# STRUCTURAL ANALYSIS OF ROLLER GEAR FOR SEMIAUTOMATIC PORTABLE NOODLES-MAKING MACHINE

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**Abstract** - This paper focuses on the design and structural analysis of roller gear for a semiautomatic portable noodles-making machine by changing roller gear face widths and three different materials (ASTM A 536, ASTM A 220 and AISI 1020). Gear corrosion occurs at contact points as a result of bending stress and contact stress. This is the major source of the gear failure of the semiautomatic portable noodles-making machine. This whole mechanism will be driven by a 180W capacitor-start induction motor according to the motor power selection. Pitch diameters of 50mm and module of 5mm spur gear is selected in the design of roller gear. In theoretical analysis, the Lewis bending stress equation is used. In the numerical analysis,the geometry is created by SolidWorks software. The minimum von-Mises stress and effective strain are found on AISI 1020 material by using ANSYS 17.0 software. In this paper, von-Mises stress and effective strainare analyzed by changing the face widths of roller gears to 8mm, 10mm, 12mm, 14mm and 16mm and using finite element analysis (FEA). Although all Face widths are safe for this design, 12mm is chosen in this paper due to power consumption and stress points of view.

**Keywords** - Noodles Roller Gears, Lewis Bending Stress, AGMA Contact Stress, Changing Face Width, FEA.

## I. INTRODUCTION

All of us enjoy eating noodles, considering them healthy and delicious fast food. Noodles are as taple food consumed in many cultures and are basically made from flours made from grains, salt, and water[6]. Instant noodles have low calories, protein, fiber, mineral and vitamins.

In the modernera, the designers have used machine to make noodles instead of making them by hand. There are many types of machines for making noodles, generally semiautomatic and fully-Automatic.

Semiautomatic noodles making machines are used for family and small unit enterprise. Proposed noodle machine with compact design, portable noodles making machine operated by the single person.

The main components of portable noodles making a chines are noodle cutting shaft, noodle rollers, noodle roller gears, noodle cutter gears and shafts.

There are one pair of rollers, converts the mixture off lour and water to sheet and the lower pair has noodle cutting shafts[6]. A flour container is a steel material cap used to add a flour and liquid mixture during the rolling process. The rollers are arranged in pairs and are hollow, allowing for a light weight.

The dough is sheeted by the same diameter of two rollers with rotational motion. There is a driving cutter shaft and a driven cutter shaft.These two shafts are fixed to a sliding sleeve that is supported by a frame.This machine employs spur pinions and gears to achieve a specific speed and output capacity.

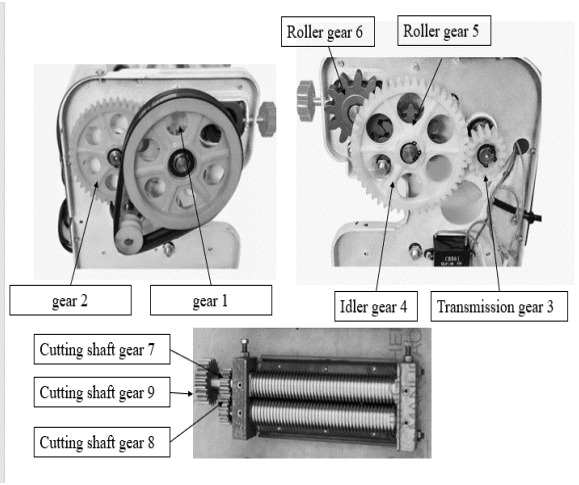



**Fig.1. Side view of semiautomatic noodles-making machine**

## II. METHODOLOGY

There are three steps in the structural analysis of the spur gear for the semiautomatic noodles-making machine. They are (i) motor power selection and design of roller gear, (ii) theoretical analysis of von-Mises stress andeffective strain of roller gear, (iii) numerical analysis of roller gear by changing materials and face widths.

### 2.1. Motor Power Selection

The motor power selection is important for the design of roller gear. The motor speed (N) is 1330rpm, motor pulleydiameter is 35mm and machine pulley diameter is 123mm.After measuring the diameters of pulleys and tip diameters of gears, the speed of pulley and gears are calculated according tothis velocity ratio equation,

$$VR=\frac{T_2}{T_1}=\frac{D_2}{D_1}=\frac{N_1}{N_2} \quad (1)$$

So, speed of machine pulley is 379 rpm.
The required power equation is,[1]

$$P=\frac{2\pi NT}{60} \quad (2)$$

The motor starting torque (T) is as high as 350 to 400 per cent of full load torque ($T_f$).[9]
Full load torque, $T_f=I_{total}\times\alpha$ (3)
Where, α is angular acceleration, $\frac{\omega}{t}=\frac{2\pi N}{60t}(rad/s^2)$

The mass of pulley 1 and pulley 2 are 0.025kg and 0.139kg. The mass of cutting shaft 1 and cutting shaft 2 are 0.56kg and 0.54kg. And then, the mass of roller 1 and roller 2 are 0.02kg and 0.025kg. These data have to be known to design the roller gear and motor power selection. Moreover, Table 1 shows the specification data for all spur gears.

| | Tip diameter (mm) | Teeth | Mass (kg) |
|---|---|---|---|
| Gear 1 | 37 | 10 | 0.0062 |
| Gear 2 | 121 | 39 | 0.095 |
| Transmission gear 3 | 42 | 10 | 0.08 |
| Idler gear 4 | 119 | 32 | 0.125 |
| Roller gear 5 | 60 | 10 | 0.1664 |
| Roller gear 6 | 60 | 10 | 0.1664 |
| Cutting shaft gear 7 | 30 | 10 | 0.0459 |
| Cutting shaft gear 8 | 30 | 10 | 0.0459 |
| Cutting shaft gear 9 | 58 | 14 | 0.15 |

**Table1: Specification Datafor Spur Gears**

The inertia of pulley 1 and 2,

$$I_{p1}=\frac{mr_{p1}^2}{2} \quad (4)$$

$$I_{p2}=\frac{mr_{p2}^2}{2} \quad (5)$$

The inertia of cutting shaft 1 and 2,

$$I_{cs1}=\frac{mr_{cs1}^2}{2} \quad (6)$$

$$I_{cs2}=\frac{mr_{cs2}^2}{2} \quad (7)$$

The inertia of roller A and B,[8]

$$I_{roller\ A}=\frac{1}{8}M_A(D^2+d^2) \quad (8)$$

$$I_{roller\ B}=\frac{1}{8}M_B(D^2+d^2) \quad (9)$$

$$I_{roller\ A,B}=I_{roller\ A}+\left(\frac{D_A}{D_B}\right)^2 I_{roller\ B}+\frac{1}{4}mD_A^2 \text{ [7] } (10)$$

Where, m is equivalent mass of the dough.
The inertia of spur gearpair,

$$I_{gear\ 1,2}=M_1\frac{r_1^2}{2}+M_2\frac{r_2^2}{2u^2} \quad (11)$$

Where, u is gear ratio.
Total inertia,

$$I_{total}=I_{p1}+I_{p2}+I_{cs1}+I_{cs2}+I_{roller\ A,B}+I_{roller\ gear\ 5,6}+I_{cs\ gear\ 7,8}+I_{gear\ 1,2}+I_{gear\ 3,4}+I_{gear\ 9,4} \quad (12)$$

| | | | |
|---|---|---|---|
| Inertia of pulley 1 | $I_{p1}$ | $3.828\times10^{-6}$ | $kgm^2$ |
| Inertia of pulley 1 | $I_{p2}$ | $2.628\times10^{-4}$ | $kgm^2$ |
| Inertia of cutting shaft 1 | $I_{cs1}$ | $1.372\times10^{-5}$ | $kgm^2$ |
| Inertia of cutting shaft 2 | $I_{cs2}$ | $1.323\times10^{-5}$ | $kgm^2$ |
| Inertia of roller A, B | $I_{roller\ A,B}$ | $1.347\times10^{-3}$ | $kgm^2$ |
| Inertia of roller gear 5, 6 | $I_{rg\ 5,6}$ | $9.6875\times10^{-5}$ | $kgm^2$ |
| Inertia of cutting shaft gear 7, 8 | $I_{cs\ gear\ 7,8}$ | $4.6875\times10^{-6}$ | $kgm^2$ |
| Inertia of gear 1, 2 | $I_{gear\ 1,2}$ | $4.409\times10^{-5}$ | $kgm^2$ |
| Inertia of gear 3, 4 | $I_{gear\ 3,4}$ | $1.256\times10^{-4}$ | $kgm^2$ |
| Inertia of gear 9, 4 | $I_{gear\ 4,9}$ | $3.3014\times10^{-4}$ | $kgm^2$ |
| Total inertia | $I_{Total}$ | $2.242\times10^{-3}$ | $kgm^2$ |

**Table 2: Result Data of Total Inertia**

According to the equation 3 and Table 2, full load torque ($T_f$) is 0.3123 Nm.

Starting torque, $T=4\times T_f=1.249$ Nm[9]

The required power is,

$$P=\frac{2\pi NT}{60}=\frac{2\pi\times1330\times1.249}{60}=173.96\ W$$

So, 180W, 220V, 50Hz and 12μF single phase capacitor start induction motor is chosen for semiautomatic noodles-making machines.

### 2.2. Design Consideration of Roller Gear

Firstly, velocity ratio is the ratio between the velocity of driver and driven gear. They can becalculated by using equation 1. The pitch diameter of all gears can be calculated by equation 13.
Pitch diameter, $D_p=T_g m$ (13)

Where, $T_g$ = number of teeth of gear
m = module

Torque transmitted by gear is,

$$T=\frac{W_t D_p}{2} \quad (14)$$

Where, $W_t$ is actual tangential tooth load.
Circular pitch,

$$p=\pi m=\frac{\pi}{\text{diametral pitch}} \quad (15)$$

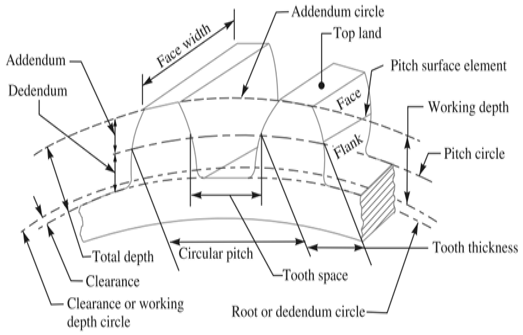



**Fig.2. Teeth of Spur Gear[10]**

| | Pitch diameter (mm) | Module (mm) | Speed (rpm) |
|---|---|---|---|
| Gear 1 | 30 | 3 | 379 |
| Gear 2 | 115 | 3 | 97 |
| Transmission gear 3 | 35 | 3.5 | 97 |
| Idler gear 4 | 112 | 3.5 | 30 |
| Roller gear 5 | 50 | 5 | 30 |
| Roller gear 6 | 50 | 5 | 30 |
| Cutting shaft gear 7 | 25 | 2.5 | 69 |
| Cutting shaft gear 8 | 25 | 2.5 | 69 |
| Cutting shaft gear 9 | 49 | 3.5 | 69 |

**Table3: Results of All Gears**

Therefore, the results of roller gear are the following:
Pitch diameter, $D_p$ = 50 mm
Module, m = 5 mm
Diametral pitch, P = 0.2 mm
Circular pitch, p =15.7 mm
Tooth thickness, t = 7.8 mm

### 2.2.1. Strength Check of Roller Gear

The tangential force ($W_t$) is the driving force of the gear mesh. According to the Lewis's equation, the permissible tangential load is,

$$W_{T(per)}=\sigma_w b\pi m Y \quad (16)$$

According to the Barth formular, the permissible working stress is,

$$\sigma_w = \sigma_0 \times c_v \quad (17)$$

Where, ($\sigma_0$)is allowable static stress 56MPa for cast iron ordinary and ($c_v$)is velocity factorfor ordinary cut gears operating at velocities up to 12.5 m/s.[10]

$$c_v=\frac{3}{3+v} (18)$$

Where, pitch line velocity, $v=\frac{\pi DN}{60}$

The value of (y) in terms of the number of teeth may be expressed as follows:

$$y=0.124-\frac{0.684}{T} \text{ for 1 14.5 °/ composite and full depthinvolute system}$$

$$y=0.154-\frac{0.912}{T} \text{ for 20° full depth involute system}$$

$$y=0.175-\frac{0.841}{T} \text{ for 20° stub system[10]}$$

So, Lewis's form factor is Y=πy = 0.176

| Materials | Cast Iron (ASTM A 536) |
|---|---|
| Actual tangential tooth load, $W_{T\,act}$ (N) | 49.96 |
| Permissible tangential tooth load, $W_{T\,per}$ (N) | 1810.44 |
| $\mathbf{W_{T\,per} > W_{T\,act}}$ Design is safe | |

**Table4: Results of Strength Check for Roller Gear**

### 2.2.2. Dynamic Check of Roller Gear

The static tooth load ($W_s$) also called beam strength or endurance strength of the toothis obtained by Lewis's formula by substituting flexural endurance limit or elastic limit stress ($\sigma_e$) 84 MPa for cast iron.[9]

$$W_s=\sigma_e b\pi m Y \quad (19)$$

The maximum or the limiting load for satisfactory wear of gear teeth, is obtained by using the following Buckingham equation,

$$W_w=D_p bQK \quad (20)$$

The dynamic tooth load ($W_D$) on the tooth by using Buckingham equation,

$$W_D=W_I+W_T \quad (21)$$

Where, Ratio factor, $Q=\frac{2VR}{VR+1}$ (22)

Load stress factor, $K=\frac{\sigma_{es}^2 \sin\emptyset}{1.4}\left(\frac{1}{E_p}+\frac{1}{E_g}\right)$(23)

Increment load due to dynamic action, $W_I$

$$W_I=\frac{21V(bC+W_T)}{21V+\sqrt{bC+W_T}} (24)$$

| Materials | Cast Iron (ASTM A 536) |
|---|---|
| Static tooth load, $W_s$ (N) | 2786.6 |
| Increment load, , $W_I$ (N) | 41.37 |
| Wear tooth load, $W_w$ (N) | 516.23 |
| Dynamic tooth load, $W_D$ (N) | 91.33 |
| $\mathbf{W_s, W_w > W_D}$ Design is safe | |

**Table5: Results of Dynamic Check for Roller Gear**

## III. STRUCTURAL ANALYSIS OF ROLLER GEAR BY THEORETICAL

Gear teeth are considered as a cantileverbeam with a tangential tooth load, $W_t$, applied at the tip. The radial component is negligible. The load is distributed uniformly across the full-face width. Forces due to

tooth sliding friction are negligible. ASTM A 536, ASTM A 220, and AISI 1020 materials are choose for this research[3].The mechanical properties of these materials are:

| Materials | Density (ρ), kg/m$^3$ | Yield Strength ($S_y$), MPa | Tensile Strength ($S_{ut}$), MPa | Young's Modulus, GPa | Poisson's ratio |
|---|---|---|---|---|---|
| Cast Iron (ASTM A 536) | 7200 | 379 | 552 | 165 | 0.23 |
| Cast Iron (ASTM A 220) | 7400 | 276 | 414 | 179 | 0.23 |
| Carbon Steel (AISI 1020) | 7680 | 352 | 420 | 207 | 0.27 |

**Table6: Material Properties of Roller Gear [4]**

The stress concentration in the tooth fillet is negligible. The bending stress can be calculated by the following equation.

$$\sigma_b = \frac{W_T}{b\pi my} \quad (25)$$

Principal stresses,

$$\sigma_{1,2} = \frac{1}{2}(\sigma_y+\sigma_x) \pm \frac{1}{2}\sqrt{(\sigma_y+\sigma_x)^2+4\tau_{xy}^2} \quad (26)$$

von-Mises Stress or Effective Stress,

$$\bar{\sigma} = \frac{1}{\sqrt{2}}[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2]^{\frac{1}{2}} \quad (27)$$

Effective strain,

$$\bar{\varepsilon} = \left[\frac{2}{3}(\varepsilon_1^2+\varepsilon_2^2+\varepsilon_3^2)\right]^{\frac{1}{2}} \quad (28)$$

First principal strain, $\varepsilon_1 = \frac{1}{E}[\sigma_1-\upsilon(\sigma_2+\sigma_3)]$ (29)

Second principal strain, $\varepsilon_2 = \frac{1}{E}[\sigma_2-\upsilon(\sigma_1+\sigma_3)]$ (30)

Third principal strain, $\varepsilon_3 = \frac{1}{E}[\sigma_3-\upsilon(\sigma_1+\sigma_2)]$ (31)

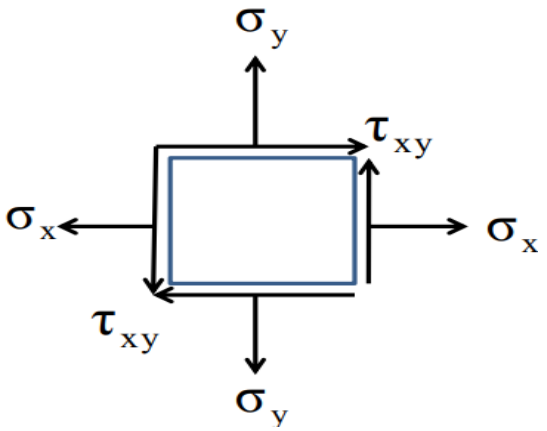



**Fig.3. Stress in XY-Plane**

| Parameter | Symbol | Value | Unit |
|---|---|---|---|
| Lewi's bending stress | $\sigma_b$ | 4.731 | MPa |
| Shear stress | $\tau_{xy}$ | 0.0509 | MPa |
| von-Mises stress | $\bar{\sigma}$ | 4.732 | MPa |
| Effective strain | $\bar{\varepsilon}$ | $2.4620\times10^{-5}$ | - |

**Table7: Theoretical Effective Stress and Strain Results for ASTM A 536**

## IV. NUMERICAL ANALYSIS OF ROLLER GEAR

In the numerical analysis, structural behaviors (von-Mises stress and effective strain) of roller gear are analyzed by changing three different materials using ANSYS software. The solid model of roller gear is created by SolidWorks software. The 3D model of roller gear is as shown in Fig.4.

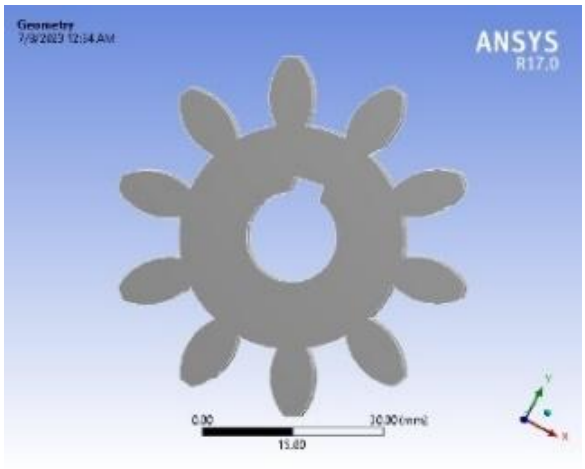



**Fig.4. Geometry of Roller Gear in XY-Plane**

Fig.5.shows the generated mesh of roller gear. This geometry model is meshed with high smoothing. The generated mesh is done by fine position to obtain the good quality of mesh.

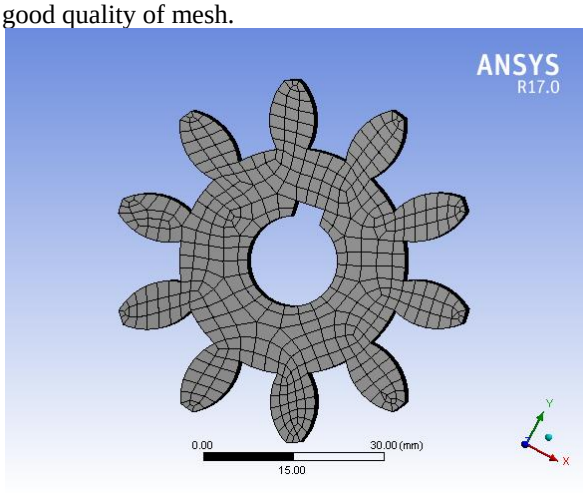



**Fig.5. Meshing of Roller Gear**

The boundary condition of the roller gear is shown in Fig.6.Firstly, fixed support is at the hub of the spur gear, the actual transmitted tooth load (Wt), the

magnitude of 49.96 N was applied with X-direction at surface of the spur gear tooth, the turning moment is 1249Nmm at the upper surface of the roller gear.

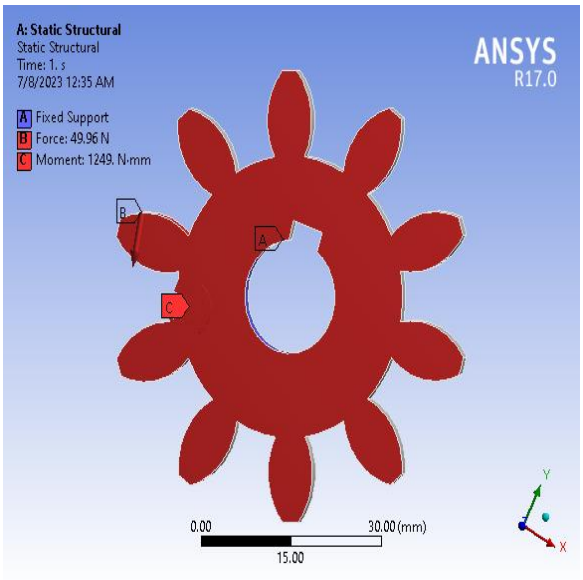



**Fig.6. Boundary Condition of Roller Gear**

After finishing set up the boundary conditions on the roller gear, run the solution and get the results for the structural behaviors of three different materials.

In the numerical analysis of 12mm face width roller gear using ASTM A 536 material,the maximum von-Mises stress of roller gear is 4.2213 MPa and maximum effective strain is 2.9651× $10^{-5}$.Fig.7. and Fig.8. show the equivalent (von-Mises) stress and effective strain of 12mm face width roller gear using ASTM A 536 material.

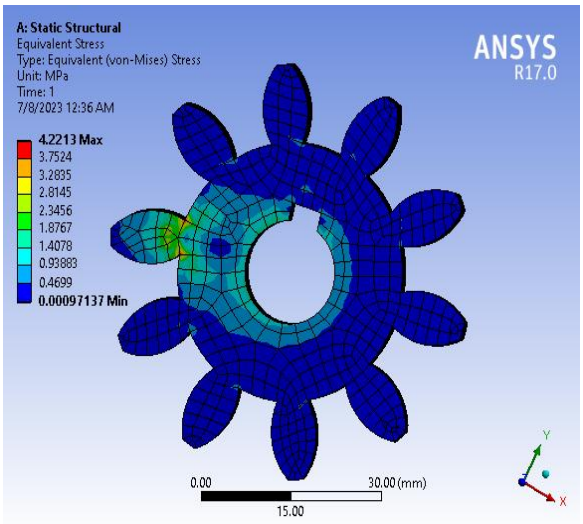



**Fig.7. Numerical result of von-Mises stress using ASTM A 536 material**

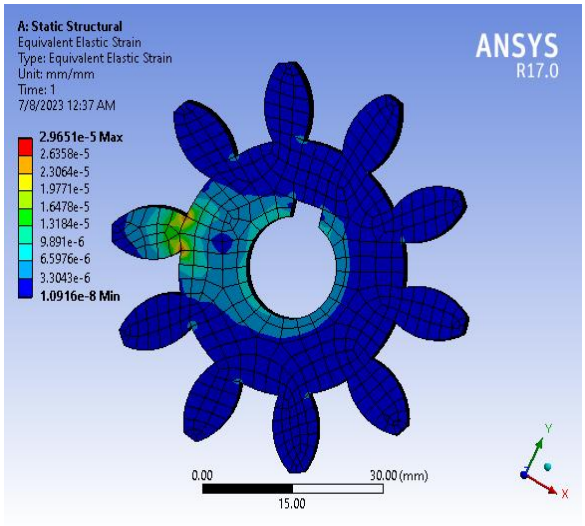



**Fig.8. Numerical result of effective strain using ASTM A 536 material**

In the numerical analysis of 12mm face width roller gear using ASTM A 220 material,the maximum von-Mises stress of roller gear is 4.2213 MPa and maximum effective strain is 2.7332× $10^{-5}$.

In the numerical analysis of 12mm face width roller gear using AISI 1020 material,the maximum von-Mises stress of roller gear is 4.1342 MPa and maximum effective strain is 2.3111× $10^{-5}$.

### 4.1. Comparison of Theoretical and Numerical Analysis

Table 8shows the comparison of theoretical and numerical results data of von-Mises stress and effective strain of roller gear (12 mm face width) for three different materials.

| Material | Von-Misesstress (MPa) | | EffectiveStrain $10^{-5}$ | |
|---|---|---|---|---|
| | Theoretical | Numerical | Theoretical | Numerical |
| ASTM A536 | 4.732 | 4.2213 | 2.4620 | 2.9651 |
| ASTM A220 | | 4.2213 | 2.2697 | 2.7322 |
| AISI1020 | | 4.1342 | 1.9978 | 2.3111 |

**Table8: Comparison of Theoretical and NumericalResults with Three Different Materials (12mm face width)**

According to the results Table 8, theoretical results data of von-Mises stress (4.732 MPa) for thesethree different materials are the same. Numerical results data of von-Mises stress and strain for three different materials are nearly equal. Fig.9. shows the comparison of von-Mises stressin theoretical and numerical results for three different materials.

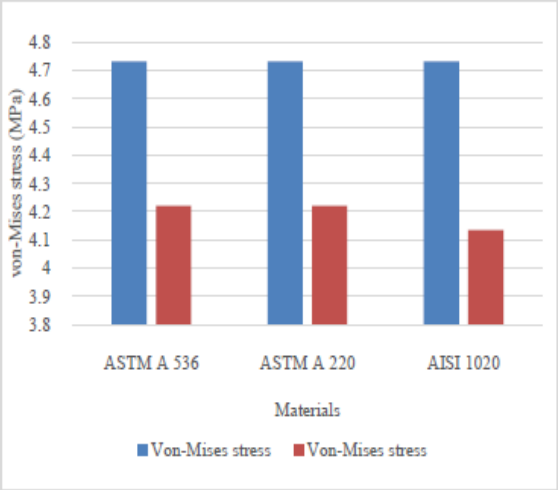



**Fig.9. Comparison of von-Mises stress in theoretical and numerical results for three different materials.**

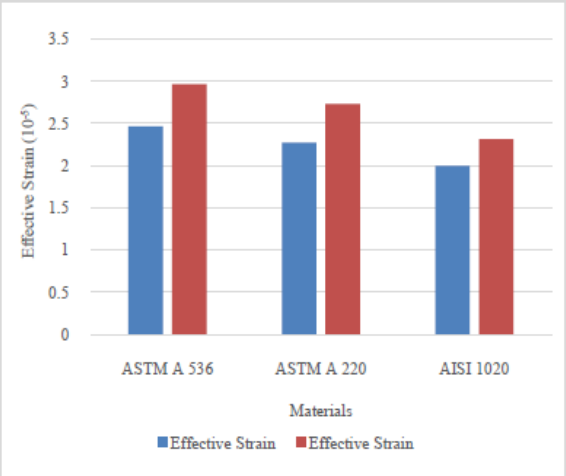



**Fig.10. Comparison of effective strain in theoretical and numerical results for three different materials.**

So, 12mm face width of roller gear design is satisfied for all these three different materials. In the comparison of results for three different materials, the results of von-Mises stress and effective strain of roller gear for AISI 1020 material are less than the ASTM A 536 and ASTM A 220 materials. So, AISI 1020 material is more suitable material of roller gear than the other two types of materials.

### 4.2. Structural Analysis of AISI 1020 Material Roller Gear with Different Face Widths

The roller gear is numerically analyzed by changing different face widths (8mm, 10mm, 12mm, 14mm and 16mm) in AISI 1020 material.

The models of the roller gear with different face widths are drawn using SolidWorks software and added to the geometry in ANSYS software. The numerical results of equivalent (von-Mises) stress and effective strain are shown in the Fig.11 andFig.12.

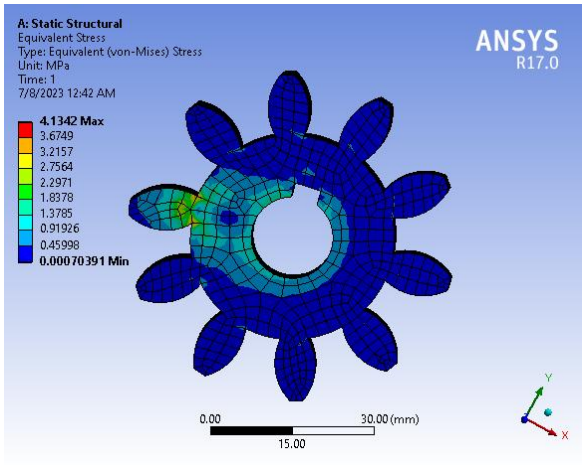



**Fig.11. Numerical result of von-Mises stress using AISI 1020 material (Face width 12mm)**

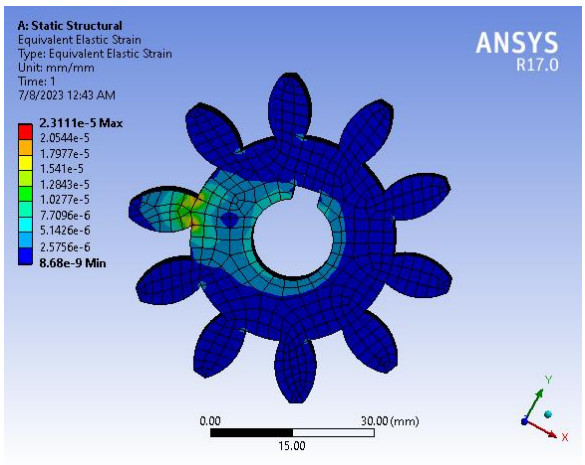



**Fig.12. Numerical result of effective strain using AISI 1020 material (Face width 12mm)**

### 4.3. Comparisonof AISI 1020 Material Roller Gear with Different Face Widths

Table 9 shows the comparison of numerical results of von-Mises stress and strain with different face widths.The maximum valuesare found in 8mm face width. The minimum value arefound in 16mm face width.

| Face Width (mm) | Equivalent Von-Mises Stress (MPa) | Equivalent Elastic Strain ($10^{-5}$) |
|---|---|---|
| 8 | 7.2482 | 4.1771 |
| 10 | 5.7571 | 3.3211 |
| 12 | 4.1342 | 2.3111 |
| 14 | 3.5186 | 2.0154 |
| 16 | 3.08 | 1.7638 |

**Table 9: Comparison of NumericalResults with Different Face Widths**

According to the Table 10, the numerical results of von-Mises stress and strain with different face widths are shown in Fig.13. and Fig.14.

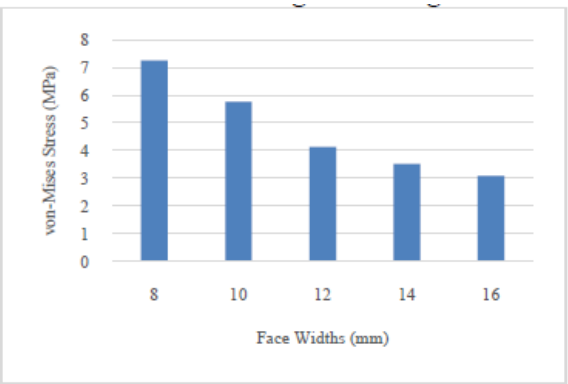



**Fig.13. Comparison of Numerical von-Mises stress with different facewidths**

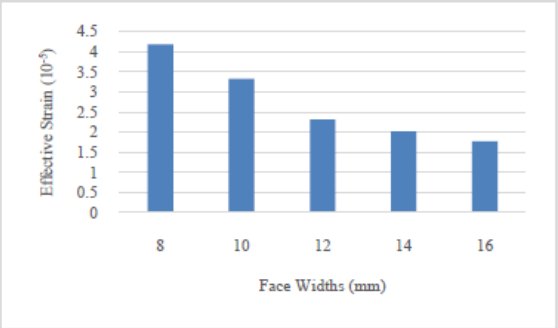



**Fig.14. Comparison of Numerical effective strain with different face widths**

## V. CONCLUSION

The power of motor is selected. A single-phase 180W capacitor-start inductionmotor is usedfor the semiautomatic potable noodles-making machine and design of roller gear are expressed by changingthree different materials (ASTM A 536, ASTM A220, AISI 1020). The module 5mm and pitch diameter 50mm spur gear is used for this machine.The strength and dynamic check of roller gear are calculated and design is satisfied.According to the results, the minimum von-Mises stress of roller gear is 4.1342 MPa and minimum effective strain is $2.3111\times 10^{-5}$ are found in AISI 1020 material. So, AISI 1020 carbon steel is chosen for roller gear. Moreover, it can be seen that the greater the face width, the smaller the von-Mises stress. And the smaller the face width, the greater the von-Mises stress and effective strain. So, the moderate value of face width 12mm is selected by considering power consumption and stress point of view.

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## REFERENCES

[1] Prof. V. S. Nikam, Mr. Nilesh Bule, Mr. Pratik Konde, Mr. Nikhil Ghongade, Mr. Zuber Sheikh, Design & Fabrication of Axial Noodle Making Machine, International Journal of Scientific Research & Engineering Trends Volume 7, Issue 4, July-Aug-2021.

[2] He Wei, Li Wanxianand Zhai Chaoka, Lanzhou Beef Noodles And Mechanical Process of Dough Machine Extrusion, 5th International Conference on Technologies in Manufacturing, Information And Computing (ICTMIC) 2020.

[3] Olutomilola Emmanuel Olatunji, Modified Design of a Selected Domestic Chinchin Cutting Machine, International Research Journal of Engineering and Technology (IRJET), 2018.

[4] Robert L. Mott, Edward M. Vavrek, Jyhwen Wang, Machine Elements in Mechanical Design, Sixth Edition, 2018.

[5] Bharathraj M, Murali Kumar L, Fabrication of portable noodle making machine, International Journal of Engineering and Applied Sciences (IJEAS) 2017.

[6] C. C. Eze, Designed and fabricated a house-hold instant noodles making machine using cassava dough, American Journal of Engineering Research, Volume 5, Issue-10, 2016.

[7] Miki Pulley, Technical document, Moment of inertia calculation formula, 2013.

[8] Richard G, Budynas and J Keith Nisbett, B., Shigley's Mechanical Engineering Design Ninth Edition, McGraw-Hill Companies, 2011.

[9] B.L. Theraja, A.K. Theraja, A Textbook of Electrical Technology, Volume II, 2005.

[10] R.S. Khurmi, J.K. Gupta, A Textbook of Machine Design, 2005.

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