

Auto-centering of seamless pipe for NDT using Three Guide Roller FLD Technique

Pankaj R. Beldar¹

¹Department of Mechanical Engineering, SPPU

Abstract— Auto-centering of shaft is the key operation in the industry. This paper deals with actual auto-centering requirement of the seamless pipe manufacturing industry. Seamless pipe is to be feed for the non destructive testing (NDT) but there were few problems such as jamming of upper roller, wearing of roller material. Sometimes offsetting of roller axis is also seen. This causes incoming pipe to struck in between. Many times manual interference is required to ensure axis alignment. Bearing failure also occurs and ultimately the proper inspection of pipe in ultrasonic setup and eddy current test setup cannot be done. Hence development of auto-centering mechanism comes in the picture. Along with the development of mechanism vibration level should be less. Free layer damping treatment is used to avoid vibrations of rollers and to reduce wear of the rollers. Tri-roller design along with FLD patches is suggested in this paper to improve the life of machine. Vibration testing is done with NI-LabView software to measure damping loss factor with MATLAB code.

Keywords— Auto-centering, Constrained layer damping, Viscoelastic material, auto-feeding, vibration testing

I. INTRODUCTION

The industry manufactures the seamless pipes for various industrial applications. After manufacturing they need to do non-destructive testing for better quality as the pipe should not be damaged. They perform the ultrasonic testing on the pipe. For the same cause they need to feed the pipe with horizontal axis aligned with NDT machine axis. Previously they use mechanism shown in fig. 1. But there were lots of problems such as bearing failure, wear and tear of guide rollers, vibrations etc. To overcome this problem three guide roller system is to be designed for stability and reduction of vibration. The line speed was 0.4 m/s and pipe diameter was restricted to 90mm. New system will deal with 0.25-0.5 m/s line speed and pipe diameter range of 20-350mm.



Fig:1 previous feeding mechanism

II. DESIGN OF ROLLING MECHANISM

In free layer damping, the layer of damping materials is attached on base structure with the help of roller or any pressure application devices.

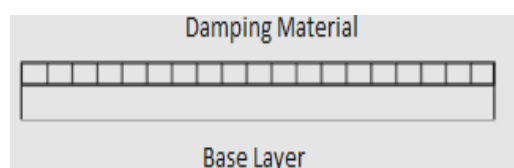


Fig: 2 free layer damping treatment[2]

- A. Rollers are the basic functional unit of the system. They guide the pipe horizontally to the testing unit. For proper gripping of pipe and continuous guiding, we have taken 3 rollers. Two of these rollers are taking care of load due to weight of pipe. Third roller which is from top side is for proper positioning of pipe to avoid offsets if any. The testing

setup has to check the pipes of different diameters, so we will have to adjust the rollers accordingly i.e. to vary the centre distance. Schematics of roller are as shown in fig.4 with Hardness: 40-44 HRC

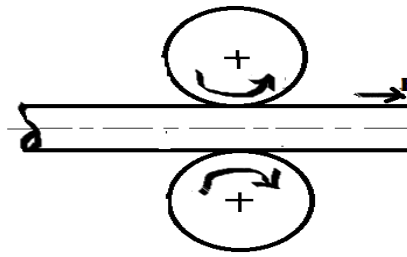


Fig: 3 Schematics of rollers

Hardness 40-44HRC

$V=0.5\text{m/s}$, $N=112\text{rpm}$

$V = (\pi DN)/60$

$D=85.26\text{mm} \sim 85\text{mm}$

Roller diameter= $85\text{mm}=0.085\text{m}$

i.e. Radius(R)= 0.0425m

Roller material= En-24(steel)

Force acting on Roller

$F=\mu W/R$ where W =load on wheel

R =Radius of wheel

Length= 2.5m , wall thickness= 15mm

OD of pipe= 88.9mm , ID of pipe= 73.9mm

Density (ρ)= 14180

Weight of maximum diameter pipe= $14180 * \pi/4 * (0.0889^2 - 0.0739^2) * 2.5 = 68\text{kg}$

$\mu=0.3$

$F=4708.8\text{N}=4.7088\text{kN}$

Power=Force*Velocity= $4.7088*0.33= 1.554\text{kW}$

B. Assumed Position of Rollers at 120°

In this first we have assumed the 3 rollers are arranged in 120° angle with each other. So it will provide proper positioning and load distribution. Weight of pipe is distributed on two bottom rollers and the third top roller is for guiding purpose.

Line speed= 0.5m/s

Maximum diameter of pipe= 90mm

Length= 2500mm

Thickness= 15mm

Weight= $14180 * \pi/4 * (0.09^2 - 0.075^2) * 2.5 = 68.909\text{kg}$

Resultant Force on Roller1

$R1=W\cos\theta1=68.909*9.81*\cos60=337.99\text{ N}$

Resultant Force on Roller2

$R2=W\cos\theta2=68.909*9.81*\cos60=338\text{N}$

Load is 338N

Calculating force exerted by cylinder on roller

$R=F/P*A=2.5 * 100000 * \pi/4 * 0.125^2 = 3108\text{N}$

Total load on roller= $338+3108=3448\text{N}$

$\sigma = (M*y)/I$

$M=3448 * 30 * 0.001/2 = 51.69\text{Nm}$

$y = 0.0425\text{m}$

$I = 3.14*64*(0.0085^4 - 0.0025^4) = 2.54*10^{-6} \text{ m}^2$

$\sigma = 86.489*10^4 \text{ N/ m}^2$

$\sigma_{\text{allowable}} = \sigma/3 = 283.33*10^6 \text{ m}^2$

$283.33 * 10^6 > 86.489 * 10^4$

Hence Design of Roller is safe

C. Analyzed Position of Rollers at 36°

For the proper gripping of pipe and due to cylinder stroke the roller position of upper roller is not exactly on axis. So there is a need of another angular position. This position can be calculated by CAD Modeling. And these positions are

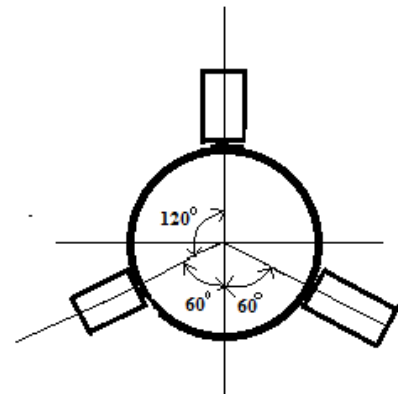


Fig. 4 Schematics of rollers(60-60-60)

totally depending on the shape of lever position with respect to cylinder rod. So from new angular positions the further forces on the rollers are calculated.

Resultant Force on Roller1

$$R1 = W \cos \theta_1 = 68.909 * 9.81 * \cos 36^\circ = 547 \text{ N}$$

Resultant Force on Roller2

$$R2 = W \cos \theta_2 = 68.909 * 9.81 * \cos 84^\circ = 71 \text{ N}$$

Load is 547N

Calculating force exerted by cylinder on roller

$$R = F \cos 25^\circ = P * A \cos 25^\circ = (2.5 * 100000 * 3.14 / 4 * 0.125^2) \cos 25^\circ = 2817 \text{ N}$$

$$\text{Total load on roller} = 547 + 2817 = 3364 \text{ N}$$

$$\sigma = (M * y) / I$$

$$M = 3364 * 30 * 0.001 / 2 = 50.46 \text{ Nm}$$

$$y = 0.0425 \text{ m}$$

$$I = 3.14 * 64 * (0.0085^4 - 0.0025^4) = 2.54 * 10^{-6} \text{ m}^4$$

$$\sigma_{\text{allowable}} = 84.43 * 10^4$$

$$\sigma_{\text{allowable}} = \sigma / 3 = 283.33 * 10^6 \text{ m}^2$$

$$283.33 * 10^6 > 84.43 * 10^4$$

Hence Design of Roller is safe. Free layer damping (FLD) patch is mounted on the roller to avoid vibrations and to improve grip as shown in fig.

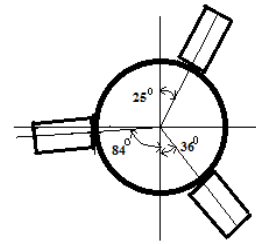


Fig: 5 Schematics of roller (84-36-25)



Fig: 6 Roller with FLD treatment

Polyurethane elastomers are special rubbers whose performance and characteristics of process ability have been designed to satisfy particular technical demands in the sector of the rubber. Excellent mechanical properties (elevated elongation and tensile strength), good elastic characteristics from -76 to 230°F (-60 to +110°C), good abrasion resistance, good compression set from -22 to +158°F (-30 to +70°C), excellent resistance to light, ozone, to the oxidation and the atmospheric agents, good/excellent chemical resistance (aliphatic aromatic solvents), low gas permeability.

III. ANALYSIS OF ROLLERS WITH FLD

Finite Element Analysis of this contact point is carried out in ANSYS workbench. The IGS model was imported as model geometry from CATIA software. Contact pressure and maximum shear stress are analysed from numerical results and compared with analytical results. Stress and Pressures developed in roller and pallet The various stresses and pressures developed are as shown in the figures below.

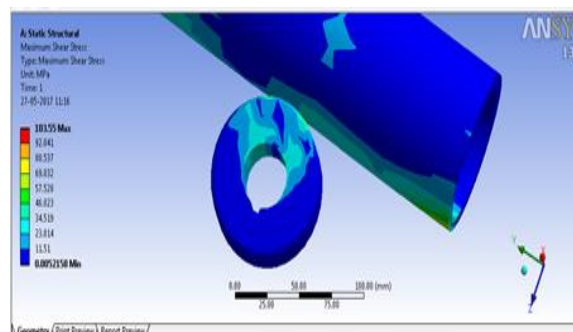


Fig: 7 Maximum Shear Stress Analysis

Contact stress analysis is very important because high localized stress is generated which will lead to fatigue of component. Finite Element Analysis is done in ANSYS Workbench in which contact pressure and shear stresses in roller contact area are analysed. As this value is below yield limit of material the plastic deformation will not be occur. The comparison between the theoretical and analytical results shows that the FEA results are acceptable. The difference between the results is within 10%. The difference in results is due to the approximation made as i.e. considering only the region where the contact occurs. The value of contact stress is very important, as the stress value changes with contact

area. Higher the contact area the stress values will be less and for lesser contact area the stress values will be higher. The contact stress between the roller and pipe is important in order to ensure the stress generated is within the elastic limits, this also helps in predicting the fatigue life by plotting the value of stress in S-N (stress v/s number of cycles) curve of the material. Based on required fatigue life the stress values can be optimized by modifying permissible load carrying capacity or by changing the dimensions of pipe and roller. The actual set up (new system) is shown in fig: 8 with three rollers without FLD treatment.



Fig: 8 new tri roller system without FLD treatment

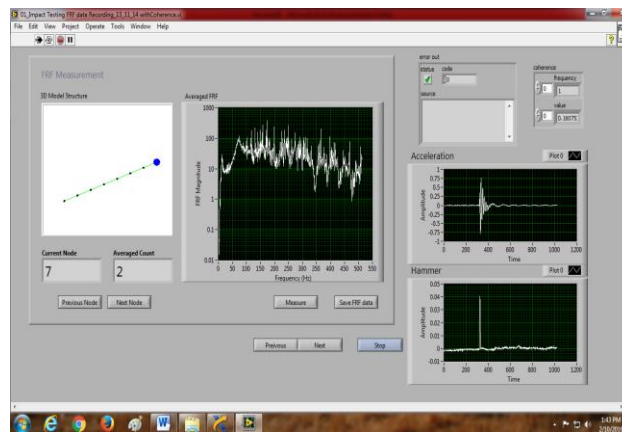


Fig:9 NI Lab VIEW FRF (Vibration Testing-Natural Frequency and Damping loss factor)

Above figure: 9 show the FRF (Frequency Response Function) data for frequency range (0-550Hz) here we can observe first natural frequency. We should take care of sharp pick for hammer impact as shown in figure: 9. here you can observe a sharp pick of hammer. On right side you can observe time domain analysis .clearly you can observe the decrement in the vibration amplitude with respect to time. Here you can calculate the logarithmic decrement of the response. With help of logarithmic decrement you can calculate damping ratio and damping loss factor.

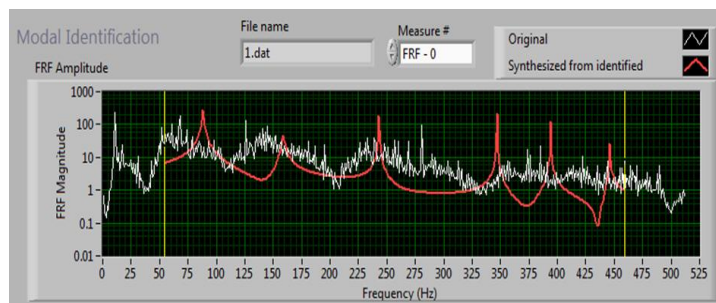


Fig :10 Roller vibration FRF testing Results

IV. RESULTS

As thickness increases the system loss factor is increases gradually. For thickness 1mm the system loss factor is 0.07413. and damping factor is half of system loss factor i.e. 0.03706. Inputs for MATLAB program are storage and shear modulus calculated from cyclic loading machine.

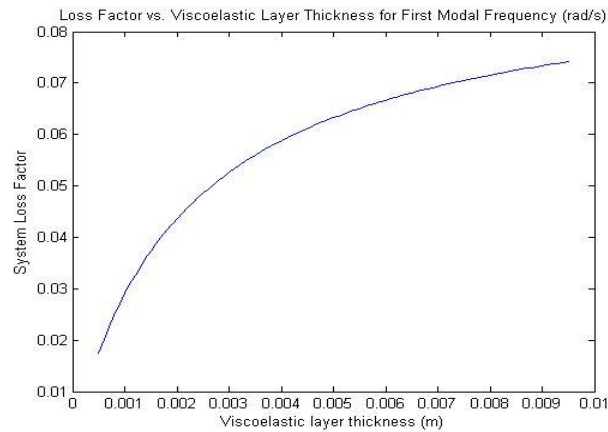


Fig: 11 Variation of FLD thickness with damping loss factor

Table 1: Result table

Parameters	Without FLD	With FLD
1 st Natural frequency	96 Hz	156 Hz
Damping loss factor	0.00015	0.07413

V. CONCLUSIONS

From the above experimentation we can conclude that new system is vibration free as there is considerable change in damping loss factor with FLD treatment. Auto-centering becomes flexible for wide range of pipe diameters up to 350mm and line speed up to 0.5m/s. auto-feeding mechanism becomes smoother than previous system hence efficiency and productivity of machines has been increased.

VI. REFERENCES

- [1] Andrzejflaga, jacekszulej, piotrwielgos, comparison of determination methods of vibration's damping coefficients for complex structures, budownictwoarchitektura 3 (2008) 53-61
- [2] Pravin hujure and anil sahstrabudhe, 'experimental verification of viscoelastic constrained layer damping', international conference on advances in manufacturing and materials engineering amme 2014.
- [3] Mohan d. Rao, recent applications of viscoelastic dampingfor noise control in automobiles and commercial airplanes, science direct, journal of sound and vibration 262 (2003) 457–474.
- [4] V. B. Bhandari, Design of Machine Element, 3rdEd. By McGraw Hill Education (India) Private Limited, 2015, pg.348
- [5] Reliability of rolled bars ultrasonic testing on the automated installation for ultrasonic testing volna-7,pg.2-3