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# Analytical Comparison of a Gas Turbine Blade Cooling Using Wet and Dry Air

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## *Abstract*
*Air cooling is widely used technique to shield the turbine aerofoils against hot flue gases. The cooling of a gas turbine blade using wet air and dry air as a coolant is analytically investigated. The investigation is carried out considering effect of rotation for inward and outward flow of coolant. Wet air cooling performance is compared with dry air cooling. It has been observed that wet air provides better cooling and the performance improves with increase in relative humidity. The temperature of blade at tip decreases from 1293.44 K to 1172.6 K when relative humidity of wet air is increased from 10% to 90%.*

***Keywords:*** *Air and wet air cooling, gas turbine, blade, rotation, outward flow, inward flow*

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## INTRODUCTION
Gas turbine converts the fuel energy in to mechanical energy. During its operation, the blade temperature may reach up to 1400 K, which may be above the melting point of blade metal; hence, it is essential to cool the blades effectively. Mostly air is used as coolant for blades for the cooling of blades.

To achieve effective cooling, various techniques have evolved recently. Cooling air, of around 800 K is from the compressor can be used as a cooling media and the temperature of the blades may be lowered to 1273 K for safer operation [1].

Albeirutty et al. [2] proposed a general model of the combined system to compare the performance of the blade cooling by air, open-loop steam and closed-loop steam. Studies on the rotating blades suggested that the rotation is an important parameter and need to be analytically investigated.

Apart from internal cooling of turbine blade, there are different techniques of providing cooling to the gas turbine blade. The conventional gas turbine blade analysis by Cohen et al. [3] does not consider the effect of rotation.

This paper presents the analysis of gas turbine blades using dry air and wet air considering rotation. The rotation introduces centrifugal force on the coolant, which may increase or decrease the temperature of coolant depending on the coolant flow direction.

## THEORETICAL ANALYSIS
A MATLAB code is made to solve differential equation implementing forward difference scheme to determine the blade temperature at various points across the length. The coolant is passing span-wise from root to tip of the blade (outward flow) or from tip to root (inward flow).

The geometry particulars and operating parameters of gas turbine blade [4] are as under Table 1.

In this analysis, the equations are written in terms of relative total temperature. The relative total temperature ($T_{oc,rel}$ and $T_{og,rel}$) is the temperature at a point when the flow is adiabatically brought to rest, with respect to non-inertial reference frame.

  

**Table 1:** *Geometry Particulars and Operating parameters of Turbine Blade.*

| | |
|---|---|
| Inlet gas angle $\alpha_1$ ( ° ) | 59.89 |
| Exit gas angle $\alpha_2$ ( ° ) | 72.38 |
| Blade chord $c$ (m) | 0.07816 |
| Total external perimeter of blade $S_g$ (m) | 0.3203 |
| Total internal wetted perimeter $S_c$ (m) | 0.1643 |
| Number of cooling passages $N_p$ | 10 |
| Blade span (m) | 0.0762 |
| Total wetted cross section area $Ac$ (m$^2$) | 2.36381E-4 |
| Blade pitch $s$ (m) | 0.11773 |
| Range of rotation $\omega$ (rad/s) | 0 to 1832 |
| Range of flow ratio $\Phi$ | 0 to 0.03 |
| Inlet relative total temperature of coolant $T_{oc, rel\ (i)}$ (K) | 800 |
| Distance of axis of blade from root $r_{root}$ (m) | 0.3 |
| Average diameter of coolant channel $D$ (m) | 0.005228 |

The external heat transfer to the coolant is given by Newton's law of cooling.

$$h_g S_g (T_{og,rel} - T_b)dr = h_c S_c (T_b - T_{oc,rel})dr \tag{1}$$

Here the suffix 'g', 'c' and 'b' represents respective parameters of gas, coolant and blade. Equation (1) can be rewritten as,

$$h_g S_g T_{og,rel} + h_c S_c T_{oc,rel} = (h_c S_c + h_g S_g) T_b \tag{2}$$

The blade temperature can be determined using equation (2),

$$T_b = \frac{h_g S_g T_{og,rel} + h_c S_c T_{oc,rel}}{(h_c S_c + h_g S_g)} \tag{3}$$

For an elemental length of the turbine blade, heat transfer to the coolant is given as,

$$m_c c_{pc} dT_{oc,rel} = h_c S_c (T_b - T_{oc,rel})dr \tag{4}$$

Substituting value of 'Tb' in equation (4),

$$m_c c_{pc} dT_{oc,rel} = h_c S_c \left( \frac{h_g S_g T_{og,rel} + h_c S_c T_{oc,rel}}{(h_c S_c + h_g S_g)} - T_{oc,rel} \right) dr \tag{5}$$

Rearranging terms,

$$\frac{dT_{oc,rel}}{dr} = \frac{h_c S_c}{m_c c_{pc}} \left( \frac{h_g S_g T_{og,rel} + h_c S_c T_{oc,rel}}{(h_c S_c + h_g S_g)} - T_{oc,rel} \right) \tag{6}$$

Multiplying by '$c_{pc}$',

$$c_{pc} \frac{dT_{oc,rel}}{dr} = \frac{h_c S_c}{m_c} \left( \frac{h_g S_g T_{og,rel} + h_c S_c T_{oc,rel}}{(h_c S_c + h_g S_g)} - T_{oc,rel} \right) \tag{7}$$

Rearranging the terms in equation (7), the heat flux per unit length can be written as,

$$\frac{dq}{dr} = \frac{h_g S_g h_c S_c (T_{og,rel} + T_{oc,rel})}{m_c (h_c S_c + h_g S_g)} \tag{8}$$

**Blade Temperature Distribution**

It is important to determine blade temperature at various points along the blade span. This helps in determining coolant mass flow rate to confine the blade temperature. Taking a control volume around the cooling channel in a non-inertial frame of reference which is rotating at the same speed as that of rotor, applying the steady flow energy equation in differential form with respect to 'r' which varies from 0 to L to get span-wise variation.

$$\frac{dq}{dr} = \frac{dh}{dr} + \frac{d(KE)}{dr} + \frac{d(PE)}{dr} + \frac{dW}{dr} \tag{9}$$

Since the rotor blades are stationary with respect to non-inertial frame, the shaft work output,

$$\frac{dW}{dr} = 0$$

In equation (9), the kinetic energy term can be written in terms of relative velocity of coolant '$v$',

$$\frac{d(KE)}{dr} = \frac{d\left(\frac{v^2}{2}\right)}{dr} \tag{10}$$

In the non-inertial frame of reference and flow of coolant from blade root to tip (outwards), along with gravitational potential, the centrifugal potential needs to be taken into account,

$$\frac{d(PE)}{dr} = \frac{d\left(gr - \frac{w^2\{r_{root} + r\}^2}{2}\right)}{dr} \tag{11}$$

Here, '$gr$' and '$\frac{w^2\{r_{root} + r\}^2}{2}$' are gravitational and centrifugal potential respectively.

By substituting equations (10) and (11) in equation (9), neglecting the gravitational potential,

$$\frac{dq}{dr} = \frac{dh}{dr} + \frac{d\left(\frac{v^2}{2}\right)}{dr} - \frac{d\left(\frac{w^2\{r_{root} + r\}^2}{2}\right)}{dr} \tag{12}$$

Introducing the term relative total enthalpy $h_{oc,rel} = h + \frac{v^2}{2}$, equation (12) is written as,

$$\frac{dq}{dr} = \frac{dh_{oc,rel}}{dr} - \frac{d\left(\frac{w^2\{r_{root} + r\}^2}{2}\right)}{dr} \tag{13}$$

$$\frac{h_g S_g h_c S_c (T_{og,rel} + T_{oc,rel})}{m_c (h_c S_c + h_g S_g)} = \frac{dh_{oc,rel}}{dr} - \frac{d\left(\frac{w^2\{r_{root} + r\}^2}{2}\right)}{dr} \tag{14}$$

It is assumed that the coolant is behaving like a perfect gas, hence $h_{oc,rel} = c_{pc} T_{oc,rel}$

Here, '$T_{oc,rel}$' is the total temperature observed by the observer attached to the non-inertial frame of reference that is rotating with the same speed as that of the blade. Putting value of '$h_{oc,rel}$', equation (14) can be rewritten as,

$$\frac{h_g S_g h_c S_c (T_{og,rel} + T_{oc,rel})}{m_c (h_c S_c + h_g S_g)} = c_{pc} \frac{dT_{oc,rel}}{dr} - \frac{d\left(\frac{w^2 \{r_{root} + r\}^2}{2}\right)}{dr} \tag{15}$$

$$\frac{dT_{oc,rel}}{dr} - \frac{w^2}{c_{pc}} \{r_{root} + r\} - \frac{h_g S_g h_c S_c}{m_c (h_c S_c + h_g S_g)} (T_{og,rel} + T_{oc,rel}) = 0 \tag{16}$$

Equation (2) can be written in terms of '$T_{oc,rel}$' as,

$$T_{oc,rel} = T_b \left(\frac{h_c S_c + h_g S_g}{h_c S_c}\right) - \frac{h_g S_g T_{og,rel}}{h_c S_c} \tag{17}$$

Differentiating the equation (17),

$$\frac{dT_{oc,rel}}{dr} = \frac{dT_b}{dr} \left(\frac{h_c S_c + h_g S_g}{h_c S_c}\right) \tag{18}$$

Substituting equation (17) and (18) in equation (16), the first order differential equation of the blade temperature can be written as,

$$\frac{dT_b}{dr} \left(\frac{h_c S_c + h_g S_g}{h_c S_c}\right) - \frac{w^2}{c_{pc}} \{r_{root} + r\} - \frac{h_g S_g h_c S_c (\ )}{m_c (h_c S_c + h_g S_g)} (T_{og,rel} + T_{oc,rel}) = 0 \tag{19}$$

Rearranging the terms, equation (20) can rewritten as,

$$\frac{dT_b}{dr} \left(1 + \frac{h_g S_g}{h_c S_c}\right) - \frac{w^2}{c_{pc}} \{r_{root} + r\} - \frac{h_g S_g}{m_c c_{pc}} (T_{og,rel} + T_b) = 0 \tag{20}$$

For the flow of the coolant inwards, 'b' varies from 0 to L the differential equation (20) can be written as:

$$\frac{dT_b}{dr} \left(1 + \frac{h_g S_g}{h_c S_c}\right) - \frac{w^2}{c_{pc}} \{r_{root} + L - b\} - \frac{h_g S_g}{m_c c_{pc}} (T_{og,rel} + T_b) = 0 \tag{21}$$

Also, the relative total temperature of the coolant is given by,

$$T_{oc,rel} = T_b \left(\frac{h_c S_c + h_g S_g}{h_c S_c}\right) - \frac{h_g S_g T_{og,rel}}{h_c S_c} \tag{22}$$

Equations (20) and (21) are first order differential equations. In the above equations, the external heat transfer coefficient is calculated as follows using the Nusselt number [5].

$$Nu_g = k(\mathrm{Re}_g)^* \left(\frac{T_{og,rel}}{T_b}\right) \tag{23}$$

  

Where,

$$k = \frac{Nu^*{}_g}{(2\times 10^5)^x} \tag{24}$$

And,

$$\mathrm{Re}_g = \frac{\rho_g V_g c}{\mu_g} \tag{25}$$

In equation (24) $x$ and $Nu^*{}_g$ are obtained from the curves given in [3]. The exponent '$x$' is a function of the ratio of inlet and exit gas angles of the blade taken into consideration in calculations. The blade geometry taken from [4] has constant gas angles across the length.
The exponent '$y$' is given by the expression [2],

$$y = 0.14\times\left(\frac{\mathrm{Re}_g}{(2\times 10^5)}\right)^{-0.4} \tag{26}$$

The external gas Nusselt number has a characteristic dimension of blade chord is given by following expression,

$$Nu_g = \frac{h_g c}{k_g} \tag{27}$$

Combining equations (23) and (27)

$$h_g = k\frac{(\mathrm{Re}_g)^*}{c}k_g\left(\frac{T_{og,rel}}{T_b}\right) \tag{28}$$

The gas flow Reynolds number '$Re_g$' given by the expression [2],

$$\mathrm{Re}_g = \frac{m_g c}{\mu_g s L\cos\alpha_2} \tag{29}$$

In above equations, the thermo-physical properties of external gas are given by Mansour and Award R. [6]. Nusselt number to compute the heat transfer coefficient on air side is calculated as follows:

$$Nu_c = 0.034\left(\frac{L}{D}\right)^{-0.1}(\mathrm{Pr}_c)^{0.4}(\mathrm{Re}_c)^{0.8}\left(\frac{T_{oc,rel}}{T_b}\right)^{0.55} \tag{30}$$

In the above equation, the coolant properties used are given by Weng et al. [7] and the wet air properties are given by Hyland and Wexter [8]. Since the relative total temperature of coolant is varying along blade span, the thermo-physical properties of coolant are also varying along with and these variations in the properties have been considered in the present analysis.

The Reynolds number for the calculation of the internal heat transfer coefficient is given as,

$$\mathrm{Re}_c = \frac{\rho_c V_c D}{\mu_c} = \frac{m_c D}{\mu_c A_c} \tag{31}$$

With, mass flow rate of coolant

$$m_c = \rho_c V_c A_c \tag{32}$$

The Nusselt number to calculate internal heat transfer coefficient has a characteristic dimension as the diameter of internal coolant channel is given as follows:

$$Nu_c = \frac{h_c D}{k_c} \tag{33}$$

The coolant properties are taken from [7]. The characteristic dimension for the Reynolds number of internal flow of coolant is diameter of coolant channel. The diameter of coolant channel is constant along blade span. Combining equations (31) and (29), the Reynolds number on the coolant side is given by

$$\mathrm{Re}_c = \phi \frac{s}{c} \frac{\mu_g}{\mu_c} \cos\alpha_2 \frac{D}{A_c} \mathrm{Re}_g\, L \tag{34}$$

The flow ratio is the ratio of mass flow rate of coolant and external gas, thus

$$\phi = \frac{m_c}{m_g} \tag{35}$$

Combining equations (30), (33) and (34),

$$h_c S_c = 0.034 \left(\frac{L}{D}\right)^{-0.1} (\mathrm{Pr}_c)^{0.4} \left(\phi \frac{s}{c} \frac{\mu_g}{\mu_c} \cos\alpha_2 \frac{D}{A_c} \mathrm{Re}_g\, L\right)^{0.8} \left(\frac{T_{oc,rel}}{T_b}\right)^{0.55} S_c \tag{36}$$

Equation (28) is written as,

$$h_g S_g = k \frac{(\mathrm{Re}_g)^*}{c} k_g \left(\frac{T_{og,rel}}{T_b}\right)^{0.55} S_g \tag{37}$$

Hence, the ratio '$\frac{h_c S_c}{h_g S_g}$',

$$\frac{h_c S_c}{h_g S_g} = 0.034 \left(\frac{L}{D}\right)^{-0.1} (\mathrm{Pr}_c)^{0.4} \left(\phi \frac{s}{c} \frac{\mu_g}{\mu_c} \cos\alpha_2 \frac{D}{A_c} \mathrm{Re}_g\, L\right)^{0.8} \left(\frac{T_{oc,rel}}{T_b}\right)^{0.55}$$

$$S_c \frac{\mathrm{c}}{\mathrm{Re}^x{}_g\, k_g} \left(\frac{T_{og,rel}}{T_b}\right)^{0.55} \frac{Z}{k S_g} \tag{38}$$

Where the parameter '$Z$' is the shape factor [1] given by the equation:

$$Z = \frac{(S_c / c)^{1.2}}{(A_c / c^2)} \tag{39}$$

Equation (21) for outward flow and equation (22) for inward flow of coolant are solved numerically by the forward difference method for each element to obtain a local blade temperature '$T_b$'. For each element coolant temperature '$T_{oc,rel}$' is obtained by equation (22). The external gas mass flow rate of 4 kg/s is assumed in the present analysis [4].

## RESULTS AND DISCUSSION

Figure 1 shows that temperature of the blade along its length for ω = 0 rad/s and mass flow rate ratio 0.01 for dry air and wet air with different relative humidity. The blade temperature increases with span length as the coolant's relative total temperature increase as it flows along span since the coolant takes up the heat during the flow. Also, at any point on blade, temperature decreases with increase in relative humidity.

This may be attributed to improved heat transfer rate due to moisture in air. The blade temperature is obtained by solving the equation (20) by applying forward difference scheme since the parameters as defined previously are varying at each point along the blade span.

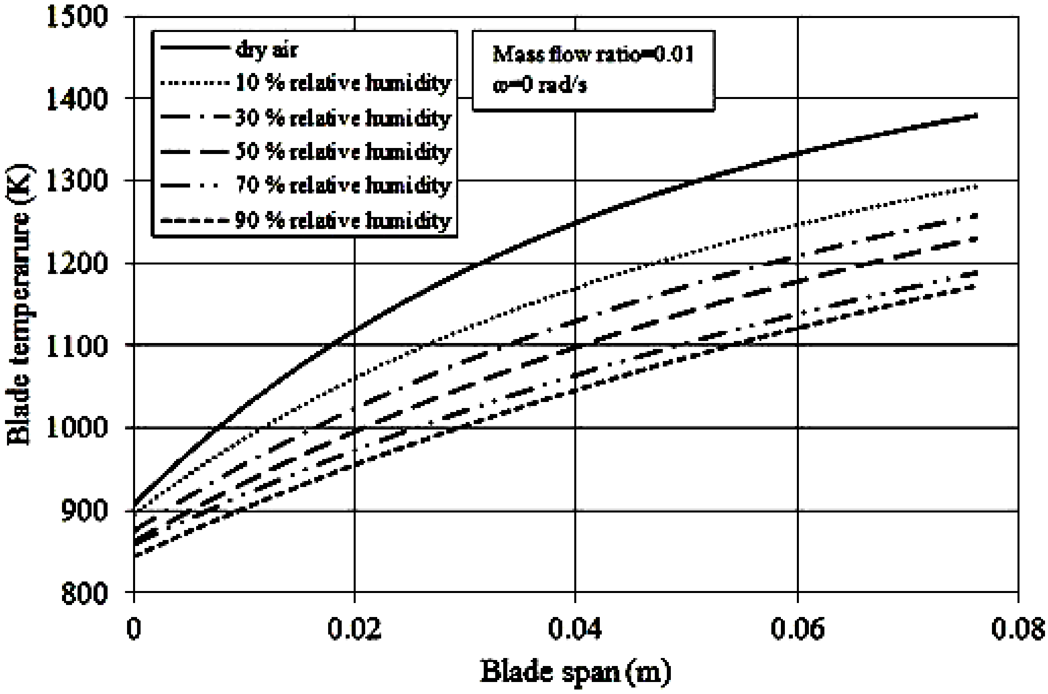


***Fig. 1:*** *Temperature Variation of Blade along span for Angular speed ω=0 rad/s and Mass Flow Ration of 0.01 for Different Types of Coolant.*

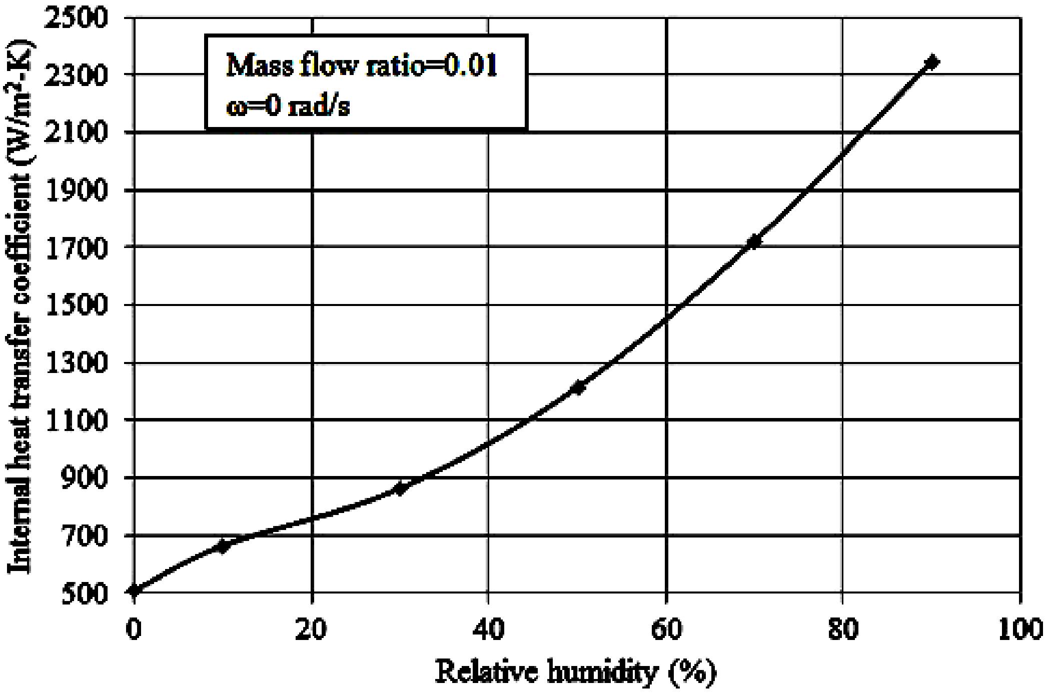


***Fig. 2:*** *Effect of Relative Humidity on Heat Transfer Coefficient.*

Figure 1 gives the temperature at various points along blade span. The temperature of blade increases from root (0 m) to tip (0.073 m) as the relative total temperature of coolant increases during the outward flow due to heat transfer between blade and coolant. Figure 1 also shows that the blade temperature decreases with increase in relative humidity. This may be attributed to improved heat transfer coefficient in two phase flow, due to presence of water in air. The same can be depicted through Figure 2, which gives the effect of relative humidity on heat transfer coefficient. The heat transfer coefficient increases from 664.1 W/m$^2$-K to 2345.32 W/m$^2$-K as relative humidity increases from 10% to 90%. Figure 3 shows the variation of blade tip temperature at different relative humidity. The blade tip temperature reduces from 1293.43 K to 1172.59 K with increase in relative humidity from 10% to 90% for mass flow ratio of 0.01. The same has been discussed earlier and shown in Figure 2 for mass flow ratio of 0.02 and ω=0 rad/s.

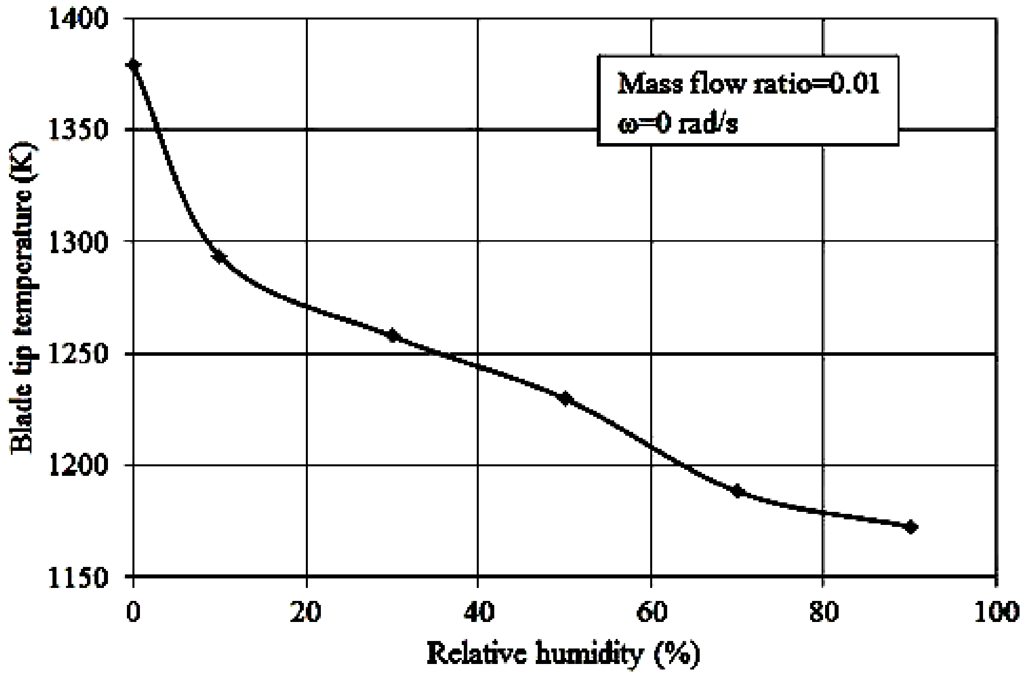


***Fig. 3*: *Temperature of Blade at Tip for Angular Speed ω=0 rad/s and Mass Flow Ratio of 0.01 for Different Types of Mist Air (based on Relative Humidity).***

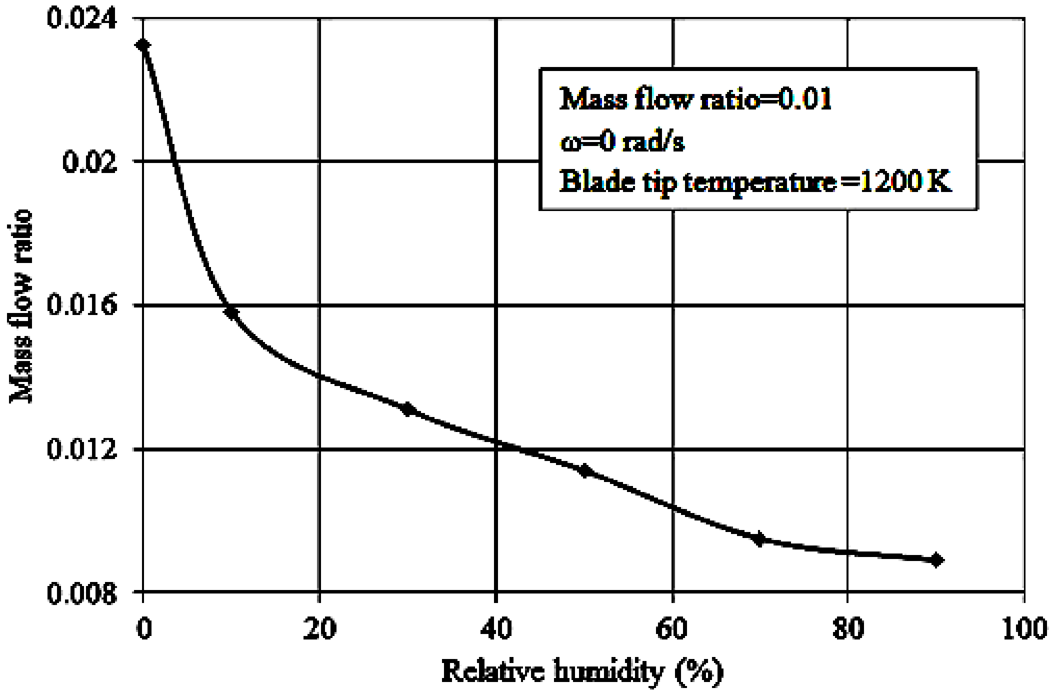


***Fig. 4:*** *Mass Flow Ratio Required to keep Blade Tip Temperature at 1200 K for ω=0 rad/s for Different Types of Mist Air (Based on Relative Humidity).*

Figure 4 show effect of relative humidity on mass flow rate. As the relative humidity increase the amount of mass flow rate reduces for a blade tip temperature of 1200 K. The mass flow ratio decreases from 0.0158 to 0.0089 as relative humidity increases from 10% to 90% to maintain 1200 K temperature at blade tip. This is due to improved heat transfer coefficient as discussed earlier. Thus, the wet air improves the cooling performance of the blade.

The theoretical results presented so far are for stationary blades. The code is further modified to investigate the effect of blade rotation on cooling performance. The investigation is carried out for both inward and outward flow of the coolant. Figure 5 shows temperature of blade at tip for different values of rotation for outward flow of coolant for different types of coolant for mass flow ratio of 0.02. It can be observed that temperature of blade at tip increases with increase in speed for outward flow of coolant. The temperature of blade at tip with dry air as coolant increases from 1235.43 K to 1293.96 K when angular speed of blade increase from 0 rad/s to 2093.33 rad/s for outward flow of coolant. This may be attributed to effect of centrifugal force on relative total temperature of coolant that increases with angular speed of blade for outward flow of coolant. Also, for any angular speed of blade, temperature of blade reduces with increase in relative humidity of coolant. The temperature of blade at tip decreases from 1193 K to 1115 K when relative humidity increase from 10% to 50%.

Figure 6 show temperature of blade at root for different angular speed for inward flow of coolant for mass flow ratio of 0.02. For inward flow of coolant 0 m in figure corresponds to blade tip and 0.073 m correspond to blade root. The coolant used is dry air and wet air with different relative humidity. It can be observed that temperature of blade at root decreases with increase in speed for inward flow of coolant. The temperature of blade at root reduces from 1235.43 K to 1178.42 K when angular speed of blade increases from 0 rad/s to 2093.33 rad/s for dry air as a coolant.

This may be attributed to the centrifugal force due to which relative total temperature of coolant decrease with increase in angular speed for inward flow. Also, for any angular speed of blade, temperature of blade reduces with increase in relative humidity of coolant. The temperature of blade at root decreases from 1108.41 K to 1043.57 K when relative humidity increases from 10% to 50%. Thus, wet air gives better cooling performance as compared to dry air and the reason has been discussed earlier.

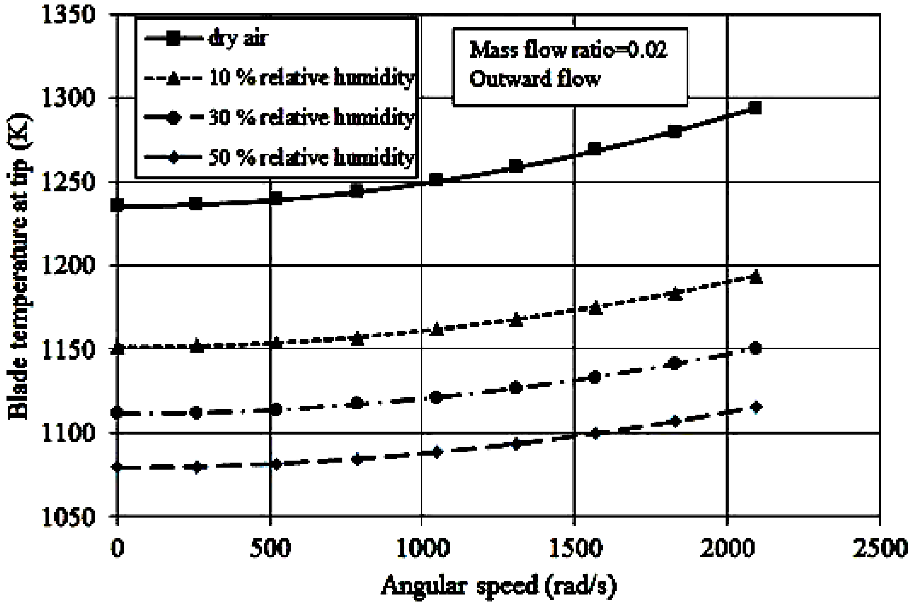


***Fig. 5:*** *Temperature of Blade at Tip for Different values of Rotation for Outward Flow of Coolant for Different Types of Coolant for Mass Flow Ratio 0.02.*

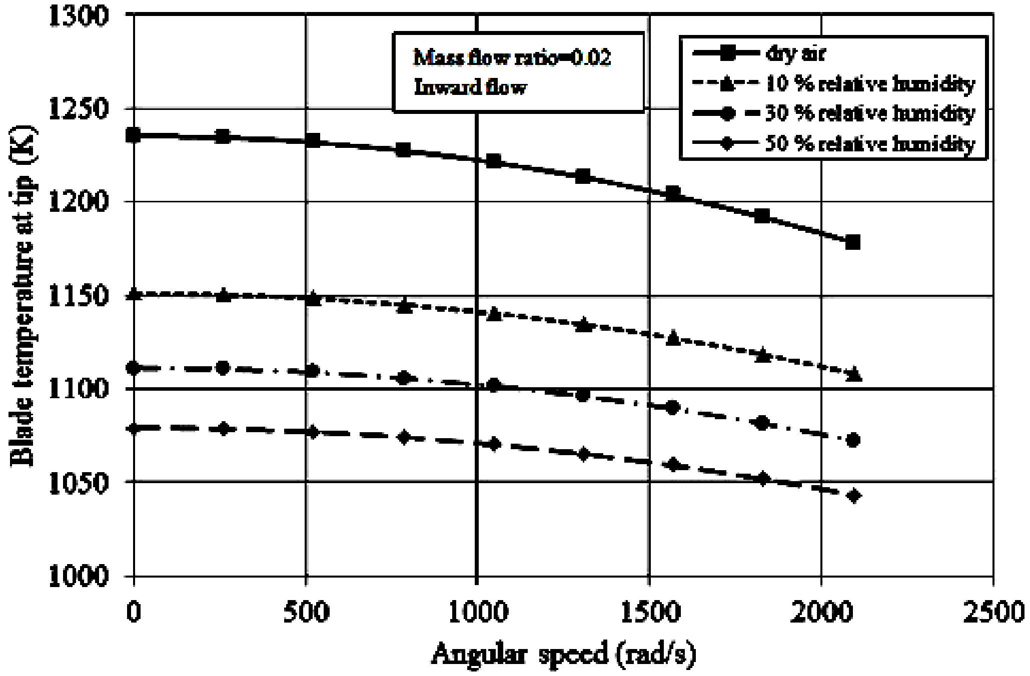


***Fig. 6:*** *Temperature of Blade at tip for Different values of Rotation for Inward Flow of Coolant for Different Types of Coolant for 0.02 Mass Flow Ratio.*

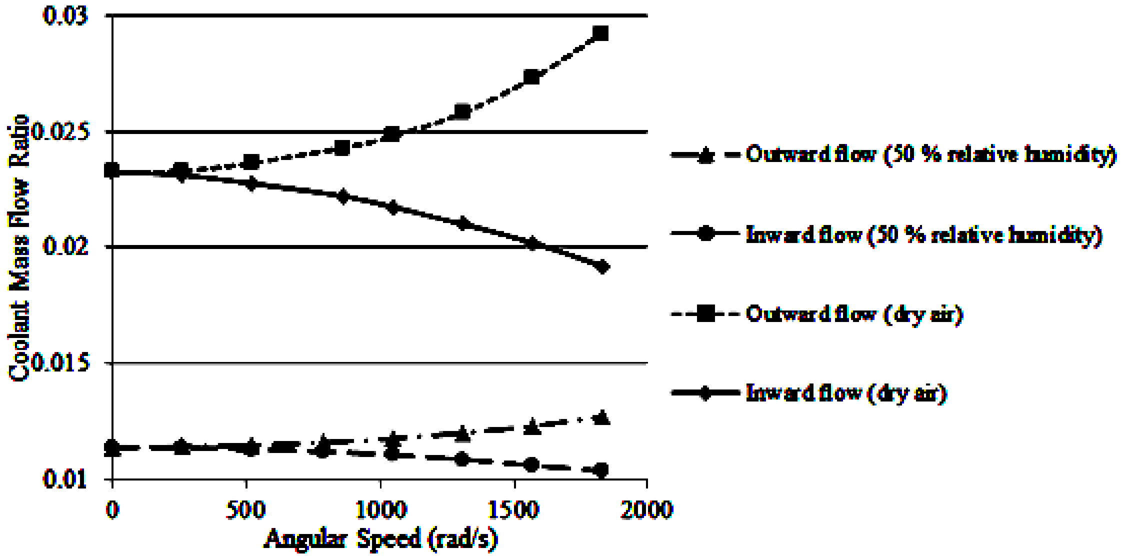


***Fig. 7:*** *Coolant Mass Flow Ration Required to Maintain Blade Tip Temperature to 1200 K at Blade Tip.*

Figure 7 show mass flow ratio of coolant required to maintain blade temperature at tip 1200 K for outward and inward flow of coolant for different types of coolant. Figure 7 shows that mass flow ratio required to maintain maximum blade temperature to 1200 K reduces with increase in speed for inward flow of coolant however it increases for outward flow of coolant. For dry air as a coolant, the coolant mass flow ratio increases from 0.02325 to 0.02925 when angular speed of blade increase from 0 rad/s to 2093.33 rad/s to maintain maximum blade temperature to 1200 K for outward flow of coolant. But the coolant mass flow ratio decrease from 0.02325 to 0.0192 when angular speed of blade increase from 0 rad/s to 2093.33 rad/s to maintain maximum blade temperature to 1200

K for inward flow of coolant. It can be observed that mass flow ratio required reduces for wet air as compared to dry air irrespective of direction of flow of coolant. The coolant mass flow ratio decreases from 0.02925 to 0.0127 at 2093.33 rad/s for outward flow of coolant for dry air and wet air with 50% relative humidity respectively.

## CONCLUSIONS

The cooling of a gas turbine blade using wet air and air as a coolant is analytically investigated. The investigation is carried out considering effect of rotation for inward and outward flow of coolant. The temperature of blade at tip with dry air as coolant increases from 1235.43 K to 1293.96 K when angular speed of blade increase from 0 rad/s to 2093.33 rad/s for outward flow of coolant due to the effect of centrifugal force on relative total temperature of coolant which increases with angular speed of blade for outward flow of coolant. The temperature of blade at root reduces from 1235.43 K to 1178.42 K when angular speed of blade increases from 0 rad/s to 2093.33 rad/s for dry air as a coolant for inward flow.

Wet air cooling performance is compared with dry air cooling. It has been observed that wet air provides better cooling and the performance improves with increase in relative humidity. The blade temperature decreases with increase in relative humidity due to improved heat transfer coefficient in two phase flow. The heat transfer coefficient increases from 664.1 $W/m^2$-K to 2345.32 $W/m^2$-K as relative humidity increases from 10% to 90%. Thus, wet air and inward flow gives better cooling in case of internally cooled gas turbine blade.

## NOMENCLATURE

| | |
|---|---|
| D | Average diameter of the coolant channel (m) |
| $r_{root}$ | Distance from the blade axis to the blade root (m) |
| L | Length of the turbine blade (m) |
| w | Angular velocity of the turbine blade (rad/s) |
| h | Heat transfer coefficient ($W/m^2$-K) |
| v | Relative velocity of the coolant in the channel (m/s) |
| $T_{b,t}$ | Temperature of blade at tip (K) |
| $T_{root}$ | Temperature of blade at root (K) |
| g | Gravitational acceleration ($m/s^2$) |
| Nu | Nusselt number |
| Pr | Prandtl number |
| $\varphi = \frac{m_c}{m_g}$ | Flow ratio |
| r | Varying radial distance from blade root |
| $T_g$ | Gas recovery temperature |
| $\rho$ | Density |
| $\mu$ | Viscosity (kg/m-s) |
| $\alpha$ | Gas angles |
| s | Blade pitch (m) |
| $\gamma$ | Specific heat ratio |
| c | Blade chord (m) |
| L | Blade span (m) |
| $A_c$ | Total cross sectional area of coolant channels ($m^2$) |
| $s/c$ | Blade pitch to chord ratio |
| Z | Passage shape parameter |
| $h_{oc.rel}$ | Relative total enthalpy (J/kg) |

*Suffix*

| | |
|---|---|
| B | Blade |
| c | Coolant |
| g | Gas |
| rel | Relative |
| t | Tip |

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