STATIC STIFFNESS ANALYSIS OF HIGH FREQUENCY MILLING SPINDLE

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Abstract

Compared with conventional spindles, motorized spindles are equipped with a built-in-motor, so that power transmission devices such as gears and belts are eliminated. This design also reduces vibrations and achieves high rotational balance, and enables precise control of rotational accelerations and decelerations. The objective of this work is to optimize the parameters influencing the stiffness of the high frequency milling spindle running at 12000 rpm with power rating of 10 KW. Theoretical analysis has been carried out to evaluate the spindle stiffness and to minimize deflection at the nose by varying the following parameters: (i) Bearing arrangement, (ii) Overhang of spindle nose from the front bearing, (iii) Spindle diameter between the bearings. The static analysis was also carried out using ANSYS and there is a good correlation between the results obtained by theoretical approach and ANSYS.

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1. PREAMBLE

1.1 Introduction

The spindle is the main mechanical component in machining centers. The spindle shaft rotates at different speeds and holds a cutter, which machines a material attached to the machine tool table. The static and dynamic stiffness of the spindle directly affect the machining productivity and finish quality of the work pieces. The structural properties of the spindle depend on the dimensions of the shaft, motor, tool holder, bearings and the design configuration of the overall spindle assembly. The bearing arrangements are determined by the operation type and the required cutting force and life of bearings.

1.2 High Speed Spindles

Spindles are rotating drive shafts that serve as axes for cutting tools or to hold cutting instruments in machine tools. Spindles are essential in machine tools and in manufacturing because they are used to make both parts and the tools that make parts, which in turn strongly influence production rates and parts quality.

High speed spindles have emerged today as the most important component of any kind of high machining process. For all kind of machining tasks, whether in the CNC, tool machining center and other process components, the use of high speed spindles always optimizes productivity. To achieve high speed rotation, motorized spindles have been developed. This type of spindle is equipped with a built-in-motor as an integrated part of spindle shaft, eliminating the need for the conventional power transmission devices such as gears and belts. This design reduces vibrations, achieves high rotational balance and enables precise control of rotational accelerations and decelerations. However, the high speed of rotation and the built-in-motor also introduce large amounts of heat and rotating mass into system, requiring precisely regulated cooling, lubrication and balancing.

1.3 Motorized Milling Spindles

Motorized milling spindles are widely used today in all mill centers for heavy duty milling. There are some specially designed motorized spindles, solely used for hard core milling while there are other motorized spindles used in all types of machine tool applications including milling, drilling etc. The use of motor-driven milling spindles is convenient compared to the gear driven or belt driven spindles in most cases and they are highly recommended for maximum milling performance and flexibility. The higher maximum speeds offered by the motor ensure optimum and economical machining from the small to the large work piece. They are suitable for both rough cutting and precise finish cutting.

1.3.1 Features of Motorized Milling Spindles

- A basic feature of this type of spindle is that it can help in reducing the cycle time
- Consider especially for complete machining in a single clamping setup.
- This type of spindle is also available with multifunctional unit for turning, drilling and milling.
- The spindle also has connection with automatic tool change providing functionality and flexibility.
- The motorized spindles ensure machining of a wide range of workpieces without changeovers.
- Integrated motor design eliminates the problems of gears and pulleys.

- The rotary motion in such spindles is accomplished by means of a torque motor as direct drive.
- Higher speed milling.
- Higher overall rigidity.

2. HIGH FREQUENCY MILLING SPINDLE

High frequency driven milling spindles are a very popular category of spindles and are highly recommended for high speed milling and engraving applications. With their ability to deliver high power and high speed, these milling spindles are being increasingly used in high-metal-removal-rate machining.

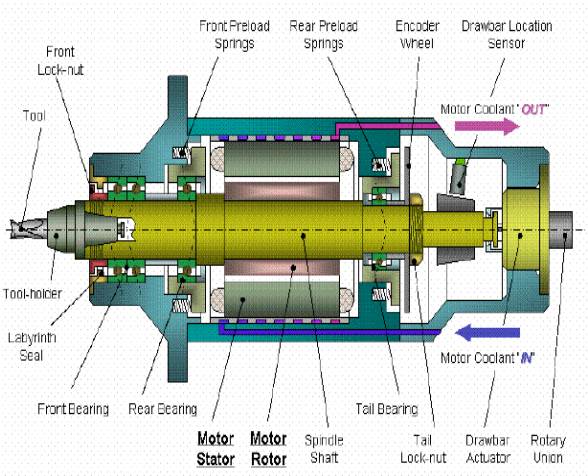

Fig: 2.1 High frequency Milling Spindle arrangements [7]

2.1 Features of High-Frequency Milling Spindles

Certain common features available in high frequency milling spindles are as follows:

- These spindles have high torque value.
- They have high power values.
- Rotational speed is very high.
- Very high axial stiffness values.
- High radial stiffness values.
- Metal removal rates are higher than other spindles.
- Usually compact design.
- Most have angular position and speed control sensors.
- Provision for Automatic Tool Change systems.
- Temperature sensors are available.

During high speed operation, the shaft will exhibit bending characteristics. The frequency at which the shaft will bend depends on the diameter and length of the spindle shaft. It is often tempting to design a very long spindle shaft, as this increases the load carrying capacity of the spindle and allows for a more powerful motor.

2.2.1 Spindle Bearing Selection

The very short length of the spindle shaft means that an unconventional bearing arrangement can be used. In our spindle shaft is held by a single assembly (of three bearings) that are mounted at the front end of the shaft (between the nose and motor) and a set of two bearings are mounted at the rear end. The selection criterions for spindle bearings are

- Speed
- Torque
- Spindle nose size / tool size / power
- Axial and radial stiffness
- Lubrication
- Preload
- Life.

2.2.2 Bearing Speeds

Once the operating speed of the spindle has been determined bearings are sized and selected. Then it is decided to run the bearings in grease rather that oil because this eliminates need for oil feed/filtering equipment. To achieve a spindle speed of 12000 rpm we are looking for bearings with a maximum speed more than that of the spindle speed between 14000 rpm and 16000 rpm.

2.2.3 Bearing Arrangement

Often, if a bearing is subjected to large loads or if a high degree of rigidity is required then two or more bearings are used. The bearing arrangements can be combined from universally matched bearings or produced at sufficient lot sizes.

The different bearing arrangements are

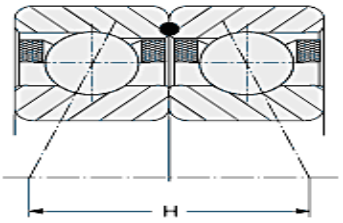

Fig: 2.4 Back-to-Back arrangement(DB)

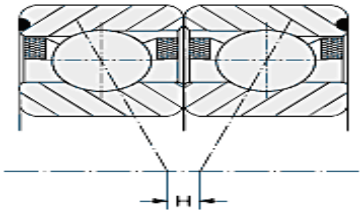

Fig: 2.5 Face-to-Face arrangement(DF)

2.2.4 Bearing Lubrication

The correct choice of lubricant and method of lubrication is as important for the proper operation of the bearing as the selection of the bearing and the design of the associated components.

Grease lubrication should be used if the maintenance-free operation over long periods of time is desired, the maximum speed of the bearings does not exceed the speed factor of the grease.

Oil lubrication should be used if the high speeds do not permit the use of grease and the lubricant must simultaneously serve to cool the bearing.

Hence grease is most preferable over oil lubrication at higher speeds.

2.3 Spindle Cooling

Both air cooled and water cooled spindle options are available. An air cooled spindle would offer the advantage of requiring a water source and pumping equipment. However, given the limited time allotted for the development of the spindle, we choose the water cooled design because it is much more robust in controlling the temperature of the housing.

3. STATIC STIFFNESS ANALYSIS

3.1 Introduction

Static stiffness is one of the important performances of machine tools. Hence it must be correctly defined and related parameters of spindle must be properly determined. Since power loss will be there in terms of mechanical efficiency, but as compared to conventional belt drive system where around 20-30%, here it has been reduced to 10-15%. Hence the available power at the spindle is assumed to be around 8.5 to 9 KW.

Stiffness of the spindle is defined as its ability to resist deflection under the action of cutting force. As shown in fig.3.2 when a force exerted on the spindle nose is F_z, the displacement at the spindle-nose in the same direction as that of F_z is '**δ**' and correlation stiffness is defined as $\mathbf{K = F_z/\delta}$.

3.2 Magnitude and Direction of Forces

In Milling, the cutting conditions are somewhat different when compared with turning as the rotating cutter has a number of cutting edges, which engage with the workpiece only in a part of its rotary path, the remainder being through air.

As shown in fig.3.1, the milling force acting in the cutting zone can be resolved into three components for face milling. The tangential component F_z, radial component F_y, axial component F_x. It has been found that the components F_x and F_y increase more rapidly than the tangential cutting force F_z, where the rake angle is changed from positive to negative value. Using carbide cutting tool, cutting speed to cut the stainless steel material is 120 m/min.[9]

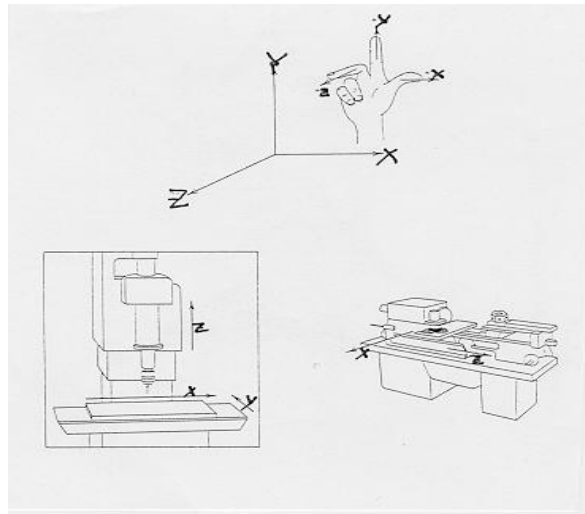

Fig: 3.1 milling force components

The tangential force F_z can be calculated from the unit power concept referring to the following equations:

$$\mathbf{F_z = N_m/v}$$

where, N_m=power of the motor at the spindle
=9 KW=$9x10^3$ W
v=cutting speed =120 m/min=2 m/sec

Therefore, F_z=$9x10^3/2$ **=4500 N**

3.3 Features of High Frequency Milling Spindle to be designed

The designed spindle operates at 12000 rpm with maximum power rating 10 KW. 25^0 angular contact ball bearings are used for mounting the spindle. In analysis of the stiffness of the spindle the outer diameter of the spindle is nothing but the bearing diameter of the inner bearing race. In general, high speed spindles that utilize grease lubrication do not allow for replacement of the grease between bearing replacements.

3.4 Analysis of Rigidity

The static rigidity is the force required to obtain a unit deflection of the spindle due to its own bending under the force in addition to the deflection caused due to bearing elasticity.

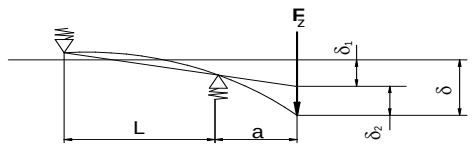

Fig: 3.2 Deflection of the spindle nose

The deflection of the spindle nose 'δ' for an unloaded spindle due to load F_z is given by [5]:

$\delta = \delta_1 + \delta_2$

$$\delta = F_z\left[\frac{1}{S_A}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L}+\frac{a}{I_a}\right)\right]$$

---eqn.3.1

Where,
δ_1= Deflection due to radial yielding of the bearings in mm
δ_2= Deflection due to elastic bending of the spindle in mm
a= Length of overhang in mm
L= Bearing span in mm (varying)
E= Young's modulus of Spindle material in N/mm^2
P= Cutting force in N
S_A=Stiffness of the front bearing in N/mm
S_B=Stiffness of the rear bearing in N/mm
I_L= Mass moment of inertia of the shaft at the bearing span in mm^4
I_a= Mass moment of inertia of the shaft at the overhang in mm^4

For diameter 65 mm front bearing, axial rigidity(C_a) for medium preloading condition is C_a=178.3 N/µm [08]

To calculate radial rigidity(C_r) for α=25^0 contact angle,

$C_r=2\times C_a$
C_r=2×178.3=356.6 N/µm

Radial rigidity of the Triplet back-to-back arrangement is
$S_A=1.36\times C_r$
=1.36×356.6=**485 N/µm**

For 55 mm diameter rear bearing, axial rigidity(C_a) for medium preloading condition is C_a=159.2 N/µm [08]

To calculate radial rigidity(C_r) for α=25^0 contact angle,
$C_r=2\times C_a$
C_r=2×159.2=318.4 N/µm

Radial rigidity of the duplex back-to-back arrangement is
$S_B=C_r$
S_B=318.4 N/µm

Table: 3.1 Specifications, size, & stiffness of different bearing arrangements.[8]

Sl. No	Details of bearing	Type and Specification of bearing	Bearing Size Details mm	Stiffness 'K' (N/µm)	Load rating dynamic C (KN)	Load rating static Co (KN)	Attainable Speed in (rpm)
1A	Front bearing	HSS 7012 ET P4S	d=60 D=95 B=18	457.2	1830	1900	15000
1B	Rear bearing	HSS 7011 ET P4S	d=55 D=90 B=18	318.4	1760	1760	17000
2A	Front bearing	HSS 7013 ET P4S	d=65 D=100 B=18	485	1900	2000	15000
2B	Rear bearing	HSS 7011 ET P4S	d=55 D=90 B=18	318.4	1760	1760	17000
3A	Front bearing	HSS 7014 ET P4S	d=70 D=110 B=20	534.2	2450	2600	13000
3B	Rear bearing	HSS 7011 ET P4S	d=55 D=90 B=18	318.4	1760	1760	17000

The stiffness of the measuring rod can be calculated from a beam model. It consists of steel with a Young's modulus of 2.1×10^5 N/mm²

The front bearing set should be positioned to minimize the overhang of the spindle nose. It is required to optimize the bearing spacing 'L_0' for maximum spindle stiffness. This requires examination of relative combinations to deflection, which arise from both bearing deflections and spindle bearing. By using bearings with a smaller cross section, larger spindle diameters can be used without changing the housing bore diameter or slightly increasing the housing bore diameter. This increases the bending rigidity of spindle and leads to increase in overall rigidity. Initially optimum bearing span is calculated using following eqn. (3.2) for each configuration. Details of different bearing arrangements [08] and calculated optimum bearing span length (L_0) are listed in table 3.2.

Calculation to find Optimum Bearing Span Length (Eqn.3.2):

$$Lo = \left[6EI_L\left(\frac{1}{S_A}+\frac{1}{S_B}\right)+\left(\frac{6EI_a}{aS_A}\right)Q\right]^{1/3} \quad \text{---3.2}$$

Where,
L_0= Static optimum Bearing Span Length in mm
E = Young's modulus = $2.1\text{x}10^5$ N/mm^2
a = Length of overhang = 65 mm
S_A=Stiffness of front bearing = $485\text{x}10^3$ N/mm (for dia. 65mm)
S_B = Stiffness of rear bearing = $318.4\text{x}10^3$ N/mm (for dia. 55mm)
I_a = Moment of inertia of the shaft at the overhang is found to be $375.5\text{x}10^3$ mm^4
I_L =Moment of inertia of the shaft at the bearing span is found to be $814.4\text{x}10^3$ mm^4
Q = Trial value for iterative determination of L_0=4xa = 4x65 = 260 mm

For First iteration, take Trial value of Q=260mm.Therefore

L_1 =209.765mm

For Second iteration, take Trial value of Q=209.77mm.

L_2=203.89mm

For Third iteration, take Trial value of Q=203.89mm.

L_3=203.19mm

For Fourth iteration, take Trial value of Q=203.19mm.

L_4=203.10mm

For fifth iteration, take Trial value of Q=203.89mm.

L_5=203.10mm= L_o

Table: 3.2 Details of different bearing arrangements [8] with optimum bearing span length (L_0).

Sl. No	Front bearings size mm	Bearing Stiffness N/µm	Rear bearings size mm	Bearing Stiffness N/µm	Attainable speed in rpm	Optimum bearing span length 'L_0' mm
'A'	Ø60x95x18 Triplet	457.2	Ø55×90×18 Duplex	318.4	15000	206.66
	Ø65x100x18 Triplet	485			15000	203.10
	Ø70x110x20 Triplet	534.2			13000	197.76
'B'	Ø60x95x18 Duplex	336.2	Ø55×90×18 Duplex	318.4	15000	227.17
	Ø65x100x18 Duplex	356.6			15000	222.90
	Ø70x110x20 Duplex	392.8			13000	216.14

Theoretical analysis has been carried out to evaluate spindle stiffness and to optimize the design to have maximum spindle nose deflection. Here the span length is varied from 140 mm to 230 mm. The variation of span length may become essential to accommodate the integral motor rotor. By considering spindle nose size BT-40 taper and also the flange, the overhang of the spindle is around 65 mm from the front bearing center. The diameter of the spindle at the front is

varied from 60 to 70 mm to evaluate its role in imparting the stiffness to the spindle system.

Calculation to find Defection i.e .$\delta = \delta_1 + \delta_2$ (From Eqn.3.1):

Where,

a = Length of overhang = 65 mm

L = Bearing span = 200 mm

E = Young's modulus of Spindle material = 2.1×10^5 N/mm^2

F_z = Cutting force =4500 N

S_A= Stiffness of the front bearing = 485×10^3 N/mm (for dia. 65mm)

S_B=Stiffness of the rear bearing = 318.4×10^3 N/mm

I_L= Moment of inertia of the shaft at the bearing span is found to be 814.4×10^3 mm^4

I_a=Moment of inertia of the shaft at the overhang is found to be 375.5×10^3 mm^4

$\delta = 30.41 \times 10^{-3}$ mm

or **δ = 30.41 µm**

and, Stiffness, **$K = F_z/\delta$**

=4500/30.41

K =147.97 N/µm

Variations of deflection and stiffness values for bearing arrangements 'A' and 'B' are given in the following table.

Table: 3.6 Variations of Deflections and Stiffness for bearing arrangements 'A'& 'B'.

Sl. no	Span length mm	Overhang mm	Front bearing dia. mm	'A'		'B'	
				δ µm	K N/ µm	δ µm	K N/ µm
1	140 to230	65	60	31.81 to32.88	141.46 to136.86	37.29 to38.93	120.69 to 115.60
2	140 to230	65	65	30.07 to31.26	149.64 to143.95	35.28 to36.70	127.56 to 122.63
3	140 to230	65	70	28.16 to29.53	159.79 to152.38	32.94 to34.08	136.61 to 132.05

The diameter of the spindle between the bearings has more influence on the rigidity as is evident from the tables 3.3 to 3.5. This diameter could be varied to the extent of a maximum of 65 mm from the practical considerations. To accommodate the integral motor rotor of 10 KW power rating, the span length of the spindle shaft is taken as 200 mm[Table.3.2] and the maximum diameter of 65 mm[Table3.1]. If the diameter of the spindle is increased, than the motor power may have to be increased and the bearing arrangements have to be reshuffled Therefore at the spindle diameter of 65 mm, the attainable speed is 15000 rpm which falls in-between the range of 14000-16000 rpm, so that the required spindle can run at the speed of 12000 rpm. Hence at the available optimum values, the stiffness of the arrangement 'A' has a value of around 147.97 N/µm and the deflection of the spindle nose based on static analysis is around 30.41 µm. From the point of view of static analysis, bearing arrangement type 'A' is chosen as an optimum design.

The variation of deflection and the stiffness of the spindle nose with varying span lengths are plotted in fig.3.3 and 3.4. The hatched part shows the optimum region of bearing span length.

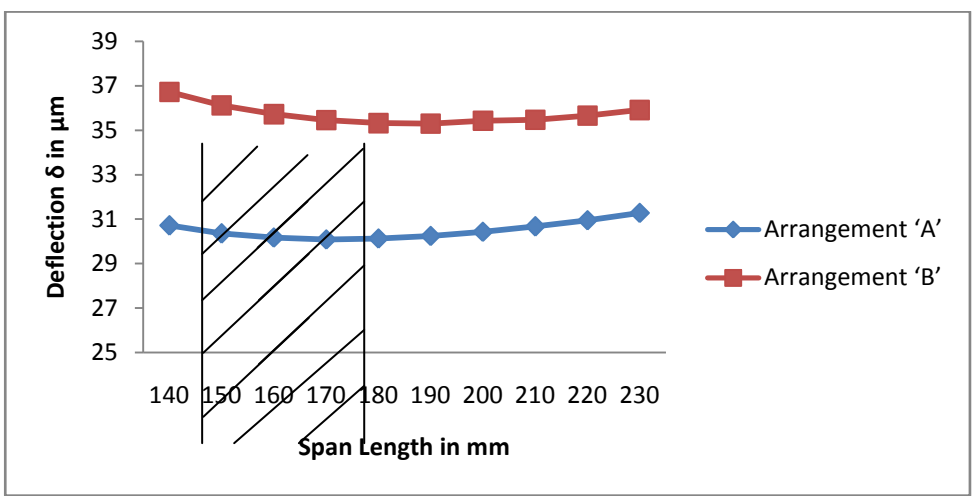

Fig: 3.3 Deflection at the nose for bearing arrangements (front bearing dia. 65mm)

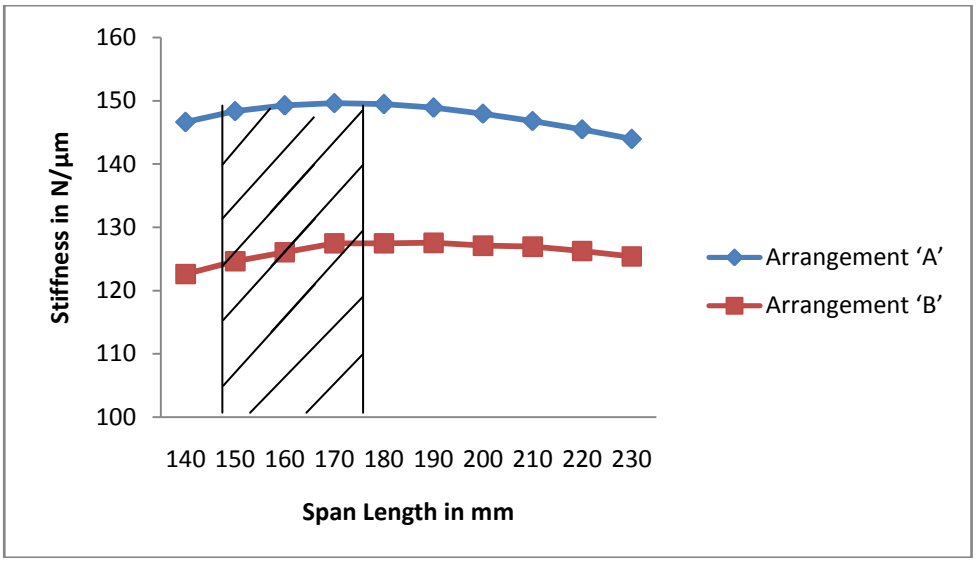

Fig: 3.4 Stiffness variations for bearing arrangements (Front bearing dia. 65mm)

The above figures show the effect of spindle elasticity and spindle deflection on the overall rigidity. The hatched part shows the optimum region of bearing span length. Variation of deflection and the stiffness when the bearing span is changed from 140 to 230 mm is not significant. This gives the designer flexibility in sizing the span to accommodate the integral induction motor.

4. STATIC STIFFNESS ANALYSIS USING FEM

4.1 Element Description: Beam Elements

There are several elements types used for FE analysis such as solid, plane and beam elements. For the spindle analysis for the bearing spans optimization, beam elements are the most suitable because the shapes of the spindle and the cutting tool are roughly cylindrical. Besides a time consuming iteration method will be used for the optimization, so the calculation time for the dynamic properties in each iteration step must be as short as possible. In order to shorten the calculation time, the simplest elements (the beam elements) are the most suitable type.

4.2 Static Stiffness Analysis

The geometric model is created in ANSYS. The finite element model is built using BEAM3 and COMBIN14 elements.

4.2.1Material Properties

For the static analysis of the spindle, the following properties are used:

- Modulus of elasticity =210 GPa
- Poisson's ratio =0.27
- Density =7.8×10^{-6} kg/mm^3

4.2.2 Boundary Conditions

The spindle is modeled using BEAM3 and COMBIN14 elements. A tangential force of **4500 N** is applied at the spindle nose as shown in fig.4.1.

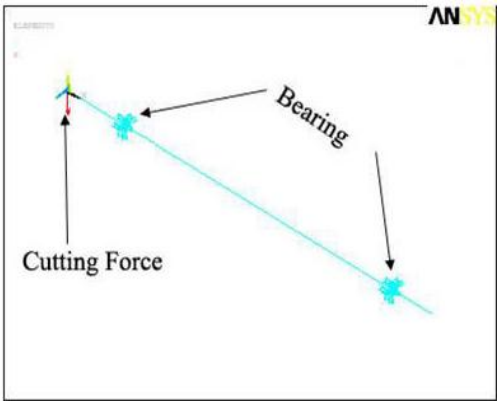

Fig: 4.1 Finite element model showing boundary conditions

4.3 Static Results

The deflection at the spindle nose in Y direction is computed for various configurations and Results are obtained for different diameters of spindle at the front bearing with 'A' type arrangement and span length of 200 mm.

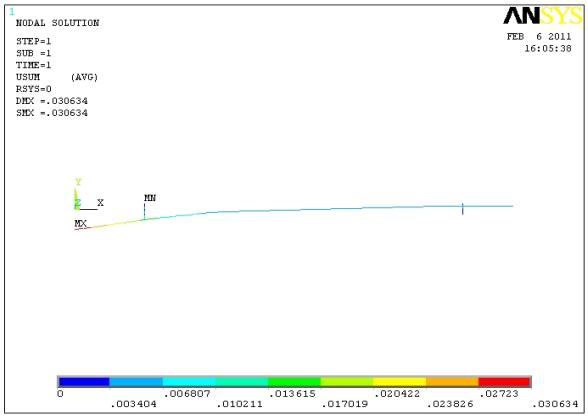

Fig: 4.2 Deflection at the spindle nose (front bearing Ø65mm, span length= 200 mm)

4.4 Analysis of Results

The deflection of the spindle nose in Y direction is function of bearing stiffness, span length and diameter of the spindle bearings for a given overhang. The deflection of the spindle nose in Y direction affects the machining accuracy.

Table: 4.1 Comparison of Theoretical & FEA values for deflection and stiffness at the spindle nose when **span length= 200 mm** with **'A'** type bearing arrangement.

Diameter at Front bearing mm	THEORETICAL		ANSYS	
	Deflection at spindle nose µm	Stiffness N/µm	Deflection at spindle nose µm	Stiffness N/µm
60	32.07	140.32	32.84	137.11
65	30.41	147.97	30.63	147.01
70	28.60	157.33	29.05	155.00

Table: 4.2 Comparison of Theoretical & FEA values for deflection and stiffness at the spindle nose when the span length various from **170 mm to 200 mm** for the front bearing with diameter of **65 mm** with **'A'** type Bearing arrangement.

Span length mm	THEORETICAL		ANSYS	
	Deflection at spindle nose µm	Stiffness N/µm	Deflection at spindle nose µm	Stiffness N/µm
170	30.07	149.64	30.44	147.92
180	30.11	149.45	30.32	148.51
190	30.22	148.90	30.49	147.68
200	30.41	147.97	30.63	147.01

The deflection at the spindle nose in Y direction and the stiffness values obtained through theoretical calculations and ANSYS are given in the table.4.1 and 4.2. The diameter could be varied to the extent of a maximum of 65 mm from the practical considerations. The stiffness obtained by ANSYS of the optimized spindle with given configuration is around 147.01N/µm at the span length of 200 mm.

In this we consider Triplet bearing set arrangement 'A' for the front end of the spindle, in which one pair of angular contact ball bearings are arranged in tandem with respect to each other and back-to-back with respect to a single angular contact ball bearing. Tandem bearing pair will carry both radial and axial loads equally shared.

4.5. Variation of Stiffness for Different Overhang Length

Fig.4.3 shows the deflection at the spindle nose for different overhang length and fig.4.4 shows the stiffness for different overhang lengths. As the overhang length increases the deflection at the nose increases and the stiffness decreases.

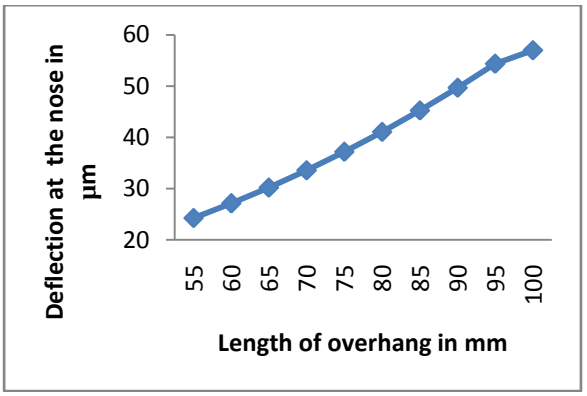

Fig: 4.3 Deflection at the spindle nose for different overhangs.

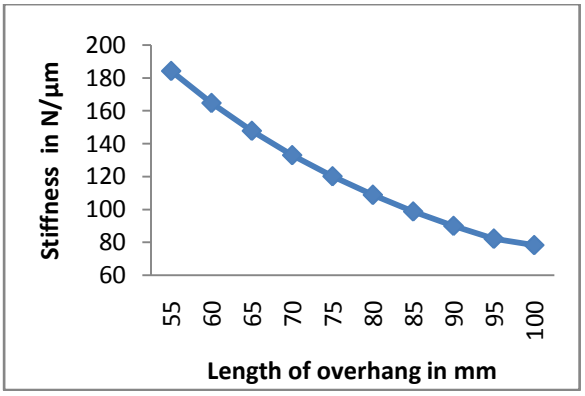

Fig: 4.4 Stiffness of spindle for various overhangs lengths.

5. CONCLUSIONS

- Variation of stiffness and deflection when the bearing span is changed from 140 to 230 mm is not significant. This gives the designer flexibility in sizing the span to accommodate the integral induction motor.
- The diameter of the spindle between the bearings has more influence on the rigidity as is evident from the results. This diameter could be varied to the extent of a maximum of 65 mm from the practical considerations of the bearing speeds. At this diameter the stiffness of the arrangement 'A' has a value of around 147.97 N/µm and the defection of the spindle nose based on static analysis is around 30.41 µm for the above arrangement. From the point of view of static analysis, bearing arrangement type 'A' is chosen as an optimum design.
- For a given overhang of 65 mm, it is evident that the bearing arrangement A, with triplet bearing set at the front, in which one pair of angular contact ball bearings are arranged in tandem with respect to each other and back-to-back with respect to a single angular contact ball bearing and the duplex bearing set at the rear mounted back-to-back shows the maximum stiffness of the spindle arrangement and is equal to 147.97 N/µm.
- The influence of the front bearing stiffness on the overall stiffness is quite considerable. Hence, the bearings with higher stiffness should be located at the front.
- Analysis of rigidity was also carried out using ANSYS and there is a good correlation between the results obtained by theoretical calculation and ANSYS.

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