VIRTUAL SENSOR FOR THE OIL TEMPERATURE ESTIMATION OF AUTOMOTIVE OTTO ENGINES

Roger T.G. Guilherme, roger.guilherme@volkswagen.com.br
Volkswagen do Brasil, Product Development - Powertrain
Prof. Dr. Silvio de Oliveira Jr., silvio.oliveira@poli.usp.br
Polytechnic School of the University of São Paulo, Mechanical Engineering Department

Abstract. *This article consists in a study of virtual sensors used for control actions of automotive Otto engines, aiming cost reduction alternatives to their management. Virtual sensor is an expression to refer an estimator of one or more state variables of a system, based on signals provided by already present physical sensors and information of the system working characteristics. In order to understand the steps and difficulties to create a virtual sensor, a simple example was developed with the objective of estimating the engine oil temperature during its operation. To accomplish this task, a physical model to represent the oil circuit and the mathematical equations to simulate the heat exchange processes that influence its temperature were proposed. The text suggests also experimental approaches to the calibration of the model parameters, and to test its viability. Finally, a discussion about limitations of the model and the proposed calibration method is presented.*

Keywords*: Internal combustion engines, state estimator, lubricating oil, heat rates.*

1. INTRODUCTION

Virtual sensor is the name given to software routines, which, based on existing physical signals and system information are capable of inferring the value of a system state variable, while it is working. The most popular example of virtual sensor in the Brazilian automotive industry is the logical fuel sensor, which detects the amount of (bio) ethanol in gasoline-ethanol mixture present in the car tank, allowing the engine management system to adapt the engine to work with any mixture of these fuels. In production since March 2003, this (renewable CO_2 friendly) flexible fuel technology, at present, equips about 80% of the cars sold in the country and it is present in about 100% of the engines produced by Volkswagen in Brazil. The success of this technology, the economic boundary conditions in the country and the cost sensitive characteristic of the market are an incentive for the study of other virtual sensors, aiming the improvement of the functions of the engine management system without (or with small) increasing the variable cost or the necessity of hardware investments.

The present work, therefore, deals with the proposal of an estimator for the lubricating oil temperature of automotive Otto engines. Some possibilities of the use of this signal were identified in the practice. An example would be the control of the electrical fans of the cooling system. Currently this control is based on the coolant temperature (and air conditioning pressure, if it is present). In some running conditions is it possible to have a high oil temperature, which can deteriorate the oil and its lubricating characteristic), simultaneously to a low coolant temperature, sufficient for not turning on the fans. It was observed that the functioning of fans can bring the oil temperature some degrees down. A sensor to indicate the oil temperature would be therefore desirable. Another situation refers to the warm up phase, when the fuel (especially ethanol) mixed with the oil, during the cold start process, starts to vaporize, causing the enrichment of the air-fuel mixture, the increasing of emissions and, in the case of flexible fuel engines, a false interpretation of the fuel in the tank. Monitoring the oil temperature allows the management system to avoid misinterpretation of the signal variation of the exhaust oxygen sensor – lambda sensor.

An abundant literature referring the use of virtual sensors in the management system of engines was not found. This term is much more common in other areas such as the control engineering. From the point of view of the control engineering, virtual sensors can be understood as state observers. Luenberger (1979), Franklin *et al* (1994), Ogata (1998) and Phillips and Harbor (2000) state a standard form of a linear system based on a set of linear first order differential equations presented in a matrix form:

$$d\mathbf{x}(t)/dt = \mathbf{A}\,\mathbf{x(t)} + \mathbf{B}\,\mathbf{u(t)} \tag{1}$$

$$\mathbf{y}(t) = \mathbf{C}\,\mathbf{x}(t) \tag{2}$$

where **A**, **B** and **C** are, respectively, the system, the input and the output matrices, **x** is the state vector (containing the variables that describe the system), **u** is the input vector (containing the variables to control the system) and **y** is the output vector, which represents the state variables that are effectively being measured.

As commented by Luenberger (1979), it is common in practice that the measurement of every component of the vector **x** is not possible. An estimation of some variables is therefore necessary for the system control. Luenberger (1979) and Franklin *et al.* (1994) show that a state estimator (also called observer) can be designed as an "open loop" considering information of the input and system characteristics, as shown with continuous lines in the Fig. 1. "Close loop" observers use also information of the output of the system, as exemplified with dashed lines in the same picture.

For linear systems, the references Luenberger (1979), Franklin *et al* (1994), Ogata (1998) and Phillips and Harbor (2000) (among others) show that a close loop state observer is possible, if the system is proved to be observable. According to Friedland (1987), these concept was first defined by Kalman in the mid of the Fifties. A system is "observable" if it is possible to determine its initial state ($\mathbf{x}$(t=0)) by using the output $\mathbf{y}$ for a finite time interval (between 0 and an instant t_1). Luenberger (1979) shows that a state estimator $\hat{\mathbf{x}}$ of a linear (and observable) system can be build so that the error of the estimated vector $\hat{\mathbf{x}}$ and the real vector $\mathbf{x}$ tend to zero with the time.

In terms of industrial application, Albertos and Goodwin (2002) present a review of this theory and some examples. In an automotive case, Gustafsson *et al.* (2001) describe a virtual sensor for estimation of tire pressure and road friction for use in automobiles. The tire pressure sensor can be determined through the observation of the variation of the angular velocity of the wheel (measured with a toothed wheel).

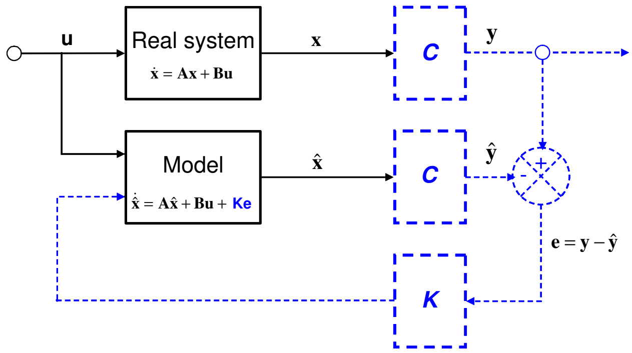

Figure 1: State estimator (FRANKLIN *et al.* (1994))

Monerat *et al*. (2000) describe the functioning of the virtual sensor to determine the amount of ethanol in a mixture with gasoline by using the signal of the oxygen sensor present in the exhaust system of almost all cars produced in Brazil. The driver can choose which kind of fuel he or she wants to tank. If a different mixture comes to the engine the oxygen sensor informs a non stoichiometric combustion. The system makes an adaptation of the amount of fuel injected until the stoichiometric value is reestablished. The magnitude of the correction is the parameter correlated to the percentage of ethanol in the mixture and it is used to correct other engine working parameter, such as the spark advance map.

In order to propose a model for the oil temperature some papers about the characteristics of the heat transfer in engines were used as references. Gruden *et al.* (1989) describe the influence of functioning parameters on the heat transfer to the lubrication oil. Part of these results was confirmed by Sebesse *et al.* (1998). The authors show also that the heat rate changes slightly with the oil temperature, tending to be higher with lower temperatures. The same occurs with the coolant. Trapy and Damiral (1994) have studied the heat transfer during the warm up of spark ignition engines. The authors observed that the heat rate to the oil is much more difficult than to the coolant. The time to reach a specific temperature is about the double for the oil.

Kaplan and Heywood (1994) proposed a model to study the warm-up phase of the engine dividing the engine in lumped masses with individual heat capacities. This procedure is interesting for the objective of this work because the oil, the carter walls, crankshaft and connecting rods were all considered a mass with uniform temperature that gains and looses heat. Alkidas (1994) showed that it is possible to make a good description of the heat rate to the coolant by using a linear correlation with engine speed and engine load (manifold pressure). The author also mentioned that the heat transfer to the oil will vary with the coolant temperature.

2. MODEL PROPOSAL

The actual oil circuit of a conventional automotive engine was simplified to the model described in the Fig. 2. A control volume analysis is carried on in order to make the assumption evident. A control volume is defined as the dashed line showed in the picture. The energy equation applied to the defined control volume gives:

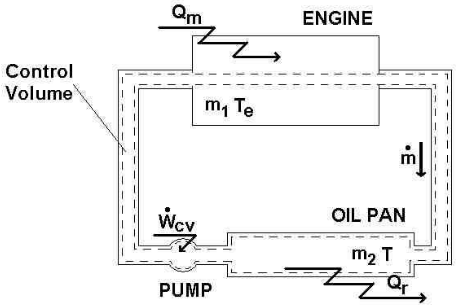

Figure 2: Model for the oil circuit

$$\dot{Q}_{cv} = \left[\frac{d}{dt}\iiint_{cv} \rho.e.dV\right] + \left[\iint_{cs} (h + \frac{v^2}{2} + zg)\rho.\bar{v} \bullet d\bar{A}\right] + \dot{W}_{cv} \quad (3)$$

In Eq.(3), e is the specific internal energy and h the specific enthalpy of a small amount of mass that crosses the elementary surface $d\bar{A}$ with a velocity vector $\bar{v}$ and a specific potential energy zg, and $\dot{W}_{cv}$ is the work entering the control volume (Van Wylen and Sonntag (1976)). In order to simplify Eq. (3), the following assumptions were considered: (a) the engine (i.e. the metallic parts, the combustion gases and the coolant) exchanges heat with the oil only during the oil flow through its channels. The net heat rate received by the oil is Q_m. It is assumed also that Q_m includes also the work done by the pump on the oil, since this work is also transformed in internal energy by the viscosity effects; (b) The oil exchanges heat (the net heat rate lost to the ambient air is Q_r) with the ambient air only through the oil pan walls via convection (the radiation and conduction is neglected); (c) The oil in the oil pan is treated as a body with uniform temperature (Biot number much smaller than 1). Its physical properties (specific mass ρ and heat capacity, c) are the same in all points of the oil circuit, and are constants; (d) The total oil mass is assumed to be constant (i.e. engine oil consumption is neglected) and it was divided into two parts (also constant in time): m_1, flowing inside the engine with temperature T_e, and m_2, laying into the oil pan with temperature T; (e) The variation of temperature in the oil pan is similar to the variation in the engine, which means: $dT/dt = dT_e/dt$.

Since it is made the assumption that there is no mass flowrate across the control volume boundaries, the second term of the right side in Eq. (3) can be vanished. As mentioned in the assumption (a), the pump power was added to the net heat entering the control volume. Thus, $\dot{Q}_{cv} - \dot{W}_{cv} = Q_m - Q_r$. Taking also into account assumptions (c), (d) and (e), Eq. (3) can be simplified to Eq. (4).

$$\frac{dT}{dt} = \frac{Q_m - Q_r}{mc} \quad (4)$$

Considering the assumption b and Newton's law for cooling [15], Q_r can be described as: $Q_r = H(T - T_{air})$, where T_{air} is the ambient air temperature. Substituting Q_r in equation (4), the following first order ordinary differential Eq. (5) is found:

$$\frac{dT}{dt} + \frac{H}{mc}T = \left(\frac{HT_{air} + Q_m}{mc}\right) \quad (5)$$

Equation (5) accepts the exact solution, as in Eq. (6), if H, mc, Q_m and T_{air} are considered constants along the time t.

$$T(t) = T_{air} + \frac{Q_m}{H} - \left[T_{air} + \frac{Q_m}{H} - T(t=0)\right]e^{-\frac{H}{mc}t} \quad (6)$$

Nevertheless, for a computation model, it is proposed to use the discretization of the differential Eq. (5), where dt becomes a finite, but sufficiently small, Δt as described in Eq. (7).

$$T(t) = T(t - \Delta t) + \left\{\frac{Q_m}{mc} - \frac{H}{mc}\left[T(t - \Delta t) - T_{air}\right]\right\}\Delta t \quad (7)$$

Equation (7) expresses the oil temperature at the instant t as a function of its value in instant $t\text{-}\Delta t$ and the heat transfer from the engine to oil and from this to the ambient air. To complete the model, it is now necessary to propose models to estimate Q_m, H and the initial oil temperature, based on existing measurements of the engine management system.

The heat transferred to the coolant can be estimated based on a linear expression involving the RPM (engine speed) and the MAP (manifold absolute pressure) as described by Alkidas (1994). Therefore, considering that the heat transferred to the oil is roughly a percentage of that added to the coolant, Q_m can be also represented similarly by Eq.(8).

$$\frac{Q_m}{mc} = K_1\left[RPM(t)\right] + K_2\left[MAP(t)\right] \tag{8}$$

As described by Özisik (1985), there are different expressions that correlate the heat transfer coefficient of a body during the forced convection through the typical flow and heat transfer dimensionless numbers, *Re* (Reynolds) and *Nu* (Nusselt). Based on these relations, to describe H/mc, Eq. (9) is suggested.

$$\frac{H}{mc} = K_3 V^n \text{ if } V>V_1 \tag{9}$$

When the car speed is very small and the forced convection ceases, an expression that describes the natural convection of a body should be used. Also based in Özisik (1985), the expression (10) is proposed.

$$\frac{H}{mc} = K_4\left[T(t) - T_{air}\right]^p \text{ if } V \leq V_1 \tag{10}$$

The parameters K_i, i=1..4, n and p must be estimated through experimental procedures.

3. OBSERVABILITY ANALYSIS

After having proposed the model for the temperature it is worthy to check if a system representing the engine has the property of observability. As mentioned in the introduction, if the system is “observable”, an estimator can be build and it will tend to reduce, for example, the error introduced by a false guess of the initial oil temperature (when the engine is turned on).

According to several references (see, for example, Luenberger (1979)), the observability of a linear system is guaranteed if the matrices **A** (system) and **C** (output) of the Eq. (1) and (2) are so that the rank of the observability matrix is equal to the rank of **A**, i.e. its determinant is not zero, as showed in expression (11).

$$\det\begin{bmatrix} C & CA & \ldots & CA^{n-1}\end{bmatrix}^T \neq 0 \tag{11}$$

To describe the state of an engine the vector $\boldsymbol{x}$ can be arranged with several components. The engine speed, the coolant, the oil and the exhaust gas temperatures are examples of state variables. Since the relation between these variables is not linear, in order to simplify the analysis, the number of state variables will be taken as a minimum: the engine speed $\dot{\theta}$ (which can be measured) and the oil temperature (to be estimated). The manifold absolute pressure will be considered the control variable $\boldsymbol{u}$. Additionally, the analysis will be performed around a specific working point (low velocity, constant coolant temperature, fixed spark advance and fixed gear ratio) to allow a linearization. With these assumptions, the following steps show how the components of the matrix $\boldsymbol{A}$ were found.

The first equation of the system will be derived from the application of Newton´s law to the engine shaft, which has a moment of inertia J, is subjected to a positive moment M_e, a resistive Moment M_r, and develops an angular acceleration $\ddot{\theta}$, Eq. (12).

$$J\ddot{\theta} = M_e - M_r \tag{12}$$

The resistive moment comes from the car movement and can be expressed by Eq.(13), developed according to Taborek (1957):

$$M_r = \left(F_w + m_v \left[\frac{r_w \ddot{\theta}}{R_{tr}} \right] \right) r_w / (\eta_{tr} R_{tr}) = \left(0.5 \rho_{atm} c_x S \left[\frac{r_w \dot{\theta}}{R_{tr}} \right]^2 + m_v g (aV^2 + bV + c) + m_v g \sin\alpha + m_v \left[\frac{r_w \ddot{\theta}}{R_{tr}} \right] \right) r_w / (\eta_{tr} R_{tr}) \quad (13)$$

In Eq.(13) F_w is the sum of resistance forces: aerodynamic drag (which is a function of the air density , aerodynamic coeficient, c_x, frontal surface, S and vehicle speed), rolling resistance, F_{rol} (that is usually adjusted by a quadratic polynomial expression of the vehicle speed V), and gradients; r_w is the dynamic radius of the car wheel; g is the gravity; R_{tr} the transmission ratio; η_{tr} the transmission efficiency; m_v the vehicle mass and α the slope of the track. The effect of the rotating inertias was ignored for simplicity.

If parameters like air density, engine displacement, heating value of the fuel, ambient temperature air-fuel ratio are considered constants in this simplified analysis, the torque that the engine delivers to the shaft is described by Eq. (14), developed based on Heywood (1989):

$$M_e = M_e(\dot{\theta}, MAP, T_1) \quad (14)$$

$$T_1 = T - T_{air} \quad (15)$$

The expressions of M_e and M_r must be linearized in order to simplify the analysis. For the equation of M_r, the aerodynamic force can be expressed around a small velocity V_{ref} , as suggested by the linearization method proposed by Luenberger (1979) . For these low velocities, the rolling resistance will be considered independent of the velocity, F_{rol} $=m_v c$ (c is the constant showed in Eq.(13)). The ramp resistance, for small gradients, is considered proportional to the ramp angle ($m_v g\alpha$). Therefore, the linear expression of F_w becomes (Eq. (16)):

$$F_w \cong V_{ref} \rho_{atm} S c_x V + m_v g c + m_v g \alpha + m_v \frac{dV}{dt} \quad (16)$$

For the expression of M_e, the linearization process suggested by Luenberger (1979) for functions of several variables it is applied,

$$M_e = \left[\frac{\partial M_e}{\partial \dot{\theta}} \right]_{ref} \dot{\theta} + \left[\frac{\partial M_e}{\partial MAP} \right]_{ref} MAP + \left[\frac{\partial M_e}{\partial T_1} \right]_{ref} T_1 + F_1 \quad (17)$$

$$F_1 = M_e^{ref} - \left[\frac{\partial M_e}{\partial \dot{\theta}} \right]_{ref} \dot{\theta}_{ref} - \left[\frac{\partial M_e}{\partial MAP} \right]_{ref} MAP_{ref} - \left[\frac{\partial M_e}{\partial T_1} \right]_{ref} T_{1ref}$$

The linearized expressions (16) and (17) were substituted in the equation (12) resulting in Eq. (18).

$$\left[J + \frac{m_v r_w^2}{\eta_{tr} R_{tr}^2} \right] \ddot{\theta} = -\frac{9{,}7 \rho_{atm} S c_x r_w^2}{R_{tr}^2} \dot{\theta} - \frac{m_v g (c + \alpha) r_w}{\eta_{tr} R_{tr}} + M_e \quad (18)$$

M_e is given by Eq. (17). Rearranging the equation in order to isolate $\ddot{\theta}$ and nominating the constants with letter F, it is possible to find the first equation of the system (Eq. (19)).

$$\ddot{\theta} = F_4 \dot{\theta} + F_3 \left[\frac{\partial M_e}{\partial T_1} \right]_{ref} T_1 + F_2 MAP + [F_1 - \frac{m_v g (c + \alpha) r_w}{\eta_{tr} R_{tr}}] F_3 \quad (19)$$

The second equation is the model proposed to simulate the oil temperature, equations (5), (8) and (9). The term of H/mc is non-linear (depends on V^n, according to equation (9)). Therefore, the same technique showed in Luenberger (1979), applied above, is used. Again, the letter F is used to represent constants that appear in the linearization process.

$$K_3 V^n T_1 \cong F_5 + F_6 \dot{\theta} + F_7 T_1 \quad (20)$$

Substituting in the Eq. (5) (using T_1 instead of T) and using the expression for Q_m proposed in Eq. (8), the second equation of the system is found (Eq.(21)):

$$\dot{T}_1 = (K_1 - F_6)\dot{\theta} - F_7 T_1 + K_2(MAP) - F_5 \tag{21}$$

The system can now be expressed in term of its matrices, considering that only the engine speed is measured.

$$\begin{Bmatrix} \ddot{\theta} \\ \dot{T}_1 \end{Bmatrix} = \begin{bmatrix} F_4 & F_3\left[\dfrac{\partial M_e}{\partial T_1}\right]_{ref} \\ (K_1 - F_6) & -F_7 \end{bmatrix} \begin{Bmatrix} \dot{\theta} \\ T_1 \end{Bmatrix} + \begin{bmatrix} F_2 \\ K_2 \end{bmatrix} \{MAP\} + \begin{Bmatrix} \left(F_1 - \dfrac{m_v g(c+\alpha) r_w}{\eta_{tr} R_{tr}}\right) F_3 \\ -F_5 \end{Bmatrix} \tag{22}$$

$$\{y\} = [1 \quad 0] \begin{Bmatrix} \dot{\theta} \\ T_1 \end{Bmatrix} \tag{23}$$

In the above system, the matrices **A**, **B** and **C** are well expressed in terms of constants that can be calculated using the expressions showed along this section. The calculation of the observability matrix and its determinant gives:

$$\det \begin{bmatrix} C \\ CA \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ F_4 & F_3 \dfrac{\partial M_e}{\partial T_1} \end{bmatrix} = F_3 \frac{\partial M_e}{\partial T_1} \tag{24}$$

The dependence of the observability of this simplified system on the variation of the engine effective torque with the oil temperature is therefore evident. Due to the variation of the oil viscosity with the temperature, it is possible to conclude that the system is observable. Therefore, theoretically, a close loop observer could be built so that, independent of the initial oil temperature assumed by the engine management unit, the error between the observer and the system would be reduced with the time. Nevertheless, measurements of this quantity carried out during this work in some engines give a value of about 0,04Nm/ºC for T_1=80ºC. This value can be even lower for higher oil temperatures.

Due to this small value, the number of simplification necessary to build a linear system, and the dependence of this system on parameters difficult to estimate in practice (e.g. the road ramp), an alternative to determine the initial oil temperature was proposed, in order to make viable an open loop estimator.

4. PROPOSAL OF A MODEL FOR THE INITIAL TEMPERATURE

The proposal consists of using a function of the engine outlet coolant temperature (T_{cool}) to estimate the initial oil temperature. This function should only be calculated after approximately 30 seconds of the engine start-up, to allow that the temperature T_{cool} stabilizes after a soak period. The format of the function is the following:

If $T_{cool} \leq T_{cool\_1}$ then $T_0 = T_{cool}$

If $T_{cool} > T_{cool\_1}$ then $T_0 = T_{cool} + f(T_{cool})$ (25)

The function f(T_{cool}) is to be found by experimental means.

5. CALIBRATION PROCEDURE AND RESULTS

The proposed model by the Eq. (7) requires a procedure to determine the value of the coefficient of convection H, which depends on the parameters K_3, K_4, n and p. In order to find a real parameter and test the proposed calibration procedures a vehicle VW-Fox (model year 2003) equipped with a 1,6L, 74kW engine was used.

To calibrate the parameter of K_3 and n of the Eq. (9), the following procedure was proposed and tested: (a) to set the car on a chassis dynamometer with a frontal fan, which velocity can be set independently of the vehicle speed; (b) to heat up the oil temperature up to 150ºC, stop the car wheels, turn the engine off and keep the dynamometer fan speed at a constant speed, for example 100km/h; (c) with a data-logger, acquire the oil temperature variation until it reaches 50ºC; (d) to calculate T-T_{air}, along the time interval; (e) using the Eq. (6), with Q_m=0, to apply the natural logarithm to both sides to find the Eq. (26). Finally, (f) Use a linear regression of the points to find the angular coefficient of the line represented by Eq. (26).

$$\ln[T(t) - T_{air}] = \ln[T(t=0) - T_{air}] - \frac{H}{mc}t \qquad (26)$$

The process from (a) to (e) was carried out with several velocities of the chassis dynamometer ventilator, making it possible to find several pairs (*V*, *H/mc*). Applying the natural logarithm in both sides of Eq. (9), the result is also the equation of a line. Applying a linear regression to the points (*lnV*, *ln*(*H/mc*)), the parameters *n* and K_3 could be estimated (*n*=0,645 and K_3=1,138*10^{-4}).

The determination of the coefficient for natural convection followed the same experimental procedure. Nevertheless, if the Eq. (10) is substituted in Eq. (5), the following differential equation (Eq. 27) is found.

$$\frac{d[T - T_{air}]}{dt} + K_4[T - T_{air}]^{p+1} = 0 \qquad (27)$$

Expression (27) is known as Bernoulli differential equation and has a solution presented by Weisstein (2006). Since this solution is difficult to transform in a line, the parameters *p* and K_4 were adapted manually to the experimental data (oil cooling with no fan velocity) and the following values were found: *p*=0,0931 and K_4= 1,9. 10^{-4}.

With the values of the forced and natural convection coefficients, it was possible to draw a graph of the heat rate rejected by the oil to the ambient air as a function of the air velocity (relative wind) and the difference between the oil and the air temperatures (Fig. 3). The graph shows also the equations found for the oil heat rejection.

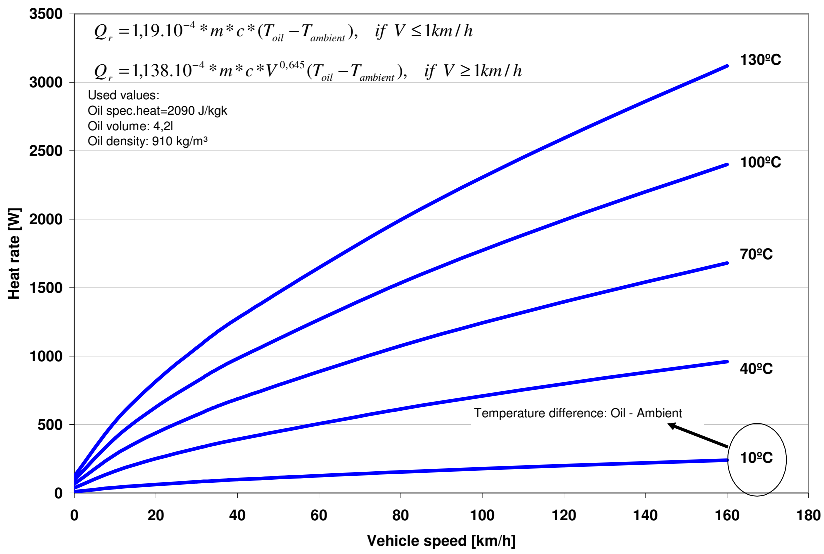

Figure 3: Heat rate rejected by the oil to the ambient air

The proposed model requires also the determination of Q_m. For this task, a method using the already determined values of Q_r is proposed. It consists in fixing a condition of *RPM* and *MAP* in the car using the chassis dynamometer. The speed of the air generated by the ventilator in front of the car is set in a constant and known value. The car runs in the same condition until the stabilization of the oil temperature (*dT*/*dt*=0). The coolant temperature was kept around 95ºC. In this condition, the Eq. (5) can be modified and Q_m can be expressed in term of the (stabilized) oil temperature and the convection coefficient that was previously found.

$$\frac{Q_m}{mc} = K_3 V^n (T_{stabilized} - T_{air}) \qquad (28)$$

This procedure was repeated for several points of (*RPM*, *MAP*). The results are showed in the graph of the Fig. (4).

The graph of the Fig. (4) shows that the heat transfer to the oil follows a linear relation with the load (*MA*P) and a quadratic relation with the engine speed (*RPM*). Because the intention is to have a model following Eq. (8), a linear (bidimensional) regression was carried out. The result is expressed by the Eq. (29).

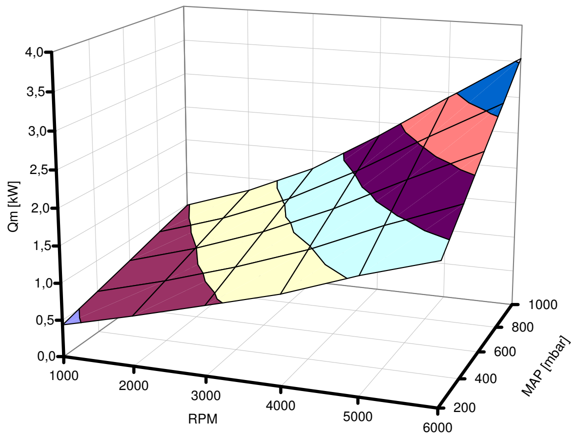

Figure 4: Heat rate rejected by the engine to the oil

$$Q_m = 3{,}90 * 10^{-4} * RPM(t) + 1{,}27 * 10^{-3} * MAP(t) - 5{,}16 * 10^{-1} \text{ [kW]} \quad (29)$$

Equation (29) of Q_m and Q_r (9 and 10 with the respective numerical parameters) were used to check if the proposed model was capable of following the real temperature in a normal running condition. The car was driven in a normal traffic and the values of velocity, engine speed and manifold pressure were saved with a data logger. The graphs in Fig. (5) and (6) show the comparison of the actual and the calculated oil temperatures. The step of time used was two or three seconds. The methods of measurements and accuracy of the parameters used (for all experimental work done for this paper) are showed in Tab. 1.

Table 1. Measurement methods and accuracy.

Parameter	Unit	Measurement Method	Accuracy
Temperatures	ºC	Thermocouple K-type	+/- 1ºC
Engine speed	RPM	Toothed wheel attached to crankshaft with magnetic pick-up	0,05% + 1RPM
Vehicle velocity	km/h	Toothed wheel in differential housing and magnetic pick-up	0,05% + 0,9 km/h
Manifold Absolute Pressure (MAP)	mbar	Voltage change due to flexure of a silicon chip	1,6kPa (or 16 mbar)

6. DISCUSSION

The proposed model is capable to follow the actual oil temperature. In Fig. (5) it is possible to observe that the calculated value remained most of the time below the real value (about 7ºC). One reason for that is that the values of Q_m used for the calculation came from a linear regression, Eq. (29), which gives, for low *RPM* and *MAP*, lower values of Q_m than the surface showed in the Fig. (4). When the car was driven in more constant velocities and higher values of *RPM* and *MAP*, the results of the calculation were better (Fig. (6)). Another possible improvement of the calculated oil temperature would be the use of a correction of Q_m with the coolant temperature.

It is important to comment the limitation of the virtual sensor, since its viability depends on which control actions will be taken due to its output. The following limitations are to be considered: (a) the mass and the specific heat of the oil is all the time considered constant. Along the use of the car, due to the mixture with fuels it is possible that the specific heat of the oil increases. Fortunately the mass goes in an opposite direction, because of the oil consumption. (b) The customer can change the type of oil used. If the characteristics of the new oil are different from that one used to calibrate the virtual sensor, an error of estimations will occur. (c) Modifications of the air flow profile around the oil pan can occur, for instance, if the customer decides to install an oil pan protector. Also, the wind or the rain can alter the convection coefficient around the engine compartment. In all cases, the control action will be taken with a different prediction. (d) The effects of the higher friction in the break-in phase (new engine) were not considered.

A special attention deserves the way of determination of the initial oil temperature. If a close loop state observer can not be developed, any error in this parameter will cause an error in the further temperature calculations. This error tends, in an open loop observer of a stable system, to become lower with the same "velocity" as the system tends to reach its equilibrium (Franklin *et al*. (1994)); but this process can last very long.

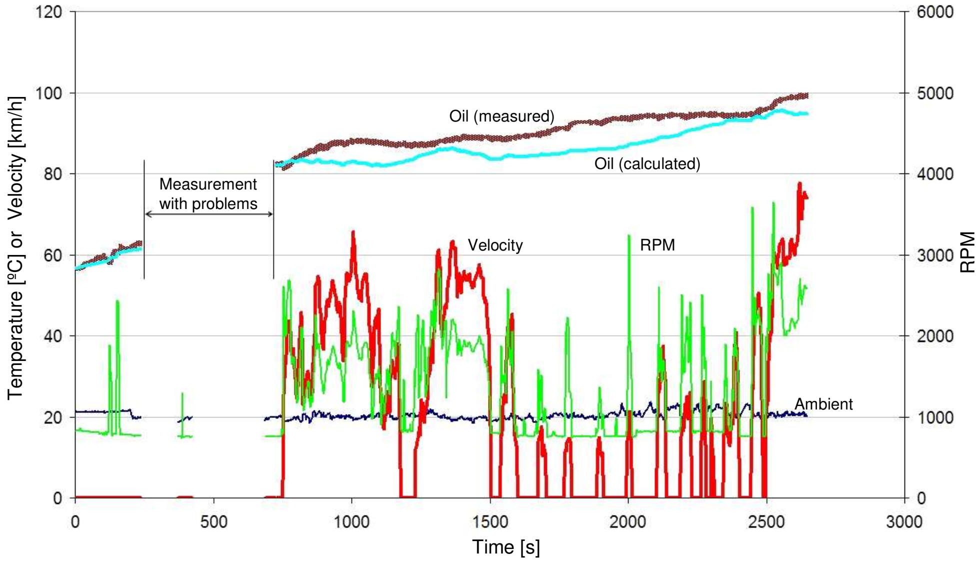

Figure 5: Test of the temperature model (urban traffic)

It was shown previously that an observable system could be proposed, so that a second state variable (engine speed) could be fed back to the (close loop) estimator, correcting an initial error in the temperature, as expressed by the equation (30).

$$\dot{\hat{x}} = A\hat{x} + Bu + K[Cx - C\hat{x}] \tag{30}$$

In this expression **x** is the actual state vector and $\hat{\mathbf{x}}$ contains its values estimated by the proposed model. For the system proposed in Eq. (22) and (23), the term [C x -C $\hat{x}$] is the difference between the real and estimated engine speed ([$\dot{\theta}$ - $\hat{\dot{\theta}}$]). An initial guess of the temperature above the real value would cause the calculation of a rotation above the measured value. The term [$\dot{\theta}$ - $\hat{\dot{\theta}}$] would be negative, causing the reduction of the temperature derivative and therefore tending to correct the initial error. The dependence of this type of observer on parameters as vehicle mass and road ramp (not all the time known) makes its use viable only for specific conditions, as the engine idle, with null vehicle velocity, and temperature sufficiently low, so that its temperature variation has an observable influence on the engine torque.

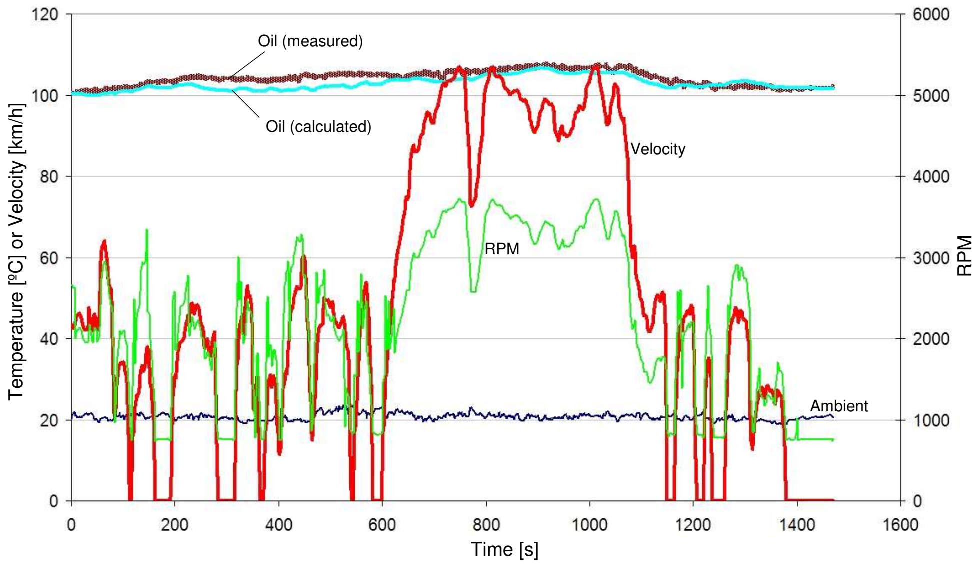

Figure 6: Test of the temperature model (urban traffic)

In addition, the estimation of the air temperature must be taken in consideration. Fortunately, this parameter can be easily determined using the sensor of the air intake temperature, already present in most engine management systems.

Regarding the calibration procedures in the chassis dynamometer, the main limitations are: (a) The determination of the heat transferred to the ambient air is carried out with engine off. That means that the velocity of the oil at the oil pan wall are zero and the heat transfer coefficient oil-wall is lower than in the real case (engine working and shaking the oil). The maximum value obtained was about 5% of the engine effective power, while the literature (see, for example Heywood (1989)) suggests about 10%. Therefore, the results obtained for Q_r, and, consequently, for Q_m are probably lower than the real. If this method is to be used for the energy balance of an engine (which could be an application for it), care must be taken to calibrate it in comparison with the results of similar engine in a test bench. Nevertheless, for calibration purposes the lower values of Q_r are compensated by lower values of Q_m in the Eq. (4) and Eq. (5). (b) To keep the engine coolant temperature low is relative difficult to accomplish in this kind of dynamometer especially for high RPM and loads. This kind of calibration (i.e. determination of Q_m with low engine temperatures) if necessary would have to count on special procedures, such as an improved vehicle radiator/ ventilator of the cooling system or even an additional (external) cooling system.

7. CONCLUSIONS

The proposed virtual sensor, even working as an open loop observer, is viable for use in control actions that do not require high precision measurements, as showed in the normal traffic condition. The system is stable and therefore the model predicts a stabilization value of the temperature, as it occurs in the car.

It is possible to build a close loop state observer using the proposed model for particular running conditions, since it was showed this system is observable. Different observers could work together in order to have an improvement in the determination of the initial temperature, which is one of the limitations of the model precision.

The model has limitations due to its assumptions (constant specific heat, constant oil mass, constant engine friction). The proposed calibration method showed to be viable for this purpose, but difficult to use, if the coolant temperature is to be kept low. Also, the measured values of the heat rejected by the oil to the ambient are probably sub-estimated. A confirmation of this fact by a comparison with a conventional measurement in an engine test bench would be necessary to validate the chassis dynamometer method to perform an engine energy balance.

Other steps are necessary to implement this function in an engine management unit. The work carried out until this paper is a base to the decision of the viability and for the preparation of the implementation phase.

The automotive industry in developing countries is under pressure to produce competitive products in terms of cost and quality. The development of software capabilities of the electronic control unit is an important tool to solve design problems, without additional variable costs. The present work has contributed with the proposal of a prediction model and test methods, but, above all, it contributed with the improvement of the competence to use such tool.

8. REFERENCES

Albertos, P.; Goodwin, G.C. “Virtual Sensors for Control Applications”. Annual Review in Control, v.26, s.n., p. 101-112, 2002. Available in: <http\\www.elsevier.com/locate/arcontrol>.

Alkidas, A. C. “Effect of Operational Parameters on Structural Temperature and Coolant Heat Rejection of a S.I. Engine”. In: Vehicle Thermal Management. Warrendale, USA: SAE, Inc., 1994. p. 203-218.

Franklin, G.F.; Powell, J. D.; Emami-Naeini, A. “Feedback Control of Dynamic Systems”. 3. ed. Reading, Massachusetts: Addison-Wesley Publishing Company, 1994. p. 35-70.

Friedland, B. “Control System Design: An Introduction to State-Space Methods”. 2. Printing. New York: McGraw-Hill, 1987. p.411-470.

Gruden, D.; Küper, P.F.; Porsche, F. “Heat Balance of Modern Passenger Car SI Engines”. In: SPALDING, D. B.; Afgan, N. H. Heat and Mass Transfer in Gasoline and Diesel Engines. New York: Hemisphere Publishing Corporation, 1989. p. 563-576.

Gustafsson, F.; Drevö, M.; Forssel, U.; Löfgren, M.; Persson, N.; Quicklund, H. “Virtual Sensors of Tire Pressure and Road Friction”. Warrendale, U.S.A. SAE Technical Paper Series 2001-01-0796.

Heywood, J.B. “Internal Combustion Engines Fundamentals”. 1. ed. Singapore: McGraw-Hill, 1989. p.668-707.

Kaplan, J. A.; Heywood, J. B. “Modeling the Spark Ignition Engine Warm-Up Process to Predict Component Temperatures and Hydrocarbon Emissions”. In: Vehicle Thermal Management. Warrendale, USA: SAE, Inc., 1994. p. 15-30.

Luenberger, D. G. “Introduction to Dynamic Systems: Theory, Models & Applications”. New York: John Wiley & Sons, 1979. p.254-328.

Monnerat Jr., P.; Gavioli, V.; Damasceno, F.; Bucci, A.; Poggio, L.; Gelmetti, A.; Ceccarini, D. Software Flex-Fuel Sensor (SFS) – “Logical sensor for the control of engine using a variable percentage of alcohol”. São Paulo. SAE Technical Paper Series 2000-01-3218PE.

Ogata, K. “Modern Control Engineering”. Portuguese edition. Rio de Janeiro: Prentice Hall do Brasil, 1998.

Özisik, M. N. "Heat Transfer: A Basic Approach". 1.ed. Singapore: McGraw- Hill, 1985. 780 p.

Phillips, C. L.; Harbor, R. D. "Feedback Control Systems". Upper Saddle River, New Jersey: Prentice Hall, 2000. p. 417-449.

Sebbesse, W. Steinberg, P.; Deussen, N.; Schlenz, D. „Engine Cooling". In: HUCHO, W.-H. Aerodynamics of Road Vehicles. 4. ed. USA: Society of Automotive Engineers, Inc, 1998. p.559-576.

Taborek, J.J.; "Mechanics of Vehicles". Cleveland, Ohio: Penton Education Division, 1957. p.31-41.

Trapy, J.D.; Damiral, P. "An Investigation of Lubricant System Warm-up for the Improvement of Cold Start Efficiency and Emissions of S.I. Automotive Engines". In: Vehicle Thermal Management. Warrendale, USA: SAE, Inc., 1994. p. 3-13.

Van Wylen, G.J.; Sonntag, R.E. "Fundamentals of Classical Thermodynamics". 2. Portuguese ed. São Paulo: Edgard Blücher, 1976. 565 p.

Weisstein, E., Bernoulli Differential Equation. Mathworld-Awolfram Web Resource. Available in: <http://mathworld.wolfram.com/ BernoulliDifferentialEquation.html> Access date: april 22nd, 2006.

9. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.