Energy Wall Losses Estimation of a Gasoline Engine Using a Sliding Mode Observer

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ABSTRACT

This paper describes an innovative method to estimate the energy wall losses during the compression and combustion strokes of a gasoline engine using the cylinder pressure measurement. The estimation during the compression and combustion strokes allows to better represent the system during the combustion. A sliding mode observer is derived from a 0-d physical engine model and its convergence and stability are proved. The complete system is validated by comparing simulated cylinder pressure with measurements.

INTRODUCTION

Through the years, the research of convective heat transfer modeling in internal combustion engines has been a hard task due to the difficult prediction of this phenomenon. Most of research has been focused on empirical laws which try to predict the heat transfer coefficient between the gases and the cylinder walls. One of the first laws was proposed by Annand [1], where an empirical law based on the thermal conductivity and the engine temperature is used to calculate the thermal coefficient of the walls heat transfer. Then, in the work of Woschni [2], a new empirical law is proposed, where the heat transfer coefficient depends on the pressure, temperature and engine speed. Woschni's law has been adopted for almost all the 0d engine models, and many works based on this law have been published, such as Hohenberg [3], Han and al.[4] and more recent Alizon [5]. Alternately, Shayler [6] takes into account the Woshni's correlation and adds a heat flow distribution depending on the surface of the chamber.

Parallel to those works, the *law of the wall* works have tried to explain the heat transfer phenomena: in Yang and Martin [7], the unsteadiness of turbulent heat transfer in piston engines depend on the flow properties, the gas conductivity and the viscosity. In Boust [8] a physical approach for wall heat transfer based on the kinetic theory of gases is proposed.

The instantaneous heat flow during the engine cycle is a necessary input for realistic cycle calculations [9], even if the existent empirical laws approximate the wall losses phenomenon, the current heat transfer models are not universally applicable and many tuning parameters remain undefined. This situation opens the perspective to the use of a different strategy than the empirical approximations.

The purpose of this work is to give an alternative method to estimate the heat wall losses. The heat wall transfer makes part of the whole enthalpy flow during the engine cycle. The whole enthalpy flow modeling requires the use of discontinuous terms to represent its dynamics. The use of an sliding mode observer is a suitable strategy for observation in these kind of systems; this technique is based on the choice of a sliding surface of the state space according to the desired dynamical specifications of the closed-loop system. The sliding choices are designed so that the state trajectories reach the surface and remain [10]. The sliding mode technique has been used by many researchers for estimation of non measurable and/or uncertain parameters in space state system. The sliding model method has been popularized thanks to the work of Utkin [11]. The main advantages of this method are its robustness against a large class of perturbations or model uncertainties, the need of a reduced amount of information and the possibility of stabilizing some nonlinear system which are not stabilizable by continuous state feedback laws.

In this work, an innovative method to estimate the wall losses during the compression and combustion stroke of a gasoline engine is proposed. To develop the observer, a 0d one zone thermodynamical model of a gasoline engine has been developed.

In the first part of this paper the one zone thermodynamical model and the combustion model are described. The result of this model is validated with cylinder pressure measurements. This model is reduced to design the sliding model observer. In the next sections, the complete observer system is developed and the estimation is compared to the cylinder pressure and the wall losses empirical model. As a result, the sliding mode strategy allows to well predict the enthalpy flow due to the gases interaction with the cylinder wall during the compression and combustion strokes using the cylinder pressure measurement.

0D ENGINE MODEL

The thermodynamical model describes the energy balance inside the combustion chamber. The 0d virtual engine model is divided into two main elements: a one zone thermodynamical model that describes the energy balance inside the combustion chamber and the combustion model.

ONE ZONE THERMODYNAMICAL MODEL

In the one zone thermodynamical model, the combustion chamber is considered as a unique open system and a uniform in-cylinder pressure is assumed. The mass flow rate in the cylinder is deduced from a balance equation corresponding to the mass transfer through intake and exhaust valves.

The energy equation for the cylinder is inferred from the first thermodynamical principle:

$$dU = -\delta Q_{th} - pdV + \Big(\sum_j h_j dm_j\Big) \tag{1}$$

where the subscript j denotes energy getting into or out of the combustion chamber, U is the internal energy of the cylinder gas mixture, δQ_{th} expresses the heat transfer of the cylinder contents to the surroundings, pdV corresponds to the work delivered by the piston, $\sum_j h_j dm_j$ is the total energy flowing into or out of the cylinder and h is the specific enthalpy.

Assuming that the specific heat constant c_v is constant, the left hand side of (1) can be written as:

$$dU = T(t)c_v dm(t) + m(t)c_v dT(t) \tag{2}$$

where $m(t)$ is the total mass of all the species in the cylinder and $T(t)$ corresponds to the temperature of the gases.

Solving (1) and (2) for $dT(t)$, an ordinary differential equation is implemented as the governing equation for the system temperature dynamics:

$$\begin{aligned} dT(t) =& \frac{1}{m(t)c_v}\Bigg(- p(t)dV(t) - \delta Q_{th}(t) \\ &+ \Bigg(\sum_j h_j(t)dm_j(t)\Bigg) - T(t)c_v dm(t)\Bigg) \end{aligned} \tag{3}$$

where $V(t)$ is the gases volume. The heat losses from the gases in the combustion chamber to the cylinder walls are given by:

$$\delta Q_{th}(t) = h_c(t)A_w(t)(T(t) - T_w) \tag{4}$$

where $A_w(t)$ is the wall transfer area, $T(t) - T_w(t)$ is the temperature difference between the gases and the cylinder walls, and h_c is the heat transfer coefficient computed from Woschni's empirical law [2]:

$$\begin{aligned} h_c(t) =& \alpha D^{-0.2} p(t)^{0.8} T(t)^{-0.53}\Big(C_1 V_p + \\ & C_2 \frac{V_s T_1}{p_1 V_1}(p(t) - p_0(t))\Big) \end{aligned} \tag{5}$$

where D is the cylinder bore, C_1 and C_2 are calibration constants, p_1 and T_1 represent the known state of the working gas related to the instantaneous cylinder volume V_1, i.e. at IVC, and p_0 is the pressure reference in the absences of combustion.

The total energy flowing into or out of the cylinder is considered as the enthalpy flow from the breathing process and the combustion:

$$\sum_j h_j(t)dm_j(t) = Q_{h_{in}}(t) - Q_{h_{out}}(t) + Q_{h_{comb}}(t) \tag{6}$$

To find the dynamics of $p(t)$, the ideal gases law is used:

$$p(t)V(t) = rm(t)T(t) \tag{7}$$

Taking the derivative of (7) and solving for $dp(t)$:

$$dp(t) = \frac{rT(t)dm(t)}{V(t)} + \frac{rm(t)dT(t)}{V(t)} - \frac{rT(t)m(t)dV(t)}{V(t)} \quad (8)$$

where $dm(t) = dm_{in} - dm_{out}$ is the mass balance through the cylinder valves. Finally, putting together equations (3), (6) and (8) and solving for $dp(t)$, the in-cylinder pressure dynamics is modeled as:

$$\begin{aligned} dp(t) =& \frac{r}{V(t)c_v} Q_{h_{in}}(t) - \frac{r}{V(t)c_v} Q_{h_{out}}(t) \\ &- \frac{r}{V(t)c_v} p(t)dV(t) - \frac{p(t)dV(t)}{V(t)} \\ &+ \frac{r}{V(t)c_v} Q_{h_{inj}}(t) + \frac{r}{V(t)c_v} LHV Q_{m_{comb}}(t) \\ &- \frac{r}{V(t)c_v} Q_{th}(t) \end{aligned} \quad (9)$$

where $Q_{h_{comb}}(t)$ has been replaced by $Q_{h_{inj}} + LHVQ_{m_{comb}}(t)$, the sum of the enthalpy supplied by the injection and the combustion process. LHV is the lower heat value: for gasoline engines it can be approximated to $4.15x10^7 MJkg^{-1}$. $Q_{h_{inj}}$ supplies a small amount of energy during few time (less than 3 CAD), in this model, its value is obtained from a map.

COMBUSTION MODEL

The combustion process is commonly defined with a burned mass fraction curve, provided by a Wiebe's law [12]:

$$\begin{aligned} Q_{m_{comb}}(\theta, u) =& m_o a e^{-a\left(\frac{\theta}{\Delta_\theta}\right)^{(m+1)}} \\ &\cdot (m+1)\left(\frac{\theta}{\Delta_\theta}\right)^m \frac{2N\pi}{60\Delta_\theta} \end{aligned} \quad (10)$$

where N is the engine speed in rev/s, m_o is the injected fuel and a, m and Δ_θ are calibration parameters.

ENGINE MODEL VALIDATION

The engine model is tested taking as reference the measurements of a 1.2 liters engine. The data to fit the model is the cylinder pressure. The results presented in this paper correspond to a test performed at $N = 2000\,rpm$ and BMEP $= 10\,bar$.

Results of the validated model are shown in Figure 1. Complementary results are shown in Figure 2, where the percentage of error between the measurements

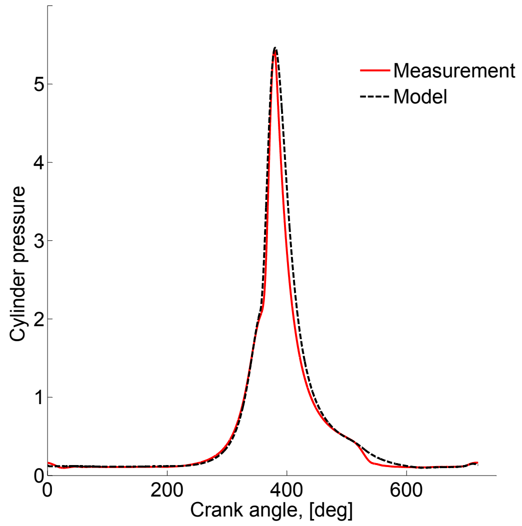


Figure 1: Cylinder pressure. BMEP=10 bar, N=2000rpm.

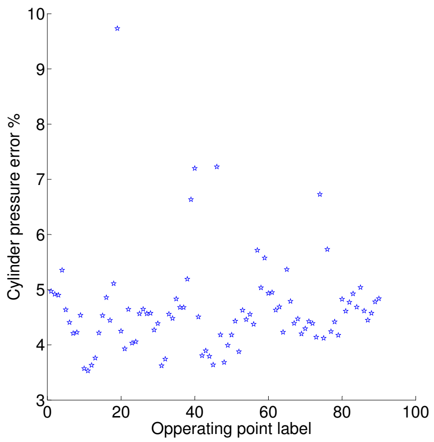


Figure 2: Cylinder pressure error for 90 operating points. BMEP=1-20 bar, N=1200-5500rpm.

and the model carried out in a data base of 90 operating points are shown. The model has less than 10% of error in the cylinder pressure prediction. Those results are accurate enough for the purpose of this work.

MODEL REDUCTION

The cylinder pressure dynamics depends on the energy balance in the combustion chamber. The enthalpy flows represent a different physical phenomenon depending on the the crank angle position (engine stroke). Figure 3 shows how are given the energy exchanges on the combustion chamber depending on the combustion stroke. In the figure:

- Inlet valve opening (IVO) - Inlet valve closure (IVC): Admission stroke

- Exhaust valve opening (EVO) - Exhaust valve closure (EVC): Exhaust stroke
- IVC - IT where IT is the ignition timing: Compression stroke
- IT-EVO: Combustion

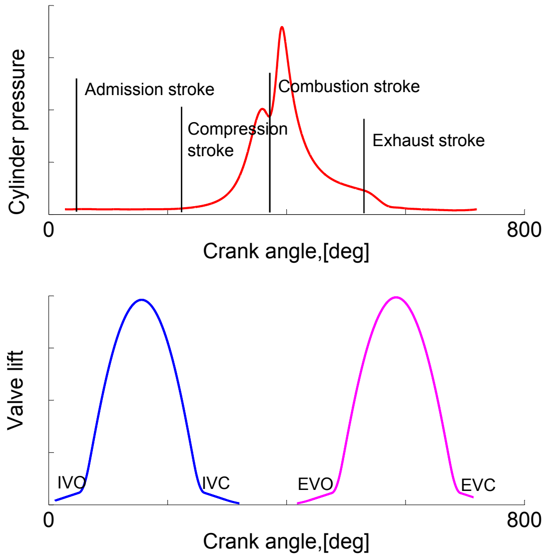


Figure 3: Engine cycles representation

As it was presented in the last sections, four main enthalpy flows are considered in the system: The enthalpy due to the valves flows $Q_{h_{in/out}}(t)$, the enthalpy due to combustion $Q_{h_{comb}}(t)$ and the enthalpy due to the wall losses $Q_{th}(t)$, the remaining energy component is the work delivered by the piston and the gases moving which depend on the cylinder pressure. Table 1 explains how the enthalpy dynamics are taken into account depending on the engine stroke (from here the time dependence notation is omitted to simplify the notations).

Admission: $Q_{h_{comb}} = 0$, $(Q_{h_{in}},\ Q_{h_{out}},\ \delta Q_{th}) \neq 0$
Exhaust: $Q_{h_{comb}} = 0$, $(Q_{h_{in}},\ Q_{h_{out}},\ \delta Q_{th}) \neq 0$
Compression: $(Q_{h_{in}},\ Q_{h_{out}},\ Q_{h_{comb}}) = 0$, $\delta Q_{th} \neq 0$
Combustion: $Q_{h_{in}} = 0$,$Q_{h_{out}} = 0$, $\delta Q_{th} \neq 0$

Table 1: Engine strokes and enthalpy flows

A transformed two states system is designed. In this system, two states are presented: The cylinder pressure $x_1 = p$ and the enthalpy flow $x_2 = Q_{h_{in}} + Q_{h_{out}} + Q_{th}$:

$$\begin{aligned} \dot{x}_1 =& -\left(\frac{r}{c_v}+1\right)\frac{dV}{V}x_1 + \frac{r}{c_v V}Q_{h_{comb}} \\ &+ \frac{r}{c_v V}x_2 \\ \dot{x}_2 =& 0 \end{aligned} \tag{11}$$

The state x_2 groups all the enthalpy flows in the system, different from the heat supplied by the combustion and the work delivered by the piston. The dynamics of x_2 is assumed to be unknown. Whit this model, during the compression and combustion strokes, the state x_2 represents only the wall losses, according to Table 1.

SLIDING MODE OBSERVER

Consider a system in additive triangular nonlinearity form:

$$\begin{aligned} \dot{z} &= A_0 x + \phi(z,u) \\ y &= C_0 z \end{aligned} \tag{12}$$

where

$$\begin{aligned} A_0 &= \begin{bmatrix} 0 & 1 & 0 \\ 0 & \ldots & 1 \\ 0 & 0 & 0 \end{bmatrix} \\ C_0 &= \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \end{aligned} \tag{13}$$

The nonlinear observer is designed according to the following theorem:

Theorem 1 [13]. If $\phi(z,u)$ is globally Lipschitz in z and u and such that $\partial\phi_i(z,u)/\partial z_j = 0$, for $j \geq i+1$, $1 \leq i,j \leq n$, then the system (12) admits an observer of the form [10]:

$$\begin{aligned} \dot{\hat{z}}_1 &= \hat{z}_2 + \phi_1(z_1,u) + \lambda_1 sgn(z_1 - \hat{z}_1) \\ \dot{\hat{z}}_2 &= \hat{z}_3 + \phi_2(z_1,\tilde{z}_2,u) + \lambda_2 sgn(\tilde{z}_2 - \hat{z}_2) \\ \dot{\hat{z}}_n &= f(z_1,\tilde{z}_2..,\tilde{z}_n) + \lambda_n sgn(\tilde{z}_n - \hat{z}_n) \end{aligned} \tag{14}$$

where

$$\tilde{z} = \hat{z} + \lambda sign(z - \hat{z}) \tag{15}$$

To analyze the observer stability, the first state space variable is analyzed:

Considering $e_1 = z_1 - \hat{z}_1 \neq 0$ and the Lyapunov function $V = \frac{1}{2}e_1^2$. To guaranty the stability, the condition $\dot{V}_1(x) < 0$ must be full filled. Considering the dynamics of e_1:

$$\dot{e}_1 = e_2 - \lambda_1 sgn(e_1) \tag{16}$$

and

$$\dot{V} = e_1\dot{e}_1 = e_1 e_2 - \lambda_1 sgn(e1) \tag{17}$$

which verifies $\dot{V} < 0$ when $\lambda > |e_2|_{max}$. As the function sgn is used, and the Lyapunov function decreases, the convergence to the sliding surface $S = e_1$ in a finite time t_0 is obtained. Thus, for $\lambda_1 > |e_2|_{max}$, $\hat{z}_1$ converges to z_1 in finite time t_0 and remains equal to z_1 for $t > t_0$.

Moreover, $\dot{e}_1 = 0$ for $t > t_0$, so from (16):

$$e_2 = \lambda_1 sgn(e_1) \tag{18}$$

and

$$\tilde{z}_2 = \hat{z}_2 + \lambda_1 sgn(e_1) \tag{19}$$

is equal to z_2 for $t > t_0$.

The same procedure if followed for the states $z_2, ..., z_n$ and the stability is shown. Refer to [10] for the complete demonstration.

SLIDING MODE OBSERVER APPLICATION

Using the sliding mode observer (14) in system (11) it yields:

$$\begin{aligned}
\dot{\hat{x}}_1 =& -\left(\frac{r}{c_v}+1\right)\frac{dV}{V}\hat{x}_1 + \frac{r}{c_v V}Q_{h_{comb}} \\
&+ \frac{r}{c_v V}\hat{x}_2 + \frac{r}{c_v V}\lambda_1 sgn(x_1 - \hat{x}_1) \\
\dot{\hat{x}}_2 =& \frac{r}{c_v V}\lambda_2 sgn(\tilde{x}_2 - \hat{x}_2) \\
\tilde{x}_2 =& \hat{x}_2 + \frac{r}{c_v V}\lambda_1 sgn(x_1 - \hat{x}_1)
\end{aligned} \tag{20}$$

Where λ_1 and λ_2 are the observer gains to be chosen to ensure the system stability. Differently from (12), in System (20) the first non zero component in matrix A_0 is different than 1. It is possible to use an equivalent transformation through a diphemorfism to obtain the exact form, however, in this work the effect of this component has been added to the observer inputs multiplying it by $r/(c_v V)$.

The Lyapunov stability theorem [14] is used to bound the choice of λ_1 and λ_1. In a first place, the bounds for λ_1 are chosen:

Consider the Lyapunov function $V_1 = \frac{1}{2}e_1^2$, where $e_1 = x_1 - \hat{x}_1$. The condition $\dot{V}_1(x) < 0$ must be full filled:

$$\dot{V}_1(x) = e_1(x)\dot{e}_1(x) \tag{21}$$

$$\dot{V}_1(x) = e_1\Big(\frac{r}{c_v V}e_2 - \frac{r}{c_v V}\lambda_1 sign(e_1)\Big) \tag{22}$$

$r/(c_v V) > 0$, then keeping $\lambda_1 > |e_2|$ ensures $\dot{V}_1(x) < 0$. The same procedure is applied to chose the second parameter λ_2.

Consider the second Lyapunov function $V_2 = \frac{1}{2}e_2^2$, where

$$e_2 = x_2 - \hat{x}_2 \tag{23}$$

The conditions $\dot{V}_2(x) < 0$ must be fulfilled:

$$\dot{V}_2(x) < 0 = e_2(x)\dot{e}_2(x) \tag{24}$$

$$\dot{V}_2(x) < 0 = e = -e_2\frac{r}{c_v V}\lambda_2 sign(e_2) \tag{25}$$

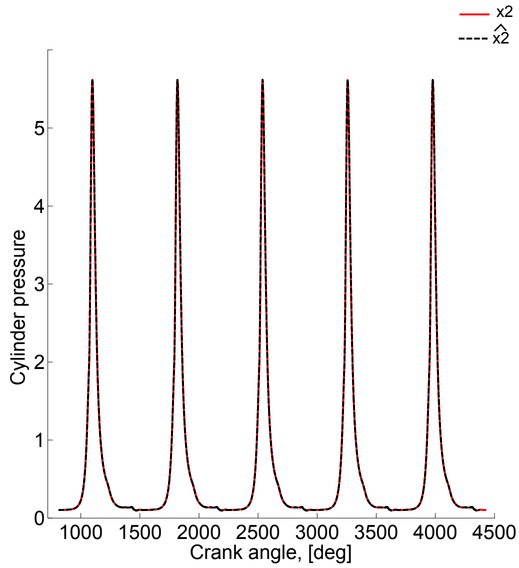
Keeping $\lambda_2 > 0$ ensures $\dot{V}_2(x) < 0$. The results of the implemented observer are presented next.

OBSERVER RESULTS

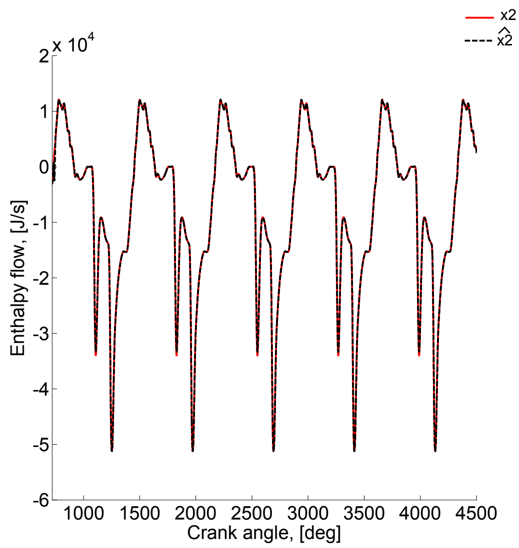
Figures 4a and 4b show the observation results of the system in equations (20), compared to the model in equations (4) and (9). For the cylinder pressure (x_1), the initial condition is the measured pressure at IVC, for the enthalpy flows, the initial condition is taken as $-0.4J/s$. The observer is efficient and effective in successive engine cycles.

Using the measured pressure $p = x_1$, the observer is able to estimate the second state x_2 that represents the enthalpy flow. When the observation is made overall the whole engine cycle, the estimated enthalpy flow represents different physical phenomena depending on the engine stroke, as it is presented in Table 1. Taking the portion corresponding to the compression and combustion strokes of one engine cycle from Figures 4a and 4b, Figure 5 is obtained. In this figure, the estimation during the admission stroke represents the enthalpy flow due to the valves flows and the wall losses, similarly during the exhaust stroke.

The remain portion of the engine cycle is extracted in Figure 6, which corresponds to the compression and the combustion strokes. In this stage of the engine cycle, besides the enthalpy flow supplied by the combustion which is represented in the model by the Wiebe's law and the work delivered by the piston, the only remaining enthalpy flow is the heat transfer to the walls. Then, the enthalpy flow shown in Figure 6b corresponds to the heat wall losses.

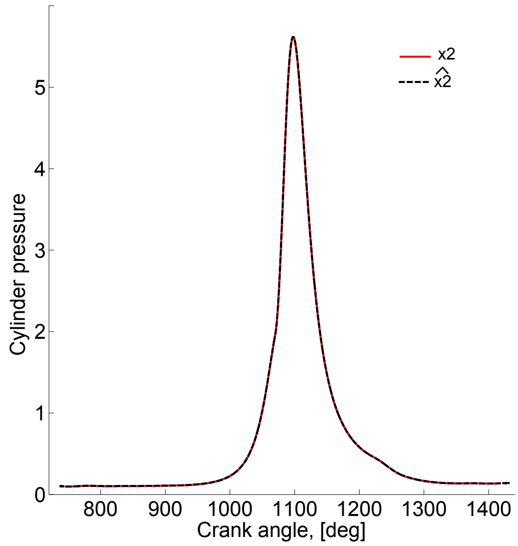


(a) Cylinder pressure estimation

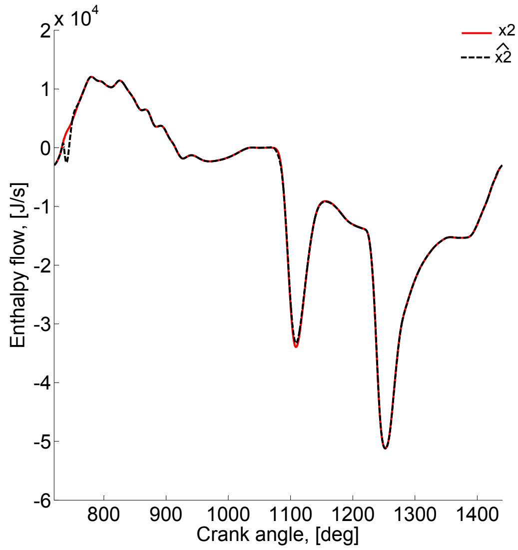


(b) Enthalpy flow estimation

Figure 4: Cylinder pressure and heat flow estimation 5 engine cycles. BMEP=10 bar, N=1200rpm.



(a) Cylinder pressure estimation



(b) Enthalpy flow estimation

Figure 5: Cylinder pressure and heat flow estimation 1 engine cycle. BMEP=10 bar, N=1200rpm.

Figures 7a and 7b show the normalized observation error for variables $\hat{x}_1$ and $\hat{x}_2$ corresponding to the cylinder pressure and the enthalpy flow for a data base of 90 operating points. The results are satisfactory as the cylinder pressure error observation remains below 0.05% while the enthalpy flow observation error remains below 3%.

CONCLUSIONS

A sliding mode observer to estimate the heat wall losses during the compression and combustion strokes has been implemented. The observer is able to estimate the wall losses even if its dynamics is unknown for the system.

The model has been validated using the validated 0d model of a park ignited engine against experimental measurements. The observer performs accurately in a large data base of operating points.

Using a sliding mode observer allows to have a simple design and implementation structure, which is robust against modeling error and perturbations due to parametric variations [15].

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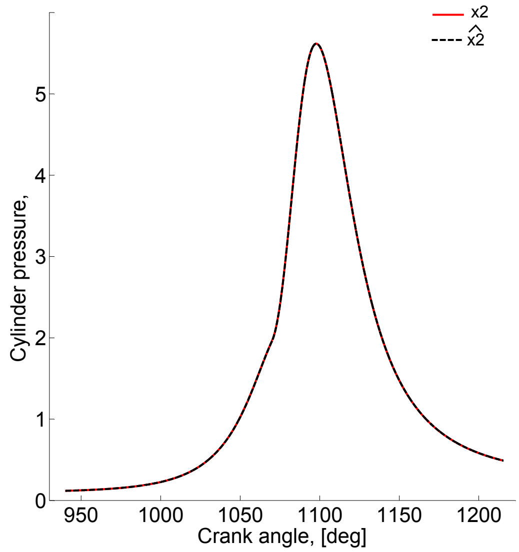
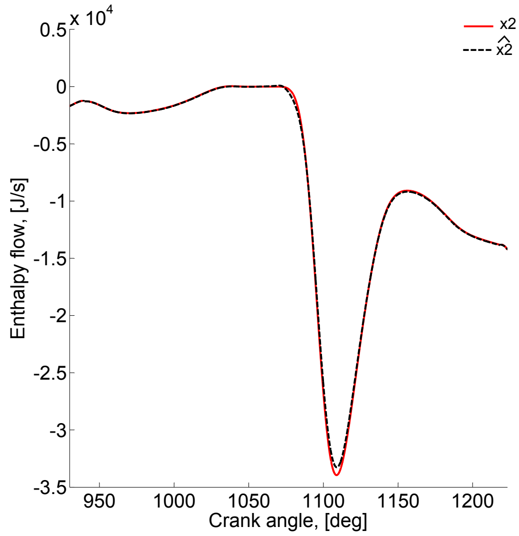
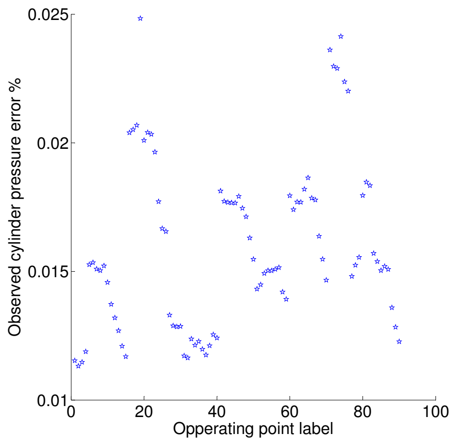


(a) Cylinder pressure estimation

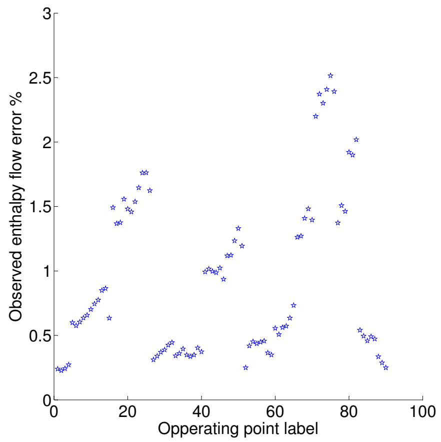


(b) Enthalpy flow estimation during compression and combustion strokes

Figure 6: Cylinder pressure estimation and heat flow estimation during compression and combustion strokes. BMEP=10 bar, N=1200rpm.



(a) Cylinder pressure estimation error



(b) Enthalpy flow estimation error

Figure 7: Cylinder pressure and enthalpy flow observation errors for 90 operating points. BMEP=1-20 bar, N=1200-5500rpm

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NOMENCLATURE

All variables are in S.I Metric Units.

α Calibration constant
ω_e Engine speed (rad/s)
IT Ignition timing
A_w Heat transfer wall area
CAD Crank angle degrees
c_v Specific heat at constant volume
C_1 Calibration constant
C_2 Calibration constant
D Cylinder bore
EVC Exhaust Valve Closure
EVO Exhaust Valve Opening
h Enthalpy
h_c Heat transfer coefficient for wall looses
H_p Piston height
IVC Inlet Valve Closure
IVO Inlet Valve Opening
k_0 Calibration constant
k_1 Calibration constant
N Engine speed (rpm)
m Total mass in the combustion chamber
p Pressure
r Specific gases constant
T Temperature
T_w Wall temperature
U Energy
V Cylinder volume