Design and Analysis of Flywheel for Weight Optimization

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Abstract— There are many causes of failure of flywheel. Maximum tensile and bending stresses induced in the rim and tensile stresses gets induced in the arm under the action of centrifugal forces are the main causes of flywheel failure. In this work stress evaluation in the rim and arm are studied using finite element method and results are validated by analytical calculations .The models of flywheel having solid, four and six no. arms are developed. The FE analysis is carried out for different loading conditions on the flywheel for different cases the maximum Von-mises stresses and deflection in the rim are determined. From this analysis it is analyzed that Maximum stresses induced are in the rim and arm junction. Due to application of tangential forces, maximum bending stresses occurs near the hub end of the arm. It is also observed that for low angular velocity the effect of gravity on stresses and deflection of rim and arm is predominant. The FEM formulation carried out in ANSYS workbench R14.

Key words: Flywheel Weight Optimization, Analysis of Flywheel

I. INTRODUCTION

A flywheel works on the principle of an inertial energy-storage. It absorbs mechanical energy and serves as a reservoir which stores energy when the supply of energy is more than actual requirement and releases it during the period when the requirement of energy exceeds than the amount supplied. Several hundred years ago pure mechanical flywheels where used solely to keep machines running smooth from cycle to cycle, thereby render possible the industrial revolution. During that time several shapes along with designs where implemented, but it took until the early 20th century before flywheel rotor shapes and rotational stress acting on them were thoroughly analyzed [1].Later in the 1970s flywheel energy storage was proposed as a primary objective for electrically operated vehicles and stationary power devices. In 1970's itself fiber composite rotors where built, and in the 1980s magnetic bearings started to appear [2]. Thus the positive perspective for using flywheels as electric energy storage device has long been established by extensive research. Recent intensive improvements in material, magnetic bearings and power electronics make flywheels a competitive choice for a number of energy storage applications.

A. *Factors Affecting Flywheel Performance*

1) *Material strength*

Stronger materials could undertake considerable operating stresses, hence could be run at high rotational speeds allowing flywheel to store more energy.

2) *Rotational speed*

It directly controls the energy stored, higher speeds tends to more energy storage, but high speeds exerts excessive loads on both flywheel and bearings during the shaft design.

3) *Geometry*

It controls the certain Energy, in other words, kinetic energy storage capability of the flywheel. Any optimization effort taken in flywheel cross-section may contribute substantial improvements in kinetic energy storage capability thus reducing both resulting shaft/bearing loads and material failure occurrences. Flywheel efficiency includes the amount of certain kinetic energy (energy per unit mass) and mechanical losses.

To improve the quality of the product and in order to have safe and well-grounded design, it is necessary to investigate the stresses induced in the component during running condition. When the flywheel rotates, centrifugal forces will act on the flywheel due to which tensile and bending stress get induced in the rim of flywheel. To counter the requirement of balancing out the large oscillations in velocity during a complete cycle of a mechanism, a flywheel is designed, optimized and analyzed. By using optimization methods various parameter like material, cost for flywheel can be optimized and by applying an approach for modification of various working parameter like efficiency, output, energy storing capacity, we can compare the result with existing flywheel result.

The most important advantages of flywheels is the ability to handle high power levels. This is a desirable quality in e.g. a vehicle, where huge peak power is necessary during acceleration and, if electrical breaks are used, a large amount of power is generated for a short period during breaking, which implies a more efficient use of energy leading to lower fuel consumption. Individual flywheels have capacity of storing up to 500MJ and peak power ranges from KW to GW, with the higher powers aimed at pulsed power applications. Stress analysis is the complete and inclusive study of stress distribution in the specimen under study.

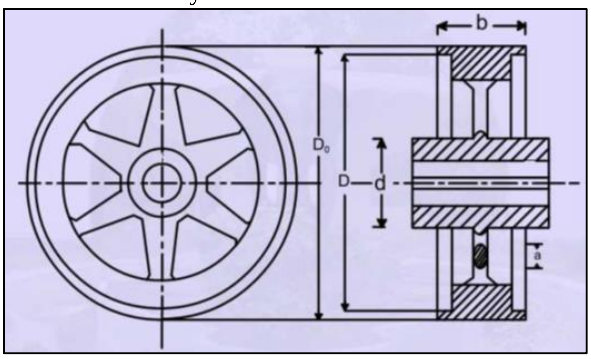

Fig. 1: Arm type of flywheel

This particular paper is focused on the analytical design of arm type of flywheel which is used for punching press operation. As per as design of flywheel is concerned it is required to decide the mean diameter of the flywheel rim, which rely upon mainly two factors such as availability of space and the limiting value of peripheral velocity of the fly wheel. However the said design problem is formulated for punching machine which has to be make holes of 30 holes/minute in a steel plate of18mm thickness along with space limitation that is the diameter of flywheel should not exceed 1000mm, hence it can be examined that the design of the flywheel is to be carried out(based) on the availability of space limitation. Weight reduction is done by using arm type flywheel instead of disc type flywheel.

II. LITERATURE REVIEW

Akshay P. Punde, 1 G.K.Gattani2 [1] have proposed, a computer-aided-designs of software for flywheels using object-oriented programming advancing towards Visual Basic. The various configurations of flywheels (rimmed or solid) formed the basis for the development of the software. The software's graphical characteristics were used to give avisual interpretation of the solutions. The software's efficiency as well as effectiveness was tested on a number of numerical examples, some of which are outlined in this work.

ITony. A. Baby, IITony Kurian, IIIMelvin Eldho Shibu had proposed [2] flywheel design, and analysis the material selection process. The FEA model is described here to achieve a better understanding of the mesh type, mesh size and boundary conditions applied to complete an effective FEA model for finding Von Mises stresses.

Arvind. S. Sorthiya, Palak. J. Patel [3] have proposed that weight reduction of flywheel improves the machine performance. Weight reduction is possible by changing shape of flywheel or by changing material of flywheel.

S. M. Dhengle, Dr. D. V. Bhope, S. D. Khamankar [4] There are many causes of flywheel failure. Among them, maximum tensile and bending stresses induced in the rim and tensile stresses induced in the arm under the action of centrifugal forces are the main causes of flywheel failure. Hence in this work evaluation of stresses in the rim and arm are studied using finite element method and results are validated by analytical calculations

Sushama G Bawane1*, A P Ninawe1 and S K Choudhary1 [5] The FEA model is described to achieve a better understanding of the mesh type, mesh size and boundary conditions applied to complete an effective FEA model. At last the design objective could be simply to minimize cost of flywheel by reducing material.

Ashish R. Sonekar1, S. G. Bahaley2 [6] This study solely focuses on exploring the effects of flywheel geometry on its energy storage/deliver capability per unit mass, further defined as specific energy. Various profiles of flywheel are studied and the stored kinetic energy is calculated for the respective flywheel. Various profiles designed are solid disk, rim type, webbed/section cut flywheel. It shows that smart design of flywheel geometry could both have a significant effect on the Specific Energy performance and reduce the operational loads exerted on the shaft/bearings due to reduced mass at high rotational speeds. Efficient flywheel design used to maximize the inertia of moment for minimum material used and guarantee high reliability and long life.

III. MATERIAL PROPERTIES

Ultimate strength Sut = 214 MpaSus = 303 Mpa
Modulus of elasticity &modulus of rigidity E = 101 Gpa G = 41 Gpa Density =7510 Kg/m3 Poisson's Ratio = 0.23

IV. ANALYTICAL METHODOLOGY

The method adopted for the design of flywheel for punching machine is Analytical method. Now let us determine the various parameters in regard with flywheel design.

– Mean speed of flywheel N= 9 Number of strokes/min =9×30=270rpm
- Maximum shear stress required to punch a hole =Shear strength × resisting area =fs× π dt =240× π×25×18×1000=339.3kN
- Energy required to punch one hole =average force × distance travelled by punch =0.5×339.3× 18=3053.7N
- Since mechanical efficiency is less than 100%, assuming as 85% , therefore
  Total energy required, E=3053.7/0.85 =3592.6N-m
- Actual punching operation lasts for the 1/10th of the cycle period. Therefore, during remaining period 9/10th of the cycle period, the energy is stored by the flywheel. Thus fluctuation of energy =9/10×E= 3233.3Nm
- Maximum space available is 100mm,therefore considering as D=800mm to carry out design
- Rim Velocity= π× 0.8×27060 =11.3m/s…which is less than the maximum permissible velocity for grey cast iron
- Mass of flywheel, m=fluctuation of energy V2 × Cs = 3233.3 = 253.3 Kg 11.32×0.1
- Assuming mass of rim as 90%of total mass, mrim=0.9×253.2=227.88Kg
- mrim= πDhp ….used to determine dimensions of rim

Where, p=7100kg/m3for C.I.
Therefore, h=90mm, b=150mm

- Outer diameter of flywheel=Do=D+h=0.89m, which is less than the maximum space of 1m, hence the design dimensions are within limit.
- To determine Shaft diameter
- Bending moment M =weight of flywheel× overhang =253.2×9.81× 0.2=496.78N-m
- Average torque =Energy required/min 2πN = 107778 2π×270 =63.53N-m
- Assuming suddenly applied load condition with shock and fatigue factor of 1.5 and 2
- Equivalent torque =((Cm×M)2 +(Ct× T)2)1/2 =755.92N-m
- Shaft is made of medium carbon steel, with shear strength 360N-mm2, Factor of safety is 4,therefore shaft diameter can be determined by using torsion equation
- Shaft diameter,ds =34.96 say=40mm
- Hub diameter, dh= 2ds=80mm
- Length of hub, lh = 2.5 ds=100mm
- To determine the Stresses in the rim of flywheel
- Stresses in unstrained rim = pv2 = 7100×11.32 = 0.9066MN/m^2
- Stresses in restrained rim = pv2(2πR2/ i2h) =4.97 MN/m^2
- Total Stress in the rim, = 0.75 Stresses in unstrained rim+0.25 Stresses in restrained rim =1.922

MN/m^2….which is less than the allowable strength of C.I, hence design of rim is safe

– To determine stress in arm of flywheel Considering arm type flywheel of four arms Bending stress in the arm =10N/mm^2 =T(D-dh)

iDz

Where, Z=modulus of elasticity=1429.4mm^3

i=numbers of arms=4

– Direct stress due to centrifugal force =pv2=0.9066N/ mm^2
– Total stress = Bending stress+ Direct stress =10+0.9066=10.9066N/mm^2
– Total stress is less than the allowable strength 20N/mm^2, hence the design of the arms are safe.

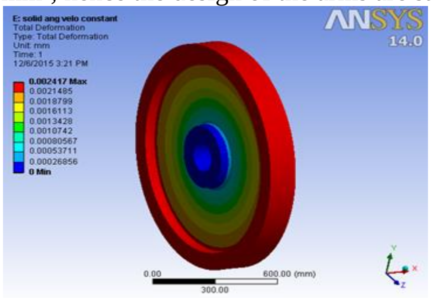

Fig. 2: Total Deformation Solid flywheel

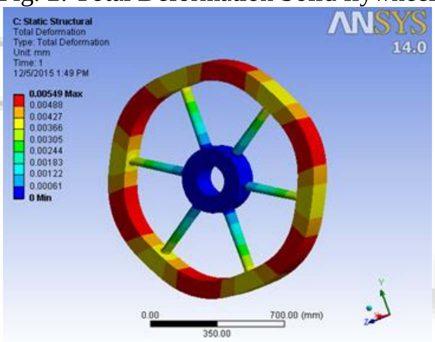

Fig. 3: Total Deformation 6 arm flywheel

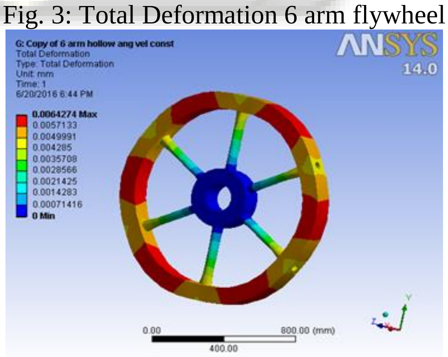

Fig. 4: Deformation 6 arm hollow arm flywheel

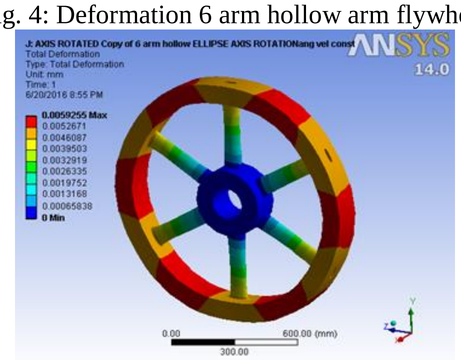

Fig. 5: Deformation 6 arm hollow arm flywheel arm c/s rotated

A. Case I: Constant Angular Velocity=25.12 Rad/Sec.

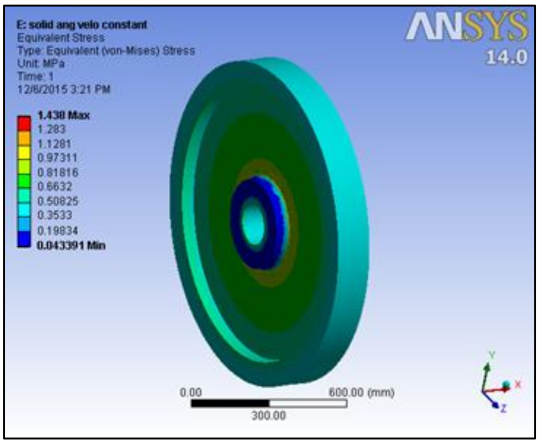

Fig. 6: Von Mises Stress Solid Flywheel

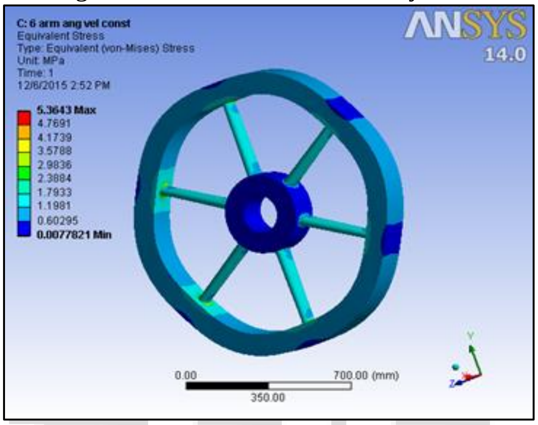

Fig. 7: Von mises stress 6 arm flywheel

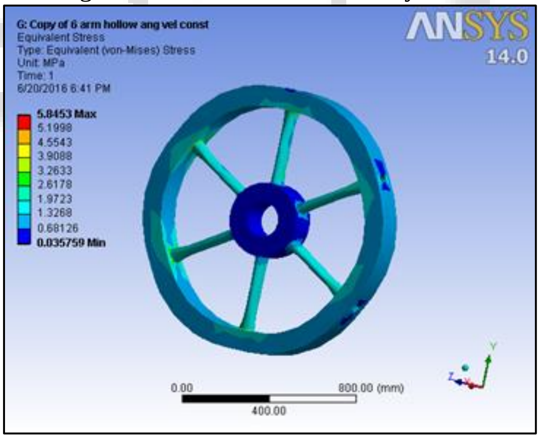

Fig. 8: Von mises stress 6 arm hollow arm flywheel

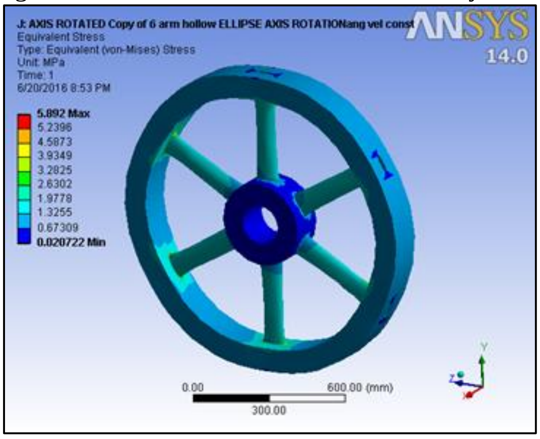

Fig. 9: Von mises stress 6 arm hollow arm flywheel arm c/s rotated

B. Case I: Constant Angular Velocity=25.12 Rad/Sec.

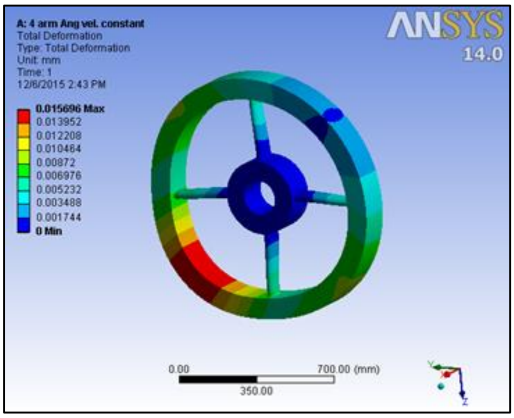

Fig. 10 Deformation 4 arm flywheel

C. Case I: Constant Angular Velocity=25.12 Rad/Sec.

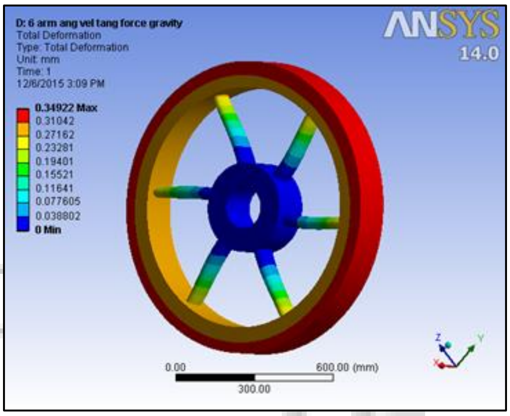

Fig. 11: Deformation 6 arm flywheel

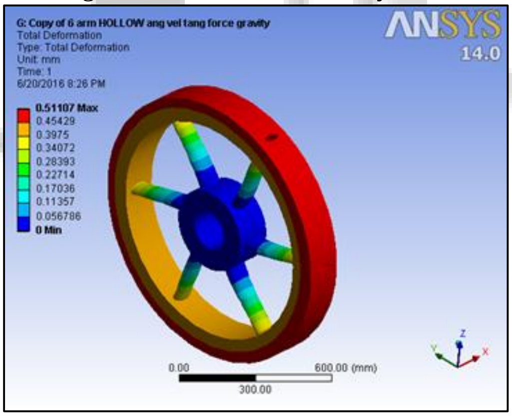

Fig. 12: Deformation 6 arm hollow arm flywheel

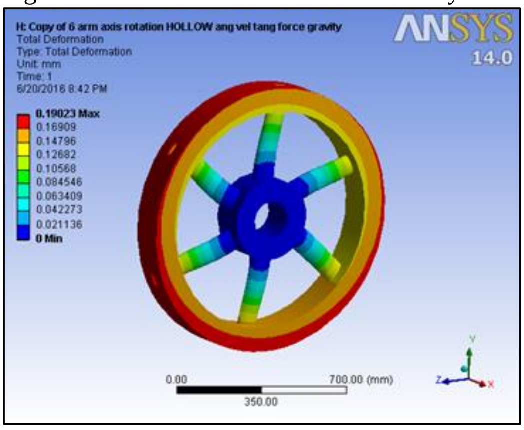

Fig. 13: Deformation 6 arm hollow arm flywheel arm c/s rotated

D. Case II: Tangential Force to Outer Rim=13376N

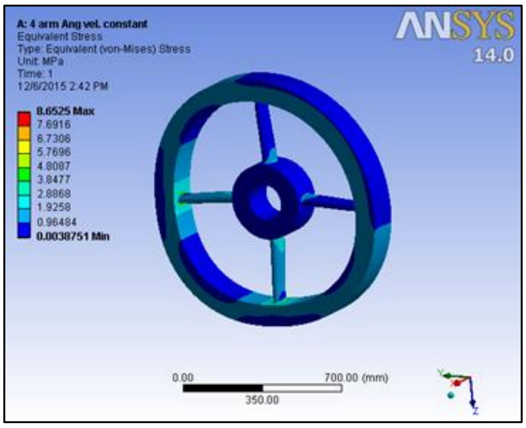

Fig. 14: Von mises stress 4 arm flywheel

E. Case I: Constant Angular Velocity=25.12 Rad/Sec.

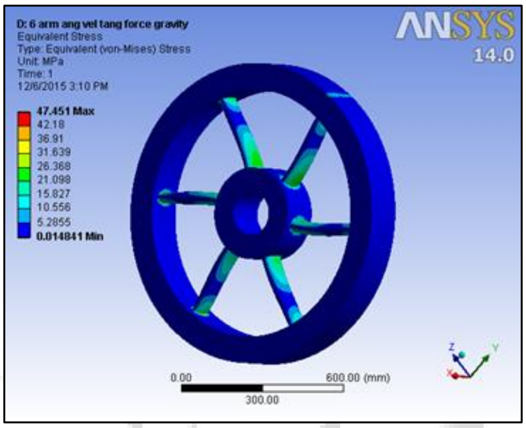

Fig. 15: Von mises stress 6 arm flywheel

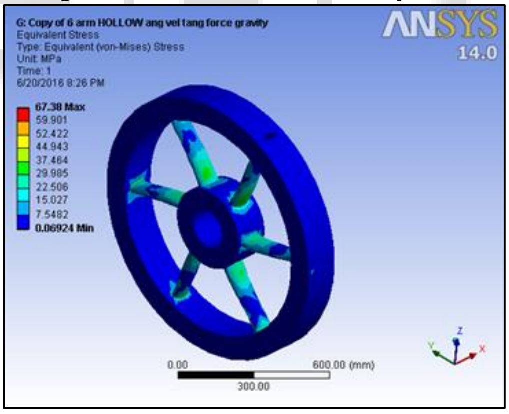

Fig. 16: Von mises stress 6 arm hollow arm flywheel

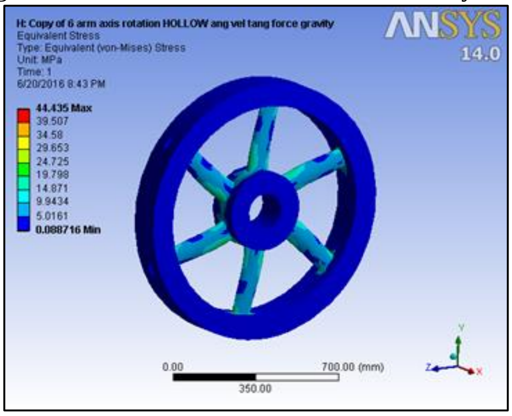

Fig. 17: Von mises stress 6 arm hollow arm flywheel arm c/s rotated

F. Case II: Tangential Force to Outer Rim=13376N

Type of Flywheel	Total Deformation(mm)	Von mises Stress(N/mm^2)
Solid	0.002184	1.438
4 Arm	0.01569	8.6525
6Arm	0.00549	5.3643
6Arm Hollow Arm	0.00643	5.845
6 Arm hollow arm Elliptical Arm c/s axis rotated by 90deg	0.00592	5.892

Table 1: Case I

Type of Flywheel	Total Deformation(mm)	Von mises Stress(N/mm^2)
Solid	0.0176	14.547
4 Arm	0.53498	76.718
6Arm	0.34922	47.451
6Arm Hollow Arm	0.511	67.38
6 Arm hollow arm Elliptical Arm c/s axis rotated by 90deg	0.190	44.435

Table 2: Case II

V. Concussions

It can be conclude that in case of flywheel design it is first important to know the design requirement such as material, type, application, desired speed (peripheral velocity), availability of space, and dimensions of the flywheel.

- The peripheral velocity of the flywheel is governed by allowable strength of the flywheel material, generally for grey cast iron it should be less than or greater than25m/s.
- In case of arm type of flywheel the numbers of arms selection depends upon the diameter of the flywheel, generally four numbers of arms are selected for diameter of flywheel which should be less than 0.75m.
- As the current problem is related to the design of flywheel with consideration(limitation) of the space availability that is maximum space should be less than1m, but after carrying out design the outer diameter of flywheel obtained is of 0.89m which is less than the required condition, hence can be concluded as design is safe.
- By replacing solid type of flywheel with arm type flywheel weight is reduced by 24% with minor effect on deformation and von mises stress. Which is cost effective.
- By making arms of flywheel hollow again weight will gets reduced by 7.3%.
- By rotation of elliptical cross section major axis with 90 degrees we will get improvement in deformation and von mises stress.

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