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# Optimization of hydro-mechanical model of a multiway valve used in automotive thermal management\*

M. KHAMMASSI, J. AUBRY, B. BARBEDETTE, M. CORMERAIS, C. LAROUCI, F. BENOUEZDOU

**Abstract**—Improving fuel economy and reducing the Greenhouse-Gas emissions have been the determining factors in the development and implementation of innovative automotive technologies. Engine thermal management is one of the research fields that tries to solve these issues. Thermal management can be applied when controlling temperatures in different cooling circuits with an electronically actuated valve, which respects certain thermal management strategies. This control varies depending on several parameters such as temperature, load and engine rotational speed. This paper is part of a larger work that aims to improve the robustness of this valve design process since the early steps while minimizing the torque requirements of the DC-actuator. The first part of the paper will describe the mechanical concept of the valve and The second part will be focused on the hydro-mechanical model that is composed of four different sub-models that has been developed in order to simulate the multi-physical environment and so was validated based on experimental measurement. The third part is dedicated to discuss optimization results of a particle swarm based bi-objective optimization (PSO) loop applied on a simpler model, which helps to determine an optimal geometrical configuration. The optimal geometrical configuration found after running the optimization loop is to be tested in the multi-physical global model and results of the simulation are to be shown in future works.

## I. INTRODUCTION

Automotive manufacturers are currently searching for means to reduce engine emissions and improve fuel economy in order to satisfy global standards, which set maximum limits for exhaust emissions of vehicles. Since many years, considerable improvements have been made by reducing: the global weight of the vehicle, friction losses in the engine, aerodynamic drag on the body, rolling resistance of the tire/road interface, by trying new materials such as new metallic alloys or composite materials. In this paper, we are interested in thermal management of the engine, and more precisely, in coolant temperature regulation by an electronically controlled ‘smart’ valve.

A solution to improve engine thermal conditions and to reduce fuel consumption is through the thermal management. The objective is to make the engine run at its optimal working temperature as quickly as possible, as *Rouaud et al.* [1] showed the influence of the start temperature on the fuel consumption on a NEDC (New European Driving Cycle). Furthermore, for short journey's, it is estimated that nearly 80% of the emissions of unburned hydrocarbons and carbon monoxide, and the majority of NOx are produced during the first 200 seconds

after a cold start [2]. So, how fuel consumption and pollutants emissions could be reduced when applying a thermal management strategy?

### A. Active thermal management

The oil temperature is directly linked to the coolant temperature. By increasing it, the oil viscosity is decreased, which is key to reducing mechanical losses due to friction [2,4], especially in the first moments when starting from a cold engine. This decreases the fuel consumption and so the CO<sub>2</sub> emissions.

In this context, MANN+HUMMEL (automotive component supplier in thermal management) has brought an innovative approach to one of the most important engine part, the ACT Valve (Active Cooling Thermo-Management Valve) which delivers an important reduction in CO<sub>2</sub> emissions and greater fuel economy (up to 1.6% on average). The electronically controlled smart ACT Valve gives a new design to a traditional engine component which has always relied on wax thermostats (or calorstats), in order to control engine cooling [1]. However, wax thermostats have the inevitable problem of slow response time, due to the time of the thermal expansion of waxes as they melt, which has a direct effect on engine temperature regulation [5].

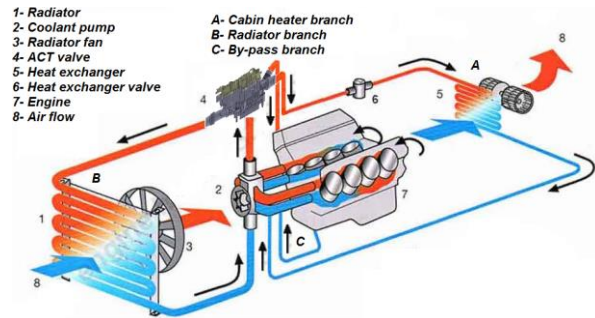


Figure 1. Typical external simplified coolant circuit

So making engine run at a higher coolant temperature, in a short warm up time, ensures a significant decrease of fuel consumption. The “no flow” position of the ACT valve guarantees a fast warm up time. In fact, when the engine is cold, the “no flow” valve position allows the engine to warm up faster, by keeping all valves closed, as the vehicle reaches the optimum operating regime faster.

The ACT valve is located on the engine breach as part of the coolant circuit, which is composed generally of three

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branches: cabin heater, radiator and bypass (Fig 1). This last branch is used to increase the temperature in the warm up phase as the coolant flows from the engine to the pump and flows back to the engine again in a closed loop.

### B. The design of the ACT valve

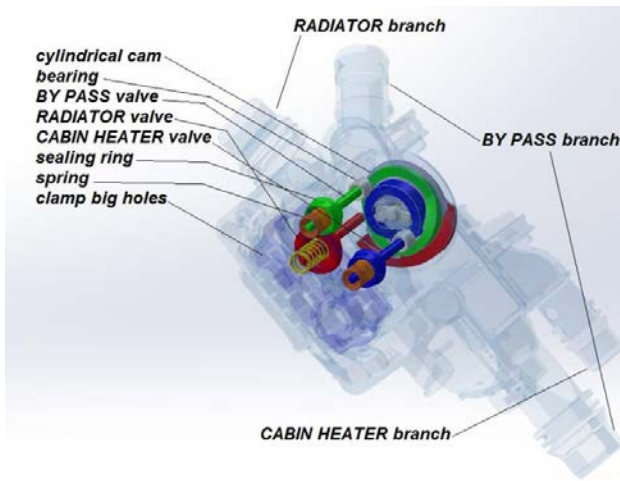


Figure 2. The ACT valve consisting parts

The ACT valve is mainly composed of a rotating cylindrical cam driving by a DC-actuator. It controls, accurately, the distribution and flow of the coolant in different branches of the cooling circuit including the engine and the interior heating/cooling system through valves that manage the opening and closing of branches. (Fig 2).

### C. The purpose of this work

This paper is part of a larger work that aims to improve the robustness of the ACT valve design process since the early steps while minimizing the torque requirements of the DC-actuator. Taking into account the multi-physical constraints of the operating environment such as water regain, mechanical load, fatigue, wear and clearances, but also of manufacturing constraints and process and geometrical tolerances related to the injection of thermoplastic parts. So in order to qualify the design process as robust, these uncertainties must be taken into account from the design phase to ensure the effectiveness and reliability of this valve over its lifetime.

This paper deals with a new hydro-mechanical model, which is composed of four different sub-models, a mechanical model, a hydraulic model, and two local models of contact and wear. Modeling has been developed in order to simulate the multi-physical environment and so was validated based on experimental data. The most interesting parameter is the torque required for the cam rotation because its maximum value corresponds to the torque requirement of the DC-Actuator. Finally, we have created a new and simpler model based on the global and more complicated one in order to use it in a particle swarm based optimization (PSO) process, to determine a robust optimal geometrical configuration.

*(Note for readers: All results presented are normalized to the maximal value, due to confidential context).*

## II. HYDRO-MECHANICAL MODELING

### A. The Global multi-physical model

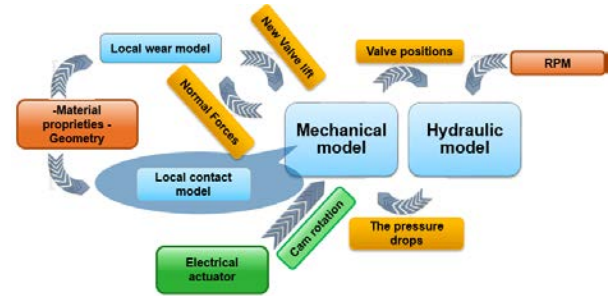


Figure 3. Flow chart for the global multi-physical model and its interactions

In order to ensure the physical coupling between the mechanical model and the hydraulic model, the coolant flow generates a pressure drop when passing through the act valve. This pressure exerts a hydro-mechanical force on the projected area of the valve. Depending on the engine speed and load, the hydraulic forces can reach 80% of contribution in the torque requirements [2].

The hydraulic model gives as output the pressure drops of the three valves according to their positions given by the mechanical model (Fig 3). The engine rotational speed in RPM are also included since the engine is coupled to the coolant pump and as the rotational speed increase, the pressure exerted by the coolant pump on the ACT valve is also increased.

The local wear model communicates with the mechanical model, as this last one gives as output the normal force exerted on top head of each valve, the wear model estimates the wear and give back new valve lift to the mechanical model.

The local contact model is implemented in the mechanical model and takes as input the material properties and the geometrical aspects of the contact (e.g. sphere to plane in our case). The electrical actuator is modeled as a constant velocity source.

#### 1) Mechanical model

All the mechanical important components were rebuilt to model the valve unit [3], which is the basic element of each ACT valve since the act valve is actually an association of several valve units distributed according to polar coordinates (radius  $r_i$ , angle  $\alpha_i$ ) (Fig 2). The modeling problem is then reduced to a single valve unit. This valve unit model was designed using the second generation of SimMechanics library (a part of Malab/Simulink/Simscape) the which is appreciated for modeling complex multi-physical systems. This software allows building digital environments based on physical equations of mechanics and it has already been used in [11, 17].

The mechanical model was fed with the same operational conditions as the tests performed on MANN+HUMMEL test benches. The mechanical model predicts the mechanical total resistive torque applied on the electrical DC actuator, taken in consideration all major mechanical parameters such as materials involved, friction coefficients, friction laws, geometric dimensions and shapes.

The mechanical total resistive torque applied on the electrical actuator is shown in the next figure (Fig 4).

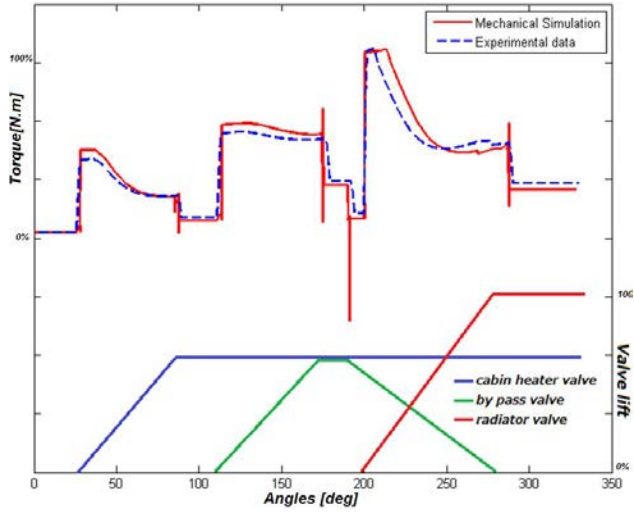


Figure 4. Experimental and computational mechanical resistive torque applied on the actuator associated with the lift computation for each valve.

The precision in the mechanical model was satisfying since there is approximatively the same order of magnitude and behavior. The horizontal gap between both simulation and experimental results is due to different geometric consideration. The vertical gap is due to difference between the friction coefficients taken into consideration and the real ones. The peaks in the simulation are due to numerical computational sources imposed by the contact library structure and limitation.

### 2) Local contact model

In order to describe the mechanical contact between two parts of the mechanical model, we have used SimMechanics Contact Forces Library, which is available on open source [6], this library contains contact 2D force models and force laws for multibody modeling in SimMechanics. Both linear and nonlinear force laws are included. The existing law force is the linear spring-damper model (known as *Kelvin Voight* contact model for others) [7] for the linear force law and the nonlinear force with an exponent on the penetration term. This contact force model was upgraded [3] in terms of precision and accuracy to a *Lankarani and Nikravesh* contact force model (1990), which is finally expressed as [8]:

$$F_N = \frac{4}{3 \left( \frac{1-\vartheta_p^2}{E_p} + \frac{1-\vartheta_s^2}{E_s} \right)} \sqrt{R_i} \times \delta^n \left[ 1 + \frac{3(1-c_e^2)}{4} \frac{\delta}{\delta^{(-)}} \right] \quad (1)$$

where

- $F_N$ : The normal contact force value;
- $c_e$ : Coefficient of restitution
- $\delta$ : The penetration value;
- $\dot{\delta}$ : The penetration velocity
- $\delta^{(-)}$ : The initial impact velocity
- $E_i$ : the young modulus
- $\vartheta_i$ : The Poisson ratio
- $E_i$ : the young modulus
- $R_i$ : The problem geometry
- $n$ : the Hertz coefficient

### 3) Local wear model

Various endurance tests showed clearly that wear is an important phenomenon to take into consideration in our multi-physical model since the valves undergo cyclical, repeated mechanical and thermal stresses.

The Global Incremental Wear Model (GIWM) [9] is the core of the local wear model, it was implemented and well explained in [3]. The GIWM was applied in each initial wear point towards the center of the valve head, as the normal load calculated in the global model is fed to the GIWM as it determines the new wear depth for each implementation on each direction. The simulation shown in (Fig 5) is for the cabin heater valve, for a complete rotation, where coolant density is  $1035 \text{ kg/m}^3$ , kinematic viscosity of the coolant is  $0.89 \times 10^{-6} \text{ m}^2/\text{s}$ , the test engine was a 1.6 liter diesel engine turbo-charged, in which the engine rotational velocity speed selected to perform the simulations and the tests is 4000 RPM.

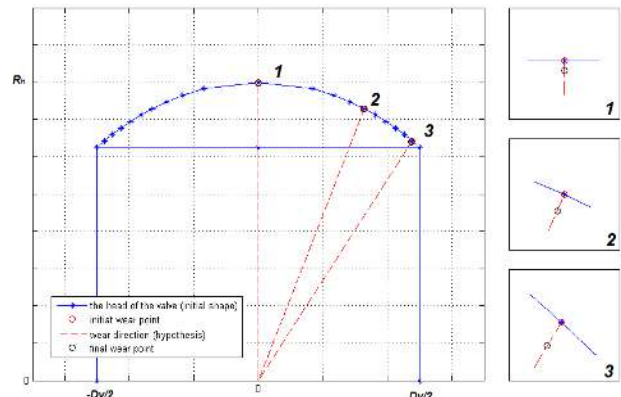


Figure 5. Wear model simulation results. (1, 2 & 3 are zoom in)

### 4) Hydraulic model

As already presented, the ACT valve controls different branches in the coolant circuit such as: the radiator branch, the cabin heater branch and the side circuit also known as the bypass branch (Fig 1). In order to model the coolant circuit all the important elements (heat exchangers, radiator, hoses and singularities ...) are modeled as hydraulic resistances in a turbulent flow [10].

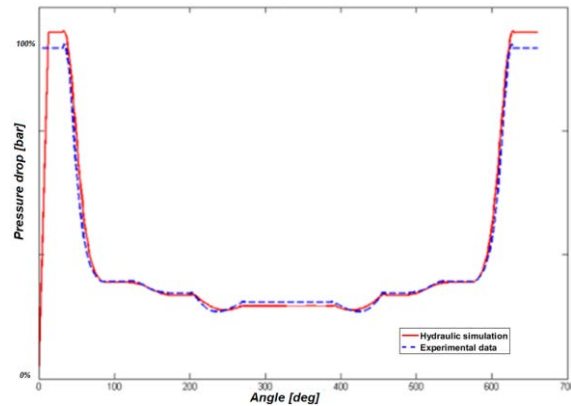


Figure 6. Pressure drop of the head of the cabin heater valve.

The global precision in the hydraulic model was satisfying since there is approximatively the same order of magnitude

and behavior. The gap between both simulation and experimental results is due to difference between the discharge coefficients taken into consideration and the real ones. In addition, there is a high sensitivity to changing hydraulic resistances

### III. DESIGN OPTIMIZATION OF THE ACT VALVE

#### A. Introduction

Due to the complexity of the multi-physical global model and to the enormous time that the simulation in a closed loop optimization might take using the global model (up to 100 000 iterations). We have created a new and simpler model based on the global and more complicated one, in which we neglect certain functional aspects of the valve, such as wear on the heads of the valves, contact, clearances, curved surfaces, and valves geometry (about 18 parameter). The optimal geometrical configuration found after running the optimization loop is to be tested in the multi-physical global model and results of the simulation are to be shown in future works.

The determination of a geometric configuration of the ACT-Valve must meet a number of constraints. Moreover, this geometry strongly influences the torque requirement of the DC motor. This geometry is controlled by many parameters and it is difficult to find by hand the configuration that meets both the specifications while minimizing the torque requirements.

The ACT Valve current design complies with customer specifications. However, it is possible to find other geometrical configurations that require a lower torque on the DC actuator. To find these other configurations, we propose to solve this problem as an optimization problem.

This requires explicit definition of variables, constraints and objectives.

We present the case of an ACT valve with three circuits: Cabin Heater, Radiator, and By-Pass circuits

#### B. Optimization problem

##### 1) Cam geometry

The set of these parameters is represented in (Fig 7)

- a) *Cabin heater valve*
  - Track radius  $R_{cabin}=[R_{cabin\ min}\cdot R_{cabin\ max}]$
  - Angular position of start of climb  $\theta_{cabin}=[\theta_{cabin\ min}\cdot\theta_{cabin\ max}]$
  - Slope of climb  $\alpha_{cabin}=[\alpha_{cabin\ min}\cdot\alpha_{cabin\ max}]$
  - Angular range of the positive slope portion  $\Delta\theta_{cabin}=[\Delta\theta_{cabin\ min}\cdot\Delta\theta_{cabin\ max}]$
  - Angular limitation  $\theta_{cabin\ limit}$
- b) *By-pass valve*
  - Track radius  $R_{BP}=[R_{BP\ min}\cdot R_{BP\ max}]$
  - Angular position of start of climb  $\theta_{BP}=[\theta_{BP\ min}\cdot\theta_{BP\ max}]$

- Positive slope of the climb  $\alpha_{BP+}=[\alpha_{BP+\ min}\cdot\alpha_{BP+\ max}]$
  - negative slope of the climb  $\alpha_{BP-}=[\alpha_{BP-\ min}\cdot\alpha_{BP-\ max}]$
  - Angular range of the full opening  $\Delta\theta_{BP}$
  - Angular range of the positive slope portion  $\Delta\theta_{BP+}$
  - Angular range of the negative slope portion  $\Delta\theta_{BP-}$
  - Angular limitation  $\theta_{BP\ limit}$
- c) *Radiator valve*
- Track radius  $R_{radia}=[R_{radia\ min}\cdot R_{radia\ max}]$
  - Angular position of start of climb  $\theta_{radia}=[\theta_{radia\ min}\cdot\theta_{radia\ max}]$
  - Slope of climb  $\alpha_{radia}=[\alpha_{radia\ min}\cdot\alpha_{radia\ max}]$
  - Angular range of the positive slope portion  $\Delta\theta_{radia}=[\Delta\theta_{radia\ min}\cdot\Delta\theta_{radia\ max}]$
  - Angular limitation  $\theta_{radia\ limit}$

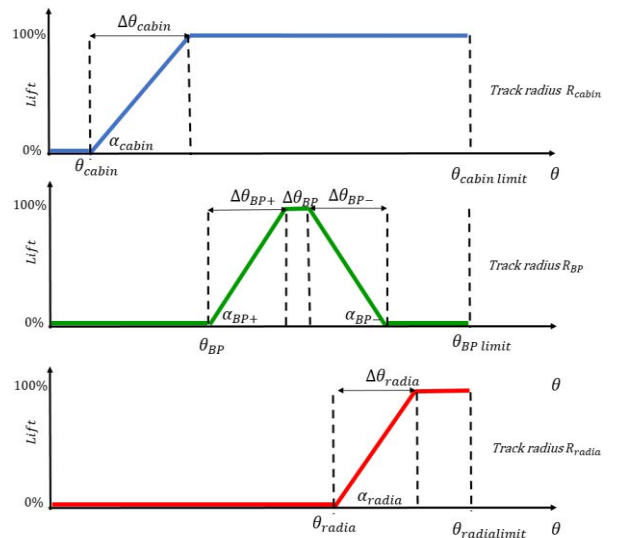


Figure 7. Cabin heater, By-pass and Radiator valves cam profil.

##### 2) Constraints of the problem

The constraints can be classified in two different categories

- a) *Geometric constraints*
  - The cam must have a sufficiently wide angular portion during which all valves will be closed:
$$\theta_{cabin\ min} \geq S_1 \quad (2)$$
  - The cam must have a sufficiently wide angular portion during which the cabin heater valve is fully open and the other valves remain totally closed:

$$\theta_{cabin} + \Delta\theta_{cabin} + S_2 < \theta_{BP} \quad (3)$$

- The lowest track radius must remain above a minimum radius value, due to the assembly solution:

$$(R_{cabin\ min}, R_{BP\ min}, R_{radia\ min}) \geq S_3 \quad (4)$$

- Track radius must be at least at a certain distance apart.

$$R_i - R_j \geq S_4 \quad i,j: \{cabin, BP, radia\} \quad (5)$$

- The largest radius must remain less than a maximum radius

$$R_{cabin, BP, radia\ max} \leq S_5 \quad (6)$$

$S_i$ : Boundary parameter: This parameter is set to the maximal or the minimal value that respects the constraints listed above.

b) *Hydraulic constraints*

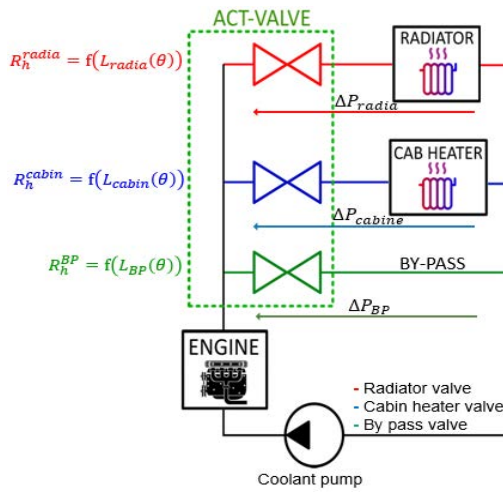


Figure 8. Hydraulic circuit diagram

The hydraulic resistances of the valves in the fully open position must remain less than customer requirements:

$$R_h^{cabin}, R_h^{BP}, R_h^{radia} \leq CR_1 \quad (7)$$

The hydraulic resistance in the fully closed position must remain above a minimum value:

$$R_h^{closed} \geq CR_2 \quad (8)$$

$CR_i$ : customer requirements

$CR_i$ : Customer requirements

3) *Optimization objectives*

a) *Torque requirement minimization*

As presented in [3], the total torque required is the minimal torque that the actuator must provide in order to rotate the cam and can be expressed as follows:

$$C_{TOTAL}(\theta) = (\sum_i F_i(\theta) \times R_i \times [\tan(\alpha_i(\theta)) + f]) + C_{Tj}(\theta) + C_0 \quad (9)$$

with

$$F_{i(\theta)} = P_c + k \times L_i(\theta) + \Delta P_i(\theta) \times A_i \quad (10)$$

where

$C_{TOTAL}$ : total resistive torque

$F_{i(\theta)}$ : normal load due to the spring and the hydraulic forces

$C_{Tj}$ : resistive torque created by the cam support plan

$C_0$ : Constant resistive torque created by the O ring seal

$R_i$ : valve track radius

$f$ : friction coefficient

$P_c$ : spring preloading

$k$ : spring stiffness

$\Delta P_i$ : pressure drops

$A_i$ : valve projected area

$\alpha_i$ : Slope of climb

$L_i$ : lift

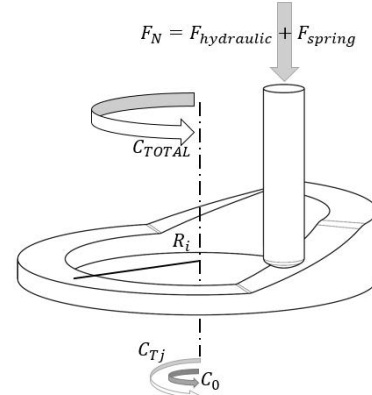


Figure 9. Total torque requirement calculation schema:  $F_N$  vertical force applied on the contact point,  $F_{spring}$  spring force applied on the contact point

As shown in Fig 9, the torque depends on the angular position of the cam. It is the maximum value of this torque on a complete rotation of the cam that we are seeking to minimize.

b) *The packaging (maximal volume occupied)*

As a start, we chose to link the packaging requirement to the maximum value of the largest track  $\max(R_i)$ . Therefore, it is this last value that we will also seek to minimize.

4) *Solving the optimization problem*

As stated, this optimization problem is a bi-objective problem. We chose to solve it using a heuristic algorithm based on the particulate swarm method [13] adapted to treat bi-objective problems. This adaptation is inspired by [14] and has already been used in [15,16,12]. The size of the swarm has been set at 1000, and 500 iterations were chosen. To ensure a good convergence of our results, the process was run 10 times. The results discussed below.

## IV. ANALYSIS AND INTERPRETATION

After running the optimization algorithm 10 times, it proposed several geometrical configurations to reduce the torque requirements on the DC-actuator and it converges towards an optimal solution which results will be presented in this paper. The optimization algorithm presents a new geometrical configuration as it changes the initial parameters of the cam and the arrangement of its tracks (Fig11). In fact, it has reduced the cabin heater lift by 17% and put it on the second cam track. It also reduced the bypass lift by 9%, reduced the first slope by 17%, increased the second one by 25% and put the By-pass configuration on the third track.

Finally, it proposed to reduce the radiator lift by 33%, the radiator slope by 32% and put it on the first cam track (Fig 10, 11).

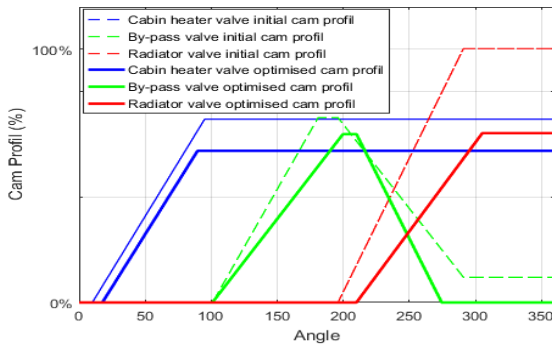


Figure 10. New suggestions for the cam profile

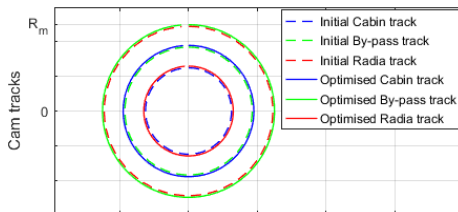


Figure 11. Tracks rearrangement

These changes caused a significant decrease of the torque requirements on the DC-actuator, 34% reduction on the maximum torque requirements (Fig. 12).

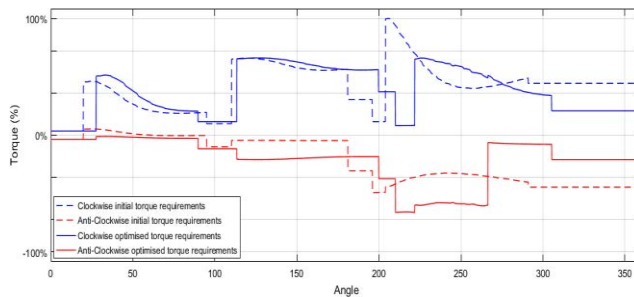


Figure 12. Cam torque requirement before and after the optimization

This new geometric suggestion proposed by the optimization algorithm constitutes a first step in the resizing the electrical actuator. The next step will be to implement this configuration on the global multi-physical model then to test it experimentally to validate this optimization and justify the probable actuator resizing.

## V. CONCLUSION

This work helped to ensure the effectiveness and reliability of this valve over its lifetime, first by developing a multi-physical core model then by optimizing the geometrical cam configuration in a simpler model in order to predict new design suggestions based on a particle swarm based optimization (PSO) loop. Only cam optimization results were presented here. In future works the optimization will include valve geometrical parameters. The suggestions of the optimization algorithm has brought a 34% reduction in the maximum torque requirement. The next step will be to

evaluate experimentally the geometrical configuration recommendations of the optimization algorithm then to seek ways to integrate these new configurations in the design process.

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