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High speed machine tool spindle cooling by air throttling - A feasibility study

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Abstract- Spindle unit is one of the most critical sub-assemblies in High speed CNC Machine Tool.The Spindle is subjected to combined bending, torsion, axial and impact loads during metal cutting process. The design of spindle is essentially required to meet both axial and radial rigidity to produce high quality components and also to have minimal structural deflections. Present day CNC machine tool spindles are designed to run at high rotational speeds in order of 20000 rpm. At that speed large amount of heat is generated due to bearing friction and viscous friction in the spindle bearing zone. The frictional heat generated will cause the spindle thermal deformation and increases the bearing preloading force. To achieve high accuracy of machine tools, it is important to find effective methods for reducing thermal errors which cause position errors during metal removal. The generated heat is to be dissipated by different heat transfer modes in order to keep the thermal-structural stability of the spindle. In this study a new approach has been introduced to dissipate the heat by throttling the air to lower temperature and this cooled air is circulated in the bearing jacket to reduce the bearing temperature and control the thermal-structural deformation of the spindle and bearings. Software Simulation and experimental verification has been carried out to study the pressure and temperature variations during throttling process.

Index Terms- high speed spindle; bearing cooling; air throttling.

1. INTRODUCTION

Metal cutting is a major manufacturing process in various industries. New technologies are developed in metal cutting to enhance reliability in accuracy of produced components with minimum operational cost. In this manufacturing process CNC machines with high speed spindle beyond 10,000 rpm widely used for high speed machining to cater conventional and newer materials. In this present study a CNC vertical machining centre spindle, bearings and housing has been considered for the purpose of thermal-structural behaviour. The main source of heat generation takes place at high operational speeds due to frictional heat caused by spindle and bearings. The spindle is thermally deformed and affects the performance accuracy of the components produced .Positioning accuracy which depends on thermal errors is one of the critical performance factors affecting the machining precision of machine tool operation **[1].** In this present study Angular contact ball bearings are cooled by a new approach by throttling the high pressure compressed air and introduced in bearing cooling jacket for dissipating the heat. This paper presents design and fabrication of throttling test set with different multi orifice (MO) plates to study pressure and temperature variations during expansion [2]. These results are compared with simulation plots.

The following CNC Vertical machining centre is chosen for study and experimentation.

Machine model: AMS spark- Vertical Machining Centre
Make : ACE Micromatic group
Spindle taper : 7/24 No. 30
Spindle speed : 60-6000 rpm
Spindle speed (opt): 80-8000 rpm
Spindle power : 5.5/3.75 kW
Rapid traverse X/Y/Z: 20/20/15 m/min
Maximum tool weight: 2.5kgf

1.1. *Importance*

Working accuracy of a spindle depends on its radial and axial true running accuracy, static and dynamic rigidity and thermal behaviour.

When a tool holder is mounted in the spindle, the accuracy of rotation is extremely important as it affects the roundness of the components produced for example boring operation. The rotational accuracy of spindle is dependent on the quality and design of the bearing arrangement used and preloading. The bearing should support the spindle radially and axially.

The accuracy and the quality of the work produced depend directly on the geometrical accuracy, running accuracy and the stiffness of spindle assembly.

The dynamic accuracy of a spindle depends on the thermal stability material used manufacturing process especially for high speed machining applications. Considering the same, appropriate bearing lubrication and spindle cooling systems should be adopted in the design of spindles for machine tools.

Therefore, designing of spindle for structural and thermal stability is very important.

1.2. *Design Approach of Spindles for Bending Criteria [3,4]:*

The spindle represents a shaft with

(a) Support length 'l' acted upon by the driving force p_2

(b) Cantilever of length 'c' acted upon by external force p_1.

The spindle is basically designed for bending stiffness such that the maximum deflection of the spindle nose should not exceed a certain prescribed value as per ISO standards.

$Y_{max} \leq [y]$

The total deflection of the spindle nose consists of:

i. Deflection Y_1 of the spindle axis due to bending forces P_1 and P_2

ii. Deflection Y_2 of the spindle axis due to compliance of spindle support.

Deflection of spindle axis due to bending as shown in fig.1:

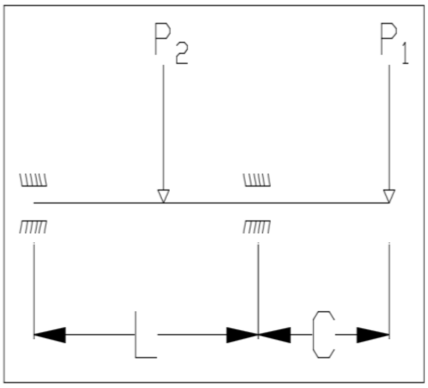

Fig. 1. Schematic diagram of spindle

$$y_1 = \frac{1}{3EI}[P_2C^2(L+C) - 0.5P_2ab\left[1-\frac{a}{l}\right] - M_{rl_c}$$

--------------(i)

M = k × M

K=coefficient which varies from k=0 at small loads to k=0.3-0.35

E=modulus of elasticity of spindle material

I=average moment of inertia of spindle section

Deflection of spindle axis due to compliance of spindle support as shown in fig. 2:

$$y_2 = p_2a - M_r + \frac{p_1(l+c)}{l.k_b}\left(1+\frac{c}{l}\right) + p_2b + M_r - p_1.\frac{c}{l.k_a}.\frac{c}{l}$$

--------------(ii)

$Y_{max} = Y_1 + Y_2$

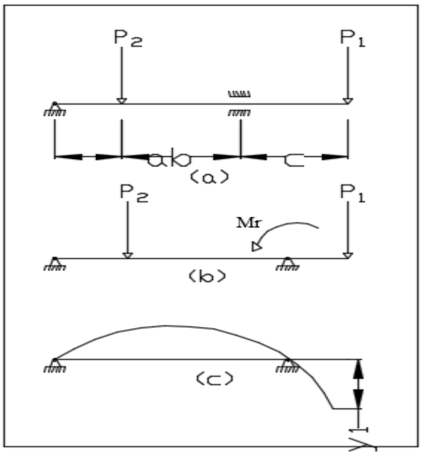

Fig. 2.

a) Schematic diagram of spindle
b) Design diagram of spindle
c) Deflected axis of spindle

1.3. *Preloading*

The ball bearings have some account of radial and axial clearances, when a main spindle is mounted on bearings, there should not be any axial and radial play in the main spindle assembly. To achieve this, the clearances in bearings have to be taken up by preloading, which is the initial load applied to the bearings as shown in fig. 3.

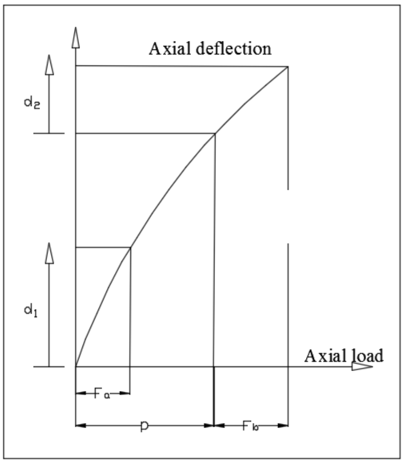

Fig. 3. Deflection curve as function of load

Fig. 3 shows the deflection curve as function of load. From the graph it is clear that deflection d_1 for a bearing without preload is much more than deflection d_2 having an initial preload P. Therefore, Preloading will reduce the deflection which occur in spindle and provides spindle stability and axial rigidity of bearings.

1.4. ***Spindle growth/ Thermal Deformation***

In high speed machining, the excessive heat generation in the spindle induces uneven thermal expansion. The heat sources within spindle system are heat generation by angular contact ball bearings under influence of speed, pre-load and lubrication. This phenomenon is leading to excessive thermal pre-loading of the bearing system resulting in bearing seizure/failure. Hence spindle and bearing units cooling is essential at very high speeds.

1.5. ***Heat Generation***

The major heat generation of the system is caused by the cutting process and the friction between the balls and races of the bearings. Assumed that the majority of cutting heat is taken away by coolant and chips, the heat generated by bearings is the dominant cause of temperature change. In angular contact ball bearings heat is generated mainly by three sources. The heat generated by a bearing can be computed as

$$H_f = 1.047X10^{-4}\ n\ M \quad \text{------------- (iii)}$$

Where H_f is the heat generated power (W), n is the rotating speed of the bearing (rpm), M is the total frictional torque of the bearing (N mm). The total frictional torque M consists of two parts, one is the torque M_1 due to applied load and the other one is the torque M_2 due to viscosity of lubricant. That is

$$M = M_1 + M_2$$

Frictional torque due to applied load (M₁):

The torque due to applied load can be empirically approximated by the following:

$$M_1 = f_1 \times F_\beta \times d_m \quad \text{……… (iv)}$$

In which f_1 is the factor depending upon bearing design and relative load. For ball bearing

$$f_1 = z\ (F_s/C_s)^Y \quad \text{…… (v)}$$

Where F_s is the static equivalent load and is given by

$$F_s = X_o\ F_r + Y_o\ F_a \quad \text{…….. (vi)}$$

F_r is the radial force acting on the bearing which is consider as zero, Fa is the axial force on bearing and the values are given at 15^o angle as Xo = 0.5, Yo = 0.46 and the axial force Fa= 945 N. By solving the above equation (vi) yields

F_s = 434.7 N

C_S is basic static load rating gives as, C_S = 4850 N. For angular contact ball bearing having contact angle 15^0 the appropriate values of Z and Y

Z= 0.001 Y = 0.33

So by solving equation (v) f_1 is obtained has

f_1= 0.00045

F_β is dynamical equivalent load, for angular contact ball bearings generally depends on the magnitude and direction of the applied load, F_β is given as

$$F_\beta = F_a - 0.1F_r \quad \text{……… (vii)}$$

Since F_r, the force in radial direction is zero then the equation (vii) gives

F_β = 945 N

Finally by solving the equation (iv) by taking pitch diameter of bearing d_m = 50 mm, the torque developed due to applied load is given as

M_1 = 21.2625 N mm

Viscous Friction Torque (M_2):

For bearings that operate at moderate speeds and under non-excessive load, the viscous friction torque can be empirically expressed as follows:

$$M_2 = 10^{-7} f_o\ (v_o n)^{2/3} d_m^{\ 3},\ v_o n \quad \text{...............(viii)}$$

In which v_o is the kinematic viscosity of lubricated oil given in centistokes, the value of v_o is taken, depending up on the temperature. Here we assumed the uniform temp as 40 and the value taken is,

v_o = 20

And n is the revolution speed of bearing in rpm

n= 2000rpm to 6000rpm

f_o factor depending upon the type of bearing and method of lubrication. The value is taken from harries [4]. It is taken as

f_o = 2

M_2 = 29.2401 N mm

The overall torque is given from

$M = M_1 + M_2$

M =50.5026 N mm

Finally the heat generated by a bearing is given as

- $\mathbf{H_f = 1.047X10^{-4}\ n\ M}$

Therefore at 2000 rpm the heat generated in angular contact ball bearing is

- $\mathbf{H_f = 10.575\ W}$

Similarly the heat generated at various speeds are calculated and listed as given in table 1 and shown in fig. 4.

Table 1. Heat Generated at Various Speeds

S No	Rotational speed (rpm)	Heat generated (W)
1.	2000	10.575
2.	3000	18.7134
3.	4000	28.3436
4.	5000	39.326
5.	6000	51.56
6.	8000	79.523
7.	10000	111.778
8.	12000	148.017
9.	14000	188.004
10.	16000	231.583
11.	18000	278.487
12.	20000	328.721
13.	22000	382.102
14.	25000	467.88

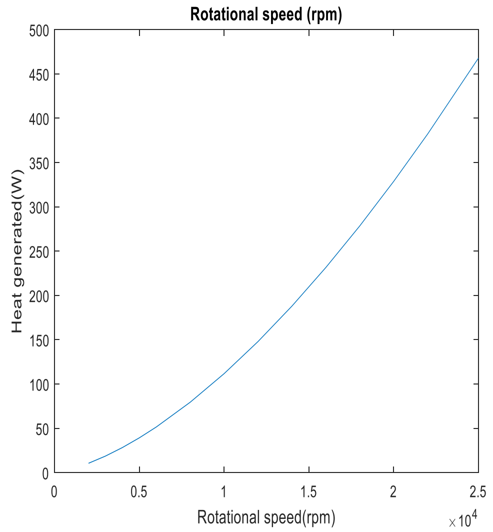

Fig. 4: Graph 1- Bearing heat generation rate as a function of rotational speed

1.6. *Present System of Cooling*

Cutting fluid is a type of coolant and lubricant designed specifically for metalworking and machining processes. There are various kinds of cutting fluids, which include oils, oil-water emulsions, pastes, gels, aerosols (mists), and air or other gases. They may be made from petroleum distillates, animal fats, plant oils, water and air, or other raw ingredients.

Water is a good conductor of heat but has drawbacks as a cutting fluid. It boils easily, promotes rusting of machine parts, and does not lubricate well. Therefore, other ingredients are necessary to create an optimal cutting fluid.

A typical CNC machine tool usually uses emulsified coolant as shown in fig. 5, which consists of a small amount of oil emulsified into a larger amount of water through the use of a detergent.

Fig. 5. Vertical Machining Centre – Metal Cutting Operation using emulsified coolant

Disadvantages

Cutting fluid contains biocides, corrosion inhibitors, tramp oils which causes ozone depletion, central nervous system and respiratory tract problems.

2. EXPERIMENTATION

2.1. *Proposed System of Cooling by Throttling Process*

2.1.1. *Throttling Process*

Whenever a fluid expands from a region of high pressure to region of low pressure through a porous plug, partially opened valve or some obstruction without exchanging any energy as heat and work with the surroundings (neglecting the change in K.E and P.E), the enthalpy of the fluid remains constant and the fluid is said to have undergone throttling process.

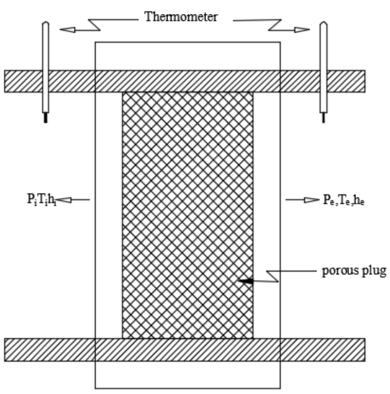

Fig. 6. Sketch of porous plug experiment of joule-Thompson

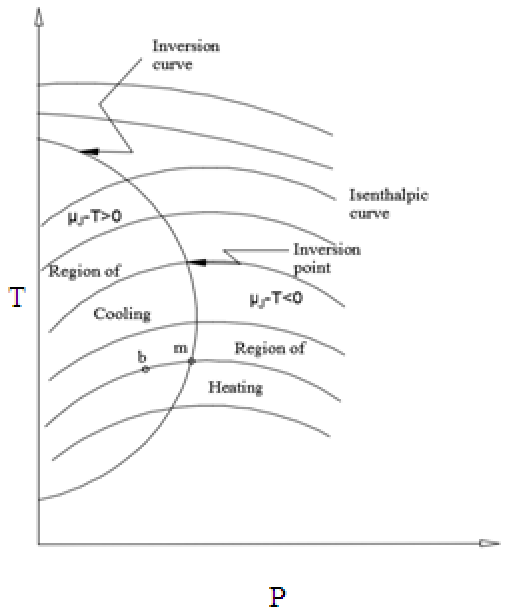

Fig. 7. Inversion and isenthalpic curve of real gas (Pressure, P vs. Temperature, T)

Suppose that a series of Joule Thompson experiments are conducted on same gas with a given initial pressure p_i and temperaturet_i but with different downstream pressures p_e different exit temperatures t_e are observed after expansion. The downstream pressures p_e is altered by controlling the opening of throttle valve. For, this set of experiments since p_i and t_i are fixed the final set of p_e,$t_e$correspond to same enthalpy. A plot of t_e,p_e yields an isenthalpic curve. The slope of isenthalpic curve is called joule-Thompson coefficient μ_{JT} and is given by:

$$\mu_{JT}=(\partial T/\partial P)_h$$

Experiments are conducted with differentp_i,t_i different isenthalpic curves are obtained. A family of isenthalpic curves is shown in fig.6

The point at which $\mu_{JT} = 0$ is called inversion point as shown in fig. 6 and fig. 7, the locus of all inversion points is called inversion curve. In the region to the left of inversion curve $\mu_{JT} > 0$, therefore whenever a real gas which is present on left of curve and subjected to throttling, temperature of gas decreases. To the right of inversion curve $\mu_{JT} < 0$, therefore any real gas which is present on right of curve and subjected to throttling, temperature of gas increases.

2.2. *Structural design for multi orifices and its experimental application*

Orifice arrangement criteria:

The random number and arrangement of orifices leads to the complexity of geometry. To avoid this complexity, the following criteria were defined for designs discussed in the present work.

(1) All holes should have a uniform size and are manufactured using the same technique. The orifice plate thickness was 3.15 mm. Detailed craft specifications were according to ISO5167-1, 2 [**5,6**].

(2) Each orifice center was located in a set of concentric circles defined as center circles.

(3) Orifice arrangement should be as symmetrical as possible, and the spacing among most orifice centers should be equal.

(4) The total orifice number ranged from 19 to 31.

(5) The various patterns of orifice arrangement can be further categorized as circular arrangement and rectangular arrangement.

Two different center circle arrangements were considered in the present work. The first mode is a circular arrangement, in which the center circle location was primarily fixed resulting in symmetric orifices located around these center circles as shown in Fig. 8, Fig. 9.

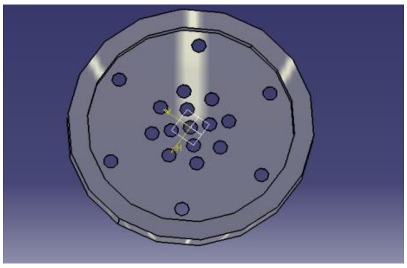

Fig. 8. (19-1-c)

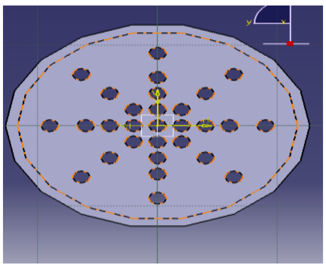

Fig 9: (31-1-c)

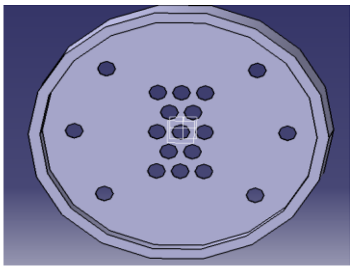

Fig 10: (19-1-c)

The second mode is a rectangular arrangement with four orifices at each center circle. This arrangement primarily considers the consistency of the spacing between each orifice center, resulting in fixed center circle locations as shown in Fig. 10.

The circular arrangement is better than the rectangular arrangement, and the location of the center circle is not restricted by the orifice spacing, while the rectangular arrangement has a more symmetric orifice distribution and the orifice spacing controls the location of the center circle.

2.3. *Key geometric parameters*

According to the above definitions, the following three geometric parameters were used to quantitatively characterize the multi orifices (MOs) geometry.

i. The total orifice number, *n*.

ii. The equivalent diameter ratio, *EDR*, which represents the square root of the total open area ratio and is expressed as $n^{0.5}d/D$. *EDR* has the same geometric meaning as the diameter ratio defined for small hole orifice (SO) specified in ISO5167.EDR should be in the range of 0.2 to 0.5

n=number of orifices

d=diameter of orifice (mm)

D=inner diameter of tube (mm)

The above arrangement criteria and geometric parameters can be used to explicitly describe the geometry.

2.4. *Design of throttling setup for cooling*

Outer Diameter of tube= 25.4mm

Length of tube=100mm

Thickness of tube wall=2mm

Thickness of plate=3.15mm^2

(Plate thickness/hole diameter)=1.0 to 3.0

Hole pitch=2mm

Reference pressure =12 bar

Reference temperature=35℃

2.4.1. *Model-1*

19-1-c (19 holes with one hole at the centre) as shown in fig.7

Analysis results on the above model from ANSYS WORKBENCH, for varying the mass flow rate as 0.1 kg-m/sec, 0.2 kg-m/sec, 0.3 kg-m/sec, 0.4 kg-m/sec and 0.5 kg-m/sec and the results were tabulated as follows.

Table 2: Model 1

Mass flow rate (kg/sec)	Upstream pressure (p_{su})in bar	Downstream pressure (p_{sd}) in bar	Inlet temp (K)	Outlet temp (K)	Velocity (m/s)
0.1	6.15	-0.89	308.2	302.4	7.642
0.2	19.2	0.285	308.3	297.4	125.2
0.3	32.6	2.011	308.4	300	104.8
0.4	46.14	4.885	308.5	289.4	138.3
0.5	58.7	7.608	308.4	287.3	145.3

2.4.2. *Model-2*

31-1-c (31 holes with one hole at the centre) as shown in fig.9.

Analysis results on the above model from ANSYS WORKBENCH, for varying the mass flow rate as 0.1 kg-m/sec, 0.2 kg-m/sec, 0.3 kg-m/sec, 0.4 kg-m/sec and 0.5 kg-m/sec and the results were tabulated as follows.

Table 3: Model 2

Mass flow rate (kg/sec)	Upstream pressure (p_{su})in bar	Downstream pressure (p_{sd}) in bar	Inlet temp (K)	Outlet temp (K)	Velocity (m/s)
0.1	38.2	0.06	308.2	304	4.64
0.2	38.2	-1.57	308.3	288.6	9.983
0.3	38.5	-0.399	308.4	285.1	108.3
0.4	38.7	-0.2882	308.5	277.1	125.4
0.5	39.6	1.382	308.4	271.3	136.6

2.5. *Experimental setup*

The experimental setup is as follows:

1) Copper plate of thickness 3mm and a standard size of 6 inch*6 inch, sheared for the sake of convenience to a size of 3 inch*3 inch.
2) 2 Galvanized iron pipes with aluminium flange setting of total length 203mm.
3) 6 M4*45 bolts and nuts for holding together the plate and pipe-flange setup as shown in fig. 11 and fig. 12 as per the bill of materials given in table 4..

The desired orifice pattern was drilled by a 2mm drill bit on the copper plate and the setup was fixed together. Facing operation had been done on the flanges, prior to the assembling of the model, to create an even surface on the side where the plate would be fixed. Also all the unoccupied spaces inside the model had been filled with parts such as bushes (in flanges), in return provided with gaskets, to ensure that the whole model would have an equal passage diameter of 13.5mm To ensure that there is no heat exchange between the surrounding environment and the model, the model has been completely insulated with Teflon tape. Two holes of diameter 6mm were drilled on either side of the pipe for the thermometer arrangement for temperature measurement at steady state during the experimentation. Air leakages may affect the pressure to some extent. To avoid this problem, the model was provided gaskets in the form of rubber and was also sealed completely with M-Seal.

With the above setup, the experiment was conducted successfully and the measured temperature was 18°c after throttling. The applications are shown in fig 13 and fig 14.

Table.4: Bill of materials of manufactured model

S No.	Name of the part	Quantity
1	Copper plate	01
2	M4*45 bolts	06

3	Flange assembly with bushes	02
4	Pipes	02

Fig 11: A portion of throttling test setup

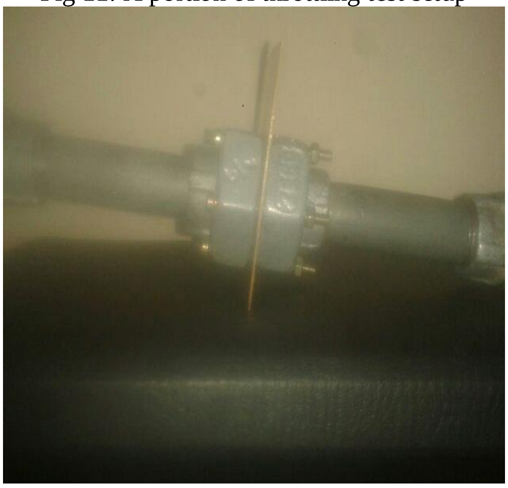

Fig 12: Experimental Setup

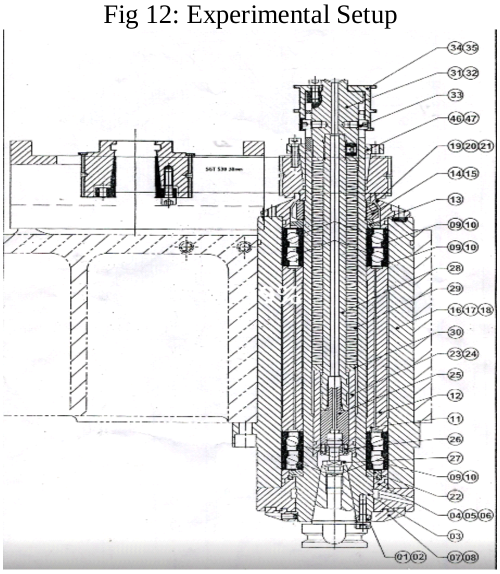

Fig 13: Illustration of the spindle and Bearing arrangement [7]

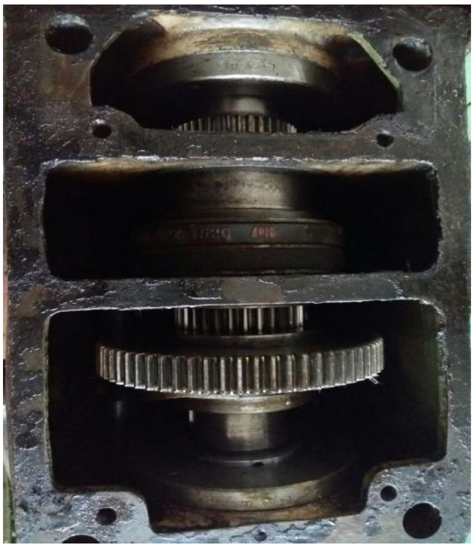

Fig 14: Spindle of lathe

3. RESULTS & DISCUSSIONS

3.1. *Analysis*

Analysis has been carried out in ANSYS CFX, by taking reference pressure as 12bar and reference temperature as 35 and the results were tabulated as follows [8, 9]. Modeling is shown in fig 15, meshing is shown in fig 16, boundary conditions applied is shown in fig 17, solution is shown in fig 18, pressure plot is shown in fig 19, temperature plot is shown in fig 20. The inlet and outlet conditions considering the throttling parameters are given in table 2 and table 3 for model 1 and model 2 respectively for conducting the throttling simulation.

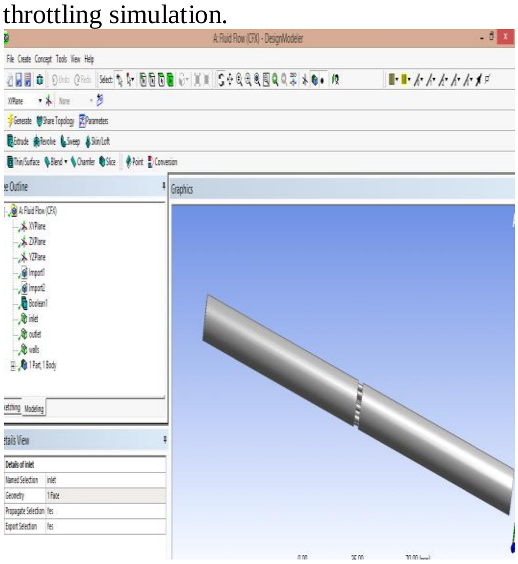

Fig. 15: Importing and Naming

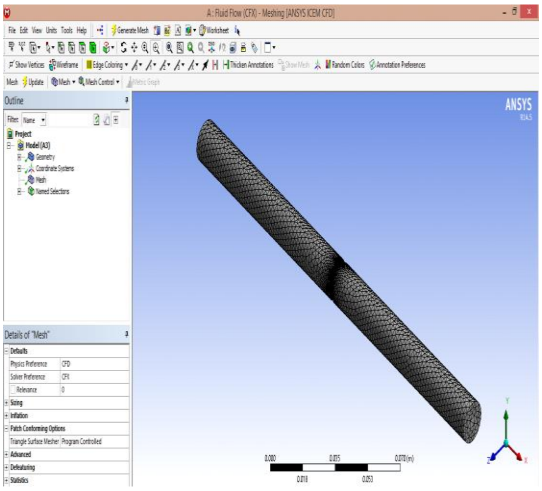

Fig. 16: Meshing

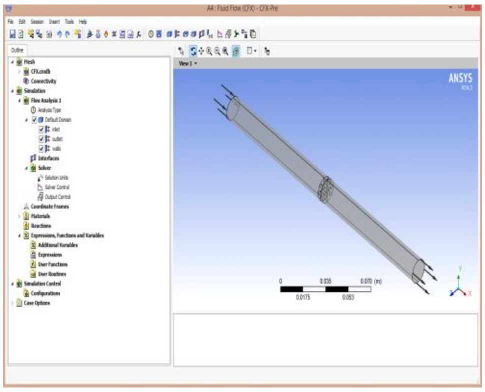

Fig. 17: Setup

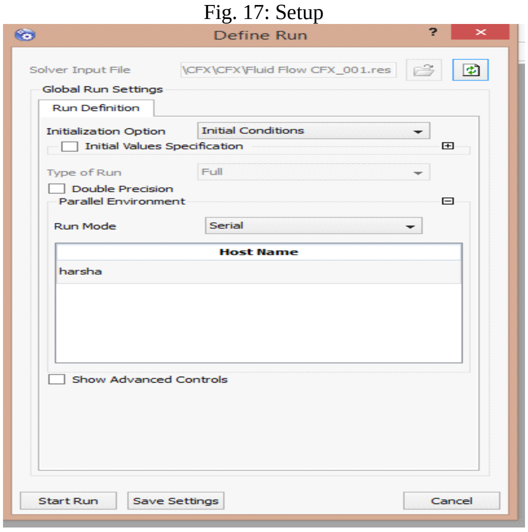

Fig. 18: Solution Run

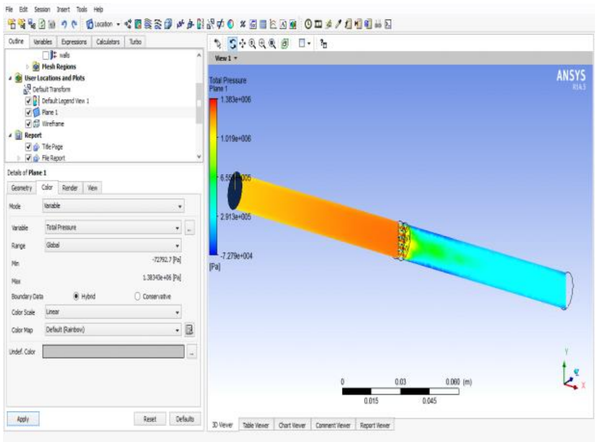

Fig. 19: Pressure Plot

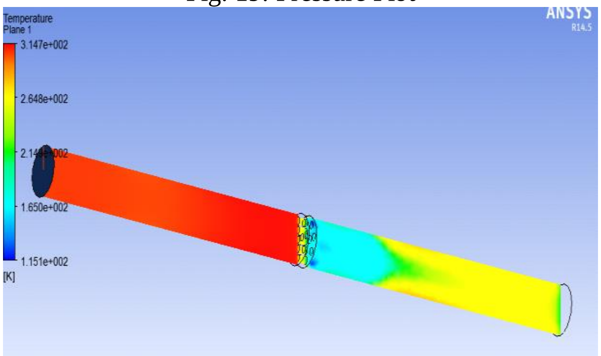

Fig. 20: Temperature plot

Table 5: Pressure and Temperature Data

Inlet	Outlet-ANSYS
Velocity (V)=1.828e+001 m/sec	V =4.244e+002 m/sec
Discharge (Q)=40.3 kg/sec	Q=251.26 kg/sec
Pressure(P)=1.291e+006	P=2.682e+005 Bar
Temperature(T)=3.147e+002	T=2.648e+002 K

3.2. *Experimental*

Pressure and temperature measurement has been carried out. It was found that the pressure is 1.9 e+005 bar and temperature is 2.92 e+002 K.

There is a deviation from ANSYS to the experimental results as the insulation (on adiabatic wall) is improper. The temperature deviations are observed because of conduction – convection losses in the downstream.

4. CONCLUSIONS

1. It is established that heat generated is a function of rotation speed of the spindle system.
2. Air throttling simulation ANSYS results show that temperature is decreasing and potential energy is converted into kinetic energy resulting in bearing

cooling and heat dissipation is taking place at high velocity through air passages.

3. Results from table 2 (model 1) and table 3 (model 2) clearly indicates the high pressure energy is converted into kinetic energy during trotting process. From model 2, the analysis work bench simulation results indicate 37.1 K temperature reduction and velocity of air at exit is increased to 136.6 m/sec at downstream with pressure of 1.382 bar and this result is use full for real time application in the bearing cooling for high speed machine tool spindle systems.
4. In, the present day cooling system use of cooling fluids are not eco-friendly and would cause ozone depletion, health problems to the workers. The proposed system of air cooling by throttling process is very economical and eco-friendly which promotes GREEN COOLING.
5. It was observed that the pressure reduction is high and the temperature at the exit is found to be comparable with ANSYS results within 10% range.

5. ACKNOWLEDGEMENTS

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