Determination of Optimal Torque in Thread Rolling Operation

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Abstract: *Manufacturing of threads, splines and gears will be conventionally done using metal cutting (chip flowing process). In the chip flow process, continuity and the strength of component gets decreased. Alternatively the above general components can also be manufactured by using metal forming in which the metal grain boundaries continuity remains and the components can withstand to higher loads. At present for cold forming for different work material and roll combinations, an empirical (approximate) solution is used for determining the torque requirements of motor. An attempt is being made to study the forming forces in thro feed of the component at roll and work intersection and to determine the optical torque requirement using ANSYS software. To generalize the thread rolling process, software code isdeveloped which can be used for different Work –Tool combinations to estimate the torque requirements.*

Keywords: *Analysis, Cold forming, Discretization, Grain boundaries, Optimal Torque, Thread rolling.*

I. INTRODUCTION

Thread Rolling is a process of making external threads. Threads are made through a die which is a hardened tool with the thread profile. The workpiece fixed with a rotor. The rotor turns around the metal piece and die creates a force, and the force results threads on the metal piece. As the force increases gradually, the accurate thread profile is transferred to the workpiece. This process produces screws with greater strength. During the process, the surface layer of the raw material is put into compressive stress.

Read forming and thread rolling are processes for forming screw threads, with the former referring to creating internal threads and the latter external threads. In both of these processes threads are formed into a blank by pressing a shaped die against the blank, in a process similar to knurling. These processes are used for large production runs because typical production rates are around one piece per second. Forming and rolling produce no swarf and less material is required because the blank size starts smaller than a blank required for cutting threads; there is typically a 15 to 20% material savings in the blank, by weight. A rolled thread can often be easily recognized because the thread has a larger diameter than the blank rod from which it has been made; however, necks and undercuts can be cut or rolled onto blanks with threads that are not rolled. Also, the end of the screw usually looks a bit different from the end of a cut-thread screw

Fig 1: Rolling

II. DIMENSIONS AND STRUCTURE OF MATERIALS

High Speed Steel (Rollers)
Composition – 18% tungsten, 4% chromium, 1% vanadium
Mild steel (Work)
Composition – 0.35-0.45% carbon, 0.05-0.35% silicon, sulphur and phosphorus less than 0.06%.

- Diameter of the roller = 170mm
- Diameter of the job = 50mm
- Length of the roller and job = 130mm
- Thread depth – 3.25mm
- Pitch – 5mm

Thread profile – **v**-thread

III. MATHEMATICAL CALCULATION FOR TORQUE

Motor power = 3.7kW = 5hp
Speed = 1500rpm
Power (p) = Torque (T) * Angular velocity (ω)
Torque, T = 3.7*1000*60/(2*π*1500) = 23.554 N-m
Maximum speed of the roll shaft = 84rpm

Power(p) = Torque (T) * Angular velocity(ω)
Torque, T = 3.7*60*1000/(2*π*84) = 420.623 N-m
Torque = Force * radius of the roller
Radius of the roller = 0.085m
Length of the roller = 0.130m
Force = 420.623/0.085 = 38.065 N
Force per unit length = Force/0.130 = 38.065/0.130 = 4.94845 N-m

IV. CONTACT STRESSES APPROACH FOR THREAD ROLLING TO FIND OUT THE TORQUE

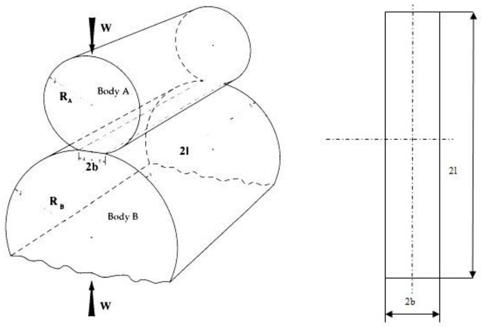

Fig 2: 2d Diagram of Rolling

where,
b is the half width of the contact rectangle [m]
l is the half length of the contact rectangle [m]
R1 is the reduced radius of curvature for two parallel cylinders in contact [m]. For the cylinders Rax = RA , Ray = ∞ , Rbx = RB , where RA1 and RB are the radii of the cylinders A and B, respectively.
1/R_x= 1/R_ax+ 1 〖/R〗 _bx=51.7645 1/m
R_ax=Radius of the roller
Rbx=Radius of the job
1/R_y= ∞
Reduced radius of curvature:
1/R^'= 1/R_x=51.7645 1/m
R^'=0.19318 m
Reduced young’s modulus
1/E^'= 1/2[(1-υ_A^2)/E_A +(1-υ_B^2)/E_B] = 5.6228*10-8 m^2/N
Where EA=Youngs modulus of the Roller=210*109N/m2
EB=Youngs modulus of the job = 200*109N/m2
Contact area dimensions
b= 4WR^'/πIE^'= 0.027m
Maximum and average contact pressure
P_max=W/πbl=4448176.615N/m2=4.4481MPa
P_average= W/4bl= 3493589 N/m2=3.4936MPa
Maximum deflection
δ= 0.319[W/E^'l][2/ 3+ln(4R_A R_B/b^2)]=0.01125m
Maximum shear stress
τ_max= 0.304Pmax
τ_max= 1.216MPa
Depth at which maximum shear stress occurs
Z=0.786b=0.021222m

T/J = τ / (radius of the roller)
J = π/32 (diameter of the roller)4 = 8.19965*10-5 m4
T =1173.03N-m

V. ‘C’ PROGRAM FOR CONTACT STRESSES

```
#include<stdio.h>
#include<conio.h>
#include<math.h>
void main ()
{
        float
R,r,rx,ea,eb,e1,b,l,w,a,deflection,pavg,pmax,shear,J,z
;
        label:read;
        printf("\nENTER POISONS RATIO OF
THE ROLLER");
        scanf("%f",&va);
        prinf("\nENTER POISONS RATIO OF
THE JOB");
        scanf("%f",&vb);
        if(va==0.5||vb==0.5)
        {
                printf("\nERROR!!!POISONS
RATIO YOU ENTERED IS EQUAL TO
                0.5”);
                goto read;
        }
        else
        {
                print("\nENTER THE ROLLER
RADIUS IN METRES");
                scanf("%f",&R);
                printf("\nENTER THE JOB
RADIUS IN METRES");
                scanf("%f",&r);
                rx=R*r/(r+R);
                printf("\nENTER THE LENGTH
OF THE ROLLER");
                scanf("%f",&l);
                printf("\nENTER THE YOUNGS
MODULUS OF THE ROLLER");
                scanf("%f",&ea);
                printf("\nENTER THE YOUNGS
MODULUS OF THE JOB");
                scanf("%f",&eb);
                printf("\nENTER THE LOAD
APPLIED");
                e1=ea*eb/((eb(1-va*va))-(ea(1-
vb*vb)))
                a=4wra/(3.141*l*e1);
                b=sqrt(a);
                pmax=w/(3.141*b*l);
                pavg=w/(4*b*l);
                shear=0.304*pmax;
                z=0.786*b
                J=3.141*2*R*R*R*R;
                T=J*shear/R;
printf("\n Half width = %f m \n Maximum pressure =
%f N-m^2 \n Average pressure = %f N-m^2 \n
```

```
Maximum shear = %f N-m^2 \n Maximum shear occurs at the depth of %f N-m^2 \n Required torque to produce maximum shear stress = %f N-m",b,pmax,pavg,shear,z,T);
            }
}
```

VI. FINITE ELEMENT METHOD

Element considered: **CST** (Constant Strain Triangle)

Table 1
Element Connectivity

Element	1	2	3
1	1	2	4
2	2	3	4

Element 1 in the following fig 3 represents job element
Element 2 in the following fig 3 represents the roller element

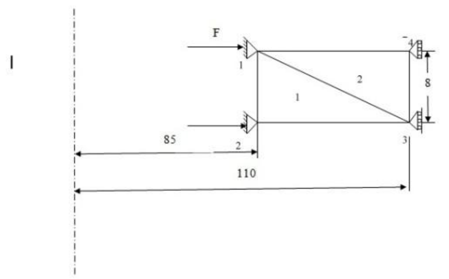

Fig 3: Free body diagram.

Table 2
Nodal points

Node	*r*	*z*
1	85	8
2	85	0
3	110	0
4	110	8

The units of millimeters for length, N for force,MgaPascals for stress and E.These units are consistent

Element 1
D_1=Elastic material matrix for roller
B_1=Strain displacement matrix for roller
K_1=Stiffness matrix for roller

$$D_1 = (E(1-\upsilon)/((1+\upsilon)(1-2\upsilon))) \begin{pmatrix} 1 & \upsilon/(1-\upsilon) & 0 & \upsilon/(1-\upsilon) \\ \upsilon/(1-\upsilon) & 1 & 0 & \upsilon/(1-\upsilon) \\ 0 & 0 & (1-2\upsilon)/2(1-\upsilon) & 0 \\ \upsilon/(1-\upsilon) & \upsilon/(1-\upsilon) & 0 & 1 \end{pmatrix}$$

E=modulus of elasticity of the roller
Roller material = High speed steel
E for Roller=210GPA
υ= poisons ratio=0.3

$$D_1 = \begin{pmatrix} 269000 & 115000 & 0 & 115000 \\ 115000 & 269000 & 0 & 115000 \\ 0 & 0 & 77000 & 0 \\ 115000 & 115000 & 0 & 269000 \end{pmatrix}$$

$$B_1 = \begin{pmatrix} z_{23}/\det J & 0 & z_{31}/\det J & 0 & z_{12}/\det J & 0 \\ 0 & r_{32}/\det J & 0 & r_{13}/\det J & 0 & r_{31}/\det J \\ r_{32}/\det J & z_{23}/\det J & r_{13}/\det J & z_{31}/\det J & r_{31}/\det J & z_{12}/\det J \\ N_1/r & 0 & N_2/r & 0 & N_3/r & 0 \end{pmatrix}$$

$$B_1 = \begin{pmatrix} -0.04 & 0 & 0 & 0 & 0.04 & 0 \\ 0 & 0.1250 & 0 & -0.1250 & 0 & 0 \\ 0.125 & -0.04 & -0.125 & 0 & 0 & 0.04 \\ 0.0035 & 0 & 0.0035 & 0 & 0.0035 & 0 \end{pmatrix}$$

$$D_1 * B_1 = \begin{pmatrix} -10357.5 & 14375 & 402.5 & -14375 & 11162.5 & 0 \\ -4197.5 & 33625 & 402.5 & -33625 & 5002.5 & 0 \\ 9625 & -3080 & -9625 & 0 & 0 & 3080 \\ -3658.5 & 14375 & 941.5 & -14375 & 5541.5 & 0 \end{pmatrix}$$

$$B^T = \begin{pmatrix} -0.04 & 0 & 0.1250 & 0.0035 \\ 0 & 0.1250 & -0.04 & 0 \\ 0 & 0 & -0.1250 & 0.0035 \\ 0 & -0.1250 & 0 & 0 \\ 0.04 & 0 & 0 & 0.0035 \\ 0 & 0 & 0.04 & 0 \end{pmatrix}$$

Now the stiffness matrix of element 1 can be found out as follows
$K^e = 2\pi\bar{r}A_e\bar{B}^T D\bar{B}$
$A_e = 0.5|\det J|$
$\bar{r}$=centroid=93.333

$|\det J| = r_{13}z_{23} - r_{23}z_{13}$
$|\det J|=(85-110)(0-0)-(85-110)(8-0)$
$=-(85-110)(8)$
$=200$
$|\det J|=200$

$A_e=0.5*200$
$=100$

$$K_1 = 53407.075 \begin{pmatrix} 1604.6203 & -909.6875 & -1215.9298 & 524.6875 & -427.1048 & 385 \\ 909.6875 & 4326.3250 & 435.3125 & -4203.1250 & 625.3125 & -123. \\ -1215.9298 & 435.3125 & 1206.4203 & -50.3125 & 19.3953 & 385 \\ 524.6875 & -4203.125 & -50.3125 & 4203.1250 & -625.3125 & 0 \\ -427.1048 & 625.3125 & 19.3953 & -625.3125 & 465.8953 & 0 \\ 385 & -123.2 & -385 & 0 & 0 & 123.2 \end{pmatrix}$$

Element2:
D_2=Elastic material matrix for the job
B_2=Strain displacement matrix for the job
K_2=Stiffness matrix
E=Modulus of elasticity of the job(mild steel)=200GPA

$$D_2= (E(1-\upsilon)/((1+\upsilon)(1-2\upsilon))) \begin{pmatrix} 1 & \upsilon/(1-\upsilon) & 0 & \upsilon/(1-\upsilon) \\ \upsilon/(1-\upsilon) & 1 & 0 & \upsilon/(1-\upsilon) \\ 0 & 0 & (1-2\upsilon)/2(1-\upsilon) & 0 \\ \upsilon/(1-\upsilon) & \upsilon/(1-\upsilon) & 0 & 1 \end{pmatrix}$$

υ =poisons ratio=0.3

$$D_2 = \begin{pmatrix} 282000 & 121000 & 0 & 121000 \\ 121000 & 2820000 & 0 & 121000 \\ 0 & 0 & 80560 & 0 \\ 121000 & 121000 & 0 & 282000 \end{pmatrix}$$

$$B_2 = \begin{pmatrix} -0.04 & 0 & 0.04 & 0 & 0 & 0 \\ 0 & 0 & 0 & -0.125 & 0 & 0.1250 \\ 0 & -0.04 & -0.1250 & 0.04 & 0.1250 & 0 \\ 0.0032 & 0 & 0.0032 & 0 & 0.0032 & 0 \end{pmatrix}$$

$$D_2{*}B_2 = \begin{pmatrix} -10892.8 & 0 & 11667.2 & -15125 & 387.2 & 15125 \\ -4452.8 & 0 & 5227.2 & -35250 & 387.2 & 35250 \\ 0 & -3222.4 & -10070 & 3222.4 & 10070 & 0 \\ -3937.6 & 0 & 5742.4 & -15125 & 902.4 & 15125 \end{pmatrix}$$

$$B^T = \begin{pmatrix} -0.04 & 0 & 0 & 0.0032 \\ 0 & 0 & -0.04 & 0 \\ 0.04 & 0 & -0.125 & 0.0032 \\ 0 & -0.1250 & 0.04 & 0 \\ 0 & 0 & 0.1250 & 0.0032 \\ 0 & 0.1250 & 0 & 0 \end{pmatrix}$$

$K^e = 2\pi\bar{r}A_e\bar{B}^TD\bar{B}$
$A_e = 0.5|\det J|$
$\bar{r}$=centroid=101.6667

$$K_2=15707.96 \begin{pmatrix} 423.112 & 0 & -448.312 & 556.6 & -12.6 & -556.6 \\ 0 & 128.896 & 402.8 & -128.896 & -402.8 & 0 \\ -448.312 & 402.8 & 1743.814 & -1056.2 & -1240.374 & 653.4 \\ 556.6 & -128.896 & -1056.2 & 4535.146 & 354.4 & -4406.25 \\ -12.6 & -402.8 & -1240.374 & 354.4 & 1261.638 & 48.4 \\ -556.6 & 0 & 653.4 & -4406.25 & 48.4 & 4406.25 \end{pmatrix}$$

Using the elimination approach, on assembling the matrices with reference to the degrees of freedom 1 & 3

$$K = \begin{pmatrix} 85698000 & -64939000 \\ -64939000 & 71077000 \end{pmatrix}$$

The force supplied by the Hydraulic power pack is a traction force.
So the Roller exhibits traction force on the job.
$F_1=F_3=2*\pi*r_1*l_e*p_i/2$
Where r_1=Radius of the roller=85mm
l_e=sample length=8mm

P_i=3.4936MPa

Hence the force vector is obtained as follows

$$F = \begin{pmatrix} 7463.9592 \\ 7463.9592 \end{pmatrix}$$

K*Q=F

$$Q= \begin{pmatrix} 0.0005 \\ 0.0006 \end{pmatrix}$$

$\sigma_1 = D_1B_1Q_3$
$Q^1= [\,0.0005 \quad 0 \quad 0.0006 \quad 0 \quad 0 \quad 0\,]^T$
$Q^2= [\,0.0006 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0\,]^T$

$$D_1{*}B_1 = \begin{pmatrix} -10357.5 & 14375 & 402.5 & -14375 & 11162.5 & 0 \\ -4197.5 & 33625 & 402.5 & -33625 & 5002.5 & 0 \\ 9625 & -3080 & -9625 & 0 & 0 & 3080 \\ -3658.5 & 14375 & 941.5 & -14375 & 5541.5 & 0 \end{pmatrix}$$

$$Q_1 = \begin{pmatrix} 0.0005 \\ 0 \\ 0.0006 \\ 0 \\ 0 \\ 0 \end{pmatrix}$$

$$\sigma_1 = D_1B_1Q_1 = \begin{pmatrix} -4.9373 \\ -1.8573 \\ -0.9625 \\ -1.2644 \end{pmatrix}$$

Where σ_1=Stress matrix for the roller

$$D_2{*}B_2 = \begin{pmatrix} -10892.8 & 0 & 11667.2 & -15125 & 387.2 & 15125 \\ 4452.8 & 0 & 5227 & -35250 & 387.2 & 35250 \\ 0. & -3222.4 & -10070 & 3222.4 & 10070 & 0 \\ -3937.6 & 0 & 5742.4 & -15125 & 902.4 & 15125 \end{pmatrix}$$

$$Q_2 = \begin{pmatrix} 0.0006 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{pmatrix}$$

$$\sigma_2 = D_2B_2Q_2 = \begin{pmatrix} -6.5357 \\ -2.6717 \\ 0 \\ -2.3626 \end{pmatrix}$$

Where σ_2=stress matrix for the job

From the σ_1 matrix, maximum shear stress
τ = 0.9625MPa
Now the torque can be found out from the torsion equation $T/J = \tau/r$
Where,
T = torque
J=Polar moment of inertia
τ = Shear stress
r = Radius of the roller = 85mm
$J = \pi/32$ (diameter of the roller)^4
$T = (\pi/32\ (170)^4)*0.9625/85$
= 928490.3457 N-mm = 928.490 N-m
Hence torque due to shear stress=928.490 N-m

VII. ANSYS-RESULTS

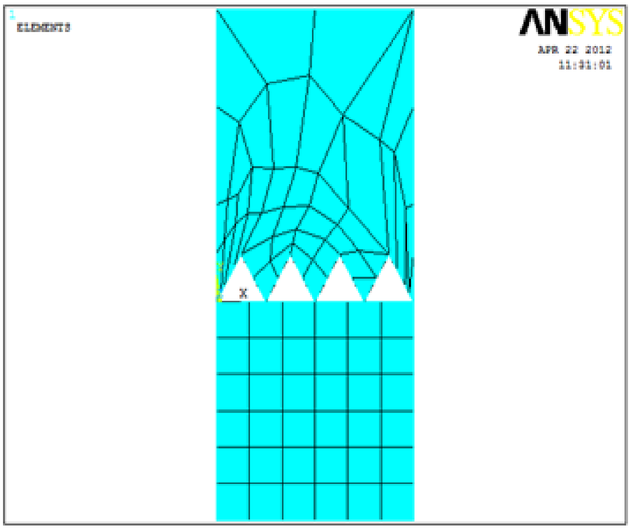

Fig 4: Stress diagram 1

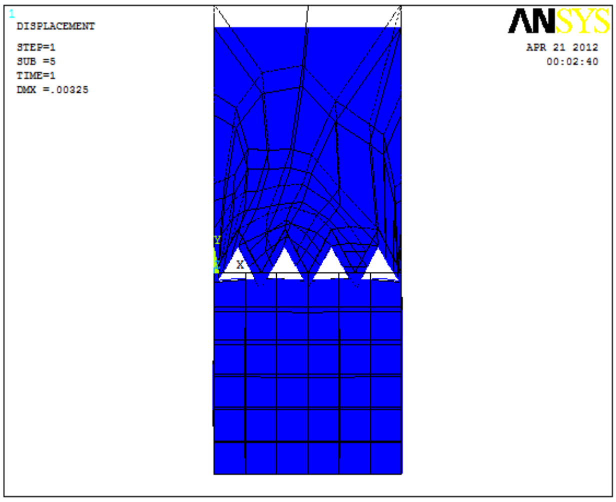

Fig 5: Stress diagram 2

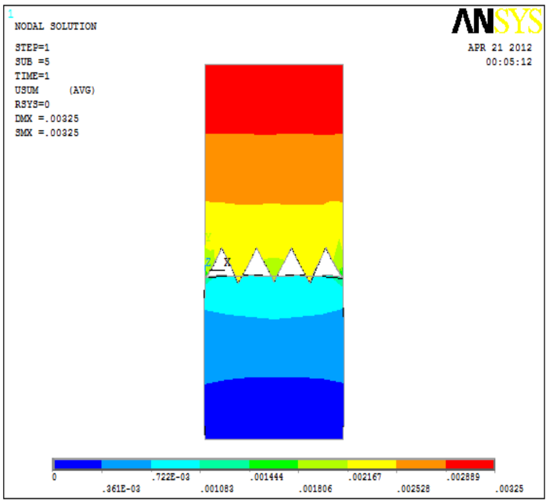

Fig 6: Stress diagram 3

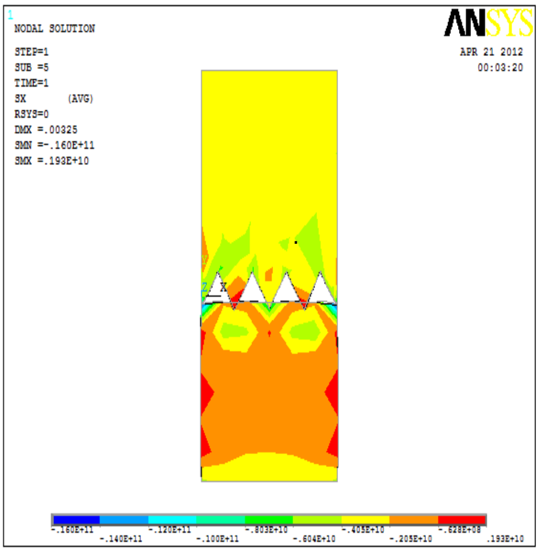

Fig 7: Stress diagram 4

VIII. CONCLUSION

The torque obtained by contact stresses approach and FEM are almost in the equal range In ANSYS, the stress intensity obtained is maximum at cutting edges and the results obtained are within the allowable stresses. The strain hardening affect is not taken into consideration in any of the approaches.

Table 2:
Comparison Table

S .NO	METHOD	TORQUE (ONE ROLLER) N-m
1	**Contact stresses**	1173.03
2	**FEM**	928.490

IX. FUTURE SCOPE

This can further be extended by taking strain hardening affect into account and also dynamic analysis can be performed to obtain optimal torque.

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Conflict of Interest: Nil

REFERENCES

[1] T. Fukuoka, *Threaded Fasteners for Engineering and Design*, Corona Publishing Co., Ltd.,pp. 263-321 2015.

[2] W.T. Wu, G. Li, A. Arvind, J.P. Tang, *Development of a Three- dimensional Finite Element Method Based Process Simulation Tool for the Metal Forming Industry*, Vol. 75, No. 3, American Society of Mechanical Engineers, Petroleum Division, New York, pp. 143–150 1996.

[3] Shigeru Yamanaka, Kenji Amiya, YasunoriSaotomeand Akihisa Inoue, "*Plastic Working of Metallic Glass Bolts by Cold Thread Rolling" Materials Transactions*, Vol. 52, No. 2 pp. 243 to 249 2011.

[4] Shao-Yi Hsia and Shuo-Kai Pan, Yu-Tuan Chou "*Computer Simulation for Flat-die Thread Rolling of Screw*" of International journal of advanced scientific and technical research Issue 2 volume 4 2016.

[5] ToshinakaShinbutsu, Shuichi Amano, TeruieTakemasu, Toshihiko Kuwabara and Jyo Shimura *Thread rolling and performance evaluations of a new anti-loosening double thread bolt combining a single thread and multiple threads* in International Conference on the Technology of Plasticity, ICTP 2017, 17-22.

[6] Z. Pater, A. Gontarz, W. Weroñski"*New method of thread rolling*" Journal of Materials Processing Technology 153–154 (2004) 722–728.

[7] L. Kukielka& K. Kukielka "*Numerical analysis of the process of trapezoidal thread rolling*" High Performance Structures and Materials IIIvol 85-2006.

[8] C Labesh Kumar, V Jayakumar, G Bharathiraja, *Optimization of welding parameters for friction stir spot welding of AA6062 with similar and dissimilar thicknesses*. Materials Today: Proceedings Volume 19 Pages 251-255.

[9] C. Labesh kumar, T. Vanaja, Dr. KGK Murti, VVSH Prasad, "*Optimization of Mig Welding Process Parameters for Improving Welding Strength of Steel*" International Journal of Engineering Trends and Technology (IJETT) Volume-50 Number-1 Year of Publication : 2017.