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# Effect of Coating Material on Wear in Internal Gears

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## ABSTRACT

Theoretical and experimental investigation of wear during coupling in internal gears coated with various polymeric coating materials was performed in this study. In the theoretical part of the study, Archards' wear formulation was adapted to internal gears and wear behavior in various conditions was determined. Moreover, a fatigue and wear testing apparatus having similar working principle with FZG (Forschungsstelle für Zahnrader und Getreibbau) closed circuit power circulation system was designed and manufactured to experimentally investigate the wear in internal gears. Internal gear-pinion couples manufactured from St50 material were coated with various polymeric materials, namely PTFE (polytetrafluoroethylene), $MoS_2$ bonded with polyamide, $MoS_2$ bonded with epoxy in the experimental study. An uncoated internal gear was also investigated to find out the performance of coated gears. Variation of wear depth on tooth profiles of internal gears were determined theoretically and experimentally. Theoretical and experimental studies showed that polymeric coated internal gears have more wear resistance than uncoated ones by means of high lubrication ability and low friction coefficient of coating materials. It was also observed that high corrosion resistance of polymeric coatings protected metallic surfaces and decreased corrosive wear.

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**NOMENCLATURE**

| | | | |
|---|---|---|---|
| $a$ | Axle offset (mm) | $N$ | Rotation cycle number (-) |
| $a_H$ | Hertz contact area (mm$^2$) | $P_1$, $P_2$ | Hertz pressures of pinion and internal gear (MPa) |
| $b_{1,2}$ | Face width of pinion and internal gear (mm) | $P_{p,(n-1)}$ | Pressure at point p at rotation cycle number n-1 (MPa) |
| $d_{b1}$, $d_{b2}$ | Tip circle diameters of pinion and internal gear (mm) | $s_{p1}$, $s_{p2}$ | Sliding distances of point p at pinion and internal gear (mm) |
| $d_{g1}$, $d_{g2}$ | Base circle diameters of pinion and internal gear (mm) | $U_1$, $U_2$ | Peripheral velocities of pinion and internal gear (m/s) |
| $d_{o1}$, $d_{o2}$ | Pitch circle diameters of pinion and internal gear (mm) | $v$ | Velocity (m/s) |
| $h_{p,n}$ | Wear depth at point p at rotation cycle number n (mm) | $w$ | Angular velocity (rad/s) |
| $i$ | Gear ratio (-) | $x_{1,2}$ | Profile offset factor of pinion and internal gear (-) |
| $k$ | Wear coefficient (-) | $z_1$, $z_2$ | Tooth numbers of pinion and internal gear (-) |
| $m$ | Module (mm) | **Greek Symbols** | |
| $n$ | Revolutions per minute (rpm) | $\alpha_o$ | Pressure angle (°) |

## 1. INTRODUCTION

Internal gears are widely used in national defense and aerospace industries as external sun gears of planetary mechanisms due to their compact structure, large torque-to-weight ratio, high gear ratio, reduced noise and vibration, etc. [1]. They are especially preferred where gearbox is required to be placed into a small space like planet gear mechanisms, transmission boxes, differential housings, cranes, hoists, automotive and aerospace industries due to short gear axle offset. Internal gears differ from external gears in respect to the orientation of the gear teeth into the gear center. Internal

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gears having concave tooth profile mate with external gears having convex tooth profile which provides some advantages such as low sliding velocity, low contact stress and high gear ratio [2]. Manufacturing of internal gears is more difficult than external gears. Thus, it is necessary to determine the working conditions of internal gears carefully. Several studies have been carried out on internal spur gears. Tooth profile geometry of the internal gears and tooth root stresses have been extensively investigated in the past studies. Certain programs were developed to ease the manufacturing process of the gear in the studies [3-5] made to determine the geometry of internal gears. By means of developed programs, internal gear geometry can be easily changed to adjust tooth force, tooth number and gear axle offset. Optimum internal gear appropriate for working conditions can be manufactured in this way. However, these programs are based on theoretical calculations without experimental verification. Effect of gear rim thickness on the tensile and compression stresses in gear root were examined in the studies [6-8] made to determine optimum rim thickness of internal gears. There are some studies [9-12] made to calculate tooth root stresses of internal gears. In these studies, place and magnitude of maximum tangential stress were determined in the tooth root of internal gears. Analytical formulations, finite element simulations, strain-gauge and photoelasticity based experimental procedures were used in stress measurements in these studies.

Rupture of small particles from the contacting surfaces of relatively moving parts which is known as wear is the most common damage mechanism on the contacting surfaces of mechanical systems [3, 13, 14]. Wear typically occurs on the contact surfaces of gear teeth due to sliding friction. In gears, two curved surfaces are in linear contact which leads to Hertzian shear stresses reaching very high values [14]. Initiation and progression of micro-pitting on the flanks of gear teeth ends up with contact fatigue damage [15]. Micropitting is generally ascribed to the stress field associated with the roughness of the contacting surfaces [16]. Mathematical modeling of wear phenomenon was first suggested by Archard [17]. Thereafter, these formulas were used in various external gear mechanisms to theoretically determine the wear behavior by Flodin and Andersson [18-23].

Contact stiffness has an important role in the design of mechanical elements [24], especially in gears transmitting high power with small contact area. Coating of gears using various coating materials is a general method for improving the crush and fatigue strength by increasing surface hardness and/or quality [15, 25, 26] as well as reducing friction coefficient [27]. Coating materials are used for increasing the corrosion strength, for preventing the discontinuities such as scratches and pores in the structure, for decreasing the friction coefficient between the contacting surfaces and for gaining higher load capacity to the parts [28, 29]. Coating of gears increases wear resistance and strength by decreasing surface toughness (and/or increasing surface hardness). In this study, it is aimed to investigate wear in the contact region of the tooth profile theoretically and experimentally by using various coating materials. Theoretical procedure for wear depth determination is given in Figure 1.

## 2. WEAR MODEL IN INTERNAL GEARS

Conjugate action starts with the contact of driving gears' (pinion) tooth root with driven (internal) gears' tooth tip and ends with separation of driving gears' tooth tip from driven gears' tooth root in a gear pair (Figure 2). Contact region is under rolling-sliding effect. Sliding is dominant in the beginning of contact and is the main reason of wear. Thus, tooth root of pinion and tooth tip of internal gear are the most critical regions in terms of wear.

Sliding velocities of pinion and internal gear are equal with opposite directions in the pitch circle (Figure 2.b). Thus, there is only rolling between meshing tooth pair in pitch circle. Gear load is subjected to single tooth pair in the pitch circle. At the end of contact, sliding occurs between pinions' tooth tip and internal gears' tooth root (Figure 2 (c)). Gear load is shared by two tooth pairs at the end of contact.

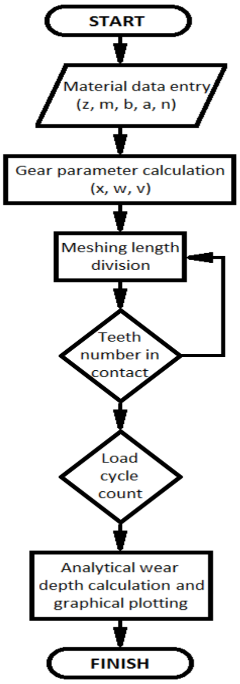


**Figure 1.** Procedure for wear depth determination

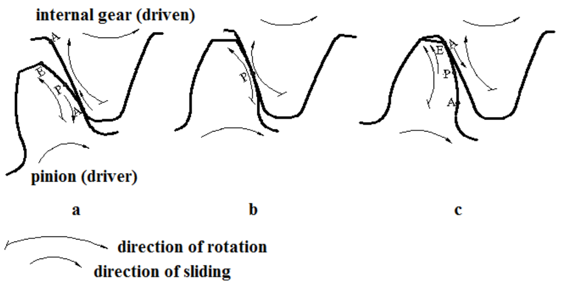


**Figure 2.** Tooth contact mechanism: (a) in the beginning, (b) in the pitch circle, (c) at the end [30]

Effect of rolling and sliding differs in each point of contact. Thus, wear has to be investigated not during the contact, but in different points of contact, individually. Equation (1) known as Archards' wear formulation can be used to calculate wear at a point p as follows;

$$h_p = \int_0^s kPds \tag{1}$$

where $h_p$ is the wear depth in point $p$, $s$ the sliding distance between two contacting surfaces, $k$ the wear coefficient and $P$ the regional contact pressure. According to Anderssons' wear model, by applying "singular point observation method [19]" to contacting gear pairs, namely expressing wear of any contacting point of tooth profiles of pinion-internal gear during coupling depending on rotation cycle count, wear depth at a point $p$ after $n$ cycle count can be expressed in Equation (2) as follows:

$$h_{p,n} = h_{p,(n-1)} + kP_{p,(n-1)} s_p \tag{2}$$

where $h_{p,n-1}$ is the wear depth of same point in the previous cycle, $P_{p,n-1}$ the pressure at point $p$ in the previous cycle and $s_p$ the sliding distance of point $p$. Distance of points on gear teeth from each other depending on the contact position during the coupling of pinion and internal gear pair is given in Figure 3.

Two opposite points ($p_1$, $p_2$) from the contacting teeth of pinion and internal gear during the coupling were considered to determine the sliding distance at contact points. These points were investigated in three different positions during the coupling. In the first position, $p_1$ and $p_2$ coincide (Figure 3.a) in the beginning of contact. In the second position; when pinions' point ($p_1$) is exiting coupling, internal gears' point ($p_2$) is still in contact region. Thus, there is a distance of $s_{p1}$ between $p_1$ and $p_2$ (Figure 3.b). In the third position, when internal gears' point ($p_2$) is exiting coupling, there is a distance of $s_{p2}$ between $p_1$ and $p_2$ (Figure 3.c). Internal gears' point ($p_2$) moves a distance of $2a(U_2/U_1)$ when pinions' point ($p_1$) moves a distance of $2a$ (Hertz contact length) along the contact length in the contacting teeth pair where $U_1$ and $U_2$ are peripheral speeds of pinion and internal gear along the contact line, respectively. Similarly, pinions' point ($p_1$) moves a distance of $2a(U_2/U_1)$ when internal gears' point ($p_2$) moves a distance of $2a$ along the contact length in the contacting teeth pair.

Sliding distance is the distance between $p_1$ and $p_2$. Thus, sliding distances between these two points for pinion and internal gear can be written in Equations (3) and (4), respectively as follows:

$$s_{p1} = 2a_H \left(1 - \frac{U_2}{U_1}\right) \tag{3}$$

$$s_{p2} = 2a_H \left(1 - \frac{U_1}{U_2}\right) \tag{4}$$

By substituting Equations (3) and (4) in Equation (2), wear depth equations for pinion and internal gear can be written in Equations (5) and (6), respectively as follows:

$$h_{p,n} = h_{p,(n-1)} + kP_{p,(n-1)} 2a_H \left(1 - \frac{U_2}{U_1}\right) \tag{5}$$

$$h_{p,n} = h_{p,(n-1)} + kP_{p,(n-1)} 2a_H \left(1 - \frac{U_1}{U_2}\right) \tag{6}$$

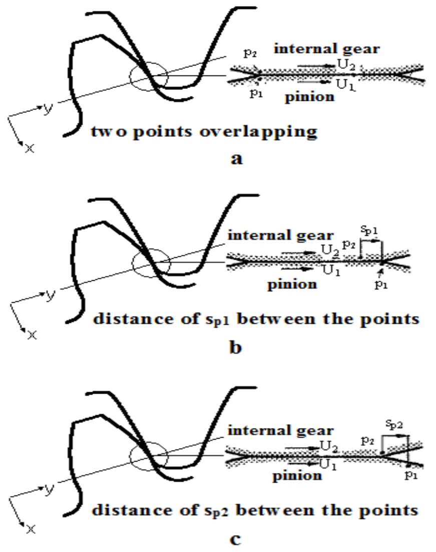


**Figure 3.** Distance of points on gear teeth from each other during coupling [20]

Pinion and internal gears' peripheral speeds ($U_1, U_2$), Hertz pressure on the contact points of the teeth profiles ($P$) and Hertz contact area of tooth profiles ($a_H$) described in the previous work [31].

## 3. EXPERIMENTAL STUDY

### 3. 1. Gears and Coating Materials

Pinion and internal gear pair used in the experimental studies were made of St50 steel having surface hardness of 160~170 HB. Properties of the gear pair were given in Table 1. Subscript "1" was used for pinion and subscript "2" was used for internal gear in the table.

In the experimental studies, three different coating materials, namely PTFE, $MoS_2$ bonded with polyamide and $MoS_2$ bonded with epoxy were used to investigate the wear of teeth surfaces during the contact length of pinion and internal gear. Powders of polymeric coating materials and bonders if required were applied to cleaned gear surfaces via thermal spraying method. Technical properties of coating materials used in the study were given in Table 2.

### 3. 2. Experimental Setup

A pinion-internal gear fatigue and wear testing apparatus having the same working principle with the FZG [32, 33] closed circuit power system was manufactured to perform wear tests (Figure 4). The apparatus which allows investigation of wear in various load and speed conditions seen in Figure 4 consists of two gear boxes having same gear ratio. One of the gearboxes transmits the power taken from the motor having 7.5 kW power to shafts. Torque applied to shaft is distributed to experiment gears by the apparatus. The other gearbox consists of pinion-internal gear pair for wear testing procedure. In the apparatus, loading is made when the system is inactive. There is a panel for temperature control and adjustment of rotation speed of driving motor in the testing apparatus which is schematically represented in Figure 5.

**TABLE 1.** Geometrical properties of test gears

| Tooth form number | Symbol | Value |
|---|---|---|
| Tooth numbers [-] | $z_1$ | 17 |
| | $z_2$ | -75 |
| Module [mm] | m | 3 |
| Face width [mm] | $b_{1,2}$ | 10 |
| Profile shift factor [-] | $x_{1,2}$ | 0 |
| Pressure angle [°] | $\alpha_o$ | 20 |
| Pitch circle diameter [mm] | $d_{o1}$ | 51 |
| | $d_{o2}$ | -225 |
| Tip circle diameter [mm] | $d_{b1}$ | 57 |
| | $d_{b2}$ | -219 |
| Base circle diameter [mm] | $d_{g1}$ | 47.92 |
| | $d_{g2}$ | -211.43 |
| Axle offset [mm] | a | -87 |
| Gear ratio [-] | i | -4.41 |

**TABLE 2.** Technical properties of coatings

| Coating Material | Technical Property | |
|---|---|---|
| | Characteristics | Value |
| PTFE | Color | Black |
| | Density at 20 °C [g/ml] | 0.95 |
| | Operated Temperature [°C] | -180~240 |
| | Film thickness [µm] | 5~20 |
| $MoS_2$ bonded with polyamide | Color | Dark grey |
| | Density at 20 °C [g/ml] | 1.10 |
| | Operated Temperature [°C] | -70~380 |
| | Film thickness [µm] | 5~20 |
| $MoS_2$ bonded with epoxy | Color | Grey |
| | Density at 20 °C [g/ml] | 1.2 |
| | Operated Temperature [°C] | -70~380 |
| | Film thickness [µm] | 5~20 |

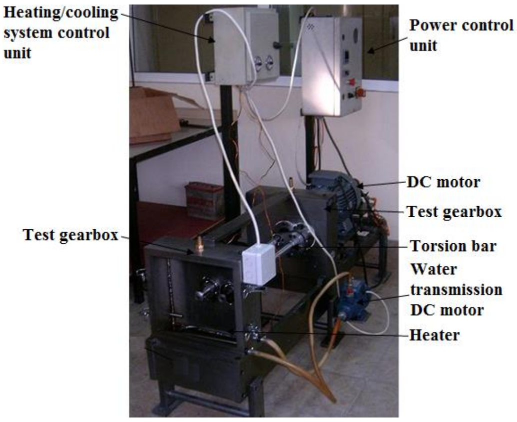


**Figure 4.** Testing apparatus

Speed control panel enables the control of rotation speed of up to 3000 rpm at 10 different speed levels. By this means, it is possible to perform fatigue and wear tests in the system at different rotation speeds. Immersion lubrication system was used in the experiments. Lubricant temperature was fixed at 23±2°C by a heating/cooling system. Properties of the lubricant can be seen in Table 3.

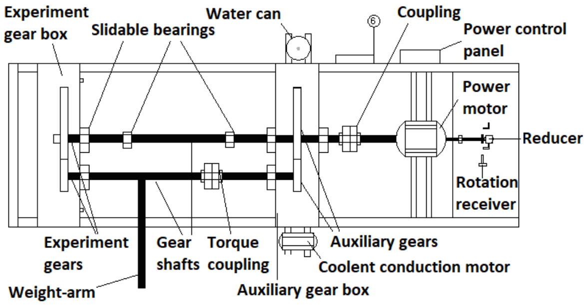


**Figure 5.** Schematic drawing of testing apparatus

**TABLE 3.** Properties of the lubricant [24]

| SAE (Society of Automotive Engineers) Number | 80W/90 |
|---|---|
| Density at 15 °C [g/ml] | 0.906 |
| Viscosity at 40 °C [mm$^2$/s] | 200 |
| Viscosity at 100 °C [mm$^2$/s] | 17.5~18.5 |
| Viscosity index | 95 |
| Flash point [°C] | 220 |
| Yielding point [°C] | -27 |

### 3. 3. Experimental Procedure

Three different coating materials, namely PTFE, MoS$_2$ bonded with polyamide and MoS$_2$ bonded with epoxy were used to investigate the effects of coating material on wear damage behaviour of tooth profiles of internal gears. Procedure of wear testing was given in Table 4.

In the experiments, system was inactivated and test gears were removed from gearbox after every 2.3x10$^4$ revolution of internal gear. Gears were cleaned to be purged of worn particles and the lubricant on them. Afterwards, an investigation was made to determine the wear that takes places on the profile of internal gear by measuring with a three dimensional coordinate measuring machine (CMM).

**TABLE 4.** Experimental procedure

| Rotation speed [rpm] | Number of revs [x10$^5$] | Wear coefficient [m$^2$/N] | Coating Material |
|---|---|---|---|
| 1500 | 0.23<br>1.15<br>2.30 | 9.14x10$^{-18}$ | No Coating |
| 1500 | 0.23<br>1.15<br>2.30 | 8.77x10$^{-18}$ | PTFE |
| 1500 | 0.23<br>1.15<br>2.30 | 7.68x10$^{-18}$ | MoS$_2$ bonded with polyamide |
| 1500 | 0.23<br>1.15<br>2.30 | 7.41x10$^{-18}$ | MoS$_2$ bonded with epoxy |

Wear on tooth contact surfaces can be seen in the optical microscope images in Figure 6.

Three different regions of tooth profiles of internal gears were determined for the measurement of wear depth on the sidewall of the tooth where the torsional moment was applied as seen in Figure 7.

Wear depths were measured at each fifty points in every region with a total number of 150 measurements, as seen in Figure 8. Afterwards, wear depth was determined by taking the average value of all measurements.

## 4. THEORETICAL AND EXPERIMENTAL RESULTS

### 4. 1. Variation of Wear Depth on Tooth Profiles

Wear depth occurred on tooth profiles of uncoated and coated internal gears was investigated theoretically and experimentally. Geometric properties of the testing gears are given in Table 1.

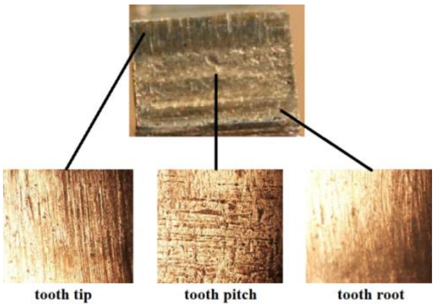


**Figure 6.** 100X optical microscope images from various regions of tooth contact surfaces

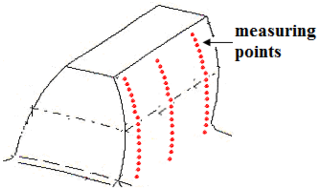


**Figure 7.** Measured points of the tooth profile

**Figure 8.** Measurement of wear from tooth profile with CMM device

Theoretical and experimental wear depth variations during meshing in internal gears were compared with each other for three different rotation cycle numbers (N) of 0.23x$10^5$, 1.15x$10^5$ and 2.3x$10^5$. Variation of theoretical and experimental wear depth through tooth profile of driven uncoated internal gear with 1500 rpm motor speed and 100 Nm torque is given in Figure 9.

Three different surface coating materials were used to investigate the effects of coating material on wear depth of internal gears. Technical properties of coating materials were given in Table 2. Theoretical and experimental wear depth values of surface coated internal gears with 1500 rpm motor speed and 100 Nm torque were given in Figure 10.

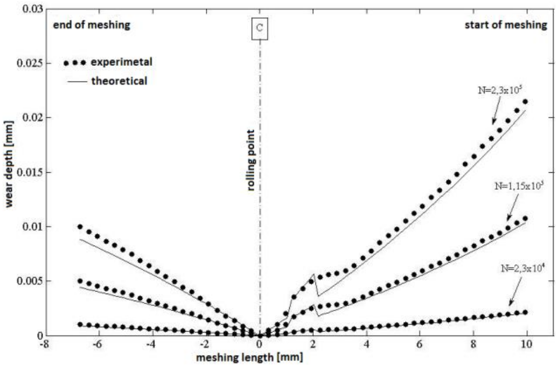
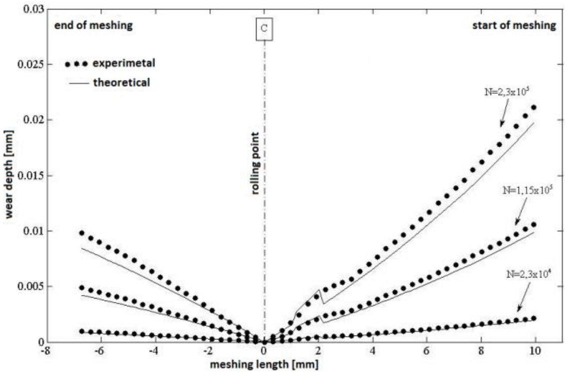
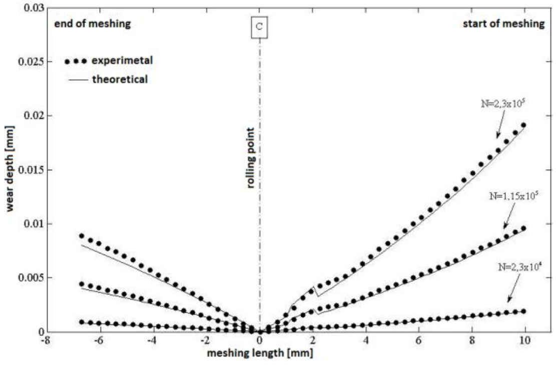


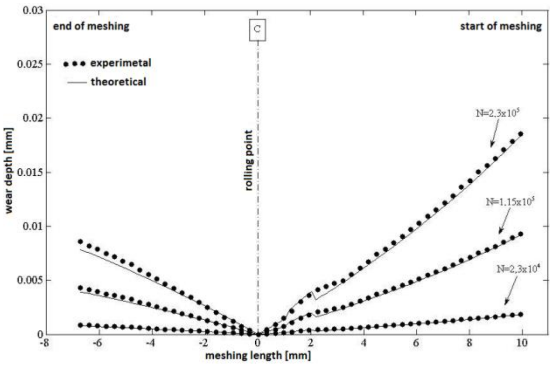
**Figure 9.** Variation of wear depth of uncoated internal gear at 1500 rpm motor speed and 100 Nm torque



(a) tooth surfaces coated with PTFE



(b) tooth surfaces coated with $MoS_2$ bonded with polyamide



(c) tooth surfaces coated with $MoS_2$ bonded with epoxy

**Figure 10.** Variation of wear depth of coated internal gears with 1500 rpm motor speed and 100 Nm torque

Theoretically and experimentally obtained values in different rotation cycle numbers and coating conditions were compared with each other. Results given in Figures 9 and 10 shows that wear depth values obtained experimentally are compatible with the theoretical ones. The results also show that maximum wear depth occurs at the start of meshing, wear depth value decreases and converges to zero at rolling point (C) where there is no relative slippage between driving and driven teeth and then increases till the meshing ends. It is clear from figures that surface coating decreases wear in internal gears. For example, maximum wear depth in internal gear without coating after 2.3x$10^5$ rotation cycle numbers is 2.3 % higher than the one in PTFE coated internal gear, 12.3 % higher than the one in $MoS_2$ (bonded with polyamide) coated internal gear and 15.9 % higher than $MoS_2$ (bonded with epoxy) coated internal gear. From Figure 9, maximum wear depth difference between experimental and theoretical values of uncoated internal gear at 1500 rpm motor speed and 100 Nm torque is 4.3%. These differences are admissible by considering the experiment conditions.

## 4. 2 Cumulative Wear in Internal Gears

After every 2.3x$10^4$ revolutions of internal gear, test gears were removed from test equipment. Accumulated lubricants and worn particles were wiped from internal gears with trichloroethylene. Wear measurement of gears was made with the accuracy of 1x$10^{-3}$ gr. Cumulative amount of wear in internal gears was also obtained theoretically by calculating the areas under theoretical wear depth graphs (Figures 9 and 10) obtained from Equation (6) and (7) using a MATLAB® program. Experimental and predicted cumulative amount of wear in internal gears at 1500 rpm motor speed and 100 Nm torque are given in Figure 11.

As it is clear from the graph, the highest value of cumulative wear was obtained in the internal gear with no coating and the lowest value of cumulative wear was obtained in the internal gear with $MoS_2$ coated which is bonded with epoxy.

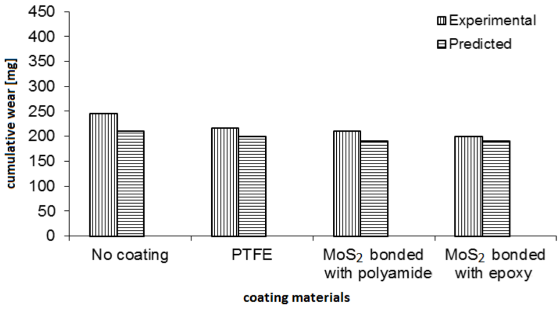


**Figure 11.** Experimental and predicted cumulative amount of wear in internal gears

Experimental value of cumulative wear decreased 11.4% in PTFE coated, 14.3% in $MoS_2$ coated which was bonded with polyamide and 18.78% in $MoS_2$ coated which was bonded with epoxy when compared with the uncoated internal gear. A difference of 5% up to 14.3% between the experimental and predicted cumulative wear values was observed. When test conditions are taken into consideration, the predicted cumulative amount of wear is in close agreement with the experimental results.

## 5. CONCLUSION

In this study, wear during coupling on the teeth profiles of internal gears coated with various polymeric coating materials (PTFE, $MoS_2$ bonded with polyamide, $MoS_2$ bonded with epoxy) was investigated theoretically and experimentally, as well as on the uncoated internal gear. A fatigue and wear testing apparatus having similar working principle with FZG closed circuit power circulation system was designed and manufactured to experimentally investigate the wear in internal gears. Archards' wear formulation was adapted to internal gears and wear behaviour in different loading and cycle time conditions was determined in the theoretical part of the study with a MATLAB® code written for the obtained formulation. Using a computer program allows the calculation of wear depth and cumulative wear amount during coupling at different motor speeds and torques in a more effective way as compared with experimental method because of the long manufacturing and experimentation time as well as the removal and calculation periods of experimental procedure. Internal gear-pinion couples manufactured from St50 material were coated with various polymeric materials (PTFE, $MoS_2$ bonded with polyamide, $MoS_2$ bonded with epoxy) in the experimental study. An uncoated internal gear was also investigated to compare with the performance of coated gears. Both theoretical and experimental results showed that maximum wear depth on the teeth of internal gears occurs in the tooth tip region where coupling with pinion starts. Thus, the critical region of internal gear in a mating internal gear-pinion couple is tooth tip. Theoretical and experimental studies showed that polymeric coated internal gears have more wear resistance than uncoated one and coating material itself by means of high lubrication ability and low friction coefficient of coating materials. High corrosion resistance of polymeric coatings protected metallic surfaces and decreased corrosive wear. It is also observed that highest wear resistance was obtained via $MoS_2$ coating with epoxy bonder because of its perfect sticking ability. Experimental study was performed in the same loading and cycle time conditions to validate the theoretical results and it was seen that the results are compatible.

## 6. ACKNOWLEDGEMENTS

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# Effect of Coating Material on Wear in Internal Gears

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## چکیده

در این مطالعه، بررسی تئوری و تجربی سایش در هنگام درگیری چرخ‌دنده‌های داخلی پوشانده شده با مواد پوشش پلی‌مری مختلف انجام شد. در بخش نظری مطالعه، از فرمول سایش آرچرز برای چرخ‌دنده‌های داخلی استفاده و رفتار سایش در شرایط مختلف تعیین شد. علاوه بر این، یک دستگاه تست خستگی و سایش مشابه با روش کار FZG (Forschungsstelle für Zahnrader und Getreibbau) سیستم گردش قدرت مداربسته چرخ‌دنده‌های داخلی طراحی و ساخته شد. به منظور آزمایش سایش در چرخ‌دنده‌های داخلی. زوج دنده‌ای داخلی ساخته شده از ماده‌ی St50 با مواد پلی‌مری مختلف، از جمله PTFE (polytetrafluoroethylene)، $MoS_2$ با پلی آمید، $MoS_2$ پیوند شده با اپوکسی در مطالعه تجربی پوشش داده شد. برای شناسایی عملکرد چرخ‌دنده‌های پوشش داده شده، یک چرخ‌دنده داخلی بدون پوشش نیز بررسی شد. مطالعات نظری و تجربی نشان می‌دهد که چرخ‌دنده‌های داخلی پلی‌مری پوشش داده شده به خاطراستفاده از قابلیت روان‌کاری بالا و ضریب اصطکاک کم مواد پوشش مقاومت بیشتری نسبت به چرخ‌دنده‌های بدون پوشش دارند. همچنین مشاهده شد که مقاومت در برابر خوردگی بالای پوشش‌های پلی‌مری موجب محافظت سطوح فلزی و کاهش خوردگی می‌شود.

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