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Analysis of Transient Thermal Behaviour of a Different Profiled Fully Wetted Longitudinal Porous Fin with Internal Heat Generation

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Abstract

The transient thermal performance of a fully wetted longitudinal permeable fin under natural convection, radiation, and internal heat generation has been studied in the present problem. Due to variation of fin thickness with respect to fin length, it is possible to obtain different fin profiles such as triangular, convex and rectangular profiles. Here, the diverse profiles of the fin have been considered for the examination and Darcy's model was implemented to investigate the porous nature of the fin. The governed partial differential equation has been solved by applying finite difference method. The influence of dimensionless parameters on the transient thermal profile as well as the heat dissipation rate of the three types of fins, has been graphically illustrated and interpreted physically.

Keywords: *Finite Difference Method, Internal Heat Generation, Longitudinal Fin*

1.0 Introduction

Fins are tiny metal strips that are affixed to a surface to improve the transfer of heat from that surface to the fluid in the vicinity. Fins are used in a variety of industrial applications, including freezers, compressors, transformers, electrical motors, heat sinks, radiators, and solar collectors. The industry's main objective is to reduce fin size and cost. This requirement was frequently supported by the greater expense of the fin's weight in addition to the higher cost of the high-thermal-conductivity metals used to create finned surfaces. In these aspects many researchers have tried to investigate the different phenomenon which helps to enhance the cooling due to transmission of excess heat among them fin structure attached to the base surface gained much focus. In this regard, Kiwan[1] has tried to analyse the significance of permeable nature with in the fin surface to augment the dissipation of excess heat. In addition, the convective effect has also been considered for the study to address the transfer of heat phenomenon by Kiwan[2]. Bouaziz and Aziz[3] have used the double optimal linearization technique in order to observe the effect of radiation and temperature dependency of thermal conductivity of fin on the dissipation of heat to the surrounding. The different boundary conditions based on the tip of the fin has been discussed by Gorla and Bakier[4]. The semi-analytical method like differential transformation technique has been considered by Torabi and Yaghoobi[5] to explore the effect of radiation and convection on the thermal

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distribution through the fin surface. The wet nature of the surface of the fin helps in dissipating more heat to the surrounding. Therefore, Hatami *et al*.[6] have modelled the energy balance equation in semi-spherical fin under wetted condition and also discuss the efficiency in it. Turkyilmazoglu[7] also discussed the same for exponential fin model and compared the results with longitudinal fin. Darvishi *et al*.[8] have applied the Darcy's rule to explore the effect of porous wetted fin on the transmission of heat. The finite element technique has been utilized by Gireesha *et al*.[9] to elucidated the influence of wetted nature of the radial fin.

The internal heat generation can be observed in many engineering applications; therefore, it grabs the attention of many researches. Hatami *et al*.[10] have elaborated the significance of thermal dependent heat generation along with different materials of fin on the heat transfer analysis. Dogonchi and Ganji[11] have considered the thermal dependent parameters and analysed their consequence on the temperature field of the fin. The fin of exponential profile moving with constant velocity has been taken for the investigation by Turkyilmazoglu[12] and described the heat generation effect. The hybrid block technique has been applied by Alkasassbeh *et al*.[13] to study the internal heat generation effect on the convective fin. Das and Kundu[14] have explained the magnetic and heat generation effect on the porous fin of radial profile. Varun Kumar *et al*.[15] have also been examined the electromagnetic field effect and heat generation on the triangular profiled porous fin.

In view of the thickness of the fin diverse shapes of the fin can be obtained. The relative analysed of diverse geometries of the longitudinal fin under nonlinearity condition has been considered by Torabi *et al*.[16] and applied the differential transformation technique to analyze it. The material effect on the diverse profile of the circular fin has been addressed by Hatami and Ganji[17]. Specifically fin made with ceramic materials has been considered by Hatami and Ganji[18] and presented the various profile of the fin and their thermal performances. Mosayebidorcheh *et al*.[19] also carried out the material and shape analysis of the longitudinal fin and also investigated the optimization under thermal dependent conditions.

The time dependent phenomenon has been a very important feature. Initially, fin heat transfer varies along with time and it reaches steady state after certain state. This has been analysed by Farvishi *et al*.[20] for permeable fin. Mosayebidorcheh *et al*.[21] have also considered the unsteady behavior of the longitudinal fin of different geometries with interior heat generation. Sowmya *et al*.[22] have used the finite difference technique to examine the various profiles of longitudinal fin under transient condition. Varun Kumar *et al*.[23] have considered the exponential plate have thermal varying properties under unsteady condition and investigated the heat transference through it.

Based on the aforementioned study, in the current analysis focused on the influence of heat generation, convection and radiation on the thermal behavior through the longitudinal porous fin under wetted condition for three particular geometries like rectangular, convex, triangular profiles. The modelled equations have been solved using finite difference method. To examine the heat dissipation from the fin's surface, a parametric analysis was conducted.

2.0 Mathematical Formulation

The unsteady one-dimensional heat dissipation through the porous longitudinal fin of length l, width w and thickness t is varying with length of the fin x i.e., $t(x)$. Based on the thickness various fin profile can be obtained. In the present study, considered three such geometries like triangular fin, convex fin and rectangular fin and are as portrayed in Figure 1. The base of the fin is kept a fixed temperature T_b and thickness t_b. Initially, there is a thermal equilibrium between the fin surface and the surrounding. Here, the fin is under completely wetted state and dissipates heat because of convection and radiation effect.

The time-dependent energy balance equation of the permeable fin with its cross section is as mentioned below

$$\rho c_p A_c \frac{\partial T}{\partial t^*} = q(x) - q(x+dx) + q^* t(x) w dx$$

$$2h_D w dx i_{fg}(1-\phi)(\omega-\omega_a) - 2\varepsilon\sigma w dx(T^4 - T_a^4) - 2\dot{m}\, c_p(T-T_a)$$

$$-2(1-\phi)wh(T-T_a)dx. \qquad (1)$$

Here, LHS is the unsteady term and in the RHS, initial two terms represent the conductive rate of heat transfer, third term represents for internal heat generation, the fourth term is because of wet condition, fifth term is radiative energy loss, sixth term stands for the loss of energy because of mass flow of inundated fluid and last term signifies the convection. In equation (1) the distinction between saturated and surrounding fluid humidity ratio is represented as $(\omega-\omega_a)=b_2\,(T-T_a)$, where b_2 is the variable constraint, $\dot{m}$ is the rate of fluid mass flow in a porous substance and is given by

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$$\dot{m} = wv(x)\rho dx, \tag{2}$$

and as per the Darcy's model the passage velocity $\upsilon(x)$ is defined as

$$\upsilon(x) = \frac{(T-T_a)\beta_f K g}{\nu_f}. \tag{3}$$

From the Fourier's law of conduction,

$$q = -t(x)wk_{eff}\frac{\partial T}{\partial x}, \tag{4}$$

where k_{eff} is the effective thermal conductivity and is noted as

$$k_{eff} = k_f\phi + k_s(1-\phi) \tag{5}$$

The internal heat generation q^* is considered as linear function of temperature and is stated as

$$q^* = q_a^*(1+\varepsilon_g(T-T_a)) \tag{6}$$

where, ε_g is a factor of internal heat generation.

The convective heat transmission coefficient is defined as

$$h = h_a\left[\frac{T-T_a}{T_b-T_a}\right]^p \tag{7}$$

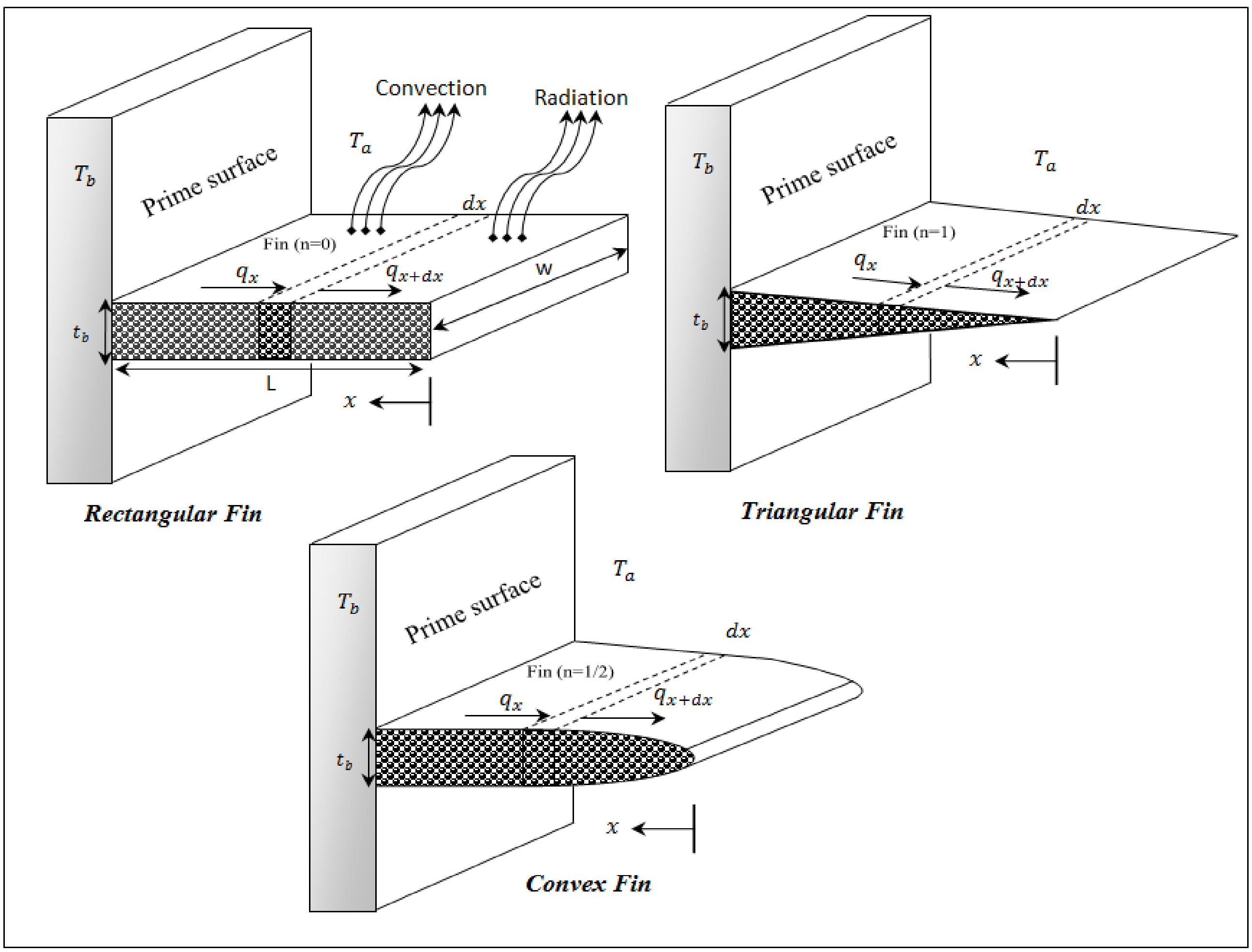

Figure 1. The geometry and coordinate system of present problem.

Here, the local semi fin thickness t(x) varies for diverse geometries i.e., case1: for rectangular shape t_b, case 2: for convex profile $t_b\left(\frac{x}{L}\right)^{1/2}$, case 3: for triangular profile $t_b\left(\frac{x}{L}\right)$.

Equation (1) is simplified by applying equations (2)-(7) and it can be rewritten as

Case1: For rectangular fin

$$\frac{\rho c_p}{k_{eff}}\frac{\partial T}{\partial t^*}=\frac{\partial^2 T}{\partial x^2}+\frac{q_a^*[1+\varepsilon_g(T-T_a)]}{k_{eff}}-\frac{2(1-\phi)h_a i_{fg} b_2 (T-T_a)^{1+p}}{c_p Le^{\frac{2}{3}} k_{eff} t_b (T_b-T_a)^p}$$

$$-\frac{2\sigma\varepsilon}{k_{eff}t_b}(T^4-T_a^4)-\frac{2\rho g\beta_f K c_p}{\nu_f k_{eff} t_b}(T-T_a)^2-\frac{2(1-\phi)h_a(T-T_a)^{1+p}}{k_{eff}t_b(T_b-T_a)^p}. \tag{8a}$$

Case 2: For convex fin

$$\frac{\rho c_p}{k_{eff}}\frac{\partial T}{\partial t^*}=\frac{\partial}{\partial x}\left(\left(\frac{x}{L}\right)^{\frac{1}{2}}\frac{\partial T}{\partial x}\right)+\left(\frac{x}{L}\right)^{\frac{1}{2}}\frac{q_a^*[1+\varepsilon_g(T-T_a)]}{k_{eff}}-\frac{2(1-\phi)h_a i_{fg} b_2 (T-T_a)^{1+p}}{c_p Le^{\frac{2}{3}} k_{eff} t_b (T_b-T_a)^p}-\frac{2\sigma\varepsilon}{k_{eff}t_b}(T^4-T_a^4)-\frac{2\rho g\beta_f K c_p}{\nu_f k_{eff} t_b}(T-T_a)^2-\frac{2(1-\phi)h_a(T-T_a)^{1+p}}{k_{eff}t_b(T_b-T_a)^p} \tag{8b}$$

Case 3: For triangular fin

$$\frac{\rho c_p}{k_{eff}}\frac{\partial T}{\partial t^*}=\frac{\partial}{\partial x}\left(\frac{x}{L}\frac{\partial T}{\partial x}\right)+\frac{x}{L}\frac{q_a^*[1+\varepsilon_g(T-T_a)]}{k_{eff}}-\frac{2(1-\phi)h_a i_{fg} b_2 (T-T_a)^{1+p}}{c_p Le^{\frac{2}{3}} k_{eff} t_b (T_b-T_a)^p}-\frac{2\sigma\varepsilon}{k_{eff}t_b}(T^4-T_a^4)-\frac{2\rho g\beta_f K c_p}{\nu_f k_{eff} t_b}(T-T_a)^2-\frac{2(1-\phi)h_a(T-T_a)^{1+p}}{k_{eff}t_b(T_b-T_a)^p}. \tag{8c}$$

In order to non-dimensionalize the above equations, following dimensionless constraints are considered;

$$\theta=\frac{T}{T_b},\qquad \theta_a=\frac{T_a}{T_b},\qquad X=\frac{x}{L},\tau=\frac{\alpha t^*}{L^2}. \tag{9}$$

After using equation (9) in equation (8), they get altered to dimensionless form as follows:

Case 1: For rectangular fin

$$\frac{\partial\theta}{\partial\tau}=\frac{\partial^2\theta}{\partial X^2}+G(1+\varepsilon_G(\theta-\theta_a))-Nc(\theta-\theta_a)^2-Nr(\theta^4-\theta_a^4)-\frac{m_2(\theta-\theta_a)^{p+1}}{(1-\theta_a)^p}. \tag{10a}$$

Case 2: For convex fin

$$\frac{\partial\theta}{\partial\tau}=\frac{\partial}{\partial X}\left(X^{\frac{1}{2}}\frac{\partial\theta}{\partial X}\right)+X^{\frac{1}{2}}G(1+\varepsilon_G(\theta-\theta_a))-Nc(\theta-\theta_a)^2-Nr(\theta^4-\theta_a^4)-\frac{m_2(\theta-\theta_a)^{p+1}}{(1-\theta_a)^p}. \tag{10b}$$

Case 3: For triangular fin

$$\frac{\partial\theta}{\partial\tau}=\frac{\partial}{\partial X}\left(X\frac{\partial\theta}{\partial X}\right)+XG(1+\varepsilon_G(\theta-\theta_a))-Nc(\theta-\theta_a)^2-Nr(\theta^4-\theta_a^4)-\frac{m_2(\theta-\theta_a)^{p+1}}{(1-\theta_a)^p}. \tag{10c}$$

Equivalent initial and boundary conditions are:

$$\theta(X,0)=0,\theta(1,\tau)=1,\frac{\partial}{\partial X}\theta(0,\tau)=0. \tag{11}$$

In above equation, X is dimensionless length, θ is nondimensional temperature, τ is nondimensional time, $Nc=\frac{\rho g\beta_f K c_p T_b L^2}{\nu_f k_{eff} t_b}$ is a convective parameter, $G=\frac{L^2 q_a^*}{k_{eff}T_b}$ is the generation number, θ_a is a dimensionless ambient temperature, $Nr=\frac{\varepsilon\sigma L^2 T_b^3}{k_{eff}\, t_b}$ is a radiative parameter, $\varepsilon_G=\varepsilon_g T_b$ is the nondimensional inner heat generation, p is the power index, m_2 is a wet permeable constraint and is a sum of $m_0=\frac{h_a L^2\,(1-\phi)}{k_{eff}\,t_b}$ and $m_1=\frac{h_a i_{fg}(1-\phi)b_2 L^2}{k_{eff}\,t_b c_p Le^{\frac{2}{3}}}$..

3.0 Results and Discussion

The parabolic partial differential equations 9(a, b, c) with conditions (10) are simplified numerically by Finite Difference Method (FDM) with the centre implicit scheme. The time and space steps are chosen as Δt = Δx = 0.001. The prominence of non-dimensional parameters is determined through the graphical illustrations. The outcomes are displayed in Figures 2-11.

The consequence of convective constraint on the temperature performance of the longitudinal permeable fin of various shapes has been displayed in Figure 2. It is observed that, as Nc decreases, thermal field enhances. This is due to the natural convection around the fin surface.

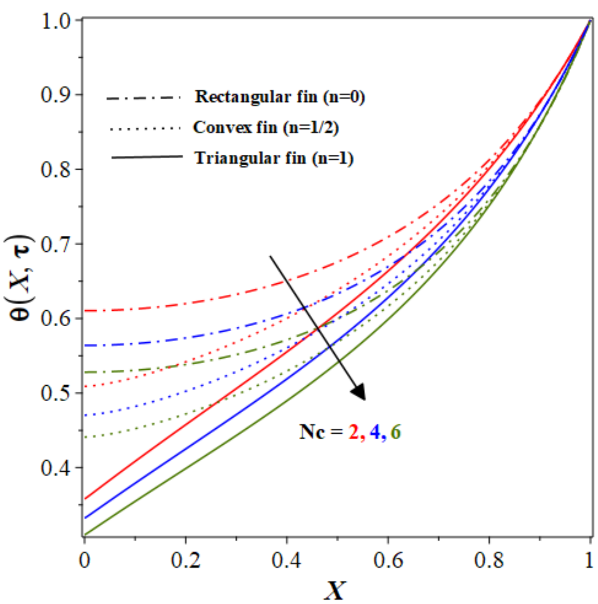

Figure 2. Effect of convective parameter on the fin temperature distribution.

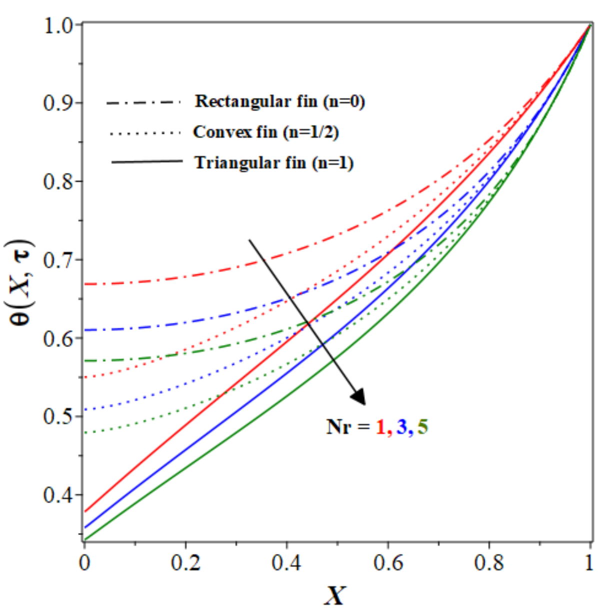

Figure 3. Effect of radiative parameter on the fin temperature distribution.

As the convective parameter upsurges, heat dissipation from surface of the fin enhances. Therefore, higher values of convective parameter are favourable for improved heat transmission rate. Additionally, the temperature transfer rate is relatively high for triangular profile than convex and rectangular profiled fin.

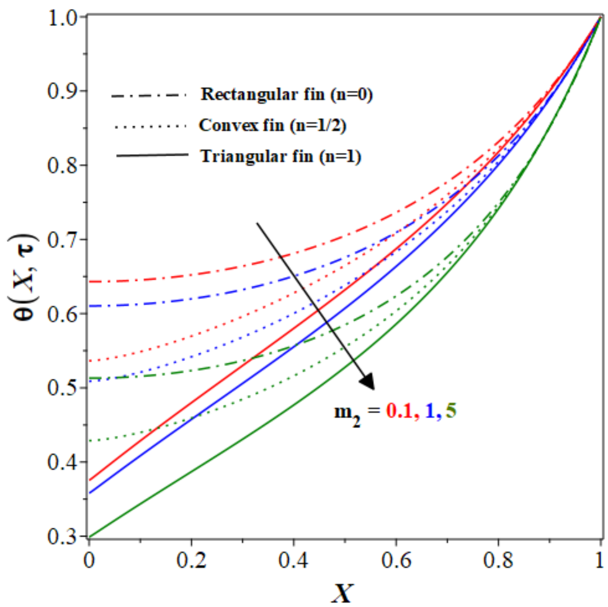

Figure 4. Effect of wet parameter on the fin temperature distribution.

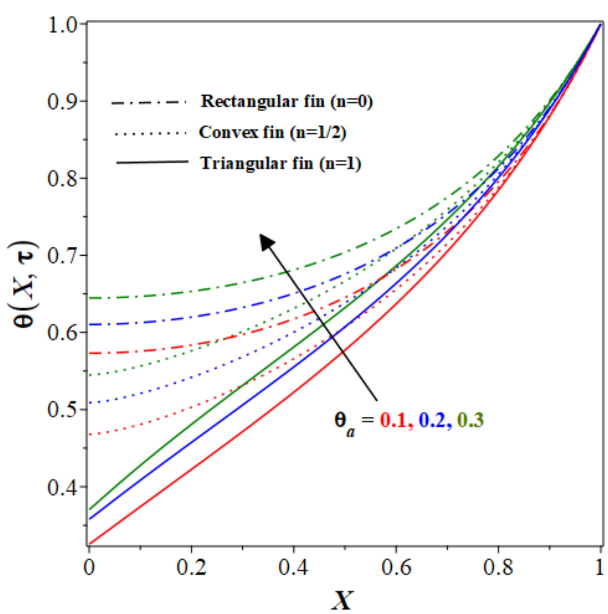

Figure 5. Effect of ambient temperature on the fin temperature distribution.

Figure 3 discloses the effect of radiative factor on the thermal behavior of different geometries of the fin. The fin surface temperature reduces with increase in the radiative constraint due to the radiation around the fin which carries the heat from the surface of the fin. Thus, the radiative parameter positively influences the thermal

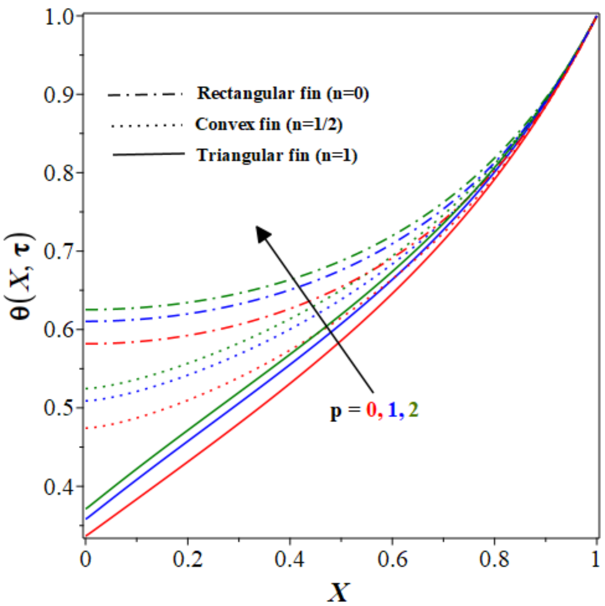

Figure 6. Effect of power index on the fin temperature distribution.

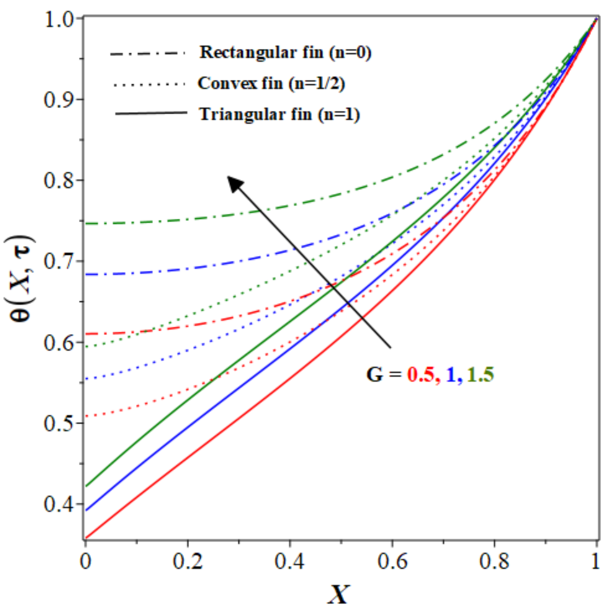

Figure 7. Effect of generation number on the fin temperature distribution.

drop rate and it is observed in all the three geometries of the fin.

Figure 4 explains the influence of wet permeable factor on the temperature profile of the fin. It is observed that the augment in wet permeable parameter leads to rise in the heat dissipation on the surface of the fin and resulting in

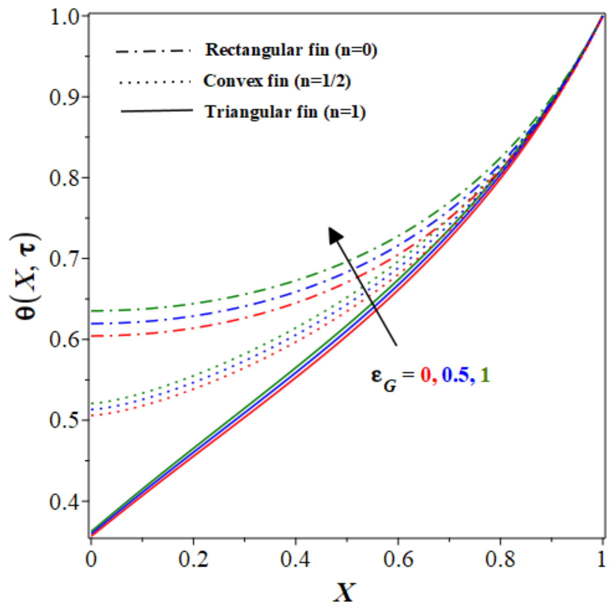

Figure 8. Effect of dimensionless internal heat generation parameter on the fin temperature profile

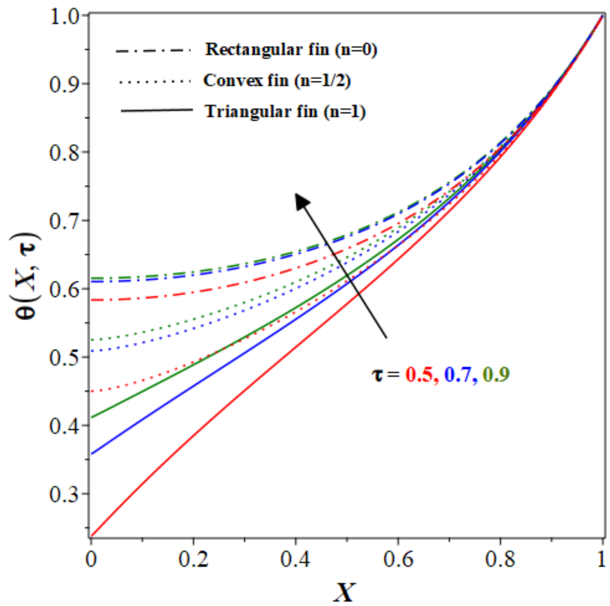

Figure 9. Effect of dimensionless time on the fin temperature profile

descending temperature profile. This is because of the wet condition around the fin surface. The wet nature helps in the augment of rate of heat exchange. On the other hand, rate of temperature reduction is more in case of triangular profile followed by convex and rectangular profiles.

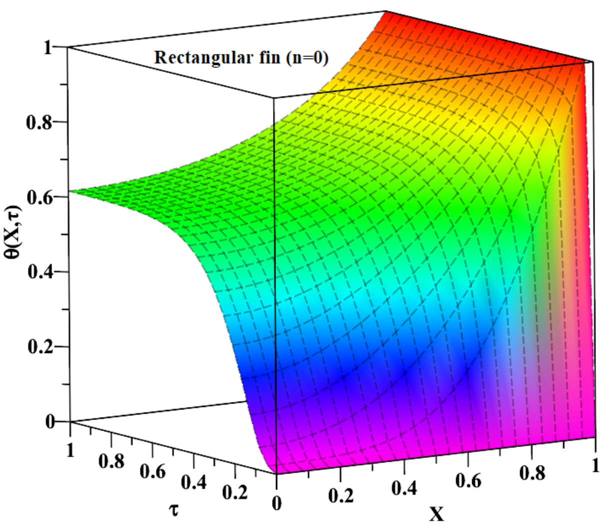

Figure 10. Surface plot of unsteady temperature profile for rectangular fin.

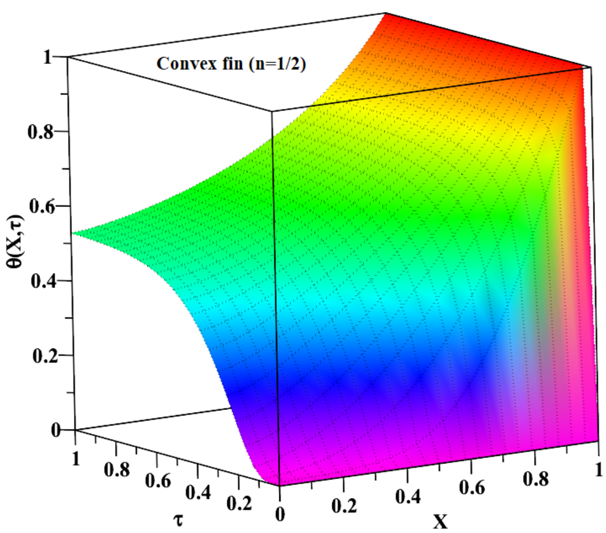

Figure 11. Surface plot of unsteady thermal profile for convex fin.

From the Figure 5, the consequence of ambient temperature on the thermal behaviour of the fin is displayed for various geometries. Here larger the ambient temperature, high will be the temperature of the surrounding. This is because of the rise in the ratio of ambient temperature to the fin base temperature. This means that, as the ambient temperature intensifies, the temperature difference of the fin surface and the surrounding lessens. As per Newton's law of cooling,

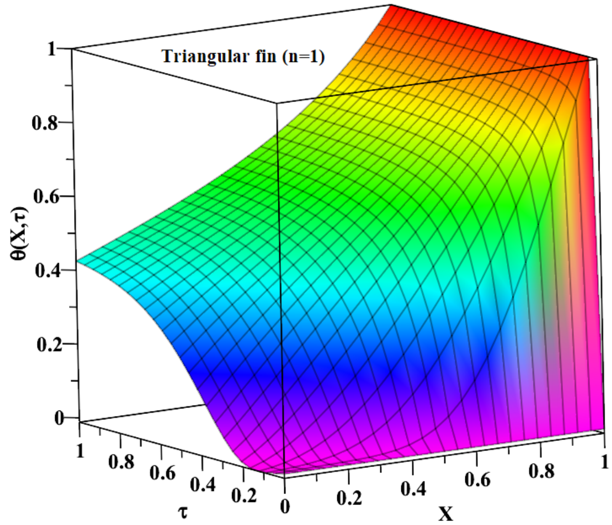

Figure 12. Surface plot of unsteady thermal profile for triangular fin.

the cooling impact reduces because of the decrease in temperature difference between surface and surrounding.

In Figure 6, consequence of power index (p) on the temperature performance of various structured permeable fin is presented. As the power index upsurges, thermal field also rises. The power index is connected with the convective heat transmission coefficient and it represents the different flow regimes. Hence, augmentation in the power index leads to nonlinearity and decreases the heat drop rate for all the three geometries of fin.

The impact of G on the temperature field is depicted in Figure 7. It is found that, the amplification in G leads to more thermal profile. This is due to the negative effect of rise in the fin temperature with the generation of internal heat on fin cooling. Also, it is observed that the thermal field is large for rectangular fin followed by convex fin and triangular fin. The impact of ε_G on the thermal performance of the fin is portrayed in Figure 8. It is determined that, the rise in ε_G tend to increase the thermal profile and reduce the heat drop rate. This is due to the enhancement in interior generation of heat.

The unsteady temperature field of the fin of various geometries is portrayed in Figure 9. It is noticed that, as time boosts thermal distribution also raises up to certain level, afterwards it turns into constant. It is very interesting fact that, the transient impact is noticed at first and later a steady state is attained and is clearly seen in surface plot for different geometries like rectangular, convex and triangular fin as in Figures 10, 11 and 12 respectively. Here, the time dependent temperature profile value enhances with time but in a while, it accomplishes a steady-state condition. Additionally, the transient temperature value of triangular profile is smaller than the convex and rectangular profiles.

4.0 Conclusion

The time dependent temperature distribution of the longitudinal permeable fin of various geometries like rectangular, triangular and convex profiles in completely wet surrounding have been investigated graphically using Finite Difference Method. The major outcomes of the current scrutinization are as follows:

- The convective as well as the radiative parameters show a predominant impact on the fin heat transmission rate.
- The heat reduction rate in fin intensifies with higher values of the wet porous constraint.
- The transient temperature behavior increases with rise in ambient temperature and power index which means, it decreases the rate of heat transmission.
- The internal heat generation number reduce the heat exchange rate.
- The various geometries of the fin have a pronounced influence on the heat dissipation amount.
- The rate of thermal transfer is high for a triangular profile followed by convex and rectangular profiled fin.

5.0 Acknowledgment

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