# 針狀太陽熱集熱器의 性能과 傳熱特性에 關한 硏究

서 정 윤*, 박 남 세**

# A Study on Performance And Heat Transfer Characteristics of Spine Fin Tube Solar Collector.

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**抄　　錄**

이 硏究는 기존의 平板集熱器와 그 型狀이 다른 Spine fin ( 針狀 ) tube 태양열집열기를 製作 使用하여 理論과 實驗을 통해 그 性能과 熱傳達 特性을 규명하고 다른 平板集熱器와 比較하므로써 Spine fin tube 集熱器의 適用可能性을 제시하였다.

또한 이 집열기의 最適 使用條件과 그 設計因子들이 제시되었다.

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## Nomenclatures

$A_c$ : Collector area $m^2$

$C_p$ : Specific heat of fluid Kcal/kg ℃

$d_1$ : Outside diameter of spine fin tube m.

$d_2$ : Outside diamerer of copper tube m.

$d_3$ : Inside diameter of copper tube m.

$G$ : Mass flow rate l/min $m^2$.

$h_1$ : Heat loss coefficient of front side of collector Kcal/$m^2$ h ℃

$h_2$ : Heat loss confficient of back side of collector Kcal/$m^2$ h ℃

$h_3$ : Heat transfer coefficient of fluid Kcal/$m^2$ h℃

$I$ : Rate of solar insolation Kcal/$m^2$h

$K_1$ : Thermal conductivity of aluminum Kcal/m h ℃

$K_2$ : Thermal conductivity of copper Kcal/mh℃

$\ell$ : Fin length m.

$\dot{m}$ : Mass flow rate kg/s

$t$ : Fin thickness for circular direction m.

$T_a$ : Ambient temperature ℃

$T_{in}$ : Inlet temperature ℃

$T_f$ : Local fluid temperature ℃

$T_{fm}$ : Local mean tube temperature with spine fin ℃

$T_p$ : Local fin temperature ℃

$T_{pm}$ : Mean fin temperature ℃

$T_{pm1}$ : Local fin temperature of front side varying with circular angle ℃

$T_{pm2}$ : Mean fin temperature of back side ℃

$T_{pm3}$ : Mean fin temperature of front side ℃

$T_o$ : Contact temperature of fin and tube ℃

$T_{o1}$ : Mean tube temperature of front side with spine fin ℃

$T_{o2}$ : Mean tube temperature of back side with spine fin ℃

$T_t$ : Skin temperature of tube ℃

$T_{t1}$ : Mean tube temperature of front side without fin ℃

$T_{t2}$ : Mean tube temperature of back side without fin ℃

$T_{tm}$ : Mean tube temperature without fin ℃

$T_{out1}$ : Outlet temperature with spine fin ℃

$T_{out2}$ : Outlet temperature without fin ℃

$U_l$ : Collector overall heat loss Kcal/$m^2$ h ℃

$y$ : Correction factor

Greek letters

$\alpha$ : Absorptance of blackened fin and tube

$\beta$ : Concentration faetor of fin

$\tau$ : Transmittance of cover-glass

$\theta$ : Circular angle of tube (deg)

$\theta_o$ : Incident angle of solar radiation (deg)

## 1. Introduction

Various types and special surface treatment for solar collecting appliance have been researched to enhance its efficiency and the Flat plate solar collector has been a major object due to its advantageous fabrication, installation and maintenance.[4)5)6)7)]

YOKOTA et el[1)] investigated the theoretical analysis and experimental measurement on roll-bond type flat plate solar collector. And this study evaluated the correlations 0.5mm thick.

for absorber plate thickness and collector pitch.

This study is aimed to search the heat transfer characteristics and performances of spine fin tube solar collector ( usually used for evaporator and condenser on commercial air-conditioner ) .

The predicted effects by the spine fin tube solar collector are as follow.

1 ) Surface absorptance of solar radiation will be higher than the flat plate solar collector resulting from the much reflectance of solar ray attached by radial fins around the spine fin tube as show in Fig 1-1, Fig 1-2.

2 ) Though the insolation rate would be changed with incident angle on the flat plate collector, the rate of spine fin tube solar collector is scarcely changed.

3 ) As the convection is restrained through the trapped air layer between the fins the collector heat loss could be reduced.

While these advantages are predicted, the collector air packing between the cover-glass and absorber surface must be maintained highly lest water should infiltrate the subtrate of aluminum fins and copper tube because of electric-corrosion.

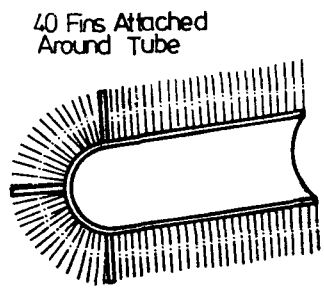


Fig. 1.1 Longitudinal cross section of spine fin tube

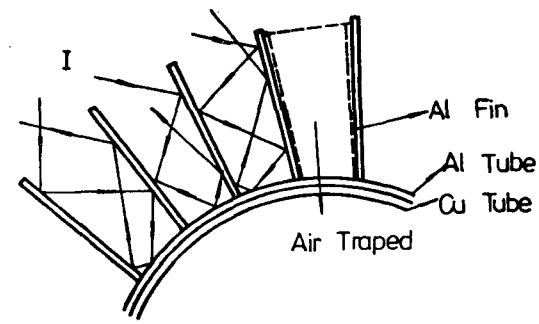


Fig. 1.2 Reflection of insolation on the fin element

## 2. Theoretical Analysis

### 2-1 Conduction of fin

Heat transfer phenomena of the spine fin tube solar collector is shown in Fig 2-1.

Energy balance on an element of the fin thickness dx as shown in Fig 2-2.

$$q_x + q_f - q_{x+dx} = 0 \quad \cdots\cdots\cdots\cdots (1)$$

$$q_x = -k_1 t \frac{dT_p}{dx},$$

$$q_{x+dx} = -k_1 t \frac{d}{dx}\left(T_p + \frac{dT_p}{dx} dx\right)$$

$$q_f = \left[ (\tau\alpha) I \cos\theta - U_l (T_p - T_a) \right] dx$$

Thus we get following equation

$$\frac{d^2\Phi}{dx^2} - m^2\Phi = 0 \quad \cdots\cdots\cdots\cdots (2)$$

where $m^2 = U_l / k_1 t$,

$$\Phi = T_p - \left[ T_a + \frac{(\tau\alpha) I}{U_l} \cos\theta \right]$$

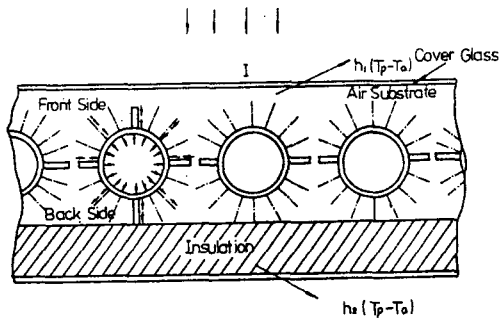


Fig. 2.1 Heat transfer behaviors of solar collector

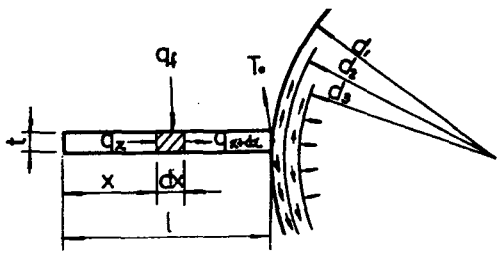


Fig. 2.2 Conduction of fin element

The solution with boundary condition

$\frac{dt_p}{dx}=0$ at $x=0$, $T_p=T_o$ at $x=\ell$)

is

$$T_p=T_o\frac{\cosh(mx)}{\cosh(m\ell)}+\left[T_a+\frac{(\tau\alpha)I}{U_l}\cos\theta\right]\times\left[1-\frac{\cosh(mx)}{\cosh(m\ell)}\right] \cdots\cdots (3)$$

Mean fin temperature $T_{pm1}$ at the front side may be obtained by $U_l=h_1$ in equation

$$T_{pm1}=\int_o^l T_p\,dx/\int_o^l dx$$

$$=T_o\eta_f+(1-\eta_f)+\left[T_a+\frac{(\tau\alpha)I}{h_1}\cos\theta\right]\cdots(4)$$

where, $\eta_f=\tanh(m\ell)/(m\ell)$ called fin efficiency.

At back side of collector, no insolation is received and mean fin temperature $T_{pm2}$ is obtainable by $I=0$ in equation (4)

$$T_{pm2}=T_o\eta_f+(1-\eta_f)T_a \cdots\cdots (5)$$

Since $T_{pm1}$ is varying with circular angle $\theta$, $T_{pm1}$ must be integrated by $\theta$ and rewr itten

$$T_{pm3}=\frac{2}{\pi}\int_o^{\frac{\pi}{2}} T_{pm1}\,d\theta$$

$$=T_o\eta_f+(1-\eta_f)\left[T_a+\frac{2(\tau\alpha)I}{h_1\pi}\right]\cdots (6)$$

Thas the mean fin temperature $T_{pm}$ at the longitudinal arbitrary cross section becomes

$$T_{pm}=\frac{1}{2}(T_{pm2}+T_{pm3})$$

$$=T_o\eta_f+(1-\eta_f)\left[T_a+\frac{(\tau\alpha)I}{h_1\pi}\right]\cdots\cdots (7)$$

## 2-2 Contact temperature $T_O$ and mean tube temperature $T_{fm}$, $T_{tm}$

1) For the case with spine fin

Energy balance on an element of differential volume in Fig.2-3, we consider the radial heat flow only for the unit depth of tube length.

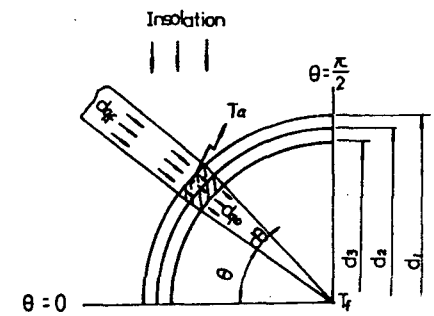


Fig.2.3 Conduction phenomena of fin tube element for differential volume (dashed line)

$$dq_f=dq_w \cdots\cdots (8)$$

$dq_f$ is obtainable from equation (3)

$$dq_f=k_1 t\frac{d_1}{2}d\theta\left.\frac{dT_p}{dx}\right|_{x=\ell}$$

$$=\frac{d_1\ell\eta_f}{2t}\left[(\tau\alpha)I\cos\theta-U_l(T_o-T_a)\right]d\theta \quad (9)$$

And $dq_w$ is expressed from the thermal resistance

$$dq_w = \frac{T_o - T_f}{\frac{\ell_n(r_1/r_2)}{k_1 d\theta} + \frac{\ell_n(r_2/r_3)}{k_2 d\theta} + \frac{1}{\frac{1}{2} h_3 d_3 d\theta}} \quad \cdots (10)$$

Let the denominator of equation (10) as $a_1$ and after some arrangement, contact temperature $T_o$ is obtained

$$T_o = \frac{a_1 d_1 \ell \eta_f [(\tau\alpha) I \cos\theta + h_1 T_a] + 2t T_f}{2t + a_1 d_1 \ell \eta_f h_1} \quad \cdots (11)$$

And because $T_o$ varies with circular angle $\theta$, the mean tube temperature $T_{o1}$ becomes

$$T_{o1} = \frac{2}{\pi} \int_o^{\frac{\pi}{2}} T_o d\theta$$

$$= \frac{a_1 d_1 \ell \eta_f [\frac{2}{\pi} (\tau\alpha) I + h_1 T_a] + 2t T_f}{2t + a_1 d_1 \ell \eta_f h_1} \quad \cdots (12)$$

For the back side, the mean tube temperature $T_{o2}$ become by $I = 0$ and $h_1 = h_2$ in equation (12)

$$T_{o2} = \frac{a_1 d_1 \ell \eta_f h_2 T_a + 2t T_f}{2t + a_1 d_1 \ell \eta_f h_2} \quad \cdots (13)$$

Accordingly at the longitudinal arbitrary section, the mean tube temperature $T_{fm}$ is

$$T_{fm} = \frac{1}{2} (T_{o1} + T_{o2})$$

$$= \frac{1}{2} (A + C) + \frac{(B + D)}{2} T_f \quad \cdots (14)$$

where,

$$A = \frac{a_1 d_1 \ell \eta_f [\frac{2}{\pi} (\tau\alpha) I + h_1 T_a]}{2t + a_1 d_1 \ell \eta_f h_1},$$

$$B = \frac{2t}{2t + a_1 d_1 \ell \eta_f h_1}$$

$$C = \frac{a_1 d_1 \ell \eta_f h_2 T_a}{2t + a_1 d_1 \ell \eta_f h_2}, \quad D = \frac{2t}{2t + a_1 d_1 \ell \eta_f h_2}$$

2) For the case of tube only

Considering on energy balance for the differential volme, $dq_t = dq_w$ ..... (15)

Where, $dq_t$ ; energy incoming, $dq_w$ : energy transfering to the fluid From Fig

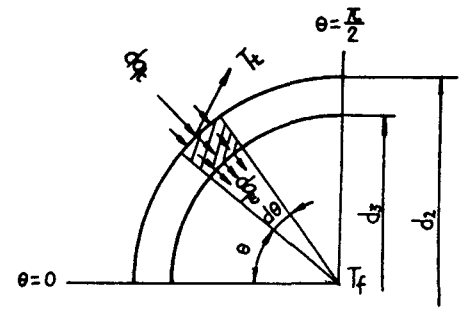


Fig. 2.4 Conduction phenomena of tube element for differential volume ( dashed line )

2-4, $dq_t$ is expressed as

$$dq_t = [(\tau\alpha) I \cos(\theta - \theta_o) - U_l (T_t - T_a)] \times \frac{d_2}{2} d\theta \quad \cdots (16)$$

And $dq_w$ is obtained in same manner of previous case

$$dq_w = \frac{T_t - T_f}{\frac{\ln(r_2/r_3)}{k_2 d\theta} + \frac{1}{h_3 (\frac{d_3}{2}) d\theta}} \quad \cdots (17)$$

let the denominator of equation (17) as $a_2$. Assuming the incident angle $\theta_o = \frac{\pi}{2}$, the local tube temperature on the front side would be written by making $U_l = h_1$

$$T_t = \frac{a_2 d_2 [(\tau\alpha) I \cos(\theta - \frac{\pi}{2}) + h_1 T_a] + 2T_f}{2 + a_2 d_2 h_1} \quad (18)$$

Thus the mean tube temperature $T_{t1}$ on the front side becomes

$$T_{t1} = \frac{2}{\pi} \int_o^{\frac{\pi}{2}} T_t d\theta$$

$$= \frac{a_2 d_2 [\frac{2}{\pi} (\tau a) I + h_1 T_a] + 2T_f}{2 + a_2 d_2 h_1} \quad \cdots (19)$$

For the back side the mean tube temper-

ature $T_{t2}$ is obtained by replacing $I=0$ and $h_1=h_2$ in equation (19)

$$T_{t2}=\frac{a_2d_2h_2T_a+2T_f}{2+a_2d_2h_2} \quad \cdots\cdots (20)$$

Accordingly, the tube temperature at the longitudinal arbitrary section is expressed as

$$T_{tm}=\frac{1}{2}(T_{t1}+T_{t2})$$
$$=\frac{1}{2}(E+G)+\frac{(H+F)}{2}T_f \quad \cdots\cdots (21)$$

Where, $E=\dfrac{a_2d_2[\frac{2}{\pi}(\tau\alpha)I+h_1T_a]}{2+a_2d_2h_1}$

$$G=\frac{a_2d_2h_2T_a}{2+a_2d_2h_2}, \quad F=\frac{2}{2+a_2d_2h_1}$$

$$H=\frac{2}{2+a_2d_2h_2}$$

## 2-3 Outlet temperature of collector

Assuming $dQ_f$ inflowing energy per unit time and $dQ_f$ outflowing energy, the useful energy gain $dQ_n$ becomes

$$dQ_u=dQ_f'-dQ_f\dot{m}c_p(T_f+dT_f)-\dot{m}c_pT_f$$
$$=\dot{m}c_p\,dT_f \quad \cdots\cdots (22)$$

useful energy gain $dQ_u$ is sum of energy from the fin tube $dQ_{fin}$ and the tube without fin $dQ_{tube}$.

$$dQ_u=dQ_{fin}+dQ_{tube} \quad \cdots\cdots (23)$$

1) Effective energy gain $dQ_{fin}$ from the fin

Effective energy gain on the projected area of the spine fin tube at the front side is expressed as

$$dQ_{f1}=y\beta(2\ell+d_1)[(\tau\alpha)I-h_1(T_{pm}-T_a)]\,dZ \quad \cdots\cdots (24)$$

where, $y$ : correction factor,

$\beta$ : concentration factor of fin

$Z$ : unit length of tube

Calculated $\beta=1$ in the case of 20 fins/inch of spine fin tube while $\beta=0$ of without fins.

And for the back side of spine fin tube no insolation was assumed and exist heat radiation only from the fin

$$dQ_{f2}=-y\beta(2\ell+d_1)h_2(T_{pm}-T_a)\,dZ \quad \cdots (25)$$

thus $dQ_{fin}=dQ_{f1}+dQ_{f2}$

$$=y\beta(2\ell+d_1)[(\tau\alpha)I-U_l(T_{pm}-T_a)]dZ \quad (26)$$

2) Effective energy gain $dQ_{tube}$ from the tube only.

From the projected area of the tube, effective energy gain is expressed as

$$dQ_{t1}=d_2(1-\beta)[(\tau\alpha)I-h_1(T_{tm}-T_a)]dZ$$
$$\cdots\cdots (27)$$

For the back side, $I=0$ and $h_1=h_2$ is fitted.

$$dQ_{t2}=-d_2(1-\beta)h_2(T_{tm}-T_a) \quad \cdots\cdots (28)$$

Thus, $dQ_{tube}=dQ_{t1}+dQ_{t2}$

$$=d_2(1-\beta)[(\tau\alpha)I-U_l(T_{tm}-T_a)]dZ \quad \cdots (29)$$

From equation (22), (25), (28), $dQ_u$ is arranged as

$$dQ_u=[(2\ell+d_1)y\beta+d_2(1-\beta)[(\tau\alpha)I+U_lT_a]dZ-U_l\Big](2\ell+d_1)y\beta\Big\{\frac{(A+C)}{2}+\frac{(B+D)}{2}T_f\Big\}\eta_f+Y+d_2(1-\beta)\Big\{\frac{(E+G)}{2}+\frac{(H+F)}{2}T_f\Big\}\Big]dZ \quad \cdots\cdots (30)$$

where, $Y=(1-\eta_f)\left[T_a+\dfrac{(\tau\alpha)I}{h_1\pi}\right]$

Equation (21), (29) is rearranged for about fluid temperature $T_f$

$$\frac{dT_f}{dZ}+JT_f=K \quad \cdots\cdots (31)$$

where $J=\dfrac{U_l}{2\dot{m}c_p}[y\beta\eta_f(2\ell+d_1)(B+D)$

$+d_2(1-\beta)(H+F)]$

$$K=\frac{1}{\dot{m}c_p}[y\beta(2\ell+d_1)+d_2(1-\beta)][(\tau\alpha)I$$

$$+U_l T_a]-\frac{U_l}{2\dot{m}c_p}[y\beta(2\ell+d_1)\{(A+C)\cdot$$

$$\eta_f+2Y\}+d_2(1-\beta)(E+G)]$$

The solution of equation (30) with boandary condition $T_f=T_{in}$ at $Z=0$, $T_f=T_{out}$ at $Z=L$ is obtained

$$T_f=C_1 e^{-jz}+\frac{K}{J} \quad \cdots\cdots (32)$$

Outlet temperature is determined

$$T_{out1}=T_{in}e^{-jz}+\frac{K}{J}(1-e^{-jz})\Big|_{\beta=1} \quad (33)$$

$$T_{out2}=T_{in}e^{-jz}+\frac{K}{J}(1-e^{-jz})\Big|_{\beta=0} \quad \cdots (34)$$

## 2-4 Collector efficiency $\eta$

Collector efficiency $\eta$ is defined as[6]

$$\eta=\frac{Q_u/A_c}{1}=\frac{\dot{m}c_p(T_{out}-T_{in})}{A_c\cdot I} \quad \cdots\cdots (35)$$

where $Q_u$ is calculated from temperature difference of inlet and outlet of the collector. Hence the performance curve is drawned $\eta$ in $y$ coordinate and

$$\left[\frac{(T_{out}+T_{in})}{2}-T_a\right]/I \text{ in } x \text{ coordinate.}$$

## 2-5 Collector loss coefficient $U_l$[2][3]

Collector loss coefficient $U_l$ is quoted by Willam Beckmans emperical formulas.[6]

$$U_l=h_1+h_2$$

$$h_1=\left[\frac{\eta}{C/T_{pm}\left\{\frac{T_{pm}-T_a}{(n+f)}\right\}e}+\frac{1}{h_w}\right]^{-1}$$

$$+\frac{\sigma(T_{pm}+T_a)(T_{pm}^2+T_a^2)}{(\epsilon_p+0.00591\,h_w)^{-1}+\frac{2n+f-1+0.133\,\epsilon_p}{\epsilon_g}-n}$$

$$\cdots\cdots (36)$$

$$h_2=\frac{K}{L} \quad \cdots\cdots (37)$$

where $n$ : number of cover-glass

$f$ : $(1+0.089\,h_w-0.1166h_w)(1+0.07866n)$

$C$ : $520(1-0.000051\beta^2)$ for $0°<\beta<70°$

$e$ : $0.43(1-100/T_{pm})$

$\beta$ : collectors tilt angle (deg)

$\varepsilon_g$ : emittance of cover-glass (0.88)

$\varepsilon_p$ : emittance of absorber plate (black paint 0.95)

$T_{pm}$ : mean temperature of absorber (°K)

$h_w$ : wind factor ($h_w=5.7+3.8v$)

$v$ : wind velocity (m/s)

## 2-6 Heat tranfer cofficient $h_3$

The coefficient of $h_3$ was calculated as below along the fluid velocities of the tube inside.[8]

i) In laminar flow

$$Nu=h_3\cdot d_3/K=4.364 \quad \cdots\cdots (38)$$

ii) In turbulent flow

$$Nu=0.023\,Re^{0.8}Pr^{0.4} \quad \cdots\cdots (39)$$

## 3. Experiment

### 3-1 Experimental apparatus

The experiment for this study was perfo-rmed on May 2 to June 5 in 1983 through the preliminary experiment and collector

located to face due south 1.5m above the roof with tilted angle 45° as shown in Fig 3-1.

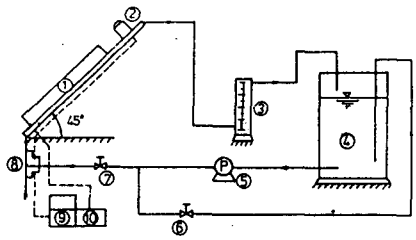


Fig. 3.1 Schematic diagram of experimental apparatus

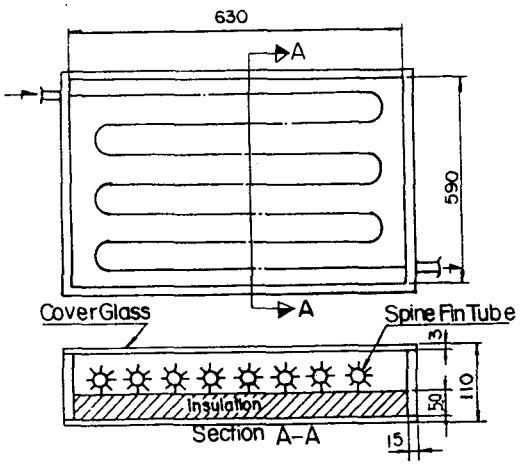


Fig. 3.2 Collector dimensions for experiment

## 3-2 Collector

Two collectors with a series copper tube and spine fin tube respectively were prepared for experiment and their surfaces were coated with acrylic black paint spray.

Each collector was consisted of 15mm thick board, 50mm glass-wool for insulation and 3mm thick common transparent glass for cover-glass.

Detailed specitications are illustrated in Fig 3-2 and Fig 3-3.

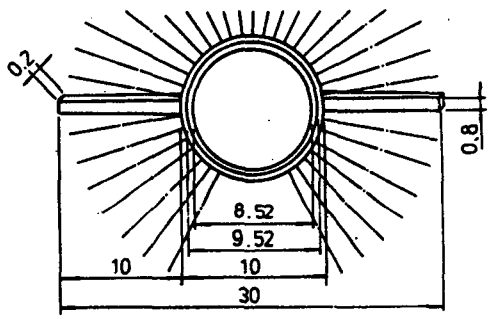


Fig. 3.3 Specification of spine fin

## 4. Discussion

*In the case of tube collector without fin* in Fig 4-1, the result showed average 2℃ of temperature difference at the tube cross section for central part of collector $\theta=90°$ and $\theta=-90°$ respectively, when the insolation rate reached over 600Kcal/m²h to 800 Kcal/m²h, but it was almost same when the rate was below 400Kcal/m²h.

Simillary in the spine fin tube collector *in Fig 4-2*, the temperature difference was resulted in avarage 1℃ below.

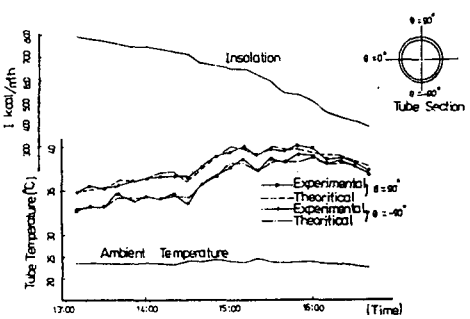


Fig4.1 Comparision of tube temperature to with no fin

This is notable that the circular heat flow around the spine fin tube is occured well compared that of no fin tube because of fin effects as indicated in preceeding

introduction.

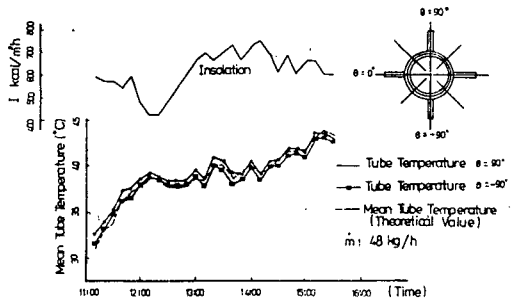


Fig. 4.2 Comparison of mean tube temperature $T_{fm}$ with spine fin ( at central part of collector )

Fig 4-3 represents inlet temperature, outlet temperature and insolation variations when mass flow rate equals 24kg/h with no fin tube collector.

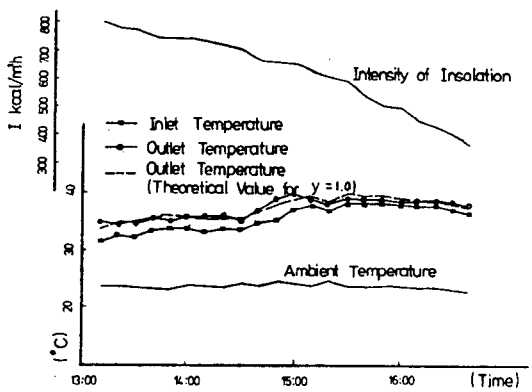


Fig. 4.3 Experimental results for no fin tube solar collector with G=1 l/min m²

In Fig 4-4 plotted as Fig 4-3, when the correction factor $y$ was 1.5 better agreement between the theoretical and experimental values was resulted rather than $y$ was 1.0.

To find out the optimum flow rate range the outlet temperature variation was plotted in accordance with various theoretical flow rates as in Fig 4-5.

The figures 4-6 and 4-7 were presented to observe the correlations between the fin efficiency, length and thickness.

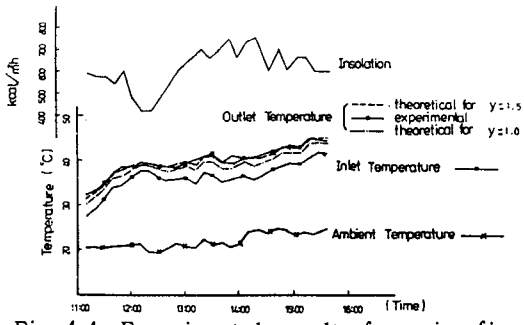


Fig. 4.4 Experimental results for spine fin tube solar collector with G=2.16 l/min m²

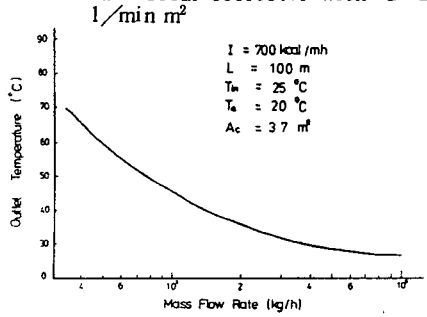


Fig. 4.5 Outlet temperature vs. mass flow rate ( theonetioal )

From these graphs a good fin efficency was resulted when the fin thickness was over 1㎜ but less affected the outlet temperature. In this experiment the 10㎜ long and 0.8㎜ thick fin was used.

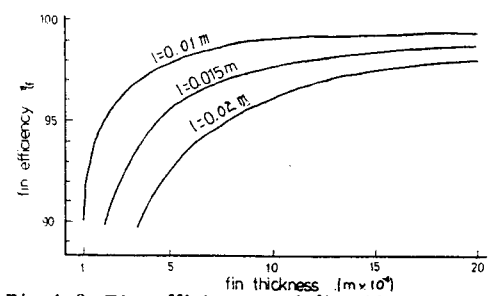


Fig 4.6 Fin efficiency and fin thickness

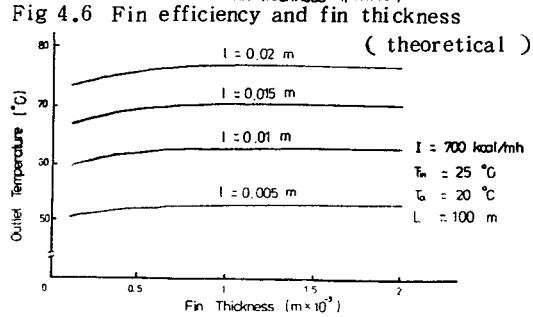


Fig.4.7 Relation between fin thickness(theoretical)

Fig 4-8 represents the experimental results of collectcr efficiencies with spine fin and without fin, and least squares line are fitted to the data.

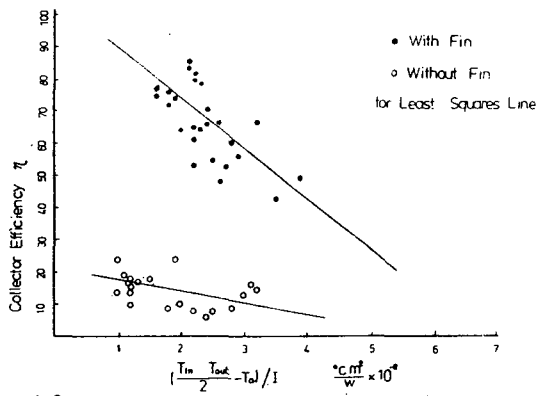


Fig. 4.8 Instantaneous efficiency of collector (experimental result)

From this figure one can see the collector efficiency of the former is higher about 4 to 5 times than the latter as expected naturally, while the efficiency decreasing rate vice versa.

The performance curves (3) and (4) of two collectors in this experiment were contrasted with those Utha university's (1) and (2) in Fig 4-9.

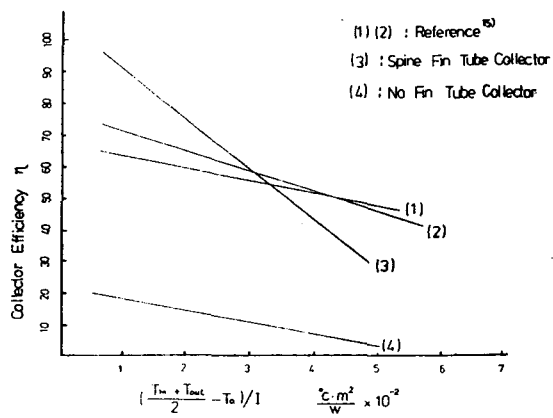


Fig. 4.9 Comparision of instaneous with reference collector [15]

Within the range of $3 \times 10^{-2}$ ℃m²/W in X coordinate, collector (3) prevails in performance over the collector (1) and (2).

More functionally, in turn, under 60℃ of the outlet temperature collector (3) shows rather higher performance than the two collectors (1) and (2).

Though the two collectors specified here were not sufficient in fabrication and black paint surface treatment, these collectors showed a comparatively good results.

Using the speciallized cover-glass and selective surface treatment, it is expected to be excellent performance for the spine fin tube solar collector.

## 5. Conclusions

Test have been performed on two solar collectors with and without spine fins to investigate the heat transfer characteristics through the experiment and theoretical analysis.

And obtained conclusions are as follow.

1) The outlet temperature of the solar collector with spine fins showed a good agreement when correction factor y was 1.5 between the experimental and theoretical results.

2) Theoretically recommended optimum flow rate range is about 0.2~1.4ℓ/min m² in the spine fin tube solar collector when the temperature difference of the inlet and outlet were chosen about 10℃ or more.

3) Fin efficiency was identified to increase linearly with the increase of fin thickness and fin length, but the former did not influence the outlet temperature to an appreciable extent when it was above 0.5 mm thick.

Thus the theoretical values of optimum range is recommended 10~15 mm long and

## References


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