

Research Article

Design of Two Stages Speed Reducer for Scraper Conveyor in CNC Machine

Nyein Aye San ^{#1}, Win Pa Pa Myo ^{*2} and Tin san ^{#3}

^{#1} Department of Mechanical Engineering, Mandalay Technological University

^{*2} Department of Mechanical Engineering, Mandalay Technological University

^{#3} Department of Mechanical Engineering, Technological University (Lashio)

Email¹: nyeinayesanmdy@gmail.com

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Abstract: This paper deals with design of speed reducer for scraper conveyor used in CNC machines. The required power and torque of the conveyor are 0.1159 kW and 102.364 Nm with the revolution of 10.8208 rpm. The drive motor used in this conveyor is the gearmotor, combination of AC electric motor and the reduction gear train. The type of speed reducer used in gearmotor is two stages helical-worm type. The design of helical gear drive and worm gear are calculated in this paper. All gears are also checked from the stand point of wear. As a results, the output power were 0.2 kW and the 0.1419 kW from motor and speed reducer, respectively. The maximum allowable speed and torque from the gearmotor were are 13 rpm and 104.223 Nm.

Keywords: Helical gear, speed reducer, worm gear, wear, maximum allowable speed and torque.

I. Introduction

Computer Numerical Control (CNC) is widely used for the automation of machines used in metal removal, shaping, and fabrication because they can increase the productivity with high accuracy and short cycle time. Conveyor which is used in CNC machine is a material transporting device and it is not made a production process. But, it can support to produce good products with smooth operation in machine. The function of conveyor in CNC machine is to remove metal chip occurring from machining process. There are two type of conveyor used in CNC machines, shaft less spiral conveyor and chain conveyor.

Scraper conveyor is the type of chain conveyor and it removes metal chip by driving scraper band. This type of conveyor is used for carrying the small metal chip under 100 mm size. So it cannot be used in turning machine. Gearmotor is used to operate the scraper band that operates with chain drive. Speed reducer is a transmission device which is reduced higher motor speed to desired lower speed. Among various types of speed reducers, speed reducer which is two stages helical-worm type is used because it can make more smooth operation and higher efficiency than single stage.

II. Theory of Conveyor System

Conveyors are the material transporting device to move the materials from one operation to another. Among various transporting devices, shaft less spiral conveyor and chain conveyor are used as a chip conveyor in CNC machine. According to the shape of the CNC machines, the chip conveyor used in CNC machines are varied which use only chain conveyor and use these combined two conveyors. Furthermore, the usages of chain conveyor is separated into

two types: slat band and scraper, by the size of chip metal produce from machine. These chain conveyors are also divided into three types: straight, straight-rising and straight-rising-straight, by the shape of the conveyor. The scraper conveyor, straight-rising-straight shape is shown in Fig.1 and this type of conveyor can carry small size of metal chips under 100 mm. So it cannot use in turning machine. The components of this conveyor are funnel, chain, scraper band, sprocket, drive shaft and gearmotor [1].

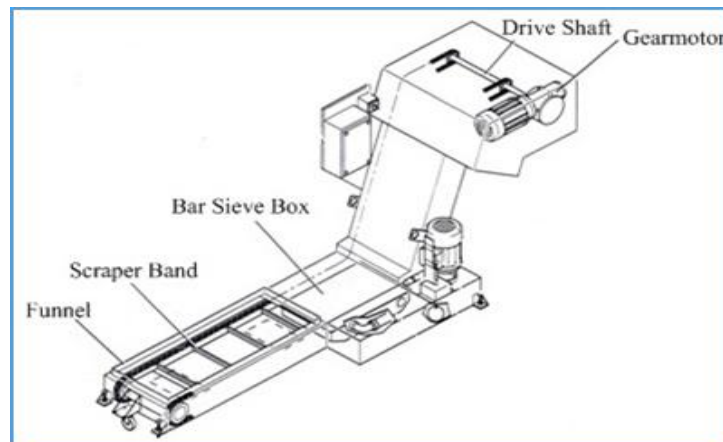


Figure 1. Scraper conveyor

A. Gearmotor

Gearmotor is a complete motive force unit consisting of an electric motor and reduction gear train integrated into one unit. The goal of using a gearmotor is to reduce the rotating shaft speed of a motor. Another goal achievable with a gearmotor is to use a small motor to generate a very large force at a low speed.

The reduction gear trains, used in gearmotors, are designed to reduce the output speed while increasing the torque. Reduction gears consist of a small gear driving a larger gear. There may be several sets of these reduction gear sets in a reduction gear box [6].

B. Gear

Gear is defined as a machine element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth and type of gear are shown in Fig.2. It is a simple device that can change the speed, direction or torque of a motor. They are widely used for transmitting small or large amounts of power from one shaft to another.

Generally, spur gear, helical gear, worm and worm gear are most commonly used in industry for power transmission purpose. Spur gear is compact, efficient, and is available in a parallel shaft arrangement. They are available in 10:1 ratio per gear stage.

The limitations are that spur gears are slightly more expensive, are more likely to produce noise and have less shock capability (compared to worm gears). Helical gears can be used on non-parallel and even perpendicular shafts, and can carry heavier loads than spur gears. They are compact, efficient, and are available in 5:1 ratio per gear stage.

Worm gears are relatively inexpensive and are available in high ratios in single gear set up to 100:1, also available in right angle configurations. They will tolerate high shock loads, and are quiet. However they are less efficient than other forms of gearing [7].

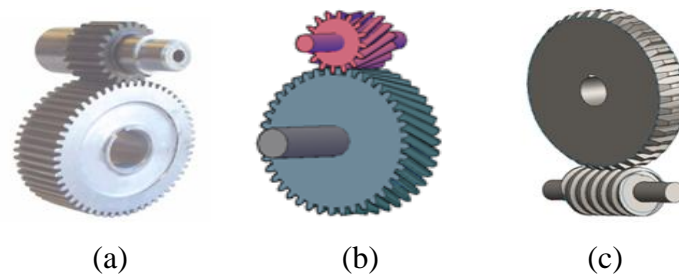


Figure 2. Types of Gear (a) Super gears (b) Helical gear (c) Worm gear

III. Design Procedure

The type of helical-worm type speed reducer is used in this design and it has two stages. The first stage is helical gear and the second stage is worm gearing. The design procedure of this speed reducer is shown in Figure 3.

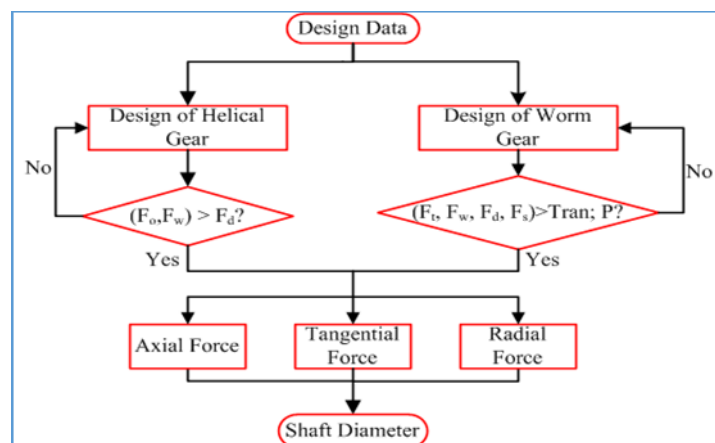


Figure 3. Design Procedure of Speed Reducer

IV. Design of Speed Reducer

The DMF 500 linear machine is the type of CNC milling machine and it cannot be operated turning operation. The type of conveyor used in this machine is the chain type scraper conveyor. The required power and torque of this conveyor are 0.115kw and 102.364Nm with the speed of 10.8208rpm. The motor to be paid input power to speed reducer is AC motor type and the selected motor specifications are

Output power = 0.2kW

Speed = 1300rpm.

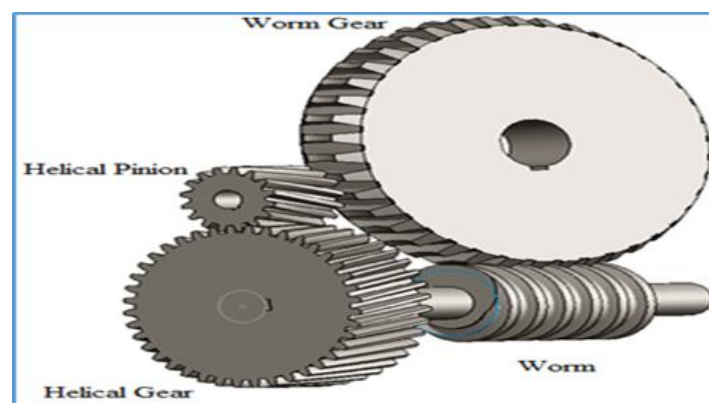


Figure 4. Assembly of Speed Reducer

A. Design of Helical Gear

The helical gear is single helical type and the helix angle of helical gear was 23 degree and the pressure angle was 14.5°. The velocity ratio of this helical gear was 2.5 and the speed of helical pinion is 1300rpm. The efficiency of the helical gear was 98%. The material for both helical gears was selected steel with Brinell Hardness Number 300 and the allowable stress of this material was 100MPa. So, the material for both gears was same and the design of helical gear was based on pinion with unknown diameter case. In the design check for strength, if the pitch diameter was unknown, the following form of the Lewis equation was used,

$$s = \frac{2M_t}{k y \pi^2 N m^3 \cos \psi} \quad (1)$$

The pitch line velocity of pinion was calculated by using diameter and speed of pinion [5].

$$v = \frac{\pi D_p (\text{RPM})_p}{60} \quad (2)$$

The allowable stress was calculated by this equation.

$$s = \frac{s_0}{5.6 + \sqrt{v}} \quad (3)$$

So the value of width of helical gears was the function of module and can be calculated by the following equation.

$$b = k \times \pi \times m \quad (4)$$

B. Checking of the Helical Gear Design

The above procedures based on strength design were considered only as a first approximation in arriving at a possible pitch and face width which were checked for wear load and dynamic load. The limiting endurance beam strength load F_o was based on the Lewis equation without a velocity factor [4].

$$\text{Limiting endurance load, } F_o = s_o \pi m y b \cos \psi \quad (5)$$

The limiting wear load F_w for helical gear may be determined by the Buckingham equation for wear.

$$\text{Limiting wear load, } F_w = \frac{D_p b Q K}{\cos^2 \psi} \quad (6)$$

The dynamic load F_d for helical gear was the sum of the transmitted load and an incremental load due to dynamic effects. F_w was greater than and equal to F_d ($F_w \geq F_d$) and F_o was greater than and equal to F_d ($F_o \geq F_d$).

$$\text{Dynamic load, } F_d = F_t + \frac{21 v_1 (C_b \cos^2 \psi + F_t) \cos \psi}{21 v_1 + \sqrt{C_b \cos^2 \psi + F_t}} \quad (7)$$

The forces acting on helical gear are tangential force, axial force and radial force. These forces was calculated as follows:

$$\text{Tangential force, } F_t = \frac{\text{Torque}}{\text{Pitch radius}} \quad (8)$$

$$\text{Axial force, } F_a = F_t \tan \psi \quad (9)$$

$$\text{Radial force, } F_r = F_t \tan \phi \quad (10)$$

The results of the helical gears design are shown in Table 1.

Table 1. Results of Helical Gear

Name of Parameters	Helical Pinion	Helical Gear
Diameter	32mm	80mm
Number of Teeth	16	40
Module	2mm	2mm
Face width	25mm	25mm
Torque	1.469Nm	3.599Nm
Speed	1300rpm	520rpm

After making the design of helical gear, check this design by endurance force, wear force and dynamic force. The results of these values are shown in Table I. According to the results of checking, the wear force, $F_w = 1354.1980$ N, and the endurance force, $F_0 = 1330.2516$ N, were greater than the dynamic force, $F_d = 1283.3516$ N. So the design was satisfied from the standpoint of wear. The results of forces acting on the helical gear were shown in Table 2.

Table 2. Results of Checking

Parameters	Unit (N)
Endurance Force, F_0	1330.2510
Wear Force, F_w	1354.1980
Dynamic Force, F_d	1283.3510

Table 3. Results of forces acting on helical gear

Parameters	Units (N)
Tangential Force, F_t	91.38125
Radial Force, F_r	23.7443
Axial Force, F_a	38.9720

C. Design of Worm Gearing

Before designing, the selection of material for worm and worm gear was necessary. Therefore, hardness steel for worm and phosphor bronze for worm gear were selected in this design, because they were used for light service. The velocity ratio of the worm gearing was 40 and the speed of worm was 640 rpm. The worm gearing design was based on the worm with unknown diameter case [3].

The lead angle of worm gearing was calculated by this equation.

$$\tan \lambda = (N_G / N_w)^{1/3} \quad (11)$$

The worm gearing design was calculated with unknown diameter case and the center distance was $x = 80$ mm. The diameter of worm was calculated by the following equation.

$$D_w = \frac{(x)^{0.875}}{1.416} \quad (12)$$

The diameter of worm gear, $D_G = 2x - D_w$.

The number of teeth on the worm gear is calculated by this equation.

$$T_G = T_w \times V.R \quad (13)$$

The axial or circular pitch can be calculated from the values of velocity ratio and number of teeth on worm gear.

$$p_a = p_c = \frac{\pi D_G}{T_G} \quad (14)$$

D. Checking of the Worm Gearing

In finding the tooth size and strength, it was safe to assume that the teeth of worm gear were always weaker than the threads of the worm. In worm gearing, two or more teeth were usually in contact, but due to uncertainty of load distribution among them it was assumed that the load was transmitted by one tooth only. According to Lewis equation the tangential tooth load was calculated as follows [4]:

$$W_T = \sigma_o C_v b.m.y . \pi \quad (15)$$

The velocity factor was given as:

$$C_v = \frac{6}{6 + v}$$

The Lewis factor, y was obtained

$$y = 0.124 - \frac{0.684}{T_G}, \text{ for } 14 \frac{1}{2} \text{ involutes teeth.}$$

The dynamic tooth load on the worm gear was calculated by,

$$W_D = \frac{W_T}{C_v} \quad (16)$$

The static tooth load or endurance strength of the tooth was obtained as:

$$W_s = \sigma_e b m y \pi \quad (17)$$

The limiting or maximum load for wear was calculated as:

$$W_w = D_G . b . K_1 \quad (18)$$

The power transmitted from force was the function of the force and the velocity and was calculated as follow:

$$\text{power} = W \times v \quad (19)$$

The efficiency of worm gearing was defined as the ratio of work done by the worm gear to the work done by the worm. In order to find the approximate value of the efficiency, assuming square threads, the following equation was used:

$$\eta_1 = \frac{\tan \lambda (1 - \mu \tan \lambda)}{\tan \lambda + \mu} = \frac{1 - \mu \tan \lambda}{1 + \mu \cot \lambda} = \frac{\tan \lambda}{\tan(\lambda + \phi_1)} \quad (20)$$

$$v_r = \text{Rubbing speed} = \frac{\pi D_w N_w}{\cos \lambda}$$

$$\tan \phi_1 = \mu,$$

For rubbing speeds between 12 and 180 m/min

$$\mu = \frac{0.275}{(V_r)^{0.25}}$$

Transmitted power from worm gear is

$$\text{Power} = \frac{2 \times \pi N_G T_{\text{Gear}}}{60} \quad (21)$$

Therefore, all power from various allowable forces were necessary to be greater than the transmitted power [4].

The results of worm and worm gear were shown in Table 4.

Table 4. Results of worm and worm gear design

No.	Particulars	Results
1.	Normal pressure angle(ϕ)	14 $\frac{1}{2}^\circ$
2.	Diameter of Worm, D_w	34mm
3.	Diameter of Worm Gear, D_G	126mm
4.	Number of Starts or threads on Worm, T_w	1
5.	Number of Teeth on Worm Gear, T_G	40

The efficiency, torque and power of worm gearing were shown in Table V. According to these results, all the powers from various forces were greater than the transmitted power from worm gear. Therefore, the worm gear design was satisfied.

The transmitted power and torque of the worn gearing were 0.1419 kW and 104.223 Nm. And the revolution of the worm gear was 13 rpm.

The forces acting on the worm and worm gear were shown in Table 6.

From the above analysis indicates that the allowable transmitted power and torque from gearmotor is limited 0.1419kW and 104.223Nm with 13rpm revolution.

The required power and torque from scraper conveyor is 0.1159kW, 102.364Nm and 10.8208rpm so this gearmotor design is suitable for the scraper conveyor and can be operated the conveyor safely.

Table 5. Results of Efficiency, Torque and Power

No.	Parameters	Results	Units
1.	Efficiency	72.39	%
2.	Torque on Worm	3.599	Nm
3.	Torque on Worm gear	104.223	Nm
4.	Transmitted Power	0.1419	kW
5.	Power from Tangential Force	0.1879	kW
6.	Power from Dynamic Force	0.1906	kW
7.	Power from Static Force	0.3812	kW
8.	Power from Wear Force	0.1858	kW

Table 6. Forces acting on worm and worm gear

No.	Parameters	Results	Unit
1.	Tangential Force on Worm, W_{TW} , or Axial Force on Worm, W_{AG}	211.7059	N
2.	Axial Force on Worm, W_{AW} , Tangential Force on Worm, W_{TG}	1654.3333	N
3.	Radial Force on Worm, W_{RW} ,or Radial Force on Worm, W_{RG}	427.8397	N

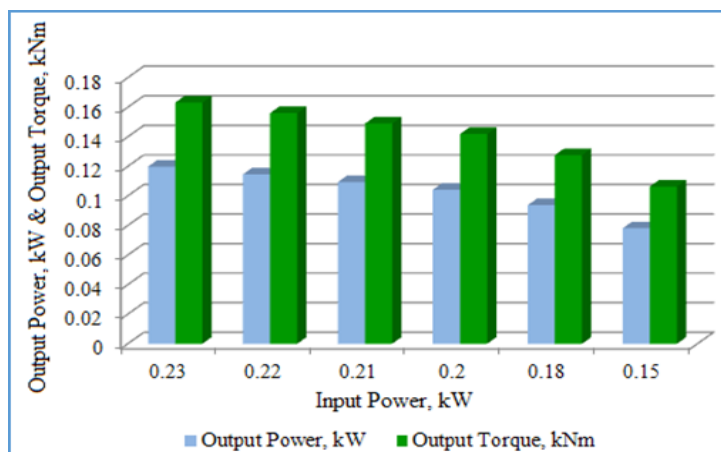


Figure 7. The Variation of Output Torque and Power of Speed Reducer with the Variation of Input Power

By consideration of the input speed of the speed reducer was constant and the input power of the speed reducer was varied, the output torque and power of speed reducer was varied. The variations of output torque and power of speed reducer with the variation of input power are shown in Fig. 4. According to this figure, the output torque and power of the speed reducer were increased by the increasing of the input power. And the reverse process was happened by decreasing of the input power of the speed reducer. From this result, the input power of 0.2kW was most suitable for the considered conveyor. Because the require power (0.115kW) and torque (102.364Nm) of the scraper conveyor were less than output power (0.1419 kW) and torque (104.223 Nm) of this input power.

V. Conclusion

The design speed reducer, two stages, supported the smooth operation for the conveyor and higher efficiency than single stage. In the speed reducer design, design of helical were satisfied from the stand point of wear and strength because of the allowable forces of this gears were greater than the transmitted force. And also the allowable power of worm gearing was greater than the transmitted power. Then, the output power and torque from the speed reducer was greater than the required power and torque of the scraper conveyor. So it was safe in operation for scraper conveyor. The variation of output power and torque of the speed reducer were also considered in the paper by changing input power with constant speed. According to this analysis the design input power (0.2kW) was the optimal condition for the desired conveyor, because it produced sufficient output power and torque for scraper conveyor.

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