Fatigue life enhancement of ratchet pawl charging mechanism through dynamic analysis

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Abstract

Ratchet pawl closing spring charging mechanism is an important building block of circuit breaker mechanism, seldom studied in past. It is utilized for transferring energy from motor and storing in the closing spring. The key challenges in designing ratchet pawl mechanism are avoiding failures such as burning of driving motor, wearing of pawl tips and ratchet teeth, wearing of pawl stoppers, breaking of eccentric shaft which drives the pawls. In order to combat these failures, this paper mainly considers minimization of driving torque required from motor by controlling friction, eccentricity and pressure angle.

The ratchet pawl mechanism is designed for four important conditions: two load-handing over positions and two extreme conditions of pawls and mathematical relations are established for the same. This paper focuses on three aspects: a) effect of friction between pawl and eccentric shaft on driving torque, b) effect of friction between pawl tip and ratchet wheel, c) effect of eccentricity, backlash, and pressure angle on driving torque. Multibody dynamic analysis showed that the driving torque has exponential relationship with coefficient of friction between pawl and eccentric shaft. A similar analysis showed that the unequal backlash in pawls results in higher impact load by ratchet wheel on pawls than the equal backlash. In the third case study, the undesirable rubbing of pawl tips with ratchet wheel non teeth portion while closing operation is avoided by an innovative design of disengaging the pawls after charging operation. The enhanced designs showed 100-900 % improvement in fatigue life in experimental validation.

Keywords: Circuit breaker mechanism, Ratchet pawl, Closing spring, backlash, eccentricity, coefficient of friction

1 Introduction

Circuit breakers are crucial devices in an electrical substation. They are essentially switches that break (i.e., open) an electrical circuit manually or automatically to protect against faulty conditions. A circuit breaker should also be able to connect the circuit back (i.e., close) after normal conditions are restored. This is achieved in circuit breaker mechanisms with the help of potential energy stored in opening and closing springs. This warrants quick closing and opening of the circuit breaker contacts. The operation of a circuit breaker mechanism is divided into three stages: charging the closing spring, closing the contact, and opening the contact. The focus of this paper is on the study of ratchet pawl charging mechanism.

It is desired to disengage the charging pawls from ratchet wheel while closing operation. Previous researchers [1-4] achieved it by using additional elements i.e. extra spring loaded catch, pawl and cam to hold the charging pawls. Kinematic analysis on duel ratchet pawl mechanism is seldom considered in past. In this paper, we propose a novel methodology of analysing ratchet pawl mechanism to design a circuit breaker spring charging mechanism. Further, the disengagement of pawls while closing operation is achieved by a novel technique. No additional

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element is required in the proposed technique. Design improvements to reduce input torque and improve fatigue life are discussed.

High reduction ratio can be easily achieved in a single stage by a ratchet-pawl mechanism. Here two pawls are provided to drive the ratchet wheel. They are referred to as big pawl and small pawl considering the height difference. One pawl drives the ratchet wheel and other trails i.e. travels in reverse direction along the path of ratchet wheel teeth. Big pawl and small pawl are mounted on 180^o opposite lobes of an eccentric shaft. Motor drives eccentric shaft. Rotation of eccentric shaft is converted into rocking motion of pawls which in turn drives the ratchet wheel. Each pawl advances and trails once during one complete rotation of the eccentric shaft. Connecting rod, spring shoe are assembled with ratchet wheel through crank bolt to form a slider crank mechanism. Ratchet wheel itself acts as crank and spring shoe as slider. Closing spring is assembled between housing (fixed) and spring shoe as depicted in Fig.1.

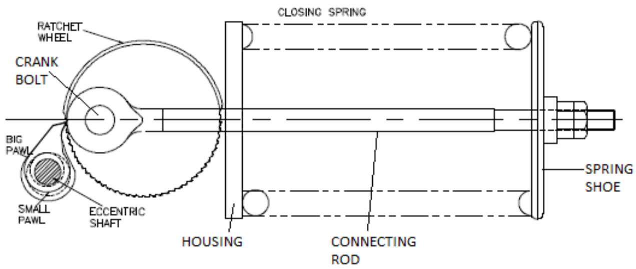

Figure 1: Ratchet pawl spring charging mechanism

2 Extreme positions of pawls

Four conditions are considered for the study of ratchet-pawl mechanism (see Fig. 2): (i) big pawl maximum up and small pawl maximum down, (ii) small pawl maximum up and big pawl maximum down, (iii) load handing over from big pawl to small pawl and (iv) load handing over from small pawl to big pawl. When the big pawl reaches the maximum height, taking complete advantage of eccentricity, the small pawl attains minimum height position. The distance between the centre of the small pawl tip and the corresponding tooth of the ratchet wheel is called the *backlash* (Fig.3).After attaining maximum up position, the big pawl starts its travel in the downward direction and the small pawl in the upward direction. Due to backlash the small pawl cannot push the ratchet wheel. Hence, the ratchet wheel rotates in the reverse direction guided by the big pawl till the small pawl touches the root of the tooth of the ratchet wheel. The small pawl takes over load from the big pawl. The big pawl starts trailing while the small pawl starts advancing. When the small pawl reaches the maximum height, the big pawl attains the minimum height. Backlash is then observed, causing reverse rotation of the ratchet wheel. Backlash is necessary to ensure the entry of pawl in the next tooth of the ratchet wheel. It can be minimized to reduce the reverse rotation of ratchet wheel.

3 Analysis of driving pawl

Kinematic diagram containing driving pawl advancing ratchet wheel is as depicted in Fig. 4. Eccentric shaft OC is pivoted at centre O with eccentricity *e.* Driving pawl CD is assembled with eccentric shaft at C and its length is *sp* or *bp* if small pawl or big pawl is driving ratchet wheel respectively. $OCDO_1$ forms a four bar linkage.

Angular displacement relationship, torque transmission ratio between eccentric shaft and ratchet wheel can be calculated by using formulations of four bar mechanism.

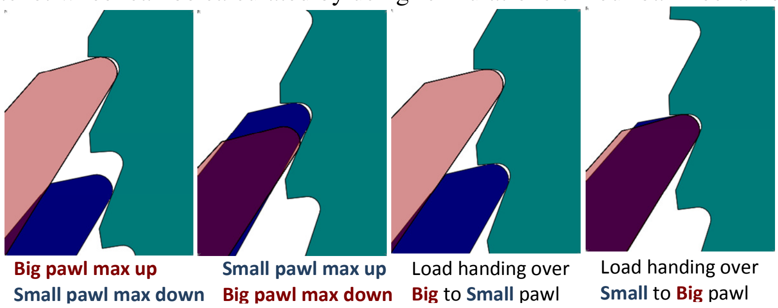

Figure 2: Important conditions of ratchet pawl mechanism

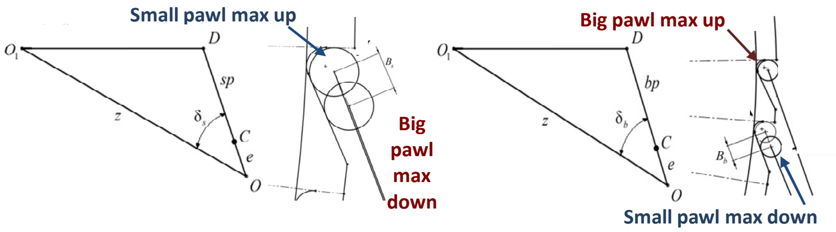

Figure 3: Backlash – Small pawl max up, big pawl max up

Pawl tip radius is *r*. Ratchet wheel centre O_1 is located at distance *z* from eccentric shaft centre O, with x and y as horizontal and vertical components respectively. Angle θ_z is inclination of z with respect to X-axis. Angular position of the eccentric shaft, ratchet wheel, big pawl and small pawl are θ_e, θ_r, θ_{bp}, θ_{sp} . O_1F represents the line joining the ratchet wheel centre and tooth corner point, measuring r_2. Line FI at angle θ_s from O_1H forms the tooth of ratchet wheel. The driving pawl tip i.e. the circle at centre D, forms tangents at line O_1H and FI at G and E respectively, as can be seen in Fig. 4.

Link lengths of $OCDO_1$four bar mechanism, OC, CD, O_1O are expressed as *e*, *sp*, *z*. Link length O_1D is not known. We assume that small pawl is driving the ratchet wheel, for deriving expressions. Further *sp* can be replaced by *bp* to analyze the case of big pawl as driver.

Length and orientation of link O_1D are expressed as

$$O_1D = \sqrt{(r)^2 + (r_2 + FG)^2} \tag{1}$$

$$\theta_{O_1D} = \theta_r - \tan^{-1}\left(\frac{r}{r_2 + FG}\right) \quad \text{Where,} \quad FG = \frac{r}{\tan\left(\frac{\theta_s}{2}\right)} \tag{2}$$

Loop closure equation for O_1DCO four bar mechanism is expressed as

$$\underline{OC} + \underline{CD} + \underline{DO_1} - \underline{OO_1} = 0 \tag{3}$$

By expressing equation (3) in terms of X and Y component equations, rearranging, squaring and adding component equations we get

$$A_1 + A_2 + A_3\cos(\theta_{O_1D}) + A_4\sin(\theta_{O_1D}) = 0 \tag{4}$$

Where,
$$\begin{aligned} A_1 &= e^2 + x^2 + y^2 + O_1D^2 - sp^2 \\ A_2 &= -2ex\cos(\theta_e) - 2ey\sin(\theta_e) \\ A_3 &= 2O_1D(-e\cos(\theta_e) + x) \\ A_4 &= 2O_1D(-e\sin(\theta_e) + y) \end{aligned} \tag{5}$$

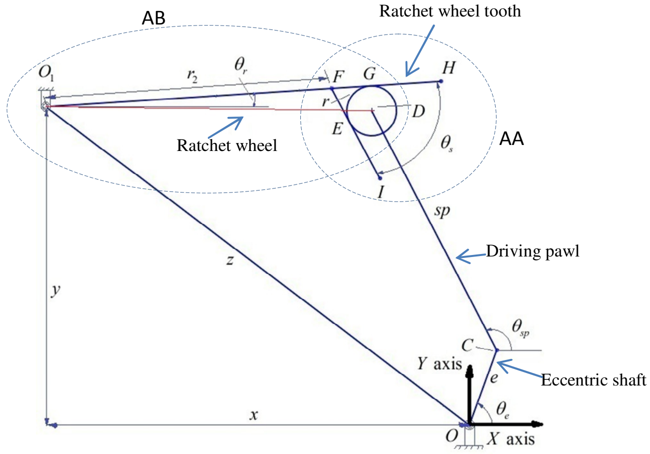

Figure 4: Kinematic diagram of driving pawl and ratchet wheel

Orientation of link O_1D is obtained by solving equation (4) as follows.

$$\theta_{O_1D} = 2\tan^{-1}\left(\frac{-2A_4 - \sqrt{4A_4^2 - 4(A_1 + A_2 - A_3)A_3}}{2(A_1 + A_2 - A_3)}\right) \tag{6}$$

Orientation of the small pawl can be calculated as,

$$\theta_{sp} = \tan^{-1}\left(\frac{-e\sin(\theta_e) - O_1D\sin(\theta_{O_1D}) + y}{-e\cos(\theta_e) - O_1D\cos(\theta_{O_1D}) + x}\right) \tag{7}$$

Orientation of Ratchet wheel θ_r can be obtained by putting value of angle θ_{O_1D} in equation (2)

$$\theta_r = \theta_{O_1D} + \tan^{-1}\left(\frac{r}{r_2 + FG}\right) \tag{8}$$

Torque transmission ratio between the eccentric shaft and the ratchet wheel when the small pawl is driving the ratchet wheel is given by

$$\frac{T_R}{T_e} = \frac{r}{e}\frac{\sin(\theta_{sp} - \theta_{rs})}{\sin(\theta_{sp} - \theta_e)} \tag{9}$$

Where, T_R and T_e are torque transmitted on ratchet wheel and torque input from eccentric shaft respectively. θ_{rs} is the value of θ_r when the small pawl is driving the ratchet wheel. Similar equation can be written for the big pawl and ratchet wheel by replacing θ_{sp} by θ_{bp} and θ_{rs} by θ_{rb}.

Torque on the ratchet wheel shaft at O_1 due to compression of closing spring, can be computed by

$$T_R = \frac{k(l_0 - (x_i - l_f))\sin(\theta_A + \delta)}{\cos(\delta)} r \tag{10}$$

Combining equations (9) and (10), input torque required at eccentric shaft, T_e can be calculated.

4 Analysis of trailing pawl

When one pawl drives the ratchet wheel, the other moves in the opposite direction following the ratchet wheel teeth surface. Position of trailing pawl is necessary to compute the backlash. Kinematic diagram of trailing pawl is as depicted in Fig. 5(a). Here, small pawl is driving the ratchet wheel and big pawl is trailing in the same driven tooth of ratchet wheel. Let d_1 be the perpendicular distance of slanted portion of the ratchet wheel teeth from O_1. Let PO be the perpendicular distance of slanted portion of the ratchet wheel teeth from O. And let $\hat{\lambda}$ be the unit vector along slanted portion of ratchet wheel teeth. Let $\hat{n}$ be the unit vector perpendicular to slanted portion of ratchet wheel teeth passing through $\mathrm{O_1}$. Let γ be angle between $\mathrm{O_1O}$ and OD.

$$\underline{OJ} + \underline{JK} + \underline{KL} + \underline{LF} + \underline{FO_1} - \underline{\mathrm{OO_1}} = 0 \tag{11}$$

By expressing equation (11) in terms of X and Y component equations and rearranging them and dividing Y component equation by X component equation we get orientation of big pawl θ_{bpt} during trailing as,

$$\theta_{bpt} = \tan^{-1}\left(\frac{(e\sin(\theta_e+\pi) + r\sin(\pi-\theta_s) - r_2\sin(\theta_r) - y)}{(e\cos(\theta_e+\pi) + r\cos(\pi-\theta_s) - r_2\cos(\theta_r) - x)}\right) \tag{12}$$

When big pawl is driving the trailing position of small pawl can be calculated by replacing value of θ_r by $(\theta_r - p)$ where, θ_r is position of ratchet wheel when big pawl is driving, p is pitch of ratchet wheel.

Applying cosine rule to triangle $\mathrm{O_1FO}$ (Fig. 5(b)), length FO can be given as

$$FO = \sqrt{z^2 + r_2^2 + 2zr_2\cos(\theta_r+\theta_z)} \tag{13}$$

Applying sine rule to triangle $\mathrm{O_1FO}$ (Fig. 5(b)), angle between $\mathrm{O_1F}$ and FO can be calculated as,

$$\theta_{O_1FO} = a\sin\left(\frac{\mathrm{z}}{\mathrm{FO}}\sin(\theta_r+\theta_z)\right) \tag{14}$$

Angle between OF and FP (Fig. 5(b)) can be given as,

$$\theta_{OFP} = (\pi - \theta_s) - \theta_{O_1FO} \tag{15}$$

Hence, backlash, length EL in Fig. 5(a), with big pawl when small pawl is driving can be computed as,

$$B_s = FP - FE - bp\cos(\theta_{bp} - (\pi-\theta_s)) \tag{16}$$

Where, $FP = \cos(\theta_{OFP})\left(\sqrt{z^2 + r_2^2 + 2zr_2\cos(\theta_r+\theta_z)}\right)$ and $FE = \dfrac{\mathrm{r}}{\tan(\frac{\theta_s}{2})}$

Similar expression can be derived for backlash when big pawl is driving by replacing θ_{bp} by θ_{sp}, bp by sp and calculating value of θ_r for big pawl as driving pawl.

The rotation of ratchet wheel with respect to rotation of eccentric shaft is as depicted in Fig. 6. The peaks in graph indicate the maximum up conditions of big/small pawl. Here, pawls take maximum advantage of the eccentricity and advance the ratchet wheel to the fullest extent. The valleys in the graph indicate reverse rotation of the ratchet wheel after reaching its peak. The reverse rotation is a function of the difference between pawl heights, i.e. ($bp - sp$), eccentricity e of the eccentric shaft and ratchet wheel pitch.

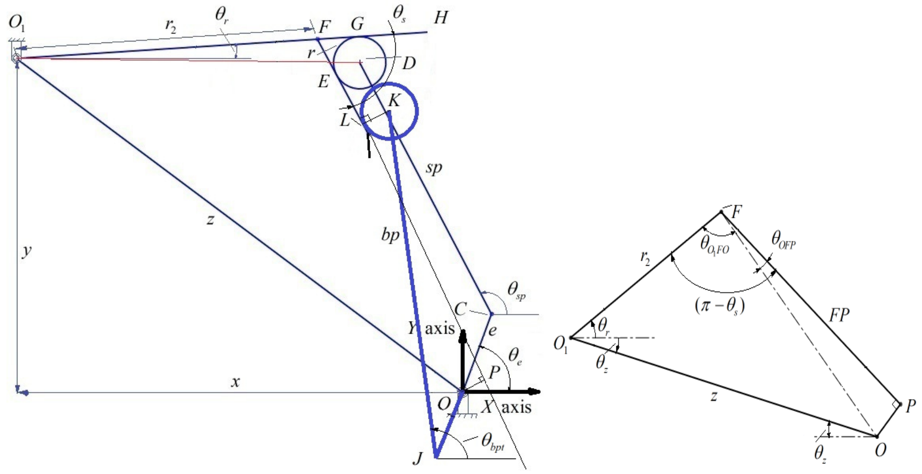

Figure 5: a) Kinematic diagram of trailing pawl, b) $OPFO_1$ from a).

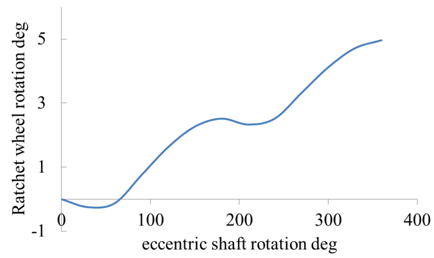

Figure 6: Ratchet wheel rotation with respect to eccentric shaft rotation

Torque required at ratchet wheel center, ratchet wheel pitch, eccentricity, friction between moving components of ratchet pawl mechanism, transmission angle between pawl and ratchet wheel are the factors affecting input torque required from motor connected at eccentric shaft. Three case studies are presented as follows.

4.1 Case study 1: Effect of eccentricity, transmission angle and backlash

Model 1 was taken as base for analytical study and experimentation. The input parameters for ratchet pawl mechanism model 1 are as mentioned in table 1. Eccentric shaft was broken after 6500 operations during experimentation. The minimum value of e is 10.2/4 = 2.55 mm. The additional 0.95 mm eccentricity resulted in unequal backlash 0.92 mm and 2.43 mm for big pawl maximum up and small pawl maximum down conditions respectively. The transmission angles for these conditions are 116.18^0 and 119.95^0. The unequal backlash led to increased impact force on big and small pawls. Alternatives to reduce impact force on pawls are 1) reduce transmission angle, 2) minimize and equalize backlash. e needs to be reduced to minimize the backlash and pawl heights need to be adjusted to make the backlashes equal. Model 2 is designed to take care of these requirements. The vertical distance between ratchet wheel and eccentric shaft is increased to 78 mm from 60 mm. *e* is reduced to 3 mm from 3.5 mm. Pawl heights are changed to 79.6 mm and 75 mm for big and small pawls respectively. The dimensional changes resulted in reduction in transmission angles to 109.7^0 and 112.76^0. Also, the backlashes reduced to 1.34 mm for both. The maximum input torque required is reduced to 22.1 Nm from 25.3 Nm. Multibody dynamic analysis study is conducted on both models to study the reaction forces on pawls and input torque. The maximum reaction force on big pawl is reduced to 15300 N from 18430 N. Also, the maximum reaction force on small pawl is reduced to 18250 N from 21245 N. Model 2 exhibited successful 15000 operations without any failure and experimentation was stopped as requirement of 10000 cycles was satisfied.

Table 1 Ratchet pawl charging mechanism parameters for case study 1

	Parameter	*Unit*	*Model 1*	*Model 2*
Inputs	Closing spring stiffness *k*	N/mm	141	141
	eccentricity *e*	mm	3.5	3
	ratchet wheel circular pitch	mm	10.2	10.2
	big pawl height *bp*	mm	60.8	79.6
	small pawl height *sp*	mm	57.2	75
	c-c distance *x*	mm	143	143
	c-c distance *y*	mm	60	78
Outputs	Transmission angle between ratchet tooth & big pawl-when big pawl is at max up	deg	116.18	109.07
	Transmission angle between ratchet tooth & small pawl-when small pawl is at max up	deg	119.95	112.76
	backlash - big pawl max up	mm	0.92	1.34
	backlash - small pawl max up	mm	2.43	1.34
	Maximum input torque required at eccentric shaft	Nm	25.3	22.1
	Experimentation results		Eccentric shaft broken after 6500 operations.	Performed 15000 operations.

4.2 Case study 2 - Effect of friction

Inner diameter pawl forms revolute joint with eccentric shaft lobe. The friction in this revolute pair results in requirement of additional input power. Effect of variation of coefficient of friction on peak input torque is as depicted in Fig.7. Input torque slightly increases from 16.5 Nm to 24 Nm for increase in coefficient of friction from 0 to 0.1. Further, input torque increases steeply till 159 Nm for coefficient of friction as 1. It is desired to keep coefficient of friction as minimum as possible so that the input torque required will be the least. The coefficient of friction will increase due to rubbing between inner surface of pawl and outer surface of eccentric shaft lobe, after several cycles of operation. This will increase the input torque required from motor. Therefore, the coefficient of friction should remain almost constant for the desired number of cycles (i.e. 10000 operations). Multibody dynamic analysis showed that the driving torque has parabolic relationship with coefficient of friction between pawl and eccentric shaft. Validation of the same is done by two variants i.e. use of dry metallic bush and needle roller bearing. The dry metallic bush wears after 1000 operations and the driving torque increases and the motor gets burned out. The needle roller bearing lasts 10000 operations without burning of motor.

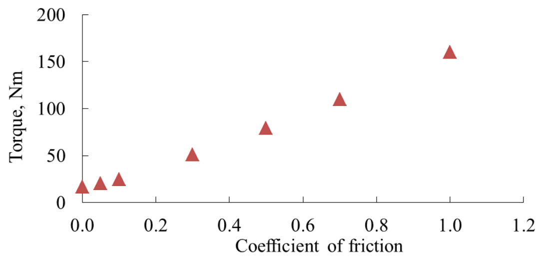

Figure 7: Effect of friction on input torque at eccentric shaft

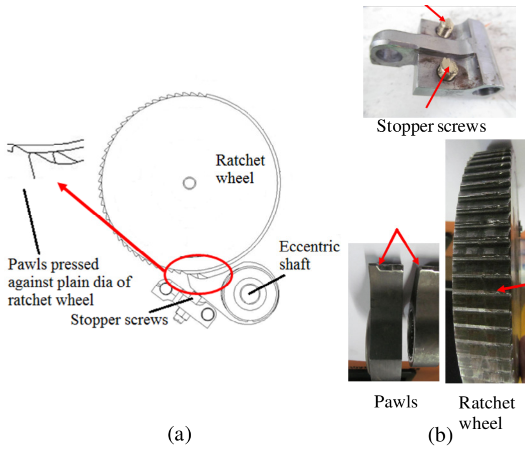

Figure 8: Before improvement - (a) Ratchet pawl during closing operation, (b) Failures after 1300 operations

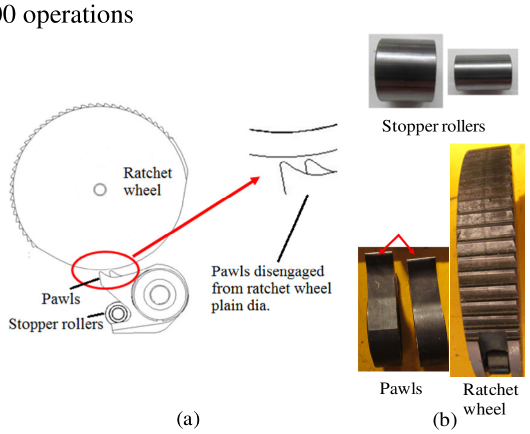

Figure 9: After improvement - (a) Ratchet pawl during closing operation, (b) Condition of components after 12000 operations

4.3 Case study 3: Effect of ratchet wheel and pawl tip wear

Pawls fly off during rotation of the ratchet wheel and produce sudden shock on the stopper screws during closing operation as depicted in Fig. 8(a). Impact of pawls on the stopper screws results in heavy wear of screw heads. Torsion springs bring pawls back to maintain contact between ratchet wheel and pawl tip. Plain diameter of ratchet wheel is in contact with the pawl tip during closing operation. It creates impression on pawl tip. The worn out stopper screws, pawl tips and ratchet teeth after 1300 operations are as shown in Fig. 8(b). This further damages the pawl tip area which is beneficial for charging. Damaged pawl tip area results in a force transfer through high points on the pawls which further results in deterioration of the ratchet wheel teeth. Wear of ratchet wheel teeth and pawl tips results in no charging.

Stopper rollers are provided instead of the stopper screws. They are pivoted on the bar-pin and hence are free to rotate. Therefore, for every closing operation, pawl will get a different surface of the roller for impact. This will cause uniform wear of roller. Prevention of pawl tips from rubbing with ratchet wheel plain surface / non teeth zone protects the pawl tips from wearing during closing operation. The non-teeth zone diameter of ratchet wheel is reduced by 2 mm. Projection on pawl is provided so that it will touch the stopper rollers and avoid contact between pawl tips and ratchet wheel plain diameter. The projections on pawls are designed such that the locus of highest point on tip should not touch the plain diameter (Fig. 9(a)). The enhancements are validated through experimentation. The condition of pawl tips, rollers, and ratchet wheel teeth after 12000 operations is as shown in Fig. 9(b). The fatigue life of ratchet wheel, pawls and stopper rollers is increased by 900%.

5. Conclusions

Working of duel ratchet pawl is studied for four extreme conditions: big pawl max up, small pawl max up and two load-handing over positions. Mathematical relations are established for these conditions. Multibody dynamic analysis showed that the driving torque has parabolic relationship with coefficient of friction between pawl and eccentric shaft. Reduction in the friction at pawl pivot joint resulted in 10 times fatigue life enhancement. Multibody analysis showed that the unequal backlash in pawls results in higher impact load by the ratchet wheel on pawls than the equal backlash. Experimental validation showed that the reduction in eccentricity from 3.5 mm to 3 mm, the improvement in pressure angle from 124^o to 116^o and the equal backlashes reduced from 3.2 mm to 1.3 mm for both pawls resulted in 100% improvement in fatigue life. The undesirable rubbing of pawl tips with ratchet wheel non-teeth portion while closing operation is avoided by an innovative design of disengagement of the pawls after the charging operation. This resulted in nine fold enhancement in the fatigue life of ratchet pawl mechanism in experimentation.

6. References

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