ENGINE SPEED LIMITER FOR WATERCRAFTS

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Abstract— Engine speed limiters are safety devices designed to prevent an engine from exceeding a predetermined high speed. The speed limiter herewith presented is designed for large and sudden load variations which require accurate and fast air-fuel ratio control. It is based on feedback linearization and on-line estimation of the load torque. Experimental testing was conducted with a Sea-Doo personal watercraft. A system was used to bypass the engine’s Electronic Control Unit (ECU) injector commands in order to use those of the new controller. Experimental results indicate that it reaches higher performances than the usual ECU injection cut-off strategy.

Index Terms— Engine Control, Fuel Control, Limiter, Kalman filters, Feedback Linearization.

I. INTRODUCTION

An engine speed limiter, also called a rev-limiter, is a safety control device which prevents an engine from exceeding a predetermined high engine speed. Usually, when an engine reaches its speed limit, the firing timing of the ignition is delayed by reducing combustion time in the cylinder in order to stop engine acceleration. In case of a large overrun, firing or injection must be disabled to prevent combustion in the cylinders and, consequently, decelerate the engine speed. When the decelerating engine reaches a lower speed limit, firing or injection is activated to prevent the engine from stalling. In other words, a relay control is activated to maintain the engine speed close to its limit value, which is specified by upper deactivation and lower reactivation values.

However, although this is an efficient strategy for automotive, it is unacceptable for a personal watercraft because it regularly operates in conditions where its propeller is partially loaded with water or completely unloaded when the watercraft takes off. In theses cases, the engine operates on the rev-limiter with the relay control. This implies a periodic deactivation of the injection in order to maintain engine speed close to its limit. For example, the engine speed will fluctuate between 7000 and 7600 rpm at a 20-Hz frequency. However, this solution generates tone variations of the noise radiated by the watercraft and, from a psychoacoustics point of view, watercraft users prefer constant sound over fluctuating variations.

This object of this paper is to propose a suitable engine speed limiter that reduces the sound associated with the sudden tone variations that occur in these conditions. Solutions could entail controlling the engine speed with a throttle-by-wire and a PID controller. This would be an efficient solution for golf course vehicles, for example [7]. However, it is inefficient when applied to watercrafts where sudden large load variations occur because the throttle-by-wire dynamic is too slow compared with the unloading dynamic of the propeller. To develop an efficient rev-limiter for large and sudden propeller unloading, this paper proposes a controller which commands the air/fuel ratio (AFR) as it can have a very strong and rapid influence on the engine torque [11-13]. The structure of the controller was suggested by the most recent developments in idle speed control (ISC), one of the most generic automotive problems [3], where fast and accurate AFR controllers are based on the estimation theory [10]. However, the ISC problem deals with low load torque modulation at low speed and does not solve problems with large load torque modulation at high speed. Consequently, original developments were achieved for specific applications to watercrafts.

Section 2 presents the proposed control strategy and introduces the six points which should be considered when studying the rev-limiter problem. Section 3 makes a detailed presentation of the event-based controller that was developed, including the design of the Kalman estimator used to assess the load torque on-line. The controller was implemented in a dSpace system to bypass the ECU injector commands. The experimental setup included a Sea-Doo personal watercraft running in a water tube. Section 4 presents the experimental results obtained in two cases: 1) a partially loaded propeller, and 2) a sudden large unloading or reloading of the propeller.

II. CONTROL STRATEGY

A. Engine Model

The rotational dynamics of an internal combustion engine can be modeled as follows:
\[ C_1 \frac{dN}{dt} = \Sigma T \]  

(1)

where \( C_1 = \frac{2\pi}{60} \) with \( I \) the equivalent moment of engine inertia, and \( \Sigma T \) is the sum of torque applied on the shaft:

\[ \Sigma T = T_{eng} - T_{Load} + w_T \]  

(2)

\( T_{eng} \) is the engine torque output, \( w_T \) is a random torque disturbance, and \( T_{load} \) is the torque resulting from the load. Thus, stationary engine speed, \( N_0 \), is characterized by a singular point defined by null acceleration, i.e. \( T_{eng}(N_0) = T_{Load}(N_0) \) if it is assumed that \( w_T = 0 \).

The load torque due to the propeller is assumed to be a static nonlinearity:

\[ T_{load} = \alpha f(N) \]  

(3)

where \( f(N) = C_{sm}N^n \) is a theoretical load curve with \( C_{sm} \) a constant and \( n \) an exponent found by experience [8]. The coefficient \( \alpha \) represents the loading of the propeller: the propeller is unloaded for \( \alpha = 0 \) and fully loaded for \( \alpha = 1 \). However, with a personal watercraft in its operating environment, \( \alpha \) is a random variable with unknown probability distribution.

\section{Problem statement}

On an engine with direct injection device, the engine torque can be controlled by modulating the AFR lambda defined as \( \lambda = m_a/14.66m_f \), where \( m_a \) and \( m_f \) are respectively the amount of air and fuel in a cylinder [5]. For ideal stochiometric combustion, this ratio is equal to one. Typically, the maximum power output is reached with a richer mixture, \( \lambda_{max} = 0.9 \). There is a decrease in engine power output when the mixture is lean, \( \lambda > 1 \), or too rich, \( \lambda < \lambda_{max} \). Moreover, a mixture that is too lean is no longer inflammable. Hence, the engine torque output is a nonlinear monotone decreasing function of the AFR engine and also a function of the engine speed: \( T_{eng} = g(N, \lambda) \) for \( \lambda \geq \lambda_{max} \). For the purpose of this article, the maximum engine torque, \( T_{eng, max}(N) \), will henceforth be defined for a wide-open throttle, where \( \lambda = \lambda_{max} \). Thus, for the highest engine power, the highest stationary engine speed for normal conditions is said maximal, \( N_{max} \), and it is reached when \( T_{eng, max}(N_{max}) = T_{Load}(N_{max}) \) for \( w_T = 0 \). Consequently, the engine speed limit, \( N_{lim} \), is set higher than \( N_{max} \).

The difficulty lies in controlling the engine power output in order to keep the engine speed, \( N \), under the engine speed limit, \( N_{lim} \). There are two cases to consider:

i) conventional control when the engine is fully loaded, \( \alpha = 1 \): the highest stationary engine speed, \( N_{max} \), reached with the highest engine torque is lower than the engine speed limit, \( N_{lim} \).

ii) rev-limiter control when the engine is partially loaded, \( \alpha < 1 \), or unloaded, \( \alpha = 0 \): the engine speed can be superior than the speed limit, then the engine power output must be reduced by fuel injection control in order to limit the engine speed at its \( N_{lim} \) limit.

There are six basic problems which arise with this formulation:

i) The maximum torque mode must be switched to the reduced torque mode.

ii) In case of a sudden unloading, the controller must react very rapidly to predict the engine torque in order to avoid an engine over-rev.

iii) In case of a rapid reloading, the controller must react very quickly to increase the engine torque output in order to prevent the engine from stalling.

iv) In case of slow time-varying loading or steady unloading, the controller must adjust the engine torque output with great precision in order to avoid engine speed fluctuations.

v) The fuel command strategy performance is limited by the time delay between the exact moment to inject less fuel in the engine and the exact moment when this action has an effect on the engine speed.

vi) The engine speed limit must be adaptable to the driving condition in order to keep the engine operating at maximum power output when the engine is fully loaded.

\section{C. The proposed approach}

The control objective is to keep the engine speed error positive: \( e_N = N_{lim} - N \geq 0 \). Alternatively, the problem can be formulated in terms of classical control where the objective is to reach the maximum speed limit, \( e_N = 0 \). Let us consider a feedback law, \( T_{eng} = T_{Load} + Ke \), under the constraints \( T_{eng} \leq T_{eng, max}(N) \) and \( w_T = 0 \).

Three cases must be considered:

Case 1: when the system is fully loaded (\( \alpha = 1 \)). The controller is always trying to reach the inaccessible engine speed limit (point 1, Fig. 1). Thus, the engine torque command is saturated to its maximum value, \( T_{eng} = T_{eng, max}(N) \), and the stationary engine speed is \( N_{max} \) (point 2, Fig. 1).

Case 2: when the system is at the loading limit, \( \alpha_{lim} \), the objective is reached, \( e_N = 0 \), without reduction of the engine torque (point 3, Fig. 1).

Case 3: when the loading is inferior than \( \alpha_{lim} \), the engine speed can exceed the speed limit if \( T_{eng, max} \) is applied (point 4, Fig. 1). However, in this case, the error is negative, \( e_N < 0 \)
so engine torque is reduced, \( T_{\text{eng}} < T_{\text{eng,max}} \), and the stationary engine speed slips from point 4 to 5.

The association of a speed regulator with command saturation will result in positive speed error by trying to reach the null value. Hence, the controller is always activated and there are no decisions to be made about switching between conventional control to the rev-limiter control (lean mode).

### III. EVENT-BASED CONTROLLER

#### A. Discrete time model synchronized with crankshaft

A convenient approach to modeling the behavior of an internal combustion engine is to use a discrete time model synchronized with crankshaft angle instead of time as the evolution variable [1]. This model can be referred to as an event-based engine model because the crankshaft angle is synchronized with many important events such as engine valve actuation and spark ignition timing.

For a four-stroke combustion engine with intermittent fuel injection, the control of fuel injected per cycle can be actualized many times per camshaft revolution. For example, on the three-cylinder engine used to develop the controller introduced in this paper, the decision angles are chosen at 130°, 370° and 610° for, respectively, the fuel injection command to cylinders 1, 2 and 3. By multiplexing the signals, the discrete time, \( k \), represents the three cylinders one after the other: \( k \) modulo 3=cylinder; i.e., for the sequence \( k=\{1,2,3,4,5,6,7,\ldots\} \), these cylinders are \( \{1,2,3,1,2,3,1,\ldots\} \). Hence, the engine torque output at iteration \( k \) is provided by the following relation (for a constant value of air mass per cylinder):

\[
T_{\text{eng}}[k] = q^{-d}g(N[k], m_f[k])
\]

where \( q^{-1} \) is the operator delay, \( g \), the engine map and \( N \), the engine speed. This model (4) takes into consideration the inherent time delay associated with the injection of fuel in a cylinder. Consequently, there is a time delay of \( d \) iterations, \( q^{-d} \), between the decision to inject an amount of fuel, \( m_f \), and the engine torque resulting from the command, \( T_{\text{eng}} \).

The discrete model transfer function of system (1) is given by:

\[
N[k] = \frac{B(q^{-1})}{A(q^{-1})}[T_{\text{eng}}[k] - T_{\text{Load}}[k]]
\]

with \( N[k] \) the engine speed, \( B(q^{-1}) = (b + bq^{-1}) \) the numerator where \( b = \frac{15t_c}{\pi} \), and \( A(q^{-1}) = (1 - q^{-1}) \) is the denominator. The sampling rate, \( t_c \), can be assumed constant because the engine speed will be controlled near the specified speed limit, \( t_c \approx 120/n_{\text{cyl}}N_{\text{lim}} \) where \( n_{\text{cyl}} \) is the number of cylinders.

#### B. The controller

The control strategy is based on AFR modulation. There are two mains nonlinearities in the system: the engine torque is a nonlinear function, \( g \), of the input, the fuel mass per cylinder \( m_f \) (5), and the load torque is a nonlinear function of the engine speed (3). The controller design is based on the feedback linearization method [6]. It was designed to find a nonlinear feedback control that would result in a linear closed loop.

Compensating the nonlinear engine map, \( g \), is simply achieved by inversing it [2]. Thus, by specifying the desired engine torque output, \( T_{\text{eng,ref}} \), the engine speed and operating conditions, the inversed engine model, \( \hat{g}^{-1} \), can be used to calculate the quantity of fuel to be injected.

\[
m_f = \hat{g}^{-1}(T_{\text{eng,ref}}, N)
\]

For perfect estimation of the parameter \( N \) and perfect measurement of the engine model, \( g = \hat{g} \), the amount of fuel to inject computed with equation (6) produces the engine power needed to reach the reference engine torque in \( d \) iterations, \( T_{\text{eng}}[k] = T_{\text{eng,ref}}[k - d] \).

To compensate load torque nonlinearity, the torque engine reference is built as follows:

\[
T_{\text{eng,ref}}[k] = \hat{T}_{\text{Load}}[k] + \Delta T_{\text{eng}}[k]
\]

where \( \hat{T}_{\text{Load}}[k] \) is the estimation of the load torque and \( \Delta T \) is called the engine torque correction. For a perfect assessment of the load torque, \( \hat{T}_{\text{Load}}[k] = T_{\text{Load}} \), and perfect
compensation of the engine map, \( T_{\text{eng}}[k] = T_{\text{eng,ref}}[k-d] \), the discrete model (4) becomes completely linear with \( \Delta T[k] \) as input and \( N[k] \) as output:

\[
N[k] = \frac{B(q^{-1})}{A(q^{-1})} q^{-d} \Delta T[k]
\]  

(8)

Hence the controller, as shown in Figure 2, includes a load estimator to provide an estimate of the load torque, \( \hat{T}_{\text{load}} \), a \( g^{-1} \) function for the linearization of the engine map, and a linear feedback controller to control engine speed precisely through the \( \Delta T \) engine torque correction.

![Controller block diagram](image)

Fig. 2. The controller block diagram

C. The feedback

To determine the \( \Delta T \) engine torque correction, the engine speed error, \( e_N = N_{\lim} - N \), passes through a numerical controller which has the following \( R \), \( S \) and \( T \) canonical structure:

\[
\Delta T[k] = \frac{T(q^{-1})}{S(q^{-1})} N_{\lim}[k] - \frac{R(q^{-1})}{S(q^{-1})} N[k]
\]  

(9)

The controller is tuned according to a pole placement method without canceling the zeros (for robustness reasons because the zeros are on the unit circle). Thus, the denominator of the closed loop transfer, \( A_BF(q^{-1}) \), is specified. The value of the polynomials \( S(q^{-1}) \) and \( R(q^{-1}) \) coefficients are calculated by resolving the following Diophantine equation:

\[
A(q^{-1})S(q^{-1}) + q^{-d} B(q^{-1})R(q^{-1}) = A_BF(q^{-1})
\]  

(10)

D. The load estimator with Kalman observer

The task of the torque estimator is to determine to what extent a change in engine speed is the result of a change in the load, a change in the engine torque output or random noise. Thus, the problem is linearized as follows. From the time derivation of equation (3), we can formulate that the load torque is resulting from the acceleration and/or loading change:

\[
\frac{dT_{\text{load}}}{dt} = \alpha \frac{df}{dN} \frac{dN}{dt} + f(N) \frac{da}{dt}
\]  

(11)

By substituting equations (1), (2) and (3) in equation (11), and considering the point at \( \alpha = 1 \) and \( N = N_{\text{max}} \), a change in the load is written as a first order transfer function:

\[
\frac{dT_{\text{load}}}{dt} = -K_{\text{lin}} T_{\text{load}} + K_{\text{lin}} T_{\text{eng}} + K_{\text{lin}} w_T + f(N_{\text{max}}) w_\alpha
\]  

(12)

with \( K_{\text{lin}} = \frac{1}{C_1} \frac{df}{dN} \frac{1}{N_{\text{max}}} \) and \( w_\alpha = \frac{da}{dt} \) a random variable.

The objective is to generate an on-line estimate of the load torque, \( \hat{T}_{\text{load}} \), from observations of the engine speed, \( N \), and the engine torque, \( \hat{T}_{\text{eng}} \). The main problem is found in the balance between measuring disturbance sensitivity (the measurement noise) and system disturbance tracking (the load torque variations). However, if the stochastic properties of the disturbances (noise and load torque) are known, the balance can be formalized and an optimal state observer, named Kalman filter, can be implemented [14].

According to the relations (1), (2) and (11), the linear state model of the system is

\[
\frac{dx(t)}{dt} = Ax(t) + Bu(t) + Gw(t)
\]  

(13)

\[
y(t) = Cx(t) + v(t)
\]

with \( x = \begin{bmatrix} N \\ T_{\text{load}} \end{bmatrix} \), \( A = \begin{bmatrix} 0 & -1/C_1 \\ 0 & -K_{\text{lin}} \end{bmatrix} \), \( B = \begin{bmatrix} 1/C_1 \\ 0 \end{bmatrix} \), \( G = \begin{bmatrix} 1/C_1 & 0 \\ 0 & f(N_{\text{max}}) \end{bmatrix} \), \( C = \begin{bmatrix} 1 \\ 0 \end{bmatrix} \), \( u = T_{\text{eng}} \) the engine torque, the outputs are the engine speed and the load torque, \( v = \begin{bmatrix} v_N \\ v_{T_{\text{load}}} \end{bmatrix} \) is the measurement noise, and \( w = \begin{bmatrix} w_T \\ w_\alpha \end{bmatrix} \), the process noise. The noises are modeled as independent white noises of zero-mean value.

The time state model (14) is written as a discrete state model with a bilinear approximation:

\[
z[k+1] = A_d z[k] + B_d u[k] + G_d w[k]
\]  

(14)

\[y[k] = C_d z[k] + D_d u[k] + v[k]
\]

Thus, the discrete time Kalman estimator is:

\[e[k] = y[k] - C_d z[k] - D_d u[k]
\]  

(15)

\[\hat{z}[k+1] = A_d \hat{z}[k] + B_d u[k] + Le[k]
\]
\[ \hat{z}[k] = \hat{z}[k-1] + M \epsilon[k] \]
\[ y[k] = C_d \hat{z}[k] + D_d u[k] \]

The estimator gain, \( L \), must be large so that old errors die out. On the other hand, a large \( L \) gain provides significant influence to the measurement noise. The Kalman filter solution provides optimal estimator gain, i.e. minimal estimation error.

**IV. EXPERIMENTATIONS**

**A. Experimental set-up**

The experimental set-up was a Sea-Doo personal watercraft GTX-4TEC running in a water tub (Fig. 3). The engine was a four-stroke Rotax engine, 155 hp, 3 cylinders, with multi-port electronic fuel injection. The GTX-4TEC is specifically designed for marine use and engineered to withstand the heavy demands placed on a watercraft engine. The propulsion system is a water jet pump with a direct drive transmission. To generate large load modulation quasi instantaneously, there is a tube located at the water intake which makes it possible to load it with air.

![Fig. 3. The global testing set-up in the water tub.](image)

Figure 4 shows the integration of the engine in dSPACE. The engine’s electronics, located on the far right, deliver five signals: encoder, camshaft and 3 injection commands. The engine receives the three injection commands from the DS402. The encoder and camshaft signals are converted into TTL format to be read by the DS402 electronic card. When the switch is in ECU position, the electronic reproduces the ECU command on the engine’s injectors. Alternatively, when the switch is in DS402 position, the dSPACE system controls the injectors.

The “S_function” blocks represent the DS402 codes which were written specifically for the engine’s application. The “Engine_data” block computes the engine speed (\( N \) in rpm) and the crankshaft angle (\( \theta \) in degrees). The S functions, “Injector master” and “Injector slave”, command the injectors according to the required injection duration (\( u \) in seconds). The S_function “ECU_data” measures the injection durations that are applied by the ECU. The “Simulink” block uses the engine speed, crankshaft angle, measured injection durations and stored engine maps to estimate the delivered engine torque precisely.

![Fig. 4. Block diagram illustrating the engine’s dSPACE interface.](image)

In open loop, there is one pole due to the denominator \( A(q^{-1}) = (1-q^{-1}) \). In closed loop, there is one pole specified by \( A_{BF}(q^{-1}) = (1-a_{BF}q^{-1}) \). Two extreme values of \( a_{BF} \) can be considered. When \( a_{BF} = 1 \), there are no modification of the open loop pole and no control. In contrast, when \( a_{BF} = 0 \), the pole is infinitely damped and such specification leads to a deadbeat controller: the reference is reached in a finite number of iterations. From a theoretical point of view, this solution could be seen as being faster. In fact, this is not a robust solution because small errors in the parameters or in the structure will dramatically decrease the performance and the closed loop stability. To design a robust controller, it is preferable to specify the placement of the closed loop pole near the pole of the open loop. The experiments indicate no improvements when the closed loop pole is set lower than \( a_{BF} = 0.7 \), which corresponds to a time constant of the closed loop equal to 16 ms.

The discrete time engine model was validated with experimental data, and the variance of the measurement noise on the engine speed was estimated at close to 600. The measurement noise on the engine torque is set to infinity because there is no load torque measure. The random torque disturbance is assumed low, at 10Nm. The standard deviation of the random variable, \( w_t \), is chosen equal to 30 s\(^{-1}\). Consequently, the matrixes \( M \) and \( L \) are computed with Matlab from the process noise covariance matrix

\[ Q = E \begin{bmatrix} 10^2 & 0 \\ 0 & 30^2 \end{bmatrix} \]

and the measurement noise covariance matrix

\[ R = E \begin{bmatrix} 600 & 0 \\ 0 & \infty \end{bmatrix} \].

B. Switching from the ECU to the controller

During testing, the personal watercraft propeller was partially loaded. Hence, when the throttle was wide open, the upper engine speed limit was reached and the speed limiter had to be activated. When the conventional speed limiter was used (the usual ECU), the rev-limiter mode generated engine speed fluctuations from 7000 to 7800 rpm characterized by a period of 0.2 s. Figure 5 shows such undesirable behavior from 0 to 0.3 s. When the engine speed is below 7000 rpm, normal injection duration occurs and maximal torque is applied; then, the engine speed increases and reaches the upper limit at 7600 rpm. Consequently, the injection is stopped (0 ms) in order to decrease the engine speed. Switching between 0 and 10 ms with peaks at 14 ms generates periodic engine speed fluctuations. Beginning at 0.3 s, the controller takes control of the injectors, stabilizing the engine speed at 7000 rpm in less than 0.2 s and maintaining an injection duration of 8 ms. This constant injection duration generates an engine torque which is exactly equal to the load torque.

C. Revolution speed control during load transition

Figure 6 presents the case of a sudden unloading of the propeller obtained by injected air in the water intake of the watercraft. In spite of the large and sudden variation of the load torque, the engine speed is maintained close to the engine reference at 7000 rpm: the mean engine speed is 7040 rpm with a peak-to-peak value of 453 rpm during the transition. The estimated load torque computed by the Kalman observer shows that the load torque moves down from 110 to 50 Nm in 0.15 s. However, in spite of this large torque variation, the injection time looks similar. This is due to the AFR influence on the engine torque: for this lean mode, a small change in the injection time leads to a large change on the engine torque.

D. Change of the speed engine limit

To show the controller’s tracking ability, Figure 8 presents the case where the engine speed limit is changed from 6500 to 7500 rpm when the propeller is partially loaded. Prior to 0.5 s, the engine speed limit is 6500 rpm. At 0.5 s, the upper limit is set at 7500 rpm, then the controller computes an injection time of 12 ms in order to apply the maximal torque. Consequently, the engine speed quickly increases from 6500 to 7500 rpm in 0.1 s. After 0.6 s, the upper limit is reached and the injection time is decreased to 8 ms in order to stop the acceleration.
V. CONCLUSION

This paper described an engine speed limiter for the particular case of large load torque modulation. The limiter works as a controller when the upper limit is reached, thus maintaining the engine speed to a fixed reference. The controller also delivers an estimation of the resisting load torque from the measurements of the injector commands and the engine speed. A dSpace system was used to bypass the engine ECU injector commands. Experimental results show that the developed controller performs better than the ECU cut off strategy.

To reduce the noise burden associated with the sudden tone variations that occur in usual watercraft driving conditions, this paper proposes a suitable engine speed controller. Moreover, the controller can be applied as an engine speed limiter at 5000 rpm for the Learning Key option. Finally, this work is an important step in implementing throttle-by-wire control.

REFERENCES