Journal of Fundamental and Applied Sciences

ISSN 1112-9867

Available online at http://www.jfas.info

Research Article

INFLUENCE OF COEFFICIENT OF FRICTION ON THE PERFORMANCE OF PARALLEL GEARS

M. Aroudj[1], F. Inel[2]

[1]Mechanical Engineerind Department,University 20Aout 1955 Skikda

[2] Mechanical Engineerind Department,Automatic Laboratory, University 20Aout 1955 Skikda

Received: 21 June 2019 / Accepted: 28 August 2019 / Published online: 01 September 2019

ABSTRACT

Gears performance is a very essential value that concluded the success of our system so researchers have worked hard to find mathematical relationships to calculate performance. Between these relationships we have the J.F. Debongnie method and the f.VILLE and VELEX method.

In this work the yield of the parallel gears was calculated with the change of the modulus and the number of teeth using the J.F. Debongnie method and comparing them with the f.VILLE and VELEX method and associating them with the experimental results to be studied which is more close to the experimental results and the effect of the parameters selected in each method.

After the comparison we found that the coefficient of friction the greatest impact on the performance whenever the lack of friction coefficient the greater the performance and the values of the J.F. Debongnie method higher than the values of the f.VILLE and VELEX method of the effect of the loss factor because it also affects the performance the greater the coefficient of loss whenever the performance.

Keywords: gear; performance; coefficient of friction; rotation speed; comparison

Author Correspondence, e-mail : aroudj.mohammed91@gmail.com

doi: http://dx.doi.org/10.4314/jfas.v11i3.6

1. INTRODUCTION

Gear power transmissions are, without doubt, the best compromise from a point of view of efficiency [1] are used as power transmission means [2], such as automobiles, industrial equipment, aircraft, helicopters and ships. These power transmission elements are often exploited at high speeds and / or torque, so their dynamic analysis becomes a major problem due to the durability of the gears and the control of vibrations and noise [3,4].

For about a century, has been the subject of much research. Among all the possible architectures, the cylindrical gears in involute of circle are undoubtedly the most used for the transfer of rotation between parallel axes because relatively easy to manufacture, they have mathematically simple geometries whose dimensional control is relatively easy. The numerous studies concerning them have in particular led to important sets of standards [5].

This improvement essentially involves taking into consideration the physical phenomena that affect the efficiency of the system. A synthesis of the different sources of power loss in a gear transmission was presented by Hoehn et al. **[**6].

The authors concluded that, regardless of the lubricant and the surface finish, a reduction in losses can be achieved by optimizing the geometry of the gears. Generally, two main contributions are involved in the total loss of power in a gear transmission [7,8].

The bute in this works the efficiency of calculation the performance of the gears by which method of J.F. Debongnie or method of f.VILLE and VELEX

2. METHODOLOGYS

In this section we study two pairs of gears example straight teeth IDEFIX 401 and example gearbox B.A.C.V In each case we apply two methods are the method of method of J.F and method of f.VILLE and VELEX To calculate the performance, and compare them.

2.1. Cylindrical toothed gear with straight teeth IDEFIX 401

Table 1 shows the gears used in straight teeth IDEFIX 401.

Table 1. Geometries of the gear wheels [9]

	pinion	*wheel*
number of Z teeth	20	20
tooth width [mm]	30	30
pressure angle [°]	20	
helix angle [°]	0	
module [mm]	10	
protrusion coefficient	1	
offset coefficient	0	
hollow coefficient	1,4	
surface roughness (Rms) [µm]	0,63	

We need some missing data from the previous table so we did the meshing simulations represented in the table on the ANSYS program to complete the remainder

Primitive diameter:

$$d_1=d_2=m.z = 10 .20 = 200 \text{ mm}$$

Head diameter:

$$d_{a1} =d_{a2} = m (z+2) = 10. (20+2)= 10.22 = 220 \text{ mm}$$

Fugure 1 shows the simulation of the gears used in straight teeth IDEFIX 401 on the program ANSYS Supplement missing data

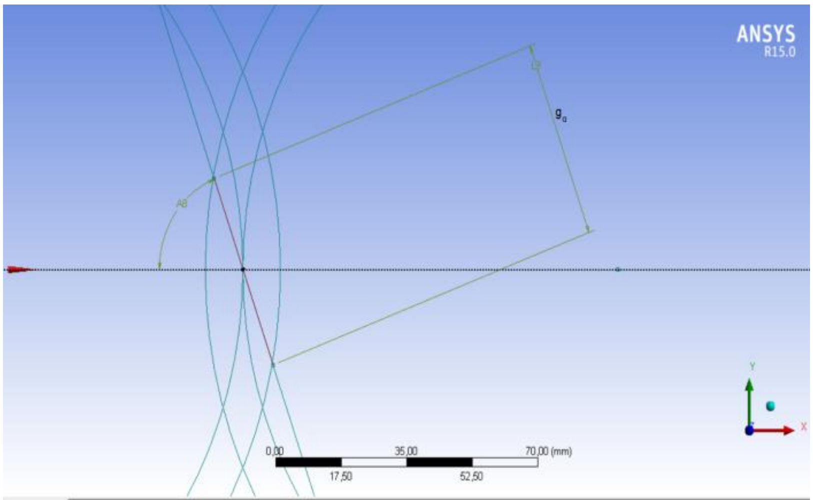

Fig.1. Graphical representation of contact line $g_{\alpha(L9)}$

The contact line ga is geven by [10]

$$g_\alpha = g_f + g_a \tag{1}$$

In this case we have:

$$g_f = g_a = 22{,}964 \text{ mm}$$

$$Z_1 = Z_2 = 20$$

$$\text{So} : u = \frac{Z_1}{Z_2} = 1$$

$$P_b = P \cdot \cos\alpha \tag{2}$$

$$P = \pi .m \Rightarrow P_b = \pi .m \cdot \cos\alpha = 3{,}14 \cdot 10. \text{ Cos } 20 = 29{,}506 \text{ mm}$$

$$\varepsilon_1 = \varepsilon_2 = \frac{g_f}{P_b} = \frac{22{,}964}{29{,}506} = 0{,}778$$

$$\varepsilon = \varepsilon_1 + \varepsilon_2 = 0{,}778 + 0{,}778 = 1{,}556$$

a) First method J.F. Debongnie

This method is based on the work produced by the friction force onWork produced by turn and the gears dimensions

the equation (3) is given by **[11]**

$$\eta = 1 - \frac{W_{fr}}{W_{Ft1}} = 1 - \mu \frac{u \pm 1}{u} \frac{\pi}{Z_1} \left(1 + \varepsilon_1^2 + \varepsilon_2^2 - \varepsilon\right) \tag{3}$$

μ : Coefficient of friction;
u : Reduction ratio;

$$u = \frac{Z_1}{Z_2} \tag{4}$$

Z_1, Z_2 : Number of wheel tines, pinion;
ε_1 : Partial driving report1;

$$\varepsilon_1 = \frac{g_f}{P_b} \tag{5}$$

ε_2 : Partial driving report 2;

$$\varepsilon_2 = \frac{g_a}{P_b} \tag{6}$$

ε : Total driving report;

$$\varepsilon = \varepsilon_1 + \varepsilon_2 \tag{7}$$

Taking coefficient of friction f =[0.04;0,08] [11]

$$\eta = 1\text{- }0{,}06 \ \frac{1+1}{1}\frac{3{,}14}{20}\ (1 + 0{,}778^2 + 0{,}778^2 - 1{,}556) = 0{,}987$$

$$\eta = 98{,}76\ \%$$

Table 2 shows change the values of parfarmonscessis by changing the friction coefficient by J.F. Debongnie method

Table 2. The results of the returns according to the coefficient of friction

μ	*0.08*	*0.07*	*0.06*	*0.05*	*0.04*
η (%)	98,35	98,56	98,76	98,97	99,17

Curve of the Fig.2 shows behavior the performance η (%) in Table 2. according to the coefficient of friction μ in same table.

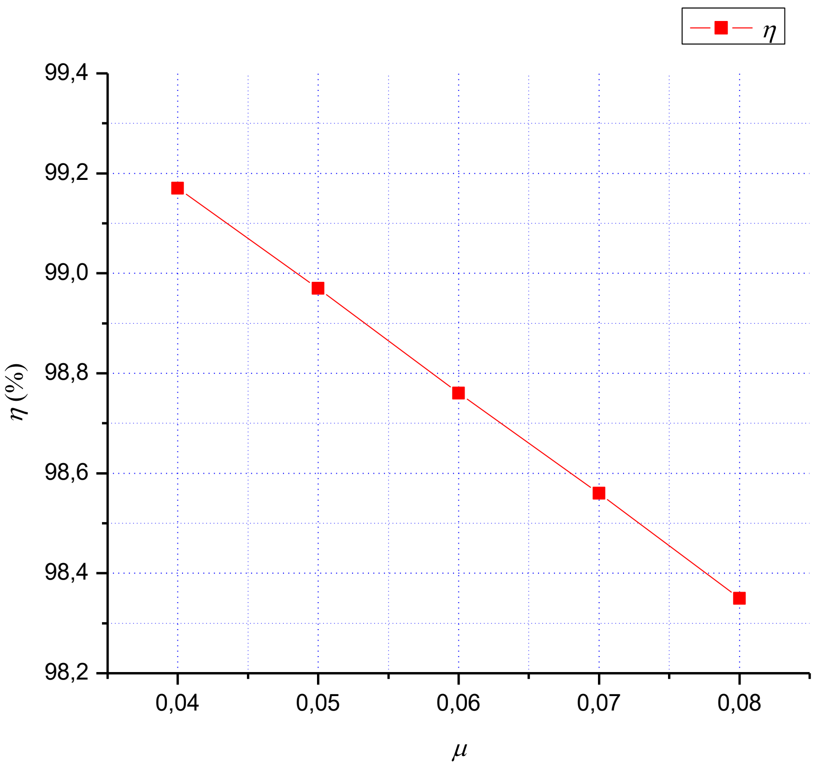

Fig.2. Curve of the performance η (%) according to the coefficient of friction μ

b) Second method f.VILLE et VELEX

This method is based on the accuracy of the toothbrush's manufacture, its dimensions, lubrication viscosity and the depth of the denture surface treatment

the equation (8) is given by [12]

$$\rho = 1 - f(1+u)\frac{1}{\cos\beta_b}\frac{\pi}{Z_1}\ \varepsilon_\alpha x \Lambda p \tag{8}$$

f : Coefficient of friction ;

u : Reduction ratio ;

$$u = \frac{Z_1}{Z_2} \tag{9}$$

Z_1, Z_2 : Number of teeth of the wheel, pinion;

ε_α : Driving report;

Λp: Loss factor. [12]

taking coefficient of friction f =[0.04;0,08] [11]

ρ pour $\Lambda p = 0.55$

$$\rho = 1 - 0.06(1+1)\frac{1}{\cos 0}\frac{3.14}{20}\, 1.556\text{x}0.55 = 0.9838$$

$$\rho = 98.38\%$$

Table 3 shows change the values of performance by changing the friction coefficient by f.VILLE et VELEX

Table 3. The results of the returns according to the coefficient of friction

f	*0.08*	*0.07*	*0.06*	*0.05*	*0.04*
$\rho(\%)$	97,85	98,11	98,38	98,65	98,92

Figure 2 shows behavior the performance ρ(%) in Table 3. according to the coefficient of friction f in same table.

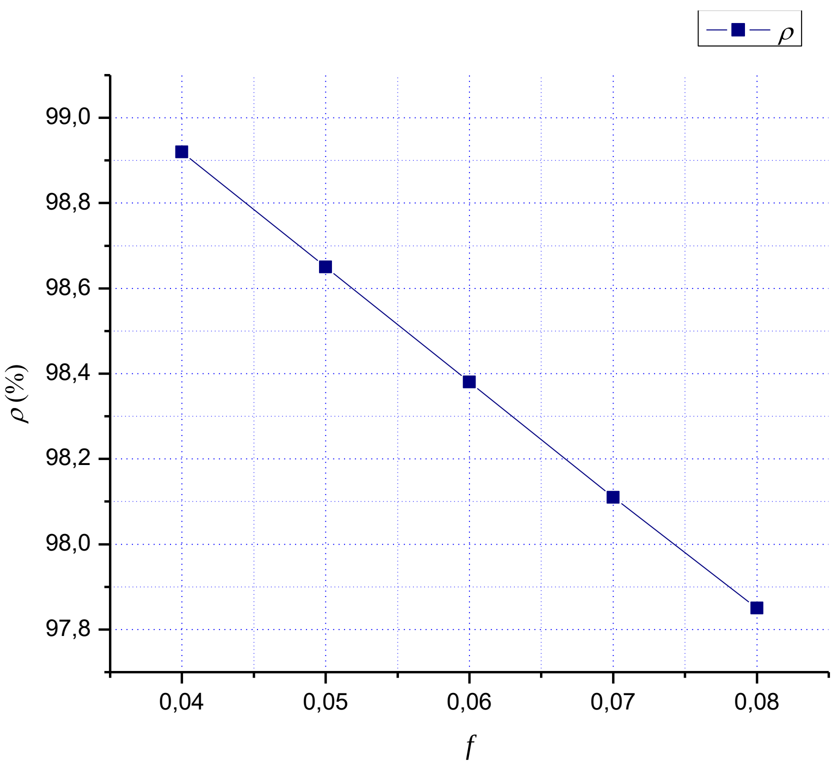

Fig.3. Curve of the performance ρ (%) according to the coefficient of friction f

The figures 4 illustrate the comparison between Fig.2. and Fig.3. The value of the module is equal to 10, where performance increases with decreasing friction coefficient as it has a direct effect on the performance of the gears, taking into account that the curve η is the results of the J.F. Debongnie equation and the curve ρ is the result of the f.VILLE et VELEX.

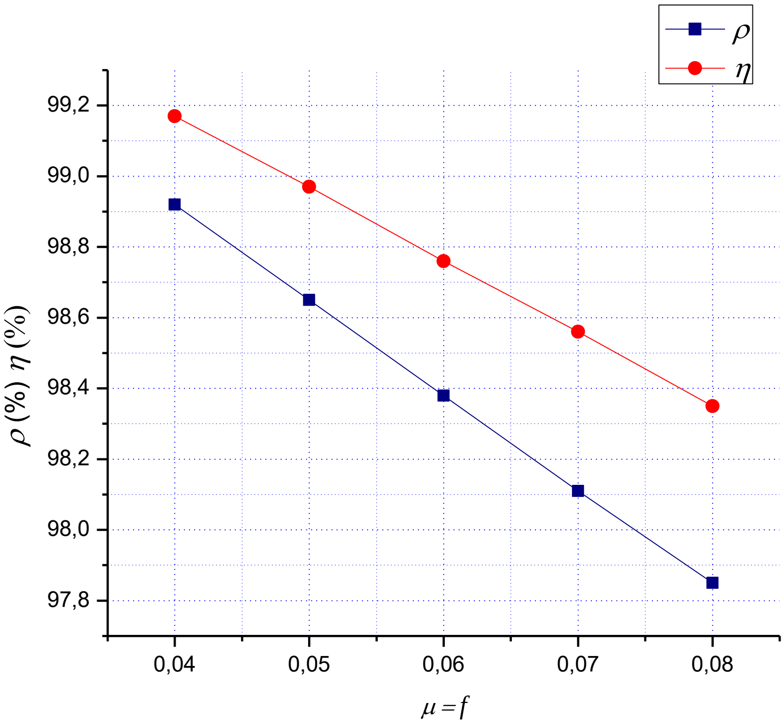

Fig.4. Comparation of Figure 2 and Figure.3

2.2. Cylindrical wheel with straight teeth having a modulus of 4 the gearbox B.A.C.V

Table 4 shows the gears used in gearbox B.A.C.V

Table 4. Geometries of the gear wheels.[9]

	pinion	*wheel*
Number of teeth	26	157
tooth width [mm]	50	40
pressure angle [°]	20	
helix angle [°]	0	
module [mm]	4	
protrusion coefficient	1	
hollow coefficient	1,4	
offset coefficient	0,16	-0,16
spacing [mm]	366	
surface roughness (Rms) [µm]	0,63	

Calculate the diameters:

Primitive diameter: $d_1 = m.z_1 = 4\ .\ 157 = 628$ mm

$d_2 = m.z_2 = 4\ .\ 26 = 104$ mm

Head diameter: $d_{a1} = m\ (z_1+2) = 4\ .\ (157+2) = 636$ mm

$d_{a2} = m\ (z_2+2) = 4\ .\ (26+2) = 112$ mm

Figure 5 shows the simulation of the gears used in gearbox B.A.C.V on the program ANSYS Supplement missing data

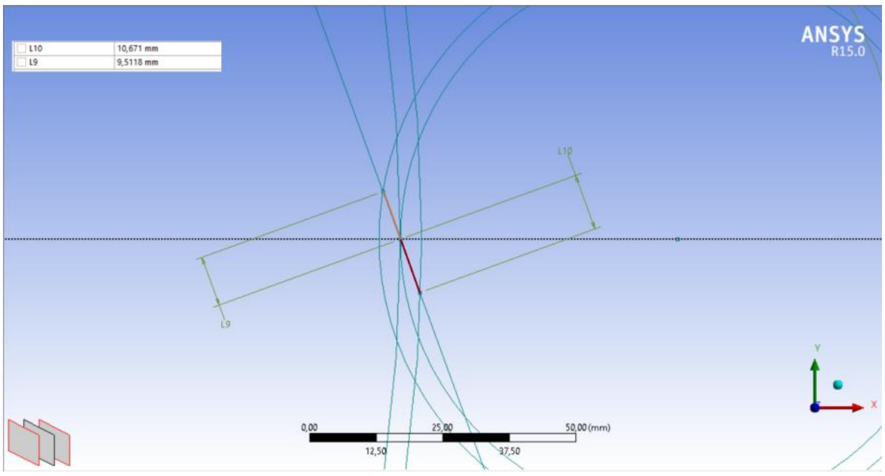

Fig.5. Graphic Representation $g_{a(L10)}$ et $g_{f(L9)}$

$$P = \pi .m \Rightarrow P_b = \pi .m . \cos \alpha = 3{,}14 . 4 . \cos 20 = 11{,}802 \text{ mm}$$

$$u = \frac{Z_1}{Z_2} = \frac{157}{26} = 6{,}038$$

$$\varepsilon_1 = \frac{g_f}{P_b} = \frac{9{,}512}{11{,}802} = 0{,}806$$

$$\varepsilon_2 = \frac{g_a}{P_b} = \frac{10{,}671}{11{,}802} = 0{,}904$$

$$\varepsilon = \varepsilon_1 + \varepsilon_2 = 0{,}806 + 0{,}904 = 1{,}71$$

a) First method J.F. Debongnie

Applied to equation number 3

Taking coefficient of friction f =[0.04;0,08] [11]

$$\eta = 1\text{- } 0{,}06 \; \frac{6{,}038+1}{6{,}038} \; \frac{3{,}14}{157} \; (1 + 0{,}806^2 + 0{,}904^2 - 1{,}71) = 0{,}998$$

$$\eta = 99{,}89 \ \%$$

Table 5 shows change the values of parfarmonscessis by changing the friction coefficient by J.F. Debongnie method

Table 5. The results of the returns according to the coefficient of friction

M	*0.08*	*0.07*	*0.06*	*0.05*	*0.04*
η (%)	99,85	99,87	99,89	99,91	99,92

Figure 6 shows behavior the performance η (%) in Table 5. according to the coefficient of friction μ in the same table.

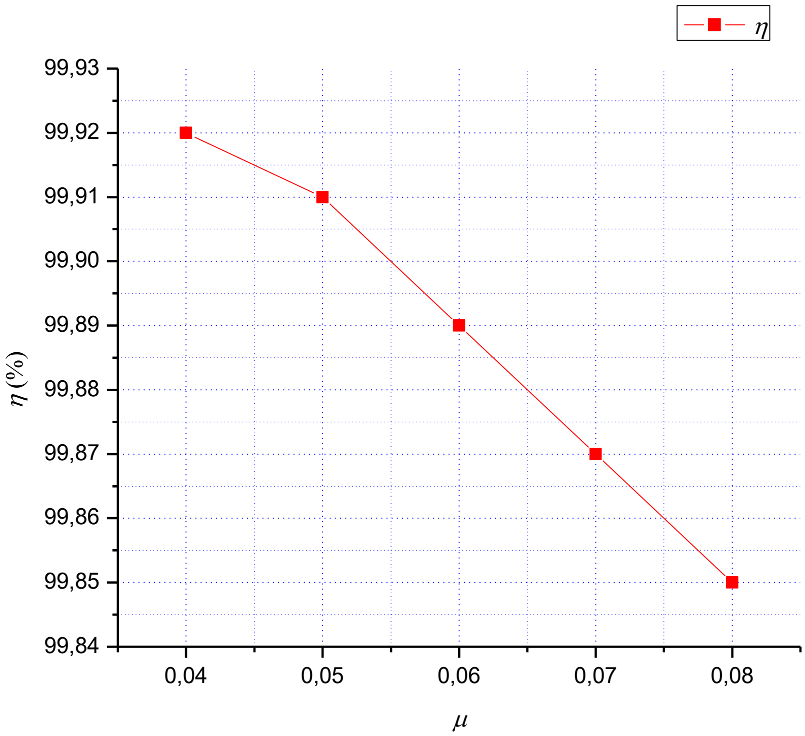

Fig.6. Curve of the performance η (%) according to the coefficient of friction μ

b) Second method f.VILLE et VELEX

Applied to equation number 8

Taking coefficient of friction f =[0.04;0,08] **[11]**

ρ pour Λp = 0.55

$$\rho = 1 - 0.06(1 + 6.038)\frac{1}{\cos 0}\frac{3.14}{157}\, 1.71\mathrm{x}0.55 = \mathit{0.9920}$$

$$\rho = 99.20\%$$

Table 6 shows change the values of parfarmonscessis by changing the friction coefficient by f.VILLE et VELEX

Table 6. The results of the returns according to the coefficient of friction

f	*0.08*	*0.07*	*0.06*	*0.05*	*0.04*
ρ(%)	98,94	99,07	99,20	99,33	99,47

Figure 7 shows behavior the performance ρ (%) in Table 6. according to the coefficient of friction f in same table

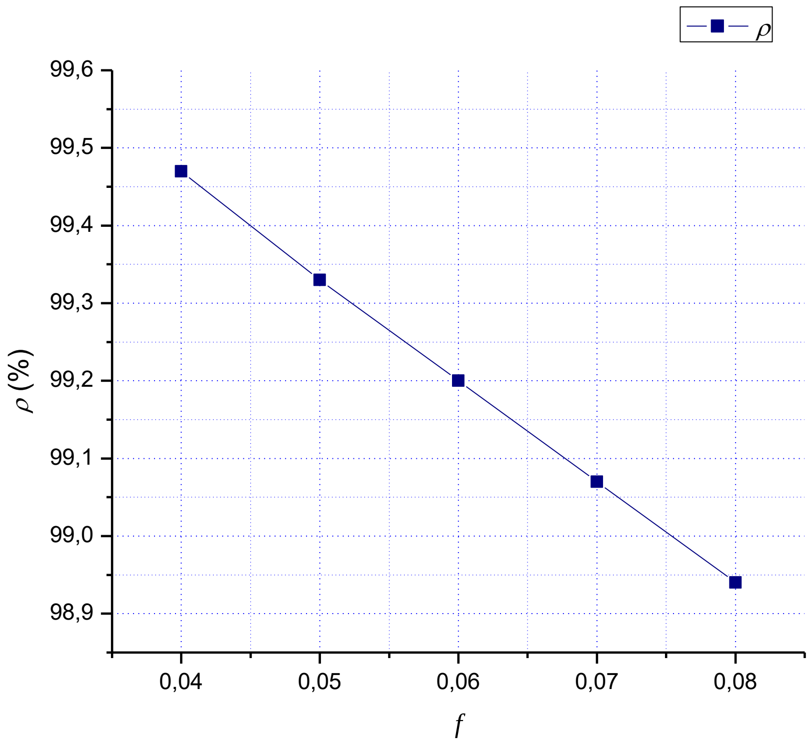

Fig.7. Curve of the performance ρ (%) according to the coefficient of friction f

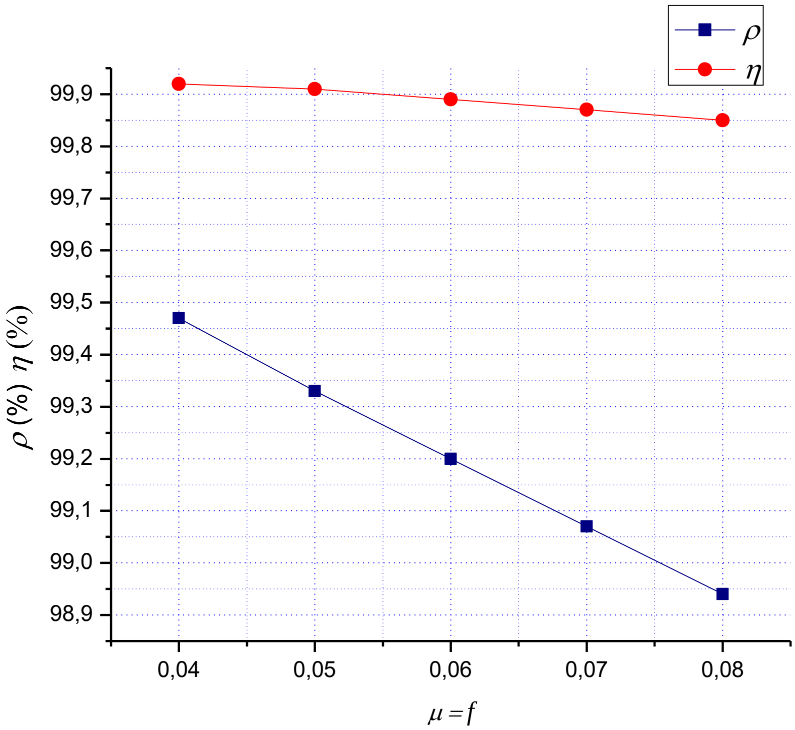

Fig.8. Compare the Fig .6 and the Figure 7

The curves in Figure 8 show a comparison between Fig.6. and Fig.7. The value of the module is equal to 4,where performance increases with decreasing friction coefficient as it has a direct effect on the performance of the gears, taking into account that the curve η is the results of the J.F. Debongnie equation and the curve ρ is the result of the f.VILLE et VELEX.

3. EXPERIMENTAL

3.1. Cylindrical toothed gear with straight teeth IDEFIX 401:

Table 7 shows experimental results of Performance in terms of speed of rotation of straight teeth IDEFIX 401.

Table 7. The results of the returns according to the speed of rotation **[9]**

N (tr/min)	*500*	*1000*	*1500*	*2000*	*2500*
ρ(%)	99,27	99,43	99,47	99,52	99,51

Figure 9 shows the shape of the Performance changes in terms of speed of rotation

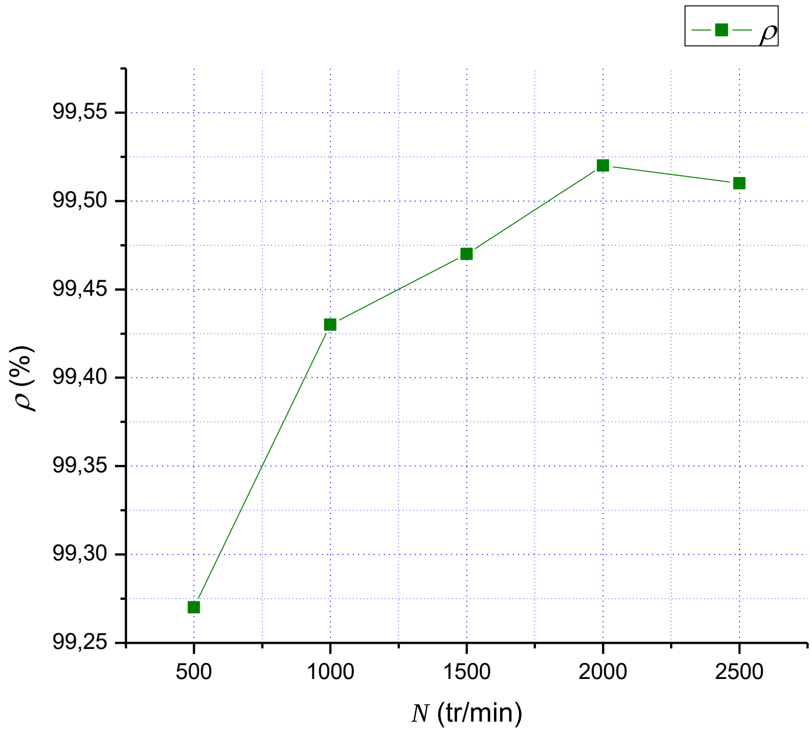

Fig.9. Curve performance ρ according to the speed of rotation N

3.2. Cylindrical wheel with straight teeth having a modulus of 4 the gearbox B.A.C.V

Table 8 shows experimental results of Performance in terms of speed of rotation of gearbox B.A.C.V

Table 8. The results of the returns according to the speed of rotation **[9]**

N (tr/min)	*1000*	*2000*	*3000*	*4000*	*5000*	*6000*
ρ(%)	99,47	99,63	99,68	99,72	99,70	99,69

Figure 10 shows the shape of the Performance changes in terms of speed of rotation.

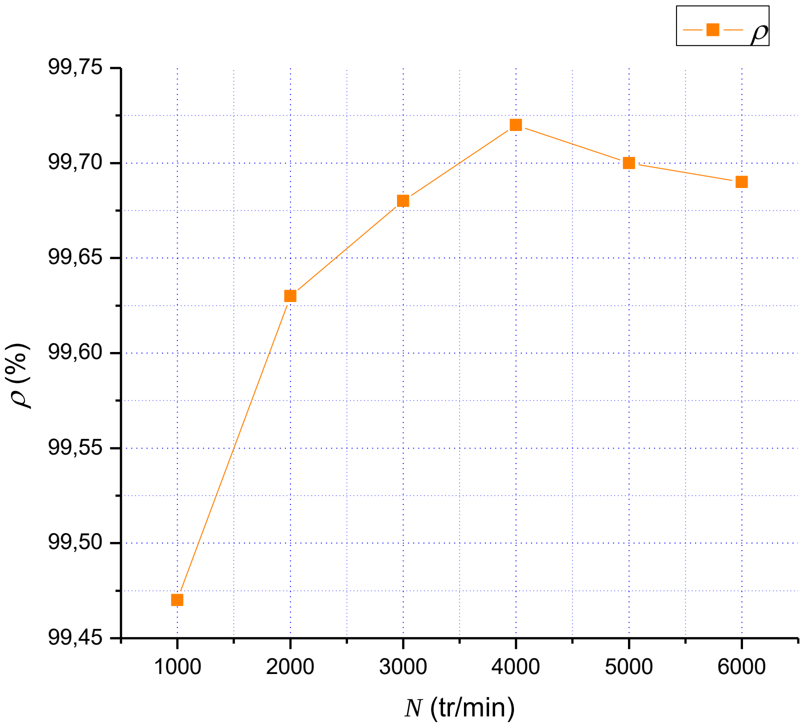

Fig.10. Curve performance ρ according to the speed of rotation N

4. CONCLUSION

After the study, we concluded that the friction coefficient had a reverse effect on the performance. The more the decrease and the more the deficiency increased, the loss factor also had a negative effect on the performance according to f.VILLE et VELEX The material of the manufacture and the depth and accuracy of the correction all have a direct effect.

When we observe the experimental results, we find that the performance have a direct effect from the speed of the rotation, which in turn has an impact on the speed of friction, means that the higher the speed of the rotation of the coefficient of friction to increase the performance and vice versa, but the validity depends on the extent of the load gears of the maximum speed, We may be counterproductive.

6. REFERENCES

[1] Thaer O, Simulation de l'usure et d'avaries sur des dentures d'engrenages cylindriques - Influence sur le comportement statique et dynamique de transmissions par engrenages, 2012

[2] Bechkri B, ANALYSE DES DÉFAUTS DE DENTURE ET LEURS INFLUENCES SUR

LES TRANSMISSIONS MÉCANIQUES PAR ENGRENAGES, 2007

[3] Stewart, R.M. *The specification and developpement of a standard gearbox monitoring*. 2nd International Conference on Vibration in Rotating Machines, Cambridge, 1980, p. 353-358.

[4] Sidahmed, M., Garnier, C. *Détection de défauts dans les engrenages*. CETIM information, 1991, n° 124, p. 71-74.

[5] Ghribi D, OPTIMISATION DES CORRECTIONS DE FORME DANS LES ENGRENAGES DROITS ET HELICOIDAUX - APPROCHES DETERMINISTES ET PROBABILISTES-, 2013

[6] Höhn, B. R., Michaelis, K., Wimmer**, A.,** "*Low Loss Gears*". AGMA Fall Technical meeting, Paper No. 05FTM11, 2005.

[7] Maifi T, ANALYSE EXPERIMENTALE DE LA FATIGUE DES ENGRENAGES A DENTURE DROITE, 2012

[8] Sellami S, ANALYSE DES FACTEURS INFLUANT SUR LA PRECISION CINEMATIQUE DES TRANSMISSION PAR ENGRENAGES, 2010

[9] Diab Y, Analyse des pertes de puissance dans les transmissions par engrenages à grande vitesse. Applications aux réducteurs industriels et aux machines textiles, 2005

[10] Johan PASCAL MESURE ET PRÉDICTION DES DISTORSIONS DES ENGRENAGES TRAITÉS PAR INDUCTION, 2012

[11] J.F. Debongnie, CONCEPTION ET CALCUL DES ÉLÉMENTS DE MACHINES VOLUME 3 Version 1, juin 2006 .pp 133-137

[12] Velex P, Ville F**.** An Analytical Approach to Tooth Friction Losses in Spur and Helical Gears—Influence of Profile Modifications, 2009, doi: 10.1115/1.3179156

How to cite this article:
Aroudj M, Inel F. Influence of coefficient of friction on the performance of parallel gears. J. Fundam. Appl. Sci., 2019, *11(3), 1135-1150.*