5.1 Introduction

The “Planetary gear and Mechanical paradox Gear design system” software can do the design of the planet gear and the mechanical paradox gear easily. It is possible that it automatically decides combination and center distance of the troublesome teeth number, etc. in the design of the planet gear and simply designs gear dimension and gear strength. And, it is possible to also easily calculate interference check of planet gear and decision of addendum modification coefficient, efficiency calculation (See Fig.5.1).

5.2 Application

(1) Type: Equal arrangement (Planetary, Solar, Star)
(2) Material of the gear: Steel, Plastic
(3) Tooth profile: Involute
(4) Option: 3K paradox type, Small number of teeth, Double pinion, Non-equal arrangement

5.3 Initial setting

In the setting the type of the tip diameter, basic rack, the design criterion (module or center distance), gear accuracy and friction coefficient (See Fig.5.2).

5.4 The selection of the planetary gear mechanism

The planet gear type is chosen (See Fig.5.3). (Planetary, Solar, Star and 3K paradox type)

5.5 Gear dimension

The gear dimension input screen is shown in Fig. 5.4. (1) The number of the planet gear is 1~21. (2) There are system for inputting directly and system for choosing from the teeth number list (Fig. 5.5) calculated from velocity ratio on the teeth number. (3) There are determination method of module which made center distance to be a standard and two methods for deciding center distance from module on the relation between center distance and module. (4) The addendum modification coefficient is calculated from module and center distance. (5) The default value of thinning for backlash is 1/2 of the JIS backlash standard middle value. (6) Though the tip diameter is calculated from basic rack and addendum modification coefficient, it is also possible on the change. (7) Root of tooth of external gear is fillet shape which is created by the generating motion. And, root of tooth of internal gear is single R type. (8) It is possible to add R in the addendum edge of the gear.
The number of teeth list (9) The other addendum-modification-coefficient is connected when the addendum-modification-coefficient changes one kind and changes. However, it is possible to enter respectively, too.

(10) It is possible to use Fig. 5.6 for clearance adjustment and tooth thickness adjustment. Then, it is possible to confirm addendum modification coefficient and tooth profile which tip diameter was changed and clearance, interference in this screen. However, the tooth profile here is only tooth surface, and the root of tooth shape is not included.

Support of the dimension (11) The calculation result after it determines the gear dimension is shown in Fig. 5.7 ~ 5.10.

Tooth profile figure 5.6 Tooth profile figure
5.6.1 Tooth profile (2D)
The 2D-meshing of the tooth profile is displayed in Fig. 5.11 and Fig. 5.12. The confirmation of the contact location in tooth surface is easy, because to display support circle and common normal line in the control screen is possible. And, it is possible to display by changing rotation angle of the gear in the zoom.
5.6.2 Meshing of a pair tooth profile (2D)

It is possible to confirm the meshing of a tooth pair in Fig. 5.13. And, it is possible to confirm the interference of internal gear and addendum of external gear and root of tooth part in detail in this screen.

5.6.3 Meshing of teeth profile (3D)

The meshing of the gear is displayed in the three-dimensional figure. Then, it is possible that the gear is made to rotate in the X, Y, Z axial direction. And, control form of the tooth profile rendering is shown in Fig. 5.16.

5.7 Sliding ratio graph

A sliding ratio graph is shown in Fig. 5.17 and Fig. 5.18.

5.8 Gear strength

5.8.1 Initial setting

It is possible to choose steel and plastic material in the setting initial screen (See Fig. 5.19). Bending allowable stress ($\sigma_{\text{flim}}$) and Hertz allowable stress ($\sigma_{\text{Hlim}}$) are chosen from Fig. 5.20. Or, it is possible to input the allowable stress directly. It is possible to choose the torque unit from "N\cdot m", "N\cdot cm", "kgf\cdot m", "kgf\cdot cm", "gf\cdot cm".
5.8.2 Input (Gear strength dimension)

The numerical value is input into the strength dimension input screen (See Fig. 5.21). Torque and rotational speed can be set even in the output side even in the input side.

5.8.3 Gear strength calculation result

The strength calculation result screen is displayed in Fig. 5.22 and Fig. 5.23. The strength calculation also considers efficiency and contact ratio.

The steel gear does the strength calculation based on JGMA401-01:1974 and JGMA401-02:1975. Allowable stress value of the plastic gear material has adopted temperature and material experimental value considering life, etc.

5.9 Hertzian stress graph

The Hertz stress graph is shown in Fig. 5.24 and Fig. 5.25. By this graph, It is possible to confirm the difference between the Hertz stress of the meshing.
5.10 Other

(1) Tooth profile output
   - DXF file : 2D, 3D All teeth meshing
   - IGES file : 3D(Tooth)
   - TXT file : 2-dimensional tooth profile coordinate

(2) Print out (Gear dimension, Strength, Tooth form figure, Sliding ratio graph, Hertzian stress graph)

(3) Saving and reading of design data

5.11 Mechanical paradox Gear design system (3K-Type)

Mechanical paradox Gear which increases a gear reduction ratio is well known but this design is very troublesome. However, it is possible to simply design Mechanical paradox Gear by using this software.

This mechanism uses the one sun gear, one planet gear, the two inside gears. Same direction deceleration and reverse-directional deceleration depend on the number of teeth of internal gear-1 and internal gear-2. Design example is shown in the following.

5.11.1 Setting of gear dimension

(1) The module standard is chosen in the setting screen dimension the initial stage (Fig.5.2).

(2) It chooses a 3 K type in the planet gear of Fig. 5.3.

(3) It inputs 135 in the gear reduction ratio as the example and it inputs three in the number of the planet gear.

(4) The teeth number list screen is displayed, and the combination of the appropriate teeth number is chosen (See Fig.5.27).

Note point in this case
(a) The error of real velocity ratio and design velocity ratio.
(b) The number of teeth isn't too small and isn't too big.
(c) That there is the relation of “z1+2×z 2” between the z3 and z4 tooth number of the internal gear.

Here, the following are chosen as an example: z1=20, z2=31, z3=82 and z4=85.

Fig.5.26 Input of gear dimension

(5) The tooth profile is confirmed in the input support screen (See Fig.5.28).

Fig.5.27 The number of teeth list

(6) Efficiency and contact ratio and sliding ratio of the size calculation result screen are confirmed, as it is shown in Fig. 5.29~5.31. The efficiency of the mechanical paradox gear of this case is 73.1%, as it is shown in Fig. 5.32.

The dimension result is shown in the following (See Fig.5.29 ~5.32).

Fig.5.28 Input support screen

Fig.5.29 Result (Gear dimension)

Fig.5.30 Result (Tooth thickness)
5.11.2 Tooth profile (2D)

A meshing figure is shown in figure 5.33. In the enlarged view of figure of 5.34, two internal gears prove well that it has meshing in planet gear each other. And, it is possible to observe the aspect of the meshing rotation of the mechanical paradox gear in tooth profile rendering shown in Fig. 5.35.

5.11.3 Mechanical paradox Gear (Spur gear)

1. It is possible to calculate gear strength calculation and sliding ratio and Hertz stress graph as well as the planet gear (explanation omission).
2. Construction example of the Mechanical paradox Gear of the spur gear is shown in figures of 5.36.

5.12 Small number of teeth (Option)

The number of teeth can design a planet gear below 4 teeth. The minimum number of teeth is 1 tooth. The small teeth number makes angle of helix increase, because the transverse contact ratio decreases. The construction example is shown in figures of 5.37 and 5.38.
5.13 Double pinion (Option)

The design example of the double pinion is shown in the following.
5.13 Outline

To the Planetary gear and Mechanical paradox Gear design system software, the Non-equality position of planet gear option was added. This can be designed by the feeling which is the same as the equality position of planet gear. But, it doesn't apply to the Mechanical paradox Gear and the double pinion.

5.14 Design example

The design example of the Non-equality position of planet gear of the planetary type is shown in the following. In the condition of the equality position, it becomes and so on sun teeth number of 15, planet teeth number of 21, internal gear teeth number of 57 in case of Fig. 5.51.

When making an internal-gear number of teeth 56 here, it doesn't become equality position of planet gear.

Therefore, it makes \( \text{Direct entry of number of teeth} \) \( \text{non-equality position of planet gear} \) valid.

The screen which changed the number of teeth of the internal-gear into 56 is shown in Fig. 5.52. The addendum modification coefficient of internal gear is big, because the module is \( m_n 1.8 \) equal to Fig. 5.51. Though the example shows the spur gear, it is possible to also design the helical gear. And, the function of \( \text{Consider input vale/tooth} \) is also usable. See Fig. 5.6 for more information.

Dimension results, etc. are shown in Fig. 5.53~5.55.
As for the Non-equality position of planet gear, the [A1] gear of figure 5.56 is a standard position. And, it chooses from angle table shown at the [B] of figures of 5.57, because to freely input the Non-equality position angle is not possible. In case of the example, configuration angle of the planet gear exists in 71 types.

The tooth profile meshing in choosing the second 10.1408 degrees from the angle of [B] of Fig. 5.57 is shown in Fig. 5.58. Enlarged view and tooth profile rendering of [C] are shown in Fig.5.59 and 5.60.
In addition, gear strength calculation, tooth profile data file output, etc. are equal to the basic software. Non-equality position of planet gear design example of the helical gear is shown in Fig. 5.61~5.63.

Fig. 5.61 Non-equality position of planet gear design

Fig. 5.62 Non-equality position-3

Fig. 5.63 Tooth rendering