Analysing the Effect of Eliminating Belt Drive in Lathe Gear Box by Directly Engaging of Gears

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Abstract— This paper offers a basic understanding of the analyzing the effect of eliminating belt drive in lathe Gear Box by directly engaging of gears. Here we lay a virtual thinking of directly engaging of gears in order to eliminate the power losses due to belt drive. This work has cherished us in realizing the complexity in design and the problems we face during removal of belt drive.

Key words: Engaging of Gears, Belt Drive

I. Introduction

The law of gearing states that for obtaining a constant velocity ratio, at any instant of teeth the common normal at each point of contact should always pass through pitch point (fixed point) situated on the line joining the centers of rotation of the pair of mating gear.A brief overview of the most popular gear types are as follows. Spur Gear Characterized by their straight cogs, these gears are mounted on the parallel shafts. You will find examples of spur gear trains in watches and clock. Bevel Gear Characterized by cogs cut in a cone shape. The gear shafts are generally mounted at 90° angles to each other. Worm Gear comprised of wheel gear with inclined cogs that is rotated by a screw thread. Helical Gear are characterized by cogs that are cut at an angle to the face of the gear can be mounted perpendicularly or parallel. Automobile transmissions make use of these gears.

II. Materials selection

The material used for the manufacture of gears depends upon the strength and service condition like wear, noise etc. the gear may be manufacture from metallic or non metallic material the metallic gears with cut The steel is mainly used for high strength gears and steel may be plain carbon steel or alloy steel. The steel gear is usually heat treated in order to combine properly the toughness and tooth hardness. The materials used to design the gear are described as follows, metallic gear-cast iron, steel. Non metallic gear – raw hide, wood.

III. Literature survey

L.Beireti (2013) proposed that work closed form relationships for the power losses in rubber V-belt CVT are obtained, together with expressions for the axial thrust on both the driving and driven pulleys, starting from simple physical hypotheses. Losses have been grouped into three main contributions: frictional sliding losses, longitudinal and lateral material hysteresis and frictional losses due to the engagement/disengagement of the belt at the entrance and exit from the pulleys. In order to keep the model simple and to obtain a closed form solution, circumferential slip is assumed to take place mainly in the driven pulley and the sliding angle is assumed constant along the sliding portion of the contact arc. The model was validated through comparison with experimental results obtained on a dedicated test bench capable of measuring the transmitted torques and the axial thrusts on both pulleys, the pulleys rotating speeds and the total belt tension. The sliding angle γ and the damping coefficient Ψ, which were introduced in the analytical relationships, were determined by a best fit procedure on the basis of the obtained experimental results. After having obtained those parameters, the model appeared to be capable of reproducing with a fairly satisfactory agreement the power losses, as well as the axial loads on the pulleys, which were obtained in different operating conditions; thus it is proposed as a useful tool for the design of this kind of transmission. Manin (2014) discussed that presents a method that permit to estimate a map of the power losses for any multi-pulley poly-v belt transmission. First, the work identifies and models the power losses in a simple two pulleys poly-v belt transmission, it focuses on the belt losses due to: the rubber hysteretic behavior, and the pulley-belt slip. The model developed is general so that it can be applied to any belt transmission. Finally, the developed model is extended to simulate the power losses of a multi pulley transmission such as a front engine accessory drives.

IV. Designing of Gears

A. *Designs for 48 Teeth*

Number of teeth	=48
Module	=1.6 mm
Outer diameter	= (N+2)xM
Tooth depth	=2.15xM
Tooth thickness	=1.157xM
Outer diameter	= (48+2) x1.6 = 80 mm
Inner diameter	=10 mm
Tooth depth	=2.15x1.6 =3.44 mm
Tooth thickness	=1.157x1.6 =1.85 mm

B. *Designs for 62 Teeth*

Number of teeth	=62
Module	=1.579 mm
Outer diameter	= (N+2)xM
Tooth depth	=2.15xM
Tooth thickness	=1.157xM
Outer diameter	= (62+2) x1.579 = 139 mm
Inner diameter	=9 mm
Tooth depth	=2.15x1.579 =3.394 mm
HRA	=1.157x1.579 =1.82 mm

V. Effects

The direct engage of gears results in reduction of power transmission in machines. The time of machining is also reduced due to the time required reduction in power transmission. The main disadvantages over this process are complexity and feasibility over the designing of machines which can satisfy all demands.

VI. Conclusion

Here we demonstrated a virtual change ofa conventional process. In a dynamically challenging environment we need to retune all the conventional ideas. In this paper we made a real attempt over it.

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