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Full Length Research Paper

An analysis into emission of a plate-fin heat exchanger with strip fins

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The objective of this study is to analyze the heat transfer by radiation in a plate-fin heat exchanger. Strip fins are employed to enhance heat transfer. Compact heat exchanger has been in use for many applications, including automobile, radiators, air-conditioning systems and electronic cooling devices. However, one of the important application of them is in microturbine cycle, where hot stream enter the heat exchanger with temperature of 950°C. So, heat transfer by radiation is important due to high temperature. However, the results show that when there is forced convection in the plate-fin heat exchanger the radiation is negligible, because of low value of the fin height and flow length.

Key words: Radiation, plate-fin heat exchanger, heat transfer coefficient, forced convection.

INTRODUCTION

One of the first papers to treat the radiation mode as the sole means of heat dissipation from the faces of a fin was that of Callinan and Berggren (1959). They considered flat and convex tubes with radiation from one side and fin-and-tube double-surface radiators. The radiation interchange between fin and tube was approximated except that interreflections were not considered for the gray body case, and no account was taken of the incident radiation from the fin on the tube. Even at this early stage in the technology pertaining to the investigation of radiative dissipation from fins, an attempt was made to maximize heat rejection on a per unit mass basic. Reynolds (1963) pointed out that typical fin-and-tube space radiator such as the one shown in Figure 1 would have manifold tubes with manifold lengths depending on the length of the fins. Reynolds (1963) considered that the mass of the system, consisting of the tube, the manifolds, the fluid that the tubes and manifolds contain, and any protective armor, may be such that shorter and thicker fins may be more desirable than fins whose design depend solely on individual fin optimization.

Keller and Holdredge (1966) conducted a numerical solution for the steady-state behavior of the annular (radial) fin of trapezoidal profile and provided charts relating the fin efficiency to other dimensionless groups defined in their analysis. Koshelyaev et al. (1969) calculated the fin efficiency and obtained optimum combinations of thermal and conduction parameters with respect to mass for straight radiating fins on tubes.

Campo and Walko (1973) investigated the conduction-radiation interplay for a longitudinal fin of rectangular profile dissipating heat to the surroundings at a constant equivalent temperature. They illustrated their mathematical scheme for obtaining the heat transferred by radiation from the fins. Delfour et al. (1983) used a finite element method as a first step toward the solution of a minimum mass radiating fin in a satellite application. Chung and Nguyen (1986) provided a general

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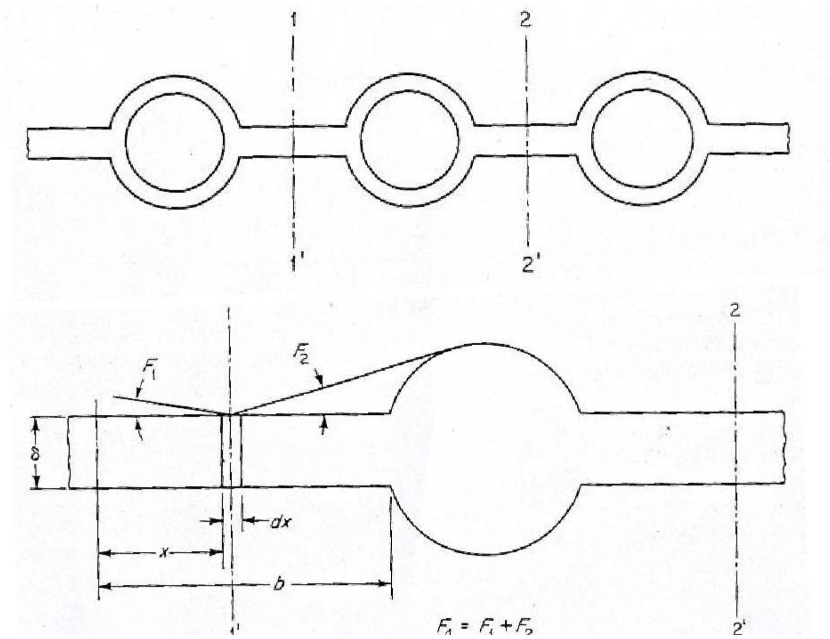

Figure 1. Longitudinal fin-radiator configuration.

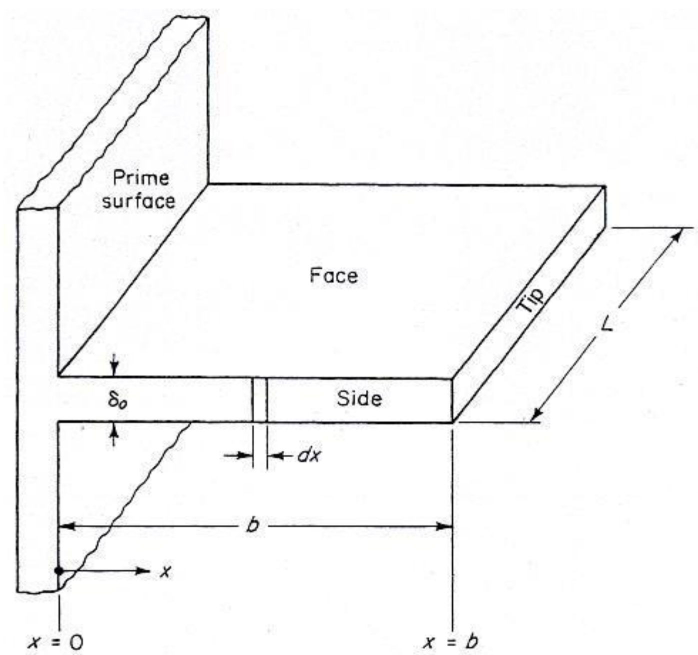

Figure 2. Terminology and coordinate system for a radiating longitudinal fin of rectangular profile.

relationship for the optimized dimensions of longitudinal fins of rectangular, trapezoidal, triangular, and parabolic profile radiating to free space, and Smith et al. (1992) presented a single equation for the profile area of longitudinal fins of rectangular, trapezoidal, and triangular profile, a function of the taper ratio.

Fin radiation analysis

The terminology and coordinate system for the longitudinal fin of rectangular profile are shown in Figure 2.

The radiant heat exchange is between the differential element of fin surface, *Ldx,* and the surroundings. This radiant heat exchange will be composed of two terms,

$$K_1 T^4 LdX \qquad (1)$$

Where the constant K_1 embraces all factors that modify the fin temperature as multipliers and where both sides of the fin dissipate, and:

$$K_2 LdX \qquad (2)$$

Where the constant K_2 consists of all terms that do not multiply the fin temperature and may include solar and/or terrestrial radiation, radiant interchange factors between the fin, and other elements in the configuration and the temperature of the surroundings. The total radiant heat dissipated by the differential fin element will be,

$$dq=\left(K_1T^4-K_2\right)LdX \qquad (3)$$

In accordance with the steady-state heat balance, this heat can be equated to the difference in heat entering and leaving the element *dX* by conduction. Thus,

$$k\delta L\frac{dT^2}{dX^2}dX=\left(K_1T^4-K_2\right)LdX \qquad (4)$$

This equation governs the temperature profile, and its solution can be obtained by successive integration. So,

$$\frac{dT}{dX}=-\left(\frac{2K_1}{5k\delta}T^5-\frac{2K_2}{k\delta}T+C\right)^{0.5} \qquad (5)$$

Where *C* is the constant of integration and where the minus sign assures a temperature gradient that is everywhere negative. The arbitrary constant can be evaluated at *X=b* , where $\frac{dT}{dX}$ is set equal to zero ($\frac{dT}{dX}=0$) and $T=T_a$. Here, after algebraic adjustment,

$$\frac{dT}{dX}=-\left(\frac{2K_1T_a^5}{5k\delta}\right)^{0.5}\left(\frac{T}{T_a}-1\right)^{0.5}\Upsilon \qquad (6)$$

Where,

$$\Upsilon=\left[\left(\frac{T}{T_a}\right)^4+\left(\frac{T}{T_a}\right)^3+\left(\frac{T}{T_a}\right)^2+\left(\frac{T}{T_a}\right)^1+1-\frac{5K_2}{K_1T_a^4}\right]^{0.5} \qquad (7)$$

If it is assumed that $v=\frac{T}{T_a}-1$, the Equation (7) will be,

$$\int_{v_b}^{v_a}\frac{1}{\left[\left(1+v^2\right)^4+\left(1+v^2\right)^3+\left(1+v^2\right)^2+\left(1+v^2\right)^1-\frac{5K_2}{K_1T_a^4}\right]^{0.5}}dv=-b\left(\frac{K_1T_a^3}{10k\delta}\right)^{0.5} \qquad (8)$$

Where the limits are:

$$\begin{cases} v=v_b=(Z-1)^{0.5} & ;\ X=0 \\ v=v_a & ;\ X=b \end{cases} \qquad (9)$$

The heat transferred from the fin faces will be equal to the quantity of heat entering the fin at its base:

$$q_b=k\delta L\left[\frac{2K_1T_a^4}{5k\delta}(Z-1)\left(\Phi-\frac{5K_2}{K_1T_a^4}\right)\right]^{0.5} \qquad (10)$$

Where

$$\Phi=Z^4+Z^3+Z^2+Z^1+1 \qquad (11)$$

correlations to calculate convective coefficient. Based their research convective coefficient is:

$$h=\frac{\text{[redacted]}\,C}{Pr^{\frac{2}{3}}} \qquad (12)$$

Here, *G* is mass velocity and is introduced by,

$$G=\sqrt{\left(\frac{StPr^{2/3}}{f}\right)\left(\frac{\Delta P}{NTU}\right)\frac{\rho_m g_c}{Pr^{2/3}}} \qquad (13)$$

In formula (13), ΔP is pressure drop and can be calculated using some correlations that Kays and London (1984) offered. Here, St and Pr are dimensionless numbers of Stanton and Prandtl respectively. *NTU* is also number of transfer units in heat exchanger. In the Equation (12), J_H [redacted]n factor that is presented using some charts for different types of fins by Kays and London (1984).

A case study

To analyze the heat transfer using radiation, a case study

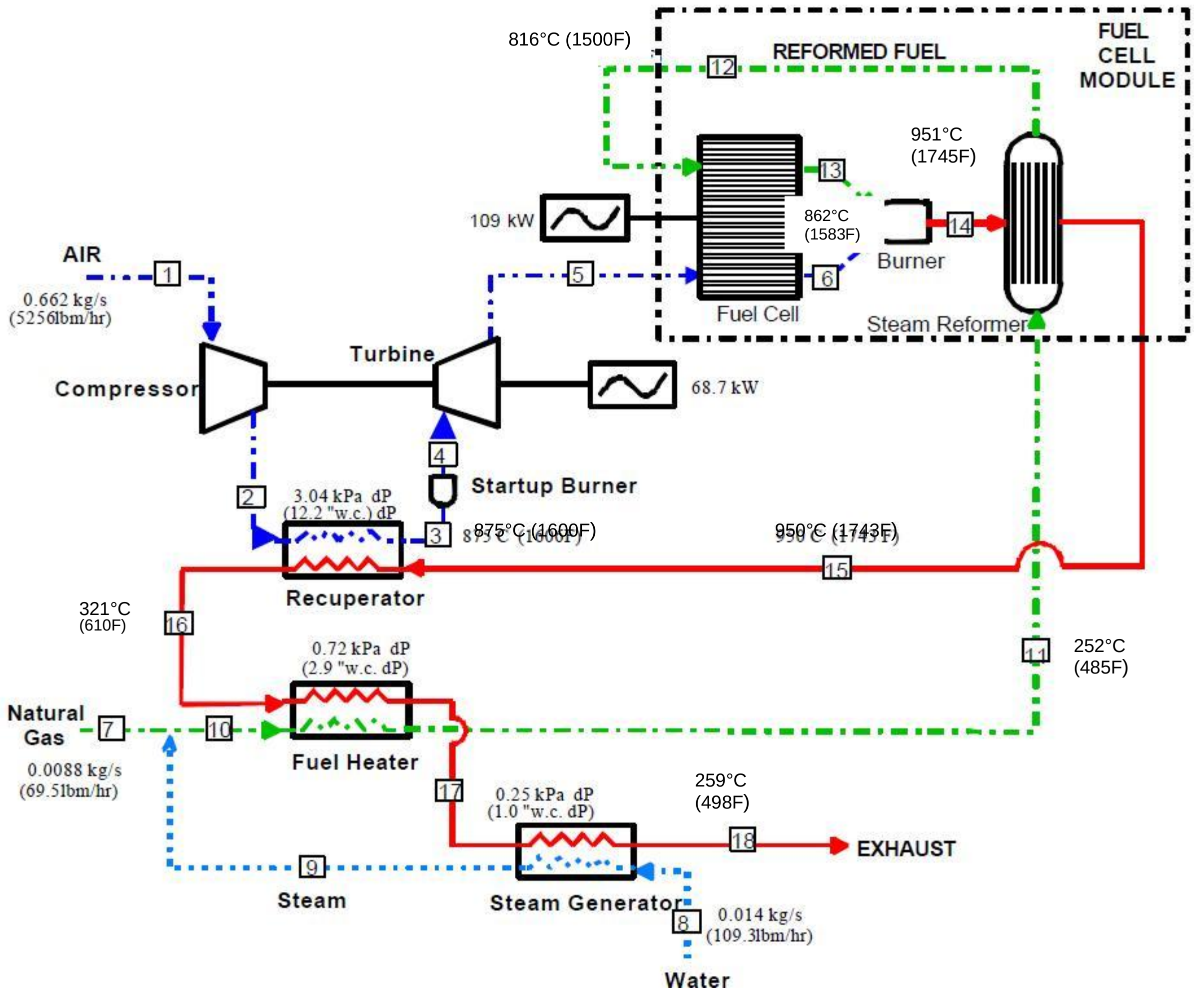

Figure 3. Microturbine cycle (180 kw) (Larry et al., 1990)

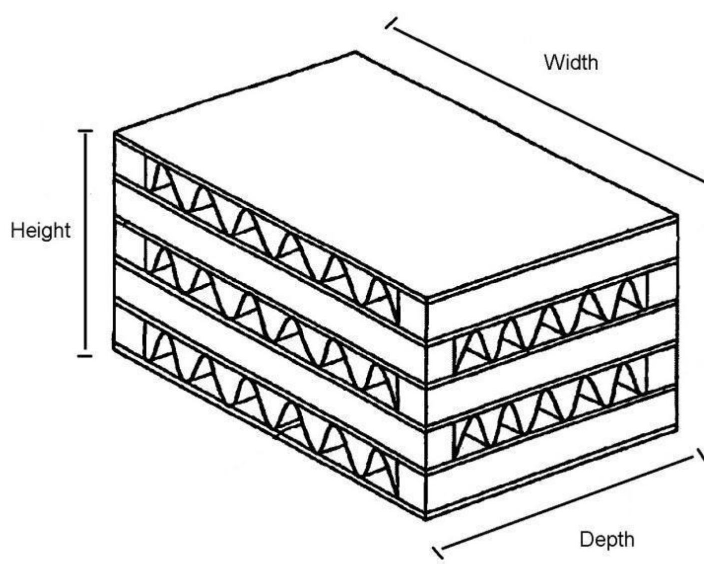

Figure 4. Plate-fin heat exchanger.

Table 1. Fluid properties (Larry et al., 1990).

State Point	Flow (kg/s)	Temperature (C)	Pressure (kPa)	Enthalpy (J/kg)
1	0.662	15	101.3	-1.35e5
2	0.662	178	304	3.11e4
3	0.662	875	304	8.05e5
4	0.662	875	304	8.05e5
5	0.662	642	106.9	5.35e5
6	0.123	862	106.8	7.83e5
7	0.009	15	204.8	-4.74e6
8	0.014	25	120	-1.60e7
9	0.014	108	120	-1.33e7
10	0.023	93	120	-9.95e6
11	0.023	252	120	-9.95e6
12	0.023	816	120	-3.16e6
13	0.036	862	120	-7.57e6
14	0.685	951	106.8	1.33e4
15	0.685	950	106.8	3.19e4
16	0.685	321	106.8	-7.17e5
17	0.685	309	106.8	-7.30e5
18	0.685	259	106.8	-7.86e5

Table 2. Geometric properties of strip fins.

Fin	$D_h\,(mm)$	$b(mm)$	$\beta\left(\frac{m^2}{m^3}\right)$	S_f/S
$\frac{1}{2}-11.94(D)$	2.266	6.02	1512	0.796
$\frac{1}{4}-15.4(D)$	1.605	5.23	2106	0.816
$\frac{1}{6}-12.18(D)$	2.63	8.97	1385	0.847
$\frac{1}{7}-15.75(D)$	2.07	7.72	1726	0.859
$\frac{1}{8}-16.00(D)$	1.862	6.48	1804	0.845
$\frac{1}{8}-16.12(D)$	1.552	5.23	2185	0.823
$\frac{1}{8}-19.82(D)$	1.537	5.21	2231	0.841
$\frac{1}{8}-20.06(D)$	1.491	5.11	2290	0.843

taken from an oil laboratory in Iran, based on Chaney et al. (1999) research, is considered. A gas–to–air counter flow heat-exchanger having heat duty of 180 kw is needed to be designed. Gas and air inlet temperatures are 950 K and 178 K where gas and air mass flow rates are 0.685 kg/ s and 0.662 kg/s respectively. Pressure drops are set to be limited to 6.36 and 12 kpa at hot and cold side. The heat exchanger material is aluminum with density of 2700 $\frac{kg}{m^3}$ .Table of 1 presents the operating conditions used in thermal design.

DISCUSSION

Two broad categories of problem specification are as follows: given the core geometry, the flow rates, and the entering fluid temperatures. The main question in the sizing problem is that what is the size of the core, given the flow rates and their entering and leaving temperatures. These in turn establish the desired heat transfer rate and exchanger effectiveness. In fact, the sizing problem is design problem, while the rating problem is a performance prediction for a specific design. Here, it is assumed that not only are the performance characteristics completely established but also that the general type of heat exchanger has been selected, the flow arrangement has been chosen, and the heat transfer surface configurations for the two fluid sides have been selected. The frontal area for the hot fluid is *HW*. Similarly, the frontal area for cold side is *HD.* The first step after having chosen the two surfaces is to assemble the geometric characteristics of the surface pair (Table 2 offer fin geometric properties): assuming a value for mass velocity (*G*) the mechanical design process will start. Its results are presented in the Tables 3 and 4.

Tables 5 and 6 show the heat transfer using radiation and forced convection in the plate-fin heat exchanger respectively. As it is clear from the tables the heat that transfer by radiation is negligible compared with forced convection. What cause the radiation to be low, despite high temperature in the heat exchanger, is the dimensions of the heat exchanger. One of the important factor in compact heat exchanger is that they have high value of the compactness factor, so the flow length and fin height is low.

Figure 5 denotes the temperature profile of strip fins. One of the main assumptions in this figure is that there is no heat lost from the end and edge of the strip fins. However, Figure 6 shows the temperature trend for parallel and counter flows. As it is evident when we use counter flow in heat exchanger, transferring heat by radiation is negligible because the temperature of hot fluid is very high (950°C), and the outlet temperature of cold fluid is high as well (875°C). The result is that net heat transfer by radiation will be negligible. This trend is approximately the same through the heat exchanger. On the other hands, for parallel flow, transferring heat by

Table 3. Design results.

Fins	Re_h	Re_c	$\Delta p_h\ (Kpa)$	$\Delta p_c\ (Kpa)$	$h_h\left(\frac{W}{m^2.k}\right)$	$h_c\left(\frac{W}{m^2.k}\right)$	U	NTU
$\frac{1}{2}-11.94(D)$	626.85	716.8	2.106	1.341	216.13	242.26	69.45	7.325
$\frac{1}{4}-15.4(D)$	531.84	590.59	2.386	0.989	256.32	281.58	68.43	8.73
$\frac{1}{6}-12.18(D)$	468	520.2	0.719	0.299	212.04	233.45	43.91	6.33
$\frac{1}{7}-15.75(D)$	436.3	484.52	2.814	1.344	288.57	340.97	59.48	9.22
$\frac{1}{8}-16.00(D)$	497.8	553.11	5.345	2.481	318.31	371.87	70.37	9.56
$\frac{1}{8}-16.12(D)$	531.84	590.59	8.669	4.028	276.04	332.78	77.06	10.16
$\frac{1}{8}-19.82(D)$	500.29	555.5	10.206	4.946	387.24	452.66	91.96	12.43
$\frac{1}{8}-20.06(D)$	496.1	551.13	9.486	4.310	376.21	437.18	93.14	12.64

Table 4. Heat exchanger dimensions.

Parameter	$\frac{1}{2}-11.94(D)$	$\frac{1}{4}-15.4(D)$	$\frac{1}{6}-12.18(D)$	$\frac{1}{7}-15.75(D)$	$\frac{1}{8}-16.00(D)$	$\frac{1}{8}-16.12(D)$	$\frac{1}{8}-19.82(D)$	$\frac{1}{8}-20.06(D)$
Width	0.622	0.654	0.728	0.712	0.691	0.613	0.613	0.618
Depth	0.384	0.321	0.451	0.397	0.403	0.416	0.415	0.397
Height	0.912	0.905	0.980	0.983	0.838	0.754	0.751	0.754
Volume	0.218	0.190	0.322	0.278	0.234	0.190	0.189	0.185

Table 5. Heat transfer by radiation.

Parameter	$\frac{1}{2}-11.94(D)$	$\frac{1}{4}-15.4(D)$	$\frac{1}{6}-12.18(D)$	$\frac{1}{7}-15.75(D)$	$\frac{1}{8}-16.00(D)$	$\frac{1}{8}-16.12(D)$	$\frac{1}{8}-19.82(D)$	$\frac{1}{8}-20.06(D)$
Hot side (Watt)	545.64	660.3	428.6	486.96	563.1	618.97	621.3	638.6
Cold side (Watt)	324.46	371.21	309.67	325.19	346.9	417.13	428.43	431.08

Table 6. Heat transfer by forced convection.

Parameter	$\frac{1}{2}-11.94(D)$	$\frac{1}{4}-15.4(D)$	$\frac{1}{6}-12.18(D)$	$\frac{1}{7}-15.75(D)$	$\frac{1}{8}-16.00(D)$	$\frac{1}{8}-16.12(D)$	$\frac{1}{8}-19.82(D)$	$\frac{1}{8}-20.06(D)$
Hot side (Watt)	967.3×10^3	979.2×10^3	954.9×10^3	990.5×10^3	990.5×10^3	1002.8×10^3	1014.6×10^3	1026.6×10^3
Cold side (Watt)	965.5×10^3	978.6×10^3	954.8×10^3	990.6×10^3	990.6×10^3	1002.5×10^3	1014.4×10^3	1026.4×10^3

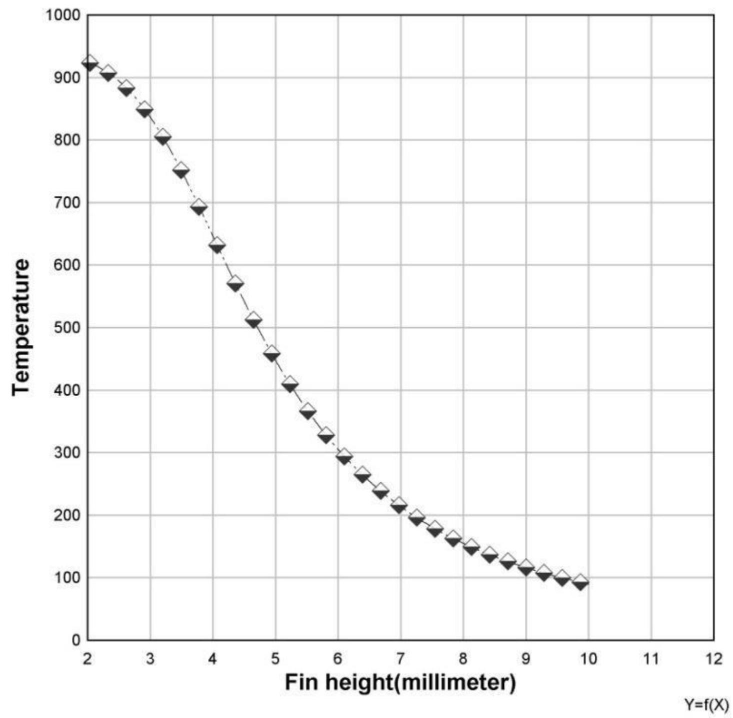

Figure 5. Temperature profile of strip fins for cold stream.

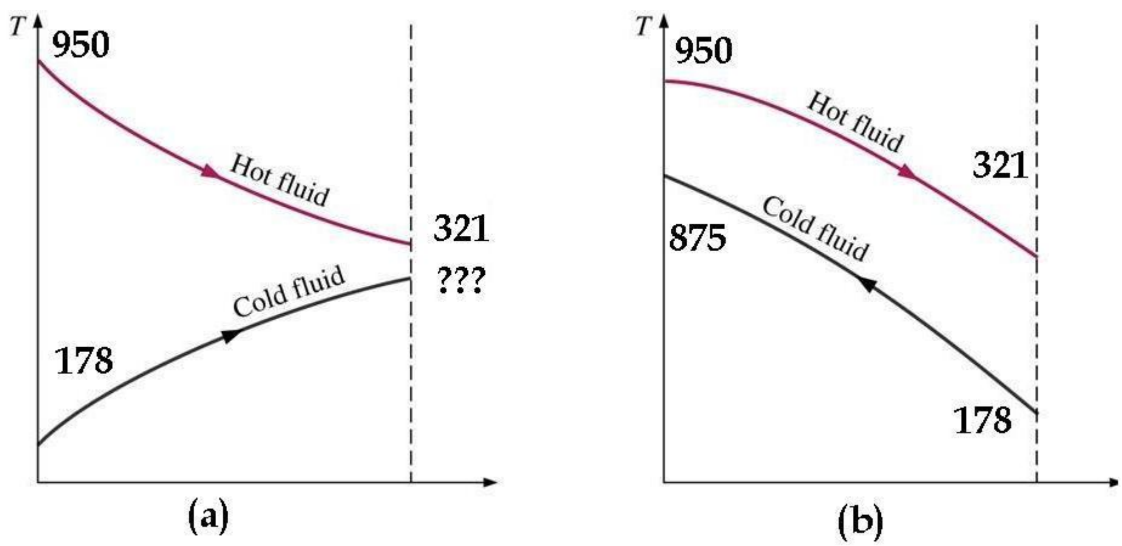

Figure. 6. Temperature profile for (a) Parallel flow (b) Counter flow.

radiation is very important, because the inlet temperature of hot fluid is 950°C while the inlet temperature is 178°C. Of course, by moving through the heat exchanger the amount of heat transfer will decrease as logarithmical function. Consequently, arrangement of heat flow plays a key role in transferring heat by radiation in compact heat exchanger.

Conclusion

Heat exchanger with different types of strip fins was analyzed, and the results, approximately all of the heat transfer from hot stream to cold stream did by forced convection. So, the convection term play an important role in calculating surface temperature. The other result is that when radiation is important that there is natural convection in system.

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