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On the Stresses of Force-Fitted Gear and the Slipping at Force-Fitted
Surface when the Influence of Centrifugal Force is considered*

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The purpose of the present paper is to examine the influence of centrifugal force on stresses at each position of a force-fitted gear under running condition, and to discuss on safe designing of a force-fitted gear when the influence of centrifugal force is considered.

The following conclusions are obtained.

(1) The stress caused by centrifugal force becomes maximum at the position where the center line of tooth space crosses the root circle, and the stresses at each position of gear and the decrease of interference at force-fitted surface are proportional to square of number of revolutions.

(2) A practical formula to decide the maximum stress, the fillet stress at the position where Hofer's weakest section crosses the trochoid curve at the root fillet of tooth and the decrease of interference can be introduced.

1. Introduction

In the present paper, the changes of stresses at each position and interference at fitted surface of force-fitted spur gears have been discussed, that is, by using finite element method (F.E.M.) the stresses in radial and tangential directions at each position of force-fitted spur gears have been calculated in the case of running condition and compared with those of disks (force-fitted disks having the outside diameter equal to the root circle diameter of gears used in theoretical calculation).

Moreover, the influence of dimensions and size (module, number of teeth, rim thickness), material and running condition (number of revolutions) of the gears on stresses and interference at force-fitted surface have been examined. On the other hand, the experimental results of some gears were compared with theoretical ones. Next, the practical formulas for the maximum stresses,

the fillet stresses at the root of tooth and the decrease of interference at force-fitted surface were introduced when the influence of centrifugal force was considered. Consequently, the influence of stresses caused by centrifugal force on strength at each position and safety for slipping at fitted surface of force-fitted spur gears have been discussed.

2. Numerical method and dimensions of gears used in calculation

Mesh division diagrams of force-fitted disk and gear are shown respectively in Figs. 1, 2. The restricted conditions of these diagrams divided into triangular elements are fixed in tangential direction at both ends and free in radial direction. Load in radial direction at each jointing point (vertices of triangular element), caused by centrifugal force was calculated by LM method in which the mass of each triangular element is divided uniformly in three jointing points of each triangular element. The calculation was done for gears made of three kinds of materials (S45C, FC25, A1080).

Dimensions of gears used in this calculation are shown in Table 1, that is,

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Number of nodal points	511
Number of elements	976

Fig.1 Mesh division diagram
(disk, $d_s = 300\text{mm}$, $T = 40.0\text{mm}$)

Fig.2 Mesh division diagram (gear, $m = 8$, $z = 40$, $T = 40.0\text{mm}$)

eighty kinds of standard involute gears having dimensions of rim thickness $T=7.9, 12.4, 30.0, 40.0$ (mm), pressure angle $\alpha=20^\circ$, tooth profile B type (top clearance $c=0.25m$), tooth width $b=10$ (mm) were adopted.

On the other hand, two kinds of force-fitted disks having the dimensions of outside diameter $d_o=300$ (mm), rim thickness $T=20.4, 40.0$ (mm) were used to compare with the case of gear. The outside diameter of these disks is equal to the root circle diameter of gear of module $m=8$, number of teeth $z=40$.

3. Stress distribution

3.1 Stress at each position of force-fitted disk, caused by centrifugal force

The calculated results of stresses (σ_r, σ_t) in radial and tangential directions are shown representatively in Figs.3,4 when the disk rotates at 1000rpm. The disks are made of S45C (moduli of longitudinal elasticity $E=205.9$ GPa, Poisson's ratio $\nu=0.30$, specific gravity $\gamma=7.86$), FC25 ($E=98.0$ GPa, $\nu=0.26$, $\gamma=7.23$), A1080 ($E=70.6$ GPa, $\nu=0.33$, $\gamma=2.71$) and have the dimensions of outside diameter $d_o=300$ (mm), inside diameter $d_i=220$ (mm), $T=40.0$ (mm). The results (actual line in these figures) calculated by theoretical equation based on two-dimensional elastic theory for disk of rotation are shown together to compare with those obtained by F.

E.M. Both results show almost the same values, and from this fact it is found that the analytical method by F.E.M. mentioned above (element-dividing method, restricted condition, loading condition) may be used for calculation of stresses at each position of a force-fitted gear, caused by centrifugal force. Stress σ_r becomes zero at force-fitted surface and outside diameter of disk, and becomes maximum at middle part of both positions.

On the other hand, stress σ_t becomes maximum at force-fitted surface (inside diameter) and is inverse to the radius r of disk. But the values of σ_t are larger considerably than those of σ_r . Fig.5 shows the calculated results of σ_r, σ_t when number of rev-

Fig.3 Relation between σ_r and r when disk is made of S45C, FC25 or A1080

Fig.4 Relation between σ_t and r ($d_o=300\text{mm}$, $T=40.0\text{mm}$, $n=1000\text{rpm}$)

Table 1 Dimensions of gears used for theoretical analysis

Module	Number of teeth	Root circle diameter	Module	Number of teeth	Root circle diameter
6	39	219.0	10	25	225.0
	53	303.0		33	305.0
	66	381.0		40	375.0
	79	459.0		49	465.0
8	30	220.0	12	21	222.0
	40	300.0		28	306.0
	50	380.0		34	378.0
	60	460.0		41	462.0

olutions of a disk made of S45C ($d_k=300\text{mm}$, $T=40.0\text{mm}$) is 1000, 2000 or 5000rpm. Both stresses (σ_r , σ_t) are proportional to the square of number of revolutions of disk, and agree well with the results calculated by theoretical equation based on two-dimensional elastic theory.

Fig.5 Relation of σ_t , σ_r with r when number of revolutions changes (S45C, $d_k=300\text{mm}$, $T=40.0\text{mm}$)

3.2 Stresses at each position of force-fitted gear rim, caused by centrifugal force

Figs.6~11 show the calculated results of stresses at the center of tooth space (A-A' section in Fig.2) and tooth (B-B' section in Fig.2), caused by centrifugal force when number of revolutions of gears (made of S45C, FC25, Al080) having the dimensions of $m=8$, $z=40$, $T=40.0$ (mm) is 1000rpm. In these figures, symbols F.S., R.C. and T.C. denote respectively force-fitted surface (inside diameter of the gear rim), root circle and tip circle. Radial stresses at A-A' section show different tendencies from those of a disk except the circumference of force-fitted surface and the maximum stress occurs at the position (A-A' section) of a gear rim near the root circle. These stresses in each figure show similar tendency and are not influenced by the dimensions of gears. On the other hand, stresses at B-B' section show similar tendency to those of a disk in the range from force-fitted surface (inside diameter of the gear rim) to root circle, that is, in any case these stresses become tensile and increase in the range from force-fitted surface to the middle position of gear rim and become maximum at the middle position, thereafter the tensile stress decreases and changes compressive one at a certain position (of r). Moreover, the compressive stress becomes maximum at circumference of root circle and changes to tensile one again, thereafter approaching zero at tip circle. These tendencies are similar in any case of gear material used for this theoretical calculation. From these calculated results, it was found that the range of radii (r) changing the tensile stress at circumference of force-fitted surface to compressive stress becomes large as T/m gets larger and the tensile stress at circumference of force-fitted surface approaches zero

Fig.6 Relation between σ_r and r (S45C, $m=8$, $z=40$, $T=40.0\text{mm}$, $n=1000\text{rpm}$)

Fig.7 Relation between σ_t and r (S45C, $m=8$, $z=40$, $T=40.0\text{mm}$, $n=1000\text{rpm}$)

Fig.8 Relation between σ_r and r (FC25, $m=8$, $z=40$, $T=40.0\text{mm}$, $n=1000\text{rpm}$)

Fig.9 Relation between σ_t and r (FC25, $m=8$, $z=40$, $T=40.0\text{mm}$, $n=1000\text{rpm}$)

Fig.10 Relation between σ_r and r (Al080, $m = 8$, $z = 40$, $T = 40.0$ mm, $n = 1000$ rpm)

Fig.11 Relation between σ_t and r (Al080, $m = 8$, $z = 40$, $T = 40.0$ mm, $n = 1000$ rpm)

as T/m is smaller.

Next, in the range from force-fitted surface to position one module inside from root circle, tangential stresses σ_t at A-A' and B-B' section decrease slightly in inverse proportion to radii r in the same manner as in the case of disk. Thereafter the stresses at A-A' section increase rapidly as radius r becomes larger, but on the contrary to this tendency those at B-B' section decrease rapidly. As the reason for this, it is considered that the tangential stresses at tooth are released free in tangential direction due to tooth space. The stresses (σ_t) of the gear rim made of Al080 are smaller than those of S45C, FC25. This is due to a small value of the specific gravity of the gear rim made of Al080. In comparison of σ_r with σ_t , σ_t at both section (A-A', B-B') is larger (about one figure) than σ_r at the same section. As obvious from the facts mentioned above, the maximum stress of force-fitted gear rim, caused by centrifugal force in the running condition, occurs in tangential direction at position (A in Fig.2) where A-A' section crosses the root circle.

Besides, when number of revolutions of gear ($m=8$, $z=30$, $T=30.0$ mm) made of S45C is 1000, 3000 or 5000rpm, the calculated results of σ_r , σ_t turn out as shown in Figs.12~15. From these figures, it is found that σ_r , σ_t are proportional to the square of number of revolutions in the same manner as in the case of disk. Gears made of FC25, Al080 have the same tendency as the gear made of S45C as mentioned above.

Fig.12 Relation between σ_r (at A-A' section) and r when number of revolutions changes

Fig.13 Relation between σ_t (at A-A' section) and r when number of revolutions changes

Fig.14 Relation between σ_r (at B-B' section) and r when number of revolutions changes

Fig.15 Relation between σ_t (at B-B' section) and r when number of revolutions changes

4. Practical formulas

4.1 Introduction of practical formulas

By the F.E.M. mentioned above, maximum stress $\sigma_{\max}$, stress σ_H at the position where Hofer's weakest section crosses a trochoid curve at the root fillet of tooth and radial displacement ΔX_c at force-fitted surface can be calculated, but the application of this calculation for many kinds of gears is too complicated. Therefore, the practical formulas to decide easily stresses $\sigma_{\max}$, σ_H and radial displacement ΔX_c were worked out.

Stress (general form σ_{ic}) and radial displacement (ΔX_c) at each position of a gear rim, caused by centrifugal force are decided by module m , number of teeth z , thickness of gear rim T , pressure angle α , top clearance c , moduli of longitudinal elasticity E , Poisson's ratio ν , specific gravity γ and number of revolutions n , that is,

$$\sigma_{ic} = F_1(m, z, T, \alpha, c, E, \nu, \gamma, n) \quad (1)$$

$$\Delta X_c = G_1(m, z, T, \alpha, c, E, \nu, \gamma, n) \quad (2)$$

Now, considering the above equation for each material in the case of standard spur gear with tooth profile of B type, α , c , E , ν , γ become constant, consequently Eqs. (1), (2) can be written in the following form.

$$\sigma_{ic} = F_2(m, z, T, n) \quad (3)$$

$$\Delta X_c = G_2(m, z, T, n) \quad (4)$$

As obvious from the results of theoretical analysis, σ_{ic} , ΔX_c are proportional to square of number of revolutions when m , z and T are constant, therefore Eqs. (3), (4) are rewritten as follows.

$$\sigma_{ic} = F_3(m, z, T)n^2 \quad (5)$$

$$\Delta X_c = G_3(m, z, T)n^2 \quad (6)$$

From above equations, practical formulas for stresses σ_{ic} (maximum stress, stress at Hofer's weakest section) and radial displacement ΔX_c will be introduced, that is, by deciding F_3 , G_3 for gears made of each material, practical formulas for σ_{ic} and ΔX_c can be obtained.

4.2 Maximum stress

As evident from the results of theoretical analysis, σ_{ic} is larger (about one figure) than σ_r and maximum ($\sigma_{\max}$) of σ_{ic} occurs at position (A in Fig.2) where A-A' section crosses the root circle in the case of gears used in this study.

The relation between $\sigma_{\max}$ and T/m in the case of $m=8$, $n=1000$ rpm is shown in Figs.16~18. The change of $\sigma_{\max}$ for T/m is hyperbolic, that is, $\sigma_{\max}$ becomes larger as T/m gets smaller. This tendency is similar to that in the case of other modules.

Now, $\sigma_{\max}$ can be written in the following form if $\sigma_{\max}$ is assumed to be a power function of T/m .

$$\sigma_{\max} = a_m (T/m)^{b_m} \quad (7)$$

Coefficients a_m , b_m are obtained as follows by power regression analysis using the method of least squares. Here, a_m , b_m are power functions of z .

$$a_m = a_n z^{a_n'} m^2 \quad (8)$$

$$b_m = b_n z^{b_n'} / m^{1/4} \quad (9)$$

Substituting Eqs.(8),(9) into Fig.(7), $\sigma_{\max}$ will be

$$\sigma_{\max} = a_n z^{a_n'} (T/m)^{b_n z^{b_n'} / m^{1/4}} m^2 \quad (10)$$

Coefficients a_n , a_n' , b_n , b_n' obtained by the method of least squares are given by the following equations for each material.

$$\text{S45C} \quad \left. \begin{aligned} a_n &= 6.97 \times 10^{-12}, & a_n' &= 2.05 \\ b_n &= -6.95, & b_n' &= -0.527 \end{aligned} \right\} \quad (11)$$

$$\text{FC25} \quad \left. \begin{aligned} a_n &= 2.26 \times 10^{-12}, & a_n' &= 2.37 \\ b_n &= -4.97, & b_n' &= -0.431 \end{aligned} \right\} \quad (12)$$

$$\text{A1080} \quad \left. \begin{aligned} a_n &= 4.73 \times 10^{-13}, & a_n' &= 2.55 \\ b_n &= -4.03, & b_n' &= -0.365 \end{aligned} \right\} \quad (13)$$

By substituting these coefficients into Eq.(10) practical formula of $\sigma_{\max}$ can be introduced and the applicable range of the practical formula is module $m=4 \sim 12$, rim thickness $T=7.9 \sim 40.0$ mm ($T/m=7.9/12 \sim 40.0/6$).

Fig.16 Relation between $\sigma_{\max}$ and T/m (S45C, $m=8$, $n=1000$ rpm)

Fig.17 Relation between $\sigma_{\max}$ and T/m (FC25, $m=8$, $n=1000$ rpm)

Fig.18 Relation between $\sigma_{\max}$ and T/m (A1080, $m=8$, $n=1000$ rpm)

4.3 Stresses at position where Hofer's weakest section crosses trochoid curve at the root fillet of tooth

The relation between σ_H and T/m is shown in Figs.19~21, and σ_H is stress at position (c in Fig.2) where Hofer's weakest section crosses a trochoid curve at the root fillet of tooth. The practical formula of σ_H is given in the following form in the same manner as in the case of σ_{max} .

$$\sigma_H = c_n (T/m)^{d_n} \dots\dots\dots (14)$$

In the same manner as mentioned above, it is found that $c_n/m^2, d_n m^{1/4}$ are power functions of z . Therefore practical formula of σ_H is expressed as follows.

$$\sigma_H = c_n z^{c_n'} (T/m)^{d_n z^{d_n'}} / m^{1/4} m^2 n^2 \dots\dots\dots (15)$$

Fig.19 Relation between σ_H and T/m (S45C, $m = 8$, $n = 1000$ rpm)

Fig.20 Relation between σ_H and T/m (FC25, $m = 8$, $n = 1000$ rpm)

Fig.21 Relation between σ_H and T/m (A1080, $m = 8$, $n = 1000$ rpm)

Coefficients c_n, c_n', d_n, d_n' for each material of gears are given by the following equation.

$$\text{S45C } \left. \begin{aligned} c_n &= 5.73 \times 10^{-12}, & c_n' &= 1.99 \\ d_n &= -4.20, & d_n' &= -0.424 \end{aligned} \right\} \dots\dots (16)$$

$$\text{FC25 } \left. \begin{aligned} c_n &= 1.66 \times 10^{-12}, & c_n' &= 2.35 \\ d_n &= -3.16, & d_n' &= -0.337 \end{aligned} \right\} \dots\dots (17)$$

$$\text{A1080 } \left. \begin{aligned} c_n &= 4.26 \times 10^{-13}, & c_n' &= 2.48 \\ d_n &= -3.34, & d_n' &= -0.349 \end{aligned} \right\} \dots\dots (18)$$

By substituting Eqs.(16)~(18) into Eq. (15), a practical formula of σ_H in the range of $T/m = 7.9/12 \sim 40.0/6$ can be obtained.

4.4 Radial displacement of force-fitted surface

Radial displacement at position where A-A' section crosses force-fitted surface is larger than that at position where B-B' section crosses force-fitted surface. Therefore, practical formula of radial displacement must be discussed for the displacement (ΔX_{CA}) at position (A' in Fig.2) where A-A' section crosses force-fitted surface (inside diameter of gear rim). Figs.22~24 show the

Fig.22 Relation between ΔX_{CA} and T/m (S45C, $m = 8$, $n = 1000$ rpm)

Fig.23 Relation between ΔX_{CA} and T/m (FC25, $m = 8$, $n = 1000$ rpm)

Fig.24 Relation between ΔX_{CA} and T/m (A1080, $m = 8$, $n = 1000$ rpm)

relation between ΔX_{CA} and T/m in the case of $m=8$, $n=1000\text{rpm}$. As obvious from these figures, ΔX_{CA} is a power function of T/m . Therefore, a practical formula of ΔX_{CA} is given in the following form.

$$\Delta X_{CA} = e_n (T/m)^{f_n} \dots \dots \dots (19)$$

Here, Eq.(19) can be rewritten as the following equation in the same manner as in the cases of σ_{max} , σ_H .

$$\Delta X_{CA} = e_n Z'^n (T/m)^{f_n} z'^n / m^{1/4} m^2 n^2 \dots \dots \dots (20)$$

Coefficients e_n , e_n' , f_n , f_n' for gears made of each material are as follows.

$$\text{S45C } \left. \begin{array}{l} e_n = 4.97 \times 10^{-15}, \quad e_n' = 2.50 \\ f_n = -8.32, \quad f_n' = -0.596 \end{array} \right\} \dots (21)$$

$$\text{FC 25 } \left. \begin{array}{l} e_n = 2.76 \times 10^{-15}, \quad e_n' = 2.87 \\ f_n = -5.87, \quad f_n' = -0.495 \end{array} \right\} \dots (22)$$

$$\text{A1080 } \left. \begin{array}{l} e_n = 8.93 \times 10^{-16}, \quad e_n' = 3.02 \\ f_n = -4.63, \quad f_n' = -0.424 \end{array} \right\} \dots (23)$$

By using the same means as mentioned above, the practical formula of ΔX_{CA} in the case of $T/m=7.9/12 \sim 40.0/6$ can be obtained.

5. Experimental method and the results

5.1 Experimental apparatus

The apparatus used in this experiment is shown in Fig.25. In this experimental apparatus, a motor (3.7KW, 4 poles, made by Toshiba Electric Co. Ltd.) is coupled directly to a stepless transmission (Kopp, 60MH3-T), which is connected to a shaft (fixed test gear) that can be revolved in the range of $n=700 \sim 8000\text{rpm}$.

Fig.25 Experimental apparatus

5.2 Measuring instruments

Instruments used in measurement of stresses are strain gauges (made by Shinko Communication Industry Co. Ltd., SR4, Type-13-FAE-0.5W-12T11), slipring (made by Michigan Scientific Co. Ltd., SR20-MSER#494), dynamic strain meter (made by Shinko Communication Industry Co. Ltd., DS6-MTC type), recorder (made by Watanabe Co. Ltd., WTR331) etc. Moreover, number of revolutions of a shaft fixed test gear was measured by using a stroboscope (made by Toshiba Electric Co. Ltd., SS6 type).

5.3 Test gears and a test disk

Dimensions of standard spur gears used in this experiment are pressure angle $\alpha=20^\circ$, tooth profile B type (top clearance $c=0.25m$), that is, three kinds: (1) module $m=8$, number of teeth $z=40$, rim thickness $T=40.0\text{mm}$, interference $\delta'=0.12\text{mm}$; (2) $m=8$, $z=30$, $T=20.4\text{mm}$, $\delta'=0.19\text{mm}$; and (3) $m=8$, $z=30$, $T=40.0\text{mm}$, $\delta'=0.07\text{mm}$. On the other hand, the dimensions of a test disk are outside diameter $d_k=300\text{mm}$, rim thickness $T=40.0\text{mm}$ and interference $\delta'=0.12\text{mm}$. Force-fitted gears and a disk are made of S45C.

5.4 Experimental method

Measuring positions of stresses of force-fitted gears and a force-fitted disk are shown in Figs.26, 27. The strain gauges are pasted on positions shown in these figures, that is, a bridge per one measuring position is made by combining a strain gauge for measurement with three strain gauges for compensation and the strain was measured by using a slipring, a dynamic strain meter and a recorder (pen type). In this case, three strain gauges for compensation are pasted in axial direction at parts of outside surface of disk and tooth crest of gear, where there is hardly any influence of centrifugal force.

Fig.26 Measuring positions of stresses (disk, $d_k=300\text{mm}$)

Fig.27 Measuring positions of stresses (gear, $m=8$, $z=40$, $T=40.0\text{mm}$)

5.5 The experimental results

Figs.28~30 show the measured stresses of a force-fitted disk and a force-fitted gear ($m=8$, $z=40$, $T=40.0\text{mm}$, $\delta'=0.12\text{mm}$) mentioned in chapter 5.3 for number of revolutions $n=1000$, 2000rpm . The experimental results in these figures are the average of five measured values, and the results calculated by theoretical equation based on

two-dimensional elastic theory are shown together in these figures to compare with the experimental results. A difference between the experimental results and the theoretical ones (F.E.M.) in both cases of force-fitted disk and gear is under 7%, and the results in both cases agree fairly well.

Fig.28 Measured results of stress σ_t (disk)

Fig.29 Measured results of stress σ_r (gear)

(→ mark in this figure shows that X mark moves to the right in parallel to horizontal line and joins ● mark)

Fig.30 Measured results of stress σ_t (gear)

6. Discussion on strength of force-fitted gear and slipping at force-fitted surface

6.1 Strength of force-fitted gear
The tangential stresses at each position of the gear rim, caused by centrifugal force become maximum (σ_{max}) at the position (A in Fig.2) where A-A' section crosses the root circle. Therefore, strength at this position must be discussed. Now, let us dis-

cuss about force-fitted gear rim (S45C, induction hardening, $\alpha=20^\circ$, tooth profile B type, $m=8$, $z=40$, $T=40.0\text{mm}$, $\delta=0.09\text{mm}$). Yield point of this gear rim is about 490MPa and it becomes 392MPa when safety is set at 80%. Limit number of revolutions calculated by the practical formula mentioned above is 10990rpm when the influence of centrifugal force only is considered. In the case of force-fitting gear, either of stresses at two positions (A and C in Fig.2) becomes maximum and its position is decided by the kind of tooth profile.

Because stress σ_{BAF} at position A, caused by force-fitted acts in tangential direction at root circle, the combined stress at position A should be σ_{max} added to σ_{BAF} as it is. In this case, the value is 88.2MPa and the limit number of revolutions becomes 9670rpm.

The stress at Hofer's section (C in Fig.2) is then discussed. σ_H of Eq.(15) mentioned above is tangential at C in Fig.2. Therefore, stress along trochoid curve at the root fillet of tooth becomes $\sigma_H \cos \theta$ when the angle between tangent-line for root circle and that for trochoid curve at position C in Fig.2 is θ , and the angle θ is $50^\circ \sim 60^\circ$ in the range of calculation and experiment of this study. Consequently the combined stress at position C in Fig.2 should be σ_R (84.7MPa in this case, caused by force-fitting) added to $\sigma_H \cos \theta$, and the limit number of revolutions becomes 9725rpm in the same manner as at position A in Fig.2. Moreover, the combined stresses at positions A and C when gear tooth is in meshing condition must be stress caused by loading added to each stress mentioned above. Therefore, the limit number of revolutions in this case becomes smaller than those mentioned above.

6.2 Decrease of interference

When the dimensions of a force-fitted gear are equal to those in chapter 6.1 and number of revolutions of gear is 6000rpm, the calculated results [by Eq.(20)] of radial displacement of gear rim and force-fitted hub (solid) are respectively 4.8×10^{-3} (mm) and 3.3×10^{-3} (mm) and the difference becomes 4.47×10^{-2} (mm), consequently interference δ decreases by about 50% of 0.09 (mm) and the dynamic limit slipping torque becomes very small. But stress σ_{BAF} , σ_R caused by force-fitting becomes small [$\sigma_{BAF}=44.1$ (MPa), $\sigma_R=42.4$ (MPa) in this case] due to the decrease of interference as mentioned above, consequently the limit number of revolutions becomes 10350rpm at position A and 10375rpm at position C and these values are larger than those of the case in chapter 6.1.

7. Conclusions

- (1) The influence of the dimensions and the rim thickness of gear on stresses at each position of force-fitted spur gear rim and interference at force-fitted surface, caused by centrifugal force has been clarified. The experimental results agree well with the theoretical ones.
- (2) Tangential stress caused by centrifugal force becomes maximum at position

A (in Fig.2) in the case of a force-fitted gear.

(3) In the case of force-fitted gear rim, radial displacement at position A (Fig. 2) of force-fitted surface, caused by centrifugal force is larger than that at position B (Fig.2).

(4) Practical formulas for the maximum stress caused by centrifugal force, stress at position where Hofer's weakest section crosses a trochoid curve at the root fillet of tooth and radial displacement at force-fitted surface are introduced.

(5) In this study, the process and the method for safe designing of strength and slipping (at fitted surface) of a force-fitted spur gear have been established when

the influence of centrifugal force is considered.

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