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Contact temperature analysis of the classical Geneva mechanism through numerical methods

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ABSTRACT

In this study, the flash temperature of the classical Geneva mechanism has been studied using the numerical method. An excessive sliding motion between the wheel and pin leads to the generation of heat at the contact surfaces, which raises the surface temperature of the two contacting bodies. The flash temperature has been anticipated for different loading cases and coefficients of friction. The analytical Blok equation results are compared with the finite element results, and the results are satisfactory for maximum temperature. Based on the simulation results, the maximum contact temperature occurred in the driving crankpin within 10-15° and 70–75° of the angular position, and as the load increased the contact temperature also increases. In addition, the angular velocity increment produced higher temperatures than the torque load increments.

1. Introduction

Geneva mechanisms are commonly used as an indexing mechanism where intermittent motion is required by transforming rotary motion into intermittent motion. The rotating drive crankpin is usually equipped with a pin that reaches into a slot located in the driven wheel that advances it in one step at a time. The Geneva mechanism is extensively used in high [1,2] - and low-speed [3] automatic machinery. It is employed for counting mechanisms, indexing [4–6], sequencing, motion-picture mechanisms, feed mechanisms, and mechanical watches, and others with smooth operation limitations due to discontinuity in the acceleration at the start and end of the intermittent motion.

The performance and life of the Geneva mechanism are adversely affected by discontinuity at the beginning and end contact points and the high contact stress between the wheel slots and the drive crankpin. The constant acceleration of the rotating crankpin is transmitted to the Geneva wheel with an impact due to the engagement and inertial force, this leads to jerks and undesirable vibrations in the mechanism. A lot of research has been conducted and proposed several methods to decrease the wheel acceleration in order to reduce the inertial forces and consequent wear. The curved slot is one way to reduce the acceleration, but it increases the deceleration and the wear on the slot [7,8]. The other approaches involve substantial changes in the slot design, such as using different radii of curvature of the entry and exit curves, using spring elements between the driving pin and the slot, and using grooved cams to drive the crankpin in a specific path [7,9]. Previous works have been conducted to minimize the jerk and undesirable vibration; however, limited studies are reported on the flash temperature that can be produced due to excessive sliding motion between the Geneva wheel and the driving pin.

The determination of the flash temperature of the contacting body will pave the way to anticipate how and where the wear will occur. The temperature affects the performance and the life of the Geneva wheel. Pearson et al. [10] investigated that temperature affects the wear rate and distribution. Surface conjunction temperature is the temperature at the interface of two contact and mutually sliding solid objects. For any specific part of the sliding surface, frictional temperature rises are very short duration, and the temperature generated is called flash temperature, it was originally formulated by Blok in 1937 and developed further by Jaeger 1944 and Archard in 1958 [11]. Most surface temperature analyses of contacting bodies have been based on the pioneering work of

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Blok [12] and Jaeger [13]. They formulate different equations based on the value of a non-dimensional number, which is called pecelet. This non-dimensional number is a function of surface velocity, contact width, and thermal diffusivity. The finite element and analytical (Block equations) are the two common methods used extensively to determine the flash and contacting temperature of different contacting bodies when there is a relative motion in between them. Chang et al. [14] use the finite element method to determine the surface temperature of the spur gear. Taburdagitan and Akkok [15] use finite element and analytical methods to determine the flash temperature of the gear, and they found that the results are satisfactory with the experimental results specifically within the highest point of single tooth contact found by Tobe and his colleagues. Patrir and Cheng [13] also use the finite element method to determine the temperature distribution of spur gear.

A finite element program is one of the commonly used methods which can very efficiently and accurately predict surface temperatures and temperature distributions in sliding systems. Kennedy [16] used a finite element program for the analysis temperature of the sliding objects. The method was applied to two different sliding systems: dry or boundary lubricated sleeve bearings and rubbing between rotating and stationary components of a labyrinth gas path seal. Comparisons between predicted and experimentally measured temperatures showed that very good agreement was obtained in both cases. Temperature predictions were significantly influenced by the velocity of the moving component, even when that velocity was not maximum. The radius of curvature of the gears and cylinders has finite so that the equivalent or reduced radius of curvature can be determined directly. But the Geneva wheel slot surface has a flat surface and its radius of curvature is infinite, while the radius of curvature of the driving crankpin is finite, therefore the equivalent radius is equaled with the radius of curvature of the pin. The contact exhibited in the Geneva mechanism can be modeled as the contact between the cylindrical object with the flat surfaces with an infinite radius of curvature. There is some literature that works on the determination of the flash temperature of cylindrical and flat body interaction. For example, Qin et al. [17] studied that the effect of local roughness on the friction and flash temperature distribution using the finite element method of cylinder-on-flat surface configuration.

In this paper, both finite element and analytical methods are employed to determine the contact temperature distribution. There are three loading cases to analyze the effects on the surface temperature value at the instant of contact. The analytical analysis is done for load case 1 and compared with the finite element results, and the results have been compared and it found that they are in good agreement. Finally, the prediction of maximum contact temperature has been done using finite element methods.

2. Design and modeling of Geneva wheel mechanism

There are different kinds of Geneva mechanisms such as external, internal and spherical Geneva mechanisms are some of them. A 3D model of the external Geneva mechanism is presented in Fig. 1. The basic modeling parameters, taken from published materials [18], are given in Table 1.

The Geneva wheel is not in motion when the pin is not in the slot and it will rotate when the pin and slot engage and we call this active phase as presented in Fig. 2. Using sine law, the following relations is obtained:

$$\frac{r_1}{\sin\theta_2} = \frac{c}{\sin[\pi - (\theta_1 + \theta_2)]} \tag{1}$$

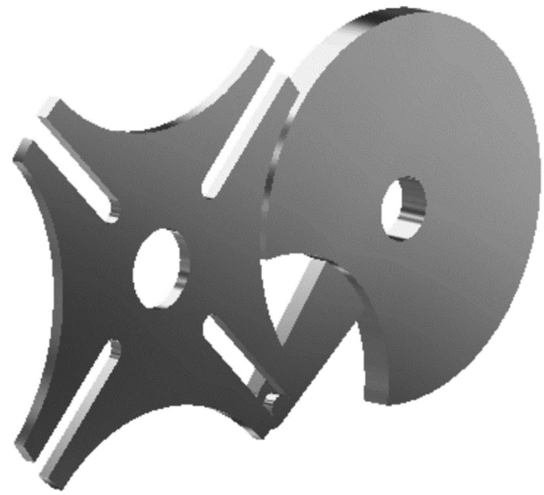

Fig. 1. 3D Geneva mechanism model.

Table 1
Design parameter of external Geneva mechanism.

Parameter	Value
Crankpin, $\boldsymbol{r}_1$	75 mm
Thickness of wheel, $\boldsymbol{t}$	4 mm
Number of slots, n	4
Drive pin radius, $\boldsymbol{R}_1$	4 mm

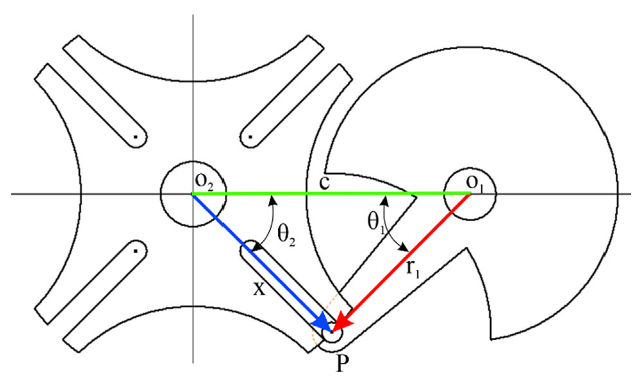

Fig. 2. Scheme diagram for determining angular position and speed.

For n number of slots, the active phase angle of the wheel (θ_2) is in the interval ± $\pi/2$. The angular position of the crankpin (θ_1) and Geneva wheel (θ_2) in terms of the Cartesian coordinates (x_p, y_p) are given in Eq. (2) (a) and (b) respectively.

$$\theta_1 = \pi - \tan^{-1}\left(\frac{y_p}{x_p}\right) \quad \text{and} \quad \theta_2 = \tan^{-1}\left(\frac{y_p}{c + x_p}\right) \tag{2}$$

From Fig. 2, the length of instant contact point P from center of wheel and angular position is given by Eqs. 3 (a) and (b) respectively.

$$x = \sqrt{r_1^2 + c^2 - 2r_1 c cos\theta_1} \tag{3a}$$

$$\theta_2 = \sin^{-1}\left(\frac{r_1}{x} sin\theta_1\right) \tag{3b}$$

The surface velocity of the Geneva wheel and driving crankpin was obtained from the above equations, and this enable to calculate the sliding velocity. Using elementary kinematics, the velocity of pin P is given by:

$$\upsilon_P = \omega_1 \times r_{P/o1} \tag{4}$$

The same vector υ_P may be obtained upon the driven wheel and expressed as

$$\upsilon_P = \upsilon_P' + V_{P/W} \tag{5}$$

where $\upsilon_P' = \omega_2 \times r_{P/o2}$ is the velocity of the point of Geneva wheel body which is instantaneously coincident with the pinpoint P, while $V_{P/W} = \upsilon_{P/W}\widehat{u_{P/o2}}$ is the velocity of point P relative to the wheel, $r_{P/o2}$ is the vector pointed from O_2 to point P, $\upsilon_{P/W} = |V_{P/W}|$, and $\widehat{u_{P/o2}}$ is a unit vector pointed from P to O_1. This, equating Eq. (4) with Eq. (5) yields

$$\omega_1 \times r_{P/o1} = \omega_2 \times r_{P/o2} + V''\widehat{u_{P/o2}} \tag{6}$$

It is straightforward to show that the above expression leads to a linear system of two equations in the unknowns ω_2 and $V_{P/W}$. Solving such a system, one obtains the general expressions

$$\omega_2 = \omega_1 r_1 \cos(\theta_1 + \theta_2)/x \quad and \quad V_{P/W} = \omega_1 r_1 \sin(\theta_1 + \theta_2) \tag{7}$$

3. Analytical modeling of contact parameters

There are three load cases in this paper used to examine the contact temperature of the wheel and crankpin, these are: load case 1 is the torque load is 7000 Nmm and 5.34 rad/s angular velocity, load case 2 is double torque load with the previous angular velocity, and load case 3 is double the angular velocity with the 7000 Nmm torque load. For the above three loading conditions the analysis is conducted, and it has been also analyzed which load case has a strong effect on the maximum contact temperature of the Geneva mechanism. The angular velocity of the wheel and the sliding or relative velocity of the pin with respect to the wheel are shown in Fig. 3.

The maximum angular velocity has occurred when the pin is engaged with the end of the slot at 45° for the four slot Geneva wheel as shown in Fig. 3 (a). The angular velocity is very small at the beginning and end of the interaction of the pin and the wheel slot. The sliding velocity of the pin with respect to the wheel slot is shown in Fig. 3 (b), and the surface velocity of the Geneva wheel is zero at 45° of the active phase of the angular position. The maximum sliding velocity is occurred at the beginning and end of the contact of the pin and wheel, in addition, the surface velocity of the wheel is varied throughout the angular position due to the variation of the angular speed and the instantaneous point of contact radius.

The normal load is varied with the angular position of the contact point. The torque load at the contact point is decomposed into tangential (W_T), radial (W_R), and normal load (W_N) as shown in Fig. 4. The normal contact load is given by Eq. (8), and the load distribution for 7000 Nmm with the angular position is shown in Fig. 5. The maximum load is obtained at half of the angular position of the active phase, the minimum load for load case 1 is about 35 N.

$$W_N = \frac{T}{r_1}\cos(\theta_1 + \theta_2) \tag{8}$$

The contact area between two parallel cylinders is circumscribed by a narrow rectangle. The contact width and the maximum contact pressure are given in Eq. (9) (a), and (b) respectively. The contact pressure is the ratio of the normal load to the true contact area.

$$b = \sqrt{\frac{4W_N R'}{\pi l E'}}; P_{max} = \frac{W_N}{\pi b l} \tag{9}$$

The above formula also is used for cylindrical and flat surface contact bodies with the infinite radius of curvature of the flat surface (wheel slotted surface). The radius of curvatures for the Geneva mechanism is expressed as $R_{1X} = \infty, R_{1Y} = R_1, R_{2X} = R_{2Y} = \infty$. The reduced radius of curvature of the Geneva mechanism is equal with the radius of the pin and it is given in Eq. (10) (a), while the reduced elastic of modules is given in Eq. (10) (b).

$$\frac{1}{R_1} = \frac{1}{R_X} + \frac{1}{R_Y} = \frac{1}{R_1} + \frac{1}{\infty} + \frac{1}{\infty} + \frac{1}{\infty} \quad and \quad \frac{1}{E'} = \frac{1}{2}\left[\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right] \tag{10}$$

The contact width and maximum contact pressure are shown in Fig. 6 (a) and (b) respectively. The maximum contact width and the contact pressure are found at 45° of the angular position of the active phase. The material and physical properties of the wheel and crankpin are given in Table 2.

There is no single algebraic equation giving the flash temperature for the whole range of surface velocities. Peclet number is a non-dimensional measure of the speed at which the heat source moves across the surface, and it has been introduced as a criterion

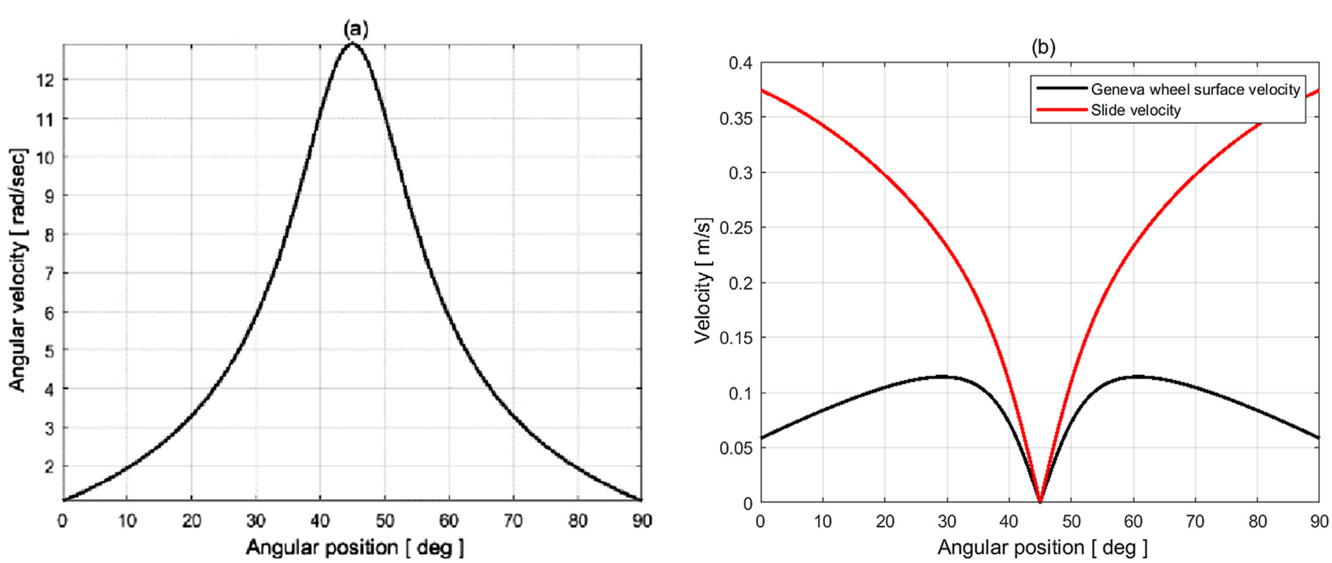

Fig. 3. a) Angular velocity of Geneva wheel, and b) sliding linear velocities.

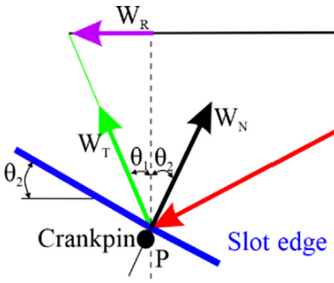

Fig. 4. Scheme diagram for force decomposition.

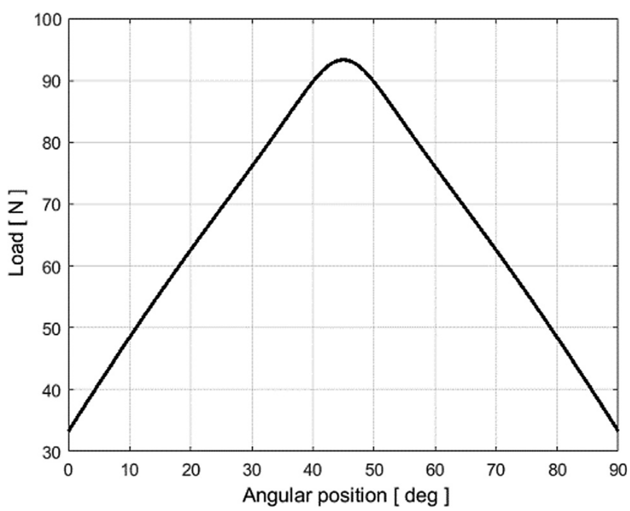

Fig. 5. Normal load distribution for load case 1.

allowing the differentiation between various speed regimes. The Peclet number is an indicator of the heat penetration into the bulk of the contacting solid bodies, which means it describes whether there is sufficient time for the surface temperature distribution of the contact to diffuse into the stationary solid.

$$L_e = \frac{bU_{1,2}}{2\chi_{1,2}} \tag{11}$$

Table 2
Material properties of the steel SAE-AISI 1045 at 20 °C [15].

Parameter	Value
Mass density, ρ	7.86×10^{-6} kg/mm^3
Modulus of elasticity, $\mathbf{E}_1$ & $\mathbf{E}_2$	207×10^3 N/mm^2
Poisson ration, ν	0.3
Specific heat, Cp	485×10^3 (Nmm)/kg°C
Thermal conductivity, κ	42.3 (Nmm)/mm°C

The value of half heat sources width (b) has been determined previously. Thermal diffusivity of the pinion and gear material is calculated using $\chi_{1,2} = \kappa/\rho C_{P1,2}$, it gives 11.0962×10^{-6} m^2/s.

Sliding friction is the dominant mechanism for heat generation. Heat is generated by sliding friction between the pin and slot surfaces. The temperature distribution is proportional to the distribution of contact pressure and sliding velocity. The flash temperature of the pin and wheel slot contact during an active phase is calculated by Blok's contact temperature theory. The contact temperature is the sum of the maximum flash temperature and bulk temperature of the contacting bodies before it enters the contact zone. The amount of heat generation is given by Eq. (12), and it shows that the heat generation is proportional to the friction coefficient, normal load and the sliding velocity of contacting bodies.

$$Q = \mu W_N |V_1 - V_2| \tag{12}$$

Fig. 7 (a) shows that the peclet number of the pin is increased slowly from 0.35 m/s at the beginning of the interaction of wheel slot to a maximum value of 0.58 m/s and after that, it decreased gently until the last point of the active phase its value to the initial pecelet number value. While the peclet number of the wheel increases gradually from the point of contact till it reaches to 35° of the angular position and automatically increases to 0 at 45° of angular position. When the angular position increases further the pecelet number start to increase again to 0.15 pecelet number and then it decreased to 0.5 at the last point of the active phase.

Fig. 7 (b) shows that the heat generation is increased from 3.8 W to a maximum value of 5.6 W at the 23° of angular position and decreased sharply to zero at 45° and when angular position increases, the amount of heat generation is also increased to the maximum value and gain it decreases. It clearly is shown that the maximum heat generation has occurred at the 23 and 63° of

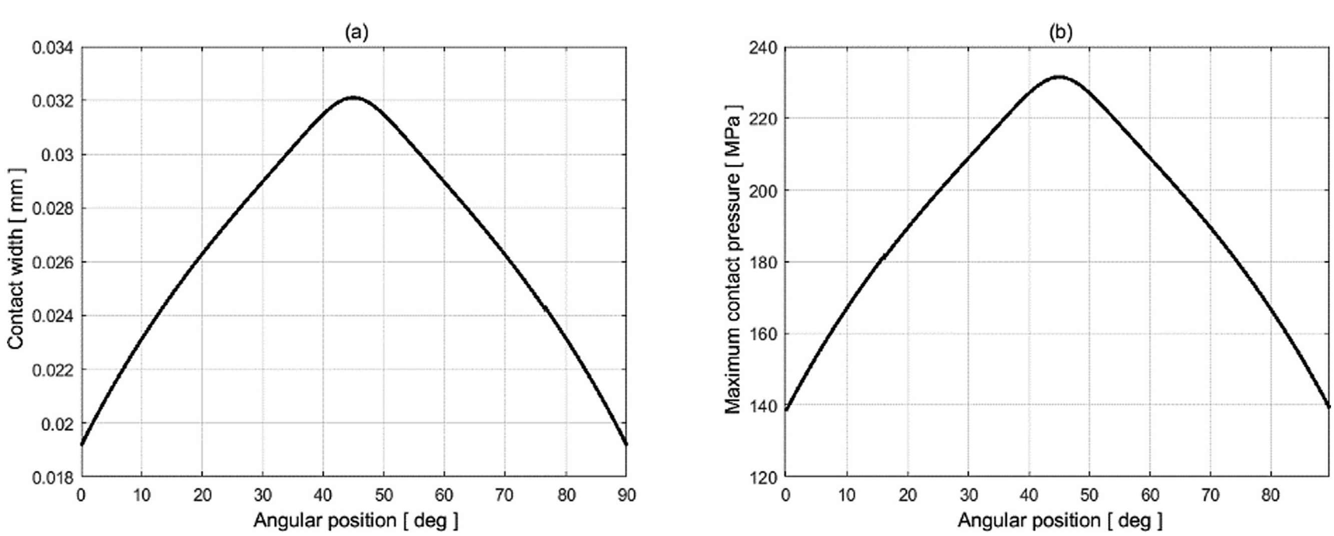

Fig. 6. (a) Half contact width, and (b) maximum contact pressure.

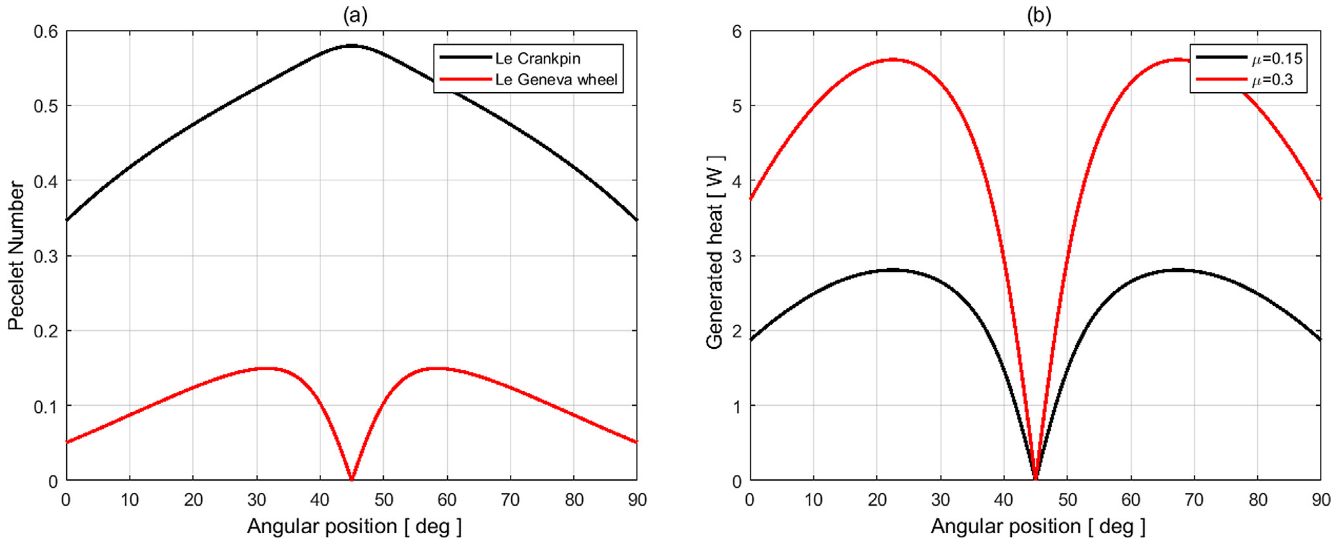

Fig. 7. a) Pecelet number, and b) Generated heat due to friction for load case 1.

the active phase. In addition to this as the coefficient of friction increases the amount of heat generation will also increase.

Flash temperature formula for line contacts for various velocity ranges is determined by using different equations depending on the value pecelet number [19]. The maximum pecelet number found in Fig. 7 (a) is around 0.58, therefore, for $0.1 < L_e < 5$, the flash temperature is given by Eq. (13) as:

$$T_f = \frac{0.159 C_4 \mu W_N |V_1 - V_2|}{KL} \left(\frac{\chi}{U_{1,2} b} \right) \quad (13)$$

The value constant C_4 of is determined from Fig. 8 as 2.5. Therefore, the value flash temperature can be determined for different loading and coefficient of friction.

4. Numerical analysis of the maximum contact temperature

The maximum contact temperature of the Geneva mechanism has been determined using a coupled temperature displacement analysis of quarter model of wheel and the driving crankpin as shown in Fig. 9 using ABAQUS software. Both contacting bodies are free to rotate about the rotational direction at the center. There are two steps created for setting the boundary and loading conditions, and in the first step the wheel is subjected to a torque based on the load case, and the speed will be granted in the second step. The initial temperature or operating temperature is added in the initial step as an initial condition.

The analysis contains a coupled thermal displacement element type C3D8T (8-node thermally coupled brick, trilinear displacement, and temperature) element type has been used for coupled temperature analysis and it has 23,645 number of elements and 40,427 number of nodes. A fine mesh is deployed at the contacting surfaces with 0.5 mm mesh size, this will enhance the results and the running cost.

The simulation results are shown in Fig. 10 (a) and (b) for wheel and crankpin respectively under load case 1 with 0.15 coefficient of friction, it indicates that the maximum contact conjunction temperature occurred at a flat surface around the circular curved faces of the wheel. The maximum temperature is observed in the crankpin as shown in Fig. 10 (b). For 0.3 coefficient of friction, the maximum contact temperature is shown in Fig. 11 (a) and (b). In both situations, the maximum contact temperature is found at the bottom edge of the wheel slot and pin portions.

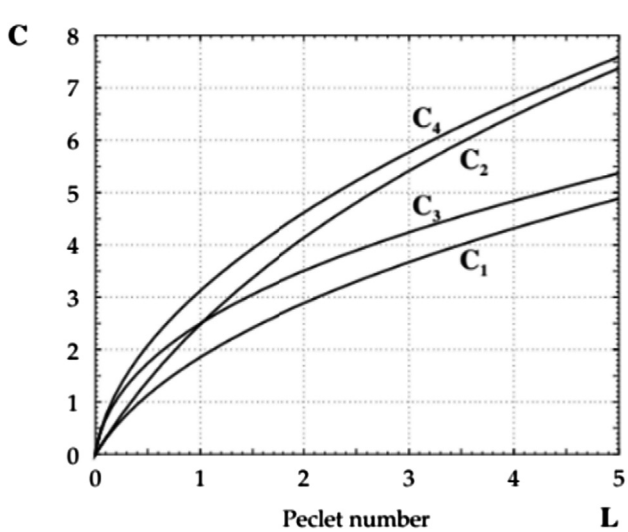

Fig. 8. Diagram for evaluation of constants 'C' required in flash temperature calculations for the intermediate velocity range [13].

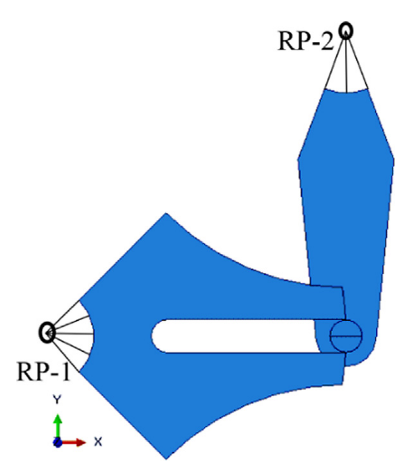

Fig. 9. Quarter Geneva mechanism model for load and boundary conditions.

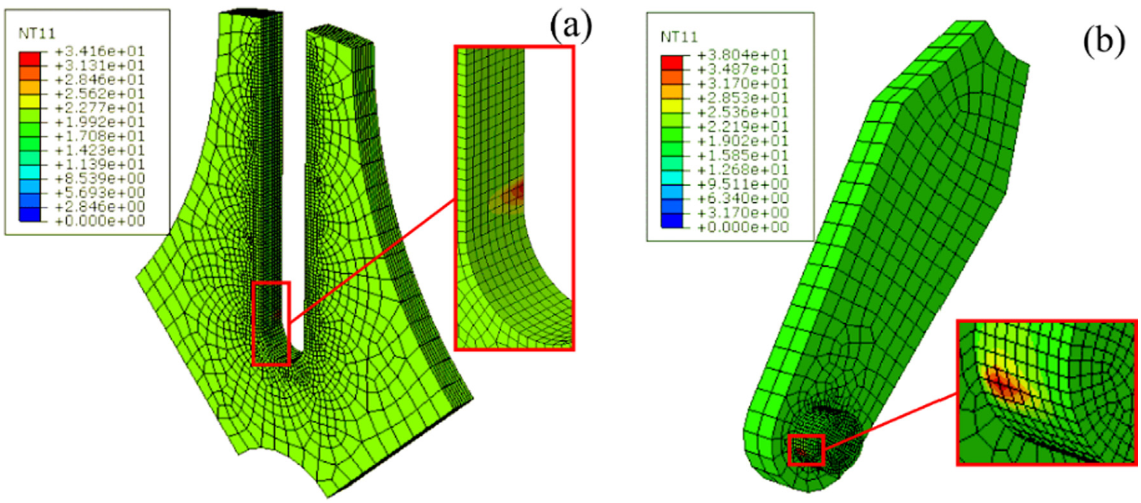

Fig. 10. Maximum contact nodal temperatures contour view with: $= 0.15 T_{Gw} = 7000$ Nmm, and $\omega = 5.34$ rad/s: a) Geneva wheel, and b) crankpin.

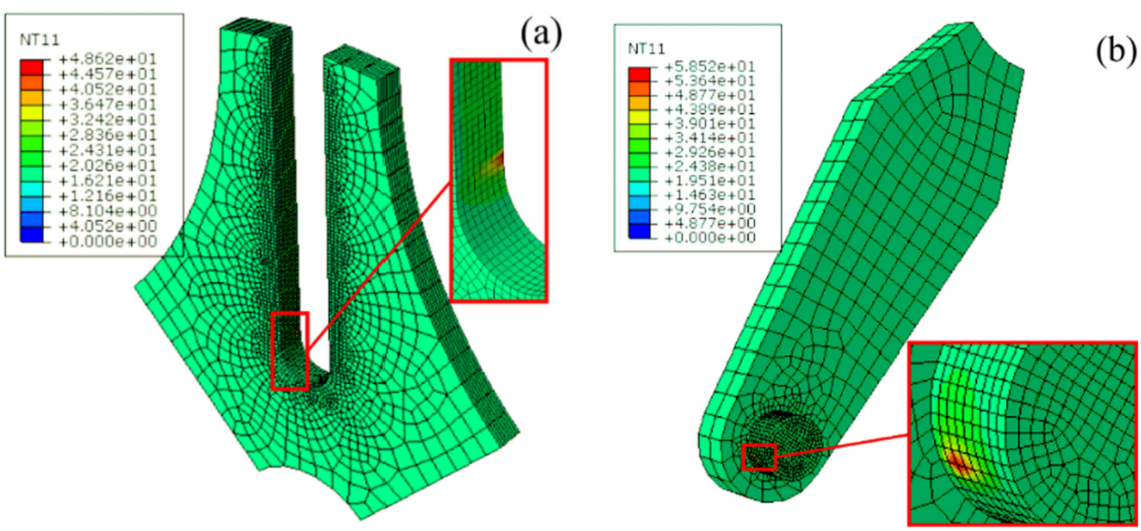

Fig. 11. Maximum contact nodal temperatures contour view with: $\mu = 0.3, T_{Gw} = 7000$ Nmm, and $\omega = 5.34$ rad/s: a) Geneva wheel, and b) crankpin.

Fig. 12 depicted the finite element and analytical Block equation maximum contact temperature results. The figure shows that the contact temperature is minimum at 45° of the angular position, while the maximum contact temperature is occurred at within 10–15 and 70–75° of the angular position. The two methods are in good agreement within 46 to 70° of angular position, in other positions there is a variation, and however, both methods show the same angular position where the maximum temperature will occur. The coefficient of friction increases the contact temperature. The flash temperature is zero at 45° of angular position in the analytical method, but not for the finite element method. This may result from, the two contacting bodies become confined to each other so that the contact area will increase and this, in turn, increases the friction.

The finite element analysis results are shown in Figs. 13–15 (a) and (b). The simulation results show that with the load 7000 Nmm and 5.34 m/s loading conditions the maximum contact temperature is found on the crankpin. Fig. 13 (a) - (b) shows the contact temperature of 0.15 and 0.3 coefficient of friction. The contact temperature is doubled and tripled of the operating temperature of the

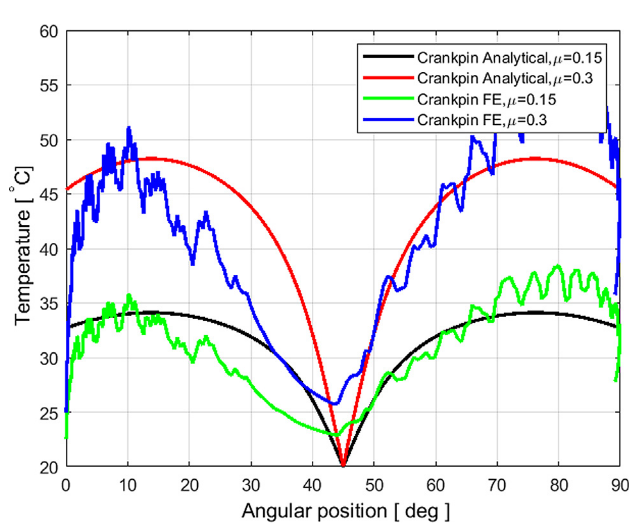

Fig. 12. Comparison of contact temperatures with: $T_{Gw} = 7000$ Nmm, and $\omega = 5.34$ rad/s.

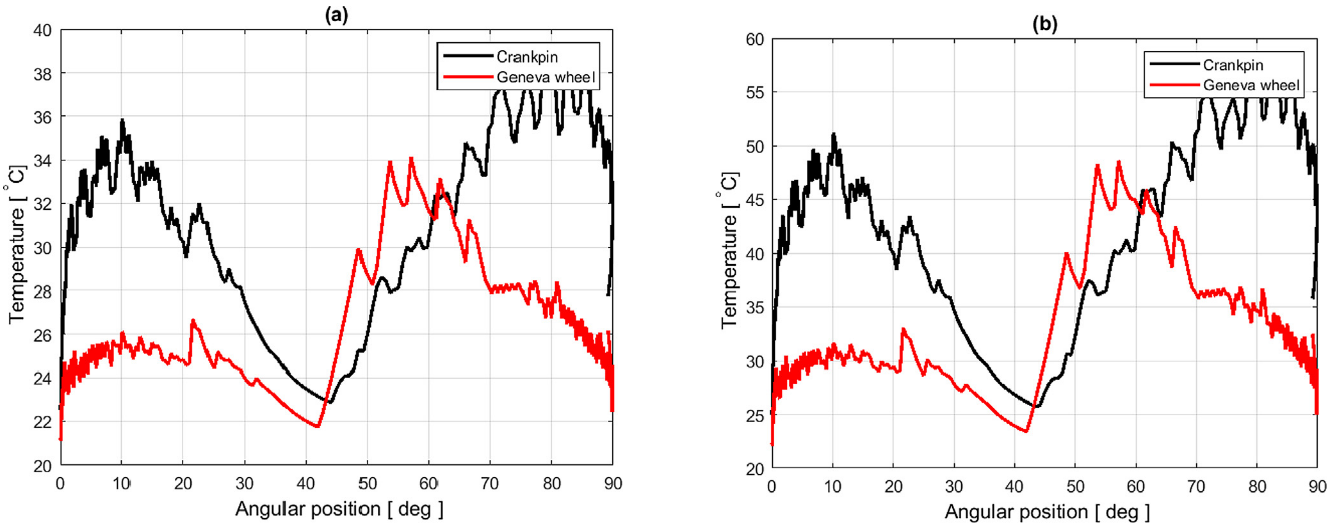

Fig. 13. Contact temperatures with : $T_{Gw} = 7000$ Nmm, and $\omega = 5.34$ rad/s: a)$\mu = 0.15$, and b) $\mu = 0.3$.

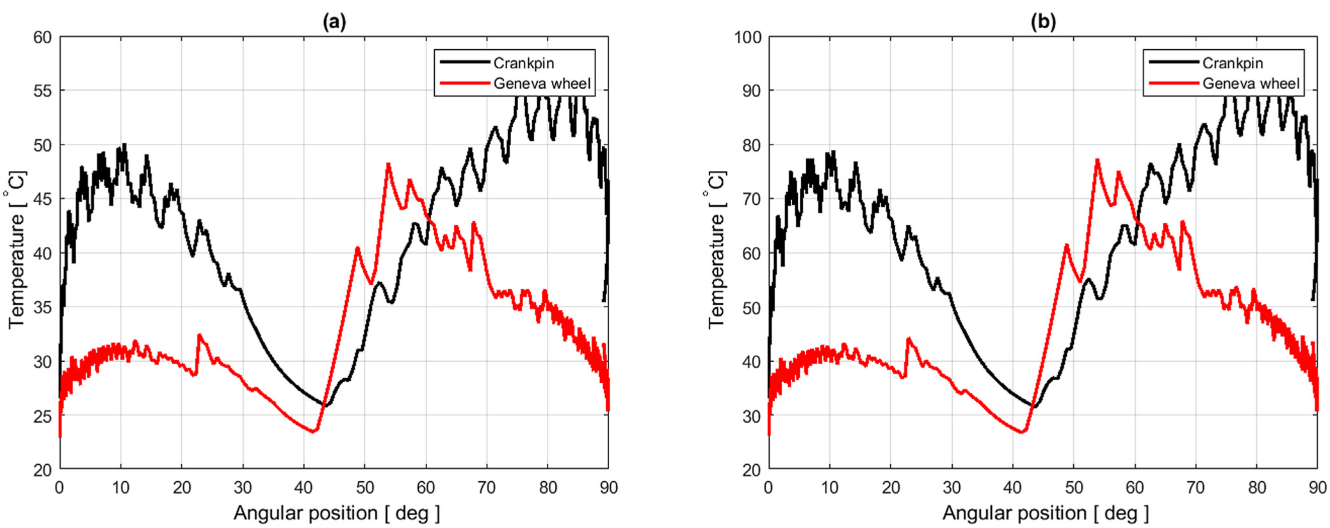

Fig. 14. Contact temperatures with : $T_{Gw} = 14000$ Nmm, and $\omega = 5.34$ rad/s: a) $\mu = 0.15$, and b) $\mu = 0.3$.

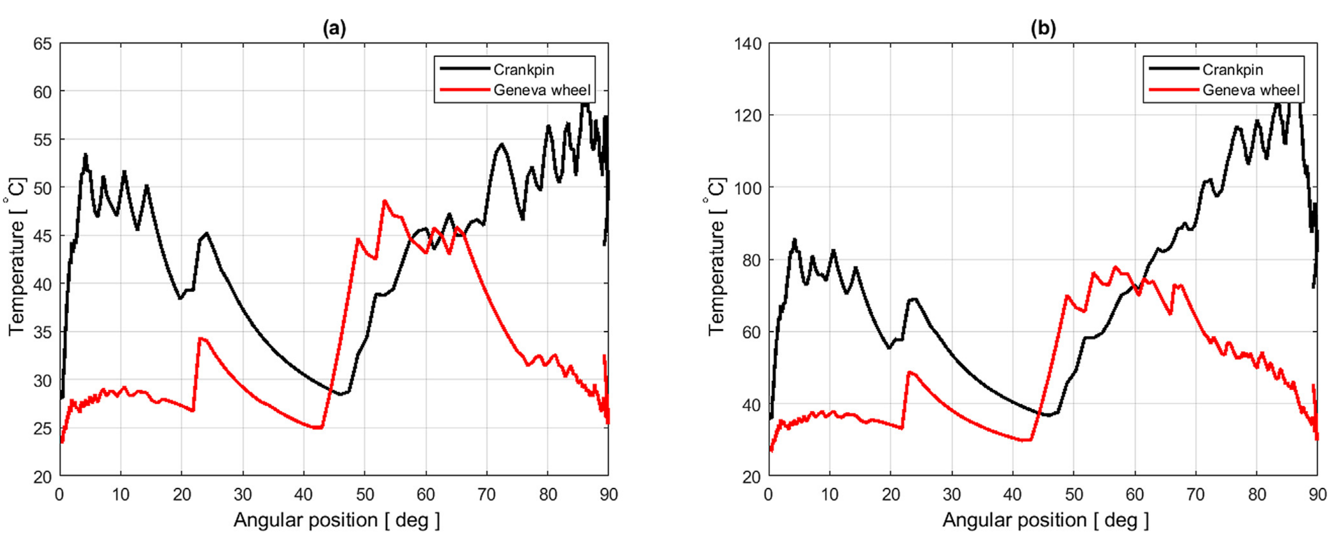

Fig. 15. Contact temperatures with : $T_{Gw} = 7000$ Nmm, and $\omega = 10.68$ rad/s: a) $\mu = 0.15$, and b) $\mu = 0.3$.

Geneva mechanism for 0.15 and 0.3 coefficient of friction respectively.

When the torque load is increased by 100% with 5.34 rad/s angular velocity, the maximum contact temperature is almost tripled the operating temperature for 0.15 coefficient of friction as shown in Fig. 14 (a). The contact temperature is increased to $95^{\circ}C$ for 0.3 coefficient of friction as shown in Fig. 14 (b). The Geneva wheel has a lower contact temperature than the crankpin. Fig. 14 (a) – (b) shows the contact temperature distribution for a 100% increment of the angular velocity of the driving pin and 700 Nmm torque load. Fig. 14 (a) shows that the contact temperature slightly varies from 100% increment torque load and 5.34 rad/s angular velocity of the crankpin. In addition, the temperature distribution profile is slightly varying from Fig. 14 (a) and (b). The same phenomenon is shown in for 0.3 coefficient of friction.

Generally, the simulation results show as the load increases the contact temperature will increase dramatically. The increment of angular velocity is highly responsible for the increment of contact temperature than the torque load. This indicates that the sliding velocity has a high influence on the flash temperature.

5. Conclusions

The quarter model of the Geneva mechanism has been modelled and analyzed for maximum contact temperature distribution for three loading cases. The analysis encompasses the analytical and finite element method and compared them for a single loading condition and the results show that the analytical and finite element is in good agreement for determining the maximum contact temperature. From the simulation results the following points are deduced as follows:

- The contact temperature of the Geneva mechanism is maximum due to a substantial sliding velocity between the wheels and pin. If the mechanism is subjected to a maximum loading its contact and flash temperature will increase drastically, and this will lead to a severe change of the mechanical property of the material.
- The maximum flash temperature is found in the crankpin at an angular position of 10–15° and 70–75°. In addition, the maximum contact temperature is maximum at the lower portion of the slot and pin. The wheel has a maximum temperature around the end of the slot of the circle.
- The angular speed of the driving pin is produced at a higher temperature than the torque load. In addition, the Geneva mechanism should be employed for a low load and slow-motion required mechanisms due to temperature rise because of a substantial sliding.

The simulation results are compared with the analytical results, and they are in good agreement, while the experimental validation will be carried on for future study.

CRediT authorship contribution statement

Getachew A. Ambaye: Investigation, Methodology, Writing – original draft. **Hirpa G. Lemu:** Conceptualization, Software, Writing – review & editing. **Mesay A. Tolcha:** Data curation, Supervision, Validation, Visualization.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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