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DEVELOPMENT OF 100 MW GAS TURBINE SHAFT SLEEVE PULLER

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Abstract

A Shaft Sleeve Puller was developed, designed and constructed in response to the need to pull-out / pull-in the 30-tonnes by 12m long rotor of the 100-MW gas turbine generator for inspection, as part of a refurbishment programme of a power station in Delta State, Nigeria. The design is a runway system, consisting of a platform incorporating rails on which the carriages run as they bear the rotor being pulled with a pulling jack. We utilized statically indeterminate approach to design the platform to very high flexural rigidity to avoid deflection, a necessary condition, to prevent damage to the generator field winding. The design was successful; the system constructed and used to pull-out/pull-in six rotors of the gas turbine generators; and is still being stored for further use when the need arises.

Keywords: shaft sleeve puller, turbine generator, pull-out/pull-in turbine generator, refurbishment, statistically determinate

1. Introduction

Power generating stations are built in many countries and oftentimes they are built in remote areas as dictated by the energy sources. There are hydro-electric, gas-turbine, steam-turbine, coal-fired, nuclear and hydro-thermal plants. New technologies, most of which are still being developed to harness different forms of energy, include wave, wind-turbine, solar-thermal, solar-photovoltaic, fuel-cells and other chemical methods. Traditional electricity generation is based on the principle of electromagnetic induction [1] and most of the sources enumerated above act as fuels to run the prime mover that drives the generator. A typical gas power plant comprises of the starter motor, the torque converter, the gas turbine, the gearbox and the generator [2]. There are, also, the air and gas intake, as well as, the exhaust systems. The central electrical source in the power plants is the generator modeled essentially of the stationary field winding and the rotor of wound coils [3].

The power plant units increase in size and weight as the generating capacity increases. The weight of the turbine of a 100-MW plant is 30 tonnes while that for a 780-MW plant is 250 tonnes. Thus, during maintenance, plants are often dismantled at location, and moved out for inspection and maintenance. The maintenance of power plants involves lifting of weights and shifting them around and this has posed a challenge to power plant maintenance. Sometimes, maintenance involves replacing turbine blades, gear boxes, alignment of units, air inlet modification and exhaust system alignment, as well as, insulation and cladding. This creates the need to pull out the rotors of the generator assembly for inspection, as part of the refurbishment/maintenance schedule. The alternative option would have been to import a pull-out / pull-in unit from the manufacturer at huge cost or have the turbine refurbished at the manufacturers facility in the US due to lack of local capacity.

2. Problem of Generator Rotor Puller

The generator rotor is a cylindrical drum of approximately 1.0m diameter × 12.0m long and has a dead weight of 30 tonnes mounted concentrically in the field winding of the generator. The main consideration in pulling out the rotor from the field winding is that the rotor must remain suspended in its horizontal position, maintaining the concentricity in the field winding so that it does not drop and scratch or damage the field winding. The space between the generator and the transformer bund-wall is only 14m and there is no space for the use of heavy duty mobile cranes.

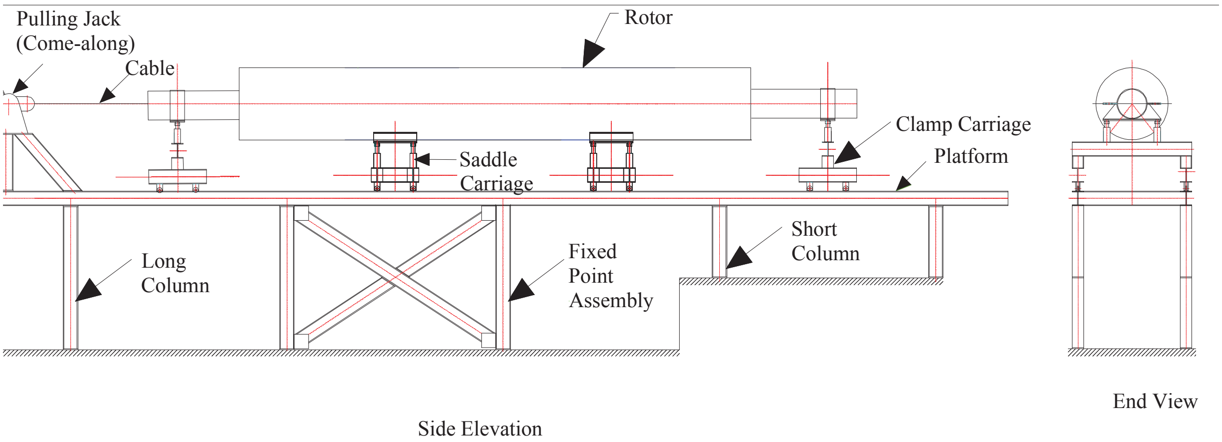

Figure 1: Conceptual design of the sleeve rotor puller mechanism.

The equipment manufacturer, General Electric, had not provided specific equipment or instructions for the pull out of rotors for inspection or description of such operation in the GE Reference Documents (GERs) [5]. However, equipment manufacturers were expected to supply special tools to be incorporated in the run-way method to pull in, pull out rotors of generators. When such are not supplied, maintenance or service teams will have to fabricate special tools for that purpose. A blog by Siswanto [6], described the removal of the rotor of a 125-MW generator. A jacking rail was constructed and a combination using jacking and chain block were installed to pull out the rotor. However, due to space and time constraints, the rails were supported by wooden framework. This illustrates decisions that have to be taken depending on site conditions [4].

2.1. Design calculations

2.1.1. Design considerations

1. A system that is rigid and strong enough to carry the weight of the rotor and whose columns should not buckle under the weight of the load, keeping horizontal members horizontal with minimal deflection.
2. The effort required to pull this dead weight out should be minimal so it must be designed for low friction.
3. The drum is not allowed to touch any part of the field winding, such that apart from horizontally, the body should remain as concentric as possible and hence must also be constrained on its lateral movement.
4. The system must be stable and not subjected to axial movement during operation.
5. The design should incorporate means of attachment for the application of the pulling force.
6. The system should be designed for mounting and dismantling because the unit should be closed after inspection and the system should be re-assembled for another inspection.

The design consists of a platform on which is mounted carriages with adjustable power screws to adjust the levels of the saddle and a come-along jack attached to a frame at one end of the platform to pull the rotor (fig. 1). The following are the features of the mechanism. The sleeve rotor puller mechanism was conceptualized to consist of a platform, the top of which will incorporate rails for tracking the rotor as it is being pulled out from the generator field winding sleeve. Pulling will be accomplished with a pulling jack (come-along) attached to a frame at the end of the platform. The following are the characteristics of the design.

1. The platform is of rigid but bolted construction and made of I-beams designed to withstand the 30 tonnes load.
2. There are two sets of columns due to the different elevations of the floor area for the mounting of the system.
3. The columns are struts or short columns with slenderness ratio ($L/r < 30$)[7] to prevent buckling due to the high load
4. Columns 3 and 4 are interconnected to provide fixed point for the structure and prevent axial movement during operation.
5. The carriages are on rollers, the ones on one side are grooved while those on the other side are plain. They are to run on rails fixed on top of the platform.
6. The grooving of only one side rollers keeps the carriage on a straight track while friction is reduced and possibility of jamming prevented.
7. All carriages incorporate adjustable screws to keep the rotor level and maintain its horizontality.

8. The end carriages have couplings to connect to the shaft of the rotor and eyes for attachment of the puller hook.
9. The inner carriages have the top parts of their saddles constructed in two halves, which were assembled in situ on top of the carriage as the rotor is being pulled out.
10. The top of the saddles and the clamps are lined with Neoprene materials to minimize scratches or damage to the rotor surface.

Fig. 1 shows the side elevation and end view of the conceptual design with the rotor fully on top of the platform, after pull out.

2.1.2. Column design

The platform is made up of the columns, major beams, cross beams and tie-beams. The column design comprised the determination of the load and reactions, stress calculations and size estimation of columns.

i. **Load and Reactions**

When the load is completely resting on the platform, the distribution of the carriages, the load transmitted through the rollers and the reactions from the column supports are as shown in the free body diagram of one arm of the platform (Fig. 2).

The load transmitted from the carriage rollers are at the mid-section between columns and are 700mm apart, which is the spacing between the rollers and are represented as W1 to W4. The reaction from the five column support have been designated RA to RE. The columns and beams are bolted and hence represent fixed supports. The structure is statically indeterminate [7] and [8]. However, the following assumptions were made to simplify the resolution of the forces and determine the reactions and bending moments at the supports.

1. The beam is considered to be made up of 4 segments each of which has both ends fixed.
2. Each segment carries 1/4 of the weight of the rotor and the weight is treated as point loads applied through the rollers 700mm apart at mid section of the span.
3. As a further simplification, the loads W1a and W1b are considered as a single load W1 acting at mid-span of segment
4. The beam is initially assumed weightless, an idealization necessary because the size of the beam was yet unknown.

Based on above assumptions, we applied the equations for the reactions and deflections of statically indeterminate beams with fixed ends [7] and [8].

$$R_A = R_{B1} = \frac{W_1 L_1}{2} \tag{1}$$

$$M_A = M_{B1} = \frac{W_1 L_1}{8} \tag{2}$$

$$\nu_{max} = \nu_c = \frac{W_1 L_1^3}{192EI} \tag{3}$$

Where, W_1 = weight of segment and L_1 = length of segment. For sections AB, BC, CD and DE

$$R_A = R_{B1};\ R_{B2} = R_{C1}\ R_{C2} = R_{D1}\ R_{D2} = R_E \tag{4}$$

Mass of rotor = 30 tonnes = 30,000 kg; hence weight of Rotor = 300 kN. Load on one beam = 150 kN; load on each segment of beam, $W_1 = W_2 = W_3 = W_4 = \frac{150}{4} = 37.5$kN. Reactions at columns $R_A = R_{B1} = R_{B2} = R_{C1} = R_E = \frac{37.5}{2} = 18.75$kN. Reactions at $R_B = R_C = R_D = R_{B1} + R_{B2} = 37.5$kN. Thus, the three inner supports are each carrying twice the weight carried by the outer supports. The load per roller (8 rollers per beam) is $W_{1a} = W_{1b} = W_{2a} = W_{2b} = W_{3a} = W_{3b} = W_{4a} = W_{4b} = \frac{150}{8} = 18.75$kN.

ii. **Size Estimation**

Since the inner columns, B, C and D, bear the highest load, maximum stress is experienced in these columns and the column design is based on forces and moments on these members. For rigidity, the columns are assumed to be short columns ($L/r < 30$); i.e. the slenderness ratio or ratio of length of column to radius of gyration is less than 30. The lengths of the long columns are taken as 2m, determined by the vertical height of the central axis of the rotor drum and after making provision for the carriage.

We assume $L/r = \lambda = 25$; therefore, $r = 80$ mm. An I-beam with a radius of gyration, $r = 80$ mm was read from American Standard I-beam. Table A-7 [7], which is S 200 × 27 for narrow beams with the nearest r of $r = 82.8$. The nearest to the choice above in the International Standard Organization reference for I-beams in the narrow flange I-beam is the ISMB 200, with corresponding radius of gyration $r = 93.1$ and has weight of 25.4 kg/m, depth of section, $D = 200$mm and width of flange, $b = 100$mm [1].

iii. **Stress Analysis**

For the short column, the stress is contributed by both the axial load and the bending moment. That is,

$$\text{Maximum Stress, } \sigma_{max} = \frac{W}{A} + \frac{M}{z} \tag{5}$$

Where, W = axial load; A = area; M = bending moment and z = section modulus. The maximum axial load and bending moment are on columns B, C and D, hence designing for either B,C and D would be adequate.

Considering column B, and substituting in Eq. 5, we have

$$\therefore \sigma_{max} = \frac{W}{A} + \frac{M}{z} = 130\text{MPa}$$

Therefore, the maximum stress due to axial load and bending moment as a result of eccentric loading is 130 MPa. We had chosen a structural steel with yield

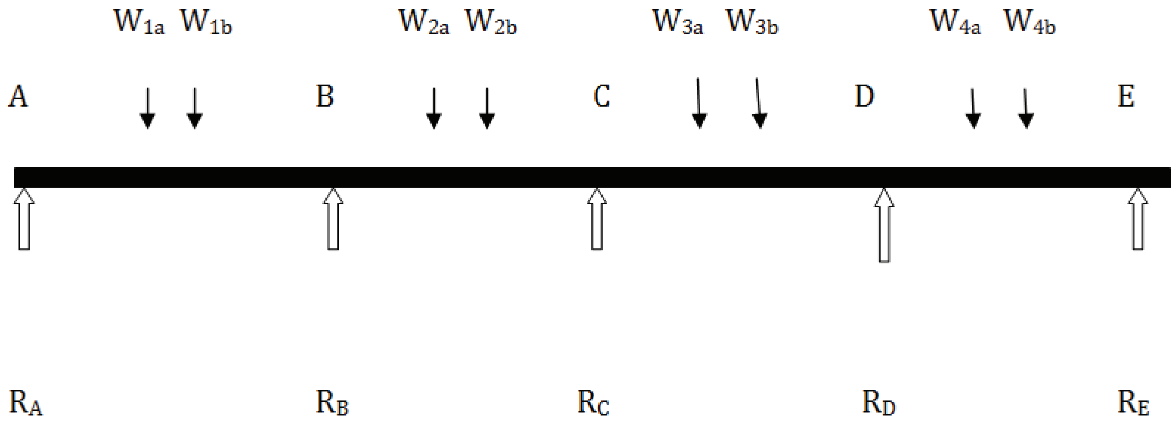

Figure 2: Free-body diagram of platform load and reactions.

stress of 250 MPa. The permissible axial load in compression for columns of slenderness ratio, $\lambda = 25$ is 146 MPa [10]. (see appendix A).

iv. **Deflection of Beam**

The deflection of the beam was investigated because the platform is required to have high flexural rigidity or near zero deflection. The deflections due to the point load and own-weight were considered in the analysis that follows.

The maximum deflection is given by Eq. 3 assuming a weightless beam with point load.

$$\nu_{max\ p} = \nu_{cp} = -\frac{W_1 L^3}{192EI} \tag{6}$$

Therefore, $\nu_{max\ p} = \nu_{cp} = -1.0986$mm. The deflection due to the point load = - 1.0986 mm.

The maximum deflection of a beam fixed at both ends is [10]:

$$\nu_{max\ d} = \nu_{cd} = -\frac{wL^4}{1384EI} \tag{7}$$

Where w = weight per m of the beam = 27 kg/m or 270 N/m. Therefore, $\nu_{max\ d} = \nu_{cd} = -0.0119$mm.

Hence, total deflection $\nu_{max} = \nu_{max\ p} + \nu_{max\ d} = 1.0986 + 0.0119 = 1.11$mm. The deflection of 1.11mm is permissible to maintain horizontality and rigidity since for this beam span/325, which in this case, would be 9.2mm, is allowed for structures [9].

v. **Bending Stress in Beam**

The maximum normal stress due to bending of this fixed beam is determined from the equation:

$$\sigma_{max} = \frac{M_1}{z} \tag{8}$$

Where

$$M_1 = \frac{W_1 L}{8} + \frac{wL^2}{12}$$

With M_1 = bending moment contributed by both point load and weight of the beam; W_1 = weight of segment 1; w = weight per m and L = length of segment. Calculated value for M_1 gives = 14062.5 (Nm) + 67.5 (Nm) = 14.13 kNm and substituting in Eq. 8 would yield $\sigma_{max} = 59.59$ N/mm^2.

The permissible maximum stress, as a function of the slenderness ratio ($D/T = 18.8$) and $r = 87$ would yield 130MPa [10]. The design of the platform was deemed satisfactory. However, due to none availability of the above size of I-Beam as determined, the platform was redesigned for a beam with slenderness ratio $\lambda = 20$ leading to the choice of wide flange I-beams, W250 x 67, used for the construction.

2.1.3. Leveling bolt

The Leveling Bolt bears the rotor drum with the four (4) carriages, each with four (4) bolts supporting the saddle. The bolts act as unrestrained simple support. Bolt design is based on tensile strength but possible failure is tested against shear force on the thread of the bolt and nut. The design stress for bolts by Seaton and Routhewaite [10] is taken as:

$$\sigma_d = \frac{\sigma_y (A_s)^{1/2}}{6} \quad \text{or} \quad F_d = \frac{\sigma_y (A_s)^{3/2}}{6} \tag{9}$$

Where σ_d = design stress; σ_y = yield stress; A_s = tensile stress area; and F_d = design load.

This is a high load bolt design and high strength material is required. Alloyed Medium Carbon Steel, Delta Steel Company Standard HTS 320, equivalent to AISI 4140 was recommended for the production of the bolt, [11], [12]. This is a heat treatable steel that could be used directly as normalized. The material, AISI 4140, does not exhibit defined yield stress so the design was based on the ultimate stress.

Usually, $0.4\sigma_y$ is used when the diameter of the bolt is above 18mm but when the determining stress is the ultimate stress, the equation will have to be modified by a new factor of safety based on ultimate stress of

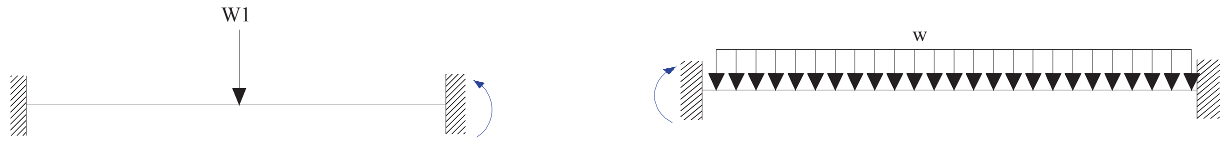

Figure 3: Free-body Diagram of Segment of Main Beam in Deflection and Free-Body Diagram of the Distributed Load of Beam Self-Weight.

at least 4. Therefore,

$$\sigma_d = \frac{\sigma_y (A_s)^{1/2}}{24} \text{ or } F_d = \frac{\sigma_y (A_s)^{3/2}}{24} \tag{10}$$

From equation 10, calculated value for $A_s = 58.989\text{mm}^2$ which is equivalent to metric thread bolt M10 with bolt diameter = 10 mm. However, considering the free length of bolt (300mm to give necessary adjustment clearance), and the need for it to be sturdy and not buckle in operation, a bolt size with slenderness ratio of 15 was chosen. This gave a bolt of 20mm radius or 40mm diameter. Actually a 2″ or 50mm diameter × 300mm length made from HTS 320 or AISI 4140 was used.

2.1.4. Others elements

Other elements like the bearings, rollers and pulling jack were chosen with reference to manufacturing manuals and accepted operating load. Bearing was Nu2208 Roller bearing for load above 22 kN. Rollers were machined from HTS 320 while an 8 Ton pulling jack was chosen when the calculated cable tension was taken as 4.5 Ton for assumed coefficient of friction of 0.15.

3. Construction

The platform consists of two straight full length of W250 × 67 × 12m long I-beam on which were laid 50 × 50 bars to serve as rails for the carriages. The beams were connected with cross beams, end-ground and welded to end-plates, to fit the web. The interconnected frame was placed on three pairs of long columns and two pairs of short columns; all with base plates for bolting to the beam and to ground. Columns 3 and 4 were interconnected as fixed point to prevent axial movement during operation. The frame for attachment of the pulling jack was fixed to the end of the platform (figure 4).

The main feature of the saddle carriage is the sliding top (figure 5). The top, made of 10mm plate, was rolled to fit using a radius of 500mm and an included angle of 97°. The curved top is welded to the sliding plate and supported by gussets. This sliding unit is cut longitudinally in two and usually slid from either side of the rotor during operation. The top is lined with conveyor belt to prevent metal-to-metal contact.

Fig. 6 showed the clamp carriage and it incorporated the clamp made from 16″ schedule 40 pipe. To this, is welded a flange containing the eye for jack connection. The clamp is also lined with conveyor belt to avoid metal-to-metal contact.

All carriages are fitted with rollers machined from AISI 4140 or Delta Steel Grade HTS320. One set of rollers was grooved while the other set was plane. The roller shafts were also machined while the roller bearings were bought off-shelf. Meanwhile, fig. 8 shows some of the elements of the mechanism for connecting the members. The scan of the photograph of the full assembly, taken after the construction, is shown in figs. 9–13.

4. The Operation

The system was assembled at point of use starting with the columns and beams to build the platform. The frame was next mounted at the far end of the platform for the attachment of the pulling jack (come-along). The first carriage, complete with clamp-assembly, was lifted on to the platform and clamped on to the rotor shaft. The pulling jack is then connected to the clamp by two cables and the pulling was ready to start. The shaft at the other end or rear shaft of the rotor is relatively long so appreciable length of the rotor (about a quarter) was pulled out before the rotor cleared the last bearing. By this time, the first saddle-carriage was mounted before the shaft cleared the bearing. The carriage is mounted in parts. First, the carriage base was lifted with crane, supported by the personnel, and maneuvered on to the platform. Next, the saddle was assembled from opposite sides of the drum and bolted in place. Finally, the bolts were adjusted to move the saddle up to bear on the drum. However, the tension on the cables and the two supports were still not enough to keep the drum from tilting over at the far end.

This means the design could not fully take care of the horizontality of the drum as envisaged. Two possible solutions were considered: (1) to extend the rear shaft or (2) to provide shims in the space between the rotor and the field winding. The first option was immediately discarded because there was no space for such attachment and the process of producing such attachment was going to be expensive and cumbersome.

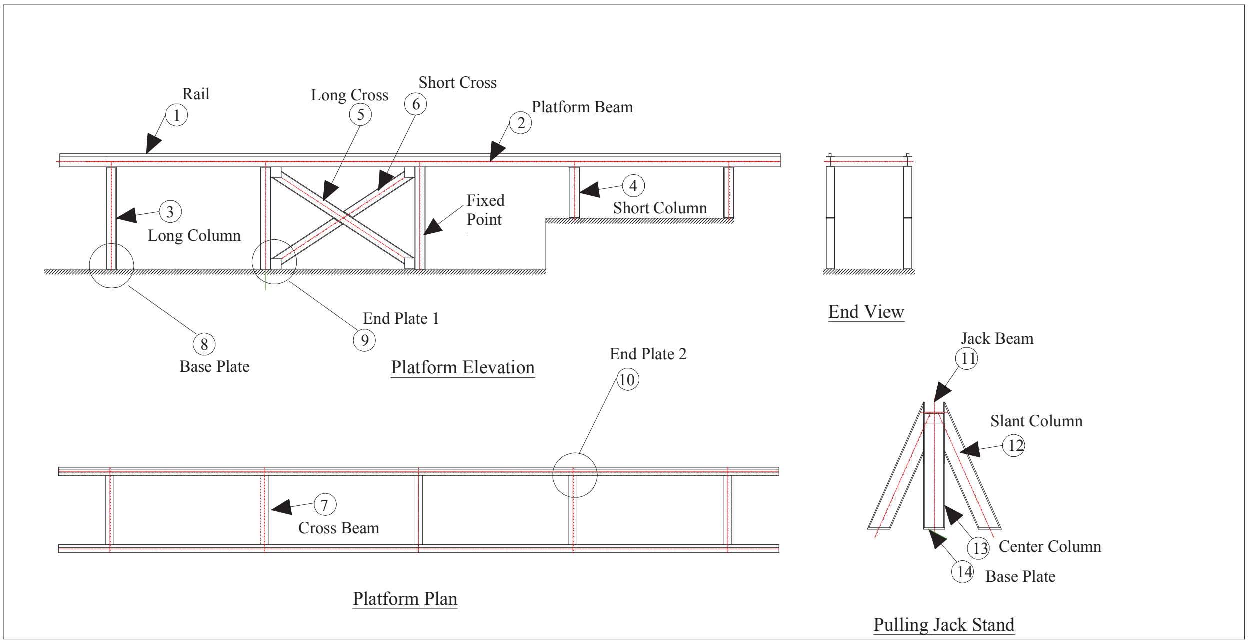

Figure 4: Details of platform construction.

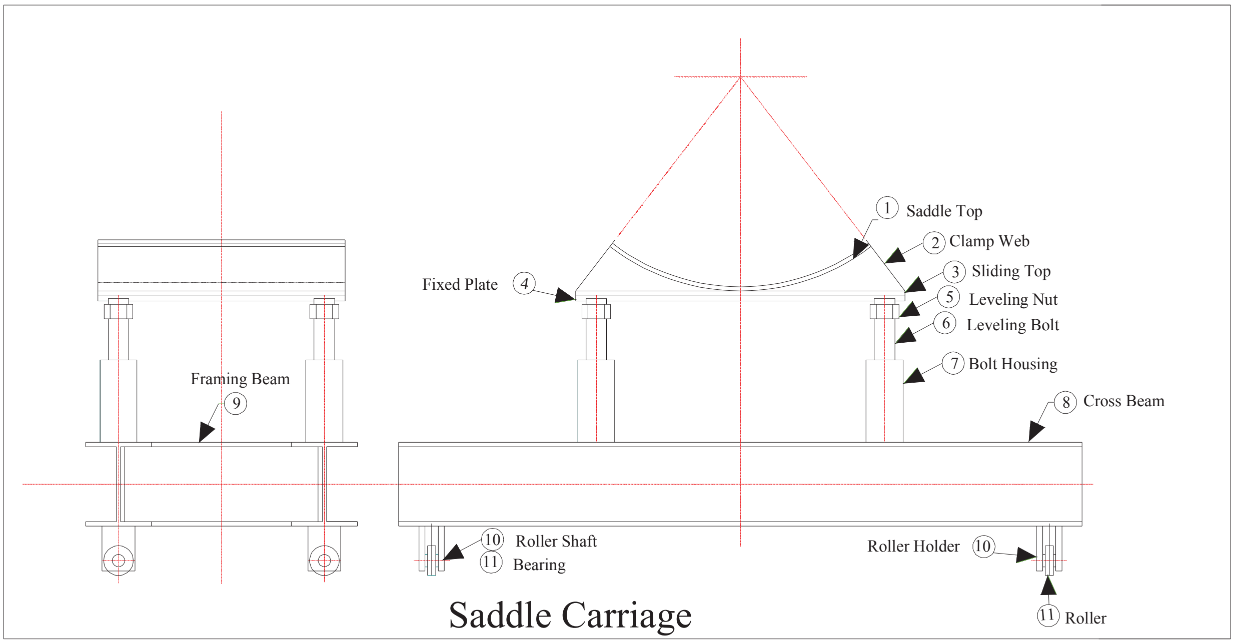

Figure 5: Details of saddle carriage construction.

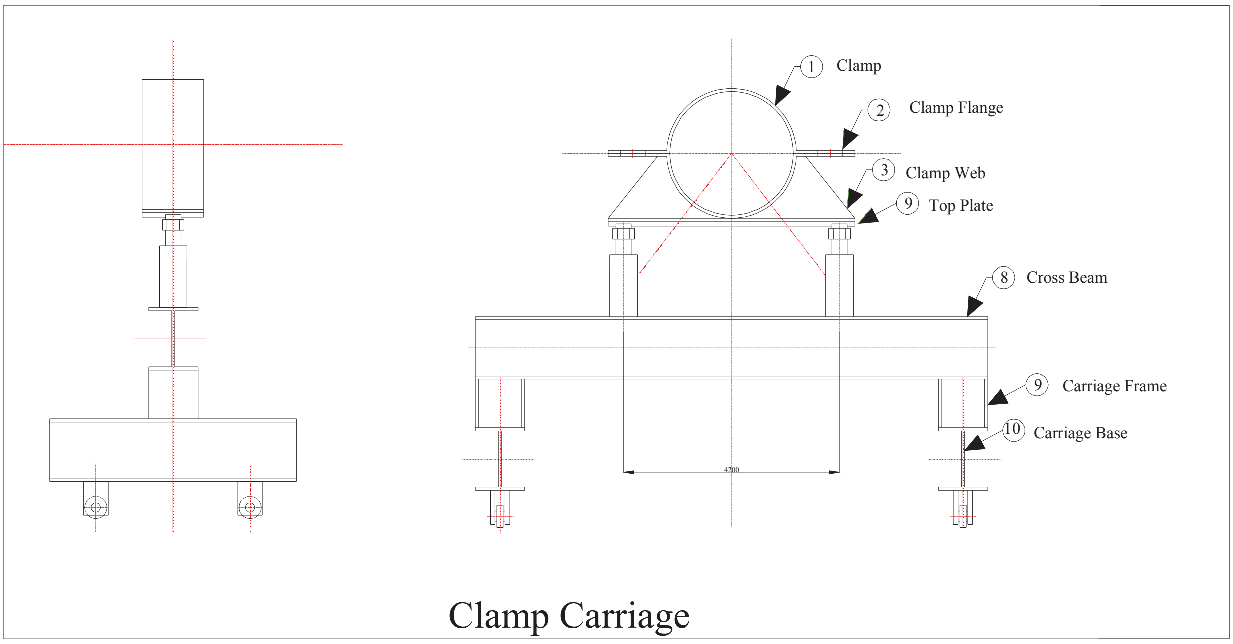

Figure 6: Details of clamp carriage.

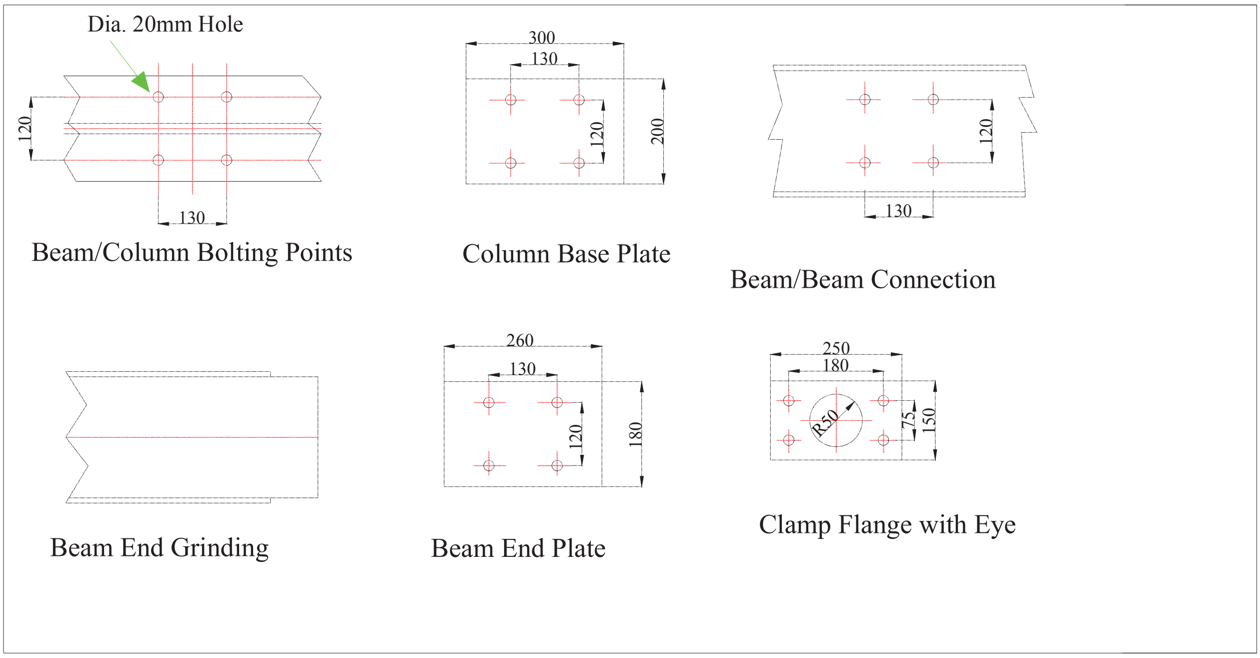

Figure 7: Details of connection points.

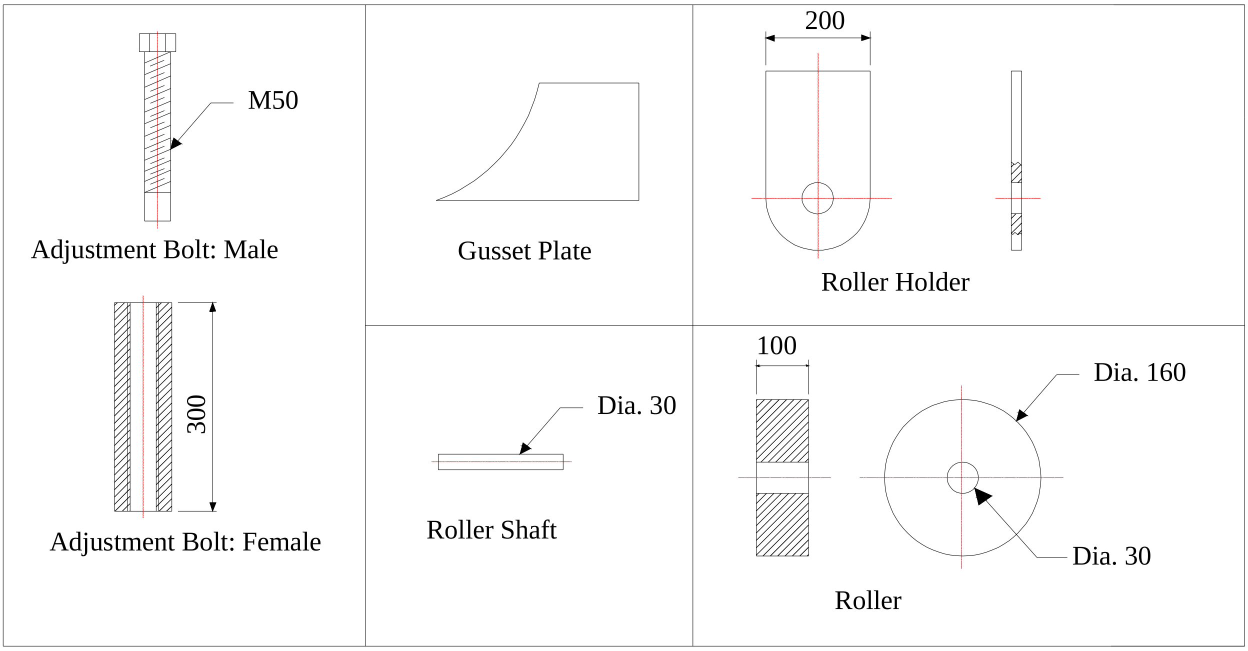

Figure 8: Adjustment bolt, roller components and gusset plate.

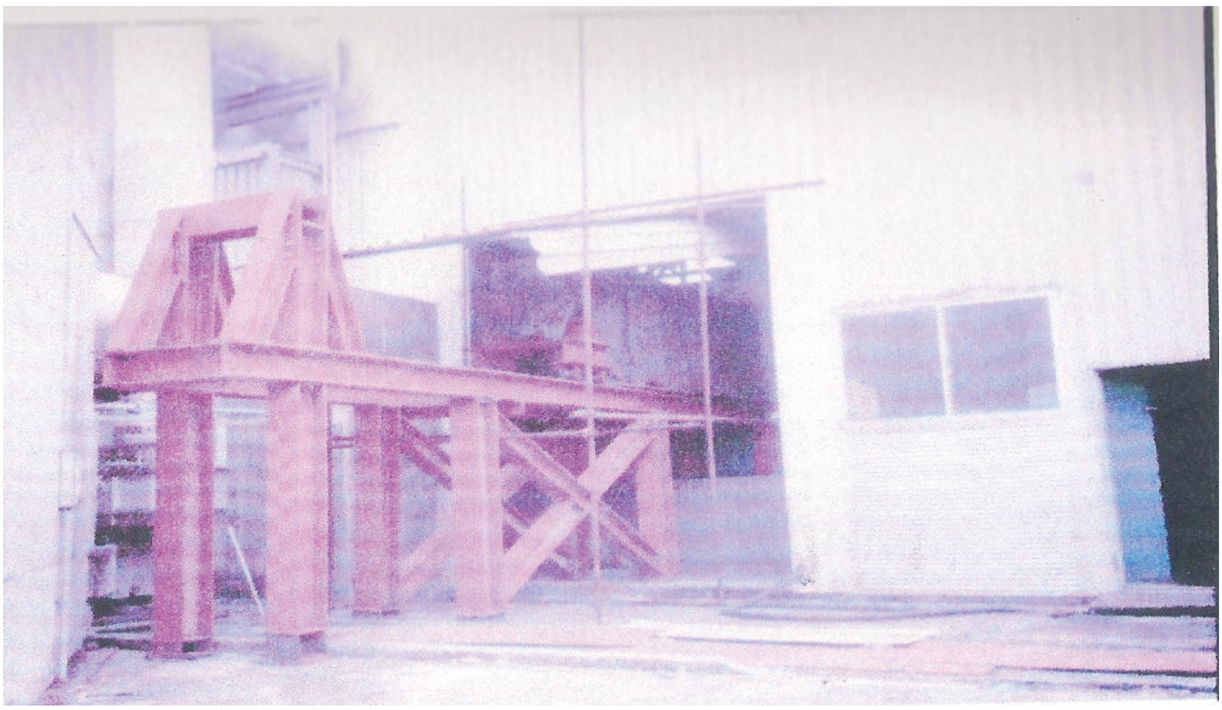

Figure 9: As-built photo of the shaft-sleeve puller mechanism.

Figure 10: Shaft-sleeve puller-platform parts.

Figure 11: Saddle carriage.

Figure 12: Part of clamp carriage showing rollers.

Figure 13: Pulling jack stand with saddle carriage.

The use of shims provided a better option though that means the rotor will have to rest on the field winding. Two sets of shims were used to overcome the problem. The lower part of the field winding was covered with shims made from Teflon and fixed so it does not move. That way, the winding was protected from scratch or damage. On the Teflon was placed hard wood planks. The rear shaft could rest and slide on the plank, which could also slide over the Teflon. This way, the rotor was pulled further onto the platform until another saddle-carriage was assembled to lift the rotor off the shims.

The mechanism was successful in pulling the rotor from the generator for inspection and the rotor was re-assembled by simply pushing it back while dismantling the supports. It was used for the inspection of five rotors, at the last count, and can still be used for any other rotor in the 100 MW Gas Turbine Electricity Generating Stations that has to be refurbished.

5. Conclusion

The mechanism, designed and constructed as described above, was successfully used to pull-out/pull-in the rotor of six 100-MW generators from their field windings. After the first rotor was pulled out, inspected, confirmed okay and pulled in, the system was dismantled and re-assembled for the same operation for another generator. The shaft-sleeve puller had been used for the six Delta IV 100-MW Gas Turbine Generators, GT15 to GT20 and is stored for the next round of inspection. The dismantled component parts are shown in Fig. 10 to Fig. 13, and can be used continuously if the components are not deliberately damaged or stolen.

The design, also, illustrates the application of equilibrium method and principle of superposition to solve statically indeterminate problem in an industrial setting. Using established formulae, the dimensions of structural members with enough flexural rigidity to build a stable frame to withstand the 30 tonne load were determined. Material choices were made from established industrial practices and/or manufactured, were possible.

Acknowledgement

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2. M. D Ode for his assistance in checking the drawings.

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Appendix

The full properties of the beam S200 $\times$ 27 are:

Structural Material: ASTM A-36
Overall depth of beam, $D = 203$ mm
Mean thickness of compression flange, $T = 10.8$ mm
Radius of gyration about x-x, $r_x = 82.8$ mm
Radius of gyration about y-y, $r_y = 24.2$ mm
Thickness of web, $t = 6.9$ mm
With of flange, $b = 102$ mm
Cross sectional area, $A = 3.5 \times 10^3$ mm^2
Moment of Inertia, $I = 24 \times 10^6$ mm^4
Section Modulus, $z = 236 \times 10^3$ mm^3
Modulus of Elasticity, $E = 200$ GPa
Yield Stress = 250 MPa
The ultimate stress tensile strength of AISI 4140 is 980 N/mm^2 (MPa).