Analytical study of an optimized limited motion actuator used in engine combustive flow regulation

Abstract. This article aims at an optimal design of a limited motion actuator regulating a flap position in order to control the combustive flow of a diesel engine. The required performances are a reduced size for a 0.4 Nm torque and an angular stroke from 0 to 80°, with operative constraint at 130°C temperature and at 9V voltage. The objective is to minimize the electrical energy consumption of an electromechanical actuator composed by a brushed DC motor and a spur gear set. A finite element model with Flux2D software is used to check the optimized actuator.

Streszczenie. Artykuł ma na celu optymalne zaprojektowanie aktuatora z ograniczonym zakresem ruchu, regulującego pozycję wahacza w sterowaniu wydalaniem spalin w silniku diesla. Wymagane działania są zredukowane do momentu 0,4 Nm i kątowego suwu od 0 do 80°, z parametrami działania: temperaturą 130°C i napięciem zasilania 9V. Funkcją celu jest minimalizacja konsumpcji energii elektrycznej w aktuatorze elektromechanicznym złożonym ze szczotkowego silnika prądu stałego i zestawu aktywacji sprzęgła. Zoptymalizowana konstrukcja aktuatora została zweryfikowana modelem numerycznym przy użyciu pakietu FLUX2D. (Badanie analityczne optymalnego aktuatora z limitowanym ruchem użytego w regulacji wydalania spalin).

Keywords: EGR valve, limited stroke, brushed DC motor, spur gear set, optimization.
Słowa kluczowe: zawór EGR, ograniczony suw, szczotkowy silnik prądu stałego, zestaw aktywacji sprzęgła, optymalizacja.

Introduction

According to the recent European Standard [1] applied to car manufacturers in order to reduce the greenhouse gases and polluting emissions of vehicles, new conceptions of internal combustion engines need a new regulation management of combustive as described in [2].

After a state of the art about several actuators which manage the combustive flow, as shown in [3], two electric topologies to produce rotary movements can mainly be of interest: the first one is an indirect drive composed of a DC motor, as the fig.1, and a tw o stage reduction gear as the fig.2 ; the second one is a permanent magnet motor direct drive.

Fig.1. Brushed DC motors

Fig.2. Motor reduction gear unit

The motor reduction gear unit is chosen to be design under constraints. The optimization is realized by genetic algorithm. Magnetostatic and transient studies are carried out with Flux 2D and the gear set freeware with Filengrene.

DC motor characterization

The DC motor is structured by one pole pair and two winding paths, according to structures quoted in [4]. Two brushes keep electrical contacts between the stator and the windings paths of rotor. The stator has two permanent magnets and a yoke to drive the magnet and wire created flux. The rotor is composed of n turns and steel plates used to drive the flux. Motor model is simplified for analytical modelling and the rotor has ten slots as presented in the following fig.3.

Fig.3. Quarter geometrical brushed DC motor definition

The rotor length is $L_r$, its radius $R_r$. The teeth length is $h_d$, their thickness $l_d$ is defined for a 1.0T flux density. The slot section under pole contains of n/2 turns in series as :

$$S_{enc} = \frac{N_{enc}}{2} \left( \frac{\pi(R_r^2 - (R_r - h_d)^2)}{N_{enc}} \right) - h_d \cdot l_d$$

Between two brushes, there are two electrical resistances mounted in parallel. Every resistance is composed by n/2 turns in series. The wire length around one pole is the sum of the length of rotor and the half perimeter concerning the head coil. The electrical resistance is:

$$R = \frac{1}{2} \rho_{\epsilon \Gamma} \left[ \frac{n}{2} (L_r + \pi \cdot R_r) - \frac{4}{\pi} d_{\\epsilon}^2 \right]$$

The rotor is made of steel and copper. The diameter $d_{\\epsilon \Gamma}$ and length wire $L_{\epsilon}$ depends on slot section, electrical resistance and filling factor. The rotor inertia is:

$$J_{\epsilon} = \frac{1}{2} \left[ \mu_{\epsilon \Gamma} \cdot \left( \pi \cdot R_r^2 \cdot L_r - 2 \cdot S_{enc} \right) + \mu_{\epsilon \Gamma} \cdot \left( \frac{\pi d_{\\epsilon}^2}{4} \cdot L_{\\epsilon} \right) \right]$$

The air gap flux density under a magnetic pole is:

$$B_g = \frac{B_{\epsilon}}{\mu_{\epsilon \Gamma} \cdot e + k_{\epsilon \Gamma} \cdot e_{\epsilon}}$$
With $B_r$, magnet remanent flux density, $\phi_a$, magnet thickness, $\mu_r$, relative permeability, $e$, airgap thickness, and $k_{ea}$, magnet concentration flux density between airgap section and magnet section.

As [4], all wires under a pole induce the back electromotive force. For a pole opening angle $\gamma_p$ equal to $120/180\pi$, the torque constant is defined by:

$$K_r = 0.5 \cdot \gamma_p \cdot R_e \cdot L_e \cdot B_r$$

In one of the two windings paths, the current value is 66 $\mu_a$ with 80MPa elastic strength. The gear thickness is:

$$C_m = 2 \cdot \frac{J}{\pi} \cdot B_r \cdot \gamma_p \cdot R_e \cdot L_e$$

Then, optimization variables parameters for DC motor are $L_e$, $R_e$, $h_b$, $\phi_a$, $n$, $d_{aw}$. 

**Spur gear set definition**

The gear set is composed of two stage reduction gears with a 81% efficiency. The gear box reduces speed motor and increases the torque motor with a reduction ratio $N$ between the speed motor $\omega_m$ and the flap speed $\omega$ equal at:

$$\frac{1}{N} = \frac{\omega_m}{\omega} = \frac{Z_m}{Z_2} = \frac{Z_3}{Z_2}$$

With $Z_2$ tooth number of motor pinion, $Z_3$, tooth number of intermediate wheel, $Z_3$, tooth number of intermediate pinion, $Z_v$, tooth number of flap wheel.

In order to insuring the power transfer between two spur gears as for example pinion motor and intermediate wheel, it is necessary to design the tooth as described in [5,6,7]. The method is based on a study of strength resistance to choose the gear module $m$ depending on operating conditions and bending stresses by tangential force $F_b$. The maximal stress in a tooth must be lower than the elastic stress $R_e$ with a service factor of 25%. Gear material is a Pa 66 with 80MPa elastic strength. The gear thickness is:

$$l_z = \frac{544 \cdot F_b \cdot f_y}{R_e \cdot \pi \cdot m}$$

Between every stage reduction gear, gear modules must be the same as $m_m = m_2 = m_3$. The tooth thickness is calculated with the maximal transmitting torque. Then the inertia $J_{red}$ of intermediate wheel and pinion can be defined with their thickness and pitch diameters. The total inertia of all moving parts returned on motor shaft $J_{tot}$ can then be calculated:

$$J_{tot} = J_m + \frac{J_{red} + J_{ch}}{N^2}$$

Consequently, the reduction gear variables parameters are $Z_m$, $Z_2$, $Z_3$, $m_m$, $m_2$, $m_3$, $l_z$, $l_2$, and $l_3$.

**Dynamics definition**

To calculate the flap position versus supply power time, the electrical balance (10) and mechanical (11) equations must be solved. The motor inductance impact is neglected to simplify this calculation as shown in [8].

$$U = R \cdot I + K_e \cdot \omega_m$$

$$J_{aw} \cdot \omega_m = K_e \cdot I - \frac{C_m}{N \cdot r_{red}}$$

According to equations (9) and (10), the load torque on the motor shaft, the motor torque constant and electrical constant, the motor speed are linked by a first order differential equation:

$$\frac{U}{K} = \frac{R \cdot C_m}{K^2 \cdot N \cdot r_{red}} = \frac{R \cdot J_{aw} \cdot \omega_m}{K^2} + \frac{\omega_m}{dt}$$

To respect constraints, actuator has to drive the flap on 80° angular range, with a 150ms response time including acceleration and deceleration for the forward and backward travel. The optimization method calculate the switching time with respect to the flap position specification on a forward travel of 0° to 80°, and a backward travel of 80° to 0°. The initial conditions of forward travel being a positive supply voltage $U$ and a null motor speed at 0°, the speed motor during the acceleration time $T_a$ is:

$$\omega_m^a(\tau) = \left[ \frac{U}{K} \cdot R \cdot C_m \cdot \left( 1 - e^{-\frac{\tau}{T_a}} \right) \right]$$

With electromechanical constant:

$$\tau_{em} = \frac{R \cdot J_{aw}}{K^2}$$

In order to obtain a soft landing at 80° flap position, the supply voltage is set at -9V to decelerate the mecanism. The initial condition is the speed given by the preceding speed trajectory and the final condition is a null speed at 80°. The motor speed during the deceleration time is:

$$\omega_m^d(\tau) = \left[ \frac{U}{K} \cdot R \cdot C_m \cdot \left( 1 - e^{-\frac{\tau}{T_a}} \right) \right]$$

And the speed motor during deceleration is:

$$\omega_m^\Delta(\tau) = \left[ \frac{U}{K} \cdot R \cdot C_m \cdot \left( 1 - e^{-\frac{\tau}{T_a}} \right) \right]$$

For the deceleration time, a simple method calculates the minimal position and the response time must be lower than the 150ms.

Then, the dynamic variables parameters are $T_a$ and $T_r$.

**Electrical energy**

Electrical energies are calculated such as for the forward:

$$E_{a\tau} = \frac{\tau}{2} \left( U - K_e \cdot \omega_m^a + \tau \right) dt$$

And the backward:

$$E_{d\tau} = \frac{-\tau}{2} \left( U - K_e \cdot \omega_m^d + \tau \right) dt$$

The objective minimizes the mean values of these two trips:

$$obj = \min(mean(E_{a\tau}, E_{d\tau}))$$

**Optimization**

Genetic Algorithm (GA) is a sequence of a selection, mutation and crossover of individuals from a population as described in [9] and [10]. An individual is a parameter vector composed by genes. A gene is a variable parameter of actuator such as presented on table 1. Elitism is used to conserve the best individuals of a generation. This strategy copies the best individual from generation $n-1$ into generation $n$. GA converges to the global best individual who defines the optimized actuator outer constraints and hypothesis.

Constraints and calculation hypothesis are:

- supply voltage 9V
- temperature 130°C
- ferrite magnet with a remanent flux density $B_r$ of 0.4 T and 1.05 relative permeability $\mu_r$
- 0.6 mm for the air gap thickness
- 5 A/mm² current density $J$ in slot section
- rotor radius is 3 times the rotor length
- a filling factor lower than 0.3.

To respect dynamic constraints and to avoid a too restrictive calculation, 5% tolerance on objective flap position is used during the calculation.

### Table 1. Variable parameters of electromechanical actuator

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Initial value</th>
<th>Min value</th>
<th>Max value</th>
<th>Optimal value</th>
</tr>
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<tbody>
<tr>
<td>$R_r$</td>
<td>mm</td>
<td>12</td>
<td>1</td>
<td>50</td>
<td>10,1</td>
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<td>$L_r$</td>
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<td>1</td>
<td>50</td>
<td>30,1</td>
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<tr>
<td>$e_a$</td>
<td>mm</td>
<td>4</td>
<td>1</td>
<td>8</td>
<td>4,7</td>
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<tr>
<td>$h_a$</td>
<td>mm</td>
<td>4</td>
<td>1</td>
<td>10</td>
<td>4,7</td>
</tr>
<tr>
<td>$d_{ma}$</td>
<td>mm</td>
<td>0.33</td>
<td>0.1</td>
<td>2.5</td>
<td>0.275</td>
</tr>
<tr>
<td>$n$</td>
<td>-</td>
<td>330</td>
<td>1</td>
<td>800</td>
<td>494</td>
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<tr>
<td>$T_a$</td>
<td>s</td>
<td>0,115</td>
<td>0,01</td>
<td>0,15</td>
<td>0,135</td>
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<td>s</td>
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<td>0,01</td>
<td>0,15</td>
<td>0,065</td>
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<tr>
<td>$Z_m$</td>
<td>-</td>
<td>11</td>
<td>9</td>
<td>17</td>
<td>11</td>
</tr>
<tr>
<td>$Z_r$</td>
<td>-</td>
<td>55</td>
<td>11</td>
<td>50</td>
<td>55</td>
</tr>
<tr>
<td>$Z_{z_r}$</td>
<td>-</td>
<td>11</td>
<td>9</td>
<td>17</td>
<td>11</td>
</tr>
<tr>
<td>$Z_{z_m}$</td>
<td>-</td>
<td>57</td>
<td>11</td>
<td>60</td>
<td>53</td>
</tr>
<tr>
<td>$m_{mr}$</td>
<td>mm</td>
<td>1,25</td>
<td>0.5</td>
<td>10</td>
<td>0.8</td>
</tr>
<tr>
<td>$m_p$</td>
<td>mm</td>
<td>2</td>
<td>0.5</td>
<td>10</td>
<td>1.25</td>
</tr>
<tr>
<td>$l_{m}$</td>
<td>mm</td>
<td>6</td>
<td>2.2</td>
<td>20</td>
<td>6,63</td>
</tr>
<tr>
<td>$l_i$</td>
<td>mm</td>
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<td>2</td>
<td>20</td>
<td>8.44</td>
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<tr>
<td>$l_p$</td>
<td>mm</td>
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<td>2</td>
<td>20</td>
<td>15.84</td>
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<td>$l_{l}$</td>
<td>mm</td>
<td>10</td>
<td>2</td>
<td>20</td>
<td>15.74</td>
</tr>
</tbody>
</table>

With initial values, analytical model calculate an actuator consuming a 5.85J mean electrical energy for forward and backward cycle. Optimal values were found by GA at the 2385th generation after 2h computation time. Global optimum is considered reached when the minimal electrical energy no longer varies after 150 generations.

**Results**

Optimized motor has a torque constant value of 16.8 mN.m/A, a 3.10 ohm electrical resistance value and a $3.42\times10^{-6}$ kg.m² rotor inertia value. Total inertia value is equal at $4.10^{-6}$ kg.m².

In the fig.4, for the forward trip, the maximal flap position is 80° at 150 ms and for the backward travel; the minimal flap position is -0.51° at 107 ms.

DC motor consumes 42W peak electrical power for the forward and 55W for the backward to decelerate the mechanism. The power electronic design could consider these values.

In the fig.5, electrical and mechanical energies are calculated. For the forward travel, the motor consumes 2.47J for the forward travel and 2.12J for the backward trip with a 24.1 gear ratio value. The mean values of electrical energy for forward and backward is 2.3 J.

In the fig.6, to check the electromagnetic behavior of the motor, Flux-2D is used with a magnetostatic resolution. The flux density in rotor teeth is equal to 1.0 T.

In analytical transient behavior, the motor speed maximal value reaches 3033 rpm in forward travel. In fig.7, at 3033 rpm, the motor torque is equal to 19.8 mN.m.

With the Flux-2D transient resolution, in the fig.8, simulation shows a ripple torque while torque values vary between 15 and 20 mN.m.
To verify the optimal DC motor, it is compared with some manufacturer DC motor, as presented in fig.1. Some performances are measured at 25°C and values are shown in the table 2. The optimal motor characteristics are close to the HC 677 ones.

Table 2. Motor DC performances comparison at 25°C

<table>
<thead>
<tr>
<th>Unit</th>
<th>HC</th>
<th>SQ2</th>
<th>HC</th>
<th>HF</th>
<th>Optim motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R$</td>
<td>$\Omega$</td>
<td>3.6</td>
<td>3.6</td>
<td>2.3</td>
<td>4.2</td>
</tr>
<tr>
<td>$K_t$</td>
<td>mN.m/A</td>
<td>17</td>
<td>18</td>
<td>20</td>
<td>19</td>
</tr>
<tr>
<td>$J_m$</td>
<td>kg.m²</td>
<td>1.39</td>
<td>1.57</td>
<td>3.4</td>
<td>1.45</td>
</tr>
<tr>
<td>$\tau_{em}$</td>
<td>ms</td>
<td>17</td>
<td>16</td>
<td>21.6</td>
<td>17</td>
</tr>
</tbody>
</table>

To respect dynamics, gear teeth are designed with the maximal torque in backward travel. To decelerate the mechanism, motor needs more current for the backward trip because of kinetic energy and the opposite load. As shown in fig.9, the motor needs 6.07A to develop 0.101N.m.

As presented in the fig 10, a gear design freeware Filengrene [11] can check the geometrical tooth of reduction gear, and operating conditions with calculated parameters.

From a geometrical point of view, the gear box was well designed as shown for the first reduction stage in fig.11.

**Conclusion**

This article proposes an analytical method to optimize a position actuator including the dynamic response and the spur gear set. After this first electromagnetic and dynamic approach, the machine design can be improved in order to refine the actuator as:
- influence of ripple torque on flap position,
- the self-heating coils on performance actuator.

To continue this optimization, a servo control as [12] is coupled with a position sensor on EGR valve. The next step is to realize a global optimization with power electronic design and feedback position in order to design an electromechanical system.

**REFERENCES**


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