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Design and Development of Experimental Test Rig for Fatigue Life Estimation of Freewheel One-Way Clutch Used in Switchgear Applications at High Overrunning Speeds

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Abstract

Freewheel one-way clutch (OWC) is a critical component used in industrial circuit breakers/switchgear to facilitate quick disengagement of the drive when an electrical circuit trips under overcurrent. The clutch acts as a safety device to endure high overrunning speeds and allows the drive shaft to rotate at normal operating speeds during each tripping cycle. The trip mechanism utilizes the stored potential energy of the helical compression spring, which is the critical design component of the circuit breaker to induce the required amount of torque at the clutch. OWC experiences fatigue during its lifespan while operating under repeated trip cycles. The fatigue testing of the clutch is carried out by developing a test bench specifically designed to simulate the tripping function of the switchgear application. The test bench is designed using the concept of a crank-gear operating mechanism, assembled with gear trains and compression springs to achieve the required transmissible torque under high overrunning speed. The design parameters like the size of gears, spring, and crank-connecting link are evaluated for the required torque capacity and the relevant size of the circuit breaker. Also, the designed torque capacity of the test bench at the clutch end is measured using a rotary torque sensor and theoretically validated. The torque sensor values differed from the theoretical values by 3-5% within the standard acceptable limit of 10%.

Keywords: One-way clutch (OWC), fatigue life, experimental test rig, overrunning speed, switchgear

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INTRODUCTION

One-way clutch (OWC) is the most commonly used unidirectional torque transmitting device in power transmission and automotive industries. It utilizes friction elements known as sprags which act as wedging elements to transmit the torque in one particular direction while running free in the reverse direction. The elemental components of the clutch include the inner race which is a driving unit, the outer race usually a driven unit, sprags, energizing springs, retainers, and bearings. A cut-section of OWC [1]. is illustrated in Figure 1.

When the inner race rotates faster than the outer race, the frictional force along with spring force tends to rotate sprags in a clockwise direction about their centers enabling them to provide a wedge between the two races for transmitting the required amount of torque by locking two races together. On the other end, when the outer race rotates faster than the inner race, the frictional force between the races and sprags enables sprags to rotate in an anti-clockwise direction about their centers, causing them to provide relative velocity between two races that are known as overrunning or freewheel function. During the overrunning period, no torque can be transmitted from either race. Chesney et al. [2, 3] described the practical applications, operational theories, system equations, element defections, stress equations, and static-dynamic behavior of one-way power-transmitting elements. The overrunning clutch design, slip angle, governing equations, and dynamic characteristics were described by Burgess et al. and Liu et al. [4, 5]. The endurance tests of OWC were carried out for the first time by K. Durkopp et al. [6]. They described the test procedure for evaluating fatigue life using an individual clamping zone. Neelakandan et al. [7] proposed an optimized design solution for OWC utilized for starter motors to improve fatigue life. OWC is a critical component employed as a torque-transmitting device in switch gear applications for the tripping mechanism. The mechanism utilizes the overrunning function of the clutch to dissipate the high-speed reverse rotation of the transition elements each time the circuit trips. The first design patent for a spring-operated circuit-breaking mechanism was proposed by Ikuo et al. [8]. Fu-Chen et al. [9, 10] described the theory of operation, kinematics, and dynamics of spring-operated mechanism for a 69 kV SF6 gas-insulated circuit breaker. Modern switchgear applications utilize gear trains made from composite materials in order to use the advantage of weight reductions and improve transmission strength. Some common raw materials used in switch gear applications are also manufactured from thermoset rigid laminates [11]. Also, the insulation layers of vacuum interrupters in solid insulated switch gear applications are made from polymer composites. [12] OWC utilized in switchgear applications is subjected to high overrunning speeds and used as a precautionary device in a circuit-breaking system to absorb high overrunning speeds while tripping off the system due to overload thereby avoiding any further damage to the drive unit. The clutch under repeated trip cycles is subjected to cyclic fatigue and it is necessary to evaluate fatigue life in order to guarantee the reliability of the clutch in this application. Therefore, a novel attempt is made to build an in-house test facility to execute the tripping function of the switchgear application utilizing the concept of a conventional slider-crank mechanism. In this way, the fatigue experiments are performed on a specifically developed test bench utilizing the freewheel OWC for direction-dependent torque transmission function under high overrunning speed. A dedicated test bench is thus designed, fabricated, and assembled to simulate the accurate functioning of the circuit breaker. The design of test bench components is carried out using a standard solid mechanic approach and the torque induced at the clutch is validated using torque measuring devices.

EXPERIMENTAL TEST BENCH FOR FATIGUE ANALYSIS OF OWC FOR SWITCHGEAR APPLICATION

OWC is used in the spring-operated circuit breaker mechanism to trip the electrical circuit under a high overload current. The application of OWC in the switchgear tripping mechanism utilizes the overrunning function of the clutch to discharge the high-speed rotation of the driven unit when the circuit trips without affecting the drive unit.

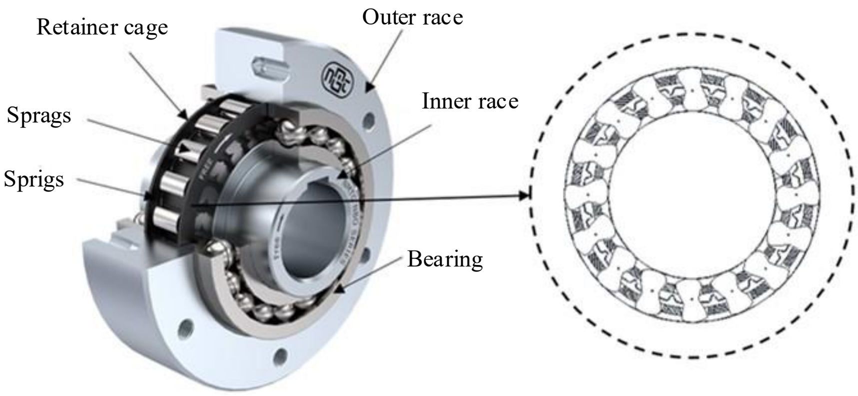

Figure 1. Sprag type one-way clutch section [1].

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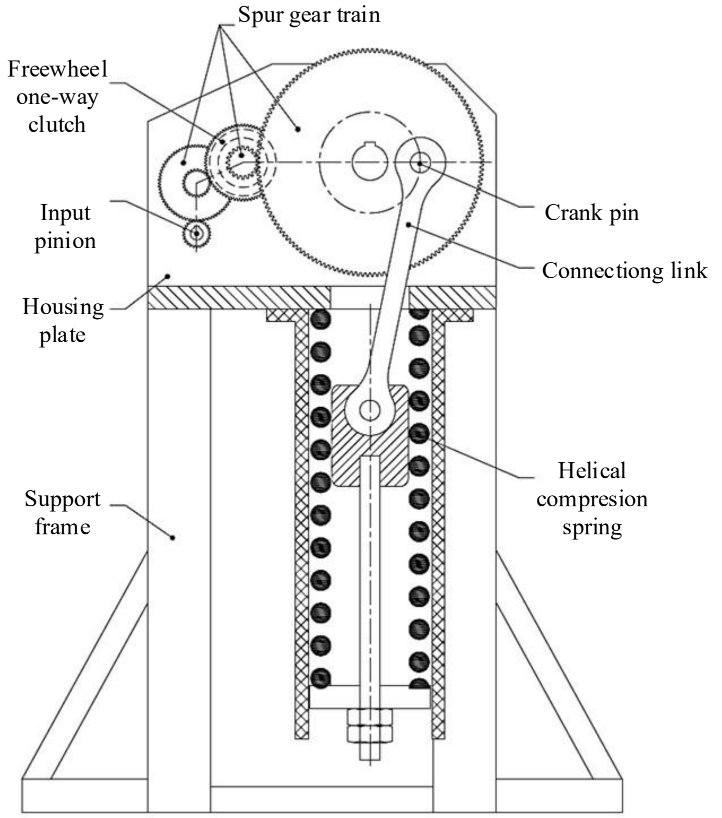

Figure 2. A schematic diagram of switchgear trip mechanism utilizing OWC.

Tripping Mechanism

In order to develop any conceptual mechanism for performing a predefined function, the following parameters are identified: (i) the magnitude of load/motion application, (ii) the method of load/motion application, and (iii) the measurement of load/motion. The schematic diagram of the operating mechanism is illustrated in Figure 2.

The mechanism unit consists of a gear motor, compound gear trains, a crank-connecting link mechanism, and a helical compression spring. The switchgear application involves two operations: (i) charging the spring by the charge motor and (ii) releasing the stored energy of the spring while tripping due to current overload. The spring is charged/compressed using a crank-link mechanism from the power delivered by the charge motor. The charging motor stops when the spring reaches its pre-set limit and the energy will be stored in the spring in the form of strain energy till the next trip cycle. The input power for charging operation against the spring load at the driven end is supplied through several gear trains using OWC installed at an intermediate stage. The helical compression spring is responsible for the magnitude of torque induced at the clutch. OWC experiences fatigue when the mechanism allows the clutch to transmit the torque during charging and allows it to overrun while discharging. The fatigue life of the clutch is experimentally evaluated by recording the number of charge-discharge cycles.

Test Bench Specifications and Material of Construction

The experimental test bench comprises several components that exhibit the predefined function of the tripping mechanism. A test bench is manufactured for introducing specified torque on OWC test specimens using a drive motor as a power input to charge the mechanism through the gear train. The load at the clutch unit is induced by compressing the helical spring which is attached to the driven end of the gear train using a crank-connecting link system. A driving member comprises a 0.75 kW, 3-phase gear motor with an output speed of 120 rpm which is capable of delivering 60 Nm of torque. The gear train consists of a motor pinion, intermediate gear, intermediate pinion, clutch gear, crank pinion, and crank gear. The intermediate gear-intermediate pinion and clutch gear-crank pinion act as compound gear trains. The helical compression spring is designed and coupled with the crank gear with the help of a crank pin and connecting link to produce the required spring load which can generate the specified torque. The gear motor output shaft is coupled with a motor pinion which then transmits the required amount of torque to crank gear through gear trains. The outer race of the OWC assembly is made integral to the clutch gear and the inner race is coupled with the crank pinion shaft through torque torque-measuring load cell. The output torque at the crank gear is defined in correspondence to the required induced torque at the clutch. The torque from the clutch gear is transmitted to the crank pinion shaft through OWC in one particular direction with a particular speed during charging. When the circuit breaker trips, the excessive rotation of the crank pinion causes the crank pinion shaft to rotate at a higher speed than the charging speed which causes the freewheeling of the clutch to restrict such motion further transmitted to avoid any damage to input elements.

The test bench components are selected, fabricated, and assembled as per the specified tripping cycle time and corresponding torque requirements. The trip mechanism is designed for 14 seconds of charge duration and 1 second of trip duration. In order to execute 4 fatigue cycles per minute, the crank gear must be rotated at the speed of 2 rpm. The clutch is subjected to 500 Nm torque capacity. The selection of test bench components and their material properties are thus carried out based on the induced torque capacity of OWC and the required fatigue cycle duration. The material properties of the respective components are listed in Table 1.

DESIGN CALCULATIONS AND SELECTION OF TEST BENCH COMPONENTS

Selection of Gear Pairs

Three gear pairs are selected and checked for required torque transmission capacity against the permissible beam strength and surface strength using Eq. (1) and Eq. (2) respectively. The gear pairs are then selected as per the required gear module using Eq. (3). The available beam strength and surface strength [13] are calculated using Eq. (4-5).

$$[\sigma_b] = \frac{K_{bl}}{n\, k_\sigma}\sigma_o \tag{1}$$

$$[\sigma_c] = C_R x HRC x k_{cl} \tag{2}$$

$$m \geq 1.26\sqrt[3]{\frac{y[M_t]}{[\sigma_b]\Psi_m z_1}} \tag{3}$$

$$[\sigma_b] = \frac{(\mathrm{i}+1)\,[M_t]}{a\, m\, b\, y} \tag{4}$$

$$[\sigma_c] = 0.74\frac{i+1}{a}\sqrt{\frac{(i+1)E[M_t]}{ib}} \tag{5}$$

Table 1. Material Properties of Test Bench Components.

S N.	Component	Material	Ultimate tensile strength, MPa	Modulus of elasticity, GPa	Poisson's ratio
1	Gear pairs	Case hardened steel (EN353), case hardness of 60±2 HRC for 1.5 mm case depth	1410	210	0.30
2	Helical compression spring	Spring steel (SAE 9254)	1579	210	0.30
4	Connecting link	Mild Steel	475	200	0.30
5	Crank pin	Alloy steel (EN19)	655	210	0.30
6	Shaft	Alloy steel (EN19)	655	210	0.30

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Selection of Helical Compression Spring

A helical compression spring is subjected to cyclic charge-discharge operations with two different speeds in one trip cycle. The charging indicates the compression of the spring when the crank moves from the Bottom Dead Center (BDC) (0°) to the Top Dead Center (TDC) (180°) at the speed of 2 rpm. During discharge operation, the spring releases its potential energy and quickly returns to its initial state at approximately 120 rpm of crank. This repeated charging and discharging continues to produce the stresses in the spring which should be within the permissible limits. The shear stress induced considering the curvature effect of the spring wire is calculated using Eq. (6) [14] and checked against the permissible value [τ_s] (710.55 MPa),

$$\tau = k_w \frac{8F_s D}{\pi d^3} \tag{6}$$

The torque induced at the crankshaft due to this spring force will depend upon the tangential effort provided by the crank rotating from 0° to 180° having a crank radius of 100 mm. Since the crank moves from BDC to TDC, the maximum deflection of the spring will be 200 mm. To determine the maximum torque induced on the crankshaft, the maximum tangential effort of the crank should be derived using Eq. (7) for a particular crank position with respect to the dead centers.

$$F_t = \frac{F_P}{\cos \emptyset} \sin(\theta + \varphi) \tag{7}$$

Design of Shaft

OWC is supported on the crank pinion shaft along with gears and bearing supports. As shown in Fig. 3, the outer race of the clutch is made integral to the hollow clutch gear, and the inner race is coupled to the crank pinion shaft using a key for transmitting the torque from the clutch gear to the crank pinion shaft. There are a number of support bearings provided to the shafts for the transmission of radial load as well as for supporting rotary motion during power transmission or overrunning. In this way, the shaft is subjected to vertical load due to the weight of the components in addition to the twisting moment due to torque transmission. Therefore, it is required to check shafts for the equivalent twisting load (T_e) due to the maximum bending moment and twisting moment as per the maximum shear stress theory [14] using Eq. (8). Refer to the line diagram shown for the available loading conditions.

$$d_{min.} = \sqrt[3]{\frac{\mathrm{Te} \times 16}{\pi \, x \, \tau_{allow}}} \tag{8}$$

Design for Crank and Connecting Link

Design of crank pin

The crank pin is designed based on maximum permissible bending stress at the pin end acting as a cantilever beam fixed at the end on the crank gear at the pitch circle of the crank surface. As the bearings are used in between the crank pin and connecting link eye, there will be no rubbing surfaces, and the permissible bearing pressure criterion is not considered. The maximum bending load considered is the same as the maximum spring force available. The minimum pin diameter required to sustain the required load considering permissible material limits is calculated using Eq. (9).

$$d_{p_{min}} = \sqrt[3]{\frac{64\, Fs\, e}{2\pi\sigma_b}} \tag{9}$$

Design of Connecting Link

The connecting link is subjected to cyclic tension and compression which causes the bending of the connecting link about the axis passing perpendicular to the plane of cyclic motion. The maximum bending moment for the connecting link is calculated using Eq. (10). The size of the connecting link is selected based on the maximum bending stress theory [14] using Eq. (11) considering the inertia forces for the maximum mass and crank revolutions. The connecting link section under maximum spring load is also checked against the permissible tensile stress limit.

$$M_{max.} = mr\omega^2 \frac{l}{9\sqrt{3}} \tag{10}$$

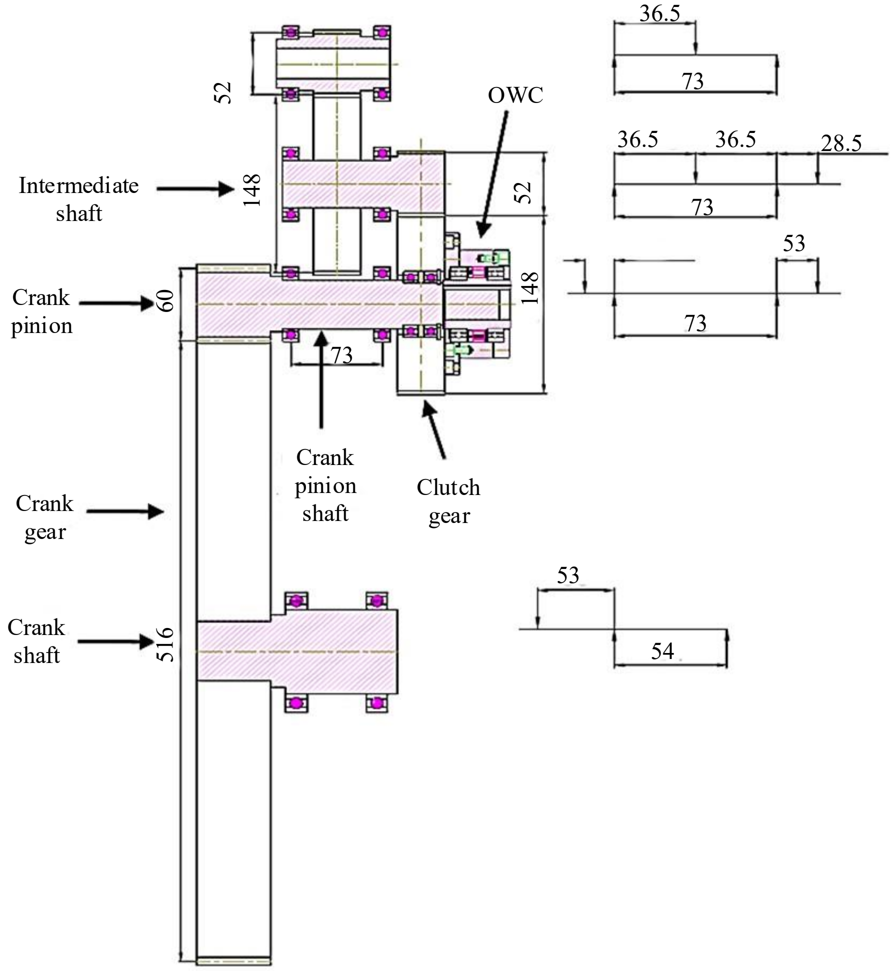

Figure 3. A line diagram of the shaft for available loading conditions.

$$\sigma_{bmax} = \frac{M_{max}}{Z_{xx}} \tag{11}$$

Where m is the mass of the connecting link, r is the crank radius (100 mm), l is the length of the connecting link, N is the maximum crankshaft speed (120 rpm at discharge), ω is the maximum angular velocity, σ_{bmax} is the maximum bending stress and Z_{xx} is the section modulus.

The connecting link eye is inserted in the crank pin which is subjected to direct tensile load under maximum spring force. The bearing area of the connecting link eye section is considered by the available permissible tensile stress of the material under maximum spring force.

Design of Spring Rod Pin

The connecting link eye is inserted in the spring rod using a spring rod pin subjected to double shear under spring load. The maximum shear stress induced in the spring rod pin is given by Eq. (12).

$$d_{min} = \sqrt{\frac{4\text{Fs}}{2\pi\tau_{allow}}} \tag{12}$$

RESULTS AND DISCUSSIONS

An experimental test bench for performing endurance tests of OWC is established using several components that can simulate the behavior of the tripping mechanism of the switch gear application.

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The corresponding test bench components are designed, fabricated, selected, synthesized, and assembled for executing specified torque requirements at the clutch end and required trip cycle frequencies as shown in Figure 4.

The design parameters for the test bench components are checked for the available size, material, and space constraints using the basic solid mechanics theories as discussed in section 3. The gear pairs are selected as per the specifications described in Table 2. The permissible beam strength and surface strength for the gear material from Eq. (1-2) are evaluated as 473.3 MPa and 1434 MPa respectively. The available beam strength, surface strength, and gear modules are checked against the permissible limits using Eq. (3-5) and listed in Table 3.

Table 2. Gear selection parameters.

Gear pair	Element	M_t	i	a	Speed (N, rpm)	Z	D	b	ϕ	m
1	Motor Pinion	59.68	2.84	100	120	26	52	35	20	2
	Intermediate Gear				42.16	74	148	35	20	2
2	Intermediate Pinion	169.87	2.84	100	42.16	26	52	35	20	2
	Clutch Gear				14.81	74	148	35	20	2
3	Crank Pinion	483.48	8.6	288	14.81	20	60	55	20	3
	Crank Gear				2	148	444	55	20	3

Table 3. Calculated beam strength and surface strength.

Gear pair	m	$\Psi_m = \frac{b}{m}$(Assumption)	Y	σ_b	σ_c
1	$0.89 \simeq 2$	15	0.4	123	766.8
2	$1.21 \simeq 2$	15	0.355	394.4	1293.7
3	$1.87 \simeq 3$	15	0.355	412.7	1382.9

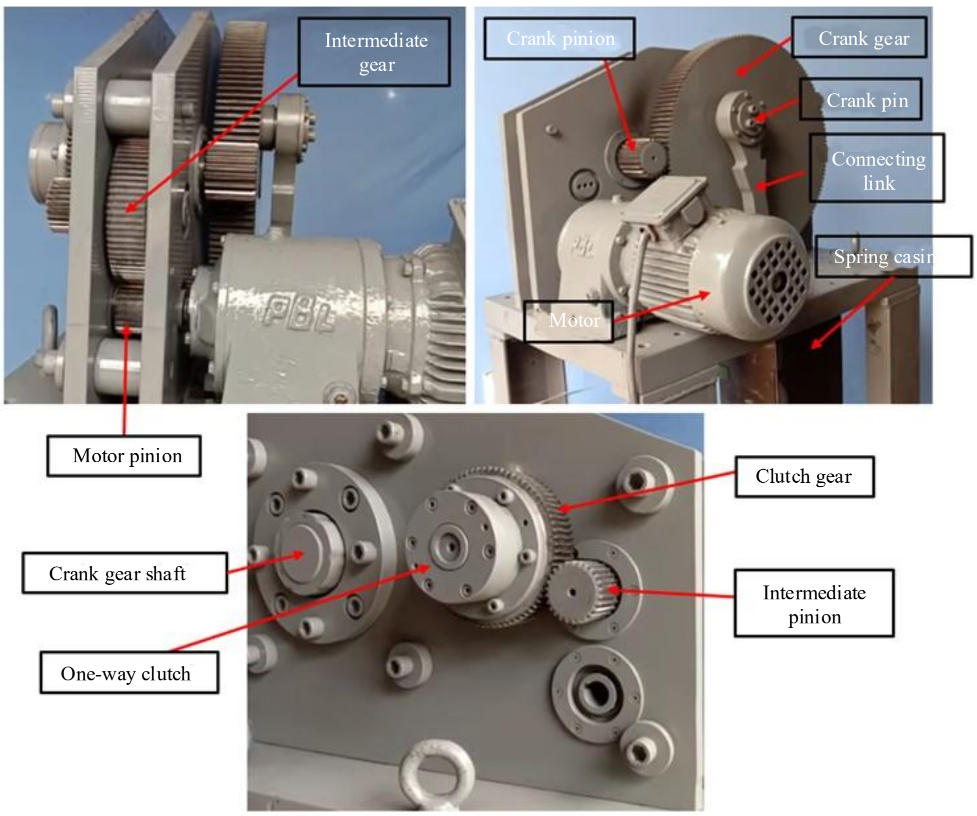

Figure 4. Experimental test rig developed for evaluating fatigue lives of owc utilized for switchgear application.

The selection of a helical compression spring to generate the required amount of torque at the OWC test specimen is carried out for the safe wire and coil diameters using Eq. (6) and the respective spring parameters are derived as shown in Table 4. The tangential effort and corresponding torque provided by the crank with respect to the crank angle measured along the line of action derived using Eq. (7) is listed in Table 5. The shafts are designed using Eq. (8) by maximum shear stress theory considering both bending moment due to the weight of the components and the available twisting moment. The derived sizes of shafts are listed in Table 6. The crank pin and spring rod pin values are also evaluated and listed in the same table for the maximum available spring force using Eq. (9) and Eq. (12) respectively. A connecting link of rectangular cross-section (b= 40 mm width, h= 45 mm depth) having 10 kg of mass, and 500 mm of length is selected and checked against the permissible bending stress using Eq. (10-11) for transmitting the rotational motion of the crank into linear motion of the spring rod.

The tripping cycles are recorded using a digital counter meter for specified charge-discharge time intervals. The induced torque at the OWC test specimen is measured using a digital torque sensor installed at the clutch end which varies with respect to crank angle during charge operation. The variation of spring force with respect to crank angle while compressing to the maximum deflection is represented by Figure 5. The corresponding variation of theoretical torque at the crankshaft with respect to crank angle while moving from BDC to TDC for available crank radius is displayed as a turning moment diagram in Figure 6 using Eq. (5). The maximum torque induced on the crank is observed when the crank is at 105°-115° position with respect to BDC. A comparison between the theoretical torque and the measured torque using the torque sensor at the clutch end with respect to crank angle is graphically illustrated in Figure 7. The reported results of the torque sensor follow quite closer values to the theoretical values having 3 to 5% of variation within the standard acceptable limit of 10%.

Table 4. Helical Compression Spring Parameters.

Spring parameters	**d**	**D_c**	**C**	**k_w**	**n_t'**	**n_t**	**δ_{max}**	**G**	**F_S**	**τ_s**
Values	38	200	5.26	1.293	12	10	200	79300	51672	620

Table 5. Turning Moment Parameters for Crankshaft.

θ	**ϕ**	**F_S,**	**F_t**	**Crankshaft M_t**
0	0	0	0	0
30	5.74	4108	2411.5	241
60	9.97	14868	14183.3	1418
90	11.54	28443	28443.2	2844
110	10.83	36971.5	32324	3232
120	9.97	40704.8	31675	3167
150	5.74	48856.1	20178	2017
180	0	51672	4.787	0.47

Table 6. Shaft and Pin Size Values.

S N.	**Parameter**	**Value, mm**
1	Motor shaft	12.21
2	Intermediate shaft	21.10
3	Clutch shaft	31.23
4	Crankshaft	54.02
5	Crank pin	40.9
6	Spring rod pin	11.47

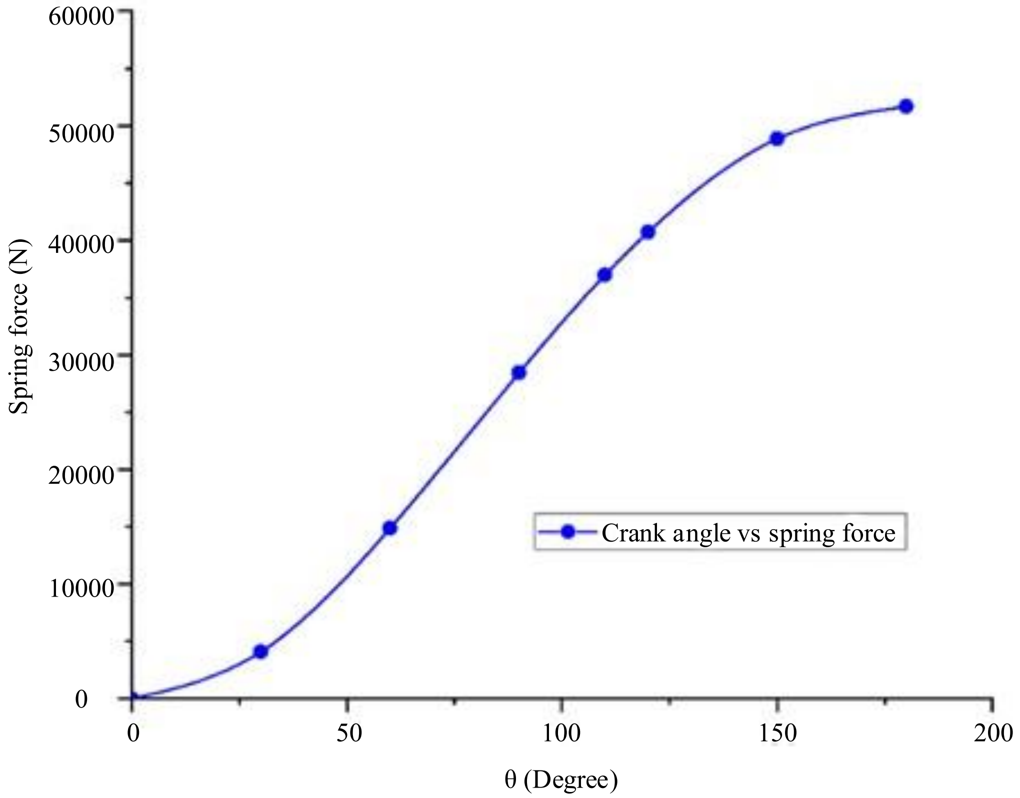

Figure 5. Spring force variation with crank angle.

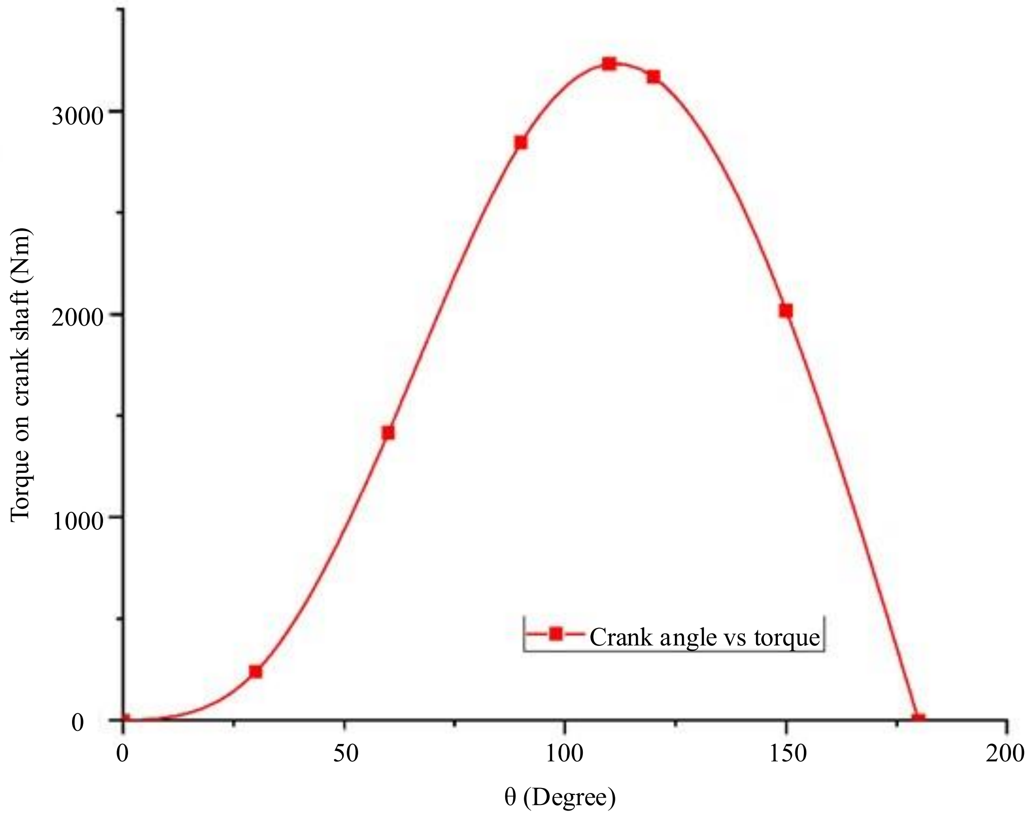

Figure 6. Turning moment diagram of crankshaft.

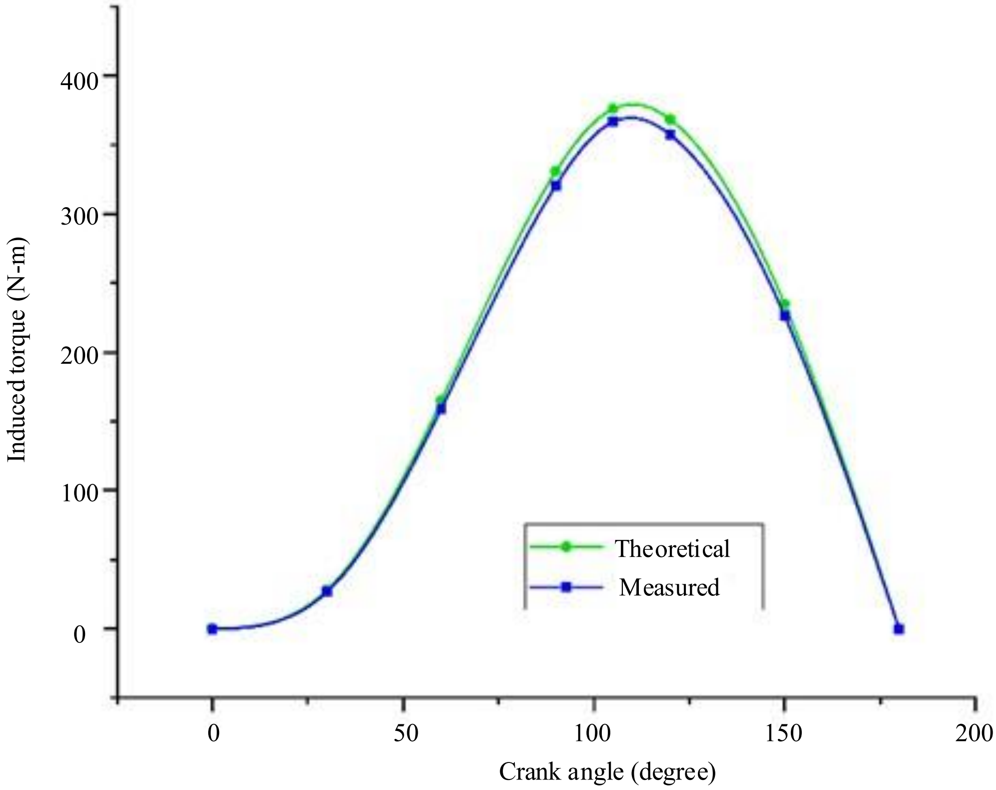

Figure 7. A comparison of theoretically induced torque with measured values at the clutch.

CONCLUSIONS

The fatigue life of freewheel one-way clutch (OWC) utilized in switchgear applications under high overrunning speed is evaluated using a specifically designed experimental test bench. Such a test bench is manufactured and assembled which implicates the behavior of the tripping mechanism of the circuit breaker. The components of the test bench like the gear train, shafts, and crank-connecting link are designed using a solid mechanics approach for the required torque capacity. The tripping mechanism actuates using the stored potential energy of the helical compression spring charged by the charging motor. The selection of the clutch depends upon the specified stiffness of the helical compression spring. The circuit breaking time is reduced by increasing the stiffness of the spring thereby increasing the torque induced at the clutch. The design parameters like gear modules, shaft sizes, and crank-connecting link sizes are evaluated for available spring force and torque requirements based on the specified size of circuit breakers. The graphical representation of the spring force and maximum torque at the driven unit with respect to the dead center is derived. The induced torque at the clutch end is measured and verified using a rotary torque sensor. The torque sensor values differed from the theoretical values by 3-5% within the standard acceptable limit of 10%.

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