KWG12 ROTOR 1ST STAGE REPAIR: STATIC BALANCING, SIMULATION AND RE-BLADING

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Abstract

Olkaria Wellheads has grappled with premature fracture of 1st stage blades on C50 geothermal steam turbines. On December 29th, 2018, KWG12 failed to progress beyond critical speed owing to high turbine vibrations after the turbine suffered two fractured 1st stage blades. The rotor needed re-blading and balancing. It was considered cost-effective to do in-house repairs using 1st stage blades recovered from a grounded C50 rotor. Repairs entailed static balancing of all 1st stage blade masses, simulation, and re-blading. Success was epitomized by timely availing of the plant after 15 days outage, at 5000kW rated capacity and normal vibrations of 30µm. The initiative has kept the plant operational as new blades are procured, besides cushioning the plant from incurring downtime losses arising from long procurement lead time. Whereas the 15 days repairs incurred KShs. 16.9 million in direct and indirect costs, the plant accrues over KShs. 17.4 million monthly in revenue.

Keywords: Re-blading, static balancing, turbine blades, vibrations, 1st stage.

1 Introduction

KWG6 and KWG8 manifested premature fracture of 1st stage blades during project implementation. As a result all C50 turbines on site were re-bladed with modified blades. Within 2½ years of operation after re-blading, four C50 turbines have again fractured 1st stage blades. The four failed turbines were statically balanced and re-bladed on-site with used blades recovered from a grounded C50 rotor. This strategy was adopted to cushion against slumping into downtime and production losses a direct consequence of long lead times associated with overseas procurements.

The threat of more C50 plants imminently capitulating to repeat blade fractures and plant shutdowns is real and needs serious attention. Blade failure mechanisms comprise fatigue fracture caused by cyclic dynamic and centrifugal steady stresses, resonance and environmental effects. Underlying factors of fracture include: inadequate blade design, material selection, manufacturing processes, operating conditions and maintenance practices. [5] [8].

On 29th December 2018, KWG12 fractured two blades on the 1st stage. This disrepair was untenable and likely to lead to serious damage. Onsite repairs was considered because it provides the most time-efficient and cost-effective means of plant maintenance.

2. Methodology

In-house re-blading was adopted on the premise that it expedited resumption of operations as procurement of new blades was pursued hence minimize downtime losses arising from long lead times. Re-using used blades was judged not to represent a significant level of risk. Blade alloy X5CrNiMo16 was selected as substitute to the current alloy X20Cr13 for its superior heat treatment, corrosion resistance and mechanical properties [7].

Static Balancing and Re-Blading

Extracted blades were cleaned and scanned for cracks by X-ray radiography as criteria for reuse and weighed on an accurate digital scale. On-bench static balancing of weighed blade masses involved arranging blade of closest masses in polar angles 180° apart so that centre of gravity of blade masses coincides with imaginary axis of shaft rotation. Blades were transferred on the rotor in same order. [1]

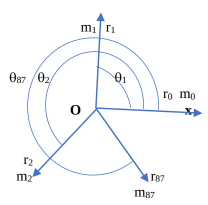


Fig. 4 Blade radial allotment

Fig 5 On-bench array

3. Results

Analytical Method

1. Radially outward centrifugal force exerted by each blade mass was determined by vector sum of centrifugal forces

   F = $F_1 r_1 \omega^2 + F_2 r_2 \omega^2 + F_3 r_3 \omega^2 + \ldots\ldots\ldots + F_{87} r_{87} \omega^2$

   If F was zero, the masses were statically balanced.

2. Centrifugal forces were resolved in the horizontal and vertical planes and their total sum determined. They didn't add to zero; implying the rotor retained residual unbalance. [1]

   Horizontal components of centrifugal force

   $\Sigma H = m_1 r_1 \text{Cos}\theta_1 + m_2 r_2 \text{Cos}\theta_2 + \ldots + m_{87} r_{87} \text{Cos}\theta_{87} + m_o r_o Cos\theta_o$

   ΣH = - 0.014

   Vertical components of centrifugal force

   $\Sigma V = m_1 r_1 \text{Sin}\theta_1 + m_2 r_2 \text{Sin}\theta_2 + \ldots + m_{87} r_{87} \text{Sin}\theta_{87} + m_o r_o Sin\theta_o$

   ΣV = -0.01891

3. Determine the magnitude of the resultant centrifugal force

   $F_o = \sqrt{[(\Sigma H)^2 + (\Sigma V)^2]} = 0.02328453$

   $m_o r_o Cos\theta_o = -\Sigma \text{ mrCos}\theta$ (1)

   $m_o r_o Sin\theta_o = -\Sigma \text{ mrSin}\theta$ (2)

   Squaring, adding and solving equ. 1 & 2 gives

   $m_o r_o = \sqrt{[(\Sigma \text{mrCos}\theta)^2 + (\Sigma \text{mrSin}\theta)^2]}$ (3)

   $= 0.02328453$

Required counter (or unbalance) mass

$m_o = \frac{\sqrt{[(\Sigma \text{mrCos}\theta)^2 + (\Sigma \text{mrSin}\theta)^2]}}{r_o} = \frac{0.023528453}{0.275}$ (4)

$= 0.085558\text{g}$

4. Angular position of resultant eccentric force with horizontal plane

   $\text{Tan}\theta_o = \frac{-\Sigma \text{ mrSin}\theta}{-\Sigma \text{ mrCos}\theta} = \frac{-0.01891}{-0.014} = 1.3507143$

   $\theta_o = \tan^{-1}\left(\frac{\Sigma V}{\Sigma H}\right) = 53.49° = 233.486°$ (5)

5. The balancing force equals the resultant force but in opposite direction.
6. Magnitude of balancing mass

   $F_o = m_o r_o$, hence $m_o = \frac{F_o}{r_o}$ (6)

A. Forced vibrations

Natural frequency is inherent in systems. It is a function of k and M; Damping factor ξ is the ratio of actual damping coefficient C, to critical damping coefficient, C_c. Where damping coefficient, $\xi = \frac{C}{C_c}$. [4]. In Wellhead rotors ξ=0.06, hence they are designed as under damped. [2]. Where: $\omega_n^2 = \frac{k}{m}$

Damped natural frequency, $\omega_d = \sqrt{1-\xi^2}\,\omega_n$ rad/sec

$\omega_d = 418.124$ rad/sec

Natural period of vibration $T = \frac{2\pi}{\omega} = 2\pi\sqrt{\frac{m}{k}}\ sec$

T = 0.015 sec

Natural freq. of vibrations $f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}\ cycles/sec$

$f_n = 66.67\ cycles/sec$

<u>Simulation</u>: <u>Forced response rotating unbalance [4]</u>

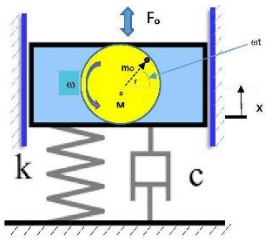


Fig. 6. Unbalance Harmonic [4]

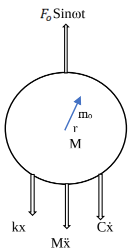
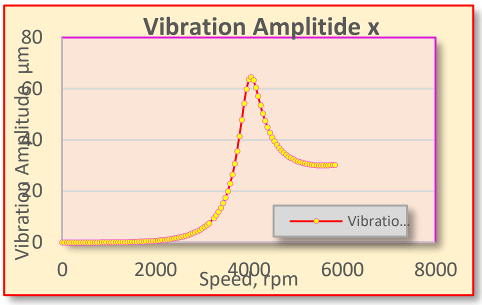


Fig. 7 Resolution of forces



Graph 1. Simulated KWG12 Vibration profile

Where:
F = Damping resistance (N)
C = Damping Coefficient (N-sec/m) = 80kN-sec/m
Critical damping Coeff. $C_c = 2\sqrt{kM}$ = 1,340kN-sec/m
k = System stiffness (N/m) = 280MN/m
m_o = unbalance mass (kg) = 8.556x10^{-5}kg
r = eccentricity of unbalance mass (m_o) = 0.275m
M = Mass of machine + unbalance mass = 1600kg
m_o makes an angle ωt with reference axis
$m_o r\omega^2$ = Centrifugal force created by m_o at r
Damping ratio, $\xi = \frac{C}{C_c}$ = 0.06
Natural speed ω_n = 4000rpm or 418.88 rad/sec. [3]

<u>Equations of motion</u>

$$(M - m_o)\frac{\partial^2 x}{\partial^2 t} + m_o \frac{\partial^2 (x + rSin(\omega t))}{\partial^2 t} = -kx - c\frac{\partial x}{\partial t} \qquad (7)$$

Generates linear non-homogeneous 2nd ODE

$$m\ddot{x} + c\dot{x} + kx = m_o r\omega^2 Sin(\omega t). \quad [4] \qquad (8)$$

From Newton's 2nd Law

$$m\ddot{x} + c\dot{x} + kx = F_o \,\mathrm{Sin}(\omega t) \qquad (9)$$

where

$$F_o = m_o r\omega^2$$

$$x = X_c(Complimentary) + X_p(Integral)$$

The differential equation describing the system

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n x = \frac{F_o Sin\ \omega t}{M} \qquad (10)$$

Solving for x, gives

$$x = A_2 e^{-\xi\omega_n t}\, Sin\left[\sqrt{1-\xi^2}\omega_n t + \varphi_2\right] + \frac{F_o}{k}\frac{Sin(\omega t - \varphi)}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\xi\left(\frac{\omega}{\omega_n}\right)\right]^2}} \qquad (11)$$

where $X_{st} = \frac{F_o}{k}$

First term is *Transient*, soon dies out leaving the second term, the *Steady State* vibrations.
The Steady State solution of Equation 11 is

$$x = X\,\mathrm{Sin}(\omega t - \varphi)$$

Where

$$x = \frac{\frac{m_o r\omega^2}{k}}{\sqrt{\left[\left(1-\frac{M\omega^2}{k}\right)^2\right] + \left(\frac{c\omega}{k}\right)^2}} \qquad (12)$$

Dividing numerator and denominator by k where
$k = M{\omega_n}^2$ = 280,736,727N/m

and substituting $\xi = \frac{c}{c_c}$ = 0.06

Gives dimensionless form:

$$x = \frac{\frac{m_o r}{M}\left(\frac{\omega}{\omega_n}\right)^2}{\sqrt{\left[\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2+\left(\frac{2\xi\omega}{\omega_n}\right)^2\right]}} \qquad (13)$$

$$\varphi = \tan^{-1}\left[\frac{2\xi\frac{\omega}{\omega_n}}{1-(\frac{\omega}{\omega n})^2}\right] \quad [4] \qquad (14)$$

Substituting for ω_n, m_o, M, r and ξ in Equation (13) gives:

$$x = \frac{8.381\ x\ 10^{-14}\omega^2}{\sqrt{[(1-(5.6993\ x\ 10^{-6}\omega^2)^2+(2.86479\ x\ 10^{-4}\omega)^2]}} \qquad (15)$$

Equation (15) relates amplitude of vibration to angular speed, ω. Represented by simulation Graph 1 below. From the graph the highest vibration of 66μm attained is within operating limits; brightening prospects of a successful run with residual unbalance of 0.0856g.

Table 1: Existing 1st Stage blades material composition

Elements	C %	Cr %	Ni %	S %	Mn %	Si %	P	Mo
Existing blades X20Cr13	0.18 ~ 0.22	12 ~ 14	-	<0.03	< 1.0	< 1.0	<0.045	-
Recommended	<0.06 max	15 ~ 17	4.0~ 6.0	<0.015	<1.50	<0.7	<0.035	0.8 ~ 1.5

Sources: MAN Energy Solutions, SA [3]

Table 2: 1st Stage blades Mechanical Properties

Standard	Yield Strength (MPa)	Ult. Tensile Strength (MPa)	Impact (DVM) J	PRENeff (>30)
Existing blades	550	800	>25	10
Recommended DIN (1.4418)	>850	>1000	>55	>20

Source: MAN Energy Solutions, SA. [3].

4. Discussion

Blade weaknesses coupled with exposure to highly detrimental working conditions is a formula for premature failure. Stresses are countered by such techniques as making the blades stronger by using superior designs, better quality materials and adding damping by caulking strips and lashing. Use of corrosion resistance materials improves reliability and longevity of service. Designs must employ a generous safety margin of at least three.

This report recommends modification of C50 turbine 1st stage blades to stainless steel X5CrNiMo16 DIN 1.4418 which is heat treated by precipitation hardening and exhibits high strength, heat and wear resistance, better corrosion resistance, enhanced reliability and is machinable. [3].

4. Conclusions

KWG12 post-reblading performance exceeded expectations. Smooth starting, normal vibrations till full speed was the first sign of success. The plant has since maintained rated capacity of 5000kW at maximum rotor vibrations of just 30/30μm with impeccable operational parameters. [2].

This report recommends blade re-design with superior quality material preferably stainless steel X5CrNiMo16 DIN 1.4418 and redesign of blade profile.[3]. The report further advocates for use of generous a safety margin of at least three.

Acknowledgement

The authors appreciatively acknowledge the invaluable supports of Eng. F. Kawa, Wellheads Team, Chief Engineers of Olkaria plants and Olkaria Steam Field Team.

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