Recent Trends in Gearing Technology*

Kiyohiko UMEZAWA**

The main aim of gearing technology, at its present state of maturity as an applied science, should be a continued effort to minimize noise and vibration in two industries where they are especially critical: office-automation equipment and automobile production. One obstacle facing the advancement of this technology is the decline in qualified gear-engineering specialists from the workforce. A solution to this problem is presented in this paper as a simplified means of analyzing low-vibration parallel gear pairs through computer-aided-design, vibration-performance diagrams. Through the development of more sophisticated computer-simulated analysis techniques, the shortage of qualified engineers may be offset by allowing less-qualified engineers greater access to this important but highly specialized technology.

Key Words : Gear, Parallel Gear, Vibration, Design, Computer, Simulation, Finishing, Trend

1. Introduction

It is probable that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in Japanese industry from heavy industry as shipbuilding to industries such as automobiles and office automation will necessitate also a refined application of gear technology.

In the automobile industry, the largest manufacturer of gears, higher reliability and lighter-weight gears are necessary as lighter automobiles continue to be in demand. In addition, the success in engine noise reduction makes the requirement to produce quieter gear pairs for further noise reduction. Noise reduction in gear pairs is especially critical in the rapidly grow-

ing field of office-automation equipment as the office environment is so easily disturbed by noise, and machines are playing an ever-widening role in that environment. Ultimately, the only effective way to achieve gear noise reduction is to reduce the vibration associated with them.

The reduction of noise through vibration control can only be achieved through a concerted research effort by specialists in the field. However, a shortage of these specialists exists in the newer, lightweight industries in Japan mainly because fewer young people are specializing in gear technology today and traditionally, the specialists employed in heavyweight industries tend to stay where they are. In order to compensate for the loss of expertise in the field, gear technology, through the use of computer simulation techniques, will necessarily have to become more accessible to the generally educated engineer.

2. A Newly Developed Method to Predict the Vibration of a Helical Gear Pair

2.1 The developed simulator⁽¹⁾⁽²⁾

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** Research Laboratory of Precision Machinery and Electronics, Tokyo Institute of Technology, Nagatsuta, Midori-ku, Yokohama, 227, Japan

2.1.1 Theoretical analysis on the vibration of a narrow facewidth helical gear pair The studies on a helical gear are very different from the ones on a spur gear. A simple outline of the theoretical analysis on the vibration of a helical gear will be discussed.

LENGTH OF PATH OF CONTACT-The plane of action of a helical gear pair is shown in Fig. 1. A pair of mated teeth starts meshing at point S on the plane of action. The meshing of the pair proceeds on the plane with the movement of the contact line (for example, CC_c in Fig. 1) and finishes at point E. The position of the line-of-contact is represented by the coordinate Y along the line-of-action of the helical gear (hereafter simply stated as the line-of-action) which is considered to be the middle of the facewidth. That is, the starting point of meshing S is substituted by S' , the position of line-of-contact CC_c by C' and the ending point E by E' on the line-of-action.

VIBRATION MODEL - When the rotational vibration of a power-transmitting helical gear pair is considered along the line-of-action, analysis can employ a single-degree-of-freedom model similar to the case of a spur-gear pair⁽¹⁾, in which the tooth is replaced by a spring and the gear blank is replaced by a mass. This model is shown in Fig. 2 and can be expressed as a linear vibration system by the following equation:

$$M\ddot{\delta} + D\dot{\delta} + K(t, \delta)\delta = W + F(t, \delta), \quad (1)$$

where δ is the relative displacement along the line-of-action, M the equivalent inertia mass of the gear pair on the line-of-action, D the damping coefficient, $K(t, \delta)$ the mesh stiffness of the gear pair, W the static transmitting load and $F(t, \delta)$ the exciting force caused by profile modification or errors $e(t, \delta)$.

STIFFNESS OF A HELICAL PAIR-Helical gear

Fig. 1 Meshing of a helical pair

teeth transmit a load on the line-of-contact inclining at an angle to the tip edge of the gear tooth as shown in Fig. 3, where x indicates the measuring coordinate and ξ is the loaded one.

Under the condition that the separation between tooth faces does not occur anywhere along the line-of-contact, the effect function of deflection $K_b(x, \xi)$, the effect function of compressive deformation by the contact between the gear teeth $K_c(x, \xi)$, the load distribution along the line-of-contact $P(\xi)$, and the deflection of a gear tooth pair W_j are related as

$$\int K_b(x, \xi, Y)P(\xi)d\xi + \int K_c(x, \xi, Y)P(\xi)d\xi = \delta_o - e(x, Y), \quad (2)$$

$$W_j = \int P(\xi)d\xi. \quad (3)$$

The total transmitting load W of a gear pair is given by

$$W = \sum_{j=1}^n W_j. \quad (4)$$

Therefore, $P(\xi)$ and W_j are evaluated with Eqs. (2) and (3).

In Eq. (2), the effect function of the deflection $K_b(x, \xi)$ was solved analytically in Ref. [3] and given by an approximate formula of deflection for the numerical analysis in Ref. [4]. The effect function of

Fig. 2 Vibration model along the line-of-action

Fig. 3 Stiffness of a helical tooth pair

compressive deformation $K_c(x, \xi)$ was also given as indicated in Ref. [5].

The stiffness K_j on the line-of-contact is defined by

$$K_j = \frac{dW_j}{d\delta} \quad (5)$$

The stiffness $K_j(Y)$ along the line-of-action is determined by each stiffness of meshing pairs on every line-of-contact.

Finally, the stiffness of a helical gear pair $K(t, \delta)$ is shown as

$$K(t, \delta) = \sum_{j=1}^n K_j \quad (6)$$

The load distribution $P(\xi)$ was obtained at first by solving Eq.(2).

REPRESENTING ERRORS ALONG THE LINE-OF-ACTION - When the profile error and lead error are discussed, it becomes necessary to consider changes in the amount of errors along the line-of-contact. However, when the equation is solved along the line-of-action of the helical gear, the distributed error along the line-of-contact should be depicted as a representative value on the line-of-action. In this study, the representative value is defined as the mean value of the distributed error.

It has been clarified by experimental results that this representation is reasonable (more precisely, see Ref. [2]).

2.1.2 Vibration simulation of a narrow-face-width helical gear The flow chart of the simulator developed for solving the vibration equation of a helical gear pair is shown in Fig. 4.

The input values initially required are the dimensions of a helical gear pair and a distribution of errors over the tooth face.

In the first step of the program, the positions of the line-of-contact are decided along the line-of-action with an interval of $1/360$ th of normal pitch. In the second step, all the effect functions of the deflection $K_b(\xi)$ and approach $K_c(\xi)$ in Eq.(2) are calculated along each line-of-contact with an use of an approximate equation for numerical solution [3]. The load distribution $P(\xi)$ is calculated with the use of Eq.(2) and the stiffness of a single tooth pair calculated against the point of the line-of-action through Eqs.(3) and (4).

The stiffness of a single tooth pair is solved along the line-of-action from the start to the end of meshing. The representative error of a pair is worked out with the mean value along the contact line.

The damping coefficient D is calculated from the relation $D = 2\zeta\sqrt{M \cdot \bar{K}(t, \delta)}$. The damping factor ζ is set at 0.07 as employed in Ref. [1].

Finally, every coefficient in Eq.(1), the equation-

of-motion, is calculated. The vibration is worked out numerically with the Runge-Kutta-Gill method producing waveform and rms value of acceleration.

2.1.3 Theoretical and experimental results The theoretically calculated and experimental results under a torque of 98 N·m at various rotational speeds from 800 to 2 000 rpm are shown in Fig. 5.

The calculated and the experimental results are in agreement with regards to the waveform, especially the change in the number of cycles within one tooth-meshing period T_z , and the behaviour of the amplitude with increasing rotational speed.

From the foregoing, it can be understood that, using mainframe computer technology, the developed simulator has made it possible to precisely depict the phenomena occurring on a parallel pair under the wide range of running conditions.

2.2 The proposed performance diagram of vibration

If we remain at a level of simulation development requiring lengthy calculation of the vibration phenomena on each parallel pair, we cannot simplify gear technology easier and universal for use by non-specialists. Hence, to go a skip beyond, the performance diagram on vibration is introduced as an useful technical information, which has been made possible by the developed simulator.

Fig. 4 Flowchart of the simulator developed for a helical pair for rotational vibration

For the following discussion, the coordinate of transverse contact ratio ε_a (as an abscissa) and overlap-contact ratio ε_b (as an ordinate) are defined as the contact-ratio-domain. The helical gear pairs occupying a point on the contact-ratio-domain are realized by an infinite number of combinations of their dimensions. It has been revealed that the stiffness of each gear pair at a point, nondimensionalized by its mean values, is given uniquely according to the position on the contact ratio domain. Using the nondimensionalized stiffnesses, and being nondimensionalized by both transmitting load W and equivalent inertia mass M , the acceleration of each pair at the same point is expressed as the same amplitude. The nondimensional accelerations are calculated over the entire contact ratio domain and their levels are expressed as contour lines of equal amplitude on the contact-ratio-domain named the performance-diagram of vibration. When the designer wants a set of design parameters for a low-vibration gear pair, or a gear pair whose vibration is less affected by errors, he

should easily know by a cursory glance how to decide their dimensions with the use of the proposed vibration performance diagram.

PERFORMANCE-DIAGRAM ON A PAIR HAVING EACH KIND OF ERROR—The performance-diagram of vibration is shown in Fig. 6 as an example of the nondimensional speed of $f_z/f_n=0.28$ (tooth-mesh frequency / resonance frequency of the pair) for three common manufacturing cases of a helical gear pair having (a) no error, (b) a convex profile or (c) a concave profile.

The interval of the nondimensional vibration level $\ddot{x}$ of the pair between adjacent contour lines is chosen as 0.005 where the nondimensional vibration level $\ddot{x}$ is expressed by the actual acceleration $\ddot{\delta}$, the transmitting load W and the equivalent inertia mass M as shown in the figure.

When the pair has a convex profile as in Fig. 6(b), the vibration becomes small over the entire contact ratio domain except around the area $\varepsilon_a=1.0$ and 2.0, for $\varepsilon_b<1.0$. That is, it is recognized that the convex profile has the ability to decrease the vibration. On the contrary, when the pair has a concave profile as in Fig. 6(c), the vibration becomes large over the contact ratio domain. That is, the concave profile should be avoided for the vibration problem.

Referencing such performance diagrams, under many conditions, an engineer can, in most cases, easily select the low-vibration gear pair, or the gear pair with only a minor possibility to be affected by some errors.

However, a more advanced stage of technology would incorporate this practical knowledge in a data base of the gear-design program, providing the general engineer access to ideal gear-pair data from only a few input items of the given conditions.

3. The Trend of Manufacturing

The preceding chapter shows a procedure by which vibration of a gear pair can be reduced in the

Fig. 5 Vibration waveforms

Fig. 6 Performance diagram on a pair having each kind of error: (a) No error, (b) Convex profile and (c) Concave profile (contour lines of equal vibration amplitude)

design stage. However, of the many areas requiring further study, the tooth surface profile is most influential in vibration of a gear pair. It is, therefore, the most important factor in decreasing vibrations. The next step is attaining a gear pair having the desired tooth profile; that should be the direction of gear manufacturing technology in the near future.

It has been stated that gear manufacturing machines have some differences from other general machine tools because the work (gear) should rotate synchronously with the rotation of the tools (hob, grinding wheel and others). This synchronous motion is called "profile generation" in gear technology, and is very important to realizing easily an involute profile gear from early age.

Profile generation has been realized by a multi-stage gear-shaft system and long gear train in which the rotational stiffness is not enough to guarantee the generation of a good surface profile. However, computer-aided technology is now making it both very easy and useful to maintain fairly accurate synchronous revolution of the work and the tool, which are set free from the long gear train. Such computer technology is called NC (numerical control). Here, NC gear cutting and grinding machines are introduced and a trend of manufacturing is discussed in relation to a recent hard-gear finishing by use of a CBN (cubic boron nitride) grinding wheel, which will be one of the main current of gear finishing in the near future.

3.1 The NC hobbing machine⁽⁶⁾

The features required for the production-type gear cutting machine are as follows: 1) high productivity, 2) ease of setup, change, and operation, 3) consistent machine precision, and 4) durability and easy maintenance.

For the realization of high productivity, a rigidity of the gear train between the head and column, are required. The NC gear cutting machine achieves this by eliminating the gear train between the hob spindle and table spindle, yielding an easy design of the head and column with a direct-drive method. The result is high-speed cutting of more than 110 m/min.

Ueno⁽⁶⁾ reported that the adaptation of CNC (computer numerical control) resulted not only in the elimination of various types of change gears for an easy setup but also in the following:

- 1) machine precision improvement and high repeatability,
- 2) reduction of the number and types of cutting tools,
- 3) mixed flow production,
- 4) process intensification by the simultaneous use of different hobs, and
- 5) special machining for noncircular gears and

helical racks.

NC technology represents a higher and easier gear production technology than that given by conventional mechanical gear-cutting machines. It is, not only a machining improvement: this technology can store production-process data and use it effectively for future production processes.

It is expected that the adoption of NC for principal machine functions will be incorporated when automated gear-manufacturing systems are developed in the near future.

3.2 Topological modification

Gear designers and production engineers now have capabilities which enable tooth flank modification varying across the facewidth while the longitudinal tooth form varies over the profile, as in the topological-modification-grinding system developed by the Maag Gear-Wheel Company⁽⁸⁾. This system applies a grid of $x-y$ coordinates to the tooth surface, and can specify a different amount of modification for each point of intersection to provide the desired tooth-flank topology (see Fig. 7). The values of the modification, z , as a function of the $x-y$ coordinates can be stored in the microprocessor control system and used for controlling the wheel positions during grinding operations.

This system does not have high productivity itself but is aiding the improvement of the tooth-surface profile by an ideal modification of shaving cutters, in a few automotive companies. The accuracy of the tooth-surface profile of a rolling gear will be improved by freely making an ideal profile of roll dies with the use of this system. It is expected to facilitate high-quality gear mass production. However, before its main stream application, it must be determined whether roll burnishing using ideal profile roll dies is better than CBN grinding.

3.3 Cubic Boron Nitride Grinding Wheel

Shaving and roll-burnishing processes are performed on a gear before heat treatment while the steel is still soft. The distortion which occurs during the heat

Fig. 7 Topological tooth flank modification (MAAG)

Fig. 8 Reishauer's new grinding method

treatment is empirically known to each company, but its estimation and control are actually very difficult.

It has long been known that gear grinding would be a solution to the problem of distortion, if the grinding process could guarantee an availability for mass production of gears in, for example, the automotive industry. Reishauer⁽⁹⁾ has developed a new technique for grinding the tooth flanks: a continuous profile-generating method using a grinding worm with a globoidal-shaped external profile based on the conventional method (see Fig. 8). The grinding worm has the geometry corresponding to the actual tooth profile of the workpiece to be ground which eliminates the axial feed movement of the workpiece during the grinding process. Reishauer reported that the main advantages are a short grinding time and high accuracy. However, it appears that this full CNC grinder uses the usual white arundum grinding wheel requiring frequent dressing.

On the other hand, using a cylindrical grinding worm as a tool having a longitudinal section corresponding to the basic profile of a rack, Kashifuji [10] has developed a full CNC grinding machine where the grinding wheel is made of CBN. CBN creates the potential of realizing high productivity through long life and high stability with respect to temperature. Haga [7] reported that the wear of the CBN grinding wheel is only 2 or 3 μm after grinding over 10,000 gears, and the surfaces ground by the CBN are in better condition than the usual WA grinding wheel. Therefore, this grinder is expected to be applied to a high-quality gear mass production system in the future.

Finally, it is noted that all recent gear grinders are also equipped with NC technology. It is expected that the full advantage of NC technology is not only its easy operation, high accuracy and repeatability,

but also its potential as a prototype for the automated gear production system of the future.

4. Conclusion

The basic idea presented here is that through the application of computer-aided technology to the manufacturing industry, the generally educated engineer should be able to gain easy access to many if not all of the specialized applied sciences. This can be accomplished in the case of gear technology through the assimilation of physical rules, and analysis of various real experiences for gear design into a computerized system. This "expert-system" can then be utilized by the generally educated engineer to achieve the same result as the specialist.

Some may hold the mistaken view that gear technology has reached a stage of maturity in its development. However, the continued demand for lightweight, efficient and quieter machines requires a continued refinement of the development of this applied science. Given the inevitable and continued reduction of specialized engineering technicians in the Japanese workforce, the problem is not limited to gear technology alone, and perhaps can be approached in many different machine production areas with solutions similar to the approach presented in this paper.

References

- (1) Umezawa, K., Sato, T., and Ishikawa, J., Simulation on Rotational Vibration of Spur Gears, *Bull. JSME*, Vol. 27, No. 223 (1974), p. 102.
- (2) Umezawa, K., and Suzuki, T., On the Design of a Low-Vibration Helical Gear for Automobiles, *Proc. 4th Int. Pacific Conf. on Automotive Eng.*, Paper No. 871222(1987-11).
- (3) Umezawa, K., Deflection and Moment Due to a Concentrated Load on a Rack-Shaped Cantilever Plate With Finite Width for Gears, *Bull. JSME*, Vol. 15, No. 79 (1972), p. 116.
- (4) Umezawa, K., The Meshing Test on Helical Gears under Load Transmission (1st Report), *Bull. JSME*, Vol. 15, No. 90 (1972), p. 1632.
- (5) Umezawa, K., and Ishikawa, J., Deflection due to Contact between Gear Teeth with Finite Width, *Bull. JSME*, Vol. 16, No. 97 (1973), p. 1085.
- (6) Ueno, K., CNC Gear Manufacturing Machine, *KIKAI NO KENKYUU (Science of Machine)*, (in Japanese), Vol. 38, No. 7 (1986), p. 789, Yokendo.
- (7) Haga, H., Proceedings of Symposium on New Technology of Power Transmission, (in Japanese), *Jpn. Soc. Auto. Eng.*, (1987-11).
- (8) Maag Company Technical Information ES-422.
- (9) Reishauer Company Catalog, CNC-Gear Grinding Machine RZP.
- (10) Kashifuji Company Catalog, KF 220 CNC.