Mechanism and Machine Theory 82 (2014) 141–153

Contents lists available at ScienceDirect

Mechanism and Machine Theory

journal homepage: www.elsevier.com/locate/mechmt

Pitch cone design and tooth contact analysis of intersected beveloid gears for marine transmission

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ARTICLE INFO

Article history:
Received 14 January 2014
Received in revised form 10 July 2014
Accepted 17 August 2014
Available online 6 September 2014

Keywords:
Beveloid gears
Pitch cone design
Tooth contact analysis
Intersected axes
Misalignments

ABSTRACT

A pitch cone based geometry design method considering line contact is proposed for intersected beveloid gears. The proposed pitch cone design applies the spatial gearing model to determine working pitch cone angles and working spiral angles. For the determination of geometry design parameters, a computer program which incorporates pitch cone design and line contact condition was developed. To illustrate the meshing characteristics, example studies with three groups of geometry parameters were performed by unloaded and loaded tooth contact analysis. Through the analysis, the influences of misalignments, torque load and angle between the first principal directions of the tooth surface curvatures (FPD-angle) on the mesh were investigated. The results show that: compared with axial position errors, shaft angle errors have a more detrimental influence on both the contact path and transmission error. Both FPD-angle and torque load have an obvious effect on this type of gearing. Finally, a practical experimental setup for intersected beveloid gearing with small shaft angle was performed and loaded tooth bearing tests were conducted to demonstrate the proposed design procedure and the theoretical simulation models. The experimental results compared well with the simulation.

1. Introduction

For power transmission between two shafts with large angles (close to 90°), hypoid and worm gears are superior to involute helical gears. However, due to design and manufacturing difficulties, it is not practical to use hypoid and worm gears when the shafts are intersected with small angles (less than 45°) in such compact gear boxes. In such applications, intersected beveloid gears provide a competitive design option because of their compactness, low manufacturing cost and flexibility in gear ratio [1]. A typical application for beveloid gears is the marine transmission with small shaft down angle as shown in Fig. 1. The transmission system consists of an engine, a beveloid gearbox and a propeller. For the beveloid gearbox, the transmission schematic of the beveloid gearbox is shown in Fig. 2. The gearbox is composed of input units, transmission parts and output components, which could operate forward and reverse for the boat. Shaft I is the input shaft, shaft II and III are respectively for running ahead and backward, while shaft IV and V are respectively for transmission and output. Gear 2 and gear 3 are conventional involute helical gears, of which the shafts are parallel; gear 1, 4 and 5 are beveloid gears. Shafts of gears 1 and 5 are intersected and the ones of gears 4 and 5 are crossed. Clutches 6 and 7 are used for running ahead and backward, respectively. For the intersected beveloid gear pair as shown in the figure, however, the main challenge is the fact that the tooth surfaces intrinsically mesh at a point with contact load concentration which leads to a lower load capacity.

In the last couple of decades, some studies have been performed on beveloid gearing theory, geometry design and manufacturing. Mitome [2–4] developed the tooth surface equation according to the velocity relation between generating hob and generated gear during the machining process and analyzed the transmission characteristics of ideal state and error-existing situation; Innocenti [5]

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http://dx.doi.org/10.1016/j.mechmachtheory.2014.08.007
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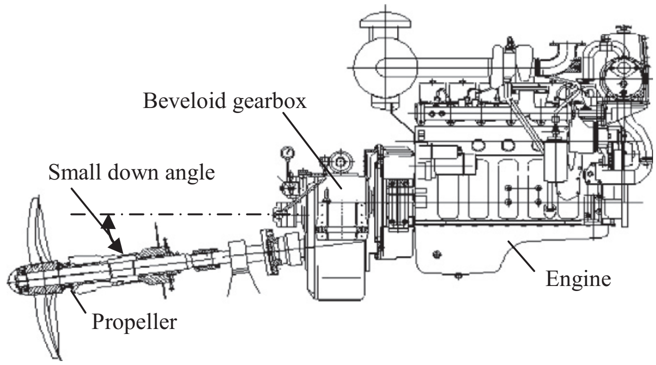

Fig. 1. Marine drive transmission with small down angle.

studied the meshing theory of crossed beveloid gears and calculated the relationship between meshing gap and design parameters; Liu and Tsay [6,7] built the meshing model of beveloid gears with parallel and crossed axis, studied mesh characteristics and developed the calculation condition to avoid undercutting. Mitome [8] proposed a kinematic analysis of taper hobbing on the strength of an imaginary generating rack and presented the generation concept for beveloid gear. Zhu and Song [9] developed the working pitch cone design considering line contact for crossed beveloid gears and built the meshing model to investigate the influence of misalignments on its tooth contact behaviors. In terms of machining principle and manufacturing method, Wu [10,11] introduced the relationship between hobbing processing method and grinding processing method of involute bevel gear. Contrary to the abundance of valuable researches on geometry design and manufacturing for parallel and crossed beveloid gears, the technical information on intersected beveloid gears is hardly available in literature pertaining to geometry design and contact analysis with line contact condition. The purpose of this paper is to present a geometry design method that can meet the line contact requirements and quantitatively analyze the mesh characteristics as well as load sharing factor of intersected beveolid gears.

2. Working pitch cone design

Working pitch cone design [12,13] is used to express the relationship between working pitch cone angle γ_{w1}, γ_{w2} and working spiral angle β_{w1}, β_{w2}. A pitch cone can be represented analytically in the local coordinate system S_i ($i = 1, 2$ where $i = 1$ for pinion and $i = 2$ for gear) as shown in Fig. 3 where θ_i and u_i represent rotational coordinates and generatrix coordinates of pitch cone, respectively. Fig. 4 shows a specific layout for the intersected beveloid gear with three distinct coordinate systems. Coordinate systems S_i ($i = 1, 2$) are attached to the pinion and gear. The global coordinate system S_f is fixed to a reference frame supporting the pinion and gear. The two working cones are tangent at point P. Unit vectors τ_1 and τ_2 denote the generatrices of pitch cones that lie in the pitch plane and intersect each other at the point P. r_{pw1} and r_{pw2} refer to the reference pitch circle radius for pinion and gear. d_1 and d_2 represent the mounting distances, and Σ denotes the shaft angle.

According to Fig. 3, the position vector of point P subject to coordinate system S_i is:

$$R_{wi} = \begin{bmatrix} x_i \\ y_i \\ z_i \end{bmatrix} = \begin{bmatrix} u_i \sin\gamma_{wi} \cos\theta_i \\ u_i \sin\gamma_{wi} \sin\theta_i \\ u_i \cos\gamma_{wi} \end{bmatrix} \quad i = 1, 2. \tag{1}$$

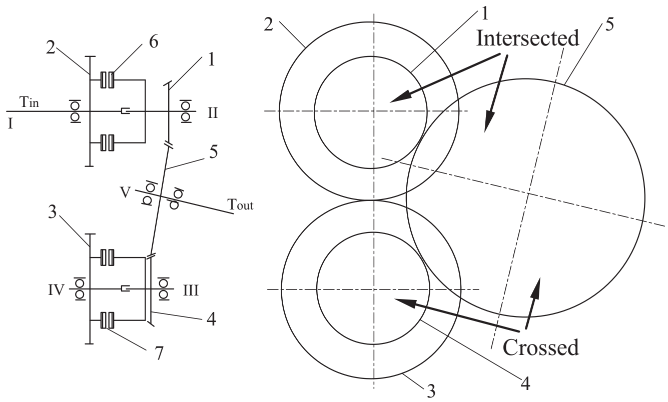

Fig. 2. The transmission schematic of the beveloid gearbox.