Surface Capacity of Gears of Circular-Arc Tooth-Profile

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سعه السطح للمسننات ذات الأسنان الدائريه الجانبيه

هذا العمل يتضمن نتائج وتحليل لبحث معملى تم إجرائه بغرض دراسة تأثير سمك طبقه التصليد للسنه ودرجه صلادتها والحمل المؤثر على سعة تحميل السطح لأسنان نوع جديد من المسننات وهى المسننات ذات الأسنان الدائريه الجانبيه. هذه المسننات تم تصنيعها من صلب (16 MnCr5) وإستخدم أسلوب الكربنه والتقسية والمراجعه وذلك للحصول على سمك طبقه التصليد للأسنان ومقداره ٠,٥ ، ١,٢ ، ٢مم وتم إختيار درجات صلاده لسطح هذه الطبقات مقدارها ٤٦، ٥٢، ٥٨ ريكول سى والحمل العمودى المؤثر على الأسنان هو ٥، ٧,٥، ١٠ كيلونيوتن. هذه المسننات وعددهم ٣٠ زوج تم تركيبهم وإدارتهم فى ماكينه إختبار المسننات بعدد من اللفات مقداره ٦,٧x١٠^٧ لفه بسرعه دورانيه مقدارها ١٤٦٠ لفه/دقيقه. تم وزن هذه المسننات عده مرات أثناء التشغيل بعد عدد ثابت من اللفات وذلك لتحديد وزن المعدن المزال. تم رسم منحنيات تمثل العلاقه بين الوزن التراكمى للمعدن المزال وعدد اللفات عند أسماك طبقه التصليد المختلفه وكذلك درجات صلادة هذه الطبقات. وكذلك تم حساب معدل التآكل ورسمة مع الحمل العمودى المؤثر على الأسنان وكذلك سمك طبقة التصليد ومقدار صلادة طبقه التصليد لسطح الأسنان. وتم عمل (Curve fitting) لهذه المنحنيات المعمليه بهدف الحصول على معادلات رياضيه يستخدمها المصمم لتصميم هذه المسننات . النتائج المعملية لهذا البحث اظهرت الآتى :-
إن معدل التآكل لسطح الاسنان يقل مع زيادة كل من سمك طبقة التصليد وكذلك درجة الصلادة ثم يزيد معدل التآكل مرة ثانيه مع زيادة أكبر لكل من سمك طبقه التصليد ودرجة صلادة الطبقة أى ان احسن سمك لطبقه التصليد لهذا النوع من المسننات هو ١,١ مم وكذلك أحسن درجة صلاده لطبقة التصليد ٥٣,٥ ريكول سى. وكذلك اوضحت النتائج ان معدل التآكل لسطح الأسنان يزيد مع زيادة الحمل المؤثر على الأسنان وان معدل التآكل للمسننات الغير مصلده بطبقة تصليد أكبر بحوالى ٤ إلى ١٩٢ مره من معدل التآكل لأسطح الأسنان التى لها طبقة تصليد. وكذلك تمت مناقشة هذه النتائج المعملية وكتبت المعادلات الرياضيه المستنتجه على المنحنيات الخاصه بها.

ABSTRACT

The paper embodies results and analysis of an experimental investigation carried out to study the effect of case depth, case hardness and tooth load on the surface capacity of a relatively new type of gearing having teeth of circular arc profiles. These gears were manufactured from 16 MnCr5. Curburizing, quenching and tempering techniques were used with 0.5, 1.2 and 2 mm case depth and 46, 52 and 58 HRC case hardness, also applied tooth loads 5, 7.5 and 10 KN were used. These thirty pairs of gears were run to 6.7×10^7 revolutions at speed 1460rpm in a power circulating gear test rig using forced lubrication technique. Test gears were weiged at interval of times, the accumulated weight of removed metal were drawn with the number of revolutions. Wear rate was calculated and drawn with the applied tooth load, case depth and case hardness. A curve fitting was made for these experimental results using Grapher software. Results show that an optimum case depth of the case hardened gears of circular arc tooth profile equal to 1.1 mm and also optimum case hardness of the case equal to 53.5 HRC. Wear rate increases with increasing the applied tooth load. Wear rate for unhardened gears increases by 4 to 192 times than that of the case hardened gears. An empirical formulae for the accumulated weight of removed metal with number of revolution, wear rate with tooth load, case depth and case hardness were derived, also presented on the curves and discussed.

INTRODUCTION

Gears are usually designed for load carrying capacity for bending strength (tooth breakage) and/or load carrying capacity for surface strength (tooth wear). Tooth surface failure results in wear which beside changing the accuracy of the designed gear running may results in decrease of tooth thickness which may cause bending strength failure. Many investigators[1-5] have studied the wear characteristic of unhardened gears of circular-arc tooth-profile.

To increase surface durability, beside main design considerations regarding the choice of material and design dimensions, several methods are adopted to decrease wear as:

a- Mechanical surface hardening (cold-Working method) such as shot-peening.
b- Surface hardening such as flame hardening, hardening in electrolytic bath and induction hardening.
c- Chemical-thermal treatments (case hardening) such as carburizing, nitriding, cyaniding, carbunitriding, plasma nitriding, salt bath nitrocarburizing, boronizing, chromizing, tufftriding and sur-sulfing.

Many investigators[6-8] studied the surface durability for involute gears using heat treatment techniques.

The aim of this work is to study the effect of case depth, case hardness and tooth load on the surface capacity (wear rate) for case hardened gears of circular-arc tooth-profile, also select the optimum conditions of the case depth and its hardness. The effect of module, helix angle, speed of rotation, and oil viscosity on the wear rate and surface capacity for case hardened gears of circular-arc tooth-profile are done [under publication].

THE TEST GEARS

Test gears are of convex all addendum pinion profile and concave wheel tooth profile[9-13]. Contact between teeth is point moving in a straight line along the helical tooth. There is no conjugacy between the teeth profiles except along the helical face. Thus such gears transmit power and motion along the path of tooth contact, with the point of contact of the two profiles changing under load to an ellipse. In these circumstances the problem of tribology applied to this type of gearing is of a complex nature. Fig.(1) shows one pair of test gears with the convex and concave teeth in mesh. Table (1) shows the specifications and the main dimensions of the test gears, while table (2) shows the chemical composition and mechanical properties.

	Pinion	Wheel
Number of pairs	30	
Normal diametral pitch or (module, mm)	6 (4.233)	
Pitch diameter, mm	91.5	
Normal pressure angle, deg	25	
Helix angle, deg	22.3	
Face width, mm	42	
Number of teeth	20	
Axial pitch, mm	35	
Radii of curvature in normal plane, mm	8.4667	9.525
Radii of curvature along helix angle, mm	314.884	341.255
Blank diameter, mm	103.735	90.170
Overlap ratio	1.2	
Surface roughness, μ	1.6	
Backlash, mm	0.423	
Bore diameter, mm	$30^{+0.009}$	

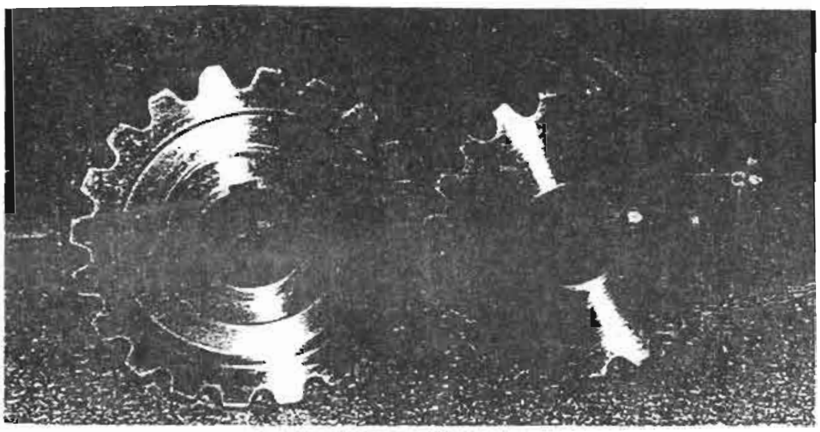
Table (1) Dimensions and specifications of the test gears

Element	C	Si	Mn	P	S	Cr
Weight percent	0.14 to 0.19	0.15 to 0.4	1.00 to 1.30	0.035 maximum	0.035 maximum	0.8 to 1.10

Table (2) Chemical composition of case hardening steel used for manufacturing test gears

Steel grade		100 to 160 mm diameter				
Code number	Material number	Yield point Kg/mm^2 min	Tensile strength Kg/mm^2	Elongation $L_o = 5d_o$ % min	Reduction of area % min	Notched bar impact strength $Kg.m/cm^2$ min
16MnCr5	1.7131	60	90	11	50	34

Table (3) Mechanical properties of case hardening steel used for manufacturing test gears



Fig(1) Pair of test gears - 22.3° helix angle.

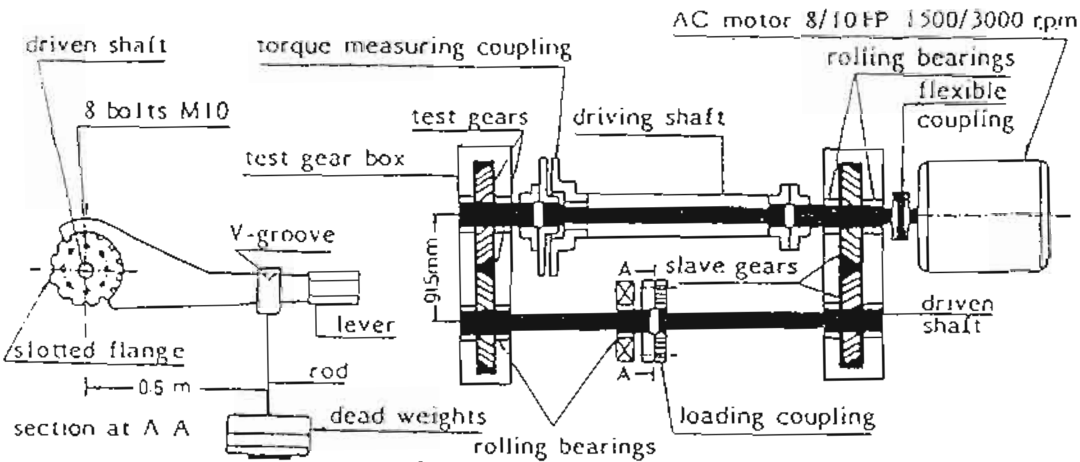
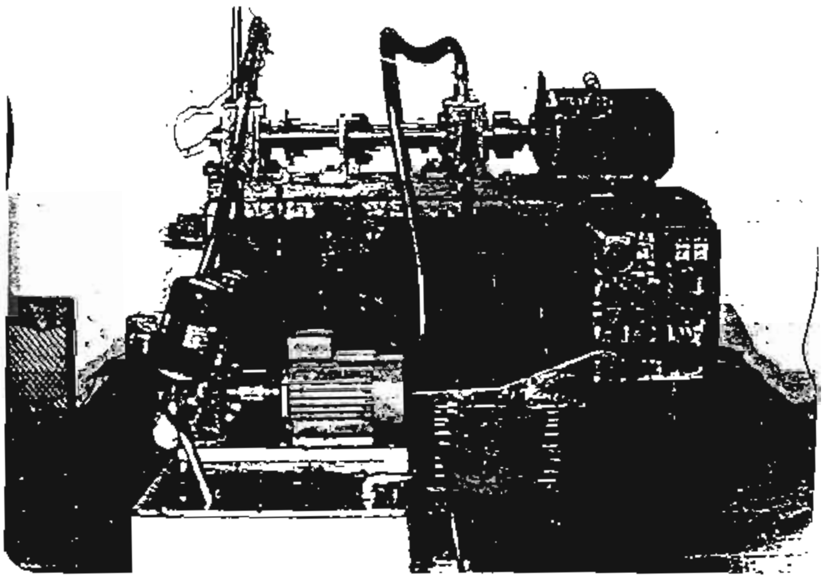


Fig 2 Layout of the Test Rig



Fig(3) General view of the test rig

THE TEST RIG

The test rig is of the power circulating type composed, as shown in Fig.(2), of two shafts carrying the two test gears at one side and a pair of slave gears at the other side The driven shaft is composed of two parts connected together by a flange coupling. The test rig is loaded, as shown in the figure, by applying dead weights at the end of a lever fastened to the slotted flange while the other flange is fixed, thus twisting each part of the shaft relative to the other. The driving shaft carries a graduated torque coupling to indicate the load on the machine. The test rig is driven by 6/7.4 KW a.c. motor when running at speeds 1500 and 3000 rpm, respectively. The test rig is forced lubrication by a gear pump driven by 2 HP, 1460 rev/min motor. Fig.(3) shows a general view of the test rig.

HEAT TREATMENT TECHNIQUE

The heat treatment technique used in this study was carburizing followed by hardening and then tempering. The test gears and other test specimens for case depth, distrbution of carbon in the case depth and microstructure examinations were completely surrounded with a mixture composed from 80% charcoal and 20% $BaCo_3$ and packed in the three carburizing boxes according to the required case depths. The boxes were covered and all openings were luted with a non-cracking loam to prevent air from penetrating into the boxes. The first box containing group A of test gears was placed in an electric furnace and heated gradually at the rate of 50°C per hour up to the carburizing temperature of 920°C and kept for a predetermined period of time = 7.5 hrs according to Fig.(4) and table (4-a). The box was cooled in furnace. The second and third boxes were placed in the same furnace and heated according to Fig.(4) and table (4-a). After reaching the room temperature, the test gear were preheated in salt bath furnace to 865°C for one hour to be hradened by oil quenching. After quenching the test gears were tempered by heating them in electric furnace according to the required case hardness as shown in Fig.(4) and table (4-b). Specifications of carburizing, hardening and tempering furnaces and quenching oil are given in appendix (1 and 2).

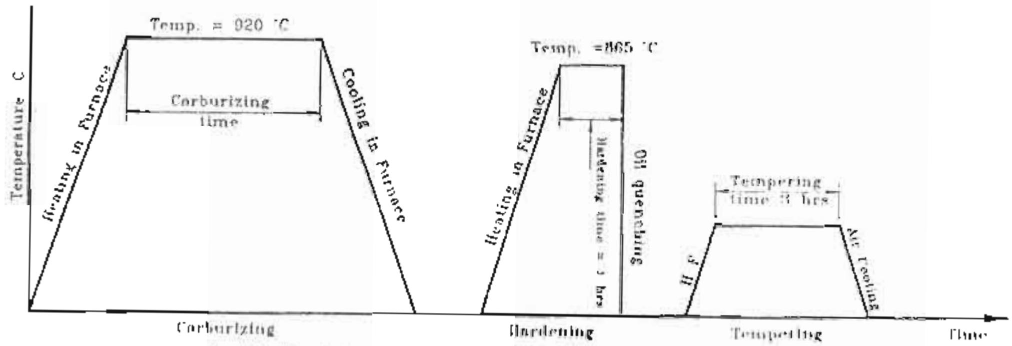


Fig.(4) Heat treatment conditions of the test gears

Group / Carburizing process	Group A	Group B	Group C
Carburizing time, hrs	7.5	18	28
Case depth, mm	0.5	1.2	2

Table (4-a) Case depth and required time

Group / Tempering process	Group D	Group E	Group F
Tempering temperature, °C	170	180	190
Case hardness, HRC	58	52	46

Table (4-b) Case hardness and required temperature

EXPERIMENTAL PROCEDURE

Tests were carried out for test gears at normal tooth loads of 5, 7.5 and 10 KN running at constant speed equal to 1460 rev/min which correspond to tangential transmitted loads, torques and transmitted powers given in Appendix (3). To study the effect of the case depth and case hardness on the wear rate and surface capacity of the gears of circular-arc tooth-profile, tests were carried out for a groups of test gears with case depths 0.5, 1.2 and 2 mm and case hardness of 46, 52 and 58 HRC. Also, the same tests were carried out on the test gears without any heat treatment to show the difference between them and the case hardened gears. Thirty pairs of test gears were used for these experiments. Tests were carried out using force lubricating system at rate of 2 L/min, specifications of the lubricating oil are given in appendix (2). Temperature of the lubricating oil in the test gear box was measured using a sensor and a temperature indicator mounted on the test gear box.

The metal removed after each run was measured by weighing the test gears before and after each test by a digital and analogue balance of accuracy 1×10^{-5} gram and capacity 2000 gram, type Chyo Jupiter C2-2000, serial No. 30559, made in Japan. The test gears were washed before weighing using an ultrasonic cleaning tank type No. 323/201, serial No. 337226, 40 KHZ frequency, 150 watts output and 220x130x150 mm internal dimensions made in Germany and then dried by a stream of compressed air.

Wear rate is calculated from the relationship

$$\text{Wear rate (WR)} = \frac{\text{Accumulated weight of removed metal}}{\text{tooth load x number of revolutions}}$$

EXPERIMENTAL RESULTS

1- Weight of removed metal during running:

Fig.(5) shows the change in accumulated weight of removed metal (AWRM) for pinion and wheel with number of revolutions at different tooth loads (pinion and wheel without heat treatment). Fig.(6, 7 and 8) show the change in AWRM for pinion and wheel with number of revolutions at different tooth loads and case hardness for case depths equal to 0.5, 1.2 and 2 mm, respectively. It is clear from these figures that the AWRM is rappidly increases with increasing the number of revolutions for the pinion and wheel without heat treatment than that of the case hardened gears at any case depth or case hardness. AWRM increases with increasing tooth load for all cases of case hardened or unhardened gears. AWRM for unhardened wheel is greater than that of the unhardened pinion. AWRM for hardened gears of case depth equal 2 mm is greater than that of the hardened gears of case depth equal 0.5 mm, and AWRM for hardened gears of case depth equal 1.2 mm give a minimum values than that of case depth equal 2 mm or 0.5 mm, this is for the same tooth loads and case hardness. That is to say an optimum value of the case depth is found. Also for all cases of tests, AWRM for the smallest and the highest case hardness (46 and 58 HRC) of the case hardened gears are greater than that of the hardened gears of case hardness equal 52 HRC, and AWRM of hardened gears of case hardness equal 46 HRC is greater than that of the hardened gears of case hardness equal 58 HRC. That is to say an optimum value of the case hardness is found.

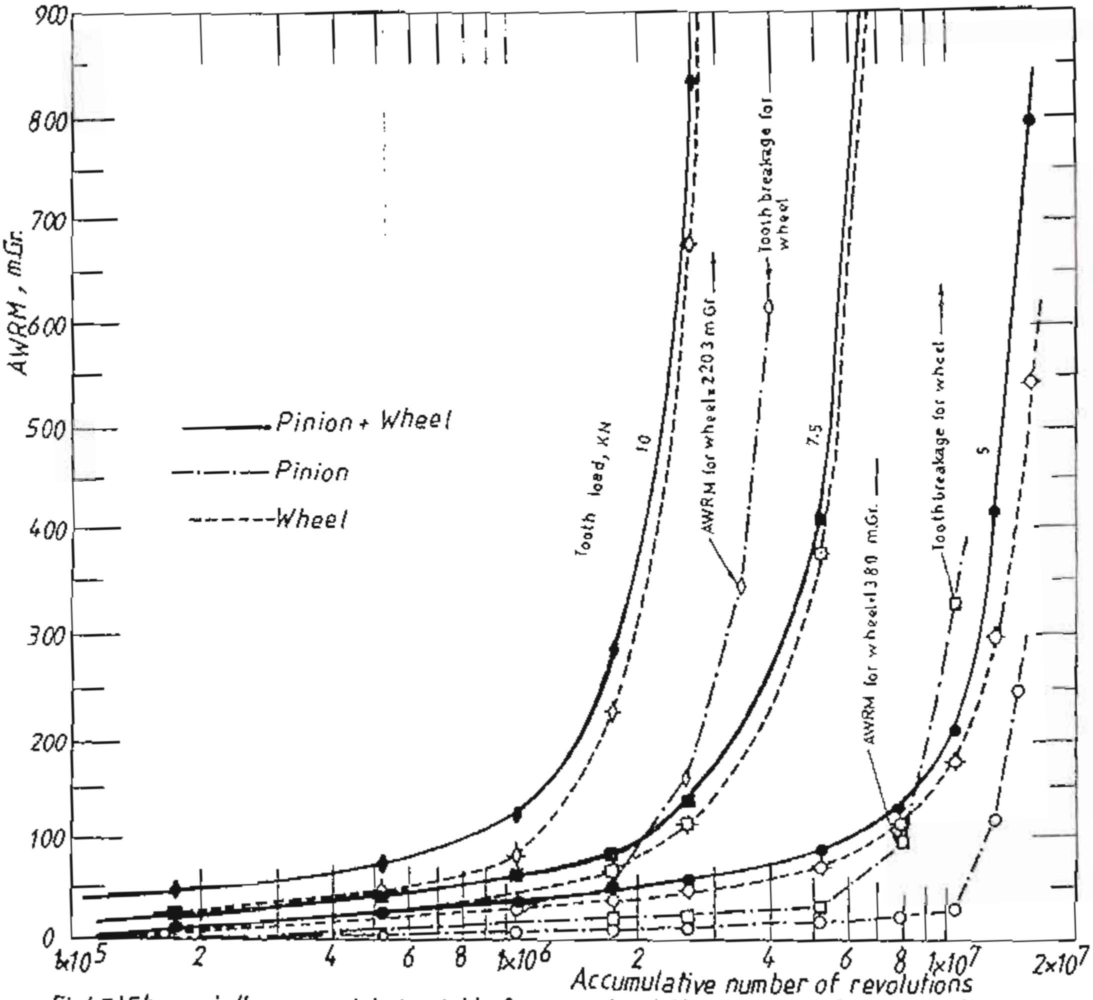
Fig.(9) shows the tooth breakage due to surface failure for wheel teeth and surface failure of the pinion teeth at tooth load equal 10 KN and 4.03×10^{6} revolutions. Fig.(10) shows the tooth breakage due to surface failure for the wheel teeth and the surface failure of the pinion teeth at 7.5 KN and 1.05×10^{7} revolution. Fig.(11) shows the surface failure of the gear teeth at tooth load equal 5 KN and 1.58×10^{7} revolutions.

For all case hardened test gears there is no tooth breakage at all tests, for example, Fig.(12) shows a picture of surface for pair of gear teeth after 6.7×10^{7} revolutions, and 7.5 KN tooth load.

A curve fitting for these results has been found using Grapher software which gives the following equation

$$AWRM = (8.75\times10^{-9} - 0.2126)\ N^{(0.4016\ -\ 1.5252)}$$

Ranges of constants given in this equation depend on the tooth load, case depth and case hardness. The fitting equation for each curve is indicated in Fig.(6, 7 and 8).



Fig(5)Change in the accumulated weight of removed metal(AWRM) for un hardened pinion and wheel with change of accumulated number of revolutions at different tooth loads

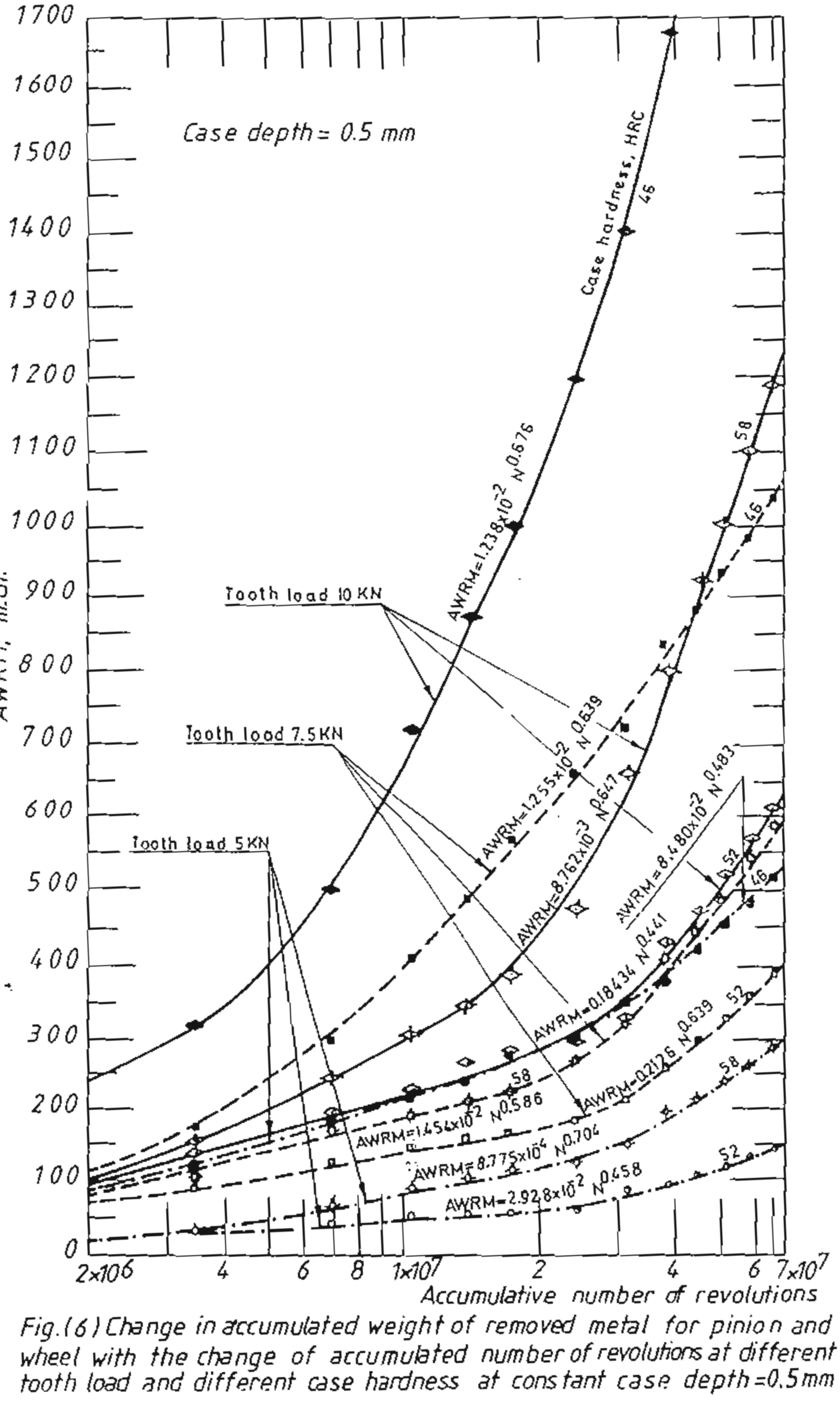


Fig.(6) Change in accumulated weight of removed metal for pinion and wheel with the change of accumulated number of revolutions at different tooth load and different case hardness at constant case depth =0.5mm

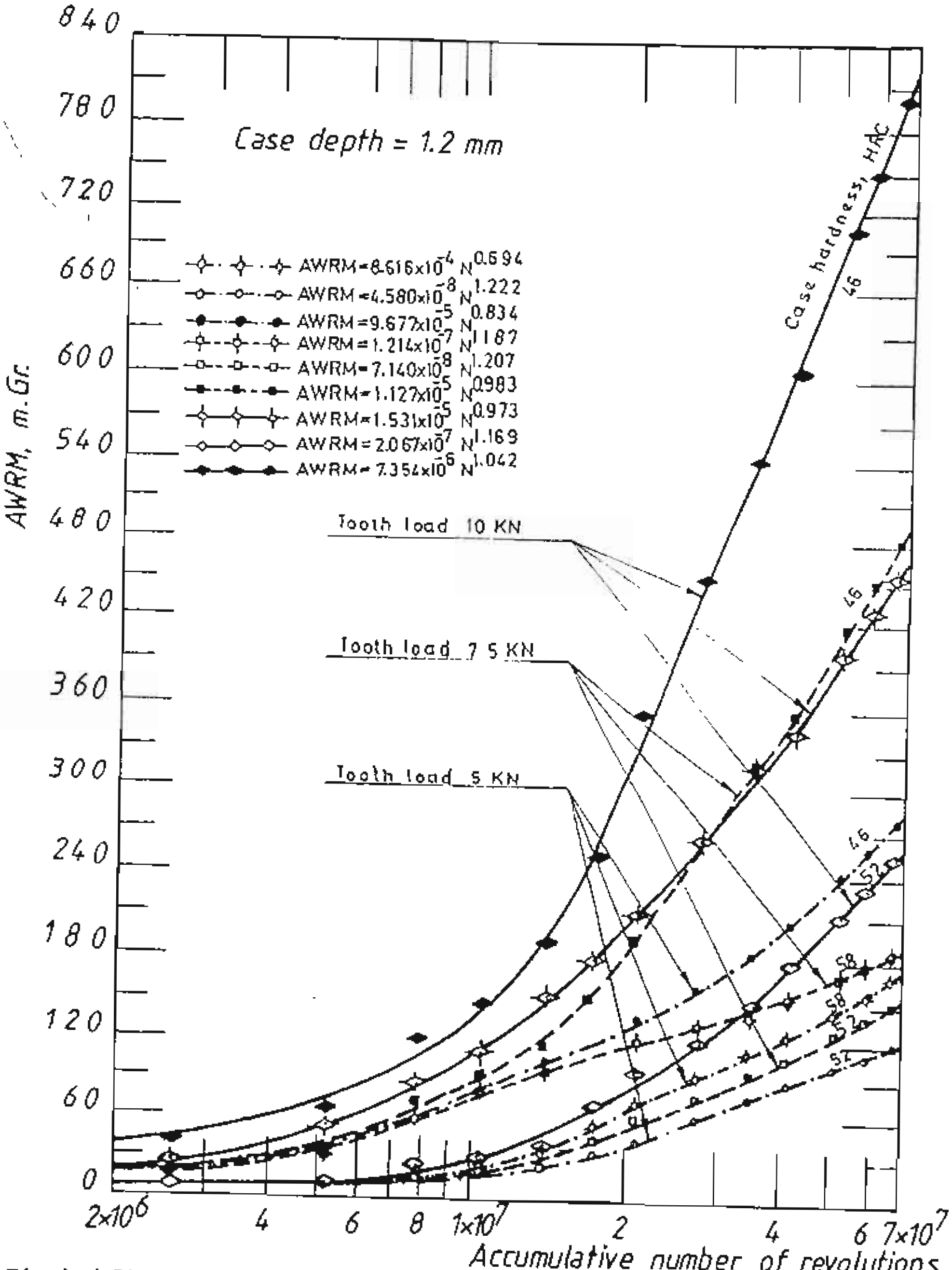


Fig (7) Change in acculated weight of removed metal for pinion and wheel with the change of accumulated number of revolutions at different tooth load and different case hardness at constant case depth=1.2mm

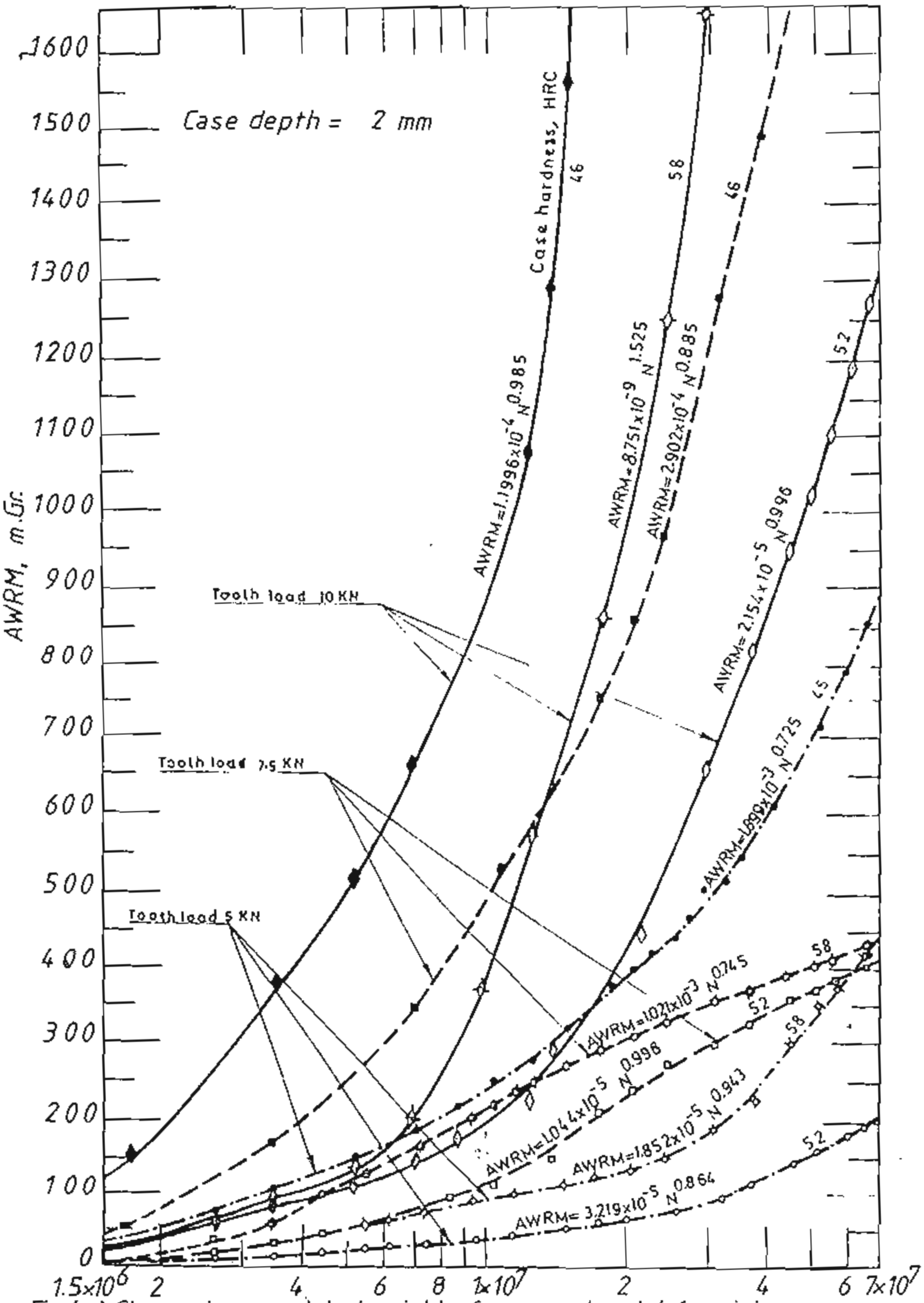
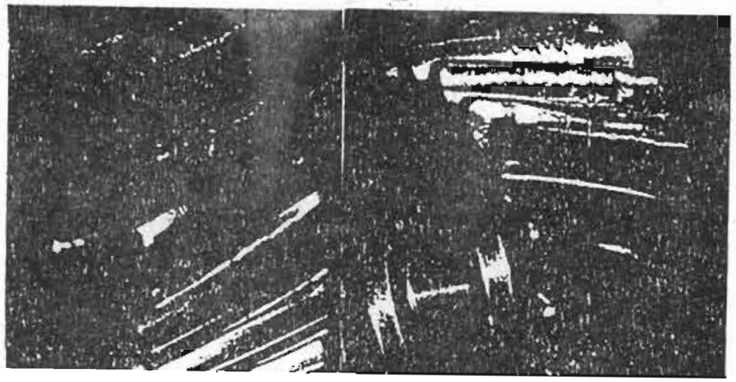
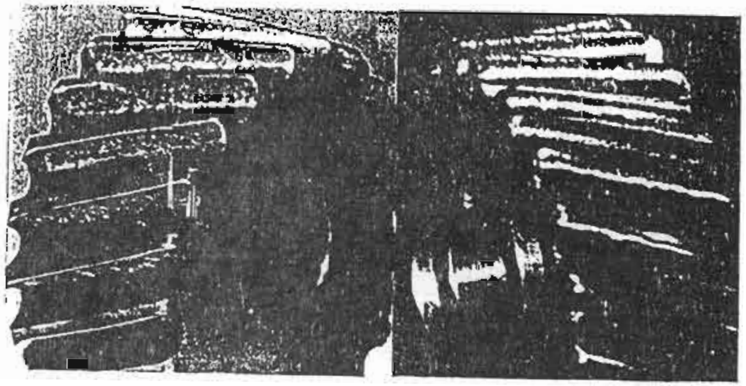


Fig (8) Change in accumulated weight of removed metal for pinion and wheel with the change of accumulated number of revolutions at different tooth load and different case hardness at constant case depth= 2 mm



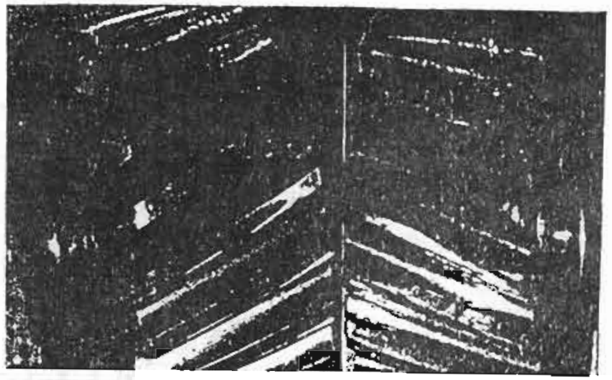
Wheel : Tooth breakage Pinion : Surface failure

Fig(9) Tooth breakage and surface failure at load=10 KN. number of revolutions= 4.03×10^{6} rev.



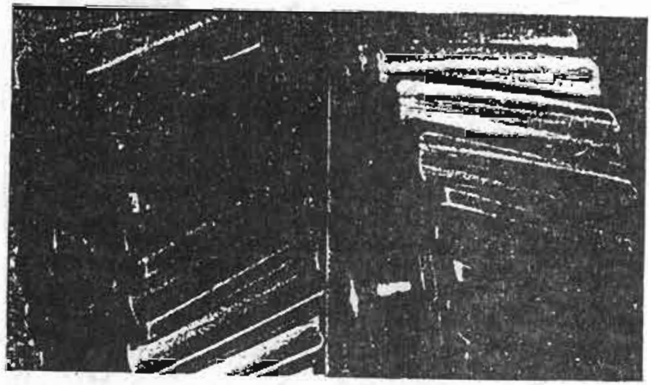
Wheel: Tooth breakage Pinion Surface failure

Fig(10) Tooth breakage and surface failure at 7.5 K N load and 1.05×10^{7} revolutions



Wheel Pinion

Fig(11) Surface failure of pinion and wheel at 5 K N tooth load and 1.58×10^{7} revolutions



Wheel Pinion

Fig(12) Surface wear for case hardened pinion and wheel at 7.5 KN load and 6.7×10^{7} rev. CD=0.5mm, CH=58HRC

2- Effect of Tooth Load on the Wear Rate

Fig.(13) shows the change of the wear rate of the hardened test gears with the change of tooth load at different case depths and case hardness. From this figure it is very clearly noticed that the wear rate increases with increasing the tooth load for tested gears. Wear rate for case depth of 2 mm is greater than that for the case depth equal to 0.5 mm, wear rate for test gears of case depth equal to 1.2 mm gives minimum values for all the tested gears. This means that an optimum case depth is found. Also for all case depths, wear rate of the test gears with hardness (46 HRC) is greater than that for the gears of (58 HRC), wear rate is minimum for test gear with medium hardness (52 HRC). This means that there is an optimum value of the case hardness.

A curve fitting for these results is derived using Grapher software which gives the equation

$$\text{Wear rate} \quad WR = (3.593\times10^{-17} - 1.008\times10^{-11})\ P^{(0.4178\ -\ 2.019)}$$

Ranges of the constants given in this equation depend on the case depth, and case hardness. The fitting equation for each curve is indicated in Fig.(13).

3- Effect of Case Depth

Fig.(14) shows the change of the wear rate of the hardened test gears with the change of the case depth for different case hardness at different tooth loads. From this figure it is noticed that, wear rate decreases with increasing the case depth to a certain value of the case and then increases with increasing the case depth for all test conditions.

A curve fitting for these results has been found using Grapher software which gives the following equation

$$WR = A + B.CD + C.CD^2$$

Where A, B and C are constants, which depending on the applied tooth load and case hardness of the test gears. The fitting equation for each curve is indicated in Fig.(15)

From the above equations and Fig.(14) an optimum value of the case depth is calculated and equal to 1.004 - 1.239 mm for all test conditions with mean value of 1.094 mm.

4- Effect of Case Hardness

Fig.(15) shows the change of the wear rate with the change of case hardness at different case depths and different tooth loads. From this figure, it is noticed that wear rate decreases with increasing the case hardness for all case depths at different tooth loads to a certain value and then increases with increasing the case hardness of the test gears. A curve fitting for these resuts has been found using Grapher software which gives the following equation

$$WR = D + E.CH + F.CH^2$$

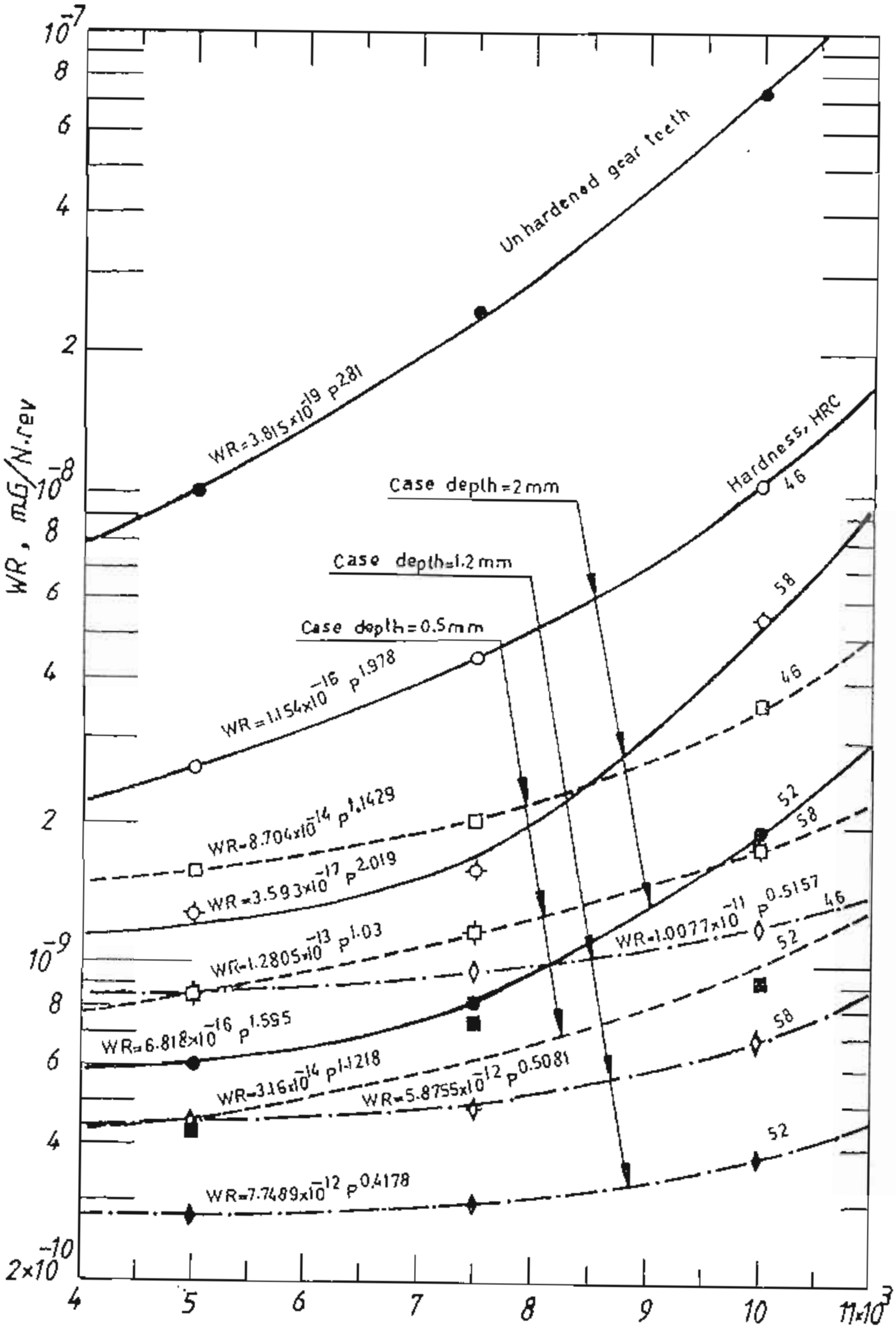
Where D, E and G are constants, which depending on the case depth and applied tooth load on the test gears. The fitting equation for each curve is indicated in Fig.(15).

From the above equation and Fig.(15) an optimum value of the case hardness of the hardened test gears is found to be 53.23 - 53.98 HRC, for all applied tooth loads and case depths, with mean value of 53.5 HRC.

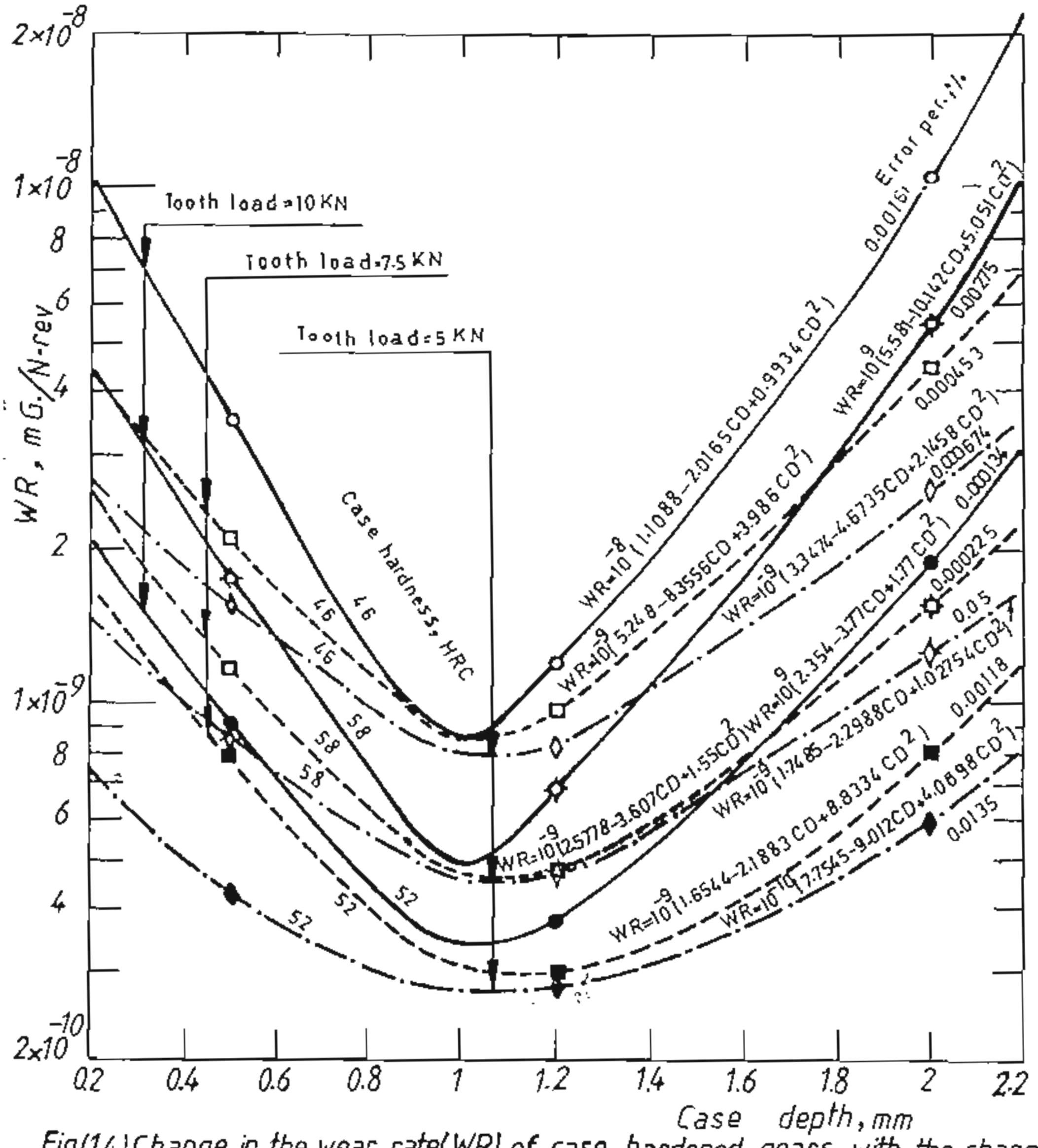
DISCUSSION OF THE EXPERIMENTAL RESULTS

During gear running, the surface and subsurface of the teeth in mesh are subjected to tension, compression and shear stresses. This surface contact is made between crests of surface waves. A number of actions take place, heavy rubbing and deformation of metal, plowing by "hills" on the harder material through the softer material, which results in breaking off worn particles and creating surfaces of different roughness and finally welding of minute high areas that have been rubbed clean. The minute welds break immediately, as motion continues, but may break at another section so that metal is transferred from one surface to the other. New surface roughness is formed, some to be plowed off to form wear particles.

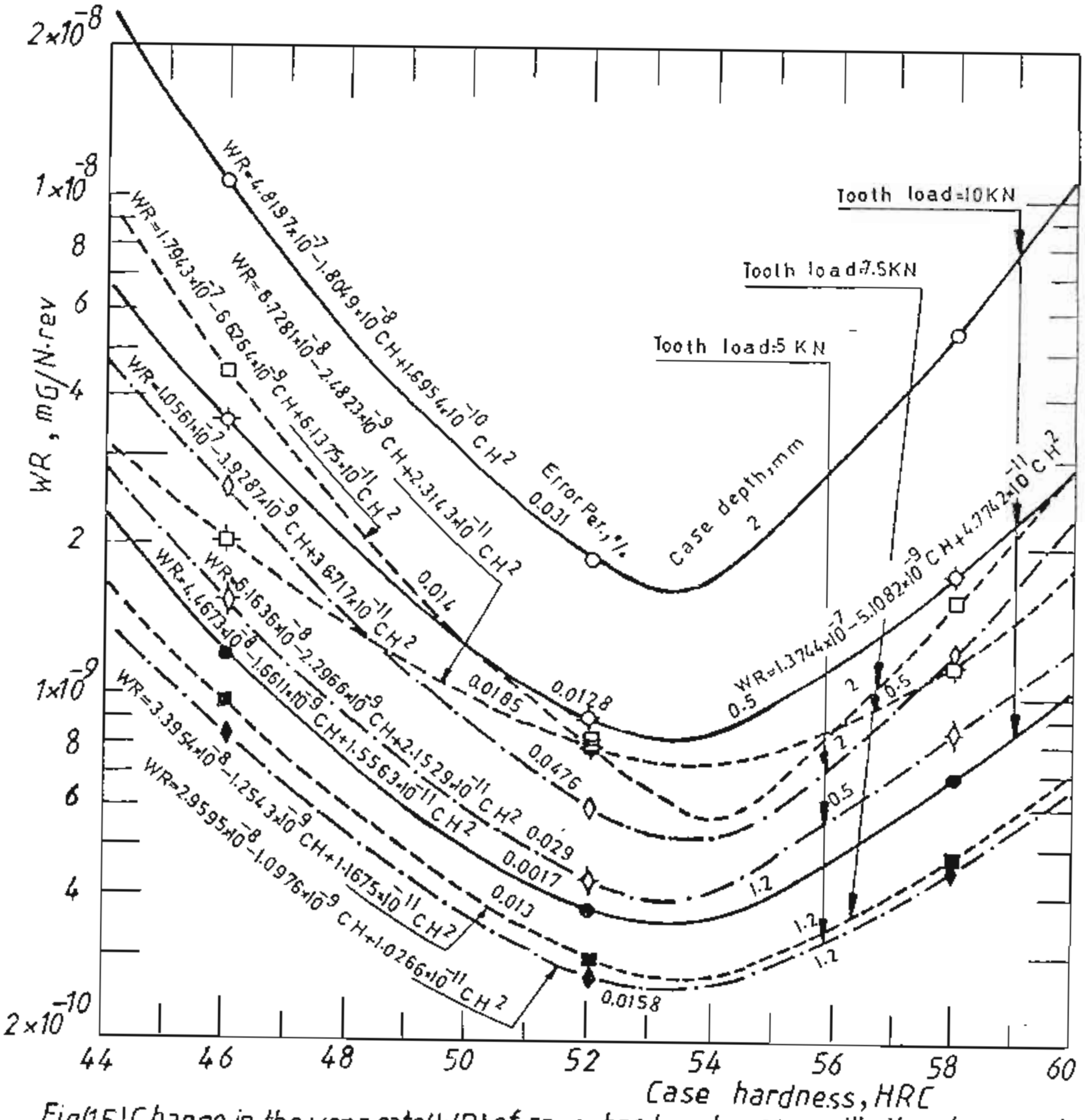
With increasing the number of revolutions, a very large number of minute cracks form in and below the surface which may grow and join togther. Eventually, small bits of metal are separated and forced out, leaving pits, surface fatigue may occur. By further increasing the number of revolutions, pits continue to form and enlarge as edges crumble or pits collides into each other. Eventually the designed tooth shape is deformed generating noise and vibration



Fig(13) Change in the wear rate(WR) of case hardened gears with the change of the applied tooth load at different case depths for different case hardness



Fig(14) Change in the wear rate(WR) of case hardened gears with the change of case depth at different tooth loads for different case hardness



Fig(15) Change in the wear rate(WR) of case hardened gears with the change of case hardness at different tooth loads for different case depths.

With Increase of amount of removed metal the tooth thickness decreases which may lead to strength failure and tooth breakage.

With Increase of applied tooth load the area of contact and the tooth deflections along and across the path of contact increase. This leads to unstable lubrication conditions and difficulty of oil film formation. Heat generation and contact temperature for the area of contact increase. strength of the material decreases, accordingly removed metal and wear rate increase.
For increasing the case depth and/or case hardness yeild limit and ultimate strength of the case increase, wear resistance increases consequently wear rate decreases to a certain value. By increasing the case depth and/or case hardness, the contact area of the teeth did not easily develop plastic deformation, therefore it induced a relatively small contact area and high contact pressure. In the hardened layer composed of fine structures, such as martensite and retained austenite, the thermal conductivity is relatively low and the flash temperature becomes high. Therefore the amount of weight of removed metal and wear rate increases.
The Increase of retained austenite resulting from further increasing the case depth, decrease the yield and ultimate stress of the material. It also decreases the fatigue limit and increase the crack propagation. This is believed to be the reason of increase of the wear rate. Also for largest case depth and case hardness, rigidity of the case hardened gear teeth increases, followed by increase of vibration level and dynamic load, tooth further increase of weight of removed metal and wear rate.

CONCLUSION

Wear rate for case hardened gears decreases with increase of case depth tending to reach a minimum at 1.1 mm case depth and increases again with further increase of the case depth.

Wear rate for case hardened gears decreases with increase of case hardness tending to reach a minimum at 53.5 HRC and increases again with increase of case hardness.

Wear rate of hardened gears of circular-arc tooth-profile increases gradually with increase of applied tooth load. For unhardened gears this rate is about 4 to 192 times the wear rate for case hardened gears. This range depending on the tooth load, case depth and case hardness.

Empirical formulae are derived to help the designer to estimate the wear rate and accumulated weight of removed metal from the design parameters:

- Wear rate with tooth load

$$WR = (3.593\times10^{-17} - 1.008\times10^{-11})\ P^{(0.4178\ -\ 2.019)}$$

- Wear rate with case depth

$$WR = A + B.CD + C.CD^2$$

- Wear rate with case hardness

$$WR = D + E.CH + F.CH^2$$

- Accumulated weight of removed metal with number of revolution

$$AWRM = (8.75\times10^{-9} - 0.2126)\ N^{(0.4016\ -\ 1.5252)}$$

Values and ranges of the constants given in the above equations depending on the experimental test conditions.

For unhardened test gears, tooth breakage occured at tooth load of 10 KN and 4.03×10^6 revolutions, also at tooth load of 7.5 KN and 1.05×10^7 revolutions. There is no tooth breakage at tooth load of 5 KN and 1.58×10^7 revolutions. For case hardened gears, there is no tooth breakage during running to 6.7×10^7 revolutions for all test conditions.

REFERENCES

1- Attia, A. Y. and Fahmy, M. A. K., "Wear of gears of double circular-arc tooth-profile", ASME paper No. 77-DET-59, June, 1978, (American Society of Mechanical Engineers, New York, NY 10017, U.S.A.).

2- El-Bahloul, A. M. M., "Wear of gears of circular-arc tooth-profile: Effect of tooth load and speed of rotation", on Proc. 3rd Int. Conf. on Production Engineering Design and Control, Alexandria University, Egypt, 12 - 14 December, 1986.

'l-Bahloul, A. M. M., "Wear of gear of circular-arc tooth-profile: Effect of helix angle and oil viscosity", Tribology International Vol. 20 No. 4, August 1987, 205 - 209.

4- El-Bahloul, A. M. M., "Wear of Involute helical and circular-arc gears (A comparative study)", Wear, 122 (1988), 103 - 114.
5- El-Bahloul, A. M. M., "Load-carrying capacity for gears of circular-arc tooth-profile", Wear, 129, 1989, 183 - 193.
6- Coy, J. J., Townsend, D. P., and Zaretsky, E. V., "Dynamic capacity and surface fatigue life for spur and helical gears", ASME, Journal of Lubrication Technology, Series F, Vol. 98, 1976, PP 267 - 276.
7- Fujita, K., Yoshida, A., Yamamoto, T., and Yamada, T., "The surface durability of the case-hardened nickel chromium steel and its optimum case depth", Bulletin of the JSME, Vol. 20, No. 140, 1977, PP 232 - 239.
8- Gang, D., Chui, X., and Yang, Y., "A method to determine the optimum effective case depth of gears", 4th European tribology congress, Ecully - France, 9-12, September 1985 PP 1 - 5
9- Davies, W. J., "Novikov gearing", Machinery, London, 96, January 13, 1960, PP 64 - 73.
10- Wells, C. F. and Shotter, B. A., "The development of CirCarC gearing", AEI Eng., March - April 1962 PP 83 - 88.
11- Chironis, N., "Design of Novikov gears", Prod. Eng. (N.Y.), September 17, 1962, PP 91 - 102.
12- French, M. J., "Conformity of circular-arc gears", J. Mech. Eng. Sci., Vol. 7, No. 2, 1965, PP 220 - 223.
13- Dyson, A., Evans, H. P. and Snidle, R. W., "Wildhaber-Novikov circular arc gears: geometry and kinematics", Proc. R. Soc., London, Ser. A, 403, 1986, PP 313 - 340.

APPENDIXES

Appendix (1)

Type / Specification	Carburizing Furnace Electric Furnace	Hardening Furnace Salt bath F.	Tepering F. Electric F
Dimension	600x1200x600	ϕ 1000x1000	ϕ 800x1000
Type	HN 27/12	DAW	KPA 20/6
Volt	3x380	3x380	3x380
Power	40 KVA	85 KVA	22 KVA
Temperature	960 °C	860 - 880 °C	650 °C
Made in	Czechoslovakia	Germany Fulmina	Czechoslovakia

Table (1) Specifications of carburizing, hardening and tempering Furnaces.

Appendix (2)

Characteristics	Lubricating oil	Quenching oil
Kinematic viscosity cSt at 40°C	599	30.9
Kinematic viscosity cSt at 100°C	32.7	4.7
Specific gravity	0.921	0.869
Pour point, max. °C	−9	-18
Flash point, min °C	174	216
Viscosity index	89	103
Timken Ok load 40 lb (18Kp)	60	–

Table (2) Characteristics of the lubricating and quenching oils.

Appendix (3)

Normal tooth load KN	Transmitted load, KN	Transmitted torque KN.M	Transmitted power KW
5	4.19	0.192	29.5
7.5	6.29	0.288	44.3
10	8.39	0.384	59.1

Table (3) Normal tooth load, transmitted load, torque and transmitted power.