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# Performance Enhancement of Parabolic Trough Collector with Oblique Delta-Winglet Twisted-Tape Inserts

Atwari Rawani*[,1], Suresh Prasad Sharma*, and Krishna Dev Prasad Singh*

***Abstract –*** *The aim of this study is to analytically investigate the performance of parabolic trough collector (PTC) with oblique delta-winglet twisted tape insert in the absorber tube. The heat transfer equations for fully developed flow under quasi-steady state conditions have been developed in order to analyse entropy generation, exergy efficiency, thermal efficiency, rise in fluid temperature and to study the effect of system and operating parameters on performance. A computer program, based on mathematical models is developed in* $C^{++}$ *language to estimate the temperature rise of fluid for evaluation of performances under specified conditions. For numerical simulations four different twist ratio, x = 1,2,3,4 and mass flow rate 0.06 kg/s to 0.16 kg/s (3000≤Re≤9000) are used. This study shows that twisted tape insert when used shows great promise for enhancing the performance of PTC. Results show that for x=1, Nusselt number/heat transfer coefficient is found to be 3.24 and 2.91 times over plain absorber of PTC at mass flow rate of 0.06 kg/s and 0.16 kg/s, respectively; while corresponding enhancement in thermal efficiency is 12.05% and 4.95%, respectively. Also, the exergy efficiency has been found to be 12.05% and 4.92% and enhancement factor is 1.121 and 1.049 for same set of conditions.*

***Keywords –*** collector efficiency factor, entropy generation, exergy efficiency, thermal efficiency.

## 1. INTRODUCTION

The worldwide energy demand is increasing day by day due to rapid industrialization. The fossil fuels fulfill more than 90% of world's energy demands. The fossil fuels are limited, exhaustible and non-renewable and they create environment pollution. The combustion of fossil fuels produces carbon dioxide, it is one of the greenhouse gases contributing to global warming. On the other hand use of renewable energy resources, can prime to decrease in fossil fuels consumption, inexhaustible, sustainable, creates zero or few greenhouse gas emissions and will never run out with negligible impact on the environment.

Solar energy is one of the largest renewable energy resources. The greatest advantage of the solar energy is that it is easily available and abundant in nature. Parabolic trough collectors (PTCs) are one of the main solar heat collector elements, which can be applied for light structure and low-cost systems for producing heat up to 400$^{o}$C.

The solar collector tube is a basic component in parabolic trough solar thermal generation system which converts solar radiation into thermal energy. The collector tube consists of an inner metal absorber tube and the outer glass cover tube. The solar radiation is redirected and concentrated on a linear receiver in focal line. The concentrator reflects solar radiation to the outer surface of the inner tube and is absorbed by tube wall. Then, a bulk of the energy will be accompanied to the inner surface of the inner tube and is transferred from the working fluid in inner tubes with the mixed convective heat transfer.

Parabolic trough collectors are attractive solar energy collection device for attaining the high temperature. The performances of parabolic trough collector can be enhanced by using heat transfer augmentation techniques such as strips, coiled wires, various kinds of twisted tape inserted for swirl generator inside the absorber tube. Bhattacharyya *et al.* [1] investigated the Nusselt number and the friction factor for laminar flow through a circular duct having roughness of integral transfer rib and fitted with centre-cleared twisted-tape. They obtained high Prandtl number ranges from 235 - 537 at twist ratio 2.5 and servotherm medium oil. In laminar flow, the combination of centre-cleared twisted tape and integral transfer rib perform better than the individual heat transfer enhancement technique in a certain level of centre-clearance. Ghadirijafarbegloo *et al.* [2] investigated the convection heat transfer coefficient in parabolic trough collector with perforated louvered twisted tape inserts. They validated the experimental data for three different twisted tape ratios 2.67, 4 and 5.33. They determined the heat transfer rate and pressure drop for fully developed condition and different Reynolds number. The result shows that the heat transfer and friction factor enhances in comparison to a typical plain twisted tape in tube. Wang and Sunder [3] investigated the heat transfer coefficient in a circular duct with twisted tape inserts. The enhancement of heat transfer coefficient were 3.0 and 3.5 times to plain tube value in Reynolds number (Re<2000) and (5000<Re<45000), respectively. Eiasma-ard and Promvonge [4] experimentally investigated the enhancement of heat transfer with alternative clockwise and counter clockwise twisted tape inserts in circular tube. Their results showed that the heat transfer enhancement of alternative clockwise and counter clockwise twisted tape with typical twisted tape and

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plain tube are 12.8 – 41.9% and 27.3 – 90.5%, respectively. Under the uniform wall heat flux condition, the influences of the tube with perforated twisted tape were experimentally investigated by Thianpong *et al.* [5]. Their results show that the maximum heat transfer in twisted tape was with the bigger holes diameter, more space between the holes and smaller twist tape ratio than plain tube and the typical tape inserts was 27.4% and 86.7%, respectively. Ray and Date [6] predicted the heat transfer in a square sectioned duct fitted with twisted tape in both laminar and turbulent flows. The heat transfer behaviours were predicted under axially and peripherally constant wall heat flux conditions. Patil [7] studied the friction and heat transfer characteristics of laminar swirl flow pseudo plastic type power law fluid in a circular tube using varying width full-length twisted tape under uniform wall temperature conditions. Nagarajan and Sivashanmugam [15] carried out CFD analysis on the heat transfer in a circular tube with right left helical twisted tape with spacer inserts. The half-length twisted tape equipped with U-bend double pipe heat exchanger for heat transfer and pressure drop were analysed by Yadav [16]. The enhancement of heat transfer coefficient was 40% higher than typical heat exchanger and thermal performances of full length twisted tape were better than half-length twisted tape. Hejazi *et al.* [17] experimentally investigated the heat transfer and the pressure drop for a tube with twisted tape inserts of twist ratio 6, 9, 12 and 15. The best thermal performance and heat transfer enhancement has been achieved with twisted tape ratio 6 and 9. Guo *et al.* [18] carried out an experimental study for the enhancement of heat transfer and friction factor for centre cleared and short width twisted tape in laminar flows and found that central cleared twisted tape have better performance. Ferroni *et al.* [19] experimentally investigated the isothermal pressure drop with separated, multiple, short length twisted tape inserts in the horizontal round tube. Their results indicate that pressure drop with full length twisted tape were 50% higher than multiple short length twisted tapes. Jaisankar *et al.* [20] investigated experimentally solar water heater fitted with rods and spacer at the trailing edge of the twisted tape for different length and twist ratios system to analyse the heat transfer and friction factor characteristic. Enhancement of heat transfer coefficient reduced by 17% and 29% for twisted tape with rod and spacer, respectively as compared with full length twisted tape.

This paper presents the mathematical models to investigate analytically the thermal performance of parabolic trough collector (PTC) with oblique delta-winglet twisted tape insert in the absorber tube for therminol VP-1fluid. The characteristics of heat transfer and thermal performance factors in the tube fitted with plain tube and oblique delta winglet twisted tape of different twist ratio are compared. The effect of mass flow rate on collector efficiency factor, collector heat removal factor, useful heat gain, efficiency and exergy efficiency have also been presented in this paper.

## 2. THEORETICAL ANALYSIS

A cylindrical parabolic concentrating collector whose concentrator has an aperture ‘$W_a$’ and length ‘$L_c$’ and rim angle ‘$\phi_{rim}$’ have been considered and as shown in Figure 1. The absorber tube has an inner diameter $d_i$ and outer diameter $d_o$ and it has a concentric glass cover of inner and outer diameters $d_{ci}$ and $d_{co}$, respectively. The fluid being heated in the collector has a mass flow rate $\dot{m}$, inlet temperature $\theta_{fi}$ and an outlet temperature $\theta_{fo}$.

The energy equations under steady state condition can be describe by the following expression for an element of thickness dx of the absorber tube, at distance ‘x’ from inlet,

$$dq_u = [I_b r_b (W_a - d_o)\rho\gamma(\tau\alpha)_b + I_b r_b d_o (\tau\alpha)_b - U_L \pi d_o (\theta_p - \theta_a)]dx \quad (1)$$

The left side term in Equation 1 represents the useful heat gain rate, the first term on right side represents the incident beam radiation absorbed in the absorber tube after reflection, while the second term represents the absorbed incident beam radiation which falls directly on the absorber tube and the third term represents the loss by convection and re-radiation.

The absorbed flux ‘S’ as follows:

$$S = I_b r_b \rho\gamma(\tau\alpha)_b + I_b r_b (\tau\alpha)_b \left(\frac{d_o}{W_a - d_o}\right) \quad (2)$$

Equation 1 thus becomes:

$$dq_u = (W_a - d_o)\left[S - \frac{U_L(\theta_p - \theta_a)}{C_R}\right]dx \quad (3)$$

where $C_R$ is the concentration ratio of the collector, it is the ratio of effective aperture area to the absorber tube area as

$$C_R = \frac{(W_a - d_o)L_c}{(\pi d_o)L_c} = \frac{W_a - d_o}{\pi d_o} \quad (4)$$

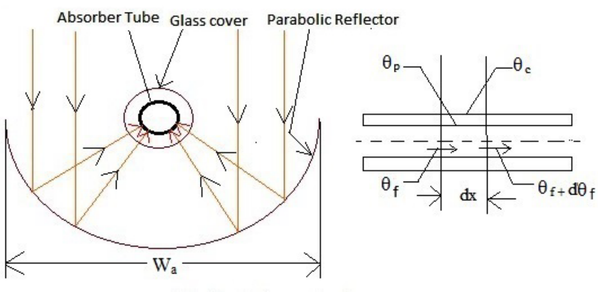


**Fig. 1. Parabolic trough collector.**

The useful heat gain rate $dq_u$ can be written as:

$$dq_u = h_f \pi d_i (\theta_p - \theta_f)dx, \quad (5)$$

where, $h_f$ and $\theta_f$ are heat transfer coefficient on the inside surface of the tube and local fluid temperature. Combine the Equations 3 and 5 and eliminate the absorber tube temperature ( $\theta_p$), the $dq_u$ can be expressed as:

$$dq_u = F_c'\left[S + (\theta_a - \theta_f)\left(\frac{U_L}{C_R}\right)\right](W_a - d_o)dx \tag{6}$$

where F′is the collector efficiency factor defined as:

$$F_c' = \frac{1}{U_L\left[\frac{1}{U_L}+\frac{d_o}{h_f d_i}\right]} \tag{7}$$

Combining Equations 2 and 5, we obtain the differential equation as:

$$\frac{d\theta_f}{dx} = \frac{F_c' U_L \pi d_o}{\dot{m}C_p}\left[\frac{C_R S}{U_L} - (\theta_f - \theta_a)\right] \tag{8}$$

Integrating and using the boundary condition at x= 0, $\theta_f = \theta_{fi}$ we have the temperature distribution:

$$\frac{\theta_f - \left(\frac{C_R S}{U_L} + \theta_a\right)}{\theta_{fi} - \left(\frac{C_R S}{U_L} + \theta_a\right)} = \exp\left[\frac{-F_c' U_L(\pi d_o)x}{\dot{m}C_p}\right] \tag{9}$$

The fluid outlet temperature is obtained by putting $\theta_f = \theta_{fo}$ and $x = L_c$ in Equation 9. Making this substitution and subtracting both sides of the resulting equation from unity, we have:

$$\left[\frac{\theta_{fo} - \theta_{fi}}{\frac{C_R S}{U_L} + \theta_a + \theta_{fi}}\right] = 1 - \exp\left[\frac{-F_c' U_L(\pi d_o)L_c}{\dot{m}C_p}\right] \tag{10}$$

Thus the useful heat gain rate as:

$$q_u = F_R(W_a - d_o)L_c\left[S - \frac{U_L}{C_R}(\theta_{fi} - \theta_a)\right] \tag{11}$$

Where $F_R$ is the heat removal factor define by:

$$F_R = \frac{\dot{m}C_p}{U_L(\pi d_o)L_c}\left[1 - \exp\left\{\frac{-F_c' U_L(\pi d_o)L_c}{\dot{m}C_p}\right\}\right] \tag{12}$$

The instantaneous collection efficiency $\eta_i$ is given as:

$$\eta_i = \frac{q_u}{(I_b r_b + I_d r_d)W_a L_c} \tag{13}$$

If ground reflected radiation is neglected. The instantaneous collection efficiency can be calculated on the basis of beam radiation alone, is given by:

$$\eta_{ib} = \frac{q_u}{(I_b r_b)W_a L_c} \tag{14}$$

### 2.1 Overall Loss Coefficient and Heat Transfer Correlations

For calculating the overall coefficient $U_L$, the correlations are required for calculating individual heat transfer coefficients. The heat loss rate per unit length can be expressed as [8]:

$$\frac{q_l}{L_c} = h_{P-c}(\theta_{Pm} - \theta_c)\pi d_o + \frac{\sigma\pi d_o(\theta_{pm}^4 - \theta_c^4)}{\left\{\frac{1}{\varepsilon_p} + \frac{d_o}{d_{ci}}\left(\frac{1}{\varepsilon_c} - 1\right)\right\}} \tag{15}$$

$$= h_{P-c}\pi d_o(\theta_c - \theta_a) + \sigma\pi d_{co}\varepsilon_c(\theta_c^4 - \theta_a^4) \tag{16}$$

Equations 15 and 16 are set of two non-liner equations which have to be solved for the unknowns $\frac{q_l}{L_c}$ and $\theta_c$ after substituting the values of $h_{p-c}$ and $h_w$.

### 2.2 Heat Transfer Coefficient between the Absorber Tube and the Cover

The natural convection heat transfer coefficient $h_{p-c}$ for the enclosed annular space between a horizontal absorber tube and a concentric cover is calculated using a correlation by Raithby and Holland [13]:

$$\frac{k_{eff}}{k} = 0.317(Ra^*)^{1/4} \tag{17}$$

$$(Ra^*)^{1/4} = \frac{\ln\frac{d_{ci}}{d_o}}{b^{3/4}\left(\frac{1}{d_o^{3/5}} + \frac{1}{d_{ci}^{3/5}}\right)^{5/4}}(Ra)^{1/4} \tag{18}$$

The characteristic dimension used for the calculation of the Rayleigh number is the radial gap, b = $(d_{ci}-d_o)/2$. Properties are evaluated at the mean temperature $(\theta_{pm}+\theta_c)/2$. It can be noted that the effective thermal conductivity $k_{eff}$ can't be less than thermal conductivity k. Hence $\frac{k_{eff}}{k}$ is put equal to unity if the Equation 18 yields a value less than unity.

The relation between the heat transfer coefficient $h_{p-c}$ and the heat exchange rate per unit length can be expressed as:

$$\frac{2\pi k_{eff}}{\ln\frac{d_{ci}}{d_o}}(\theta_{pm} - \theta_c) = h_{p-c}(\theta_{pm} - \theta_c)\pi d_o \tag{19}$$

Thus, $h_{p-c} = \frac{2k_{eff}}{d_o \ln\frac{d_{ci}}{d_o}}$

The limitations on using Equation 17 are that $Ra^*$ should less than $10^7$ and b should be less than $0.3d_o$.

### 2.3 Heat Transfer Coefficient on the Outside Surface of the Cover

The convective heat transfer coefficient $h_w$ on the outside of the cover has been calculated using Hilpert [11] correlation and is given in Equation 20:

$$Nu = C_1 Re^n \tag{20}$$

Where $C_1$ and n are constant having the following values:

For 40< Re <4000, $C_1$ =0.615, n =0.466
For 4000< Re <40000, $C_1$ =0.174, n =0.618

For 40000< Re <40000, $C_1$ =0.0239, n =0.805.

Churchill and Bernstein [12] have made a comprehensive analysis of the data available for cross flow across a cylinder and developed the following correlation;

$$Nu = 0.3 + \frac{0.62Re^{1/2}Pr^{1/3}}{[1+(0.4/Pr)^{2/3}]^{1/4}}\left[1+\left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5} \tag{21}$$

Equation 21 is valid for all values of Reynolds number up to $10^7$. For the range 20000< Re < 400000, Churchill and Bernstein recommended that the last term $[1+(\frac{Re}{282000})^{5/8}]^{4/5}$ be modified to $[1+(\frac{Re}{282000})^{0.5}]$. The $d_{co}$ is characteristic dimension to be used in Equations 20 and 21. Properties are evaluated at the mean temperature $(\theta_c+\theta_a)/2$.

Equations 20 and 21 have been obtained for the cross flow and at low level of turbulence intensity. In practice, the flow may not be right angles and the turbulence intensity in the wind may not be insignificant. As a result, there is an uncertainty in the value of $h_w$ predicted by Equations 20 and 21. Fortunately, this uncertainty does not affect the value of overall loss coefficient significantly.

### *2.4 Heat Transfer Coefficient on the Inside Surface of the Absorber Tube*

The convective heat transfer coefficient ($h_f$) on the inside surface of the absorber tube can be calculated under the assumption that the flow is fully developed. For a Reynolds number less than 2000, the flow is laminar and the heat transfer coefficient may be calculated from equation.

$$Nu = 1.86(RePr)^{0.33}(d_i/L_c)^{0.33} \tag{22}$$

On the other hand, for a Reynolds number greater than 2000, the flow is turbulent and heat transfer coefficient may be calculated from the well-known Dittus-Boelter equation [9]

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{23}$$

The characteristic dimension used for calculating Nusselt number and Reynolds number in Equations 22 and 23 is $d_i$. Properties are evaluated at the mean temperature $(\theta_{fi}+\theta_{fo})/2$.

To enhance the heat transfer, it is desirable to use some kind of augmentative technique to increase the heat transfer coefficient. One of the effective technique is to enhance heat transfer rate by using an oblique delta winglet twisted tape of width $d_i$ inserts all along the inside absorber tube as shown in Figure 2. Eiamsa-ard *et al.* [10] derive the generalized correlations to evaluate the Nusselt number in the tube with oblique delta winglet twisted tape inserts as:

$$NU = 0.18Re^{0.67}Pr^{0.4}X^{-0.423}\left(1+\frac{d}{w}\right)^{0.982} \tag{24}$$

Where X = tape twist ratio = $H/d_i$ and H= length over which the tape is twisted through $180^o$.

**Fig. 2. Oblique delta winglet twisted tape.**

### *2.5 Analysis of the Parabolic Trough Collector Based on the Second Law of Thermodynamics*

In this subsection we developed a model for estimating the Second Law efficiency of the PTC. The set-up is defined by the receiver tube, where different heat transfers occur across the wall-fluid, and the fluid flow. The system comprises dissipative phenomena (or spontaneous non-equilibrium processes) since the natural tendency of system is to achieve equilibrium with their surroundings, and therefore the irreversibilities always occur in the actual process.

The overall entropy generation rate $S_{gen}$ of the PTC can be assessed by considering the simpler form [14]:

$$\dot S_{gen} = \left(\frac{Q_{loss}}{\theta_a} - \frac{Q_*}{\theta_s} + \frac{Q_u}{\theta_{fi}}\right) + \left(\frac{m\Delta P}{\rho\theta_a}\right) \tag{25}$$

Where the first term in parenthesis is due to the heat transfer rate, and the second term is due to the irreversibility caused by the fluid friction. In Equation 25, $\theta_a$ is the ambient temperature and $\theta_s$ is the apparent temperature of the Sun as an exergy source which is of the order of 4500K. The pressure difference is defined as $-\Delta P>0$ since there is a pressure drop between the inlet and outlet of the absorber tube. The PTC has an aperture area, $A_a$, that receives direct solar radiation, $G_B$, at an energy rate from the sun $Q_*$ as it is shown in Equation 26.

$$Q_* = A_a G_B \tag{26}$$

In Equation 25, the useful heat gain $q_u$, is established by Equation 6, the heat transfer $Q_{loss}$ represents the heat loss to the ambient established by:

$$Q_{loss} = Q_* - q_u \tag{27}$$

and the pressure drop ΔP can be calculated by:

$$\Delta P = f\,\frac{4l}{D_i}\,\frac{\rho V_W^2}{2}$$

Where L is the length of the absorber tube, $d_i$ is the internal diameter of the tube, ρ is the density of the fluid, f is the friction factor and $V_w = m/\rho A$ is the velocity of the fluid.

Rearranging Equation 25, the entropy generation rate, $S_{gen}$, of the PTC can be written as:

$$\dot S_{gen} = \frac{1}{\theta_a}\left(Q_*\left(1-\frac{\theta_a}{\theta_s}\right) - Q_u\left(1-\frac{\theta_a}{\theta_{fi}}\right) + f\frac{32m^3L}{\pi^2\rho^2 d_i^5}\right) \tag{28}$$

The correlation of the friction factor for plain tube can be expresses as [14]:

$$f_{pt} = 0.376Re^{-0.259} \quad (29)$$

The friction factor when oblique delta winglet twisted tape is used is calculated as:

$$f_{O-DWT} = 24.8Re^{-0.51}\,(x)^{-0.566}\,(1+z)^{1.87} \quad (30)$$

The exergy supplied via solar energy to the PTC is calculated by:

$$\dot{E}_s = Q_*\left(1-\frac{\theta_a}{\theta_s}\right) \quad (31)$$

some of this exergy supply is destroyed due to irreversible processes. The exergy destruction $E_D$ of the system is calculated by considering its irreversibility, established by the Gouy-Stodola theorem as:

$$E_D = \theta_a \dot{S}_{gen} \quad (32)$$

and refers to the degraded useful energy when real processes are carried out.

The exergy efficiency $\eta_{IIO-DWT}$ is defined as [14]:

$$\eta_{IIO-DWT} = 1-\frac{E_D}{\dot{E}_s} \quad (33)$$

The enhancement factor, $\Delta\eta_{II}$ established by considering the second law of thermodynamics as follows:

$$\Delta\eta_{II} = \frac{\eta_{IIO-DWT}}{\eta_{IIpt}} \quad (34)$$

## 3. RESULTS AND DISCUSSION

In the following section, results of performance, such as Nusselt number, heat transfer coefficient, collector efficiency factor, collector heat removal factor, rise in fluid temperature, thermal efficiency, entropy generation rate and exergy efficiency of the proposed solar parabolic trough collector are presented. For performing the calculation of this study a computer program in C++ language was developed, considering the following system, operating and metrological parameters as given in Table 1. The physical properties including density, the specific heat, the dynamic viscosity and the thermal conductivity of Therminol VP-1 has taken at mean temperature.

**Table 1. Values of system, operating and metrological parameters.**

| | |
|---|---|
| Width of aperture | 3 m |
| Length of absorber | 7.8 m |
| Inner diameter of absorber | 0.0381m |
| Outer diameter of absorber | 0.04315 m |
| Glass cover inner diameter | 0.0560 m |
| Glass cover outer diameter | 0.0630 m |
| Beam radiation | 705 $W/m^2$ |
| Diffuse radiation | 244 $W/m^2$ |
| Intercept factor | 0.95 |
| Absorber tube emissivity | 0.95 |
| Absorber tube absorptivity | 0.95 |
| Transmissivity | 0.85 |
| Mass flow rate | 0.06-0.16kg/s |
| Solar flux | 486.03 $W/m^2$ |
| Tilt factor for beam radiation | 1.0143 |
| Tilt factor for diffuse radiation | 0.993 |
| Slope | $9.625^o$ |
| Declination angle | $9.415^o$ |
| Angle of incidence | $398^o$ |
| Emissivity of glass cover | 0.88 |
| Absorptivity of glass cover | 0.88 |
| Inlet temperature | $120^oC$ |
| Wind velocity | 5.3 m/s |
| Ambient temperature | 31.9 $^oC$ |

### *3.1 Variation of Nusselt Number with Reynolds Number and Heat Transfer Coefficient with Mass Flow Rate for different Tape Twist Ratio*

Figure 3 shows the variation of heat transfer coefficient with Reynolds number and heat transfer coefficient with mass flow rate for different tape twist ratio of oblique delta winglet twisted tape inserts inside the absorber tube of parabolic trough collector. Results indicated that the Nusselt number obtained from the absorber with inserts is higher than plain absorber for all values of Reynolds number. Also by decreasing the tape twist ratio, the value of Nusselt number is increased. It is clearly seen that the effect of tape inserts increased at high Reynolds number due to intensive mixing of fluid which increased the heat transfer rate and high flow velocity. The effect of mass flow rate on heat transfer coefficient with oblique delta winglet twisted tape inserts curves show that as the mass flow rate increases, heat transfer coefficient for plain as well as oblique delta winglet twisted tape inserts absorber tube linearly increases. This is because of increases in mass flow rate, increase the Reynolds number and heat transfer coefficient. It has been found that heat transfer coefficient increases 3.24 times and 2.91 times with use of oblique delta winglet twisted tape insert having x=1 at mass flow rate of 0.06 kg/s and 0.16 kg/s, respectively.

The rate of enhancement of Nusselt number increases with increase in Reynolds number. Figure 4 shows the ratio of Nusselt number of oblique delta winglet twisted tape inserts ($Nu_{O\text{-}DWT}$) to the plain tube ($Nu_{Pt}$) absorber ($Nu_{O\text{-}DWT}/Nu_{Pt}$) as a function of Reynolds number. It has been found that this ratio decreases with increasing Reynolds number and this express the fact that the oblique delta winglet twisted tape is better for low turbulence flow. Increase in Nusselts numbers and heat transfer coefficients at various values of mass flow rate of fluid for different values of tape twist ratio is given in Table 2.

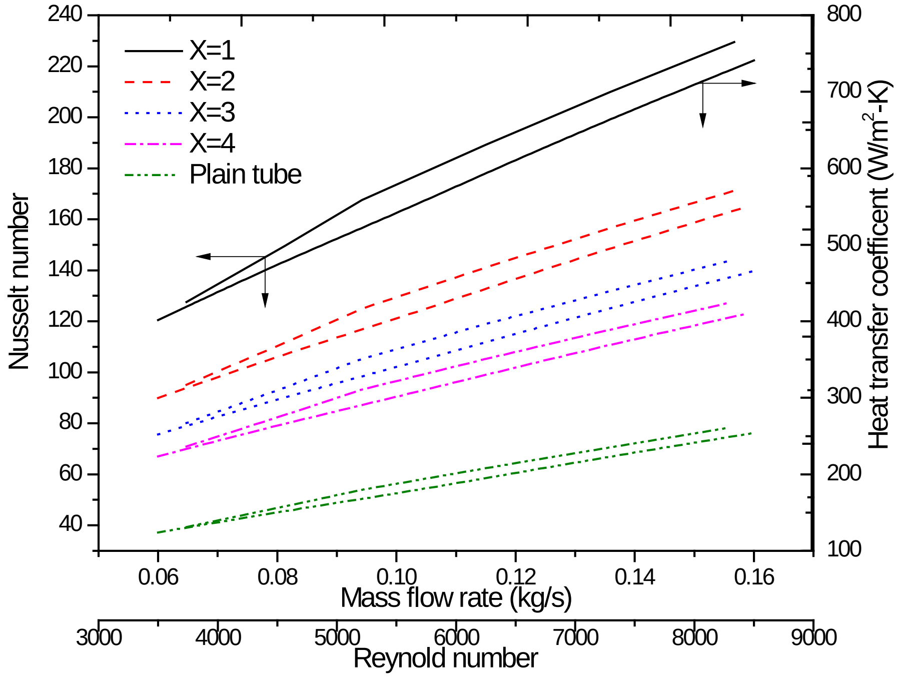


**Fig. 3. Variation of Nusselts number with Reynolds number and heat transfer coefficient with mass flow rate for different tape twist ratio.**

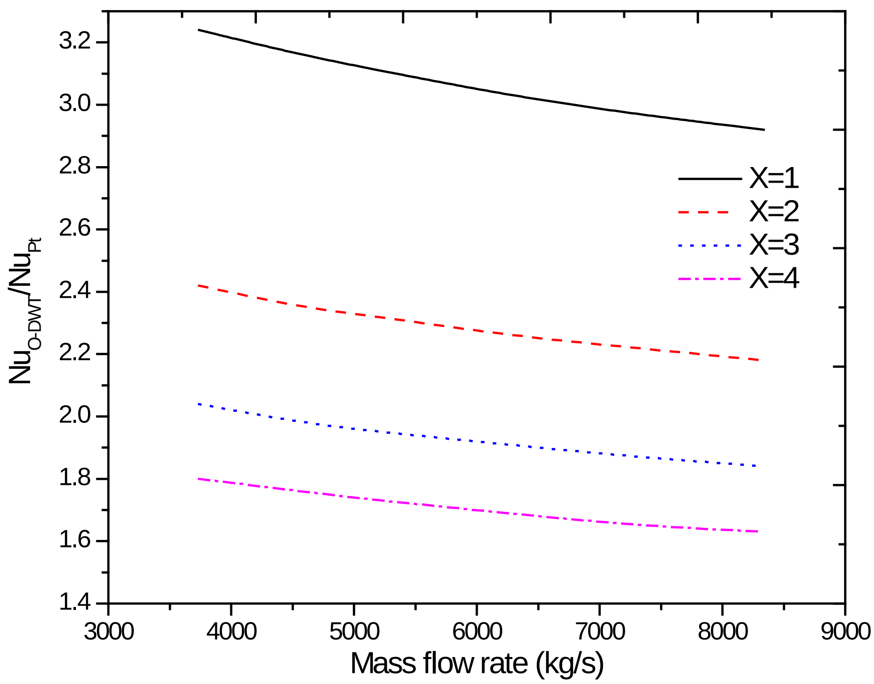


**Fig. 4. Variation of ratio of Nusselts number ($Nu_{O\text{-}DWT}/Nu_{Pt}$) with Reynolds number at different tape twist ratio.**

**Table 2. Nusselt number and heat transfer coefficient with different twisted tape inserts for various values of x and m.**

| ṁ (kg/s) | Plain tube | | Tape Twist ratio | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | X=1 | | | X=2 | | | X=3 | | | X=4 | | |
| | Nu | h | Nu | h | $\varphi^*$ | Nu | h | $\varphi^*$ | Nu | h | $\varphi^*$ | Nu | h | $\varphi^*$ |
| 0.06 | 39.37 | 124.01 | 127.53 | 401.65 | 3.24 | 95.12 | 299.58 | 2.42 | 80.13 | 252.37 | 2.04 | 70.95 | 223.45 | 1.80 |
| 0.08 | 47.43 | 150.64 | 149.68 | 475.36 | 3.15 | 111.64 | 354.56 | 2.35 | 94.05 | 298.67 | 1.98 | 83.27 | 264.45 | 1.76 |
| 0.10 | 53.95 | 174.59 | 167.52 | 540.80 | 3.09 | 124.95 | 403.37 | 2.31 | 105.25 | 339.79 | 1.95 | 93.19 | 300.86 | 1.72 |
| 0.12 | 62.42 | 202.01 | 189.28 | 611.07 | 3.02 | 141.18 | 455.78 | 2.26 | 118.93 | 383.94 | 1.90 | 105.30 | 339.95 | 1.68 |
| 0.14 | 70.61 | 228.52 | 209.88 | 677.55 | 2.96 | 156.54 | 505.37 | 2.21 | 131.87 | 425.72 | 1.86 | 116.75 | 376.94 | 1.65 |
| 0.16 | 78.57 | 254.28 | 229.51 | 740.96 | 2.91 | 171.19 | 552.66 | 2.17 | 144.21 | 465.56 | 1.83 | 127.69 | 412.22 | 1.62 |

$\varphi = Nu_{TT}/Nu_{Pt} = h_{TT}/h_{Pt}$

### *3.2 Effect of Mass Flow Rate on Collector Efficiency Factor and Collector Heat Removal Factor*

Figures 6 and 7 show the collector efficiency factor and collector heat removal factor as a function of mass flow rate of fluid for various values of tape twist ratio of oblique delta winglet twisted tape inserts for $I_b$ = 705W/m$^2$ and $I_d$ =244W/m$^2$. It is seen from the figures that collector efficiency factor and heat removal factor increases with increase in mass flow rate at all values of the tape twist ratio and also for plain tube. This is due to the fact that increases in mass flow rate of fluid increases the convective heat transfer coefficient and consequently increases the collector efficiency factor and collector heat removal factor. It is also observed that, there is a substantial enhancement of these factors due to decrease in the tape twist ratio. This is because of decrease in twist tape ratio enhances the heat transfer rate due to increase in heat transfer coefficient between tube surface and working fluid by generating turbulent swirling flow. The maximum enhancement in the collector efficiency factor and heat removal factor has been found to be 9.86% and 10.09%, respectively, with a tape twist ratio x = 1 as compared to plain tube.

### *3.3 Effect of Mass Flow Rate on Rise in Fluid Temperature at Different Tape Twist Ratio*

Figure 7 shows the variation of the rise in fluid temperature of fluid with function of mass flow rate for various values of tape twist ratio. It is clearly seen from the figure that rise in fluid temperature sharply decreases with increase in mass flow rate and moderately decreases as the mass flow rate increases beyond 0.10 kg/s. The higher mass flow rate show a less steeper fall because of relatively higher rate of energy collection as a result of higher heat transfer rates and relatively lesser thermal losses. Here, again the most efficient tape twist ratio (x=1) shows highest temperature rise under similar operating condition because of maximum rate of energy gain for this collector.

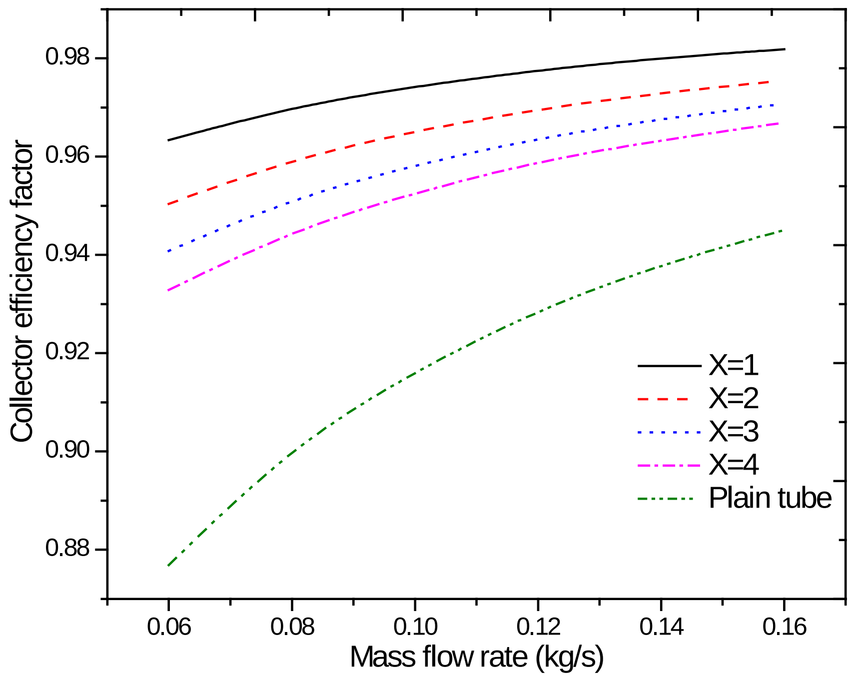


**Fig. 5. Collector efficiency factor as a function of mass flow rate for different tape twist ratio.**

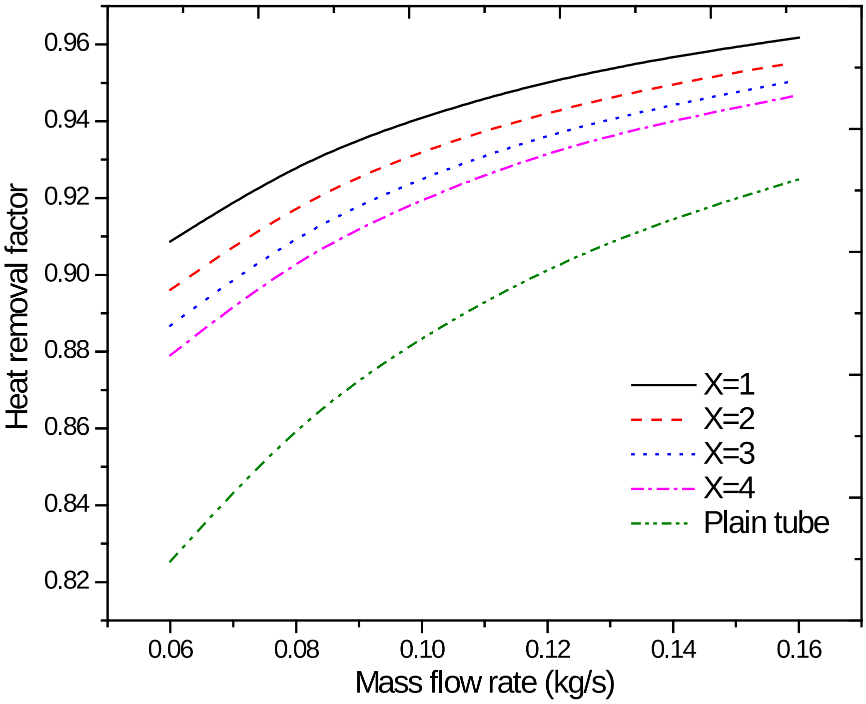


**Fig. 6. Heat removal factor as a function of mass flow rate at different tape twist ratio.**

### *3.4 Effect of Mass Flow Rate of Fluid on Thermal Efficiency for Different Twist Tape Ratio*

The effect of mass flow rate of fluid on collector efficiency has been shown in Figure 8. It is seen that the useful heat gain as well as efficiency increases with increase in mass flow rate. This is due to enhance heat transfer rate at a higher mass rate. It is also seen that useful heat gain and efficiency increases with decrease in tape twist ratio. This is because of heat flow increases between tube surface and working fluid by generating turbulent swirling flow with decreasing values of tape twist ratio. It has been found that tape twist ratio x=1 attains maximum useful heat gain and efficiency in the range of mass flow rate investigated. The percentage enhancement in efficiency with use of the oblique delta winglet twist tape insert is shown in Table 3. Inspections of this table show that there is a considerable enhancement in performance with use of oblique delta winglet twisted tape insert. It has been found that the maximum enhancement in efficiency with tape twist ratio (x=1) are 12.06% and 4.96% at mass flow rate 0.06 kg/s and 0.16 kg/s, respectively as compared to plain tube. It is also found that the enhancement in efficency with tape twisyt ratio (x=1) are 7.71% to 12.06% at lower mass flow rate 0.06 kg/s and 2.91% to 4.96% at higher mass rate 0.16 kg/s, respectively as twist ratio(x) decreased from 4 to1.

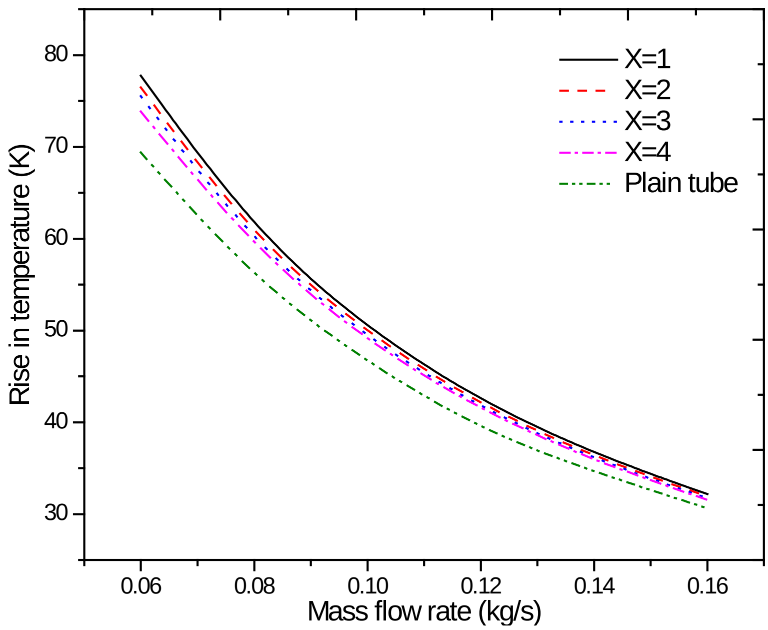


**Fig. 7. Variation of rise in fluid temperature with mass flow rate at different tape twist ratio.**

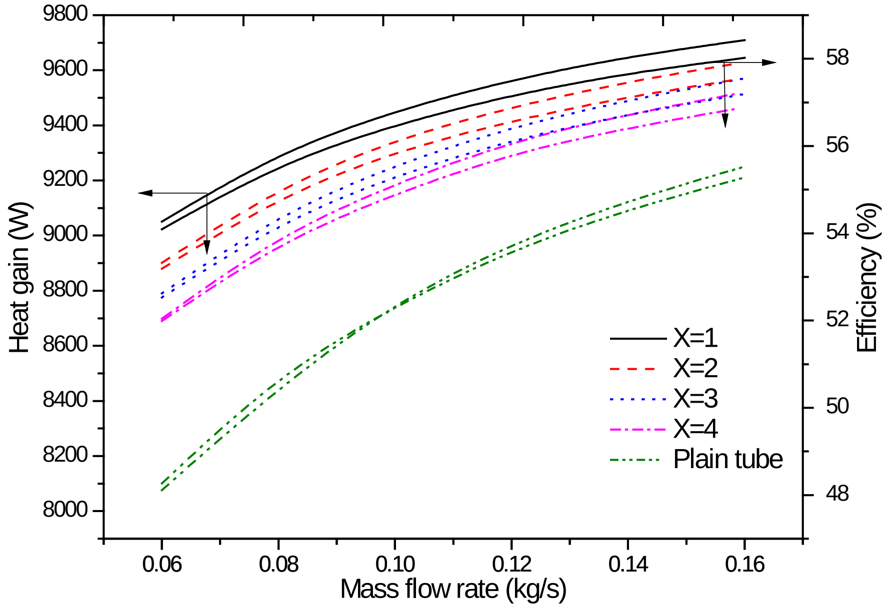


**Fig. 8. Useful heat gain and efficiency as a function of mass flow rate at different twist tape ratio.**

### *3.5 Effect of Mass Flow Rate of Fluid with Oblique Delta Winglet Twisted Tape Insert on Overall Loss Coefficient.*

Figure 9 shows the overall loss coefficient as a function mass flow rate for various value of tape twist ratio. The overall loss coefficient for plain absorber has also been plotted for comparison. The overall loss coefficient for parabolic trough collector with oblique delta winglet twisted tape as well as plain absorber decreases with the increase in mass flow rate. As obvious, increase in mass flow rate decreases the surface temperature of absorber; there by decreases the value of $U_L$. It is also seen from the figure that overall loss coefficient is higher at lower mass flow rate and lower at higher mass flow rate. These types of characteristics behaviour observed for all type of oblique delta winglet twisted tape insert absorbers as well as plain absorber. It may also be noted that rate of decrease of overall loss coefficient is more pronounced at lower mass flow rate as compared to higher mass flow rate. Further, most efficient absorber (x=1) exhibits lowest values of overall loss coefficient in the entire range of mass flow rate investigated.

### *3.6 Effect of Mass Flow Rate of Fluid with Oblique Delta Winglet Twisted Tape Insert on Entropy Generation*

Figure 10 shows the variation of the entropy generation with mass flow rate for different values of tape twist ratio. The entropy generation is calculated from Equation 28. It can be seen from figure that the entropy generations tend to decreases with increasing mass flow rate. For specific mass flow rate at constant inlet temperature, the entropy generation rate increases with decreasing tape twist ratio.

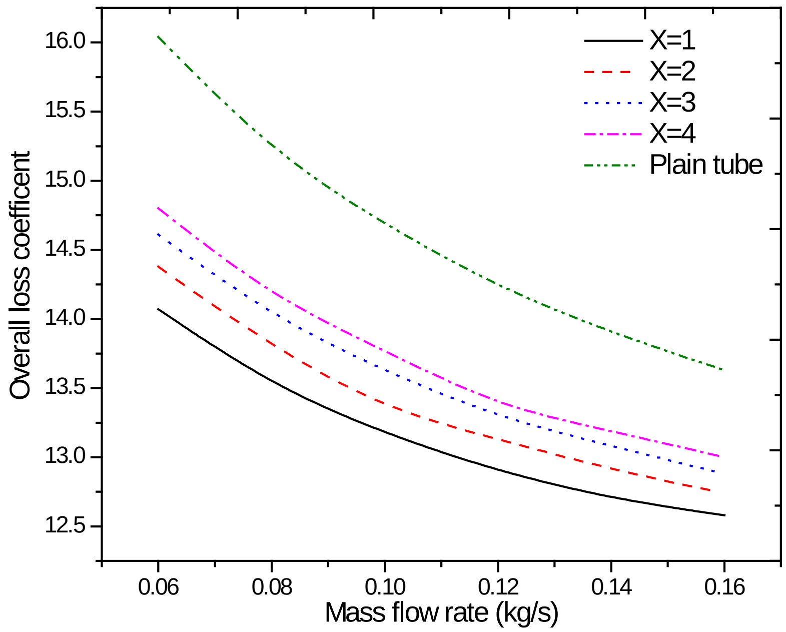


**Fig. 9. Overall loss coefficient a a function of mass flow rate with different twist ratio.**

**Table 3. Enhancement in thermal efficiency (*Φ)* with oblique delta winglet twisted tape for different values of x's and m's.**

| ṁ (kg/s) | Plain tube | Twisted tape X=1 | | X=2 | | X=3 | | X=4 | |
|---|---|---|---|---|---|---|---|---|---|
| | $\eta_{Pt}$ | $\eta_{O\text{-}DWT}$ | Φ | $\eta_{O\text{-}DWT}$ | Φ | $\eta_{O\text{-}DWT}$ | Φ | $\eta_{O\text{-}DWT}$ | Φ |
| 0.06 | 48.27 | 54.09 | 12.05 | 53.19 | 10.19 | 52.53 | 8.82 | 51.99 | 7.71 |
| 0.08 | 50.76 | 55.59 | 9.52 | 54.83 | 8.02 | 54.24 | 6.85 | 53.79 | 5.96 |
| 0.10 | 52.34 | 56.49 | 7.92 | 55.88 | 6.76 | 55.32 | 5.69 | 54.91 | 4.91 |
| 0.12 | 53.62 | 57.17 | 6.62 | 56.58 | 5.52 | 56.14 | 4.69 | 55.83 | 4.12 |
| 0.14 | 54.55 | 57.67 | 5.72 | 57.13 | 4.73 | 56.73 | 3.99 | 56.42 | 3.42 |
| 0.16 | 55.28 | 58.02 | 4.95 | 57.55 | 4.11 | 57.19 | 3.46 | 56.89 | 2.91 |

$Enhancement\ in\ thermal\ efficiency(\Phi) = (\eta_{O-DWT} - \eta_{Pt})/\eta_{Pt}$

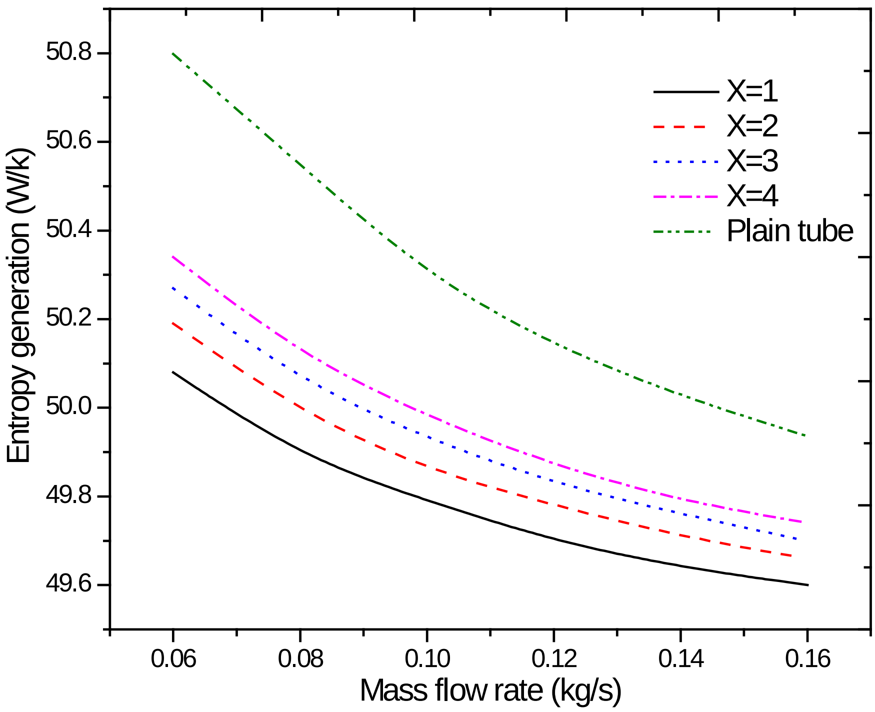


**Fig. 10. Variation of entropy generation with mass flow rate for different values of tape twist ratio.**

### *3.7 Effect of Mass Flow Rate on Exergy Efficiency and Enhancement Factor*

Figure 11 shows the variation of the exergy efficiency and enhancement factor *i.e.* for augmentation of exergy efficiency with mass flow rate for different tape twist ratio. Results of exergy efficiency of plain tube are also plotted for comparison. It is seen from the figure that the exergy efficiency increases and enhancement factor decreases with increase in mass flow rate. It is also observed from the plot that the enhancement in exergy efficiency i.e. enhancement factor is considerably higher at lower mass flow rate and it is lower at higher mass flow rate. This is due fact that at higher mass flow rate the entropy generation decreases. It is also seen that the exergy efficiency and enhancement factor increases with decrease in tape twist ratio for entire mass flow rate investigated. The maximum exergy efficiency with x=1 is observed to be 12.05% and 4.92% at lower and higher mass flow rate of 0.06 kg/s and 0.16 kg/s, respectively and corresponding enhancement factor is 1.121 and 1.049, respectively.

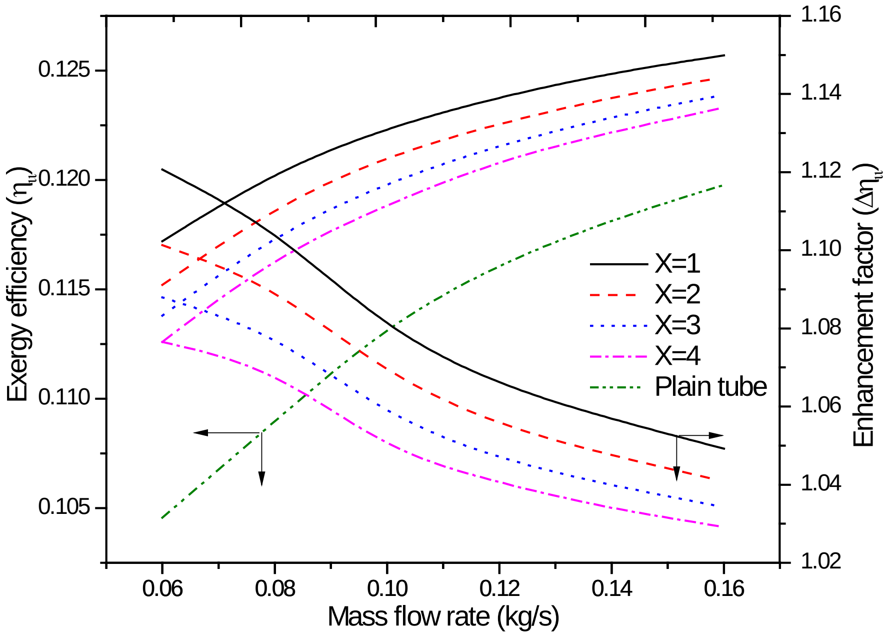


**Fig. 11. Variation of mass flow rate on exergy efficiency and enhancement factor.**

## 4. CONCLUSION

On the basis of analytical investigations, the following conclusions can be drawn:

- The heat transfer equations have been developed in order to analyze the thermal performance of parabolic trough collector with oblique delta winglet twisted tape inserts in the absorber tube.
- A computer program in $C^{++}$ language is developed in order to study the effect of system and operating parameters on thermal performance.
- It has been found that the heat transfer coefficient increases 3.24 and 2.91 times to that of plain tube with use of oblique delta winglet twisted tape inserts (x=1) in the absorber tube of parabolic trough collector at mass flow rate of 0.6 kg/s and 0.16 kg/s, respectively, while corresponding enhancement in thermal efficiency is found to be 12.09% and 4.96%, respectively.
- It has been found that there is a considerable enhancement in performance with use of oblique delta winglet twisted tape insert. The maximum enhancement in efficiency with tape twist ratio (x=1) are 12.06% and 4.96% at mass flow rate 0.06 kg/s and 0.16 kg/s, respectively as compared to plain tube. It is also found that the enhancement in efficency with tape twist ratio (x=1) are 7.71% to 12.06% at lower mass flow rate 0.06 kg/s and 2.91% to 4.96% at higher mass rate 0.16 kg/s, respectively as twist ratio(x) decreased from 4 to1.
- Best performance of PTC is achieved with a low tape twist ratio (x=1) operating at low mass flow rate (0.06 kg/s).
- It has been found that entropy generation decreases with increase in mass flow rate and decease in tape twist ratio.
- The exergy efficiency having tape twist ratio, x=1, at mass flow rate of 0.06kg/s and 0.16kg/s has been found to be 12.05% and 4.92%, respectively while corresponding enhancement factor is found to be 1.121 and 1.049, respectively.

## NOMENCLATURE

| | |
|---|---|
| $A_c$ | surface area of collector, $m^2$ |
| $C_p$ | specific heat, J/kg-k |
| $C_R$ | concentration ratio |
| $dq_u$ | useful heat gain rate |
| $E_D$ | exergy destruction |
| $\dot{E}_s$ | exergy supply |
| $F'_c$ | collector efficiency factor |
| $F_R$ | collector heat removal factor |
| $f_{Pt}$ | friction factor of plain tube |
| $f_{O-DWT}$ | friction factor of oblique delta winglet tape |
| $h_f$ | inside surface heat transfer coefficient, $W/m^2 K$ |
| $h_{p-c}$ | heat transfer coefficient between absorber plate and glass cover, $W/m^2$-K |
| $h_w$ | wind heat transfer coefficient, $W/m^2$-K |
| $I_b$ | incident beam radiation, W/m2 |
| k | thermal conductivity, W/m-K |
| $K_{eff}$ | effective thermal conductivity, W/m-K |
| K | thermal conductivity of fluid, W/m- K |
| H | pitch length of twisted tape, m |
| $L_c$ | length of collector, m |
| $\dot{m}$ | mass flow rate |
| Nu | Nusselt number |
| Pr | Prandtl number |
| $\theta_a$ | ambient temperature, K |
| $\theta_c$ | temperature of cover, K |
| $\theta_f$ | local fluid temperature, K |
| $\theta_{fi}$ | temperature of fluid at inlet, K |
| $\theta_{fo}$ | temperature of fluid at outlet ,K |
| $\theta_s$ | apparent temperature of the sun, K |
| $Q_*$ | solar beam radiation collected by PTC |
| $\theta_p$ | local temperature of absorber tube, K |

| | |
|---|---|
| S | incident solar flux, W/m$^2$ |
| $W_a$ | width of aperture, m |
| X | twist tape ratio |
| Re | Reynolds number |
| Ra | Rayleigh number |
| $r_b$ | tilt factor |
| Ra* | modified Rayleigh number |
| ν | kinematic viscosity, m$^2$/s |
| $\dot{S}_{gen}$ | entropy generation |

*Greek letters*

| | |
|---|---|
| α | absorptivity of absorber |
| $\varepsilon_p$ | emissivity of absorber surface |
| $\varepsilon_c$ | emissivity of cover |
| φ | enhancement in thermal efficiency |
| $\phi_r$ | rim angle |
| ρ | specular reflectivity of concentrated surface |
| σ | Stafan's-Bltzmann constant, W/m$^2$-K$^4$ |
| τ | transmissivity of glass cover |
| $\eta_i$ | instantaneous collection efficiency |
| $\eta_{O\text{-}DWT}$ | thermal efficiency of oblique delta winglet tape |
| $\eta_{Pt}$ | thermal efficiency of plain tube absorber |
| $\eta_{IIO-DWT}$ | exergy efficiency |
| $\Delta\eta_{II}$ | enhancement factor |

*Subscripts*

| | |
|---|---|
| Pt | plain tube |
| O-DWT | oblique delta winglet twisted tape |

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