial function FPR = $P(V)$ (Fig. 8) before being introduced in the model. In addition, given the existing relation between the fan characteristics and air density in the fan blade region, which in turn depends on the air temperature at the estimated working condition in the car, the curve FPR = $P(V)$ should be correlated to its estimated air temperature for the specific load case.
2 *The Multiple Reference Frame (MRF) Model* [33,36,37]: this modeling scheme considers the geometry of the fan blades as an axisymmetric zone which rotates around a fixed axis (axis of the fan) with a specified rotational speed. Then, two terms of Coriolis and centrifugal forces are added to the momentum equations based on the operating angular velocity in order to take the rotation of the defined zone (Multiple Reference Frame) into consideration.

Savory et al. [74] performed experimental tests and computational fluid dynamics (CFD) simulations to investigate the effects of the fan support hub geometry on the component heat transfer and cooling airflow through a simplified model of an electric motor for an automotive cooling-fan system. They found that the presence of radial ribs on the fan hub has a significant effect on drawing cooling air through the motor, particularly at lower airflow rates, regardless of the rotational speed.

The widely used solution for the heat exchangers packaging [75–77] in the cooling module is the "*surface*" solution, where the heat exchangers are superposed in the airflow direction (Fig. 9a). In this case, surface solution, where heat exchangers are placed sequentially, the different heat exchangers interfere with each other. The performance, represented by the total heat rejected $\dot{Q}$, of an isolated heat exchanger varies when it is integrated into a full cooling module, that is, when placing the heat exchangers sequentially. It is clear, as shown in Fig. 9, that a heat exchanger in row 2 is cooled by air at a temperature T_2 higher than T_1. Then, being directly proportional function of the extreme temperature difference (which decrease when $T_{in,air} = T$ increase), the total heat rejected $\dot{Q}$ is smaller in the case of surface solution (Fig. 9a) than its isolated counterpart (Fig. 9b). In this context, a numerical investigation, using spatial optimization technique, was carried out to place optimally the heat-exchanger in the vehicle's cooling module was presented in Ref. [78]. It was found that heat exchangers that are placed in-series "*Reference*" configuration in the cooling module, that is, different heat exchangers are superposed in the airflow direction, have inferior performance when compared to their in-*parallel* configuration, different heat exchanger surfaces are in a row with respect to the airflow direction, counterparts. In fact, in-parallel configuration increases the thermal power of the heat exchangers by 4.4% and decreases the pressure losses by 0.9%. Moreover, the pressure drop of the surface heat exchangers packaging is higher than the sum of the pressure drops of the different heat exchangers measured individually as shown in Fig. 10.

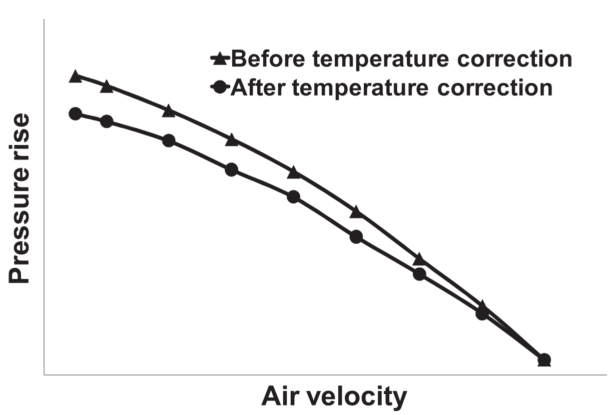

Fig. 8. Characteristic curves of polynomial fan pressure rise as a function of the air velocity.

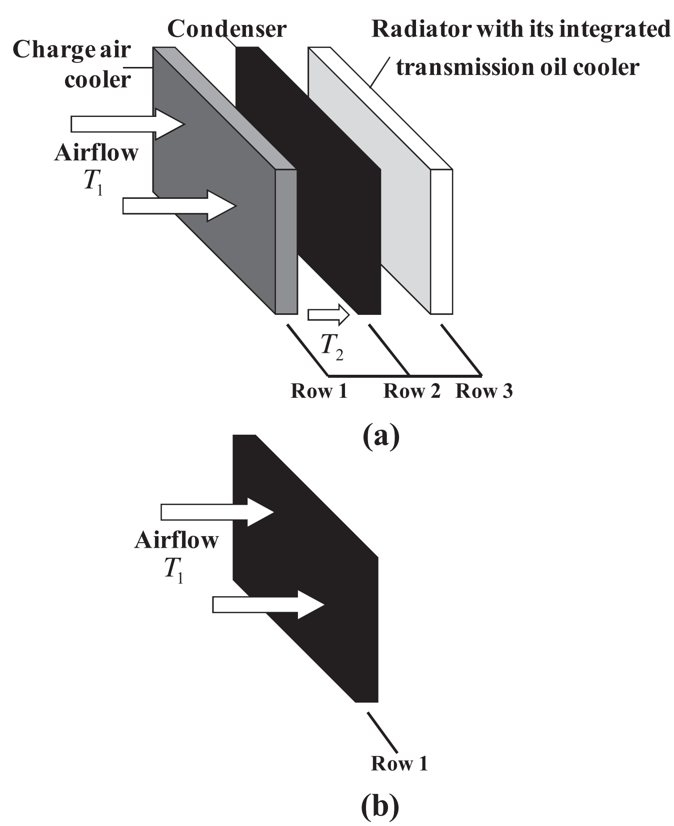

Fig. 9. (a) "Surface" solution of heat exchangers' packaging and (b) heat exchanger isolated from the surface combination.

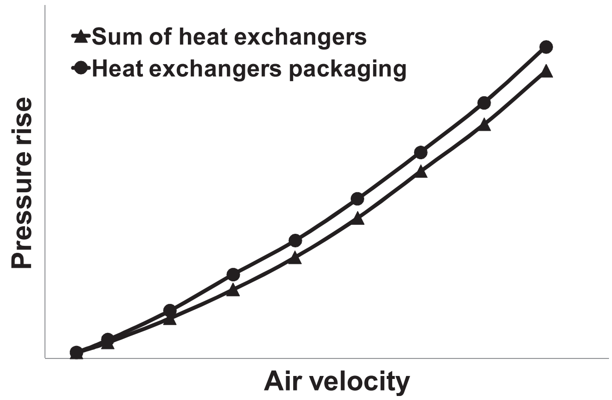

Fig. 10. Interference effect.

As an example, the cooling module of BMW series 3 vehicles has been developed in collaborations between BMW and Modine [79]. The goal was to produce a compact module to control the oil transmission temperature on departure conditions and during normal driving conditions. With special arrangements, the cooling module thickness in the *X*-direction (vehicle length) can be reduced, which reduces the pressure losses across the module and then the fan power required to overcome the losses. Indeed, the water radiator of the BMW series 3 has a double function by cooling the engine and automatic gear box, in addition to the presence of a compact and flat oil-water heat exchanger, which made it possible to eliminate row 3 (Fig. 9a).

A new monitoring concept was proposed in Refs. [80–82] for optimizing the underhood aerothermal management. The concept is based on adjusting the heat exchanger's performance to the engine energy requirements and increasing the heat exchanger's thermal performance in critical situations such as vehicle slowdown and thermal soak phases (vehicle stops after a significant heating load). The principle of the control approach is to separate, in *Y*-direction (vehicle width), the vehicle condenser (heat exchanger at row 1 in Fig. 9a) from the radiator (heat exchanger at row2 in Fig. 9a) in order to increase the heat extraction by the radiator in critical situations where the engine overheats.

El-Sharkawy et al. from Chrysler Group LLC [83] presented a new design methodology for automotive heat exchangers which permits to maintain an optimal coolant temperature and to limit the vehicle underhood air temperature within a tolerable limit. On the other hand, Song et al. from Hyundai-Kia America Technical Center Inc [84] discussed the integrated low temperature coolant LTC loop configuration and others. The LTC system prototype design allowed vehicle performance data to prove that a simplified front end cooling module package allows better airflow cooling efficiency.

3.3. Heat sources

The most important heat sources in the underhood compartment are the engine, turbocompressor, and exhaust manifold. Fig. 11 illustrates the distribution of the heat dissipated by the engine in the underhood compartment which is primarily produced by the fuel combustion [45,85].

From the aerodynamic point of view, the engine position (longitudinal or transversal) may have different impacts on the cooling air drag. Indeed, the engine position changes the underhood overall architecture and influences the air circulation in the compartment. From the thermal point of view, the engine releases heat by convective heat transfer and by radiative transfer to its environment [81]. The engine involvement in the convective heat transfer with the radiator coolant is greatly related to the dimensioning of the radiator which in turn depends on the dimensioning of the air inlets and outlets to extract the necessary amount of heat transferred between the engine and coolant.

An experimental study of the aerothermal phenomena in the vehicle underhood compartment as investigated by measuring temperature, convective heat flux, and radiative heat flux was presented in Ref. [12]. Measurements were performed on a real PSA Peugeot Citroen vehicle for three thermal functioning conditions while the engine is in operation and the front wheels are positioned on the test facility with power-absorption-controlled rollers. In the thermal analysis, particular attentions were given to measuring absorbed convective heat fluxes at component surfaces. It is shown that, in some components, the outside air enters the engine compartment (cooling certain components) can in fact heats other components. This problem arises from the underhood architecture, specifically the positioning of some components downstream of

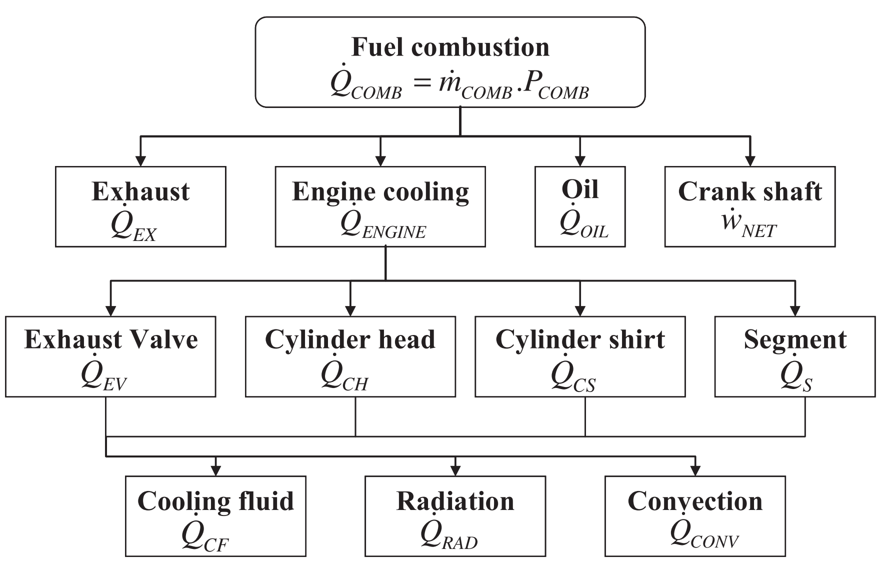

Fig. 11. Distribution of the heat dissipated by the engine in the underhood compartment [45].

warmer components in the same airflow. Optimized thermal management suggests placing these components further upstream or isolating them from the hot stream by deflectors. Based on the aforementioned, a new underhood control procedure was presented in Refs. [81,82] in which static and mobile deflectors are implemented in the car underhood space in order to protect certain components from hot air circulations. These deflectors can also direct the warm air passing over low-temperature components towards higher-temperature components.

4. Components of second order impact

In this section, components of the second order impact and their involvements in the aerothermal phenomena are presented. They are mainly: coolant pumps (Section 4.1), thermostats (Section 4.2), cooling circuits (Section 4.3), air conditioning elements (Section 4.4), air admission elements (Section 4.5) and engine batteries (Section 4.6).

4.1. Coolant pumps

The coolant pump is an essential component of the cooling system. It permits circulations of a volume flow rate $\dot{C}$ of a coolant in a cooling circuit by compensating the pressure losses of this latter (which the coolant loses during its circulation) with a network. Generally, this work is furnished to the pump electrically. In other terms, at its passage through the pump, the coolant acquires a pressure rise which it loses in the other parts of the cooling circuit. Therefore, a pump is generally characterized by the curve of the pressure rise ΔP_p or the necessary hydraulic power $\dot{P}$ as a function of the volume flow rate $\dot{C}$ which are related to the engine requirements.

From the volume flow rate point of view, there are two types of pumps [86]: pumps with constant volume flow rate of the coolant; in this case, the pump functioning corresponds to a fixed point $(\dot{C}_f; \dot{P}_f)$ in the characteristic diagram of Fig. 12a; and pumps with a variable volume flow rate of the coolant; the functioning of the pump corresponds to a series of points $(\dot{C}_i; \dot{P}_i)$ in the characteristic diagram as a function of the engine requirements (Fig. 12b). For this type of pump (variable speed), an electrical drive is recommended.

There are two types of the pump driving devices; namely: electrical driven pumps; in this case the hydraulic power for the pump to circulate the coolant is furnished electrically; and belt driven pumps; in this case the hydraulic power for the pump, to circulate the coolant, is part of the engine power and then is considered as a power consumption from the total power (0.5–1%). Recently in 2013, BorgWarner Inc. in collaboration with University of Michigan-Dearborn [87] developed a dual mode coolant pump with its corresponding active thermal management.

In belt driven pumps, the volume flow rate is related to the engine rpm (Fig. 13). On the contrary, in electrical driven pumps, the volume flow rate is related to the engine cooling requirements. Hence, the electrical driven pump provides reduced power consumptions compared to belt driven pumps which eliminates the need to the electric pump power. The characteristic curve of electrical driven pumps is sometimes given in a non dimensional form based on the two following terms [86,88]:

$$\psi_p = \frac{\Delta P_p}{\frac{\rho_w}{2}u^2} \quad \text{and} \quad \varphi_p = \frac{\dot{C}}{u\frac{\pi d_p^2}{4}} \tag{9}$$

where ψ_p and φ_p are the dimensionless pressure rise and volume flow rate of the pump, respectively, ρ_w is the water density, u is the linear velocity of the pump water and d_p is the pump diameter.

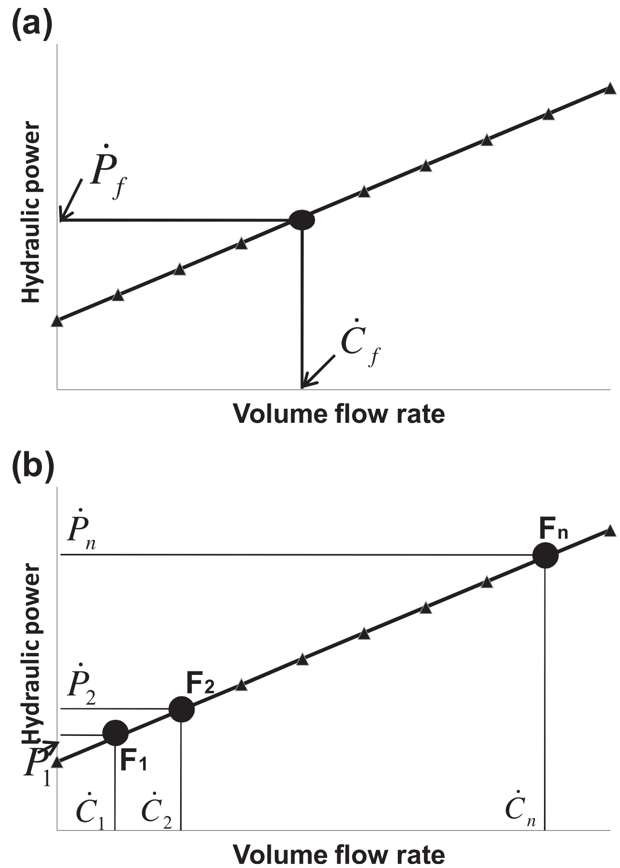

Fig. 12. (a) Functioning point of constant volume flow pump on the characteristic diagram ($\dot{P} - \dot{C}$), (b) functioning points of variable volume flow rate pump on the characteristic diagram ($\dot{P} - \dot{C}$).

Mounting the pump on the engine appears actually to be the most promising solution since [86] the engine should be tested independently from the vehicle and is typically used in several different car platforms. In addition, mounting the pump on the body leads to more complexity and requires a longer bypass circuit and consequently more coolant.

Finally, the coolant pump is an essential element in the functioning of the cooling system since it is contributing to the functioning of its main element, the radiator. The, the coolant pump is considered a component of second order impact.

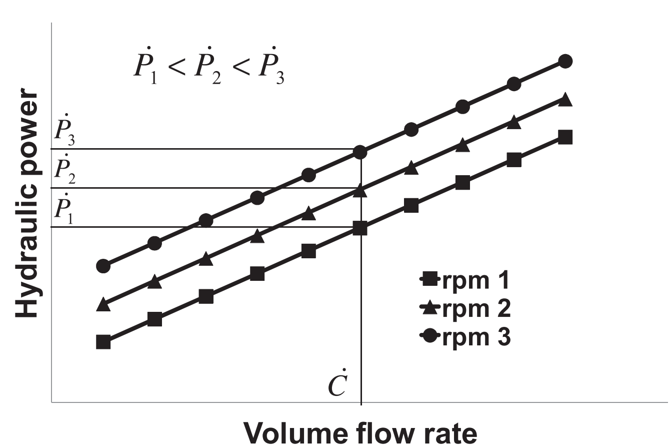

Fig. 13. Characteristic curves $\dot{P} = f(\dot{C}; n)$ of a belt driven pump.

4.2. Thermostat

The thermostat is an essential element of the cooling system. It provides the control of the coolant flow temperature through the cooling system [88,89]. The thermostat is generally characterized by a temperature above which it opens and permits the coolant to pass through the engine block. The time-variation of the coolant temperature from the engine warm-up phase up to the thermostat opening has a typical shape as shown in Fig. 14 [89,90]. The temperature above which the valve opens is called the "*start to open STO*" temperature of the thermostat. The opening of the thermostat at its STO temperature is generally achieved by a thermal sensitive "*wax*" power element. Indeed, when this element reaches the STO temperature, it expands and allows the valve to lift off its seats.

From the thermostat position point of view, there are two types of thermostats [89,90]: "*outlet thermostat*", when the thermostat is placed between the engine outlet and radiator inlet and controls the coolant outlet temperature (Fig. 15a); and "*inlet thermostat*", when the thermostat is placed between the radiator outlet and coolant pump inlet and controls the engine block and head temperatures in order to protect the engine from thermal shock (Fig. 15b).

From the point of view of opening command nature, there are two types of thermostats [89,90]: *mechanical thermostats* – functioning is through the mechanical expansion of its integrated "*wax*" power element described above; and *electronic thermostats* – functioning is through an electronic command of the wax element. At extreme operating conditions of the engine, in other words when a fast reaction of the cooling system to lower the coolant temperature is rapidly needed the electronic thermostat heats up the wax element by electrical power and forces the thermostat to open.

The thermostat is a component which contributes to the functioning of the radiator (component of first order impact). It is worth noting that the thermostat is considered to be as a component of a second order impact based on the impact order classifications.

4.3. Cooling circuits

The cooling circuit is a combination of different pipes relating essential elements of the cooling system: radiator, thermostat, coolant pump, and engine inlet and outlet [89,90]. By following this cooling circuit, the coolant loses from its pressure. The pressure losses are the sum of linear and singular pressure losses.

The linear charge losses are losses of energy due to friction between the fluid and a tube of constant section. They are expressed in fluid head (meters) or Pascals and calculated from the following expression [88]:

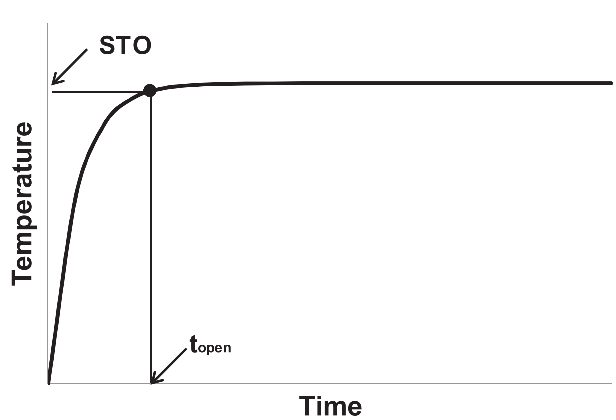

Fig. 14. Typical time-variation of the coolant temperature controlled by the thermostat.

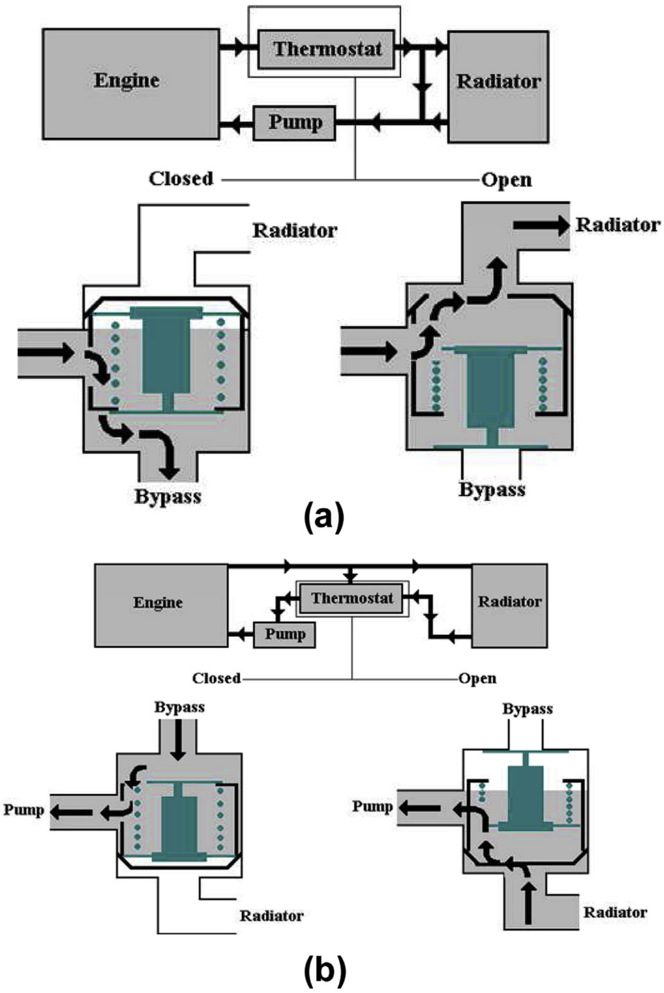

Fig. 15. Functioning (open-closed) of: (a) inlet thermostat, (b) outlet thermostat [89,90].

$$\Delta h_r = f \frac{L_c}{d_c} \frac{V_f^2}{2g} \tag{10}$$

where f is the moody friction coefficient, L_c is the conduit length, d_c is the conduit diameter, V_f is the conduit fluid velocity and g is the gravity acceleration.

The singular charge losses are losses due to change in direction, to the bends, branches in the circuit, section reduction, passages through machines, and entry and exit. They are expressed in fluid head (meters) or Pascals and calculated from the following expression [88]:

$$\Delta h_s = K \frac{V_f^2}{2g} \tag{11}$$

where K is the resistance coefficient of the bend, branch, section reduction, section enlargement, passage through machine, and entry or exit.

Finally, the total charge loss in a hydraulic circuit is expressed as follows:

$$\Delta h = \sum_{i=1}^{M} (\Delta h_r)_i + \sum_{j=1}^{N} (\Delta h_s)_j = \sum_{i=1}^{M} f_i \frac{L_{c_i}}{d_{c_i}} \frac{V_{f_i}^2}{2g} + \sum_{j=1}^{N} K_j \frac{V_{f_j}^2}{2g} \tag{12}$$

where M is the number of conduits of different diameters and N is the number of singularities in the circuit.

Similar to the coolant pump, the coolant circuit is an essential element to the functioning of the cooling system and contributes to the functioning of the radiator which is a component of first order impact. The coolant circuit is also considered as a component of a second order impact.

4.4. Air conditioning elements

Aside from the condenser which is considered as a component of a first order impact, the fundamental elements of the air conditioning system are [91–93]: the evaporator, compressor, expander and air conditioning circuit.

Fig. 16 shows the typical refrigeration cycle of the air conditioning system on the Pressure-Enthalpy P-h diagram and the basic equations for the cycle heat balancing [94]. In Fig. 16, $\dot{Q}$ is the heat rate exchanged in each process, $\dot{m}_{\text{ref}}$ is the refrigerant mass flow rate, h is the enthalpy and η is the efficiency. The different abbreviations are: *evap* for evaporator, *ovh* for overheating, *comp* for compressor, *isent* for isentropic, and *cond* for condenser. These elements contribute to the functioning of a component of the first order impact (condenser) [95,96], and are considered as components of a second order impact.

4.5. Air admission elements

Aside from the turbocompressor and the charge air cooler (components of first order impact), the fundamental elements of the air admission system are: the entry, flowmeter, admission manifold, and air admission circuit.

These elements are influenced by the underhood convective heat transfer with the environing air and restrict the airflow passages. They are elements contribute to the functioning of the air admission system which its essential task is achieved by the charge air cooler and the turbocompressor. In addition, these elements are part of a system contributes to the functioning of the engine. Therefore, the air admission elements are components contribute to the functioning of three first order impact components Based on the impact order classifications, the air admission elements are considered to be second order impact elements.

4.6. Engine batteries

The essential function of the battery is to provide electric power to the engine start. The battery has additional functions; its energy serves to power the alarm and remote control the command of unlocking the doors, provide the electrical source of the dashboard, power the headlights when the engine is not in function, and power the memory of the electronic devices [97,98].

This energy is produced by a chemical reaction between two electrodes dipped in an electrolyte. The dimensioning of the battery is dependent on the starter and the consumers installed on the vehicle [97,98]. Usually, the battery is placed close to the engine at its left. Hence, it is strongly influenced by the thermal radiation of the engine. For this reason, the battery is usually protected by a thermal shield in order to avoid its thermal failure (Fig. 17). In this context, Chrysler Group LLC [99] showed that the electric vehicle battery life can be optimized through battery thermal management. In addition, the battery is a component contributes to the functioning of the engine (first order impact), so it is considered to be a second order impact element.

5. Components of third order impact

Basically, a component of third order impact is defined as a component with its functioning influencing the aerothermal phenomena indirectly and its effect may be replaced and even eliminated in order to simplify the underhood and give more accessibility in the simplified models. Aside from the first and second order classifications for the different components described in Sections 4 And 5, a component of a third order can then be defined or selected in different manners as a component:

- Does not satisfy the conditions to be classified as a component of first or second order impact;
- Influenced by the aerothermal phenomena without contributing to this phenomena and this influence can be studied separately from the underhood compartment management;
- Does not contribute to the aerothermal phenomena but, at the same time, restricts the air passages of the underhood airflows;

Among the components of the third order impact, the most important ones are the distribution belt, transmission, and shield under the engine, cradle, apron, thermal shield, accessories, and cables. It should be recalled that there are a large number of underhood components. Hence, from the classification point of view of the present paper, one should opt to the identification of the fundamental components of first and second order satisfying the predefined conditions. It is not the aim of this paper to list all the remaining components that appear to be as components of third order impact, one can consider that all the components in the underhood not listed above are of third order impact.

6. Underhood models and vehicle simplified body

As shown above, the underhood compartment is composed of a large number of components that intervene in complex aerothermal phenomena in the underhood compartment. In addition,

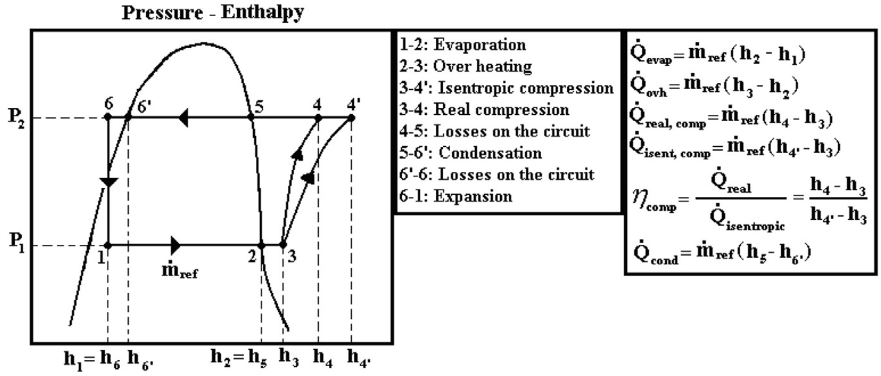

Fig. 16. Typical refrigeration cycle of the air conditioning system on the Pressure-Enthalpy P-h diagram and the basic equations for the cycle heat balancing.

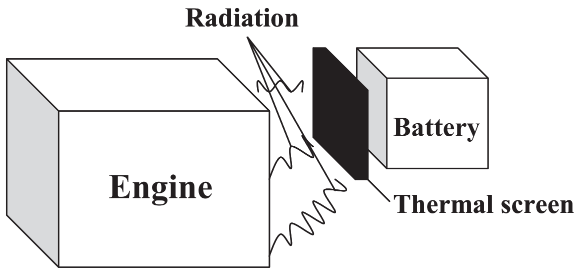

Fig. 17. The battery protected by thermal screen form the engine radiation.

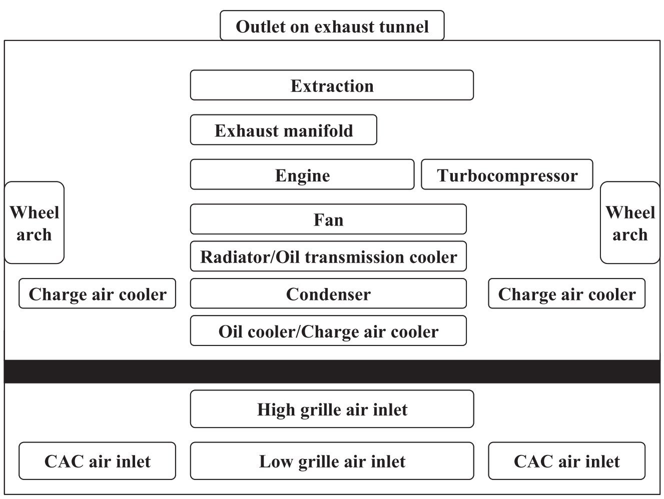

Fig. 18. Underhood simple model.

due to the order of impact classification presented here, one could highlight the most important components which are aerothermally active. Once the degree of the impact of different components is determined, simplified models of the underhood compartment can be built by eliminating components of lesser impact. The first order model, shown in Fig. 18, is a simplified model in which only components of the first order impact are retained. The second model presented in Fig. 19 is more complete, in which not only the components of the first order impact but also those of the second order are also retained.

Due to the high cost and the practical difficulties of full scale rolling and stopping vehicle tests, numerical simulation proves to be a good alternative. Finite element codes provide very satisfying results. Nevertheless, long modeling and computation times implied by these approaches do not allow this technology to be fully efficient in the early design stages. In these cases, a more adequate solution is to use vehicle simplified models which reproduce in each study the main affecting parameters.

These simplified models in automotive applications are mainly used in crash tests, aerodynamic and rarely in aerothermal analyses.

Gillespie and Karamihas [100] presented an analysis of passenger car response to road roughness on a quarter-car model. Yoshimura et al. [101] worked on the design of an active suspension system for one-wheel car models using the sliding mode control. Halgrin et al. [102] developed and implemented an automatic method for the localization and the non-linear response of collapse mechanisms, based on the simultaneous use of analytical results and of a global beam finite element model. The robustness of the model was validated against experiments under frontal crash conditions on a commercial front car part. Mundo et al. [103] proposed a methodology for the concept design of vehicle bodies, based on the use of reduced models of beams, joints and panels. Simplified models of beams, joints and panels of the upper region of a vehicle's body are created and validated through full vehicle analyses.

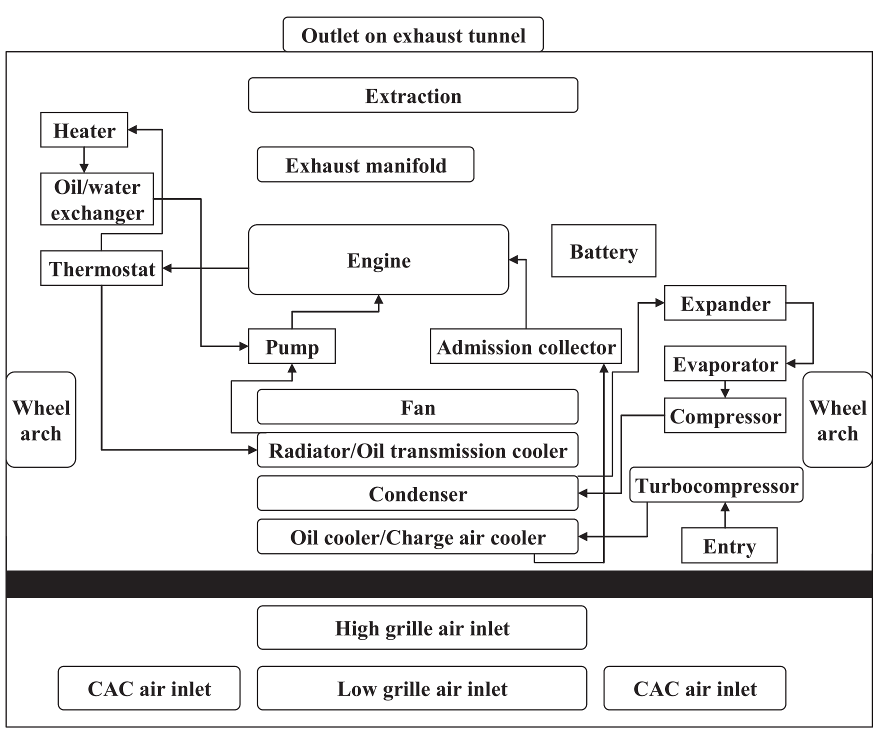

Fig. 19. Underhood advanced model.

In aerodynamic analyses, one of the test cases most often adopted is the generic car model proposed by Ahmed et al. [104]. This is because of its geometrical simplicity and the realistic complex flow field which is reproducibly generated in its wake and makes it a selective test case. Cogotti [105] investigated the effect on the aerodynamic coefficients produced by important geometric changes which affects the flows under the car in proximity of the ground. The model chosen for the work was that defined by the SAE "Open Jet Interference Committee" as a reference model to be used for investigating wind tunnel interference. In particular, it has front and rear wheel-housings of three different sizes, wheels of 2 different sizes/covers, some different wheel tracks, three different front and rear tracks. Guilmineau and Chometon [106] presented an analysis of the instability of passenger vehicles associated with transient crosswind gusts. Experimental and numerical tests were performed on the squareback Willy test model, which is realistic, compared to a van-type vehicle. Slaboch et al. [107] described window buffeting measurements acquired on a full scale vehicle as well as two different simplified scale models. The implication of this result is that simplified laboratory models of a vehicle are sufficient to study the various aspects of window buffeting in full scale vehicles. Huminic and Huminic [108] investigated numerically the flow around the Ahmed body for the rear slanted upper surface of 35^0, fitted with a simple underbody diffuser, without endplates, in order to find the influence of the later one on the aerodynamic characteristics, drag and lift. The study was performed for different geometrical configurations relevant for hatchback passenger cars. Chen et al. [109] (General Motors Company) uses also a simplified full-scale underhood with open enclosure to study numerically the buoyancy driven flow at vehicle stopping conditions. The simplified model includes an open enclosure which has openings to the surrounding environment from the ground and through the top hood gap, an engine block and two exhaust cylinders mounted along the sides of the engine block. Aerodynamic studies are also performed on simplified bodies by Kawakami et al. [110] (Toyota) to model transient aerodynamic loads in sinusoidal motion.

Passing through the different simplified models mentioned above, one can see that they can be classified into the two categories of simple and advanced simplified models of Figs. 18 and 19. However, it can be found that simplified models in aerothermal analyzes are scarce. The next paragraphs describe a new vehicle simplified body designed and implemented mainly for aerothermal analyzes. This model presents the advantages of permitting accessibility to the advanced thermal and aerodynamic measurement techniques and parametric geometrical configurations to be reached.

The vehicle simplified body is designed and implemented based majorly on the simple model of Fig. 18. This model consists of a representative simplified form of a front vehicle block designed basing on a Peugeot 207 with tangential manner (Fig. 20a). The model is about 1.7 m width in Y-direction, 1.3 m length in X-direction, and the height in the Z-direction is variable between 0.6 and 1 m. A real cooling module (radiator, condenser and fan) and a

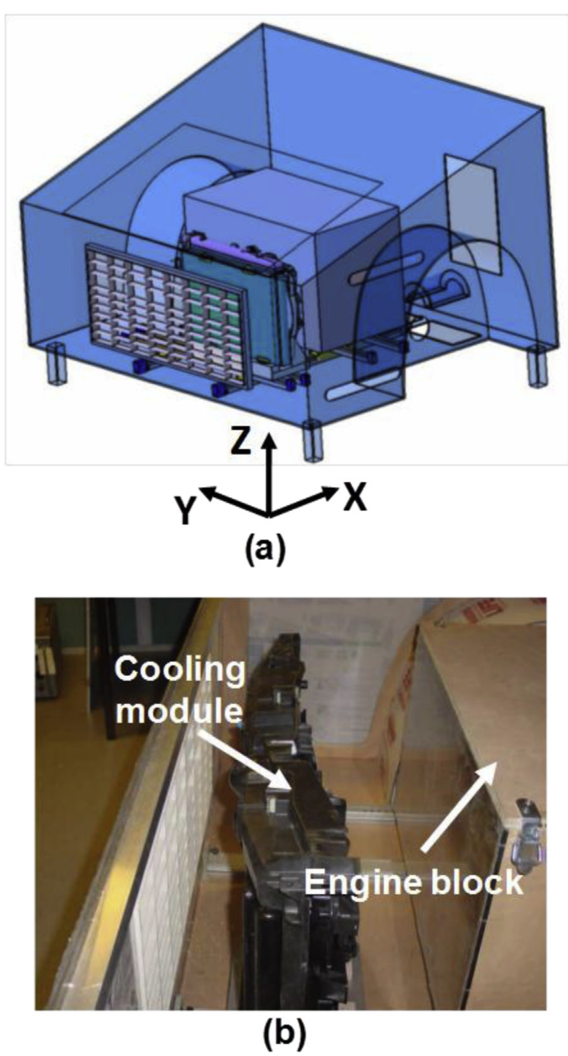

Fig. 20. (a) Schematic view of the simplified model and (b) Cooling module and the simplified engine block.

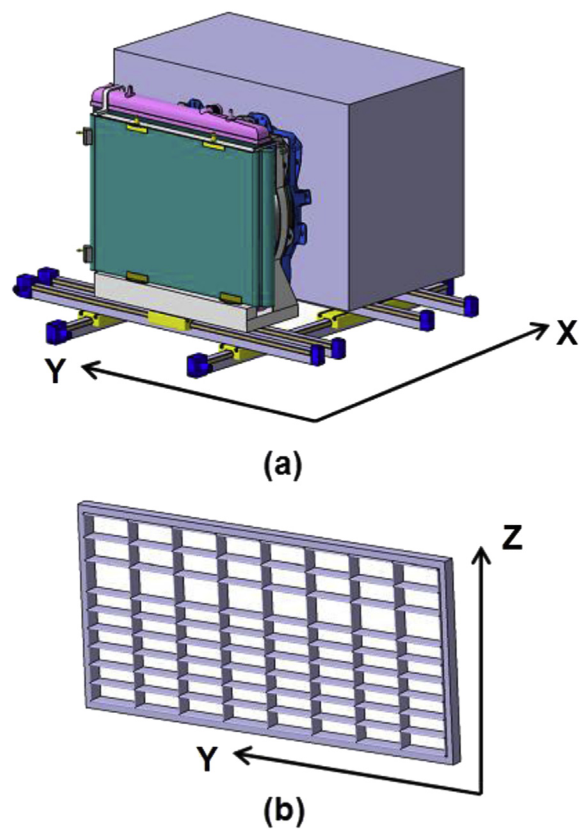

Fig. 21. (a) Rail system for two-dimensional motion in XY plane and (b) modular air inlet opening in YZ plane.

simplified engine block are implemented inside the model (Fig. 20b). Both elements are positioned on a rail system (X, Y) which permits moving these elements in a two-dimensional plane XY (Fig. 21a). The air inlet opening in YZ plane (Fig. 21b) is also movable in such a way to modify the flow entrance configuration by varying the inlet cross section. There are four types of air outlet openings in the simplified model (Fig. 22): Vertical air outlet and outlet in the tunnel, air outlet in the wheel arches and air outlets under the ground. The vertical air outlet is positioned at the rear face of the model. This outlet is not existent on real vehicles since it is located at the same region of the driver but constitutes a typical reference configuration in most aerodynamic studies. The outlet in the tunnel constitutes a real outlet existent in all actual vehicles and represents the region enclosing the exhaust tunnel of the vehicle.

Fig. 22. Air outlet openings in the simplified model.

Wheel arches air outlets are exactly openings in the wheel arches. Air outlets under the ground of the vehicle are extractions under the engine block of the simplified body. Air outlets in the wheel arches and under the ground represent real air outlets not necessary existent in all series car.

7. Conclusions

The present work concerned a review of underhood aerothermal management. The underhood main aerothermal phenomena as well as the involvement of the main components in these phenomena are described. The different components are then classified according to three orders of impact from the more active or involved to the less ones.

Once the degree of the impact of different components was determined, simplified models of the underhood compartment were built by eliminating components of lesser impact. The first order model is a simplified model in which only components of the first order impact are retained. The second model is more complete, in which not only the components of the first order impact but also those of the second order are also retained.

Finally, a new automobile simplified body was mainly designed and built for the purpose of aerothermal analyses. This model provides accessibility to the advanced thermal and aerodynamic measurement techniques. Furthermore, it facilitates the performance of parametric geometrical configurations. Moreover, this car model is based primarily on the aforementioned first order model.

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