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REVIEW ON DESIGN AND DEVELOPMENT OF DUAL PURPOSE GEAR DRIVE FOR BLISTER MACHINE APPLICATION

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ABSTRACT

Blister packaging machine are used by pharmaceutical industry for packing number of product like tablet, capsule, syringes or liquid material. Blister packs are perforated plastic packs useful for protecting product against external factors, such as humidity and contaminations for extended duration of time. Blister packs are portable, can help patients follow drug regimens, and can protect drugs over a long shelf life. Advocates cite several aspects in which blister packaging is better than conventional packaging, including product integrity, product production, tamper evidence, reduced possibility of accidental misuse, and patient compliance.

Blister machine consists of many units and subunits. Conveyor is used to carry blister strip in forward direction, on which various operations are performed. One of the operations is printing on the blister strips. It was found that after certain period of time, there was misprinting occurring on Blister strip. To overcome the problem of misprinting, a gear box was developed, which will be synchronized with the conveyor system. This paper present analytical method for designing bevel gears, using Lewis and Buckingham Equations. The developed gearbox perform dual function who's one output shaft is use to drive conveyor and other is connected to the printing unit. 3D model of gearbox is made using Catia V5. Static structural analysis of components like gear, shaft is performed using Ansys. Experimental setup will be developed to verify the motion is precise and well synchronized.

Keywords: Blister Machine, Blister Strip, Lewis and Buckingham Equations

I. INTRODUCTION

1.1 Problem Statement

Blister strip undergoes various process and operations within the Blister machine. Out of which one process is to drive the conveyor system, and other process is defined exclusively for printing batch code on the blister strip. The motor drive was provided as input power to drive the conveyor system using belt drive. And batch code printer was attached to same conveyor system near to the packaging unit of blister machine. Therefore the same belt drive was used to drive the roller of the batch printer. After certain period of time, sagging stretching of belt was observed in the conveyor system. Due to which idler pulley need to be repositioned frequently. These leads to problem of misprinting of Batch code on the blister packs. It results to inappropriate and incomplete information on the blister strip. To avoid it, the operator needs to be adjusting the idler pulley every time.

Therefore now we have arrived to the solution for the above problem by introducing synchronized and precise motion gear drive. This gear drive would serve dual purpose, one of its output shafts will drive the conveyor roller and other of its output shaft will drive the printer unit. By this sagging of belt will not have any effects on batch code printing, and both the process will carry out efficiently.

1.2 Objective

Our main objective is to overcome the problem of misprinting on the blister strip. This can be done by synchronizing two processes i.e. driving the conveyor and other printing process, using single gear drive. The batch printer will be relocated below the conveyor drive. So that slacking or stretching of belt will have no longer effects on the batch printing process. To accompany our objective we will design bevel gearbox having two output shafts. One of the output shafts will drive the roller of the conveyor and other output shaft will drive batch printer. Such that both the process are well synchronized. The input power is provide using servomotor of 0.3Kw. This motor will be coupled with input shaft of bevel gearbox to transmit required power at both of output shaft.

1.3 Methodology

The solution for the above problem was given by selecting dual purpose Gear Drive. The gear drive was selected due to reasons like,Gear drives are less noisy and responds well on different loading, whereas Chain drive worn, breaks easily on impact loading and its maintenance is difficult. In belt drive some adjustment of center distance or use of an idler pulley is necessary for wearing and stretching of belt drive compensation. Belt drive mechanism has significant uncertainties in determining torsion behavior as the belt stiffness tends to be nonlinear and highly depended on the belt tension. Therefore Gear drive was selected over belt or chain drives for transmitting synchronized and precise motion to printing unit of the blister machine.

The bevel gear box arrangement is made for driving conveyor roller and batch printer of the blister machine. Various design and analysis calculations are justified for the safe design. Mechanical design of components like gears, shafts is done using various theories of failure, selecting appropriate material. The selected components will be manufactured using various machine like lathe machine, Hobbing machine, milling machine, electrical arc welding machine etc.

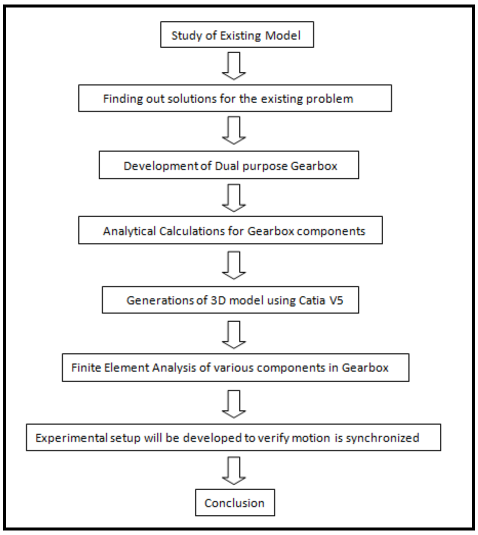


Figure 1: Work flowchart

II. LITERATURE REVIEW

In this paper, literature has been critically reviewed involving various studies carried out by various researchers related to the field of designing and analysis of gearbox. Gearbox is an important part of various applications like automobile, hoists, cranes and machines which is used for transmitting different or same torque, power, or speed ratio as per the application.

A. A. Pandharabale, Asst. Prof. A. J. Rajguru,[1] The main objective of their paper was to design a model of dual worm system for optimal load lifting capacity, optimal factor of safety and optimal efficiency for reduced power consumption. They have derived the optimal power for individual motor and select the motor for the application so as to make the device compact. The experimental validation part of the lifting force developed by the dual worm system is validated using test-rig. Various characteristics graph were plotted like Torque Vs. Speed, Power Vs. speed, Power consumption of motor under rated load, Efficiency of system Vs. speed. They concluded that the torque increases with the decrease in the output speed, Graph of power output indicates a rising trend up to certain output speed and then slightly drops indicating that indicating that the device will slow down slightly if the load is increased.

J. D. Chougule1, R. G. Todkar,[2] showed the influence of a cam shaft in the transmission system of a blister packaging machine used for packaging of tablets. It is observed that the cam shaft fails due fatigue loading into two pieces during operation. To find out cause of failure, a finite element analysis was carried out. Results of stress analysis reveal that the highest stressed area coincides with the fractured regions of the failure of the shaft. The theoretical stress fairly matches with the sub-model stress values. The failure analysis shows that the fatigue failure of the shaft is due to weak section at the step provided for cam shaft mount. To enhance service durability of the transmission system of Blister Packing Machine, stress concentration at cam step was modified and material with high service durability, mechanical characteristics such as fatigue strength, ultimate tensile strength, and fracture toughness was selected.

Ashish N. Taywade, Dr. V. G. Arajpure,[3] Their paper deals with the idea of gear designing and development for automobile application. Basically the driver and driven gear material plays vital role for the better performance in wear. Replacement of metal gears by plastic gear is continuing in automobiles, appliances and machinery due to its merits of low noise, less wear, self-lubrication, economic considerations, light weight, simple design and manufacturing. Due to thermal wear acetyl gear failed, so to overcome the problem of acetyl gear, nylon 66 material was selected. They concluded that, the nylon 66 is better option due to its superior properties and it also meet extreme performance challenges.

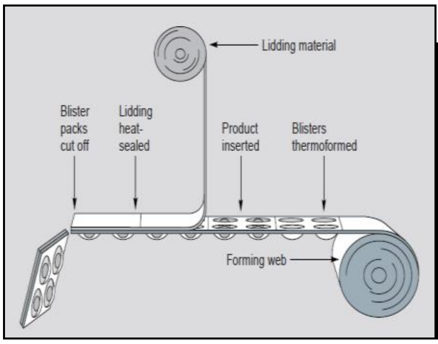
S.S. Kachare, R.M. Ghodake and V.D. Ghogare, [4] Have focus their work on Tracked vehicles which are equipped with Hydraulic drum type winch. The main function of winch is to carry out self-recovery of the vehicle, limited recovery of the other vehicles and assisting in amphibious operations. During the self-recovery of the vehicle it is proposed that the vehicle speed and the winch speed should be same, for this particular requirement constant speed winch is required. It was required to design and develop two stage dual planetary gear box for 8 tonne capacity winch which will drive the winch drum.The first stage gear pair is a helical and second stage is a planetary where sun is input and carrier will be output, when Ring gear is stationary and fixed to the drum. They emphasise on the fact that in gear design, surface contact strength and bending strength of the gears are assumed to be major contributors for the failure of the gear pair. Therefore bending stress and contact stress are determined using American Gear Manufacturing Association standard to make the design fail safe in bending and pitting failure. Their proposed design focus on reduction of weight and producing high accuracy gears.

M. B. Raut, Prof. S. L. Shinde, [5] they have focused their research work in planetary gears.The rim thickness of the annular gear is one of the important parameter of planetary gearbox to meet design objectives. An optimum rim thickness is required to meet the strength criterion at the same time to reduce mass of the gearbox and to save on material cost. Stresses obtained in their design were very less. Further reduction of the annular gear rim thickness was done without hampering the strength of annular gear.In this manner they concluded with optimum rim thickness of the rim for the gear.

III. PROJECT OVERVIEW

3.1 Blister Machinery Process

The sequence of Blister Machinery Process involves heating the plastic, thermoforming it into blister cavities, loading the blister with the product, placing lidding material over the blister, and heat-sealing the package. This can be a simple manual process, or it can be partially or fully automated. Although purchasing empty, preformed blisters and lidding material and then filling the product in a separate step is possible, this is rarely done. Instead, the package is created and filled on the same machine. [10]



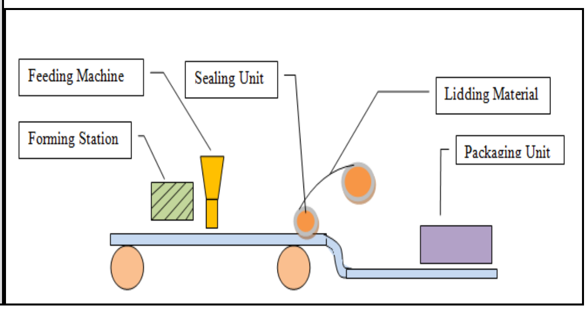


Figure 2: Blister Strip PackagingFigure 3: Detail Assembly of Blister Machine

3.1.1 Detail Assembly of Blister Machine

The essential parts and functions of an intermittently operating packaging machine include the following. The unwinding station:The unwinding station supplies the forming films and thelidding material at a rate corresponding to the speed of the packaging machine (seeFigure 4)The heating station:The heating station raises the temperature of the plastic formingfilms to a level suitable for deep drawing. Forming films containing the polyvinyl chloride (PVC) support material are heated to 120–140 C. Polypropylene (PP) forming films are heated to 140–150 C. Forming films containing aluminium are not heated before the forming process.

The forming station:The forming station forms the plastic blister cavities via compressed air or die plates. Films containing aluminium are formed with mechanicalforming tools only.The feeding machine:The loading area fills the blister cavities with product. The feeding machine can be linked, or the product to be packaged can simply be sweptinto the blisters.The sealing station:The sealing station heat seals the lidding material to the formingfilm that contains the product. All heat-sealing methods mate the blister and lid under constant pressure for a specified time, during which heat issupplied. The mating surfaces fuse and bond, setting almost instantaneously whenheat input stops. Depending on the type of machine, the sealing temperature typicallyranges between 140 and 340 8C.Labelling through packaging:Packages are labelled, notched, and then marked with abatch number at the coding station. The perforating device makes a cross-shaped perforation along the sealing seams. At the punching station, the packages are then separated into sheets that typically contain from 10 to 20 individual blisters. The vision system checks the filled packages for defects. Finally, a multipacking machine packs the individual packages into bigger cartons.Printing process: These simple & low cost coders are ideal for continuous coding on various packaging machines.The message to be printed is very easily composed by just assembling the stereos (easily changeable) on to the print drum. The Inking system consists of a rechargeable circular cartridge, which is fully enclosed allowing the use of fast-drying & indelible solvent-based inks for porous and non-porous surfaces.The whole assembly is directly mounted on the packaging machine with the supplied bracket & as the film passes over the roller of the bracket, it gets printed continuously. [10]

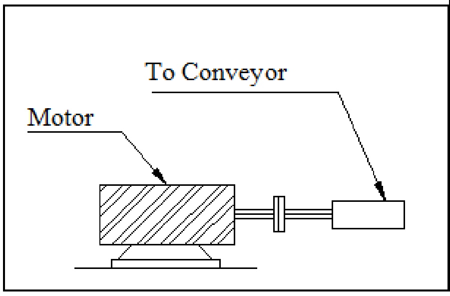


Figure 4: Existing Setup on Blister Machine

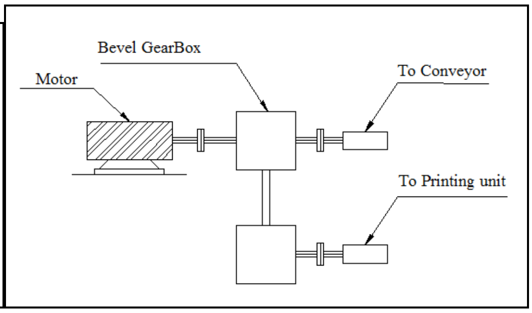


Figure 5: Proposed Setup on Blister Machine

IV . DESIGN CALCULATIONS

Gear design is based on Lewis and Buckingham Equations. [9]

Material Specifications = En 34

Material Strength = S_b = 330 N/mm2

No. Of teeth on pinion, Z1 = 28

No. Of teeth on Gear, Z2 = 28

Gear Ratio, i = Z2/Z1 = 1

Module, m = 2.5

Pressure angle, $\varphi = 20$

Pitch diameter, d = m * Z1

d = 70mm

Pitch cone angle, δ = tan-1(Z2/Z1)

= 0.7853 rad

Cone distance, R = d/(2 sin δ)

= 49.497 mm

Face Width, b = R/3 = 16.50 mm

Addendum, ha = 1 * m = 1 * 2.5

ha = 2.5 mm

Deddendum, hf = 1.16 * m

= 1.16 * 2.5

hf = 2.917 mm

Virtual number of teeths,

Zv = Z/ cos δ

Zv = 39.598

Lewis form factor,

Y = π* (0.154 – 0.912/ Zv)

= π * (0.154 – 0.912/ 39.598)= 0.41

Velocity,

V = π d N / (60 * 1000)

= (π * 70 * 36) / (60 * 1000)

= 0.132 m/sec

Load carrying capacity of gear tooth or strength of gear tooth is given by, [9]

Fs $=\frac{Sb * b * Y}{Pd}(1-b/R)$

Fs $=\frac{330 * 16.50 * 0.411}{0.4}(1-16.50/49.497)$

Fs= 3733.716 N

Buckingham's Dynamic load on tooth per unit area is given by, [9]

F_d = Cv * Nsf * Km * Ft

Now, N_{sf} depends on service factors.

Now, We want to transmit Power of 0.3 Kw using gear drive. Therefore Force transmitted, Ft can be calculated as follow.

F_t = P * (75/V)

= 0.3 * (75/ 0.132)

Ft = 170.45 N

Driven Machinery			
Source of power	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

Table No 3: Service Factor Coefficient

We assume driven machine is uniform in nature, therefore N_{sf} = 1

Mounting type	Mounting rigidity
Both gears are straddle-mounted	1.0 to 1.25
One gear straddle-mounted; the other overhung	1.1 to 1.4
Both gear overhung	1.25 to 1.5

Table No 4: Mounting Factor K_m for Bevel Gears

According to our assembly of bevel gearbox we select Km = 1.25

Now For straight bevel tooth Velocity factor, Cv is given by

Cv $=\frac{3+V}{3}$

$=\frac{3+0.132}{3}$

Substituting all the values in equation of Stress occurring due to Buckingham's Dynamic load on tooth per unit area, we get

Fd = Cv * Nsf * Km * Ft

Fd = 1.044 * 1 * 1.25 * 170.45

Fd = 222.43 N

According to Lewis and Buckingham Equations, dynamic load on the tooth is less than the strength of the tooth of gear, i.e. Fd<Fs, Our gear is safe in design.

P = Fs * V / Cv

= (3733.716 * 0.132) / 1.044

P = 471.89 W

Torque transmitted at Output shaft is given by

T = (P * 60) / (2 π N)

= (471.89 * 60) / (2* π * 36)

T = 125.17 Nm

B. Design of Shaft according to ASME code [1]

Material Specification = EN 9

Ultimate Tensile strength = 650 N/mm2

Yield Strength = 480 N/mm2

σs max = 0.18 * σ ult

= 117 N/mm2

σs max = 0.30 * σ yld

= 144 N/mm2

Considering minimum of above values,

σs max = 117 N/mm2

Shaft is provided with keyway, which will reduce the strength. Hence reducing the above value by 25 %

σs all = 87.75 N/mm2

This is allowable shear stress that can be induced in the shaft material.

T = 125 * 10^3 N-mm

Assuming 20 % of overload.

T design = 1.20 * 125 * 10^3

T design = 150 * 10^3

Now the lowest diameter on output shaft is 50mm, we check Torsional shear failure of shaft,

Td * σs act * d3

σs act = (16 * 150 * 10^3) / (π * 50^3)

σs act = 6.11 N/mm2

As, σs act <σs all; Shaft is safe in Torsional load.

Now the lowest diameter on intermediate shaft is 40mm, we check Torsional shear failure of shaft,

Td * σs act * d^3

σs act = (16 * 150 * 10^3) / (π * 40^3)

σs act = 11.93 N/mm2

As, σs act <σs all; Shaft is safe in Torsional load.

Now the lowest diameter on second output shaft is 45mm, we check Torsional shear failure of shaft,

Td * σs act * d^3

σs act = (16 * 150 * 10^3) / (π * 60^3)

σs act = 8.38 N/mm2

As, σs act <σs all; Shaft is safe in Torsional load.

Static structural analysis of gears, shafts, using Finite Element Method software was done under specific boundary condition. It was found that Gears and shafts are within the safe limit under the loading conditions.

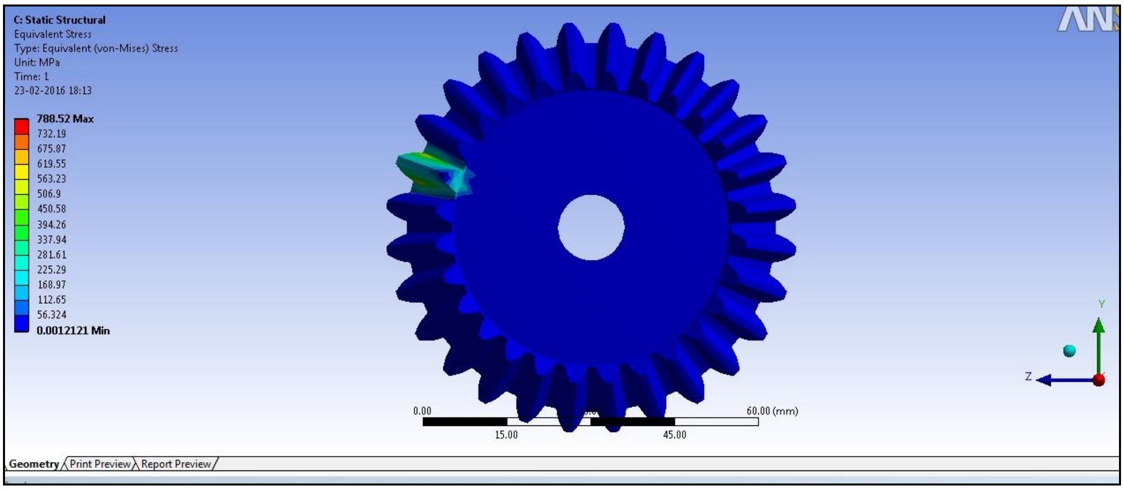


Figure no 6: Static structural analysis of Gear

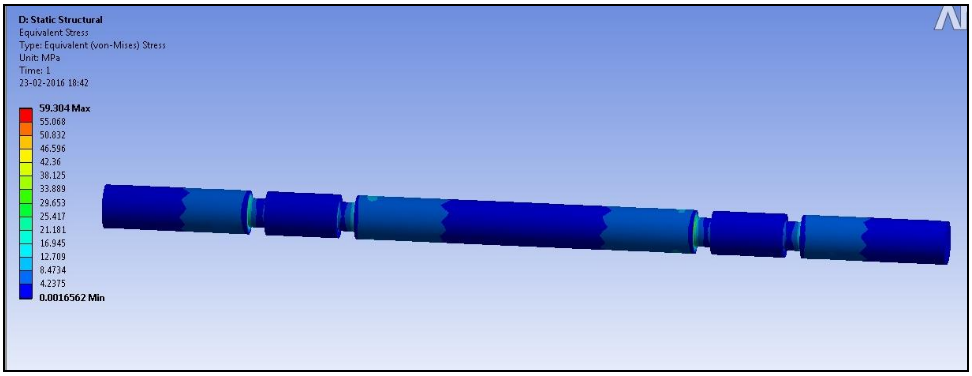


Figure no 7: Static structural analysis of Input shaft

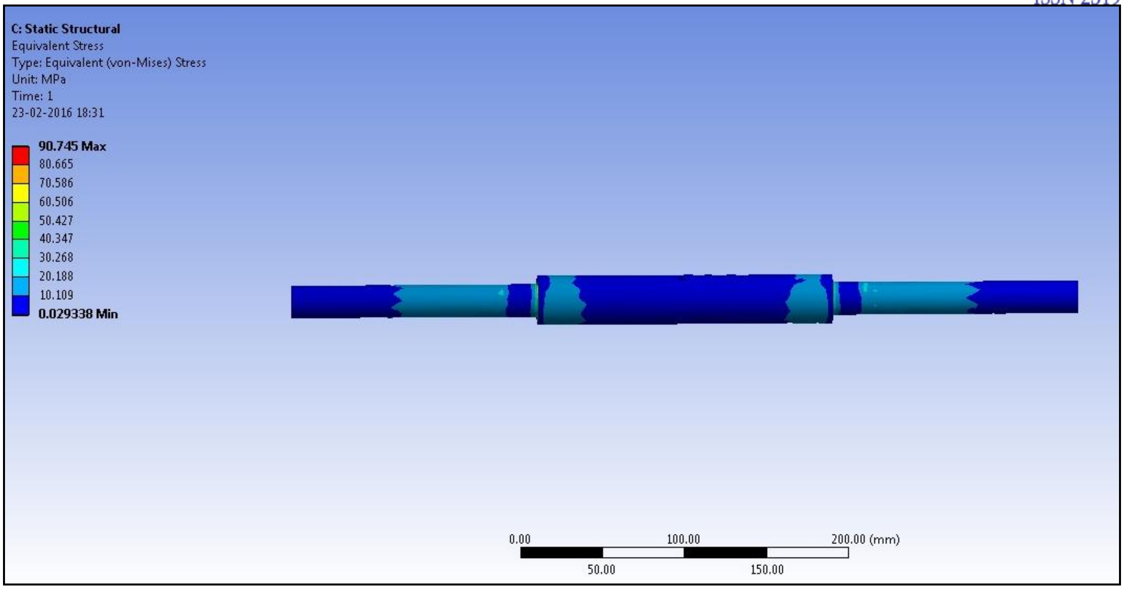


Figure no 8: Static structural analysis of Intermediate shaft

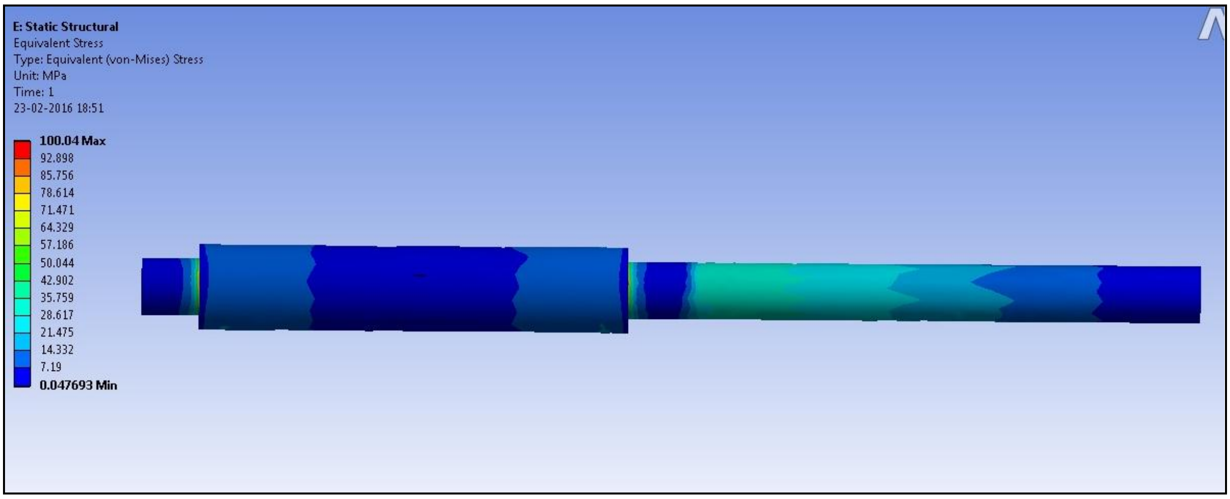


Figure no 9: Static structural analysis of Output shaft

V. CONCLUSION

The dual purpose gearbox has been designed to transmit torque of 125 Nm at the output shafts, such that conveyor driving process and batch printing process are well synchronized. Analytical calculationsfor designing bevel gears, is performed using Lewis and Buckingham Equations. Analytical calculations for component within the gearbox like shafts are checked against the limit of stress developed within them and found safe for operations. The developed gearbox perform dual function who's one output shaft is use to drive conveyor and other is connected to the printing unit.

5.1 Future Scope of Work

Experiment test rig will be made to validate the motion for the process is well precise and well synchronized. This will overcome the problem of misprinting or incomplete printing of batch code on the blister strip.

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