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Material Selection for High Pressure (HP) Compressor Blade of an Aircraft Engine

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Abstract

Trade-offs are usually allowed during the material selection process of a given component, but the designer must have proper understanding of a component's loading conditions, to be able to select materials with suitable properties that can eliminate or minimise failure during the duty cycle of such component. High strength-to-weight ratio is one of the most important requirements in the aerospace industries in terms of components performance. High Pressure (HP) compressor blades with a wide range of other parts in an aircraft engine require these properties for optimum performance. This is due to the in service condition (such as operation principles and the high temperature (450-600°C) environment of the HP compressor blade which subjects the component to Radial loads (caused by centrifugal forces acting on the blade) Bending Loads, Thermal Loads etc. thereby, inducing stresses, cracks and fractures (caused by compressor surging), fatigues and other material defects that can result in failure or limit the longevity of the blade. In this study, detailed material indices were derived for price, specific stiffness, yield strength, including fracture toughness, and a search was performed using CES software to determine the materials having the desired material indices. Since the material should be able to withstand the operating temperature of the HP compressor, a limiting factor of 500°C was set to filter off materials whose service temperature is below this value. Four materials passed the search including Low alloy steel, Nickel-based super alloys, Stainless steel and Titanium alloys. However, titanium alloys were found to meet the high strength to weight ratio at 500°C despite its high cost, while steel materials in this category had relatively high densities and some could not match up the temperature requirement. Above 500°C, more dense Nickel based super alloys were found to be preferable. Hence, Nickel based super alloys are suitable for applications requiring high performance at elevated temperature, but the high density might limit its suitability during material selection.

Keywords

Compressor Blade, Material, Temperature, High Strength, Low Density, Loading Condition

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1. Introduction

The application of Gas turbine cuts across many industries, some of which includes power generation, oil and gas and Aerospace industries. The Aircraft gas turbine engine is an internal combustion engines which consist of four main parts: the compressor, the combustor, the turbine and the exhaust nozzle [6]. With the help of a dovetail (shape) component known as the blade, the compressor inducts and compresses ambient air into the gas turbine engine. Depending on the engine capacity, the ratio of inlet to outlet pressure at the end of each compression cycle can fall between 30:1 and 40:1 [10]. As shown in Figure 1, The compressed air intake flows through multiple stages of rotating (N_1) and stationary (N_2) blades which further increases the air pressure, reduce the air volume and forces it into the combustion chamber. In other words, the High pressure compressor blades compresses, pressurises and delivers the air to combustion chamber [12, 13]. In addition, the combustion chamber of a gas turbine

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engine is annular, with an exit ring installed at the back to control exhaust gas. The gases exiting the combustion chamber are released at a temperature of about 1700°C, while the shaft rotates at a speed in excess of 12,000rpm [9].

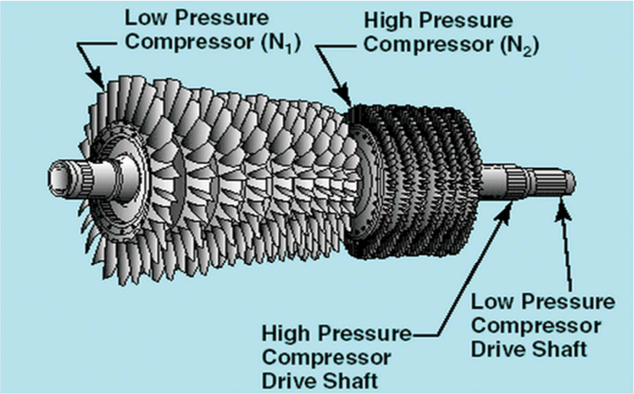

Figure 1. Typical Compressor Blade of an Aircraft gas turbine engine [13].

The air entering the Aircraft gas turbine air inlet from fan to low pressure compressor operates with temperature of -50 to 40°C and flows to the 8th stage intermediate pressure compressor with increasing temperature of 50 to 300°C. When the air intake flows to the 6th stage high pressure compressor, the temperature increases from 300 to 650°C, and by the time it enters the combustion chamber, the air increases to 2000°C respectively. The air fuel-mixture exiting the combustion chamber is used to power the turbine as it enters through the 5th stage low pressure turbine at a temperature of 900°C, 1st stage intermediate pressure turbine at a temperature of 1200°C and the 1st stage high pressure turbine at a temperature of 1500°C [13].

The compressed air enters the combustion chamber where it is mixed with fuel, ignites and burns. The burning process results in a high temperature and high pressure gas which is expanded through the turbine blades. Part of the energy of the gas is used in turning the turbine shaft. The compressor is mounted on the shaft and rotates along with it, thus the whole process is sustained. The hot gases leave the engine through the exhaust nozzle with enough energy to overcome the drag forces and hence move the aircraft forward [11]. For effective performance, attention should be given to aerodynamic features of compressor blades such as the shape, as properly streamlined blade can play a significant role in static and dynamic stress resistance when the aircraft is in operation. However, Compressor surge may occur when there is a momentary reversal and flow back of compressed air towards the intake of the compressor. This happens when the mass of air flowing into the compressor drops below certain critical level and results in a high pressure difference across the compressor. Consequently, enough air might not be available to immediately replace those being push forward; the already compressed air reverses its direction and flows backwards. The air flowing backward provides more mass of air to the suction side and the compressor pushes it forward again. This forward and backward flow can result in rapid fluctuation in the pressure of the system. The fluctuations can set up excessive vibrations on the blades which can damage them [6, 8].

Atmospheric and environmental conditions can also affect the performance of the HP compressor. At high altitude and on hot days, the density of air is decreased and as a result, the mass of air entering the compressor decreases. This often results in lower output power of the turbine [5, 15]. Sand and dust particles within the operating environment (operation in desert environment for example) can be ingested into the compressor which may result in erosion of the blades [5]. In addition, when the ambient air contains corrosive substances, corrosion and pitting of the blade can occur, from which cracks can develop [8].

The centrifugal forces acting on the high pressure compressor blade during operation subjects the blade to high temperature and stresses in adverse environmental condition, which induces various loading effects and material defects on the blade material, thereby, limiting the longevity of the turbine blade in the long run [6, 7].

This service temperature over time can result in creep [8].

2. Loading Conditions of High Pressure Compressor Blade

The blade is subjected to the following cyclic loading modes:

2.1. Radial Centrifugal Loading

This loading mode is as a result of rotation of the blade under high speed and acts as to pull the blade away from its root. This force increases as the speed of rotation increases and is given by;

$$F = mr\omega^2 \tag{1}$$

Where F is the centrifugal force;

m is the mass of the blade;

r is the radius of rotation; an

ω is the angular velocity.

The cyclic tension and compression as the speed of rotation of the compressor increase or decrease can result in fatigue failure of the blade [4]

2.2. Bending Load

The blade acts as a cantilever since one end of the blade is fixed on the hub, whereas, the other end is unrestrained. The air being compressed subjects the blade to bending load [3, 4]. Assuming the blade to be flat beams, the bending stresses on the beam is given by

$$\sigma_y = \frac{My}{I} \tag{2}$$

Where σ_y is the yield strength of the material

M is the bending moment,

y is the distance from the neutral axis and

I is the second moment of area.

This bending force subject the blade to bending which can distort its shape and geometry.

2.3. Thermal Loads

This arises due to uneven expansion of sections of the blades especially during start-up and shut down [8]. This uneven expansion can initiate crack or can be superimposed on those caused by other load types to initiate crack [2].

The thermal stress is given by

$$\sigma_T = E\alpha\,(T_2 - T_1) \tag{3}$$

Where

σ_T is the stresses due to uneven expansion

E is the Young's Modulus of the material

α is the coefficient of thermal expansion

$(T_2 - T_1)$ is the temperature gradient.

2.4. Residual Stress

The residual stress is induces due to the vibration of the blade as a result of other loads modes and service conditions. If these vibrations are close to the natural frequency of the material, resonance may occur. Such high vibration frequency can initiate crack [4].

2.5. The Material Characteristics Needed for the High Pressure Compressor Blades

To withstand the aforementioned loading modes and operating conditions, the material must:

1. Be light weight so as to minimize both the power needed to drive the compressor the total weight of the engine, hence should have low density ρ.
2. Be resistant to crack and hence, should have high fracture toughness K1c.
3. Have low Coefficient of thermal expansion in order to keep strain and thermal stresses as low as possible.
4. Be able to withstand the bending loads and hence should have high stiffness value (high value of Young's Modulus).
5. Be able to withstand many cycles of fluctuating loads and hence should have high fatigue strength.
6. Be able to withstand the axial tensile stresses due to centrifugal forces acting on the blade and hence should have high yield strength.
7. The material should have high natural frequency so as not to be excited into resonance.
8. The material should also be resistant to erosion and corrosion.

2.6. Determination of Material Performance Indices

The following assumption has been made in determining the performance indices:

1. The blade is a cantilever fixed at one end and uniformly loaded as shown in Figure 2.
2. The cross sectional area of the blade is a free variable with a constant aspect ratio α

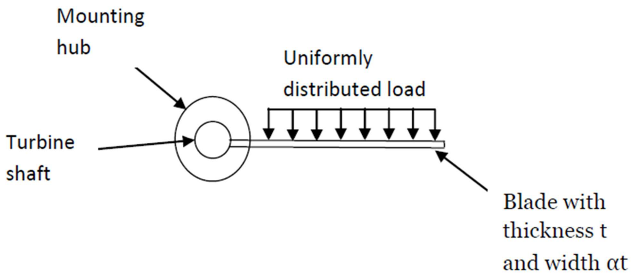

Figure 2. HP compressor blade as a cantilever with uniform loading.

2.6.1. Material Index to Resist Bending

From the generic beam bending equation

$$\frac{M}{I} = \frac{\sigma}{y} \tag{4}$$

Where:

σ is the stress at distance y from neutral axis of beam;

M is the bending moment of the blade;

y is the distance from the neutral axis; and

I is the second moment of area.

For a uniformly distributed load on a cantilever,

$$M = \frac{wl^2}{2} \tag{5}$$

Where

w is the uniformly distributed load in N/m;

l is the length of the blade in meter.

$$I = \frac{\alpha t^4}{12} \tag{6}$$

Where

t is the blade thickness;

α is the aspect ratio.

For stress (σ) = σ_{max}

$$y = \frac{t}{2} \tag{7}$$

From equation 4

$$\frac{I}{y} = \frac{\left[\frac{\alpha t^4}{12}\right]}{\left[\frac{t}{2}\right]} = \frac{\alpha t^3}{6} \tag{8}$$

Cross sectional Area,

$$A = \alpha t^2 \tag{9}$$

Substituting the value of t from equation 9 into 8 gives

$$\frac{I}{y} = \frac{\alpha}{6}\left(\sqrt{\frac{A}{\alpha}}\right)^3 = \frac{A^{\frac{3}{2}}}{6\sqrt{\alpha}} \tag{10}$$

From equation 4,

$$\frac{M}{\sigma_{max}} = \frac{wl^2}{2\sigma_{max}} = \frac{A^{\frac{3}{2}}}{6\sqrt{\alpha}} \tag{11}$$

$$A = \left(\frac{3\alpha^{\frac{1}{2}}wl^2}{\sigma_{max}}\right)^{\frac{2}{3}} \tag{12}$$

Mass of the blade, m_1 is given by

$$m_1 = Al\rho \tag{13}$$

Where

A is the cross-sectional area, and

ρ is the material density

Substituting equation 12 into 13,

$$m_1 = \left(\frac{3\alpha^{\frac{1}{2}}wl^2}{\sigma_{max}}\right)^{\frac{2}{3}} l\rho \tag{14}$$

For $\sigma_{max} = \sigma_y$ (yield strength)

$$m_1 = \left(3\alpha^{\frac{1}{2}}wl^2\right)^{\frac{2}{3}} l\frac{\rho}{(\sigma_y)^{\frac{2}{3}}} \tag{15}$$

Therefore, to minimize mass, the material index $\frac{(\sigma_y)^{\frac{2}{3}}}{\rho}$ should be maximized.

2.6.2 Material Index to Resist Fatigue

It is desired that the Fatigue strength endurance limit σ_e be as high as possible. Hence,

$$\frac{wl}{A} \leq \sigma_e \tag{16}$$

Where wl is the total load on the blade.

$$A \geq \frac{wl}{\sigma_e} \tag{17}$$

Mass, $m_2 = Al\rho$

$$m_2 = wl^2\frac{\rho}{\sigma_e} \tag{18}$$

In order to optimize performance indices $m_2 = m_1$

To minimize mass, $\frac{\sigma_e}{\rho}$ should be maximized

2.6.3. Material Index to Maximize Fracture Toughness, K1c

Assuming that the blade follows the equation of a centre cracked plate with a very large width [1].

$$\mathrm{K}_{1c} = \sigma(\pi c)^{0.5} \tag{19}$$

Where

K1c is the fracture toughness

σ is the applied stress and

c is a very small crack.

$$\mathrm{K}_{1c} = \frac{wl}{A}(\pi c)^{0.5} \tag{20}$$

$$A = \frac{wl}{\mathrm{K}_{1c}}(\pi c)^{0.5} \tag{21}$$

$$m_3 = Al\rho$$

$$m_3 = \frac{wl}{\mathrm{K}_{1c}}(\pi c)^{0.5} l\rho \tag{22}$$

$$m_3 = wl^2(\pi c)^{0.5}\left(\frac{\rho}{\mathrm{K}_{1c}}\right) \tag{23}$$

In order to optimize performance indices $m_3 = m_1$

To minimize mass, $\frac{\mathrm{K}_{1c}}{\rho}$ should be maximized.

2.6.4. Material Index to Maximize Specific Stiffness

The stiffness of a beam is given by [1],

$$S = \frac{F}{\Delta} \tag{24}$$

Where

S is the stiffness of the material

F is the load applied

Δ is the deflection.

In this case,

$$S = \frac{wl}{\left(\frac{wl^3}{8EI}\right)} = \frac{8EI}{l^2} \tag{25}$$

Where E is the Young's Modulus of the material

Substituting equation 6 into 25

$$S = \frac{8E\alpha t^4}{12l^2} \tag{26}$$

$$t = \left(\frac{12Sl^2}{8E\alpha}\right)^{\frac{1}{4}} \tag{27}$$

$$m_4 = Al\rho = \alpha t^2 l\rho \tag{28}$$

Substituting equation 27 into 28,

$$m_4 = \alpha\left(\frac{12Sl^2}{8E\alpha}\right)^{\frac{1}{2}} l\rho \tag{29}$$

$$m_4 = \sqrt{\alpha}\left(\frac{12S}{8}\right)^{\frac{1}{2}} l\left(\frac{\rho}{E^{\frac{1}{2}}}\right) \qquad (30)$$

In order to optimize performance indices $m_4 = m_1$

Therefore to minimize mass, $\frac{E^{\frac{1}{2}}}{\rho}$ should be maximized.

2.6.5. Material Index to Maximize Natural Frequency

The natural frequency of a body is given by [1]

$$f = \frac{1}{2\pi}\sqrt{\left(\frac{K}{m}\right)} \qquad (31)$$

Where K is the stiffness constant and is given by

$$K = \frac{AE}{l} \qquad (32)$$

$$m = Al\rho \qquad (33)$$

Substituting equation 32 and 33 into 31 yields

$$f = \frac{1}{2\pi}\sqrt{\left(\frac{AE}{Al^2\rho}\right)} \qquad (34)$$

$$f = \frac{1}{2\pi}\, l\sqrt{\left(\frac{E}{\rho}\right)} \qquad (35)$$

To maximize natural frequency, $\left(\frac{E}{\rho}\right)^{0.5}$ should be maximized.

The material indices to be maximized are

$$\frac{E^{\frac{1}{2}}}{\rho}, \left(\frac{K_{1c}}{\rho}\right), \frac{\sigma_e}{\rho}, \frac{(\sigma_y)^{\frac{2}{3}}}{\rho}$$

3. Methodology

Considering the aforementioned materials indices to be maximised, a search was performed using the CES software to determine the materials having the desired material indices. Since the material should be able to withstand the operating temperature of the HP compressor, a limiting factor of 500°C was set to filter off materials whose service temperature is below this value and is presented as follows;

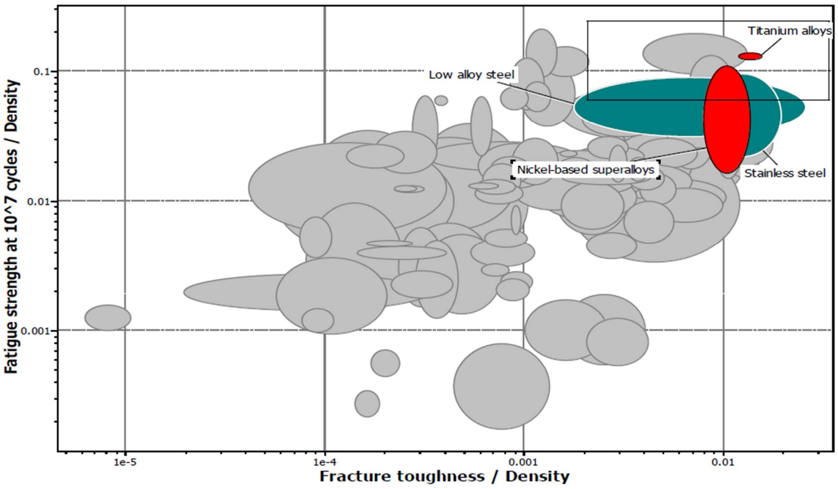

Figure 3. Graph of fatigue strength $\frac{\sigma_e}{\rho}$ against fracture toughness $\left(\frac{K_{1c}}{\rho}\right)$.

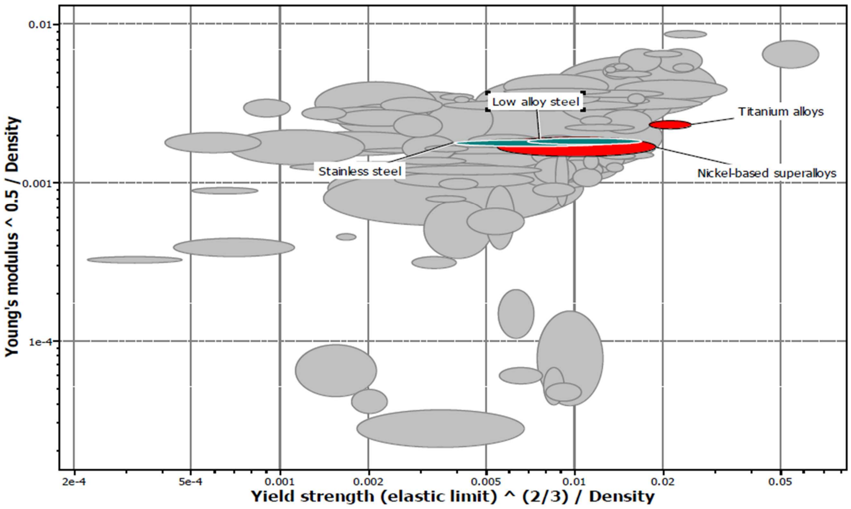

Figure 4. Graph of specific Stiffness $\frac{E^{\frac{1}{2}}}{\rho}$ against yield strength $\frac{(\sigma_y)^{\frac{2}{3}}}{\rho}$.

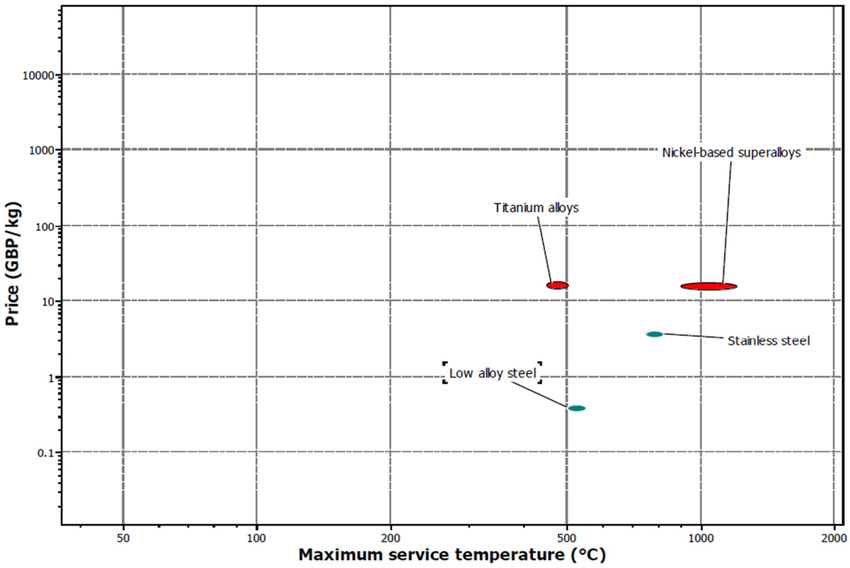

Figure 5. Graph of price against Maximum service temperature of 500°C.

4. Discussion

Having used CES Edupack software 2014 version to search for materials with the following material indices $\frac{E^{\frac{1}{2}}}{\rho}$, $\left(\frac{\mathrm{K}_{1c}}{\rho}\right)$, $\frac{\sigma_e}{\rho}$, $\frac{(\sigma_y)^{\frac{2}{3}}}{\rho}$, the first result for the search as presented in Figure 3 showed some material for high pressure compressor blade that can exhibit good fatigue strength $\frac{\sigma_e}{\rho}$ and fracture toughness $\left(\frac{\mathrm{K}_{1c}}{\rho}\right)$ when exposed to maximum temperature of 500°C. Four materials passed the search including Low alloy steel, Nickel-based super alloys, Stainless steel and Titanium alloys. Also, result for the second search as presented in Figure 4 showed some material needed for specific Stiffness $\frac{E^{\frac{1}{2}}}{\rho}$ and yield strength $\frac{(\sigma_y)^{\frac{2}{3}}}{\rho}$ in high pressure compressor blade when exposed to maximum temperature of 500°C and Low alloy steel, Nickel-based super alloys, Stainless steel and Titanium alloys equally passed the test. Considering the density of these four material, titanium alloy weighs lighter, followed by low alloy steel, stainless steel and Nickel-based super alloys, as Nickel-based super alloys turns out to be the heaviest of all the four potential materials that can withstand in-service temperature of 500°C and above when used in manufacturing High Pressure (HP) compressor blades for aircraft gas turbine engines. The exceptional properties of titanium alloy with excellent corrosion resistance property is one of such, is one of the reasons why it is mostly used in manufacturing HP compressor blades despite its high cost [7]. However, for HP compressors with operating temperature exceeding 5000C, the blades are usually made of Nickel based super alloys. However, for HP compressors with operating temperature exceeding 500°C, the blades are usually made of Nickel based super alloys. For example, in the Trent series where the temperature of the HP compressor gets to about 700°C, Rolls Royce prefers Nickel based super alloys for the HP compressor blades, as the yield strength of steel and even titanium drops off when 40-50% their melting point is approach, whereas Nickel alloys restrain their yield strength up to a melting point of over 85% [14]. Although steel and its alloys may not have very good high temperature (can withstand up to 450-550°C) resistance, their strength and surface hardness are one of the outstanding properties of the steel family. A disadvantage of this is that the increased density of Nickel would require more power to drive the compressor [7]. Nickel based super alloy are chosen over stainless steel and low alloy steel because of their higher service temperature. The third results for the search to determine the price of materials that can show resilient when the high pressure compressor blade is exposed to maximum service temperature of 500°C as presented in Figure 5, showed the same materials (Low alloy steel, Nickel-based super alloys, Stainless steel and Titanium alloys) obtained from previous searches. The result shows Titanium alloy and Nickel-based super alloys to be more expensive compared to steel in the steel family. Since high performance is greatly desired in aero-engines, Titanium alloys and Nickel based super alloys are usually chosen despite their relative higher cost.

5. Conclusion

The report aimed at selecting a suitable material for high pressure compressor blade of an aero-engine. The operating conditions and the loading modes of the blade were considered after which materials search was done using the CES software. Titanium alloys gave the best performance with operating temperature of 500°C. Above this temperature, Nickel-based Super alloys were found to be denser, but resisting the increasing temperature, whereas, apart from the high density of steel materials, their strength slightly diminishes when the operating temperature approaches 500°C and above. Most materials such as Titanium alloy might be more expensive than some of its contemporary alloys, but manufacturers should not compromise, as failure of components due to inferior materials or poor material selection might result in more catastrophes and loss of lives than the cost of acquiring suitable materials for optimum performance and safety.

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