Gearbox Design For A Cnc Lathe

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Abstract--The main objective the development of numerical technology has been reductions of production cost by reducing the development time. CNC machines are most housefuls nowadays. The maintenance of CNC headstock is very difficult due to high precision and accuracy. Hence our project "DESIGN OF GEAR BOX" for CNC lathe is to reduce vibration and difficulties in maintenance. This design helps to save the time and there by the cost of maintenance. The function of the gear box is to transfer the power effectively to the head stock from the motor. We are provided with the certain specification of the machine. By knowing this data, we are successful in creating the design by selecting the best materials and optimum size. Here we have designed the V-Belt, Shaft, Selection of bearings, Spline and Gears.

Keywords-- V-Belt, Shaft, Bearings, Spline, Gears.

INTRODUCTION

With the advancement in technology relative, motion between tool and work piece can be controlled automatically. Electronic Industries Association defines numerical control as a “System in which actions are controlled by direct insertion of numerical data at some point. The system must automatically interrupt at least some portion of this data”. CNC (Computer Numerical Control) may be defined as an NC system in which dedicated stored program computer is used to perform some or all the basic NC functions in accordance with control programs stored in the read/write memory of the compute.

The SBCNC Tuning Centre is designed to perform a variety of machining operations such as straight and taper turning drilling boring contouring with linear and circular interpolation, Internal and external threading (parallel or tape) etc. The machine is suitable for chuck, shaft and bar type of work pieces. To solve any problem in engineering there should be a pre planning. This planning may be called designing. It may be concerned with the selection or designing of physical components for a physical structure or to find an optimum solution for an existing problem.

This machine design can be considered as the scientific art of efficient dealing with materials and forces to achieve economic design assuring most advantageous combination of accuracy safety, durability, speed simplicity, and efficiency. The design of three step variable speed gear box for SB CNC lathe is mainly based on considerations like compactness, longer service life with minimum breakdown periods.

DESIGN MORPHOLOGY

MACHINE DESCRIPTION

The SBCNC Tuning Centre is designed to perform a variety of machining operations such as straight and taper turning drilling boring contouring with linear and circular interpolation, Internal and external threading (parallel or tape) etc. The machine is suitable for chuck, shaft and bar type of work pieces.

The basic structure of the machine is computer optimized, with a large uninterrupted torque-tube for maximum...
rigidity and stability. The bed guide ways for saddle and the saddle guide ways for cross slide are hardened and ground. Anti-friction liners (ticulty) are fixed to the saddle guide ways mating with the bed as well as to the cross slide guide ways mating with the saddle.

The headstock contains the main spindle bearing system and the drive shafts system. The spindle is powered by a 22.57.5KW variable-speed motor through solenoid actuated. Hydraulically shifting, 3-speed gear in the headstock thus providing an overall constant power speed range of 128. The turret is a 12-station, disc-direct, suitable for accommodating VDI 3425 60nm-diameter shank tool-holders. The turret actuation is hydraulic. The tailstock is mounted on separate hardened and ground bed guide ways to avoid interference with the saddle. The tailstock body is clamped rigidly to the bed using hydraulic pressure and be positioned along with the bed either manually or automatically through programming. Quill movement can be obtained either manually or through programming. Quill thrust can be varied by adjusting the oil pressure.

Movements both saddle and cross slide are effected through precision ball screws driven by servo-motors. The spindle and tailstock bearings are oil-lubricated from the hydraulic power pack. The slides, the ball screw-nut and end bearings and the worm wheel of the turret are oil lubricated by the centralized, automatic pulse lubrication system. The chip conveyor is mounted in the front of the machine for continuous and uninterrupted removal of chips. All electrical cabinets allied equipments and the hydraulic power pack are kept at the rear of the machine. The machine is controlled by a micro-processor based CNC system. This system accepts part Program from a paper lape reader, PC or through Manual Data Input using keyboard.

The operator control station is located at the front of the machine near the face of the headstock, for operation convenience. The position and velocity feedback devices and the limit/proximity switches indicate the machine status continuously on the CRT monitor at the CNC operator panel.

**MACHINE DESIGN**

To solve any problem in engineering there should be a pre planning. This planning may be called designing. It may be concerned with the selection or designing of physical components for a physical structure or to find an optimum solution for an existing problem. This machine design can be considered as the scientific art of efficient dealing with materials and forces to achieve economic design assuring most advantageous combination of accuracy safety, durability, speed simplicity, and efficiency.

**A. Gearbox**

Gear boxes provide for a wide range of cutting speeds and torque from a constant speed power input enabling proper cutting speeds or torque to be obtained at the spindles as required in the case of cutting drives and desired feed rates in the case of feed drives. Gear boxes are also used to affect interrelated motion between the work piece and the tool as in the case of screw cutting and gear cutting.

With a constant speed power source there is a need for some method of varying the speed over this range. Step less mechanical and electrical drives can provide infinitely variable speed variation. However, the torque or speed characteristics of available steeples drives do not meet the requirements of spindle drives which demand an increased driving torque to the spindle at lower output speed in order to maintain a constant rate of metal removal.

The steeples drives which do possess the torque or speed characteristics are limited by the speed range over which these characteristics can be maintained. In order to provide for a wide range of operating speed together with adequate torque at lower spindle speed range to be covered in a number of discrete steps.

Gear boxes with large number of steps within a given range would be bulky and expensive. Hence they should be so designed that while fulfilling the functional requirement, they are also economical to manufacture.

The cost of a gearbox is related to the number of shafts and bearings required and the total number and the size of the gears. From possible arrangement of gears, the layout, which promotes compact size and lower cost, while still fulfilling the technical requirement of the system should be chosen.
B. V-Belt Design

The V-belt is being extensively used as the driving element in different types of equipments. It has replaced the conventional flat belts drives in many fields of application because of its inherent and many sided advantages. It provides a compact, quite and resilient type of power transmission. Its advantage over chain and gear drive is flexible, elasticity and simplicity. As against flat belt drive and rope drive, it requires considerably less space. It is versatile in application, requires little maintenance, is cheap and has a long life. Because of it’s practically slip free power transmission, its efficiency is nearly 100%. It also has impact vibration and damping. Due to wedge action, the required pretention and consequently also the leads on shafts and bearings are about 60% smaller than those in the case of flat belts drive under similar conditions. The greater force transmitting capacity of the V-belts is explained as follows.

when a compressive force \( F_A \) acts at right angles on a flat belt, the relevant frictional force up to which the circumferential force can develop given by

\[
F_r = \mu F_A
\]

Where,

\( \mu \) = coefficient of friction between the belt and pulley.

Because of the wedge shape of the V-belt, a gripping force greater than in case of a flat belt is developed due to the following reasons.

Force \( F_A \) is resolved into two lateral forces \( F_x \) which is greater than \( F_A \) and is effective gripping forces on each side of the groove, this side force is given by,

\[
F_x = \frac{F_A}{2 \sin \frac{\alpha}{2}}
\]

For belt drives which drive without slip timing belts should used. V-Belts on higher power duties generally have to be matched to ensure the drive power is shared.

For a flat belt drive the tangential friction force at the point of slipping is \( \mu R \). (\( \mu \) = coefficient of static friction- \( R \) = radial force between the belt and the pulley). For a vee belt drive the equivalent friction force at the point of slipping = \( \mu R / \sin \beta \). (\( \beta \) is the vee half angle). Therefore a vee drive has a maximum friction = \( 1 / \sin \beta \) x the flat belt friction.

A v belt drive system, when correctly specified, can be expected to deliver 25000 hours of service under continuous, or 5 years normal use) before belt replacement is required.

V Belt Drives achieve drive efficiencies of about 95%.

Vee belts are generally manufactured from a core of high tensile cord in a synthetic rubber matrix closed in a fabric reinforce rubber lining. The vee belt sections and pulley groove dimensions are in accordance with British/European standards.

1) Motor to gear box

Nominal pitch length \( (L) = 2202 \) mm

2) Modification Factors (From PSG Data book)

Length correction factor \( (F_c) = 1.05 \)
Correction factor for arc of contact \( (F_d) = 0.96 \)
Service factor \( (F_a) = 1.3 \)
Number of belts required \( (N_b) = 8 \) belts

3) Gear box to spindle

Nearest Nominal pitch length \( (L) = 2710 \) mm

4) Modification Factors (From PSG Data book)

Length correction factor \( (F_c) = 1.04 \)
Correction factor for arc of contact \( (F_d) = 176.66^\circ \)
Correction factor, \( F_d \) (Arc of contact) \( = 0.99 \)
Service factor \( (F_a) = 1.3 \)
Number of belts required \( (N_b) = 11 \) belts

C. BEARINGS DESIGN

A bearing is a device to allow constrained relative motion between two or typically linear movement. Bearings may be more parts, rotation or classified broadly according to the motions they allow and according to their principle of operation as well as by the directions of applied loads they can handle.
Plain bearings use surfaces in rubbing contact, often with a lubricant such as oil or graphite. A plain bearing may or may not be a discrete device. It may be nothing more than the bearing surface of a hole with a shaft passing through it, or of a planar surface that bears another (in these cases not discrete device or it may be a layer of bearing metal either fused to the substrate (semi-discrete) or in the form of a separable sleeve (discrete) With suitable lubrication, plain bearings often give entirely acceptable accuracy, life, and friction at minimal cost. Therefore, they are very widely used.

However, there are many applications where a more suitable bearing can improve efficiency, accuracy, service intervals, reliability, and speed of operation, size, weight, and costs of purchasing and operating machinery.

Thus, there are many types of bearings, with varying shape, material, Lubrication, principal of operation, and so on. For example, rolling-element bearings use spheres or drums rolling between the pans to reduce friction: reduced friction allows tighter tolerances and thus higher precision than a plain bearing and reduced wear extends the time over which the machine stays accurate. Plain bearings are commonly made of varying types of metal or plastic depending on the load, how corrosive or dirty is the environment, and so on. In addition, bearing friction and life may be altered dramatically by the type and application of lubricants. For example a lubricant may improve bearing friction and life. But for food processing a bearing may be lubricated by an inferior food-safe lubricant to avoid food contamination: in other situations a bearing may be run without lubricant because continuous lubrication is not feasible and lubricants attract dirt that damages the bearings.

1) **Life of bearing**

\[ L_h = \frac{106 \times L}{60} \]

Where, \( L = \frac{C}{F_m} \)

\( \Sigma = 3 \) for ball bearings

\( \Sigma = 3.33 \) for roller bearings

For general purpose machine tools \( L_h \) should be about 24,000 hrs , considering vibration of load and speed.

2) **Input shaft**

Drive motor power \( P = 37 \text{ KW} \)

Radius of motor pulley \( R_m = 70 \text{ mm} \)

Radius of gear box input pulley, \( R = 163 \text{ mm} \)

Torque on input shaft \( F = 3365.64 \text{ N} \)

Tangential force acting on \( Z=45 \text{ teeth gear} \) \( F_t = 6095.6 \text{ N} \)

Similarly force acting on \( Z=60 \text{ teeth gear} \) \( F_t = 4571.66 \text{ N} \)

Similarly force acting on \( Z=22 \text{ teeth gear} \) \( F_t = 11082.83 \text{ N} \)

When 45 teeth gear engaged

\( R_B = 987.24 \text{ N} \)

\( R_A = 8474 \text{ N} \)

When 60 teeth gear engaged

\( R_B = 1444.42 \text{ N} \)

\( R_A = 6492.88 \text{ N} \)

When 22 teeth gear engaged

\( R_B = 8301.92 \text{ N} \)

\( R_A = 6146.55 \text{ N} \)

3) **Output shaft**

Rpm of the output shaft \( = 244 \text{ rpm} \)

Torque on output shaft \( T = 1.45 \text{ KN-m} \)

Force acting on \( Z=45 \text{ teeth gear} \)

Module, \( M = 4 \)

Pitch circle diameter, \( D = Z \times M \)

\( = 45 \times 4 \)

\( = 180 \text{ mm} \)

Tangential force, \( F_t = 16111.11 \text{ N} \)

force acting on \( Z=30 \text{ teeth gear} \) \( = 24166.66 \text{ N} \)

force acting on \( Z=58 \text{ teeth gear} \) \( = 11111.11 \text{ N} \)

Force acting on output pulley \( = 12888.88 \text{ N} \)

When 45 teeth gear engaged

\( R_B = 28999.655 \text{ N} \)

\( R_A = 7733.33 \text{ N} \)

When 30 teeth gear engaged

\( R_B = 28516.652 \text{ N} \)
When 58 teeth gear engaged 

\[ R_A = 8538.89 \text{ N} \]

\[ R_B = 25599.99 \text{ N} \]

\[ R_A = 1600 \text{ N} \]

Selection of bearings

Input shaft (input side)

Cylindrical roller bearing, Single row \{55/140×33\}

Dynamic load rating, \( c \) = 180KN (manufactures catalogue)

Static load rating \( c_o \) = 140KN

Axial load, \( F_a \) = 0

Radial load, \( F_r \) = 8474 N (higher value of reaction at input side)

Equivalent dynamic load, \( F_m = F_a + F_r \) = 8474 N

Life of bearing, \( L_h = \frac{L \times 10^6}{Nm \times 60} \)

Where \( L \) = life in millions of revolutions

Life in hours, \( L_h = 148858.16 \) hours

Input shaft (outside)

Cylindrical roller bearing, Single row \{55/100×65\}

Dynamic load rating, \( c \) = 136 KN (manufactures catalogue)

Axial load, \( F_a \) = 0

Radial load, \( F_r \) = 8301.92 N (higher value of reaction at input side)

Equivalent dynamic load, \( F_m = F_a + F_r \) = 8301.92 N

Life of bearing, \( L_h = \frac{L \times 10^6}{Nm \times 60} \)

Life in millions of revolutions,

\[ L = \left[ \frac{c}{F_m} \right]^2 \]

For general purpose machine tool \( L_h \) should be about 24000 hours considering variation of load and speed.

(Reference: HMT design calculation manual)

Output shaft (output side)

Single row cylindrical roller bearing, \{70/150×51\}

Dynamic load rating, \( c \) = 204 KN (manufactures catalogue)

Axial load, \( F_a \) = 0

Radial load, \( F_r \) = \( \frac{28516.65 + 1600}{2} \) = 15058.305 N

Equivalent dynamic load, \( F_m = F_a + F_r \) = 15058.305 N

Life of bearing, \( L_h = \frac{L \times 10^6}{Nm \times 60} \)

Life in millions of revolutions,

\[ L = 70201.55 \text{ hours} \]

D.GEARS

A gear or cogwheel is a rotating machine part having cut teeth, or cogs, which mesh with another toothed part to transmit torque, in most cases with teeth on the one gear being
of identical shape, and often also with that shape on the other gear. Two or more gears working in tandem are called a transmission and can produce a mechanical advantage through a gear ratio and thus may be considered a simple machine. Geared devices can change the speed, torque, and direction of a power source. The most common situation is for a gear to mesh with another gear; however, a gear can also mesh with a non-rotating toothed part, called a rack, thereby producing translation instead of rotation.

The gears in a transmission are analogous to the wheels in a crossed belt pulley system. An advantage of gears is that the teeth of a gear prevent slippage.

When two gears mesh, and one gear is bigger than the other (even though the size of the teeth must match), a mechanical advantage is produced, with the rotational speeds and the torques of the two gears differing in an inverse relationship.

1) **Gear calculation**

   Power, \( P = 37 \text{ KW} \)  
   = 50.27 HP  
   Shaft speed, \( N_p = 644 \text{ rpm} \)  
   \( T_p = 55905.6 \text{ Kgmm} \)

2) **Calculation of module**

8.4.1 For Gear \( z = 45 \) teeth

   Module, \( M \geq 1.26 \left( \frac{1}{3} \right) \sqrt{\frac{|M_t|}{y [\sigma] m z_t}} \)

   \[ M = 3 \text{(Nearest standard size)} \]

   To avoid correction in gears the module is taken as 4 for both the pair gears.

   For Gear \( z = 30 \) teeth

   \( M = 3 \text{(Nearest standard size)} \)

   To avoid correction in gears the module is taken as 4 for both the pair gears.

   For Gear \( z = 22 \) teeth

   \( M = 4 \text{(Nearest standard size)} \)

   To avoid correction in gears the module is taken 4.5 for both the pair gears.

   **E. SPLINE CALCULATION**

   1) **Output shaft**

   Torque, \( T = 147545.082 \text{ Kgf-mm} \)

   Mean Radius, \( R_m \)  
   = \( (D+d)/4 \)  
   = 38.5 mm  
   \( F = 147545.08 / 38.5 \)  
   = 3832.33 Kgf  
   \( A = K \times n \times h \times l \)  
   \( K = 0.5 \)

   Total Face width of the spline engaged

   \( L = 165 \text{ mm} \)

   \( 5 = (D-d)/2-g-f \)  
   \( h=(82-72)/2 \)  
   = 4  
   \( A = K \times n \times h \times l \)  

   \( = 0.5 \times 6 \times 4 \times 165 \)  
   = 1980 mm\(^3\)

   Bearing Pressure \( F/A \)  
   = 3832.33/1980  
   = 1.93 Kgf/mm\(^2\)

   From Table 4

   For Shaft and hub both are C2R Toughened, the allowable bearing pressure

   \( P_a = 6 \text{ Kg / mm}^2 \)

   Therefore \( P < P_a \), the selected spline is safe.

   Here 1.93 Kgf / mm\(^2\) \( > 6 \text{ Kg / mm}^2 \)

   Therefore the selected size of spline is safe.

   2) **Input shaft**

   Torque, \( T = 55902.17 \text{ Kgf-mm} \)

   Mean Radius, \( R_m = (D+d)/4 \)
   = 30.25 mm
F  = 55902.17 / 30.25  
  = 1848.005 Kgf 

A = \( K \times n \times h \times l \) 

K  = 0.5 

n  = 6  

(From Design Data Book Pg.No.8.14)  

Bearing Pressure , \( F/A \) = 1848.005 / 1732.5 
  = 1.06 Kgf/mm\(^2\)  

From Table 2, bearing pressure  
\( P_a \) for \( C_2M = 6.61 \) Kgf / mm\(^2\), \( P < P_a \) so design is safe. 

**F. SHAFT DESIGN**  

1) **Calculation of input shaft for torsion**  

Torque,  
\( T = 55902.17 \) Kgf-mm  

\( T = 55902.17 \) Kgf-mm  

\( D = 35.05 \) mm  

Shaft under combined bending and torsion  

When 45 teeth gear is engaged  

Maximum bending moment \( M \) = 33656.4  

Maximum bending stress allowable, \( \sigma_{ba} = 6.95 \) Kgf/ mm\(^2\)  

(Tables 2 for C2M Toughned)  

\[ \sigma_b = \sqrt{\frac{M^2 + \pi^2}{Z}} \]  

\[ Z = \pi d^3 / 32 \]  

\[ Z = 92062.83 / 6.95 \]  

\[ = 13246.45 \]  

\( d = 51.29 \) mm  

When 60 teeth gear is engaged \( d = 45.18 \) mm  

When 22 teeth gear is engaged \( d = 49.07 \) mm  

Torsional stiffness of input shaft \( d = 63.56 \) mm  

The minimum diameter of the spline shaft is selected in 65 mm. So the design is safe. 

**HYDRAULIC SYSTEM**  

A hydraulic system may be defined as a system of power transmission in which an incompressible fluid is used as a power transmitting medium. The primary function of hydraulic system is the transmission of energy from one location to another. The energy level of the hydraulic system determines its use as a control or power application. It is an essential requirement that the approach be as to provide an economical proposition by the utilization of minimum number of components and conservation of energy, without however affecting the reliability of operation.  

Hydraulic power pack generally implies a source of supply of pressurized fluid in a condition acceptable to the drive and control unit. It consists essentially of a reservoir, a pump and its drive motor, a strainer at the pump inlet, a relief valve, a pressure gauge with shutoff and filters for the system fluid and manifold assembly. The pump is either mounted within the reservoir or on the top cover. Both methods are widely practised. 

**CONCLUSION**  

The design of three step variable speed box for SB CNC lathe as per the requirement was done successfully. This design is mainly based on considerations like compactness, longer service life with minimum breakdown periods. Here instead of spur gears, we can use helical gears, which would render smooth and silent transmission. But these calls for good thrust bearings and wear may be greater. Thus this adds up cost and reduces service life. The economic viability of design can be greatly increased while choosing factor of safety for each component. However on choosing an optimum factor of safety, demands sufficient industrial experience on the part of design engineer. With developments in material science and manufacturing technology, few materials with greater design stresses and reduced stress concentration and precision machined will be available. This will result in a more compact, safe and more economical design. 

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