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Kinematic Analysis and Design of a Geneva Stop Mechanism Teaching Aid for Intermittent Motion

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Abstract-The importance of intermittent motion in many equipment and machinery has called for the necessity to fully understand the various concepts of those systems where this type of motions are required in between continuous motions and also in those systems designed to perform automatic timing operations in robotic applications. This paper examines the kinematics analysis of a four (4) slot Geneva mechanism. The arbitrary model was developed and the effects of the force were neglected. The angular displacement of the wheel was investigated as is the function of crank angle and found to occur between $\pm \frac{\pi}{4}$; $\frac{7\pi}{4}$ to $\frac{9\pi}{4}$; $\frac{15\pi}{4}$ to $\frac{17\pi}{4}$; $\frac{23\pi}{4}$ to $\frac{25\pi}{4}$ radians for the 1st, 2nd, 3rd and 4th revolutions of the crank angle respectively, in between these angles, there are dwelling periods. Also, for an assumed speed of the crank (12 rpm), the angular velocity of the wheel was obtained to be maximum and minimum between ± 3.0338 rad/sec at crank angles of 0° , 2π , 4π and 6π radians, for each of the 4 revolutions and that of angular acceleration was ± 8.5161 rad/sec² at ± 0.19 , 6.10 and 6.47 , 12.38 and 12.75 , 18.66 and 19.04 radians respectively. The timing of the tangent drive was expressed by the arc of each revolution of the driver or crank that the wheel was being indexed and the arc of each revolution that the wheel was at rest (idle).

Keywords- Geneva Wheel; Crank; Intermittent Motion; Kinematics; Kinetics

I. INTRODUCTION

Geneva mechanism which is also called Maltese cross is a gear like mechanism that translates continuous motion to intermittent motion through the driving wheel whose crank pin is shown in Fig. 1. It drives the Geneva wheel as it slides into and out of the slot of the Geneva wheel. It thus advances it - one step at a time when engaged [1]. The driver wheel usually consists of both the crank and a raised circular blocking disk that locks the Geneva wheel in position between steps to avoid excessive vibration while rotating [2, 3].

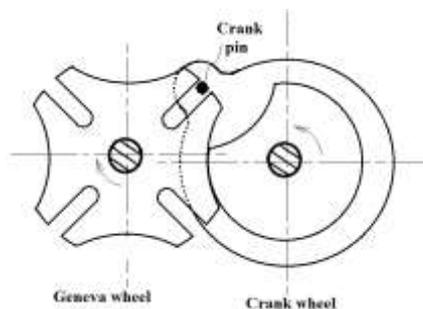


Fig. 1 Four external slot Geneva mechanism in starting position

Geneva mechanism's popularity as a means of producing incremental motion arises from its simplicity in design and construction, it is therefore a relatively low cost indexing device [4]. Geneva mechanism must have at least three slots for any intermittent motion to be achieved [5]. It represents a form of intermittent gearing where the driver wheel has a pin that reaches into a slot of the driven wheel and thereby advances it by one step at a time. A conventional Geneva mechanism generated intermittent motion differs from that of the cam mechanism in that; for each of its stops there is a corresponding and particular motion period and a motion law, unlike the cam mechanisms where there is a free choice of motion period and motion law [6].

Numerous areas of application of these mechanisms has been highlighted by Akpaidu and Dahunsi, Hseih, Figliolini et al., and Zhang et al. [7-9]. These application areas include mechanical wound-up watches, timing instruments, precision measurement equipment, film projectors, machine tools, discrete motion drives with high load capacity applications in robotic manipulators, printing and machinery presses, automated packaging, weaving looms, and transfer lines [7-10].

The greatest motivation for industrial use of Geneva mechanism is its simplicity and inexpensive design and construction. This however limits its operations to low speed machinery applications because of problems like high impact force and jerking motion at the point of engagement and disengagement of the crank pin, shock loading, vibration and wear [7-10]. These performance challenges have been confronted by selecting appropriate displacement programmes such that zero displacement

and acceleration are avoided at the point of engagement and disengagement [10]. Other works dealt with synthesis of conjugate Geneva mechanisms and redesigning of the slots [2, 6, 8, 10, 11].

These mechanism design improvements affect the kinematic properties of the Geneva mechanism and are always successful in introducing some considerable level of complications to its design. Meanwhile, the greatest threat to industrial application of Geneva wheel remains the stepper motors. The Geneva mechanism, however, remains a classic tool in teaching kinematic analysis and motion conversion to students. The objective of this paper is to design a conventional Geneva mechanism with four external slots for use in teaching of kinematics of mechanisms and intermittent motion.

II. SPECIFICATIONS, DESIGN AND KINEMATIC ANALYSIS

A. Wheel Geometry and Specifications

Figs. 2 (a) and (b) show the geometry of the Geneva wheel. In this study, three main (i.e., crank angle, angular velocity, and acceleration) are most essential when specifying particular geometry in the design of Geneva mechanism [1, 2, 12]. The crank pin, P always engage the slots on the Geneva wheels tangentially such that the center line of the slot and the line connecting the crank pin form a right angle with the crank's rotation center when the crankpin enters or leaves the slot.

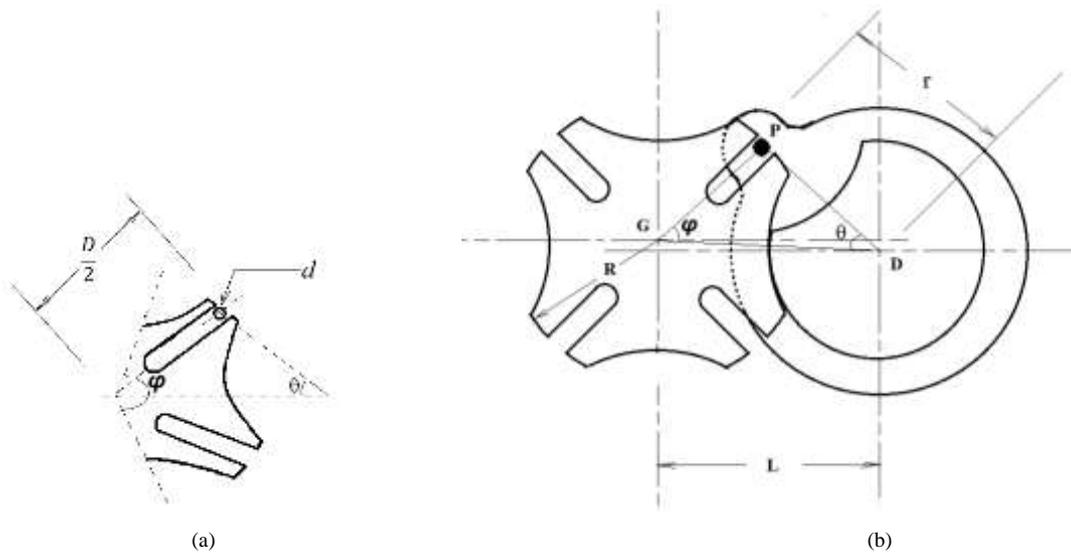


Fig. 2 Geneva wheel geometry

The crank (driven) wheel has n number of slots, and made to advance by $360/n$ for each full rotation of the crank wheel. The other variables are: Geneva wheel diameter, D ; crank or drive pin diameter, d , and the locking radius, r . It is also usually convenient to use both pin diameter, d and the tip thickness, t of the wheel as the ratio of the wheel diameter, D to give what can be described as normalized variables. Other geometrical dimensions are available the detail design drawings.

For any set of n - slots wheel, the normalized pin diameter d' and tip thickness t' can be considered constant or, any given d' and t' can be used to define a proportional set of Geneva wheels [7, 12]. Also, the thickness of the Geneva wheel cannot be considered as an independent parameter, but was taken to be equal to the pin diameter. This particular assumption was made to ensure that the drive pin load across the thickness of the wheel is approximately uniform. If the pin diameter was made small with respect to the Geneva wheel thickness, then the loading will be concentrated near the fixed end of the beam and will not be uniformly distributed across the face of the wheel which will make the kinematics analysis unrealizable [12]. For further analysis, the path followed by the crank pin was traced out and used to outline the triangle PGD in Fig. 3.

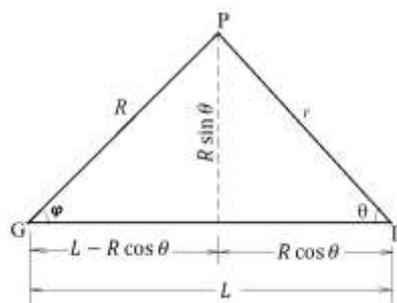


Fig. 3 Path triangle of the Geneva and the drive wheels

Since the Geneva wheel under consideration in this work is the four external straight slot wheel, triangle PGD is an isosceles right angled triangle such that:

$$\varphi = \tan^{-1} \frac{\sin \theta}{\sqrt{2} - \cos \theta} \quad (1)$$

Equation (1) provides a relationship between the angle θ and α , this relationship is valid within $-\frac{\pi}{4} < \theta < \frac{\pi}{4}$. Outside this range, there is no physical contact between the wheel and the Geneva wheel undergoes a dwell period while the crank wheel completes its revolution and positions the pin to engage the next slot.

Similarly, by taking the first and second differential derivatives of both sides, velocity and acceleration relationships for both wheels could be derived

$$\omega_G = \dot{\varphi} = \omega_C \frac{\sqrt{2} \cos \theta - 2}{3 - 2\sqrt{2} \cos \theta} \quad (2)$$

and

$$\alpha_G = \ddot{\varphi} = \alpha_C \frac{\sqrt{2} \cos \theta - 2}{3 - 2\sqrt{2} \cos \theta} \quad (3)$$

where α_G represents the angular acceleration of the Geneva wheel, ω_G and ω_C represents the angular velocity of the Geneva and crank wheels respectively. θ and φ represents the angular displacements of the crank and Geneva wheels respectively.

B. Conceptual and Detail Design Drawings

The pictorial rendering of the model for the conceptual design of the Geneva stop mechanism as well as an exploded diagram showing and listing of the components was done using Professional – Engineer (Pro-E) design software and is presented in Fig. 4. Provision was made for transparent perspex or glass cover to enable viewing while keeping dust away.

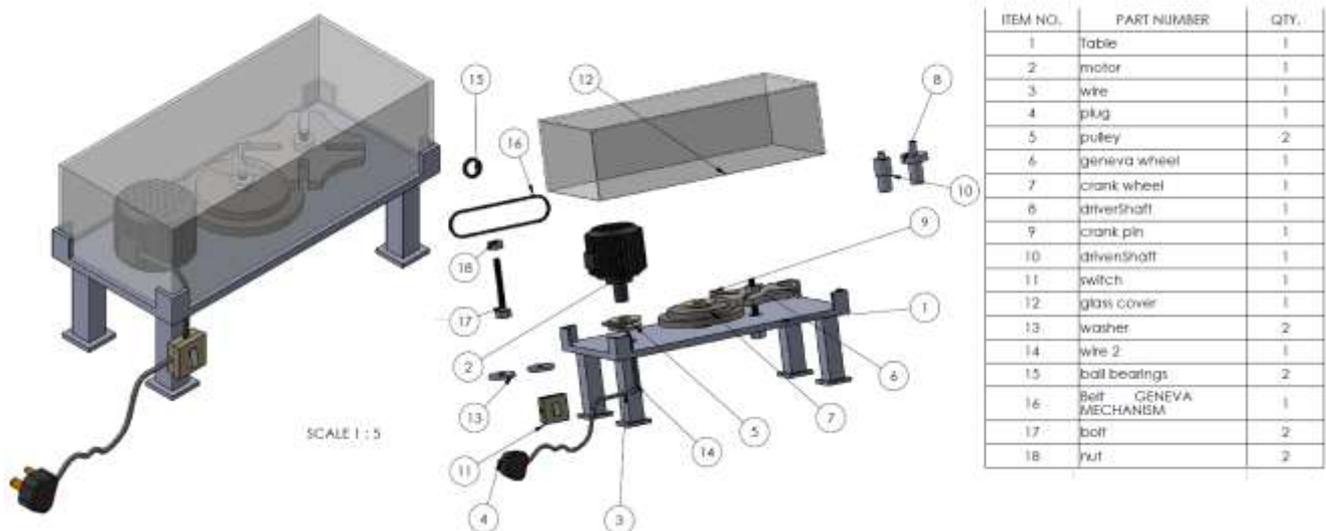


Fig. 4 Conceptual and exploded view of the Geneva stop mechanism

Detail drawings of the assembled component that could be used for physical construction purposes are presented in Fig. 5.

Other specifications used in the design are:

| | |
|-------------------------------|----------|
| Number of slots (n) | 4 |
| Radius of crank wheel | 65 mm |
| Roller pin diameter | 10 mm |
| Half angle subtended by slots | 45° |
| Length, L | 91.92 mm |

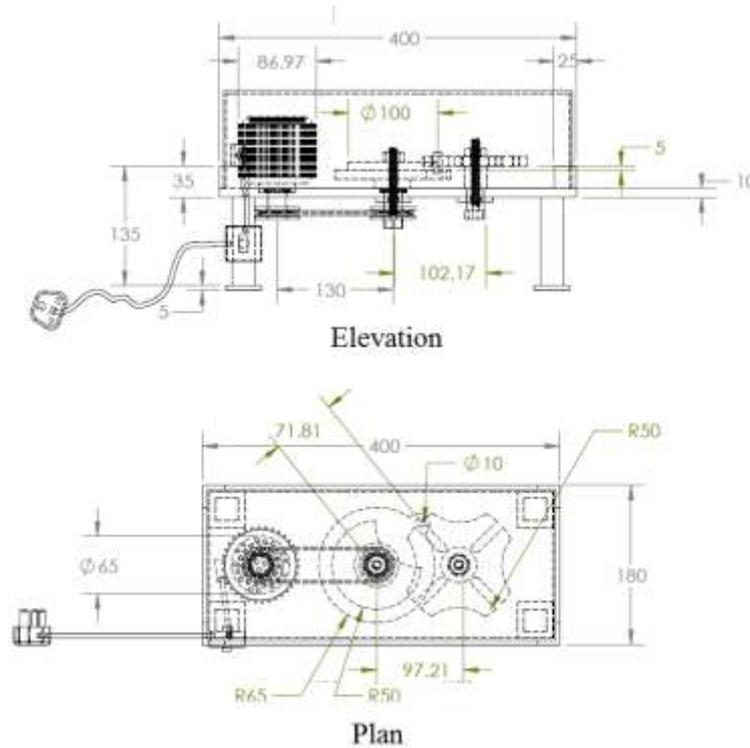


Fig. 5 Detailed drawings of the assembled components

III. KINEMATIC AND STRESS ANALYSIS

Kinematics analysis of a four external slots Geneva stop mechanism is the subject of this work. Relative motion analyses between the two wheels have been analyzed in Equations (1)-(3) with numerical experimentation based on constant crank wheel speed of $N_c = 12 \text{ rev/min}$ ($\omega_c \approx 1.257 \text{ rad/sec}$) was carried out using MATLAB. Stress analysis to investigate for critically loaded points on the two wheels was executed based on finite element analysis with generous mesh used SOLIDWORKS.

A. Results of Kinematic Analysis

Experimentation for relative angular motions of the wheels is presented in Fig. 6. The motion in each slot represented by continuous thick lines of different colours, while the broken lines shows the dwell periods. Each motion period of the Geneva wheel is equivalent to a quarter of the crank wheel’s motion.

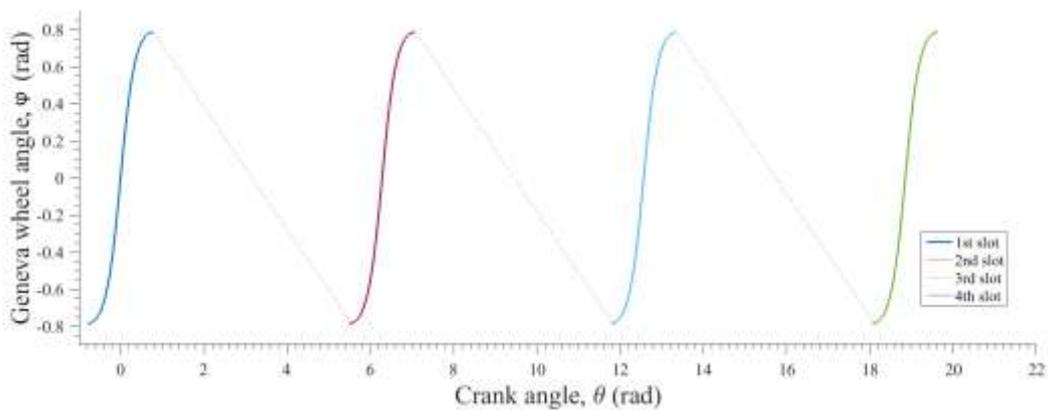


Fig. 6 Relative angular displacements of the Geneva and crank wheels

The corresponding angular velocity relationship between the wheels is shown in Fig. 7. The motion starts from zero velocity, reaching a spiked peak of about 3 rad/sec before returning to zero at the exit of the crank pin from the slot. The sharpness of the spike edge characterizes “jerking” experienced in the mechanism’s motion. Achievement of smoother motion is achieved by designing to reduce the sharpness of the spike and other discontinuities.

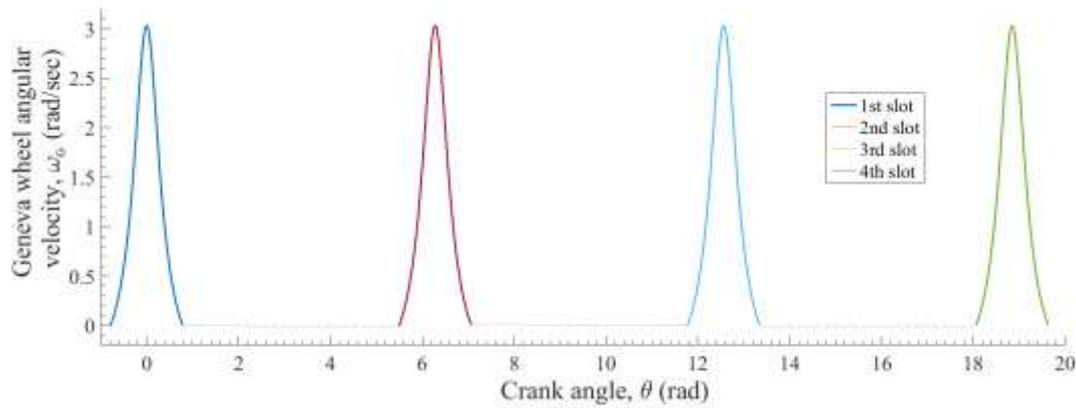


Fig. 7 Relative angular velocity of the Geneva and crank wheels

Fig. 8 presents a picture of the relative motion of the wheels in terms of acceleration. The Geneva wheel’s angular acceleration ranged between $\pm 8.5161 \text{ rad/sec}^2$ and occurs at crank angles of $\pm 0.19, 6.10, 6.47, 12.38, 12.75, 18.66$ and 19.04 radians respectively. These are also the positions of maximum angular velocities of the Geneva wheel.

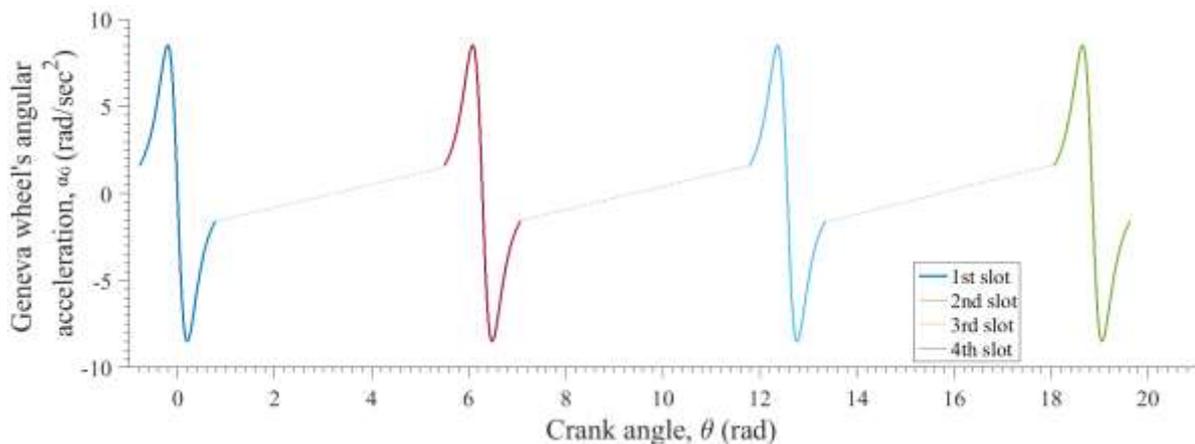


Fig. 8 Relative angular acceleration of the Geneva and crank wheels

B. Results of Stress Analysis

Finite element based stress analysis was carried out on the crank and Geneva wheels using linear elastic isotropic solid mesh model. The analysis was carried out for a nominal temperature of 25°C based on the Von Mises failure criterion. The crank wheel solid member was constrained to turn about a drive shaft passing through its center based on a motion propelled by a resultant force of 407.58 N which yielded an applied torque of 96.77 Nm . The Geneva wheel was also constrained to rotate about its own shaft, passing through its center. The maximum resultant force driving the Geneva wheel’s motion reached a maximum of 126.29 N . A maximum normal force of 126.28 N was acted perpendicularly against the face of the Geneva wheel slots due to the motion of the crank pin.

AISI 1060 high carbon steel alloy was selected for the construction because the high strength requirement. The properties of the material selected include; yield strength of 2.757 Nm^{-2} , Poisson ratio of 0.33 , elastic modulus of $6.9 \times 10^{10} \text{ Nm}^{-2}$, shear modulus of $2.7 \times 10^{10} \text{ Nm}^{-2}$ and thermal expansion coefficient of $2.4 \times 10^{-5} \text{ K}^{-1}$. The crank wheel after construction will have a weight of 5.172 N and a volume of $1.955 \times 10^{-4} \text{ m}^3$, while the Geneva wheel will have a weight of 2.333 N and volume $8.817 \times 10^{-5} \text{ m}^3$. Fig. 9 shows the mesh arrangement for the two wheels while Table 1 presents their mesh parameters.

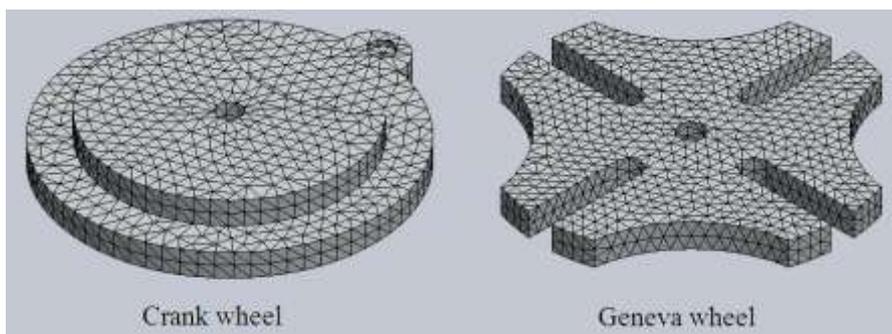


Fig. 9 Solid mesh arrangement on the crank and Geneva wheels

TABLE 1 MESH PROPERTIES FOR THE WHEELS

| | Crank wheel | Geneva wheel |
|--------------------------|-------------|--------------|
| Mesh element size | 5.805 mm | 4.452 mm |
| Total nodes | 13030 | 13573 |
| Total number of elements | 8029 | 8043 |

Fig. 10 presents the stress distribution pattern for the two wheels. The minimum and maximum stresses of 27.209 kNm^{-2} and 162.925 MNm^{-2} respectively. The minimum stress was prevalent throughout the crank wheel except for the shaft position where there was a stress jump to the maximum value. This was expected and stress concentration should occur around the key. This information is vital and was useful in designing the crank's shaft and its key.

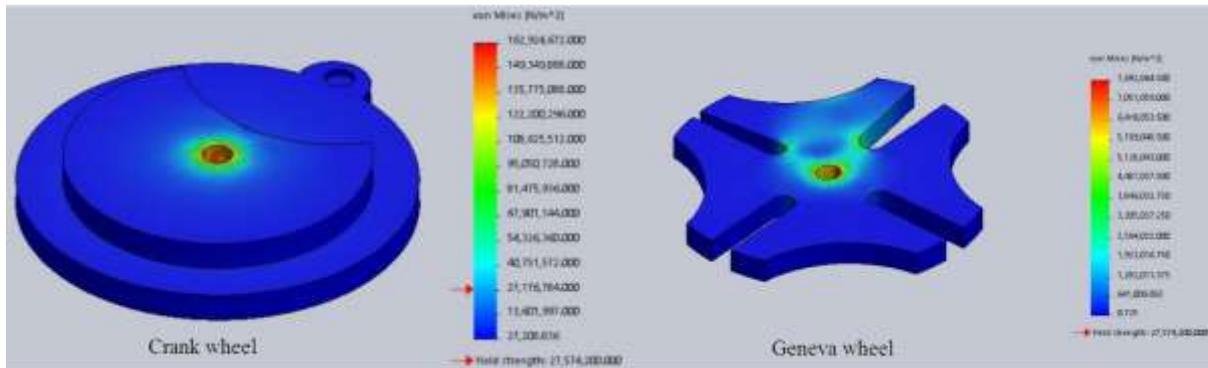


Fig. 10 Stress distribution on the crank and Geneva wheels

There was a similar trend for the Geneva wheel but the stress distribution showed that the necks of the slots form critical points as well. Its minimum and maximum stress values were 0.725 Nm^{-2} and 7.692 MNm^{-2} respectively. It is noteworthy that the order of stress values of the crank wheel greatly exceeds those of the Geneva wheel.

The motion of the Geneva wheel with the crank pin engaged in one its slot was simulated in COMSOL Multi-Physics to show three instances during the engagement of the pin, this is shown in Fig. 11. The first instance is when the pin is at the tip, entering the slot. The second instance is when it reaches the mid-point in the slot and the last instance is when the pin reaches the extreme position in the slot. The colour legend accompanying each figure indicates the point of action of the load due to the crank pin.

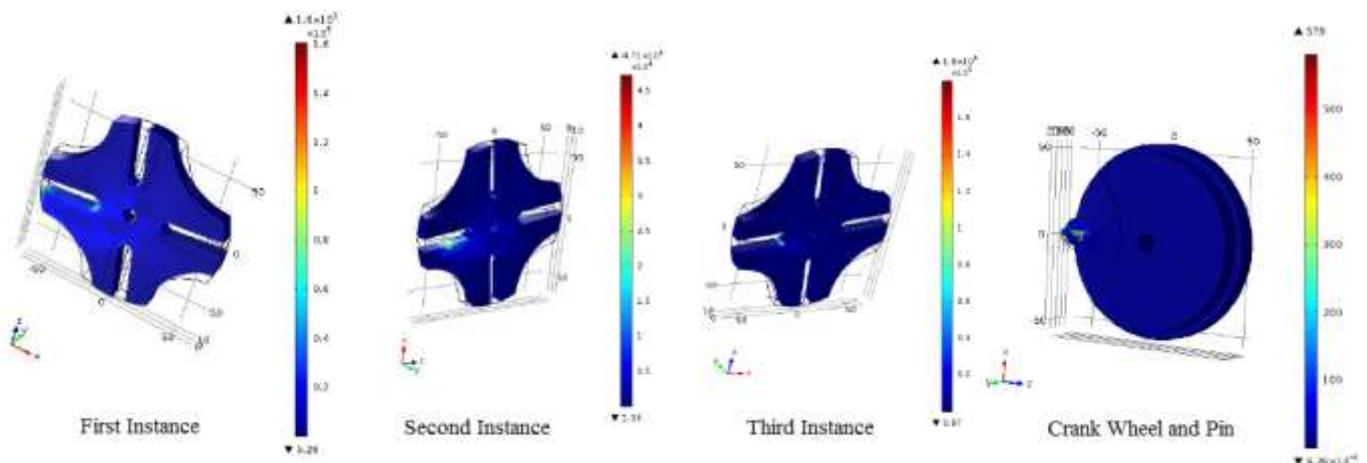


Fig. 11 Stress distribution as the pin traverses within the Geneva slot

Expectedly, the pin was stressed most on its sides as it is constrained to move through the slot. It was subjected to compressive stresses due to reactions from the walls of the slot. Optimal design decision requires to be taken to ascertain the proper diameter of the pin to be used in order to avoid resist deformation.

IV. CONCLUSION

Kinematic analysis and computer aided mechanical engineering design of a four – external slots Geneva stop mechanism designed for use in classroom illustration and teaching of kinematics of intermittent motion has been provided. A finite element based stress analysis was carried out on the crank and Geneva wheels using linear elastic isotropic solid mesh model. Static structural analysis carried out on the two wheels show that the material AISI 1060 selected is suitable. High stresses occur around the point of fixed boundary condition of the wheels (that is, the shaft at the center of the wheels), this is expected

because stress concentration is normally the point of connection of the wheels to the shaft (that is, the key). The spikes in the velocity and acceleration diagrams show the source of the jerks in the motion of the Geneva wheel that will be experienced. Elimination of this source of discontinuity in motion will be the subject of subsequent works since the focus of this work is mainly to design a teaching aid material basic mechanisms and applied mechanics courses for undergraduate engineering degree programmes.

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