Proceedings of the 2nd International Conference on Systems and Control, Marrakech, Morocco, June 20-22, 2012

Diagnosis-Oriented Modeling for Marine Diesel Engine Injection System

Yassine Khelil, Guillaume Graton, Mohand Djeziri, Mustapha Ouladsine, Rachid Outbib
Laboratoire des Sciences de l'Information et des Systèmes
UMR CNRS 7296, Domaine Universitaire de Saint-Jérôme
Avenue Escadrille Normandie-Niemen, 13397 Marseille Cedex 20
yassine.khelil@lsis.org.

***Abstract*— This paper aims to highlight existing relationships between faults and symptoms on marine Diesel engines. These causal relationships are usually described by maintenance experts and characterized by uncertainties and fuzziness. In order to cope with these drawbacks an analytical model is built, putting in evidence physical parameters subjected to faults. This procedure is applied to the injection system that is equipped with an in-line injection pump and standard sensors. After defining the most frequent and hazardous faults on the system, sensitive parameters to these faults are identified. Then, variations are introduced on each parameter to simulate the faults and sensor outputs are monitored. The observed changes caused by each fault are used to build causal relationships. A validation of these relationships is made by introducing real faults on the six-cylinder marine Diesel under study. The results show good concordances between simulation and real results.**

I. INTRODUCTION

In control engineering, a fault detection and isolation (FDI) function is compounded by two major tasks: the fault detection task and the isolation task. Whatever is the nature of the fault detection task (model-based [1] [2], knowledge-based [3] [4], empirical or signal-based [1] [5]), features must be checked if they belong to normal operating spaces. In last decades and concerning the detection issues on Diesel engines, many papers tackling this problem are published. The mentioned references are some application examples treating fault detection issues upon engine subsystems.

As it is the detection task, the isolation task in Diesel engines is not an easy task. The traditional approach consisting of finding the measurements subject to faults is not sufficient. In fact, if for example, one of the temperatures raises, it is important to know it. But, it is more important to know the cause of this raise (fault). Concerning Diesel engines, solutions are proposed to resolve this issue and among them two major orientations can be listed: knowledge-based approaches and model-based approaches.

Knowledge-based approaches use maintenance expert knowledge that is formulated, for example in FMECA, HAZOP ... or use analytical knowledge, usually transformed to qualitative methods. These knowledge-based approaches can regroup some studies. In [6], a diagnosis qualitative model of the common rail injection system is proposed. In the same context, rules related to faults in the air path system and the combustion chamber are transformed to a fuzzy model in [7]. In [1], a model-based approach combining physical equation with neural networks and knowledge symptom-fault table, for fault diagnosis in the air path and injection systems is presented. In [8], a failure tree is used to establish causal rules able to isolate faults on the air path system.

The published contributions concerning the model-based approaches are more widespread. In fact these approaches can be divided in two sets. The data-driven models where known faults are introduced and a classification is applied using pattern recognition methods. This subset of methods can be illustrated with [3] where isolation between some faults on the injection system is realized after feature extraction. Neural networks are used in [9] to distinguish between normal and faulty operating conditions of an air inlet valve. In [10], SVM-classifier is trained to differentiate a normal state from air and exhaust gas leaks and faulty injectors.

The second subset contains analytical models where usually few faults and components, often very instrumented, are monitored. Some examples can be given, like in [11], faults on a common rail injection system are studied, in [12], adaptive fixed and variable gain observers are used for air leaking diagnosis in the air path system. In [2], an adaptive and extended Kalman filter are implemented to diagnose air leaking and EGR problems. Many other studies are carried out and the given list of publications is not exhaustive.

Our contribution consists in highlighting the likely relationships that exist between faults and symptoms that can be observed on standard installed sensors. The goal of this approach is to take advantage from the simplicity of knowledge-based approaches and to cope with theirs drawbacks. Notice that, the model-based approaches (data-driven and analytical models) cannot be used since no data for classification are available and no specific instrumentation is available to perform analytical isolation. To this end, a diagnosis-oriented model of a marine Diesel engine is built. This paper presents the part related to the injection system. Hence, in section II, a description of the entire engine, under study, is given and the injection system is detailed. A model is chosen from the literature in a way to meet some specifications and is presented in section III. In section IV, the causal relationships for non-combined faults are established and an experimental validation is also given. Section V is dedicated to the conclusion and perspectives.

II. DIESEL ENGINE DESCRIPTION

Diesel engines are mainly compounded of six subsystems. The combustion system hosts the fuel combustion reaction and transforms the liberated energy into a useful power.

This work is sponsored by CG13, CG83, CG06, CR PACA, TPM, FUI and MPM within the framework of the BMCI project (Bilan de santé pour la Maintenance et la Conduite Intelligente).

It contains mechanical elements like the engine block, the crankshaft, camshafts, gears, pistons, connecting rods ... The cooling system maintains the engine compartment and its elements under an acceptable temperature. It is compounded of one or many pumps, exchangers ... The lubrication system has a double role, the lubrication of the moving elements to reduce the friction and a cooling function at the same time. It is mainly made up of one or many pumps, exchangers like the cooling system. The air path system manages the circulation of the admitted air and the exhaust gas. It includes an air filter, a turbocharger, manifolds and an air cooler. Diversified type of start systems can be found (hydraulic, pneumatic or electrical) depending on the power of the engine. The injection system is designed to supply fuel under a desired pressure into cylinders at specific moments. It is mainly made up of a lift pump assuring the suction of fuel from the tank and the minimum pressure necessary for the proper operating of the injection pump. Moreover, the fuel is filtered before reaching the injection pump. The injection pump supplies the fuel to the injectors under a high pressure and specifies the time and the quantity to be injected. The excess of fuel returns to the tank. The figure 1 gives a global illustration of the considered injection system.

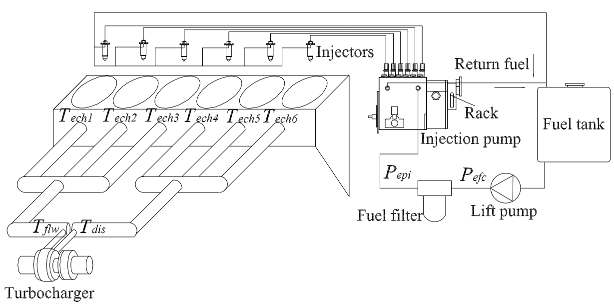

Fig. 1. Scheme of the injection and exhaust systems

The health of the injection system is of crucial importance. Troubles on it are equivalent to an increase in pollution, a reduced fuel economy and may engender other hazardous faults in the engine. The studied engine is a *Baudouin 6M26SRP1*™direct injection six-cylinder Diesel engine equipped with an in-line pump and a twin-inlet turbocharger. To monitor health of this system, a list of standard sensors are implemented:

(S1) engine speed (ω)
(S2) injection pump rack position (x_{rack})
(S3) engine power (Po_{eng})
(S4) fuel filter inlet pressure (P_{efc})
(S5) fuel injection pump inlet pressure (P_{epi})
(S6) cylinders outlet temperature ($T_{ech1},\dots,T_{ech6}$)
(S7) exhaust gas temperature at the the turbocharger inlet situated at the flywheel side (T_{flw})
(S8) exhaust gas temperature at the turbocharger inlet situated at the gear distribution side (T_{dis})

III. Injection system model

This section aims to build a model of the injection system able to estimate measurements provided by the sensors and offering the possibility of simulating faults. The purpose is to predict measurement behavior in fault presence. Hence, the causal relationships are established. At the exception of the engine speed ω and the rack position x_{rack} that are considered as inputs, all the other measurements are estimated.

A. Model

The relationship between the power of the engine Po_{eng} and the injected fuel flow $\dot{m}_f$ is given by (1) as formulated in [13], [14]:

$$Po_{eng} - Po_{fric,org} = \eta_e \dot{m}_f LHV \tag{1}$$

where $Po_{fric,org}$ corresponds to the power lost in friction and within the auxiliary elements (like pumps, alternator...), LHV the fuel lower heating value. The overall efficiency η_e is expressed by an empirical expression given in [13]:

$$\begin{aligned}\eta_e = \; & \lambda(d_{c1} + d_{c2}\lambda + d_{c3}\lambda^2 + d_{c4}\lambda\omega + d_{c5}\lambda^2\omega \\ & + d_{c6}\lambda\omega^2 + d_{c7}\lambda^2\omega^2)\end{aligned} \tag{2}$$

where $\lambda = \frac{\dot{m}_{ei}}{\dot{m}_f}$ is the air/fuel ratio and $d_{c1},\dots,d_{c7}$ are constants to be identified. The injected fuel flow is approximated by the following formula taken from [15]:

$$\dot{m}_f = \omega(b_{c1} + b_{c2}x_{rack} + b_{c3}x_{rack}^2) \tag{3}$$

where b_{c1}, b_{c2}, b_{c3} are constants to be identified.

The air flow admitted to cylinders is estimated using the expression (4) given by [13], [14]:

$$\dot{m}_{ei} = \eta_v \dot{m}_{ei,th} \tag{4}$$

where $\dot{m}_{ei,th}$ is the theoretic admitted air flow, η_v the volumetric efficiency and their expressions are given respectively in (5) and (6),

$$\eta_v = a_{c1} + a_{c2}\omega + a_{c3}\omega^2 \tag{5}$$

where a_{c1}, a_{c2}, a_{c3} are constants to be identified,

$$\dot{m}_{ei,th} = n_{cyl}\frac{V_{cyl}\,p_a}{rT_a}\frac{\omega}{4\pi} \tag{6}$$

where p_a and T_a are respectively the pressure and the temperature in the admission manifold, n_{cyl} the number of cylinders, V_{cyl} the displacement volume of a cylinder and r the specific air constant.

By introducing the notion of mechanical efficiency η_m, the lost power $Po_{fric,org}$ is expressed by means of the produced power Po_{eng} as follows [14]:

$$Po_{eng} - Po_{fric,org} = \eta_m Po_{eng} \tag{7}$$

Assuming that η_m is constant, the expression of the engine power becomes:

$$\begin{aligned}Po_{eng} = \; & \tfrac{1}{\eta_m}\dot{m}_{ei}LHV(d_{c1} + d_{c2}\lambda + d_{c3}\lambda^2 + d_{c4}\lambda\omega \\ & + d_{c5}\lambda^2\omega + d_{c6}\lambda\omega^2 + d_{c7}\lambda^2\omega^2)\end{aligned} \tag{8}$$

Using equations (3), (4), (5) and (6), λ becomes:

$$\lambda = \frac{\dot{m}_{ei}}{\dot{m}_f} = \frac{n_{cyl}V_{cyl}\,p_a(a_{c1} + a_{c2}\omega + a_{c3}\omega^2)}{4rT_a\pi(b_{c13} + b_{c23}x_{rack} + b_{c33}x_{rack}^2)} \tag{9}$$

The temperatures of the exhaust gas at the cylinder outlet and at the turbocharger inlets are estimated using the formula (10) reported by [13]:

$$T_\alpha = T_a + \frac{a_{c4\alpha} + a_{c5\alpha}\lambda + a_{c6\alpha}\lambda^2}{1.2 + \frac{\lambda}{15}} + \frac{a_{c7\alpha}}{\omega} + a_{c8\alpha} \tag{10}$$

where the subscript α takes one of following values: $\{ech1, ech2, ech3, ech4, ech5, ech6, flw, dis\}$. Hence, it allows to write the corresponding variables $\{T_{ech1}, T_{ech2}, T_{ech3}, T_{ech4}, T_{ech5}, T_{ech6}, T_{flw}, T_{dis}\}$. $a_{c4\alpha}$, $a_{c5\alpha}$, $a_{c6\alpha}$, $a_{c7\alpha}$, $a_{c8\alpha}$ are constants to be identified.

It should be underlined that the estimation of the exhaust gas temperatures at the cylinder outlet are done by considering that the same quantity of fuel is injected into each cylinder ($\dot{m}_{fi} = \dot{m}_f / n_{cyl}$). As the two inlets of the turbine (turbocharger) collect the gas from respectively the two sets of cylinders ($set_1 = \{1,2,3\}$ and $set_2 = \{4,5,6\}$), as it can be seen in Fig. 1, the temperatures at these places are estimated by summation of the individual fuel quantities injected in the corresponding cylinders. Notice that the same reasoning is also applied to the admitted air flow $\dot{m}_{ei}$.

The hydraulic heads of the injection system are modeled as a succession of cylindrical pipes. The estimation of the differential pressure at the fuel filter is made using the equation (11) taken from [16]:

$$P_{dfc} = 32 f \frac{L_{c1}}{\pi^2 \rho_c D_{c1}^5} D_{carb}^2 \tag{11a}$$

$$f = \begin{cases} \frac{16}{Re} & \text{laminar flow } (Re < 2000) \\ 0.079 Re^{-0.25} & \text{turbulent flow } (Re > 2000) \end{cases} \tag{11b}$$

$$Re = \frac{D_{carb}}{\frac{\pi}{4}\mu_c D_{c1}} \tag{11c}$$

where f corresponds to the friction factor, L_{c1} and D_{c1} the length and the diameter of the fictive cylindrical pipe (representing the filter), ρ_c the fuel density, μ_c the dynamic viscosity and Re the Reynolds number.

Assuming that fuel flowing among the filter is proportional to the injected fuel flow ($D_{carb} = k\dot{m}_f$). Equation (11) becomes:

$$P_{dfc} = \begin{cases} C_{c1}(1 + \frac{b_{c21}}{b_{c11}} x_{rack} + \frac{b_{c31}}{b_{c11}} x_{rack}^2)\omega \\ (\text{if } Re < 2000) \\ \\ C'_{c1}((1 + \frac{b_{c21}}{b_{c11}} x_{rack} + \frac{b_{c31}}{b_{c11}} x_{rack}^2)\omega)^{1.75} \\ (\text{if } Re > 2000) \end{cases} \tag{12}$$

where $C_{c1} = 128 \frac{\mu_c b_{c11} L_{c1}}{\pi \rho_c D_{c1}^4}$, $C'_{c1} = 1.7876 \frac{\mu_c^{0.25} b_{c11}^{1.75} L_{c1}}{\pi^{1.75} \rho_c D_{c1}^{4.75}}$. Notice that the probable dependency of the dynamic viscosity and the density with the temperature is neglected.

In a similar manner, the pressure at the inlet of the injection pump is estimated according to:

$$P_{epi} - P_{ref} = \begin{cases} C_{c2}(1 + \frac{b_{c22}}{b_{c12}} x_{rack} + \frac{b_{c32}}{b_{c12}} x_{rack}^2)\omega \\ (\text{if } Re < 2000) \\ \\ C'_{c2}((1 + \frac{b_{c22}}{b_{c12}} x_{rack} + \frac{b_{c32}}{b_{c12}} x_{rack}^2 \\ )\omega)^{1.75} \ (\text{if } Re > 2000) \end{cases} \tag{13}$$

where $C_{c2} = 128 \frac{\mu_c b_{c12} L_{c2}}{\pi \rho_c D_{c2}^4}$ and $C'_{c2} = 1.7876 \frac{\mu_c^{0.25} b_{c12}^{1.75} L_{c2}}{\pi^{1.75} \rho_c D_{c2}^{4.75}}$.

The pressure at the fuel filter inlet is simply computed by:

$$P_{efc} = P_{epi} + P_{dfc} \tag{14}$$

B. Model validation

All model constants are identified using either least square method or Nelder-Mead optimization algorithm minimizing the error between the estimation and the real values.

The performances of the model are tested upon a new dataset different from the one used for the identification. The results are summarized in Fig. 2, 3, 4 and 5.

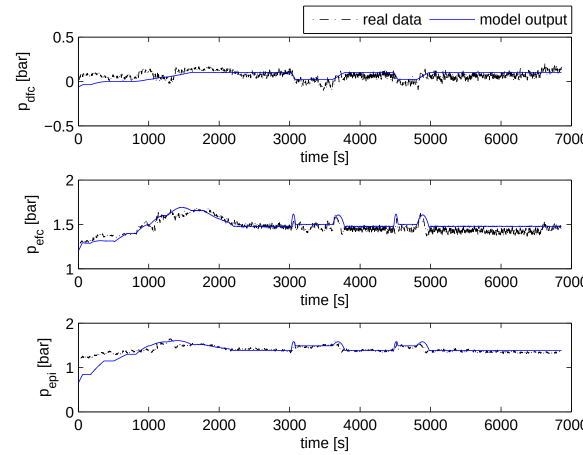

Fig. 2. Estimated pressures versus their real values

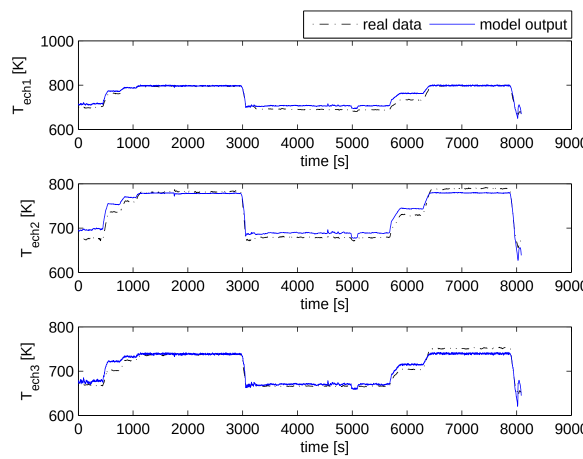

Fig. 3. Estimated temperatures versus their real values

Some small biases can be noticed but in general, model estimations are accurate. Remind that the principal purpose of the model is to give information about the tendency of the measurements and not to give a very precise estimations. Hence, the validation of the model is accepted.

IV. Causal Relationship Definition

A. Monitored faults and faults associated to parameters

1) Monitored faults: The injection system can be subjected to many faults and some of them are frequent and have

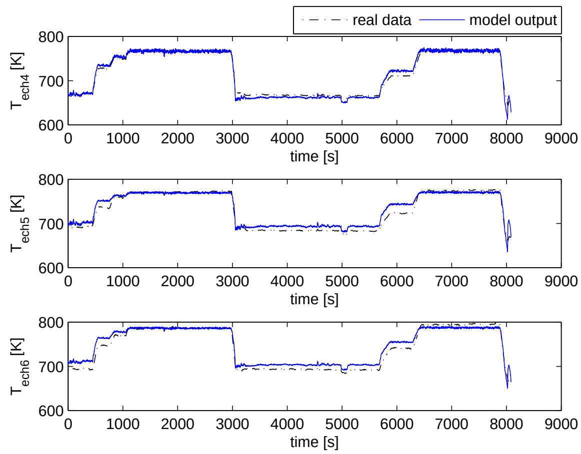

Fig. 4. Estimated temperatures versus their real values

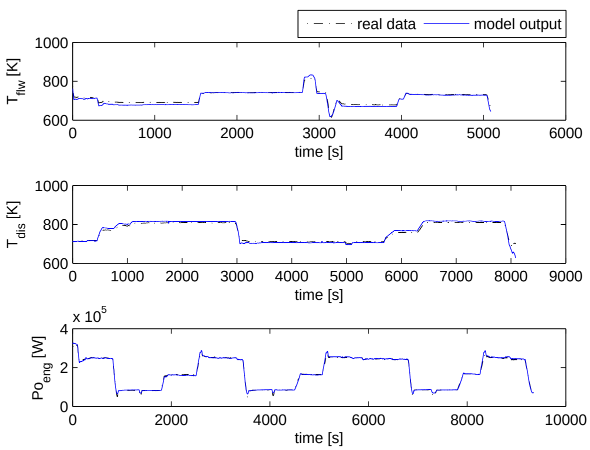

Fig. 5. Estimated power and estimated temperatures versus their real values

hazardous consequences. A particular attention is paid to this kind of faults. Hence, a list of these faults is established in collaboration with maintenance experts on Diesel engines. This list is constituted of:

(F1) a faulty injector (nozzle stuck open, closed...)
(F2) a fuel contamination (water, air...)
(F3) a clogged fuel filter
(F4) a defective lift pump
(F5) a defective injection pump
(F6) a fuel leak (cracked lines, sealing aging)

It has to be noted that in the following sections only non-combined faults are considered.

2) Parameter variation for fault description: In this section, definitions of listed faults are firstly given. Then, the way of introducing faults is presented and modeled by varying parameters or variables.

A faulty injector (F1) is by definition an injector unable to perform correctly the injection function. Hence, the injector may be leaking, partially or completely clogged due to, for example, the deposit of the combustion products in the injector nozzles. Hence a faulty injector, that injects the fuel in the cylinder i, is simulated by increasing or decreasing the corresponding injected fuel flow quantity $\dot{m}_{fi}$. A decrease of it can be introduced to simulate problems of opening and an increase to simulate problems of closing (leaks). The following equation summarizes the introduced variation:

$$\dot{m}_{fi} = \dot{m}^*_{fi}(1 + \delta \dot{m}_{fi}) \tag{15}$$

where $\dot{m}^*_{fi}$ is fuel flow injected into the i^{th} cylinder in a non-faulty state. These parameter variations affects equations (8) and (10). However, as the speed in our study is considered as an input, a particular attention must be paid when simulating this fault. In fact, in a normal state operating mode, additional fuel flow causes an increase of the speed or the torque as it can be noticed in the following equation [13]:

$$J_{vil}\,\omega \frac{d\omega}{dt} = \eta_e \dot{m}_f LHV - C_r \omega \tag{16}$$

where C_r is the resistive torque, J_{vil} the crankshaft inertia.

As no accessibility is available for modifying the resistive torque, each action on the fuel flow must be accompanied with an action on the engine speed to keep (16) satisfied. The variation is introduced according to the following equation:

$$\omega = \omega^*(1 + \delta\omega) \tag{17}$$

where ω^* is the engine speed in a non-faulty state and $\delta\omega = \pm 0.01$. The simulation of the two possible cases of a faulty injector (closing and opening matters) are then done by setting $\delta \dot{m}_{fi} = \delta\omega = \pm 0.01$.

A fuel contamination (F2) is due to the presence of foreign elements in the fuel (water, air, soot ...). These elements contribute to change the physical and chemical properties of the fuel. Hence, (F2) is simulated by an action on the low heat value LHV, the density ρ_c and the dynamic viscosity μ_c. These parameters are directly linked with the equations (8), (10), (12), (13), (14). As the contamination effect on the parameters is not known, all the probable effects are checked one by one. The susceptible variations affecting the dynamic viscosity are given by the following formula:

$$\mu_c = \mu^*_c(1 \pm \delta\mu_c) \tag{18}$$

where μ^*_c is the dynamic viscosity in a non-faulty state.
The susceptible variations affecting the density are given by:

$$\rho_c = \rho^*_c(1 \pm \delta\rho_c) \tag{19}$$

where ρ^*_c is the density in a non-faulty state.
LHV is just subjected to a decreasing pattern because the contamination elements have usually less heat value than the fuel. At the same time, this introduced decrease in LHV must be associated with a decrease in the engine speed to keep (16) satisfied like in the case of a faulty injector. This can be summarized by the following sub-equations:

$$\begin{cases} LHV = LHV^*(1 - \delta LHV) \\ \omega = \omega^*(1 - \delta\omega) \end{cases} \tag{20}$$

To simulate the fuel contamination, the introduced variations of the parameters ($\delta\mu_c$, $\delta\rho_c$, δLHV and $\delta\omega$) are set to 0.01.

A clogging (F3) is linked to a deposit phenomenon leading to the reduction of the flow area. Therefore (F3) is simulated by reducing the diameter of the representative hydraulic charge of the filter D_{c1} as follows:

$$D_{c1} = D^*_{c1}(1 - \delta D_{c1}) \tag{21}$$

where D^*_{c1} indicates the diameter in a non-faulty state. This parameter is directly linked with the equations (12) and (14). A reduction of 1% of the diameter D_{c1} ($\delta D_{c1} = 0.01$) is introduced to simulate a clogging.

A defective lift or injection pump (F4, F5) is by definition the inability of the pump to provide the necessary requested flow. Consequently (F4, F5) are simulated by reducing the fuel flows ($\dot{m}_f$ and D_{carb}) as follows:

$$\begin{aligned} \dot{m}_f &= \dot{m}^*_f(1 - \delta \dot{m}_f) \\ D_{carb} &= D^*_{carb}(1 - \delta D_{carb}) \end{aligned} \tag{22}$$

where the superscript * is put to indicate a non-faulty state of the corresponding variable.
This variable is directly linked with (8), (10), (12), (13), (14).

A fuel leak (F6) is by definition an escape of some quantity of fuel, which leads to the decrease of the principal fuel flow. Accordingly, to simulate the leak (F6), the fuel flows ($\dot{m}_f$ and D_{carb}) are reduced as showed in equation (22). These variables are directly linked, like the precedent case, with (8), (10), (12), (13), (14).
Defective pumps (F4, F5) and leaking of the fuel (F6) are simulated by a reduction of 1% of the fuel flows ($\dot{m}_f$, D_{carb}).

B. Fault simulation and symptom monitoring

In this section, the listed faults are simulated by acting on parameters already listed in the previous subsection.

The simulation of closing and opening matters of a faulty injector, that injects fuel in cylinder i, causes respectively increase and decrease in:

- T_{echi}
- T_{flw} or T_{dis} depending on the membership of i
- Po_{eng}

An increase or a decrease of the dynamic viscosity induces respectively an increase or a decrease in the pressures P_{efc} and P_{epi}. An increase or a decrease of the fuel density provokes respectively a decrease or an increase in the pressures P_{efc} and P_{epi}. A decrease in LHV and ω leads to decreasing pattern in the gas exhaust temperatures at the outlet of the cylinders, at the two turbine inlets and in the engine power.

A decrease in D_{c1} causes the increase of the filter differential pressure P_{dfc}.

(F4), (F5) and (F6) result in decreasing patterns on all the variables (cylinders outlet temperatures, turbine inlets temperatures, engine power, filter inlet and injection pump inlet pressures).

These results are summarized in Table I.

C. Experimental validation

In this section, some real faults are introduced in the engine injection system. The objective of this action is to validate the causal relationships made upon a simulation which are presented in the previous subsection. Unfortunately, not all the simulation results can be checked, out of fear to

TABLE I

FAULT SIMULATION AND OBSERVED SYMPTOMS

Fault	Variations	Symptoms
(F1)	$\dot{m}_{fi} \nearrow$ $\omega \nearrow$	$T_{echi} \nearrow$ T_{β}[1] $\nearrow$ $Po_{eng} \nearrow$
	$\dot{m}_{fi} \searrow$ $\omega \searrow$	$T_{echi} \searrow$ T_{γ}[2] $\searrow$ $Po_{eng} \searrow$
(F2)	$\mu_c \nearrow$	$P_{efc} \nearrow$ $P_{epi} \nearrow$
	$\mu_c \searrow$	$P_{efc} \searrow$ $P_{epi} \searrow$
	$\rho_c \nearrow$	$P_{efc} \searrow$ $P_{epi} \searrow$
	$\rho_c \searrow$	$P_{efc} \nearrow$ $P_{epi} \nearrow$
	$LHV \searrow$ $\omega \searrow$	$T_{ech} \searrow$ $T_{\gamma} \searrow$ $Po_{eng} \searrow$
(F3)	$D_{c1} \searrow$	$P_{dfc} \nearrow$ $P_{efc} \nearrow$
(F4)	$D_{carb} \searrow$ $\dot{m}_f \searrow$	$P_{efc} \searrow$ $P_{epi} \searrow$ $T_{ech} \searrow$ $T_{\gamma} \searrow$ $Po_{eng} \searrow$
(F5)	$D_{carb} \searrow$ $\dot{m}_f \searrow$	$P_{efc} \searrow$ $P_{epi} \searrow$ $T_{ech} \searrow$ $T_{\gamma} \searrow$ $Po_{eng} \searrow$
(F6)	$D_{carb} \searrow$ $\dot{m}_f \searrow$	$P_{efc} \searrow$ $P_{epi} \searrow$ $T_{ech} \searrow$ $T_{\gamma} \searrow$ $Po_{eng} \searrow$

[1] $\beta = \{flw\}$ *or* $\{dis\}$ depending on the membership of i
[2] $\gamma = \{flw\}$ *and* $\{dis\}$ both variables undergo the specified change

damage the engine. Remind that the faults under study are the most frequent and have hazardous consequences. Hence, only filter clogging (F3) and fuel leaking (F6) are introduced on the Diesel engine. In fact, two levels of clogging and leaking were introduced. A soft clogging at 950s followed by a rough one at 1640s until 2200s. At time 2595s a small leaking flow is introduced followed by a rough one at time 3190s. The results are presented in figures 6, 7, 8 and 9.

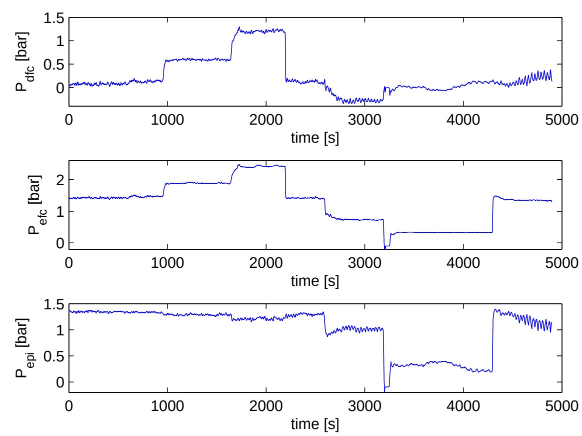

Fig. 6. Variables states in the presence of faults (clogging and leaking)

It is clear from Fig. 6 that a clogging of the filter (F3) causes an increase of the differential pressure like already showed in simulation. Hence, the causal relationship established by simulation is validated. The causal relationship corresponding to the presence of a leaking (F6) indicates that decreasing patterns in the variables should be noticed. Referring to the figures, only the rough leaking are observed and some variables like T_{ech1}, T_{ech2} and T_{flw} are less sensitive than the others. In fact, the soft leaking introduces very small changes that are hidden by the presence of noise.

V. CONCLUSIONS AND FUTURE WORKS

A. Conclusions

A diagnosis-oriented model for an in-line injection system configuration is proposed. This model provides the option

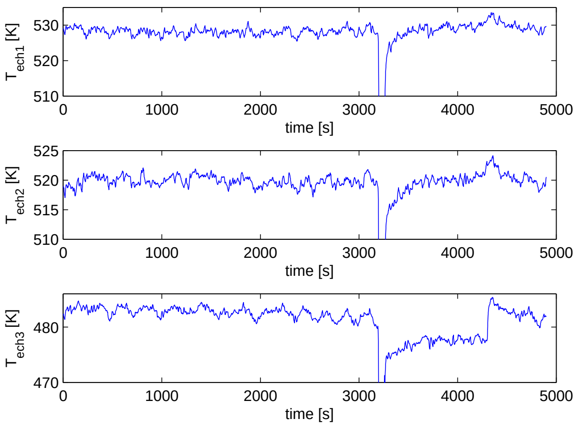

Fig. 7. Variables states in the presence of faults (clogging and leaking)

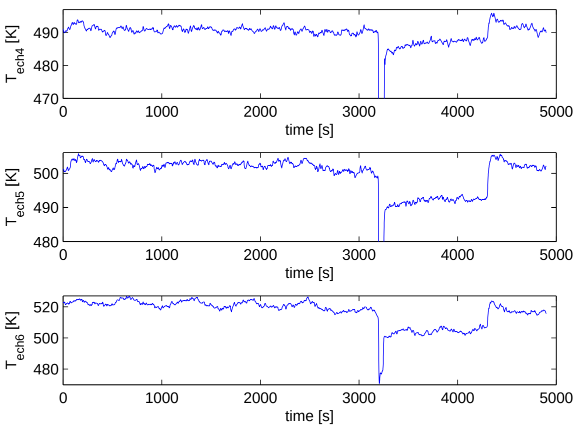

Fig. 8. Variables states in the presence of faults (clogging and leaking)

of output estimation of standard sensors and at the same time the possibility to simulate faults. The model shows good accuracy compared to real data taken from the engine. The simulation of faults allowed the setting of causal relationships. These cause to effect relationships are validated upon some real faults introduced on the engine. The results show good concordances between the simulation and real situations. However, more attention must be paid to fault sensitivity since fault effect on variables is not the same. Same methodology is applied on other engine subsystems and similar results are obtained.

B. Future Works

To complement this work, studies on fault sensitivity, isolation enhancement and the effect of non-detection on the isolation are currently started. Solutions for these issues are planned.

VI. ACKNOWLEDGMENTS

Authors would like to thank BMCI's partners: DCNS, PREDICT, SIMB, CMR, ECT, ECM, CRAN, FEMTO, LEAT and EDF.

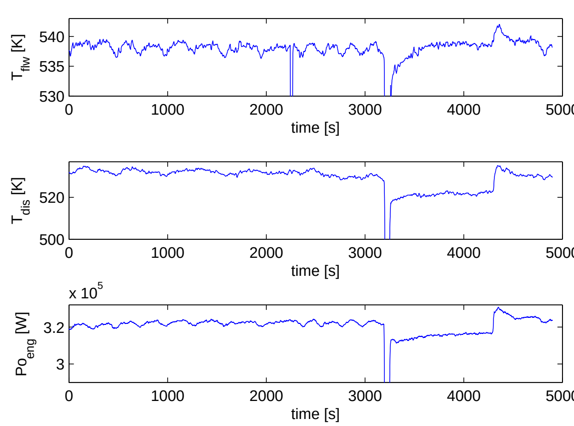

Fig. 9. Variables states in the presence of faults (clogging and leaking)

REFERENCES

[1] F. Kimmich, A. Schwarte, and R. Isermann, "Fault detection for modern diesel engines using signal and process model-based methods," *Control Engineering Practice*, vol. 13, pp. 189–203, 2005.

[2] M. Nyberg and T. Stutte, "Model based diagnosis of the air path of an automotive diesel engine," *Control Engineering Practice*, vol. 12, pp. 513–525, 2004.

[3] R. Sun, F. Tsung, and L. Qu, "Combining bootstrap and genetic programming for feature discovery in diesel engine diagnosis," *International Journal of Industrial Engineering*, vol. 11, no. 3, pp. 273–281, 2004.

[4] A. Berton, E. Frisk, and M. Nyberg, "Improving diagnosis performances on a truck engine making use of statistical charts," *IFAC Symposium on Advances in Automotive Control*, pp. 421–426, 2004.

[5] Y. Khelil, G. Graton, M. Djeziri, M. Ouladsine, and R. Outbib, "Marine diesel engine faults diagnosis - a generic approach," *1st PAPYRUS WORKSHOP - Fault Diagnosis and Fault Tolerant Control in large scale processing industries*, 2011.

[6] M. Sachenbacher, A. Malik, and P. Struss, "From electrics to emissions: Experiences in applying model-based diagnosis to real problems in real cars," *Proc. 9th Int. Work. on Principles of Diagnosis*, pp. 246–253, 1998.

[7] R. Antonic, Z. Vukic, and L. Kuljaca, "Fuzzy modeling and control of marine diesel engine process," *Proc. of 11 th Mediterranean Conf. on Control and Automation*, 2003.

[8] I. Golub, R. Antonic, and A. Cibilic, "Fault diagnosis and isolation of the marine diesel engine turbocharger system," *MIPRO 2009, Microelectronics, Electronics and Electronic Technology*, pp. 208–213, 2009.

[9] G. Chandroth, A. Sharkey, and N. Sharkey, "Artificial neural nets and cylinder pressures in diesel engine fault diagnosis," *Proceedings of INMARCO98*, 1998.

[10] S. Xiaochun and H. Hongying, "Diesel engine fault diagnosis and classification," *8th International Conference on Signal Processing*, 2006.

[11] G. Graton, J. Fantini, F. Kratz, J. Ragot, and P. Dupraz, "Diagnosis of diesel injection systems using finite memory observers," *IFAC Symposium on Advances in Automotive Control*, pp. 19 – 23, 2004.

[12] R. Ceccarelli, C. C. de Wit, P. Moulin, and A. Sciarretta, "Model-based adaptive observers for leakage detection in diesel engines," *Proc. of American Control Conference*, pp. 1128 – 1133, 2009.

[13] R. Omran, R. Younes, and J. C. Champoussin, "Neural networks for real-time nonlinear control of a variable geometry turbocharged diesel engine," *International Journal of Robust and Nonlinear Control*, vol. 18, p. 1209 1229, 2007.

[14] J. Heywood, *Internal combustion engine fundamental*. McGraw-Hill Inc, New York, USA, 1988.

[15] R. Younes, *Élaboration d'un modèle de connaissance du moteur Diesel avec turbocompresseur géométrie variable en vue de l'optimisation de ses émissions*. Phd thesis, École Centrale de Lyon, 1993.

[16] R. King, *Introduction to practical fluid flow*. Butterworth-Heinemann, 2002.