Simulation of Motorization Cooling System of Low-cost Tractor based on AMEsim

Yassine Zahidi[1,2], Mohamed El Moufid[2], Siham Benhadou[1,2], Hicham Medromi[1,2]
[1] *EAS Research team, Laboratory of Research in Engineering, ENSEM Casablanca, Morocco*
[2] *Foundation of research development and innovation in sciences and engineering (FRDISI)*

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Abstract: The low-cost tractor is equipped with a hybrid powertrain, which consists of an electric motor and an internal combustion engine linked together by a planetary gearbox. Both of them generate a considerable amount of heat during the driving cycles. In order to be able to manage this produced temperature, a robust thermal management system must be implemented for the safety of the tractor. with the help of the simulation established by the Amesim software, the proposed cooling system should able to stabilise the heat of the thermal engine at a maximum of 95 degc and that of the electric motor at 92 degc, working with the most energy-intensive tool.

1 INTRODUCTION

All-electric vehicles were presented as a promising solution, thanks to their independence from oil. But their low autonomy does not allow them to replace conventional vehicles definitively (Ahmed Neffati, 2020). In order to combine the advantages of both architectures, the so-called hybrid architecture is developed. It combines the advantages of electric and thermal power (GINDROZ, 2014). In general, The Hybrid is a technology that combines a combustion engine with another electric motor to drive a vehicle (Destraz, 2008) (Kermani, 2009). Hybridization is one of the promising alternatives in the short and medium term (Zahra Hajirahimi, 2020). Having two types of source makes it possible to recover energy and choose the most efficient operating point for the primary source (A.Khanipour, 2007). The automotive industry has already started to move towards the construction of green vehicles. This approach focuses mainly on three areas of emissions, noise and fuel consumption. The aim is to significantly reduce vehicle emissions and increase fuel economy, which is good for the environment (Latha KannusamyMY, 2019).

Agriculture is the most important industry, given its main role in providing food (Fedoroff, 2015). For our part, we have projected the hybrid structure on this industry through the realization of a low-cost tractor. The latter will allow us to minimize energy consumption and the rate of rejects.

Engines are the crucial source of energy, as well as the source of heat, in the powertrains of hybrid vehicles (Tatehito Ueda, 2004). An adapted thermal control system should remove heat loss from these constituents in order to maintain prescribed working temperatures while minimizing the energy consumption of the auxiliary cooling actuators.

As part of this work, we will propose a system that will ensure that powertrain temperatures are maintained at the correct level. The system will be evaluated by the Amesim software, by attaching the low-cost tractor to the Bette Harvest tool. The work of the latter requires the supply of energy from the power train.

Table 1. Main notations used in this paper.

Q_e	Thermal capacity of the internal combustion engine
$M_{w,e}$ and $M_{w,r}$	Coolant mass in the engine and radiator
$M_{a,r}$	the radiator air mass
$c_{p,w}$	the specific heat of coolant water
T_e, and $T_{a,o}$	denote engine temperature, coolant water temperature at engine outlet, coolant water temperature at radiator outlet,

	and air temperature at radiator outlet respectively.
h_e	equivalent heat transfer coefficient in the engine, which is also a function of the coolant mass flow rate, $\dot{m}_{air}$
T	the continuum temperature,
k and d	the thermal conductivity and specific heat respectively
q	denotes the heat flux
$\vec{t}$	the nodal temperature vector of the finite element mesh
K	the finite element matrices which correspond to thermal conductivity
D	is the finite element matrices which corresponds to specific heat.
$\vec{q}$	vector corresponds to the excitation of the thermal model, including heat transfer on cooling surface, friction and windage losses, conduction and core losses.
$\vec{t_s}$ and $\vec{t_r}$	the temperature nodes of motor stator and motor rotor respectively
h_m	conduction heat transfer in the air gap between the stator inner surface and the rotor external surface
$\vec{q_s}$ and $\vec{q_r}$	heat flux vector
h_m	the heat transfer coefficient
$\dot{m}_c$	Air mass flow rate
K_2	Positive constant

2 STATE OF ART

Previous studies have examined the overall cooling system architectures for conventional and hybrid vehicles and their effect on cooling system energy consumption (Sungjin Park, 2010)

Salah et al (Mohammad Salah, 2008) (Mohammad Salah T. H., 2010) applied non-linear control theory to the design of advanced thermal management systems for the cooling fan of the ICE and showed increases in temperature monitoring and energy efficiency. Cho et al (Hoon Cho, 2006) investigated the advantage of a controllable pump in a truck-engine cooling system. The study results demonstrated a clear decrease in pump power consumption and a possibly reduced size of the heat exchanger.

Shams-Zahraei et al (Mojtaba Shams-Zahraei, 2012) presented an energy management system for hybrid vehicles that incorporates a method of engine thermal management that provides overall optimal battery charging and discharging. A top-level optimisation of the propulsion system management has been realised using a dynamical programming methodology by Zhang et al (Xueyu Zhang, 2014). They have taken into account the impact of the losses in cooling of the auxiliary batteries and improved the global fuel efficiency.

3 PROPOSED COOLING SYSTEM

The low-cost tractor is an agricultural hybrid tractor, weighing 4.5 t, equipped with a 4-stroke V8 combustion engine with a total displacement of 12 L and a power of 120 KW. In The electric part, the tractor is equipped by the permanent magnet synchronous motor (MSAP) seems to be the most optimal solution for automotive traction thanks to its technical performance and in particular, its compactness and efficiency. It has been selected by Toyota in the Prius for its best efficiency, good dynamic performance thanks to the low stator inductances due to the large width of the apparent air gap, large magnetic field in the air gap and no DC voltage source for excitation. For this, the low-cost tractor was equipped with 20 KW synchronous motors. To ensure stable powertrain temperatures, the tractor equipped with a thermal management system. The system consists of an engine block (cooling jackets), thermostat, Coolant Temperature Sensor, radiator, and coolant circulatory pump. The proposed system is illustrated by the model built on AMEsim and shown in figure 1.

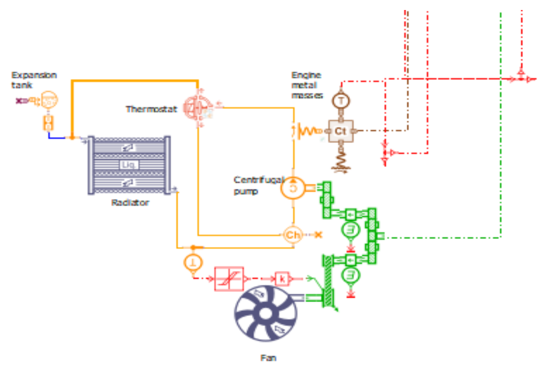

Figure 1: Sketch of Model 1D parallel hybrid tractor

4 MATHEMATICAL MODEL

The main objective of the suggested thermal management system is to eliminate the heat from the main components of the Hybrid Electric Vehicle's Powertrain, which includes the electric motors and the internal combustion engine. The electric engines and the internal combustion engine are both cooled by a circulated coolant. Thermal analysis of each component is detailed in section 3.1.

4.1 Thermal Engine Model

The (Internal Combustion Engine) ICE is the vehicle's main source of energy. It transforms about one third of the chemical energy in the fuel into propulsion power and the other third is absorbed by the exhaust gases. The remaining combustion energy is directed to the waste heat that must be removed by the cooling system. The conventional cooling system for the engine consists of a belt-driven coolant pump, a clutchable radiator fan and a wax thermostat. Different parameters describing characteristics of engine cooling system are represented by following equations (Gu Ning, 2009).

$$Q_e \frac{dT_e}{dt} = h_e(T_h - T_e) + Q_e \quad (1)$$

$$M_{w,e}c_{p,w}\frac{dT_e}{dt} = h_e(T_e - T_c) + \dot{m}_w c_{p,w}\ (T_c - T_h) \quad (2)$$

$$M_{w,r}c_{p,w}\frac{dT_c}{dt} = h_r(T_{amb} - T_h) + \dot{m}_w c_{p,w}\ (T_c - T_h) \quad (3)$$

$$M_{a,r}c_p\frac{dT_{a,0}}{dt} = h_r(T_h - T_{amb}) + \dot{m}_{air}c_p\ (T_{amb} - T_{a,0}) \quad (4)$$

4.2 Thermal Model of Electric Motor

The electric motor is another essential part of the powertrain of hybrid electric vehicles. Its propulsive power is restricted by temperature limits. Real-time thermal monitoring with knowledge of the inside temperature of the machine is very useful for improved operational safety and for determination of the torque/power capacity at any time during a driving cycle. A thermal model with clustered parameters has been introduced to record the thermal performance of the electric motor machine. Mathematical descriptions are very quick but are not able to demonstrate the temperature distribution inside the machine. To accurately estimate the internal temperature of the electrical machine, a finite element thermal model was also developed. Unfortunately, thermal finite element models for electrical machines with complex geometries are too cumbersome to calculate for the design of a real-time controller. In this work, a reduced order dynamic 3D thermal model based on the finite elements of an electric motor is chosen to explore a new concept of thermal control. The differential equation of thermal conduction is developed by (Kan Zhou, 2011) (Ayman M. El-Refaie, 2004).

$$\mathrm{d}\frac{dT}{dt} - k\nabla^2 T = q \quad (5)$$

The aforementioned equation can be expressed in discrete form using finite element techniques.

$$\mathrm{D}\dot{\vec{\mathrm{t}}} - \mathrm{k}\vec{\mathrm{t}} = \vec{\mathrm{q}} \quad (6)$$

By fundamentally changing and reducing the order of the model, this model can be refined into a state space form that matches the stator and rotor, respectively (Dezheng Wu S. D., 2010).

$$\dot{\mathrm{x}}_\mathrm{s} = \mathrm{A_s}\overrightarrow{\mathrm{x_s}} + \mathrm{B_s}\overrightarrow{\mathrm{q_s}} \quad (7)$$

$$\overrightarrow{\mathrm{t_s}} = \mathrm{V_s}\overrightarrow{\mathrm{x_s}} \quad (8)$$

$$\dot{\mathrm{x}}_\mathrm{r} = \mathrm{A_r}\overrightarrow{\mathrm{x_r}} + \mathrm{B_r}\overrightarrow{\mathrm{q_r}} \quad (9)$$

$$\overrightarrow{\mathrm{t_r}} = \mathrm{V_r}\overrightarrow{\mathrm{x_r}} \quad (10)$$

The heat transfer is calculated as

$$H_s = min\left[h_m(T_{so} - T_c), (T_{so} - T_c)\dot{m}_c c_{p,w}\right] \quad (11)$$

The model of the cooling system for electric motors is built on the same approach as outlined in equations (3) and (4).

4.3 Thermal Control Algorithms

A range of monitors, based on thermal models of each power train component, will be presented in this section. A non-linear controller is developed for the operation of the engine cooling system to monitoring the temperature of the coolant at the engine output and to reduce the energy consumption of the actuators. By using the high-fidelity thermal model of the electric motor, a new cooling strategy concept is developed to maintain the hot-spot temperature in the electric motor stator by following an ideal rate of heat removal from the cooling surface.

4.3.1 Control of Engine Cooling

The ICE is the largest source of heat in a vehicle. In this conception of the thermal management of the ICE, a liquid-air cooling cycle is applied. In the proposed system, both the coolant pump and the cooling air fan are electrically controlled. This improves temperature monitoring, optimises coolant flow and reduces cooling power consumption. Wang and his colleagues (Tianwei Thomas Wang, 2014) studied an optimal strategy for controlling the fan matrix to minimise the power of the radiator fan. The thermal model of the engine cooling system described in equations (1), (2), (3), (4) is applied to design a non-linear controller to stabilize the coolant temperature at the engine outlet by regulating the coolant and cooling air flow rates. Three assumptions are made to facilitate the controller design process,

A1. The thermal valve is fully open so that all coolant leaving the engine passes through the radiator.

A2. All temperature states can be measured with the available sensors and used as feedback signals in the control system.

A3. The size of the radiator is large enough to meet the heat dissipation requirements.

Define the desired coolant temperature at the engine outlet as Th,d, and define the control error as the difference between the actual value and the desired temperature.

$$e_c = T_h - T_{h,d}$$

The Lyapunov stability theorem (Marcio S. de Queiroz, 2001) concludes that, if a positive definite function composed of states of the system has a zero point only when the states are zero, and its time derivative is negative semi-definite, then the equilibrium point is stable. Based on this theory, a monitor can be designed to control the coolant mass flow rate and to monitor temperature. The failure dynamics can be written as shown below.

$$\dot{\mathrm{e}}_\mathrm{h} = \dot{\mathrm{T}}_\mathrm{h} - \dot{\mathrm{T}}_\mathrm{h,d} \qquad (12)$$

The coolant mass flow control law can be derived using the following procedures

$$\dot{\mathrm{e}}_\mathrm{h} = \dot{\mathrm{T}}_\mathrm{h} \qquad (13)$$

Define a Lyapunov cost function as

$$\mathrm{V}_1 = \frac{1}{2}\mathrm{e}_\mathrm{h}^2 \qquad (14)$$

So that

$$\dot{\mathrm{V}}_1 = \mathrm{e}_\mathrm{h}\dot{\mathrm{e}}_\mathrm{h} = \mathrm{e}_\mathrm{h}\dot{\mathrm{T}}_\mathrm{h} \qquad (15)$$

To eliminate tracking error, define the mass flow control law as

$$\dot{\mathrm{m}}_\mathrm{w} = \frac{\mathrm{M}_\mathrm{w,e}\mathrm{c}_\mathrm{p,w}\mathrm{K}_1\mathrm{e}_\mathrm{h} + \mathrm{h}_\mathrm{e}(\mathrm{T}_\mathrm{e} - \mathrm{T}_\mathrm{c})}{\mathrm{c}_\mathrm{p,w}(\mathrm{T}_\mathrm{h} - \mathrm{T}_\mathrm{c})} \qquad (16)$$

the dynamics of the tracking error as follows

$$\dot{\mathrm{T}}_\mathrm{h} = -\mathrm{K}_1\mathrm{e}_\mathrm{h} \qquad (17)$$

and equation (15) can be rewritten as

$$\dot{\mathrm{V}}_1 = -\mathrm{K}_1 e_\mathrm{h}^2 \qquad (18)$$

which indicates that the fault e_h is near zero with the coolant mass flow rate provided.

The fan powered by the cooling air consumes a large amount of energy and its speed must also be monitored. The cooling air mass flow control law is obtained by following the temperature of the coolant at the radiator outlet (also at the engine inlet), T_c, to a target value, $T_\mathrm{c,d}$ so that the heat removed by the coolant flow is the same as the heat transferred by the engine. The wanted coolant temperature at the radiator exit is expressed as

$$\mathrm{T}_\mathrm{c,d} = \frac{-\mathrm{K}_1\mathrm{e}_\mathrm{h}\mathrm{M}_\mathrm{w,e}\mathrm{c}_\mathrm{p,w} - \mathrm{h}_\mathrm{e}(\mathrm{T}_\mathrm{e} - \mathrm{T}_\mathrm{c})}{\dot{\mathrm{m}}_\mathrm{w}\mathrm{c}_\mathrm{p,w}} \qquad (19)$$

To minimise the consumption of energy by the cooling air fan, it is reasonable to find the minimum cooling air mass flow rate required. With assumption A_3, the heat transfer can be fully developed inside the radiator. The cooling air and coolant temperatures are very close to each other when they exit the radiator. Therefore, the required cooling air mass flow rate can be calculated by changing the temperature of the cooling air at the radiator outlet, $T_\mathrm{a,o}$, to the required coolant temperature at the engine inlet, $T_\mathrm{c,d}$, which is calculated in equation (19). Define another Lyapunov cost function as

$$\mathrm{V}_2 = \frac{1}{2}\mathrm{e}_\mathrm{a}^2 \qquad (20)$$

Where

$$e_a = T_{a,o} - T_{C,d}$$

such that

$$\dot{\mathrm{V}}_2 = \mathrm{e}_\mathrm{a}\dot{\mathrm{e}}_\mathrm{a} = \mathrm{e}_\mathrm{a}\left(\dot{\mathrm{T}}_\mathrm{a,0} - \dot{\mathrm{T}}_\mathrm{c,d}\right) \qquad (21)$$

With reference to equation (4), the control law for the desired air mass flow rate is set as follows

$$\dot{\mathrm{m}}_\mathrm{air} = \frac{\mathrm{M}_\mathrm{a,r}\mathrm{c}_\mathrm{p}(-\mathrm{K}_2\mathrm{e}_\mathrm{a} + \dot{\mathrm{T}}_\mathrm{c,d}) - \mathrm{h}_\mathrm{r}(\mathrm{T}_\mathrm{h} - \mathrm{T}_\mathrm{amb})}{\mathrm{c}_\mathrm{p}(\mathrm{T}_\mathrm{amb} - \mathrm{T}_\mathrm{a,0})} \qquad (22)$$

The term $\dot{\mathrm{T}}_\mathrm{c,d}$ can be obtained from equations (1),(2),(3) and the measured coolant mass flow rate. Enter the prescribed air mass flow rate into equation (4), the dynamic change of the air temperature at the radiator outlet can be expressed as

$$\frac{\mathrm{dT}_\mathrm{a,0}}{\mathrm{dt}} = -\mathrm{K}_2\mathrm{e}_\mathrm{a} + \dot{\mathrm{T}}_\mathrm{c,d} \qquad (23)$$

By substituting the term $\dot{\mathrm{T}}_\mathrm{a,0}$ in equation (20) with equation (22), we can observe that the dynamical change of the error cost function V_2 becomes

$$\dot{\mathrm{V}}_2 = -\mathrm{K}_2\mathrm{e}_\mathrm{a}^2 \qquad (24)$$

This suggests that the proposed mass-flow rate will remove the error e_a, since the coolant and cooling-air temperature converge at the heat exchanger outlet, the temperature of the cooling water at the engine inlet also converges to its desired value. Equations (16) and (22) provide the coolant mass flow control law for temperature monitoring and the minimum cooling air mass flow rate to meet the heat removal rate required.

4.3.2 Control of Electric Motor Cooling

The form of the state-space of the thermal model allows the design of an optimal regulator with five inputs; only the third input state (heat transfer by conduction cooling on the surface of the electric motor) can be controlled in practical terms. It is reasonable to take only the stator part of the electric motor as the cooling subject, since the heat source is located there and the hottest points are always in the stator. the optimal regulator can be designed on the basis of the thermal model of the state-space stator using the As, Bs and Vs matrices.

generally, the thermal model of the stator shows that the hottest points inside are the elements with indices between 2 and 12 in the output vector, $\vec{t}_r$ corresponding to the positions close to the windings. With these observed simulation results, the control objective can be reduced to only 10 elements by properly weighting the matrix design. In order to develop an optimal controller to calculate the ideal heat dissipation requirement, the cost function should be defined as

$$J_b = \int_{t0}^{t1} \left\{ \left[\vec{t}_s - \vec{t}_{s,r}\right]^T R_3 \left[\vec{t}_s - \vec{t}_{s,r}\right] + \tilde{q}_s R_4 \tilde{q}_s \right\} dt \quad (25)$$

where, R3 and R4 are positive symmetrical weighting matrices. In this case, R3 is designed to stabilise only the highest internal temperature of the stator, elements 2 to 12 of the model output vector, but does not control the other elements of the output vector.

This term $\tilde{q}_s = \vec{q}_s - \vec{q}_{s,0}$ defines the range of heat flux variation, and $\vec{t}_{s,r}$ is the reference temperature vector of the stator.

The optimal linear control law for an ideal heat flux on the cooling surface is derived as follows

$$\vec{q}_{s,d} = -F_1^{-1}\vec{x}_s + \vec{q}_{s,0} \quad (26)$$

The feedback and feed forward gain is defined as follows

$$F_1 = R_4^{-1} B_s^T P_2 \quad (27)$$

And

$$\vec{q}_{s,0} = C_s(R_3(A_s - B_s F_1)^{-1} B_s)^{-1} \vec{t}_{s,r} \quad (28)$$

Where P_2 is solution as the negative non-defined solution of the respective Riccati's algebraic equation. The term $\vec{q}_{s,d}$ is the goal input of the stator model: core loss; conduction loss; cooling heat flux; heat flux at the end of the machine and heat flux at the air gap surface. Therefore, the optimum rate of heat dissipation at the cooling surface of the machine is determined by the third element $\vec{q}_{s,d}$ of equation (26) which reduces the specified cost function J_2 to a minimum. To emphasize the important limitation that only the third element of this model is controllable, design the weighting matrices for the second element of the cost function as shown in the following example.

$$R_4 = \begin{bmatrix} 5.10^7 & 0 & 0 & 0 & 0 \\ 0 & 5.10^7 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 5.10^7 & 0 \\ 0 & 0 & 0 & 0 & 5.10^7 \end{bmatrix}$$

In other words, the weight of the third entrance is much smaller than that of the other four uncontrolled entrances. To reduce the prescribed cost function, the controller would concentrate on the feedback benefit on the third input. The heavy weights of the other inputs reduce the demands on the uncontrollable range of variation in heat transfer. The optimum rate of heat dissipation to the motor surface, $\vec{q}_{s,d}$ obtained by the optimal linear regulator based on the thermal model of the motor, is provided to the cooling system controller as a reference for monitoring. A tracking non-linear regulator is developed to operate the cooling pump and fan speed to track the optimum rate of heat dissipation to the motor surface. On the basis of the rate of heat transfer as defined in equation (11), the error of the heat dissipation rate tracking control is defined as follows

$$e_H = H_{s,d} - \min\left[h_m(T_{so} - T_c), (T_{so} - T_c)\dot{m}_c c_{p,w}\right] \quad (29)$$

Where $H_{s,d}$, is the desired cooling surface heat flux. The term h_m is the heat transfer coefficient defined as a function of the refrigerant mass flow rate. Now design a cost function

$$V_3 = \frac{1}{2} e_H^2 \quad (30)$$

And its temporal derivative expressed in the form of

$$\dot{V}_3 = -e_H(\dot{H}_{s,d} + \dot{H}_s) \quad (31)$$

The regulator can be obtained by computing an ideal variation in the rate of heat removal, for example

$$\dot{H}_s = \dot{H}_{s,d} + K_3 e_H (32)$$

Where K3 is a positive constant. If we study the relationship between the rate of heat loss and the mass flow rate of the coolant, then the heat transfer over the cooling surface of the stator follows the rule of dynamic change as follows

$$\dot{H}_s = (T_{so} - T_c)\frac{\partial h_m}{\partial \dot{m}_w} - h_m \frac{dT_c}{dt} \quad (33)$$

The ideal coolant mass flow rate is designed as follows

$$\frac{d\dot{m}_w}{dt} = \frac{\dot{H}_s + K_3 e_H - h_m \frac{dT_c}{dt}}{(T_{so} - T_c)\frac{\partial h_m}{\partial \dot{m}_w}} \quad (34)$$

So that the control law for obtaining the cooling water mass flow rate is

$$\dot{m}_w = \int \frac{\dot{H}_s + K_3 e_H - h_m \frac{dT_c}{dt}}{(T_{so} - T_c)\frac{\partial h_m}{\partial \dot{m}_w}} \quad (35)$$

By substituting the term $\frac{\partial h_m}{\partial \dot{m}_w}$ in equation (33) with equations (34) and (35), we can get that $\dot{H}_s =$

$\dot{H}_{s,d} + K_3 e_H$. The dynamic change of the tracking error function $\dot{V}_3$ becomes

$$\dot{V}_3 = e_H(\dot{H}_{s,d} - \dot{H}_{s,d} - K_3 e_H) = -K_3 e_H^2 \qquad (36)$$

And this demonstrates that the mass flow rate of coolant proposed in equation (35) will result in the monitoring $H_{s,d}$. In the electric engine cooling system, all four on-board engines are cooled by the same radiator at the same time. To decrease the waste of energy in the radiator fan, the airflow can be optimized. To investigate the minimum required airflow, it is presumed that the heat removal by the radiator cooling airflow is the same as the heat removed from the engines by the coolant. The mass air flow rate must therefore be adjusted in proportion to the desired heat extraction rate in the radiator, which is four times the required heat flow over the cooling surface for each electric motor. The radiator fan in the cooling cycle of the electric motor operates in the following way

$$\dot{m}_{air} = \frac{4H_{s,d}}{c_p(T_{a,o} - T_{amb})} \qquad (37)$$

For comparison, a Conventional controller P_I is included to monitor the stator hot-spot temperature and is given as follows:

$$\dot{m}_w = K_1 \int R_3(\vec{t}_s - \vec{t}_{s,r})dt + K_p R_3(\vec{t}_s - \vec{t}_{s,r}) \qquad (38)$$

For cooling the electric motor and the ICE, the mass flow rate of the coolant is in proportion to the pump speed and the mass flow rate of the air is in relation to the fan speed. The fan and pump power consumption rates are reduced to be in proportion to the air mass flow and coolant mass flow, respectively. In simulation, the cooling system power consumption requirement is provided by the fan/pump sub-model of the AMESim.

5 RESULTS AND DISCUSSION

The results of these simulations are carried out with the low-cost tractor. The latter is an agricultural tractor, weighing 4.5 t, equipped with a 4-stroke V8 combustion engine with a total displacement of 12 L and a power of 120 KW. In addition, it is equipped with a 20 KW electric motor to give a hybrid structure. The outputs of the two engines are linked by a planetary gear train. The output power is adapted by a manual gearbox with 4 gears. In addition, it is equipped with a generator that allows us to recover the electrical energy from the thermal engine and store it in a lithium battery. The tractor is attached to the Bette Harvest tool, which is an energy-demanding tool. The speed has been maximized to 9 km/h, as recommended by the American Society of Agricultural Engineers. The adopted driving cycle is shown in the figure 2.

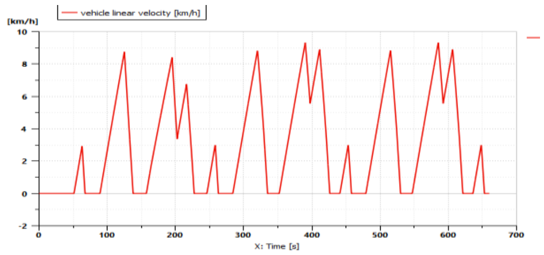

Figure 2. Tractor driving cycle with the Bette Harvest tool attached

The simulation results show that the electric motor, as in figure 3, has a low contribution during the driving cycle due to the high-power demand of the tool. The state of the internal combustion engine, as in figure 4, illustrates a contribution in the cycle. With a generator contribution during the operation of the combustion engine ensures an accumulation of energy in the battery. The battery has its role in supplying power to the electric motor. It can be seen that the addition of the electric motor significantly reduces the use of the combustion engine.

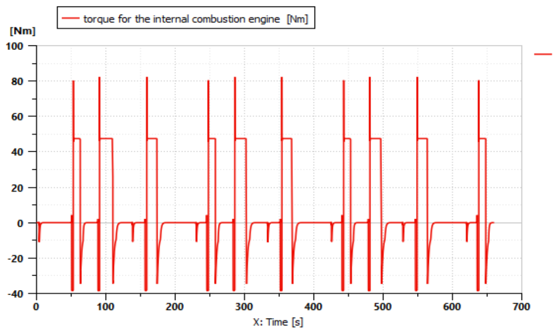

Figure 3. Variation of the thermal motor torque

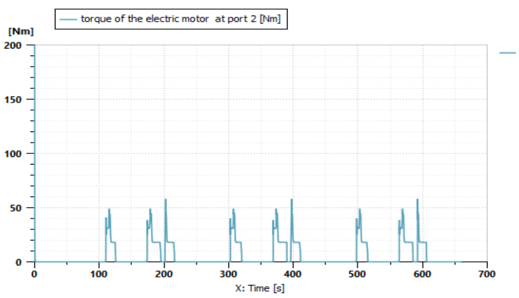

Figure 4. Variation of the electric motor torque

The results of the simulation, shown in Figure 5, illustrate that the heat of the combustion engine shows a continuous rise for 300 s and then it is stabilised at 96 degc.

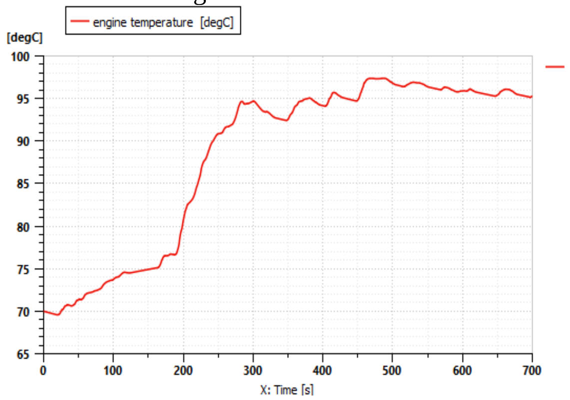

Figure 5. Variation of the temperature of thermal motor

The results of the simulation, shown in the figure 6, illustrate that the heat of the electric motor shows a continuous rise for 350 s and then it is stabilised at 95 degc. From the second 400 the temperature starts to decrease until it reaches 70 degc.

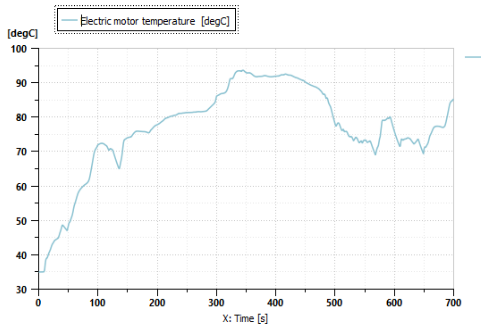

Figure 6. Variation of the temperature of electric motor

6 CONCLUSION

Agriculture is one of the most important industries in the world. The tractor is mainly regarded as the axial machine in this sector. The present work is an evaluation of the low-cost tractor with a parallel hybrid engine. The simulation was carried out using the AMEsim software, attaching it to the Bette Harvest tool and equipping it with a cooling system. The Bette Harvest is a tool requiring very high power. The cooling system consists of a radiator, a thermostat, a centrifugal pump and a fan. The results showed that the proposed cooling system was able to guarantee a stabilisation of the temperature of both engines.

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