Thermodynamical optimization a plate and frame heat exchanger for microturbine applications

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Abstract: In this study a plate and frame heat exchanger is designed for microturbine applications. In a microturbine cycle, normal efficiency is about 15%, but if heat of outlet gases from microturbine uses to warm outlet air from compressor, total efficiency will up by 30%. So,designing a suitable heat exchanger to transfer this heat from outlet gases to inlet air to heat exchanger is so important, and has intense influence on heat exchanger performance. In this study, two type of heat exchangers are designed, plate and frame heat exchanger and plate-fin compact heat exchanger, and compared in different terms. After evaluating plate and frame heat exchanger, thermodynamical optimization has done to improve its performance. Eventually, after modifying mass flow rates based on thermodynamical optimization, outlet air temperature have increased about 6 C°.

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1. Introduction

The plate and frame heat exchanger was originally introduced in the 1930s and is used extensively in the food industries. Plate and frame heat exchangers consist of several metal sheets with corrugated surfaces that are clamped together [1]. A frontal view of a plate having a herringbone pattern is shown in the Figure of (1).

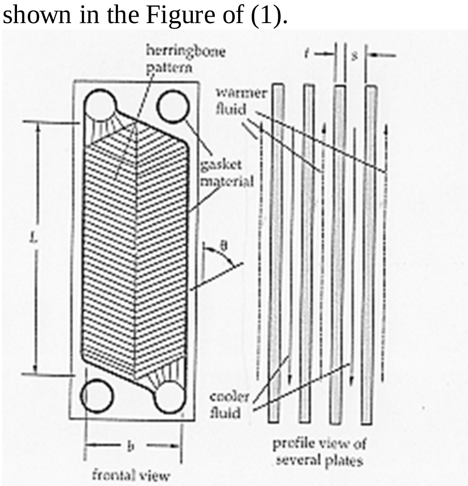

Figure 1. Frontal view of a plate and frame heat exchanger[1]

In the herringbone pattern, the angle made between adjacent ribs and the vertical is called the chevron angle, θ. Performance of heat exchanger is a function of the chevron angle where plates can be made with small chevron angles or large angles. Low-θ plates provide low heat transfer rates with low pressure losses. In addition, the converse is true.

The plates are grouped into passes with each fluid being directed evenly between the paralleled passages in each pass. Whenever the thermal permits, it is desirable to use single pass, counter flow for an extremely efficient performance. Although plate and frame exchangers can accept more than two streams, this is unusual. Two-pass arrangements are, however, common. Figure of (2) demonstrates the flow path in such a unit.

Plates can be made from all pressurable materials. However, where corrosion is a problem, some manufactures offer plate and frame heat exchanger in non-metallic materials, such as a graphite/fluoroplastic composite or a polymer.

Gasket properties have a critical bearing on the capabilities of a plate and frame heat exchanger, in terms of its tolerance to temperature and pressure. Originally, most manufacturers used glue to fix the gaskets to the plates. Several proprietary fixing techniques are available that eliminate the need to use glue, and most manufacturers have adopted these methods. These so-called *glueless* gaskets are suitable for some heavy duty industrial applications.

Nomenclature

$A_\circ$	surface area $\left(m^2\right)$	R_i	ideal gas constant
A_*	dimensionless form of heat transfer area	R	heat resistance
b	plate width	Re	Reynolds number
C_p	specific heat $\left(\frac{J}{kg.k}\right)$	$\dot{S}_{gen}$	entropy generation
C_A	price per unit area $\left(\frac{\$}{m^2}\right)$	S	plate spacing
D_h	hydraulic diameter(mm)	t	plate thickness
E_C	efficiency	$U_\circ$	overall heat transfer coefficient
f	friction factor	V	velocity
F	correction factor	V_t	volumetric flow
G_*	dimensionless form of mass velocity	Greek symbols	
h	convective coefficient $\left(\frac{W}{m^2.k}\right)$	ρ	fluid density $\left(\frac{Kg}{m^3}\right)$
K	thermal conductivity $\left(\frac{W}{m^2.k}\right)$	μ	dynamic viscosity $\left(Pa.s\right)$
K_{el}	electrical energy price $\left(\frac{\$}{MW\ \ h}\right)$	τ	hours per year
L	flow length	θ	chaveron angle
m	mass flow rate $\left(\frac{Kg}{s}\right)$	η	pump/compressor efficiency
N	heat transfer unit	Subscripts	
N_p	plate number	1	input
N_S	entropy generation number	2	output
Nu	Nusselt number	W	warmer fluid
T	temperature of fluids	C	cooler fluid

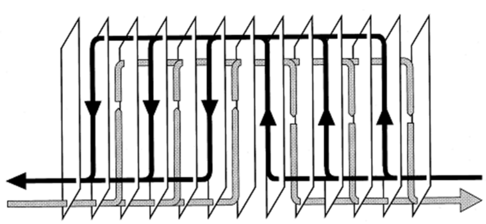

Figure 2.A two-pass Plate and frame arrangement

2. Thermal Analysis

A profile view a plate within an exchanger and the associated resistances to heat transfer is shown in Figure of (3), where for the sake of discussion, the warmer fluid is on the left, and then heat is transferred through the plate to the cooler fluid. The resistances include a convection resistance on the warm side, a conduction resistance through the plate, and a convection resistance on the cooler side [1]. Furthermore, the heat transfer area A_0 is the same as the surface area, and equals plate width b times height L. Hence, the sum the resistance is

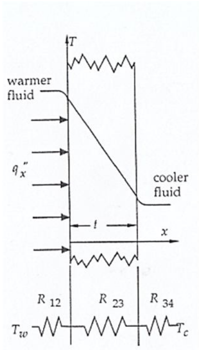

Figure 3. Profile view of one plate and the resistances to heat transfer

$$\Sigma R = R_{1\text{-}2} + R_{2\text{-}3} + R_{3\text{-}4} \tag{1}$$

$$\Sigma R = \frac{1}{h_i A_0} + \frac{t}{k A_e} + \frac{1}{h_0 A_0} \tag{2}$$

Here t is the thickness of the plate, also h_i and h_0 are the convection coefficient between the warmer fluid and the plate, and between cooler fluid and the plate respectively. Although, usually, the temperature drop across a thin walled metal is virtually negligible; but for the plate and frame exchanger, such may not be the case [1]. According to the convection coefficients are usually so high, the conduction resistance is the same order of magnitude as the convection resistances. Consequently, an overall heat transfer is defined by

$$A_0 \Sigma R = \frac{1}{U_0} = \frac{1}{h_i} + \frac{t}{k} + \frac{1}{h_0} \tag{3}$$

Where the overall heat transfer coefficient, U_0, is based on area $A_0 = bL$.

The heat transferred within the heat exchanger equals the product of the overall heat transfer coefficient U_0, the total surface area of N_p plates, which is A_0N_s, and a temperature difference. So

$$\begin{aligned} q &= U_0 A_0 N_p \Delta T = \dot{m}_w\, c_{pw} \left(T_1 - T_2\right) \\ &= \dot{m}_c\, C_{pc} \left(t_2 - t_1\right) \end{aligned} \tag{4}$$

In this formula Δt is considered as the temperature difference.Here, the pure counterflow does not exist, but if the flow through the exchanger is entirely counterflow or parallelflow, then, as a result, Δt would be the log mean temperature difference. So, the plate and frame heat exchanger has features of both flows. Furthermore, the established method of analysis involves the use of the log mean temperature difference for counterflow with a correction factors, F. So

$$F = 1 - 0.0166N \tag{5}$$

Where N is the number of transfer unit and is defined as

$$N = \frac{U_0 A_0 N_p}{(\dot{m} C_p)_{min}} \tag{6}$$

Here $(mC_p)_{min}$ is the minimum mass flow rate-specific heat product for either fluid. Thus, after defining correction factor, F, the temperature difference in the heat balance equation becomes

$$F\left(\text{LMTD}_{\text{counterflow}}\right) = F\frac{(T_1 - T_2) - (t_2 - t_1)}{Ln\ \left[(T_1 - T_2)/(t_2 - t_1)\right]} \tag{7}$$

The flow passage is bounded by the distance between the plates, s, and the plate width, b. So, the hydraulic diameter for a rectangular flow section is

$$D_h = \frac{4sb}{2s + 2b} \tag{8}$$

For a two-dimensional flow passage in which b>>s, the hydraulic diameter would be

$$D_h = 2S \tag{8-a}$$

It is important to notice that the spacing between plates, s, varies typically from 2 to 5 mm.

For laminar flow through a plate and frame heat exchanger, the Nusselt number is

$$Nu = \frac{hD_h}{K_f} = 1.86 \left(\frac{D_h\, Re\, Pr}{L_1}\right)^{1/3} \tag{9}$$

Here, $Re = \dfrac{VD_h}{v}$. In general, one of the most widely used relationships, for turbulent flow is

$$Nu = \frac{hD_h}{K_f} = 0.374\ Re^{0.668} Pr^{1/3} \tag{10}$$

So, h_i and h_0 can be calculated from this equations.

Besides, the pressure drop encountered by the fluid is

$$\Delta P_{\text{plates}} = f\ \frac{L}{D_h} \frac{\rho V^2}{2g_c} \tag{11}$$

In this formula f and L are friction factor and the plate length respectively. The friction factor varies over the range of Reynolds numbers according to the following table (1):

Table 1.Friction factor

Reynolds Number Range	Dorcy – Weisbach Friction Factor
1-10	$F = \dfrac{280}{Re}$
10-100	$F = \dfrac{100}{Re^{0.589}}$
>100	$F = \dfrac{12}{Re^{0.183}}$

Also, there is another loss, where fluids will be entering and exiting the heat exchanger through standard piping connections, and so this loss is associated with these sudden changes in geometry, where it is called port loss. So

$$\Delta P_{port} = 1.3\frac{\rho V_p^2}{2g_c} \quad (11\text{-a})$$

$$\Delta P_t = \Delta P_{plates} + \Delta P_{port} \quad (11\text{-b})$$

For an odd number of plates, the flow velocity between plates is

$$V = \frac{2\dot{m}}{\rho A\ (N_p + 1)} \quad (12)$$

This equation would use to both fluids. However, when the number of plates is even, one of the fluids will have a velocity that is defined by

$$V = \frac{2\dot{m}}{\rho A\, N_p} \quad (12\text{-a})$$

So, the other fluid velocity is

$$V = \frac{2\dot{m}}{\rho A\ (N_p + 2)} \quad (12\text{-b})$$

Typically, inlet temperatures and flow rates would be known and the outlet temperatures must be calculated. So, the outlet temperature of warmer and cooler fluids are respectively:

$$T_2 = \frac{(1-R)T_1 + (1-E_c)Rt_1}{1-RE_c} \quad (13)$$

$$t_2 = t_1 + \frac{T_1 - T_2}{R} \quad (13\text{-a})$$

Here, R and E_c are respectively

$$R = \frac{(\dot{m}C_p)_c}{(\dot{m}C_p)_w} \quad (13\text{-b})$$

$$E_c = \exp\left[\frac{U_\circ A_\circ N_s F}{m_c C_{pc}}\ (R-1)\right] \quad (13\text{-c})$$

3. Thermodynamic Optimization

The irreversibility of any heat exchanger is due to two factors; The transfer of heat across the stream-to-stream temperature difference and the frictional pressure drop that accompanies the circulation of fluid through the apparatus[8]. The fluid friction and heat transfer irreversibility can systematically be reduced by showing down the movement of fluid through the heat exchanger. In other words, this technique is synonymous with employing larger heat exchanger, where there is more heat transfer area and more heat exchanger volume.

Typically, in the caunterflow heat exchanger, the irreversibility due to heat transfer is

$$\dot{S}_{gen} = (\dot{m}C_p)_1 \ln\frac{T_{1,out}}{T_{1,in}} + (\dot{m}C_p)_2 \ln\frac{t_{2,out}}{t_{2,in}} \quad (14)$$

Here, entropy changes associated with the frictional pressure drops $(p_{in} - p_{out})_{1,2}$ have not been included. Also, the entropy generation number is

$$N_s = \frac{\dot{S}_{gen}}{(\dot{m}C_p)_2} \quad (15)$$

$$C = \frac{(\dot{m}C_p)_1}{(\dot{m}C_p)_2} \quad (16)$$

In this formula, subscript 2 is associated with the smaller capacity rate.

In the industry, the heat exchanger irreversibility is caused not only by flow imbalance, but also by an insufficient amount of stream-to-stream heat transfer area plus a set of insufficiently wide flow passage [8]. However, if be considered a balanced counterflow arrangement (C=1), the entropy generation rate in this arrangement is

$$\dot{S}_{gen} = \dot{m}C_p \ln\frac{T_{1,out}}{T_{1,in}} + \dot{m}C_p \ln\frac{t_{2,out}}{t_{2,in}} - \dot{m}R_i \ln\frac{P_{1,out}}{P_{1,in}} + \dot{m}R_i \ln\frac{P_{2,out}}{P_{2,in}} \quad (17)$$

Where the first two terms on the right side refer to the heat transfer irreversibility, and the last two terms consider for fluid friction.

Nevertheless, in actual applications one or more geometric parameters are constrained based on economic considerations. Hence, the minimization of irreversibility subject to constant area is important in cases that the cost of building the heat transfer surface is a major component in the overall cost of the plate and frame heat exchanger.

The heat transfer area for one side by defining of hydraulic diameter is

$$A = \frac{4L}{D_h} A_c \quad (18)$$

Where L and A_c are flow path length and flow cross-section respectively. Also, the dimensionless form of Eq.(18) is

$$G_* = \frac{G}{(2\rho p)^{0.5}} \quad (19)$$

Consequently, when A and Re are fixed, the one-side entropy generation number can be minimized by properly selecting G_*:

$$G_{*,opt} = \left\{\frac{\tau^2}{3A_*^2\left(\frac{R_i}{C_p}\right) f\ st}\right\}^{\frac{1}{4}} \quad (20)$$

$$N_{s,min} = \frac{4}{3^{\frac{3}{4}}} \frac{\tau^{\frac{3}{2}} \left(\frac{R_i}{c_p}\right)^{\frac{1}{4}} f^{\frac{1}{4}}}{A_*^{\frac{1}{2}} \; St^{\frac{3}{4}}} \qquad (21)$$

Where R_i and St are ideal gas constant and Stanton number. τ (for example, for warmer fluid) and G_* are also defined by:

$$\tau = \frac{|T_1 - T_2|}{(T_1 T_2)^{0.5}} \qquad (22)$$

$$G_* = \frac{G}{(2\rho p)^{0.5}} \qquad (23)$$

Here G is considered as mass velocity. Also, G_* is the dimensionless form of it.

4. Results and discussion

To demonstrate procedure, a case study is considered as follows.

It is assumed that the plate and frame heat exchanger is going to employ in microturbine applications. After manual designing the plate and frame heat exchanger, its results have been compared with a plate-fin compact heat exchanger, and then by thermodynamic optimization its performance will be improving.

Input data in order to design a plate and frame heat exchanger are according to below table.

Table 2.Input data to design a plate and frame heat exchanger

Variables	Data
Gas mass flow rate	$1.4676 \frac{kg}{s}$
Gas density	$0.399 \frac{m^3}{kg}$
Gas fluid thermal Conductivity	$43.9 \times 10^{-3} \frac{W}{m^2.k}$
Gas Viscosity	$2.85 \times 10^{-5} \frac{N.s}{m^2}$
Gas Specific heat	$1.58 \frac{kj}{kg.k}$
Prandtl number of gas	0.683
Gas inlet temperature	865 k
Air mass flow rate	$1.45 \frac{kg}{s}$
Air density	$2.934 \frac{m^3}{kg}$
Air fluid thermal conductivity	$0.05 \frac{W}{m^2.k}$
Air viscosity	$3.55 \times 10^{-5} \frac{N.s}{m^2}$
Air specific heat	$1.04 \frac{kj}{kg.k}$
Prandtl number of air	0.735
Plate width	323 mm
Plate height	675 mm
Plate spacing	5.09 mm
Plate thickness	1.015 mm
Number of plates	9
Thermal conductivity of plate	$18.19 \frac{W}{m^2.k}$

So, Table of (3) provides the result of this designing for microturbin applications.

Table 3.Results of designing

Variables	Data
Hydraulic diameter of flow passage	10.18 mm
Plate surface area	0.218 m^2
Flow area	1.61×10^{-3}
Reynolds number for warmer fluid	104.75
Reynolds number for cooler fluid	83.1
Nusselt number for warmer fluid	7.28
Nusselt number for cooler fluid	1.81
Heat transfer coefficient for warmer fluid	$31.39 \frac{W}{m^2.k}$
Heat transfer coefficient for cooler fluid	$8.88 \frac{W}{m^2.k}$
Number of transfer unit	7.16
Overall heat transfer coefficient	$4.99 \frac{W}{m^2.k}$
Outlet air temperature	775 k
Pressure drop for warmer fluid	0.034 kpa
Pressure drop for cooler fluid	0.017 kpa

One of the common heat exchanger in microturbine applications is plate-fin compact heat exchanger. To evaluate the plate and frame heat exchanger designed, a plate-fin compact heat exchanger with Wavy fin is also designed. This type of heat exchanger is assembled from a series of flat sheets and corrugated fins in a sandwich construction. Parting sheets provide the primary heat transfer surface.

Parting sheets are positioned alternatively with the layers of fins in the stack to form the containment between individual layers.

The heat transfer fins provide the secondary heating surface for heat transfer. Fin types, densities and heights can be varied to ensure that exchangers are tailor-made to meet individual customer requirements in term of heat transfer performance versus pressure drop. In this work, the type of Wavy fin for both warmer and cooler fluids is $11.5-\frac{3}{8}(\mathrm{W})$, which its geometric properties is according to Table of (4).

Table 4. Geometric properties of Wavy fin

	b(mm)	D_h(mm)	σ(mm)	$\beta\left(\frac{m^2}{m^3}\right)$	$\frac{S_f}{S}$
$11.5-\frac{3}{8}(W)$	9.25	3.023	0.254	1138	0.822

Here b, D_h and σ are plate spacing, hydraulic diameter and fin metal thickness respectively. β and $\frac{S_f}{S}$ are total heat transfer area/volume between plates and fin area/total area respectively. And eventually, the output data so as to design a compact heat exchanger with wavy fins are:

Table 5. Output data to design a plate-fin compact heat exchanger

Variables	**Data**
Fin pitch	453 per m
Plate spacing (fin height), b	9.25 mm
Flow passage hydraulic diameter, D_h	3.023 mm
Fin metal thickness, δ	0.254 mm
Parting sheet thickness, a	0.152 mm
Total sheet transfer area/volume between plates, $\boldsymbol{\beta}$	1138 $\frac{m^2}{m^3}$
Total heat transfer area/total volume, $\boldsymbol{\alpha}$	525.16 $\frac{m^2}{m^3}$
Fin area/total area, $\frac{S_f}{S}$	0.822
Contraction coefficient, $\boldsymbol{\sigma}$	0.468
Thermal conductivity of fin, k	18.19 $\frac{w}{m^2.k}$
Height of heat exchanger, H	1013mm
Width of heat exchanger, W	700mm
Depth of heat exchanger, D	500mm
Reynolds number of warmer fluid, Re_w	483.014
Reynolds number of cooler fluid, Re_c	449.98
Pressure drop for warmer fluid, ΔP_w	2.085kpa
Pressure drop for cooler fluid, ΔP_c	0.813 kpa
Heat transfer coefficient for warmer fluid, h_w	196.99 w
Heat transfer coefficient for cooler fluid, h_c	193.54 w
Overall heat transfer coefficient, U	50.505 w
Number of transfer unit, NTU	6.17

Table 6. Comparing between a plate and frame and a plate-fin heat exchanger

Variables	**PFHE**	**CHE**
Re_w	104.75	483.01
Re_c	83.1	449.98
h_w	31.39	196.99
h_c	8.88	193.54
U	4.99	50.505
NTU	7.16	6.17
t_2	775	779.2
ΔP_w	0.034	2.085
ΔP_c	0.017	0.813

As it is clear in the Table of (6), despite of the large difference in heat transfer coefficient, Reynolds number and overall heat transfer coefficient, NTU for PFHE (plate and frame heat exchanger) is more than CHE (plate-fin compact heat exchanger). This factor leads to negligible difference between output air temperature. However, the important note, is that, pressure drop for PFHE is much less than CHE, where it has direct impact on annual costs. The total annual costs can be estimated by G.N,Xie, B.Sunden and Q.W.Wang researches [12].

$$\mathrm{TAC} = \mathrm{C_{in}} + \mathrm{C_{out}}$$

$$\mathrm{C_{in}} = \mathrm{C_A} \times \mathrm{A^n}$$

$$\mathrm{C_{op}} = \left\{k_{el}\,\tau\frac{\Delta PV_t}{\eta}\right\}_w + \left\{k_{el}\,\tau\frac{\Delta PV_t}{\eta}\right\}_c$$

Where C_A and K_{el} are the price per unit area and electrical energy, respectively, n and τ are the exponent of nonlinear increase and the hours of operation per year, respectively. ΔP, V_t and η are pressure drop, volumetric flow rate and pump/compressor efficiency respectively.

Table 7. Cost function variables

Variables	**Data**
Price per unit area, C_A	$100\ \frac{\$}{m^2}$
The exponent of nonlinear increase with area increase	0.6
Electrical energy price	$30\ \frac{\$}{MW.h}$
Hours of operation per year	6500 hr
Pump/compressor efficiency	0.6

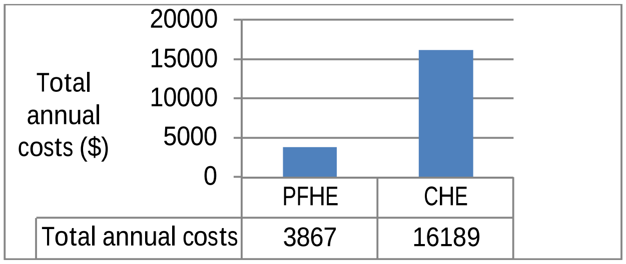

Figure 4.Comparing total annual costs between PFHE and CHE

Then, according to the total annual costs and outlet air temperature it is reasonable to claim that the PFHE has better performance compared with CHE, and in the next stage its performance will be improving by themodynamical optimization process. As it is mentioned previously, minimization of irreversibility subject to constant area is a major component in the overall cost for the heat exchanger system. So, by using this procedure the higher outlet air temperature is possible, where with editing mass flow rate it can be done. Its results is according to Table of (7).

Table 7. Thermodynamical optimization data

	$G_{*,opt}$	$\dot{m}$	$N_{s,min}$
Warmer fluid	25.88	$5.62\ \frac{kg}{s}$	1.23
Cooler fluid	21.44	$4.67\ \frac{kg}{s}$	1.54

Table 8. Output data after thermodynamical optimization

variables	**Manual designing**	**Optimization**
$\dot{m}_W$	$1.4676\ \frac{kg}{s}$	$5.62\ \frac{kg}{s}$
$\dot{m}_C$	$1.45\ \frac{kg}{s}$	$4.67\ \frac{kg}{s}$
Re_w	104.75	1848.31
Re_c	83.1	1228.72
Nu_W	7.28	50.16
Nu_C	1.81	39.08
h_w	$31.38\ \frac{w}{m^2.k}$	$216.30\ \frac{w}{m^2.k}$
h_c	$8.88\ \frac{w}{m^2.k}$	$191.94\ \frac{w}{m^2.k}$
NTU	6.17	13.1
t_2	775 k	781 k
ΔP_w	0.034 kpa	0.668 kpa
ΔP_c	0.017 kpa	0.132 kpa

4. Conclusion

Having increase mass flow rate for both fluids, Reynolds numbers rise dramatically, where this increasing has been coincided with climbing NTU and outlet air temperature. The main reason refer to irreversible free expansion. So, with constant heat transfer area compared with previous state heat transfer coefficient increase sharply, and it causes to rise NTU and heat exchanger efficiency in general. About pressure drop, increasing mass flow rates for both warmer and cooler fluids directly influences pressure drop. So, it is reasonable to claim that thermodynamical optimization process had been successful.

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