

Profile Optimization of the Teeth of the Double Rack-and-Pinion Gear Mechanism in the MCE-5 VCRi Engine

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ABSTRACT

MCE-5 DEVELOPMENT has been developing its variable compression ratio engine (VCRi) for over a decade aiming at reducing fuel consumption and pollutant emissions. In order to transmit power from the piston (combustion) to the crankshaft, the MCE-5 VCRi technology is based on three innovative components: a gear wheel and two racks. This gear mechanism ensures a very low friction compared to other continuous VCR solutions based on bearings. However, this transmission is used in nonstandard conditions: the direction of rotation is reversed repeatedly, and the parts are submitted to high and rapidly varying loads. To avoid interferences and alteration caused by high contact pressure at high load, and ensure a regular transmission at low load, the profile of the teeth is carefully considered. A crowning shape is placed on the teeth in the direction of the gear axis, and a correction is applied to the tooth active profiles in the area of tooth roots and tooth tips. A Design of Experiment (DOE) is realized on a light Finite Element 2D model in order to highlight the influence of the parameters of the correction on the maximum stress and the contact pressure on the teeth, and on the speed ratio between the gear and racks. A refined Finite Element 3D model is developed in order to obtain more accurate results concerning the contact pressure and the stress on the teeth, and to take into account the crowning shape. Endurance experiments were conducted on engine test benches before and after optimization of the tooth profile. Observations of the teeth show the benefice of this method, with a suppression of the wear and pitting on the surface contact.

CITATION: Duchemin, M. and Collee, V., "Profile Optimization of the Teeth of the Double Rack-and-Pinion Gear Mechanism in the MCE-5 VCRi Engine," *SAE Int. J. Engines* 9(3):2016, doi:10.4271/2016-01-1013.

INTRODUCTION

In the MCE-5 VCRi engine, the power transmission from the piston (combustion) to the crankshaft is ensured by three innovative components: a gear wheel and two racks (see [Figure 1](#)). This transmission is used in nonstandard conditions:

- The up and down movement of the piston makes the direction of rotation of the gear wheel reverse constantly
- The combustion cycles apply high and rapidly varying loads to the parts.

In order to keep a constant centre distance between wheel and racks, these parts have 2 rows of teeth, separated by a track on the symmetry plane. The widths of the teeth are also varying, depending on the maximum load they must endure. [Figure 2](#) shows the combustion rack and the wheel of this gear mechanism.

This configuration gives them very specific stiffness. Therefore, the profile of the teeth must be carefully considered in order to:

- avoid interference and alteration caused by high contact pressure at high loads
- ensure a regular transmission at each load level
- avoid rattle noise

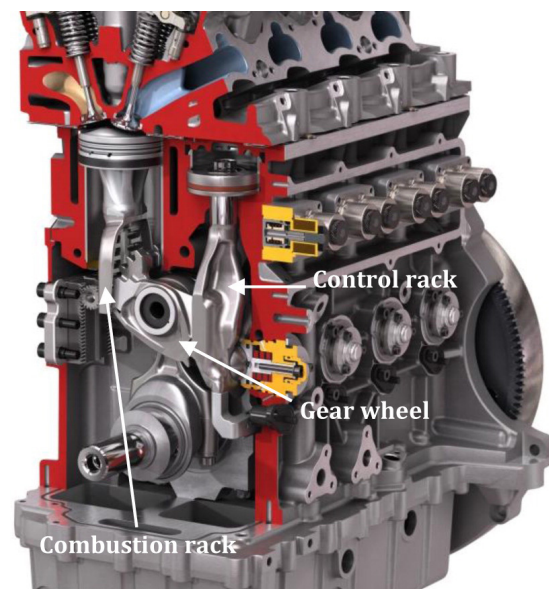


Figure 1. MCE-5 VCRi 4 cylinders engine

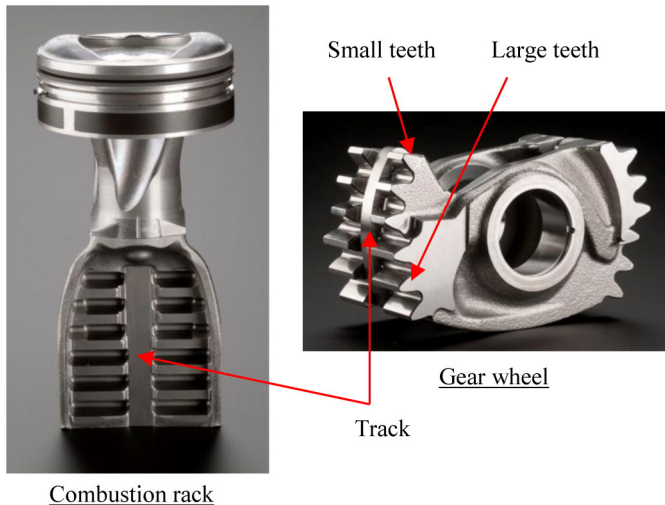


Figure 2. Gear mechanism of the VCRi engine on the combustion side

The 2 possible profile modifications on teeth are:

- a correction applied to the tooth active profile in area of tooth roots and tooth tips
- a crowning shape in the direction of the gear axis

Initially, these profiles had been determined using an off the shelf specialized gear analytical software. Yet, this software was limited to wheel gears, and was not able to calculate rack gears accurately enough. Moreover, the peak cylinder pressure in the VCRi engine was originally of 85 bars, but in the recent years, it has been increased to 120 bars. Therefore, the teeth correction previously calculated is not adapted anymore. As a consequence, on some endurance tests, plastic deformation, micropitting and PSO macropitting (see Figure 3), has been observed on the wheel teeth.

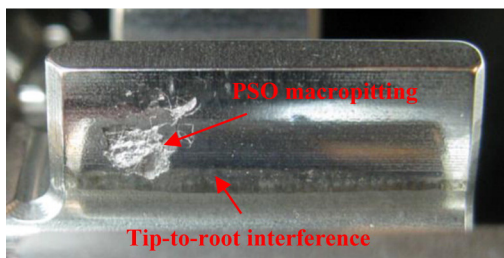


Figure 3. Tip-to-root interference and PSO macropitting on a tooth

This type of damage has been described by Errichello (1). His study shows that the pitting is initiated at the upper edge of the tip-to-root interference, where the damage is caused by corner contact, i.e. the contact between the edge of a tooth and the root of the corresponding tooth. In other words, the damage is caused by high Contact Pressure due to corner contact. His study describes the damage process but does not provide solutions. Over the past few years, a lot of studies have focused on the effect of a tooth correction on the Transmission Error, but very few on the induced contact pressure, and none of them have been identified to deal with rack gears. Rigaud (2) deals with estimation of static transmission error and mesh stiffness variations of spur and helical gears, with no correction. The static and dynamic Transmission Error was studied by Podzharov (3). But for the VCRi

wheel/racks system, the speed rotation changes constantly and rapidly, so the dynamic Transmission Error is of no interest. Kissling (4) analyses the effect produced by short linear, long linear, short arc-like, long arc-like and fully crowned profile corrections on gears with different transverse contact ratio. Even if some results are displayed for the Contact Pressure, this study is mainly focused on the Transmission Error. Beghini (5) presents a new tip relief profile constructed as a classical linear profile modified with a parabolic fillet, but does not give results for the Contact Pressure.

Amongst the solution to calculate a tooth profile, the simple 2D mapping procedure is the most commonly used. But most of the usual dedicated software calculates the transmission characteristics under fixed load and fixed speed. Additionally, due to the specific design of the VCRi parts, the stiffness of each tooth is different. These solutions are therefore not applicable to the VCRi wheel/racks system. Palmer (6) indicates that a 3-D Finite Element Method is more flexible and gives better accuracy on the results. Figure 4 displays a 3D Finite Element model confirming the interference between the tip of a rack tooth and the root of a wheel tooth.

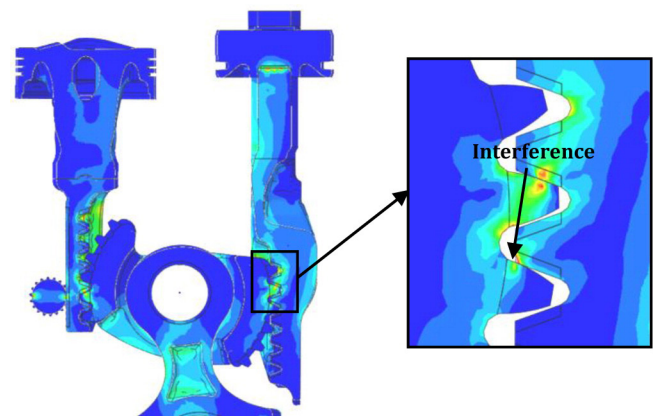


Figure 4. Interference observed on a 3D Finite Element explicit model

Previously, a Finite Element Calculation would require hours to complete. Due to improvements in computer power and FEA solver performances, the calculation can be reduced to minutes, thus enabling to use this method to carry out a complete study. This paper presents a method of profile optimization using Finite Element Analysis, performed in 2 steps. First, the influence of the correction parameters (length, depth, shape, position) on the Transmission Error, Contact Pressure and Maximum Principal Stress on the tooth root are studied on a fast 2D Finite Element Model. Then a fine tuning is realized on the contact pressure, taking into account the crowning shape in the direction of the gear axis, on a 3D Finite Element Model. To conclude, the result is validated by an endurance test on a single-cylinder engine.

INFLUENCE OF THE CORRECTION PARAMETERS

Correction Parameters

The tooth is defined as an involute profile, expressed in a cylindrical system (r, θ, z) :



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$$\theta = \frac{180}{\pi} \left(\frac{\sqrt{r^2 - r_b^2}}{r_b} - \arccos\left(\frac{r_b}{r}\right) \right) \quad (1)$$

where r_b is the base circle radius.

The gear correction can be placed on the tip and/or on the root of the gear teeth and/or the rack teeth. In a prototyping phase, for machinability reason, the gear teeth correction is easier to obtain than the rack teeth correction. Hence, for this parametric study, it was chosen to place the correction only on the tip and root of the gear teeth. These corrections can be either linear or parabolic. The parabolic profile is known to be more efficient with varying loads than the linear profile. Therefore, only the parabolic profile is studied here. Thus, the studied parameters are restrained to the length L_t and the depth D_t of the tip relief and the length L_r and the depth D_r of the root relief.

The analytical equations of the profile in a cylindrical system (r, θ, z) are:

$$\text{Tip relief:} \quad \theta = \frac{180}{\pi r} \frac{D_t}{L_t^2} (r - R_t + L_t)^2 \quad (2)$$

$$\text{Root relief:} \quad \theta = \frac{180}{\pi r} \frac{D_r}{L_r^2} (R_r + L_r - r)^2 \quad (3)$$

where R_t and R_r are respectively the maximum and minimum altitudes of the tooth profile, as shown in Figure 5.

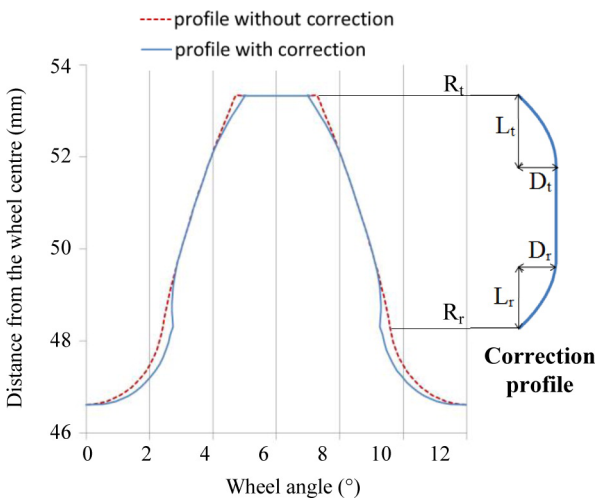


Figure 5. Tooth profile and applied correction

2D Finite Element Model

The 2D Finite Element Model enables to have a very fine mesh with a very low computation time (less than 2 mn). In this model, a load is applied on the rack and a rotational displacement is imposed to the gear wheel (see Figure 6).

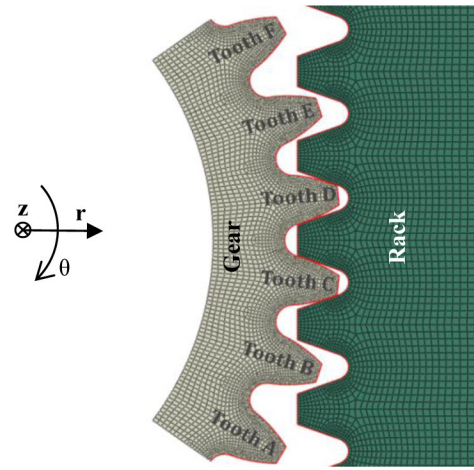


Figure 6. 2D Finite Element Model for the Design of Experiment

A comparison between this model and a 3D Finite Element Model gives a good correlation of the behaviour of the gear transmission. The maximum principal stress on the root of the teeth is 8% higher on the 2D model. The contact pressure is 10% lower, and does not take into account the edge effect and the crowning of the tooth. The deformation of the teeth, as well as the load distribution between the different teeth is very well correlated. Hence, if this 2D FE Model is not precise enough to give absolute results, it is sufficient to highlight the influence of the tooth profile.

Each tooth profile is meshed automatically, using the involute profile. The teeth correction profile is added to that original profile. This fast FE Model enables to conduct a Design of Experiments to quantify the influence of the correction parameters on the gearing characteristics.

Design of Experiments Criteria

Because of the damage due to corner contact observed on the gear tooth, the first criterion for the Design Of Experiments is the maximum Contact Pressure (CP). The second criterion is the maximum Principal Stress (σ_{max}) at the root of the teeth to minimize the risk of fatigue failure of the gear teeth.

In this FEM calculation, the position of the wheel is imposed. Hence, the Transmission Error is defined as the difference between the effective and the ideal positions of the racks with reference to the wheel. The Transmission Error is usually used to evaluate the whining noise in gears, by calculating its Fast Fourier Transform. This relation is true when dealing with gears used in quasi-constant speed condition. In the wheel/racks transmission of the VCRI engine, the speed varies rapidly. As a consequence, the FFT of the Transmission Error is not representative of the transmission noise. The shocks that occur during gearing are more relevant. Therefore, the second derivative of the Transmission Error (d^2TE) is chosen as criterion. Considering that the load also varies rapidly during the engine cycle, the shocks at low load must be minimized as well as the shocks at high load. Therefore, 2 criteria are used separately: the second derivatives of the Transmission Error at low (d^2TE_L) and high load (d^2TE_H).