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OPTIMIZATION OF THE CONTROL PARAMETERS OF A VEHICLE'S ELECTRIC FAN ACTIVATED BY PWM

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Abstract. *It is increasingly common the presence of new technologies in the current automotive projects, with energy efficiency as the main objective, and, consequently, the reduction of fuel consumption. The majority of these technologies require a specific calibration for each project, to enhance the efficiency of the component while maintaining the performance in acceptable limits. One of the newest technologies used in the cooling system is the PWM (Pulse Width Modulation) controlled fan, aiding in the energy efficiency of the vehicle with less demand from the alternator. One of the problems facing this component is its calibration, as the PWM fan is needed to fulfill its energetic function and reach the thermal balance performance targets as well; keeping the temperature at acceptable levels. At most times, the calibration of this component is done experimentally. In such a way, that many tests need to be run until satisfying values to the calibrated parameters are found. The present work shows a computational method to calculate the optimum calibration parameters for the fan. In this way, it is possible to find the optimum parameters in a relatively faster and less expensive way, avoiding a large number of physical tests.*

Keywords: *Optimization, Electric Fan, Fuel Consumption, Energy Efficiency*

1. INTRODUCTION

Once the radiator in a new cooling system is designed, the definition of the cooling control parameters is the next step, so that it presents the necessary performance to cool the engine, but requires a minimum possible energy. There are several ways to control the component that are used extensively in the industry, among them the use of a table (lookup table) or a PID (Proportional-Integrative-Derivative) controller.

Due to cost, easiness to implement, understanding, and control issues, the direct control, performed by the lookup table is more common. PID control is also used, but on a smaller scale.

It is proposed to present a comparison of the performance of a vehicular electric fan, by using a control proportional to the error, considering a target temperature, and an algorithm that uses the lookup table. The main purpose of this work is to present a comparison of the energy spent by the electric fan, when controlled by these two control forms, modifying parameters that define the rotation of the component.

2. THEORETICAL FORMULATION

Thermal systems modeling in general is present in several previous studies (Shome and Joshi, 2006; Thombare and Dhananjay, 2012; Rosa et al, 2009; Gupta, 2015; Swarthmore, 2015; Yoo et al, 2000), in such a way that the models and the equations presented by some of them will be used as a basis for the modeling of the cooling system of a vehicle. In (Yoo at al, 2000) it is presented a simple model used for the modeling of the cooling system, in which the system is understood as an energy storage device composed of cooling fluid, oil, engine block , such so that the energy level of the system is expressed as the system temperature.

In a simplified manner, the cooling system of a vehicle contains a source of heat from the combustion of the fuel in the engine and various sources of energy loss, illustrated in Fig 1.

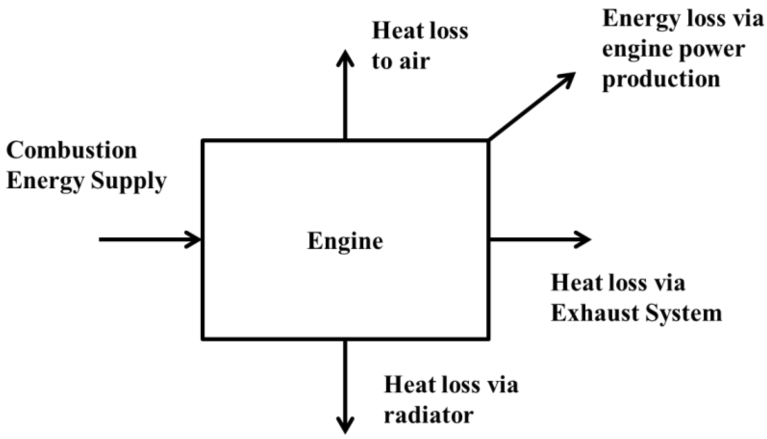


Figure 1: Energy sources associated to na Internal Combustion Engine (Yoo et al, 2000)

The source of energy loss treated in the present study will be the loss of heat to the cooling liquid. After combustion, part of the heat generated inside the cylinders is transferred to the cooling fluid, heating up the cooling fluid. When the fluid leaves the engine, the thermostatic valve distributes the liquid to the components of the cooling and heating system of the vehicle. When the coolant temperature exceeds thermostatic valve opening temperature, part of it is directed to the radiator, and then cooled (Yoo at al, 2000). In most passenger vehicles, the cooling module is composed of the radiator, the heat exchanger of the air conditioning system (condenser) and the electric fan.

The cooling system’s electric fan is the component responsible for forced ventilation in the radiator when the vehicle stops or operates at low speeds, because under these conditions the engine still needs to be cooled (Cardoso, 2011). This component may have discrete control, in which it is driven at certain speeds, depending on the vehicle’s coolant temperature, and the pressure of the air conditioning system; or continuous control, gradually changing the speed of the fan. Most fans are made of plastic and their operation power varies from vehicle to vehicle (Bosch, 2005).

The new automotive designs, aiming at greater energy efficiency are more frequently using variable-speed electric fans, controlled by PWM (Pulse Width Modulation), which may have different forms (Vasca, 2012). In addition to the advantages found in conventional cooling fans, PWM controlled fans have a higher mechanical reliability (Paparrizos, 2003).

The PWM control can be understood by an analogy to a circuit being modulated by a switch connection, which shows a simple circuit of a motor, turned on when the circuit is closed and off when the circuit is open.

The pulse width is defined by the time that the circuit remains closed. The sum of the closed and open circuit time represents the period. The relationship between the time of the pulse and the duration of a complete cycle of circuit opening and closing defines the Duty Cycle of the pulse.

Pulse rate is one of the most important parameters when defining a PWM control, and can be constant or variable (Vasca, 2012). With the use of different Duty Cycles it is possible to control the average power applied to a load. Figure 2 shows a simple illustration of the Duty Cycle definition of a Pulse Width Modulation. The active cycle, measured in percent, is the ratio between the time of 100% of voltage pulse and the time between two pulses.

The present work shows an optimization methodology of the control parameters of an electric fan controlled by PWM, since it presents a better possibility of control and a better charge consumption in relation to the electric fan controlled by discrete voltages.

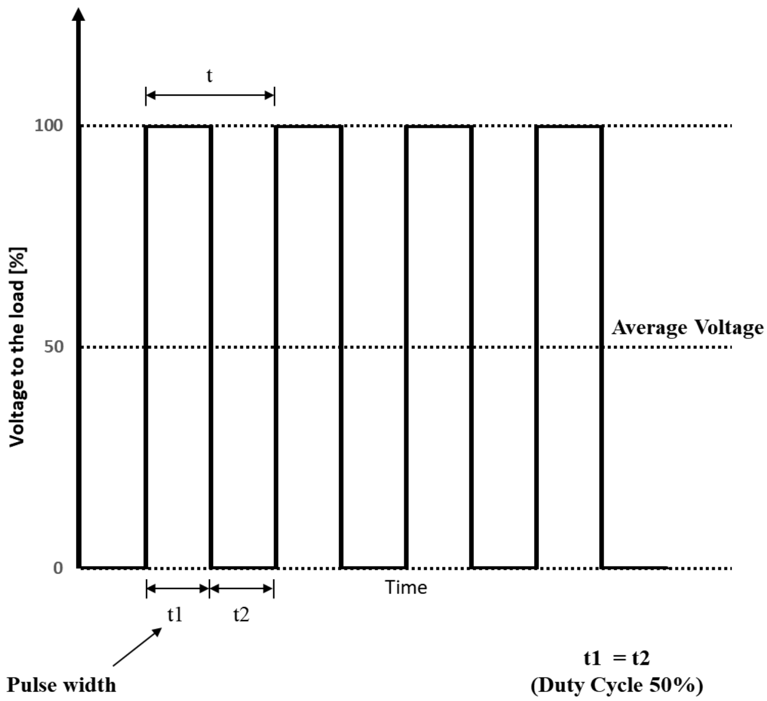


Figure 2: Analogy of a PWM, where the motor is fed when the switch is closed and not fed when the switch is open (Rodrigues, 2015).

3. METHODOLOGY

In this section, it is discussed the thermal models, of the system as a whole and also of the heat exchanger used in the simulations of the proposed cycles. After discussing the model used for the simulation of the cooling system and control of the electric fan, it is presented the control strategy and optimization method used to calculate the lowest energy value with the strategy studied. Finally, the proposed cycles are presented for the thermal simulation of the cooling system of the vehicle in question.

3.1 Input Data

For the simulation of each step with a constant speed for each cycle with variable speed, it is necessary to define the environment initial data in which the cycle will run: the characteristics of the vehicle, the thermal rejection curves of the engine and the cooling module's flow data, dimensions and pressure drop.

To run the simulations, the present model uses a commercial radiator as well as its data collected on a test bench.

3.2 Thermal modeling

The thermal model considers the cooling fluid as an energy storage, and temperature is the parameter that indicates the system's energy level. In the present model, the fluid receives heat of the engine, and consequently increases its internal energy level, and it loses heat in the radiator, reducing its energy. When the energy balance between these two sources of heat reaches steady state situation, it is also noticed the thermal equilibrium. Figure 3 illustrates a schematic of the simulated model, taking into account the heat flux absorbed by the fluid in the engine and the rejected heat flux in the radiator, as well as the main parameters governing the simulation, also expressed in the Eq. 1 to Eq. 4.

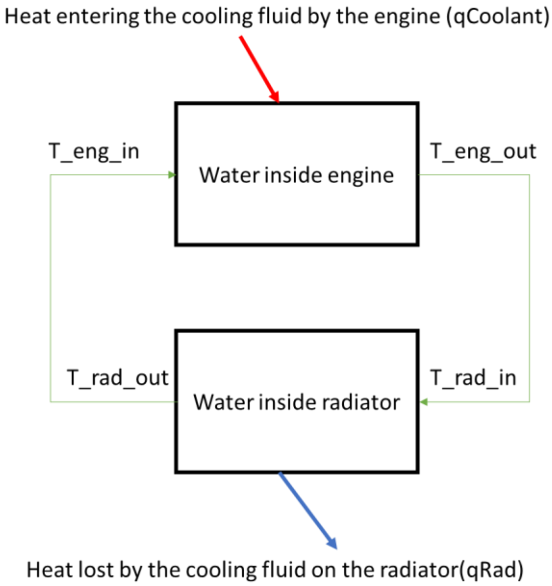


Figure 3: Illustration of the energy balance applied in the studied model (Yoo et al, 2000).

Where, qCoolant is the heat rejected from the engine to the coolant. In the model, this value is found from a thermal map of the internal combustion engine, previously defined. The heat lost by the fluid in the radiator (qRad) is given by Eq. (2), explained in the next topic.

$$qTot=qCoolant-qRad \tag{1}$$

To calculate the transient regime, thermal rejection of the vehicle engine in various situations are estimated through the required torque and engine speed. Using Eq. (1), Eq. (2) and the estimated values for each of the parameters the thermal rejection in the radiator is calculated, it is possible to estimate the evolution of the coolant temperature along the simulation time. The coolant temperature calculation is repeated until the proposed cycle ends.

With the intention of understanding and certifying the consistency of the values found by the model, the results of the thermal modeling were correlated with data collected from experimental tests, performed in a controlled situation in a test cell.

3.3 Heat Exchanger modeling

From the input data of the cycle speed and the initial parameters of the vehicle (such as gear ratio, tire, mass, etc.) it is estimated the coolant mass flowing through the radiator and the parameters of thermal exchange on the "water side".

On the other hand, the "air side" heat exchange parameters, such as airflow, pressure drop, etc., are calculated by the air inlet velocity at the front of the vehicle and the airflow rate caused by the pressure rise due to the electric fan. The studied model applied in the present work act in the calculation of thermal exchange on the "air side".

The heat exchanged in the radiator was estimated through the efficiency table of the heat exchanger, in such a way that from the liquid and air flow estimated on the radiator, the value of the radiator's efficiency is obtained, and then used in Eq. (2).

$$qRad=e\times C_{min}\times(T_{rad\_in\_liq}-T_{rad\_in\_air}) \tag{2}$$

Where, e is the effectiveness of the radiator, estimated from the efficiency table, and C_{min} is the minimum thermal capacity between the liquid and air thermal capacities. These values are estimated through Eq. (3) and Eq. (4).

$$C_{ar}=c_{p\_air}\times\dot{m}_{air} \tag{3}$$

$$C_{liq}=c_{p\_liq}\times\dot{m}_{liq} \tag{4}$$

Where c_{p_air} and c_{p_liq} are the specific heats at constant pressure for air and liquid, respectively, and $\dot{m}_{air}$ and $\dot{m}_{liq}$ are the mass flow rates of the air and liquid, respectively, estimated in the model, taking into account the radiator's dimensional parameters and vehicle's front grille (Silva, 2010; Incropera and de Witt, 2003; Barros and Baêta, 2006).

4. ELECTRIC FAN CONTROL

The electric fan's control logic studied in the present model is the PID. PID is already used in production vehicles and the important controlled parameter is the coolant temperature. As the electric fan control acts indirectly in the cooling system's temperature, the amount of flow available by the component must increase or decrease according to the temperature of the system and the target temperature. The model compares the current temperature with the target temperature of the cooling system and, according to this error, positively or negatively increases the Duty Cycle of the electric fan (percentage of PWM pulses). Similar to the PID, a simpler logic was used in this work: the Duty Cycle increment value is calculated as a linear function of the error:

$$DelDC(Err)=m\times Err+n \tag{5}$$

In addition of being a function of the error, a minimum and maximum duty cycle increment value is defined, so that from a certain error value, both negative and positive, the reduced and increased value of PWM pulse percentage remains the same.

Figure 4 illustrates the control performed. It is possible to notice three main points that influence the system's temperature behavior:

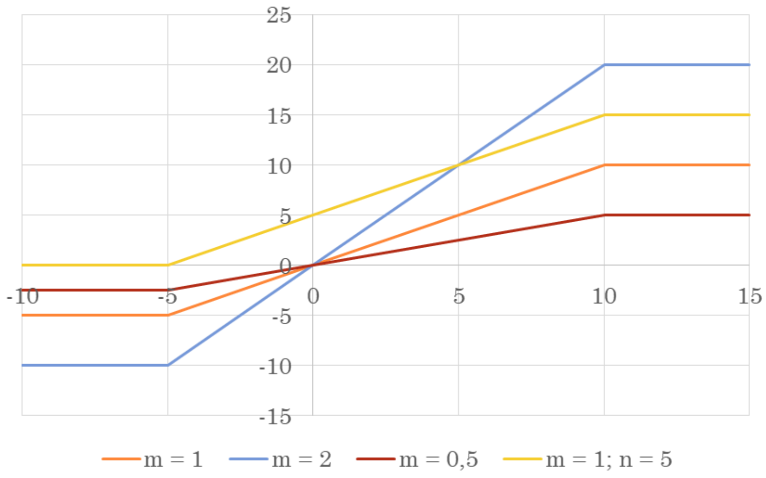


Figure 4: Variations of the Duty Cycle increment curve to the electric fan control

- Angular and linear coefficients modify the curve Error x DeltaDC
- Maximum and minimum increments influence the response of the fan
- Target temperature influences the amount of DC set

For simplicity, the algorithm uses only the angular coefficient – "m" – of the linear curve shown Fig. 4. The value of the linear coefficient remains zero and the minimum and maximum Duty Cycle increment values were set to -5 and 10 respectively. The angular coefficient value m = 1 is used as the initial value for the stabilized speed cycles. For the dynamic cycle, it is used all the values of the coefficient that presents lower energy required by the electric fan in this cycle. Those coefficients are calculated through a simple search, where a new coefficient value is the previous coefficient multiplied by two. Next, a new value is calculated again by multiplying the previous value by two, and so on. If the value of energy calculated with the new angular coefficient is greater than the energy previously estimated, this coefficient is replaced by the average between the previous coefficient and the current coefficient until a lower energy value is found. This procedure repeats until it is no longer possible to find a lower energy value than previously calculated. Figure 5 represents the described procedure.

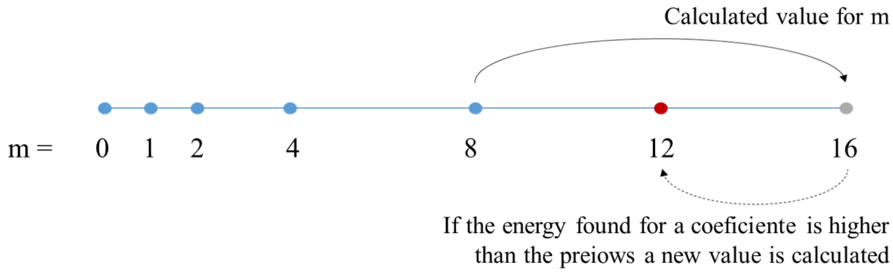


Figure 5: Calculation procedure of the angular coefficient values for the control model. From a certain value the average of the two lasts values is used in order to find the optimum value in the cycle.

Five coefficient values are calculated and the energy level with all five coefficients are shown in the present work. In addition, the target temperature is set as 90°C.

5. COMPUTATIONAL SIMULATION

The first simulation experiment consists in the analysis of stabilized cycles, with several circuits, in order to correlate the thermal model in different situations. After simulating the stabilized cycles, a comparison with real vehicle test and simulated on the same cycles is performed. It is worth noting that in this first stabilized test, each of the phases is simulated following the ambient conditions, vehicle configurations, slope shown in tab. 1.

Table 1: Summary of the simulated cycles for each of the phases performed in the present study

Cycle Type	Speed [km/h]	Time	Slope [%]	Ambient Temperature [°C]	Air Conditioning
Stabilized Cycles	23	NA	9	30	Off
	42		9	30	On
	42		6	30	On
	140		Max	30	On
	0		NA	30	On
Stabilized Speed Cycles	23	30 minutes	9	40	Off
	42	30 minutes	9	40	On
	140	30 minutes	0	40	On
	0	30 minutes	NA	40	On
Unstable Cycle	0 - 23 23 23 - 40 40 40 – 23 23 23 - 140 140 140 - 23 23 23 - 0 0 23 23 - 0	5 seconds 30 seconds 10 seconds 5 seconds 5 seconds 60 seconds 30 seconds 5 minutes 30 seconds 60 seconds 10 seconds 60 seconds 25 seconds 10 seconds	6%	40	On

Next, the model real test correlated parameters is run in a cycle, still with stable speeds, but this time considering the transient regime. The temperature of the liquid starts at ambient temperature and all the components, electric fan, as well as, thermostatic valve are operating normally. In this section, it is presented the temperature evolution for the first angular coefficients simulated by the model.

The vehicle is simulated in a simple unstable cycle, obtaining the temperature and energy required by the vehicle's electric fan. In this simulation, a comparison is made between the results found with each angular coefficients found before. This cycle is simulated with presence of air conditioning. Table 1 shows schematically the performed simulations.

Figure 6 shows the vehicle speed in the unstable cycle. In this cycle, the speed consists of all the simulated velocities in a stabilized cycle, run sequentially.

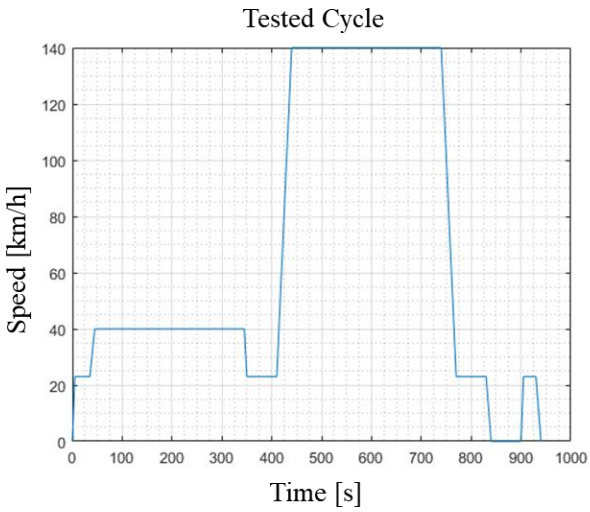


Figure 6: Unstable cycle composed by the stabilized speeds separately simulated before to calibrate the model. Once the model is calibrated, the speeds were used to create the model, simulating a normal driving condition.

6. RESULTS

In this section, the results for the three proposed cycles are presented. First, it is showed the correlation of the model with the stable test results, followed by the simulation with the stable speed cycles showing the transient behavior of the coolant temperature. Both first results are run with the initial angular coefficient, before the optimization. The lasts results show the comparison with all the coefficients for the unstable cycle, presenting also a comparison of the energy consumed by the electric fan with all the parameters calibrated.

6.1 Stabilized cycles following situations previously established in standard (calibration of the model)

Table 2 presents the comparison between computational simulation, and a real vehicle, tested in a controlled test cell situation.

Table 2: Stabilized tests results. Correlation between real results simulated in test cell and computational simulated results, using the model of the present study

Test type	Speed [km/h]	Grade [%]	Real Temperature [°C]	Simulated Temperature [°C]
Stable test results	23	9	66,9	67,55
	42	9	81,6	82,46
	42	6	74,3	76,22
	140	Max	73,5	73,57
	0	-	62,9	62,57

6.2 Stable Speed Cycles

The simulation was performed with the proportional control, using the first value of the angular coefficient (m = 1). With the proportional control, there is the evolution of the temperature until the stabilization in a certain value; however, there is the presence of overshoot until stabilization. Figures 7 to 10 present the results for the simulations of the stable speed cycles, demonstrating the temperature evolution, Electric fan duty cycle and energy from the fan.

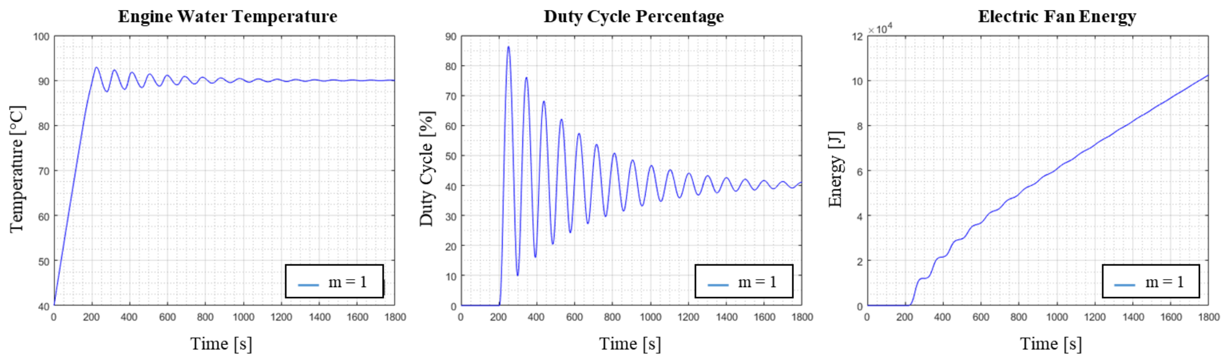


Figure 7: Results found for the computational simulation of the transient model for the speed of 23 km/h.

Figure 7 presented the results for the first speed, 23 kilometers per hour, with evolution of the temperature, duty cycle during the cycle and total energy requested from the fan. It is possible to notice the overshoot present during some steps of the simulation. This overshoot is characteristic of PID control and especially because the control studied, uses only proportional control to the error. One strategy that can be used to enhance this behavior is including a derivative term, to decrease the overshoot present. Similar to the results for 23 kilometers per hour, Fig. 8 shows the results for the next phase, 42 kilometers per hour.

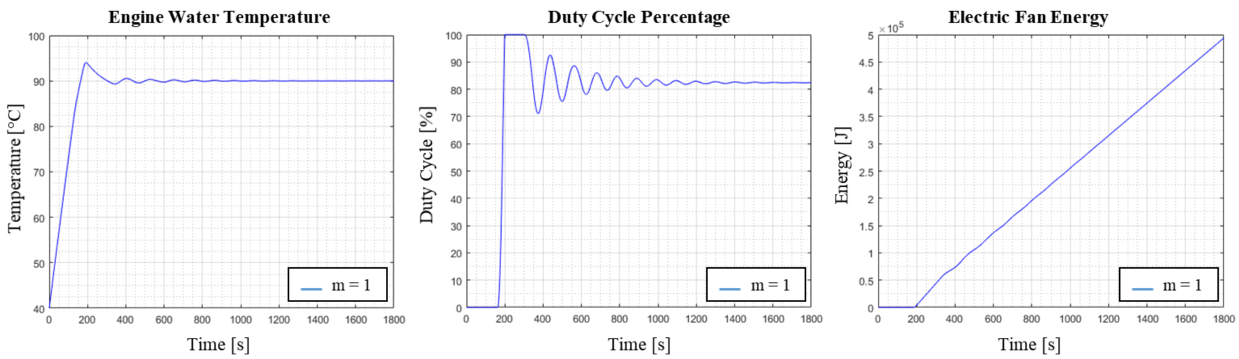


Figure 8: Results found for the computational simulation of the transient model for the speed of 40 km/h.

Similar to the results found with 23km/h, the results of Fig. 8, 40km/h, presents an electric fan Duty Cycle overshoot, but with less expression in engine water temperature. The amount of energy is higher, since it is a more critical phase, demanding more presence of the electric fan to maintain the temperature on the target 90°C.

Next phase simulated, 140 kilometers per hour is shown in Fig. 9. Differently from the two phases presented before, this phase does not present electric fan Duty Cycle or energy demanded. As the speed is too high, and the thermal equilibrium between engine heat rejection and radiator heat rejection happens before the target temperature, the air flowing on the radiator due to the vehicle speed is sufficient to maintain the temperature stable and cool the vehicle.

Finally, the engine idle is shown in Fig. 10, presenting similar results as 23km/h, since the speed is also similar for these two phases. The amount of energy requested by the fan is also similar.

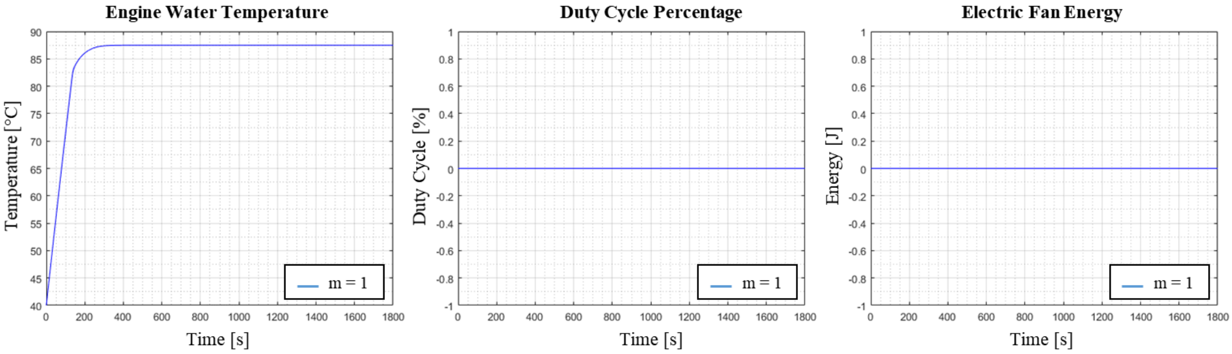


Figure 9: Results found for the computational simulation of the transient model for the speed of 140 km/h.

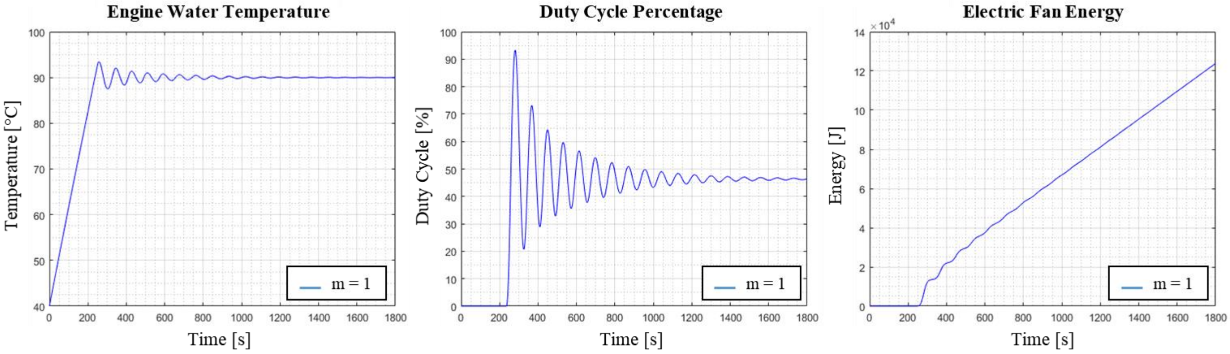
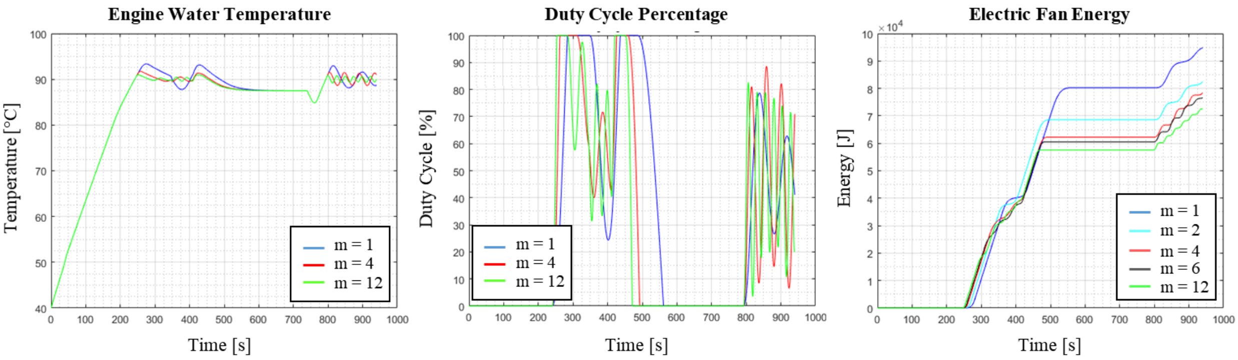


Figure 10: Results found for the computational simulation of the transient model for the idle condition test.

6.3 Unstable Cycle

Finally, the next simulated and analyzed topic is the unstable cycle, with all the five values of angular coefficient. The model simulates all the five parameters to find the value that requires less energy. Figure 11 shows the results found with three of the simulated values for engine coolant temperature, and the energy value for all the five values.



a) b) c)

Figure 11: Comparison of the results found with the simulated model for the unstable cycle. 11a. present the temperature behavior during cycle; 11b. shows the electric fan duty cycle; 11c. presents accumulated energy of the electric fan during unstable cycle.

Figure 11a. and 11b., illustrates the temperature behavior and the electric fan Duty Cycle during the unstable cycle. It can be noticed that the temperature level reached by the shortest coefficient value is higher than the highest coefficient. It happens due to the reaction of the fan to the temperature behavior. This reaction can be noticed on figure 11b, showing the electric fan duty cycle. Figure 11c. presents the energy cumulated during the cycle, tab. 3 shows the final value of energy required by the fan at the end of the cycle for each one of the angular coefficients, the lowest energy value was reached with the angular coefficient, m = 12..

Table 3: Total energy accumulated after the unstable cycle for the five angular coefficient simulated. The last value, as expected, represent the lower value of energy consumed.

Unstable test results	Angular Coefficient	1	2	4	6	12
	Energy [kJ]	94.860	82.447	78.514	76.542	72.601

An interesting point perceived from the simulation results is that as the coefficient value m = 8 did not present less energy compared to m = 4, the value found was m = 6 and continuing the seeking process, the next value found is m = 12. Regarding that as m = 8 did not show a smaller value than m = 6 neither than m = 12, the function cannot be considered as a convex function. This behavior shows that the objective function has different local optimum values, presenting opportunities for using heuristics to find the local optimum values during different cycles.

7. CONCLUSIONS AND FUTURE WORKS

From the present work, it may be observed that the logic made by proportional control presents an opportunity for control and possible optimization of parameters to find the best energy or temperature of the liquid using the model in question. By increasing the angular coefficient values of the curves, the response of the electric fan becomes more sensitive to the temperature, and it is possible to find a point demanding less energy during the cycle.

Future works can also present, more complex control logic such as including derivative logic creating a Proportional-Derivative controller. Logics can also be created based on lookup table, which consists of a table with the Duty Cycle percentage of the fan directly defined. Another control logic also possible to be used is fuzzy logic, based on nebulous logic, capable of controlling the fan by different ranges or states of temperature and heat rejection by engine and radiator.

Finally, another improvement to be made concerns the thermal model. As previously quoted, the model considers the fluid as an energy storage whose temperature indicates the level of energy present in it. Another way of performing the calculation is by modeling the fluid through its flow in the cooling system, i.e., the fluid receives heat from the engine and loses it in the radiator, which has an inlet and outlet temperature on each of these components.

The main advantage of the model is the possibility of using different optimization algorithms directly linked to the behavior of the cooling system of the vehicle, and easily calibrating electric fan control parameters before running physical tests, which can cost time and money if performed indiscriminately.

8. ACKNOWLEDGEMENTS

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