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Design and Analysis of Steam Turbine Blade using FEA

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Abstract: Steam turbine converts the heat energy of steam into useful work. Steam jets strike the moving rows of blades mounted on rotor causes change in the direction of steam which imparts momentum. Thus blades convert the kinetic energy of steam into the rotational energy of shaft. Moving blades in a turbine are loaded by centrifugal forces and steam forces. Depending upon the design and operating conditions, centrifugal force may develop tensile, compressive or torsional stresses in moving blade. Steam turbines are subjected to number of start-ups and shut-downs during its life span. That means it is subjected to repetitive cyclic loading conditions which causes a fatigue failure of moving blades. This project summarizes structural performance of the blade due to centrifugal loading that acts on the blade due to high angular speeds. Also fatigue or service life of blade is estimated.

Keywords: Steam turbine; Moving blade; Fatigue life.

I. INTRODUCTION

In this study, first strain-controlled deformation and fatigue life are calculated for Stem turbine blade and then they are compared with ANSYS results. Various approaches to estimating mean stress effects on strain-life analysis are Morrow method and Smith, Watson, and Topper (SWT) method employed here to estimate the fatigue life of steam turbine blade. Morrow using the true fracture strength is a considerable improvement. However, the Morrow expression employing the fatigue strength coefficient σ_f' may be grossly non conservative for metals other than steels. The Smith, Watson, and Topper (SWT) method is a reasonable choice that avoids the difficulties. A steam turbine is a device that extracts thermal energy from pressurized steam and uses it to do mechanical work on a rotating output shaft. In this case, the pressure and flow of steam rapidly turns the rotor. The nozzles and diaphragms in a turbine are designed to direct the steam flow into well-formed, high-speed jets as the steam expands from inlet to exhaust pressure. The rotating blades convert the kinetic energy into impulse & reaction forces, caused by pressure drop, which together result in the rotation of the turbine shaft or rotor. The blades in the high-pressure (HP) and Intermediate pressure (IP) turbine are small and medium because of the low volumetric steam flow. Basic features of the short blades typical of HP turbines are identified. Shrouds (also called covers or connecting bands) provide a sealing surface for radial steam seals and are used to minimize losses due to leakage. While many parts may work well initially, they often fail in service due to fatigue failure caused by repeated cyclic loading. Characterizing the capability of a material to survive the many cycles a component may experience during its lifetime is the aim of fatigue analysis. In a general sense, Fatigue Analysis has three main methods, Strain Life, Stress Life, and Fracture Mechanics.

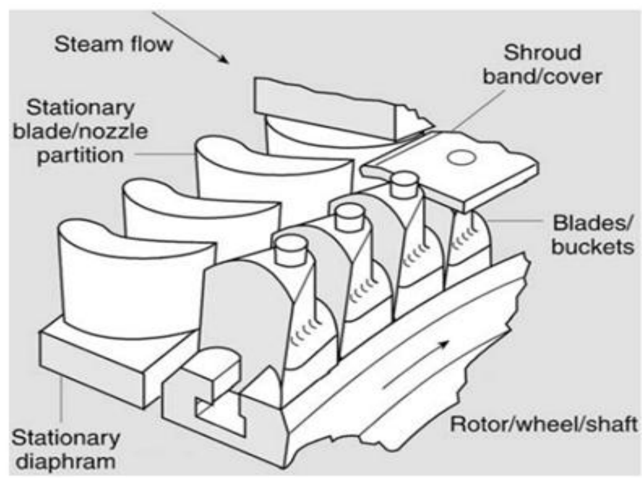

Fig.1. Schematic arrangement of stationary and rotating blades.

II. LITERATURE SURVEY

A. Moving blade

The moving blades convert the kinetic energy into impulse and reaction forces, caused by pressure drop, which together result in the rotation of the turbine shaft or rotor. Any moving blade can be briefly nomenclature as profile and root. But these can be further named as follows

- Tang: It is the base portion of the root.
- Neck: It is the middle portion of the root.
- Shoulder: It is upper portion of the root.

- Airfoil section: Airfoil is cross section of the blade profile.
- Tenon or Tip: Tenon is top most portion of the profile. It is used as rivet for shroud attachment.

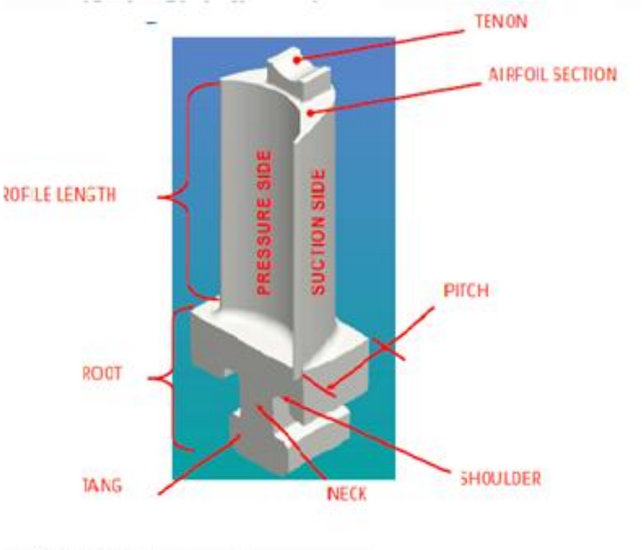

Fig.2. Moving Blade Nomenclature.

B. Moving Blade classification

Moving blades are classified into many types based on their working principle, type of profile, type of root and so on. All those classifications are shown in below figure 3.

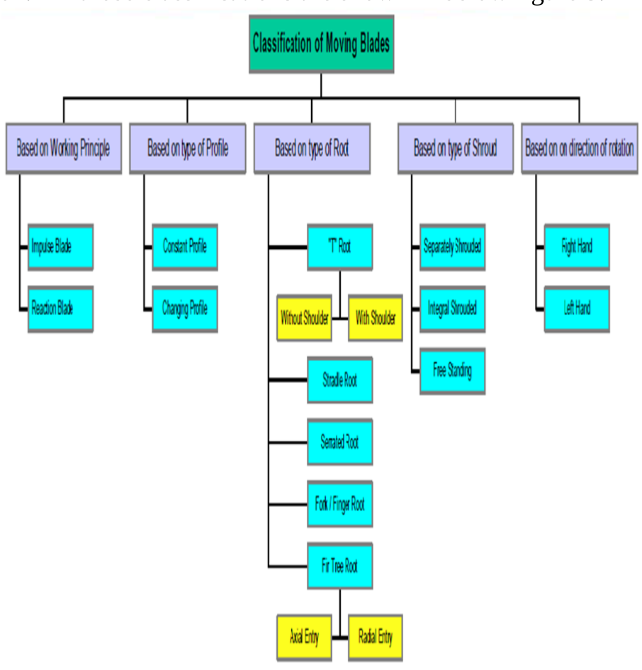

Fig.3. Classification of moving blades.

C. Required Material Properties for steam turbine blades

The selection of materials for steam turbines is very competitive and an important factor in the overall cost. Generally shafts of these steam turbines are manufactured from either rolled bars or forging. The materials that are used for these steam turbines are different types of steels such as stainless steel and alloy steels like12% Chromium steel and 2NiCrMoV steel. Choosing the optimum blade material is an ongoing tradeoff between desirable material properties. In addition, it is important that blading material be weldable, particularly last-stage LP blading, as many designs require that cover bands, tie-wires, and erosion shields be attached by thermal joining.

D. Stresses in Steam Turbine Blade

There is a myriad of static and dynamic stresses and loads on turbine blades, particularly the longer blades of the steam turbine. Centrifugal Stresses, Centrifugal loads, caused by rotation, are the primary source of stress on blades. In contrast, as a rule of thumb, in typical turbine blades (un-shrouded), the steady stresses will be roughly $0.5S_y$ over about half the blade airfoil length, and in excess of $0.25S_y$ over about 80% of that length, where S_y is the yield strength of the material. Centrifugal stresses are generally proportional to the square of the speed. As a result, during over speed tests, the centrifugal stresses on a blade can increase to as high as 75%of the material's yield strength.
Centrifugal stresses can also have a dramatic effect in those locations where there are stress concentrations such as in the root attachment and at tie-wire holes.

E. Blade Root Geometry and Load Transfer

The blade forces, centrifugal forces (radial) and into axial and circumferential bending forces. For the consideration of stiffness effects of both the rotor (drum or disc) and the blade root, the reduction of the bending force into its axial and circumferential component is required.

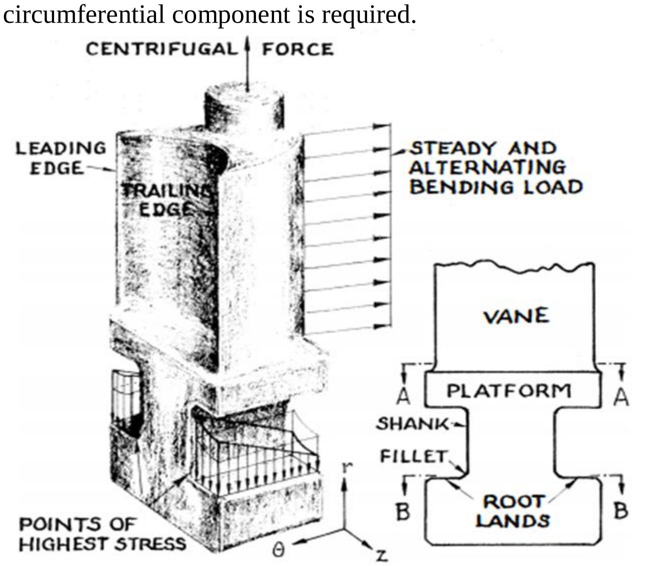

Fig.4. Load Distribution on Root Lands of an Impulse Blade.

Starting with the pure bending moment, which is caused by the axial component of the blade force on the root, one finds that this moment is transferred right into the rotor, if the shank portion is tightly held by the rotor or disc.

F. Stress Distributions in Blade Roots

Extensive studies of stress distribution in root cross sections have been performed by manufacturers and research groups. The photo elastic analysis technique is one of the common tools for obtaining information of stress distributions and stress concentrations. Figures 5 show the stress pattern in a T-root and more detailed in the vicinity of the fillet. One should note the much localized stress concentration just above the root land in the fillet and shank portion. Most photo elastic analyses concentrate on two-di-mensional configurations, loaded by a radial force only. The results reported in correspond to such tests. Photo elastic measurements for the moment resulting from the axial blade force , similar to the case shown in Figure 4, could also be easily obtained. If considering the moment caused by the circumferential blade force on a simple T-root, the problem becomes three dimensional. Since the stress concentrations occur for this configuration in rather small regions, results are more difficult to produce. For more complicated shapes, the problem becomes even less translucent.

Fig.5. Stress concentration in fillet of T-root.

Compared with other networks, ZigBee has the following advantages: low power, low cost, short time delay, network large capacity, reliability and safety.

III. CALCULATIONS & ANALYSIS OF STEAM TURBINE BLADE

A. Material Selection

As we discussed in literature survey under subtitle "Required material properties for steam turbine blades" blade material should have a Creep Strength, Creep-fatigue Resistance, Notch sensitivity and damping property. X22CrMoV121 is one such a material which has all these properties. And it is the one most commonly used material for HP blades in steam turbine.

B. Centrifugal force calculation

Centrifugal force is directed outwards, away from centre of curvature of the path. A simplified 2D figure of the blades under discussion is shown in figure 6. The general equation for centrifugal force is

$$F = mr\omega^2 \qquad (1)$$

Where m is the mass of the moving object, r is the distance of the object from the centre of rotation (the radius of curvature) and ω is the angular velocity of the object.

Consider a small segment of mass δ_m, of having width δ_r at a distance r from the centre. Then the equation for the centripetal force δ_F on this small segment is given by:

$$\delta_F = \delta m r\omega^2 \qquad (2)$$

The blades have a cross sectional area A (mm^2) and material density ρ (kg/mm^3). Then we can write the mass of the element

$$\delta_m = \rho_A \delta_r$$

Equation (2) can be write as

$$\delta_F = (\rho_A \delta_r) r\omega^2$$

Or formally it can be writing as

$$d_F = \rho_A \omega^2 r dr \qquad (3)$$

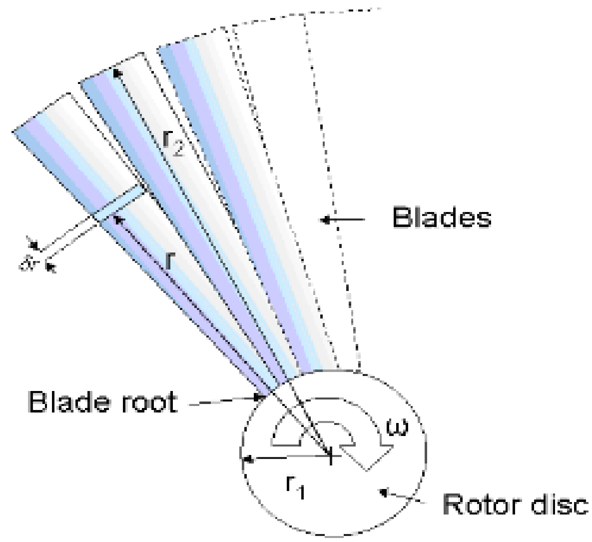

Fig.6. Simplified blade dimensions.

Then, integrating equation (3) along the total length of the blade, the total centrifugal force acting on the blade is given by

$$F = \rho A \omega^2 \int_{r1}^{r2} r dr$$

$$F = \rho A \omega^2 \int_{r1}^{r2} r dr \qquad (4)$$

The following data is considered for design and centrifugal force estimation

Blade speed	N	= 8000 rpm
Blade cross-sectional area	A	= 165.161 mm2
Material density	ρ	= 7850x10-6 kg/mm3
Blade tip radius	r_2	= 267.5 mm
Blade root radius	r_1	= 220.5 mm
Blade length	r_2-r_1	= 47mm

Substituting the all above values in equation (4)

Centrifugal Force F = 10,436.2N

C. Fatigue life calculation

There are many methods to calculate the Fatigue life. Based on the available data, accuracy and ease Smith, Watson and Topper (SWT) Mean Stress Correction for Strain Life method used for the present work. SWT equation for Fatigue Analysis is given below

$$\sigma_{Maximum} \frac{\Delta\varepsilon}{2} = \frac{(\sigma'_{failure})^2}{E}(2N_{failure})^{2b} + \sigma'_{failure}\varepsilon'_{failure}(2N_{failure})^{b+c}$$

Where, σ_{max} = Maximum stress

Δε/2 = Total strain applitude

$\sigma'_{failure}$= Fatigue Strength Coefficient or Effective strength

$\varepsilon'_{failure}$ = Fatigue ductility coefficient
E = Modulus of Elasticity
$N_{failure}$= Number of reversals
B = Fatigue strength exponent
c = Fatigue ductility exponent

D. FE Analysis of Steam Turbine Blade

The intention of the FE analysis was to determine the stress and Fatigue life of the components at the critical location of the blade. It has been found out that the critical location of the blade is situated at the T root of the blade. More precisely, at the convex side of the neck. However, it is required to assess the fatigue life of the whole blade-disc connection. For the purpose of simple solving and solution time in ANSYS the 3D model of blade is simplified by removing the tenon on the blade.

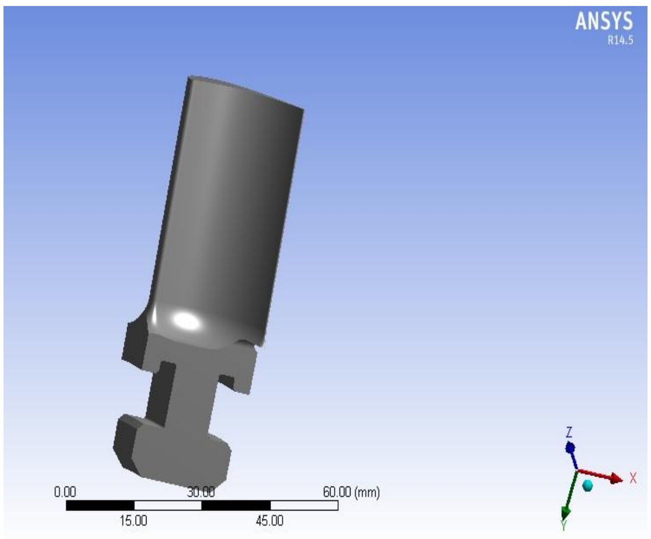

Fig.7. Imported Blade model.

As discussed in literature survey centrifugal force is the major force acting on blades. When compared magnitudes of all other forces acting on blade with centrifugal force magnitude they can be negligible. So in this analysis centrifugal force only considered as load of application.

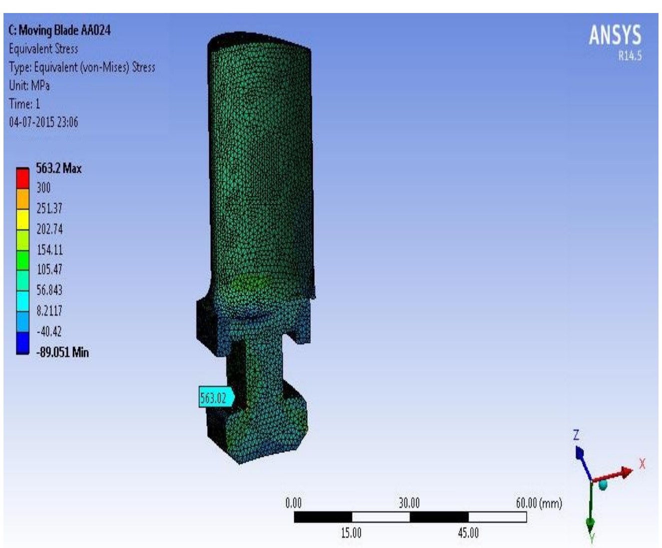

Fig.8. Equivalent stress of the steam turbine blade.

Equivalent von mises stresses observed (563 MPa) on the fillet region of the blade as depicted in the figure 8.

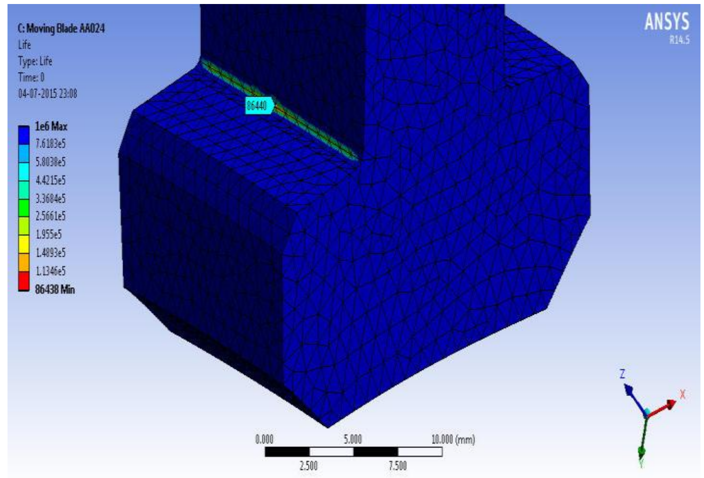

Fig.9. Fatigue life of the steam turbine blade

Minimum Fatigue life observed at the fillet region is 86438 as depicted in the figure 9.

IV. RESULTS & DISCUSSIONS

Existing blade design is good enough for fatigue life in theoretical calculation. But there is a problem in finite element analysis. In theoretical calculation blade model is getting infinite life (2.438e6). By running ANSYS software existing blade design is getting only 86436 is number of cycles as fatigue life.

A. Modifications suggested in design of steam turbine blade

Here failure of blade mostly occurs in T root. So it requires some modifications to get the infinite life ($1e^6$). By doing some trial and error methods in changing the dimensions of T root. Finial a modification is suggested to turbine designer as below.

1. Neck width of the blade is increased by 1mm. i.e. Neck width is modified to 11mm from 10mm
2. Fillet radius of the root is modified to 0.8 mm from 0.5mm
3. Chamfer dimensions of the tang (bottom part of the root) is changed to1x45° and 2.77x45° from 1.25x45° and 3x45° respectively.

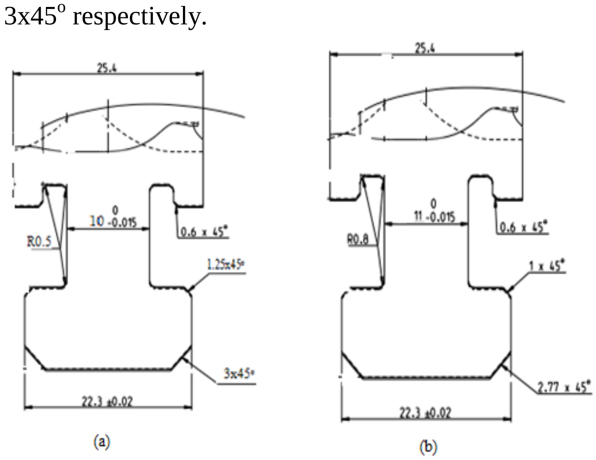

Fig.10 . (a) Existing design (b) Modified design.

After making modifications in blade root design. Same loads and constrains are applied to check the strength of the blade. Then the results are as follows

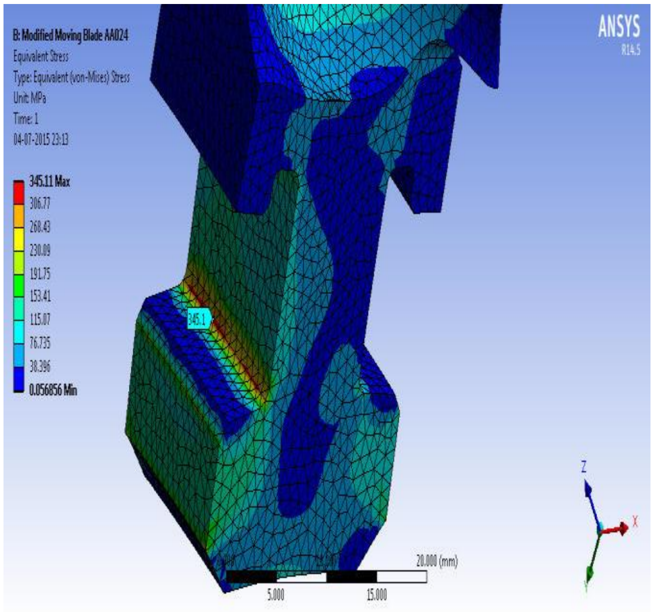

Fig.11. Equivalent stress of modified steam turbine blade

With the implementation of design modification, the Equivalent von mises stresses observed (345 MPa) on the fillet region of the blade as depicted in the fig above. The stress levels reduced from 563MPa to 345 MPa which will helps in improving the fatigue life.

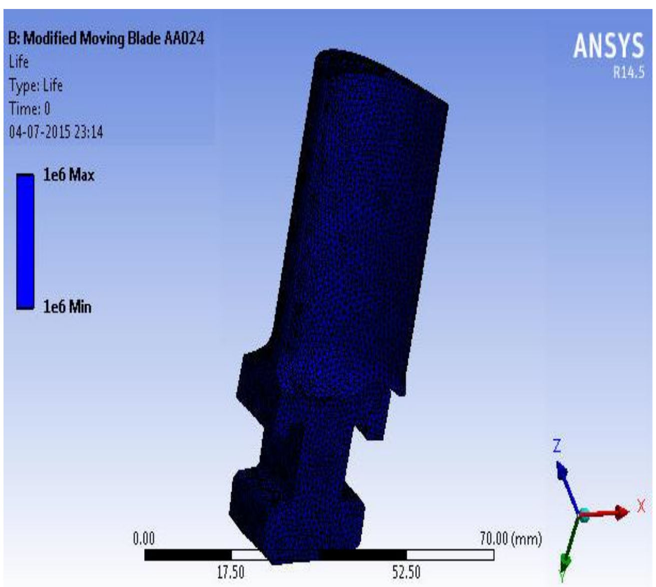

Fig.12. Fatigue life of modified steam turbine blade.

Table1. Comparison of results

S.No.	Parameter	Existing Blade	Modified Blade
1	Equivalent stress (Mpa)	560	345.1
2	Fatigue life “N”	86438	1000000

V. CONCLUSION

This project has attempted to investigate the fatigue response of the steam turbine blade in terms of high cycle fatigue. Existing blade design is good enough for fatigue life in theoretical calculation. But there is a problem in finite element analysis. In theoretical calculation blade model is getting infinite life (2.438e6). But during run of ANSYS software existing blade design is getting only 86436 is number of cycles as fatigue life. Some modifications are suggested to steam turbine blade designer which is able to achieve the life of 1e6 cycles as fatigue life.

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