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A Comparative Second Law Analysis of Microchannel Evaporator with R-134A & R-22 Refrigerants

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Abstract— In paper second law analysis is applied to study the thermodynamic losses caused by heat transfer between finite temperature difference, and pressure drop for the fluid inside microchannel evaporator which is employed for power electronics cooling. A general expression for second law efficiency is obtained by considering control volume around microchannel evaporator. The conservation equation for mass and energy along with entropy balance are applied in microchannel. Inside the microchannel analytical/empirical correlations are used for friction factor, inlet and exit losses. With the help of expression a comparative study is performed with two different refrigerants viz. R-134A and R-22 on different designs of microchannels to obtain the best design. A parametric study is also performed to show the effects of different parameters on the overall performance of microchannel evaporator.this

Index Terms— Exergy destruction, Heat transfer, Microchannel evaporator, Pressure drop, Second law efficiency.

1 INTRODUCTION

Recently, Electronics industry is advancing at a significant pace.Ecpecially the integration of devices has become much denser than before, which results in increased heat dissipation rate from devices. Thus, a high level performance from cooling technology is required for the optimum performance of devices. After the pioneered work of Tuckerman and Pease [1] ,Microchannels have received considerable attention especially in microelectronics ,that has limited space and where fins ,fans and baffles cannot be used because for the reason of size. Microchannel heat exchanger provides powerful means for dissipating high heat flux with small allowable temperature difference. The important characteristic of microchannel heat exchanger is smaller hydraulic diameter of channel result in large heat transfer coefficient in microchannels.

In the present work second law analysis is applied to determine the overall performance of microchannel evaporator, which is employed in a miniaturized refrigeration system [2] and is in direct thermal contact with the CPU.

By taking in account combined effect of heat transfer and pressure drop expression for second law efficiency is obtained by using the conservation equation for mass and energy with the entropy balance..On the basis of second law efficiency a comparative study is performed with two different refrigerants viz. R-134a and R-22 on different designs of microchannels to obtain the best design.

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2 METHODOLOGY

2.1 Model development

The geometry of microchannel evaporator is shown in fig. (1).The length of the evaporator is L and the width is W .The top surface is insulated and the bottom surface is uniformly heated. A refrigerant passes through a number of microchannels and takes heat away from the heat dissipating electronic component attached below. There are N channels each of height H_c and width w_c, the thickness of each fin is w_w .

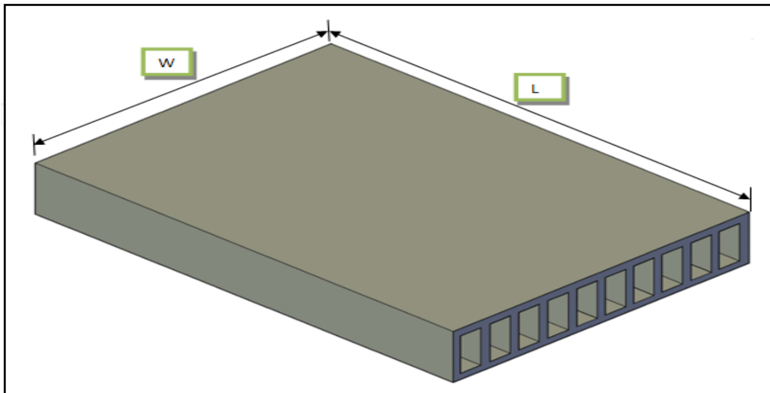

Fig. 1. geometry of microchannel evaporator

Taking advantage of symmetry, a control volume is selected for developing the second law efficiency model, as shown in fig. (2) .

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Fig. 2. Control volume comprises of half channel and fin thickness chosen for analysis

This control volume includes half of the fin and half of the channel along with the base. The ambient temperature outside microchannels is taken as T_a and the base surface temperature of microchannel is taken as T_b .The bulk properties of refrigerant are represented by h_1 , P_1 and s_1 at the inlet and by h_2 ,P_2 , and s_2 at the outlet respectively. The irreversibility of this system is due to heat transfer across finite temperature difference $T_b - T_a$ and to friction. To simplify the analysis follow1g assumptions were employed.

1. Fully developed heat and fluid flow
2. Steady flow
3. Uniform heat flux on bottom surface
4. incompressible fluid with constant thermo physical properties
5. Change in kinetic energy and potential energy is negligible.

Now, applying mass balance for steady state condition reduces to:

$$\dot{m}_1=\dot{m}_2=\dot{m} \quad (1)$$

Neglecting change in the kinetic and potential energies, the first law of thermodynamics for a steady-state condition can be written as

$$\dot{E}_1 = \dot{E}_2 \quad (2)$$

Energy balance rate reduces to

$$Q = \dot{m}\ (h_2 - h_1) \quad (3)$$

Now apply1g entropy balance for steady state condition

$$S_{gen} = \dot{m}\ (s_2 - s_1) - \frac{Q}{T_b} \quad (4)$$

Where T_b represents the absolute temperature of the heat sink base.

Integrating Gibb's equation from inlet to the outlet gives

$$h_2 - h_1 = T_a (s_2 - s_1) + (1/\rho)(P_2 - P_1) \quad (5)$$

Combining (3),(4), and (5),the exergy destruction rate can be written as

$$X_{dest} = Q\left[1 - \frac{T_a}{T_b}\right] + \frac{\dot{m}_a}{\rho}\Delta P \quad (6)$$

If G is the volume flow rate, then the total mass flow rate can be written as

$$\dot{m} = \rho G \quad (7)$$

The pressure drop associated with the flow across the channel is given by

$$\Delta P = (\rho u^2/2)\ [k_{ce} + f\ (L/D_h)] \quad (8)$$

Where the friction factor f for turbulent flow inside the channels depends on the Reynolds number, Roughness value, given by S.E. Haaland in 1983 can be written as

$$1/\sqrt{f} = -1.8 \log[(6.9/Re) + \{(\varepsilon/D_h)/3.7\}^{1.11}] \quad (9)$$

Kleiner [3] used experimental data from Kays and London [4] and derived the following empirical correlation for the entrance and exit losses k_{ce} in terms of channel width and fin thickness

$$k_{ce} = 1.79 - 2.3[w_c/(w_c + w_w)] + 0.53[w_c/(w_c + w_w)]^2 \quad (10)$$

Substituting (7) and (8) into (6) , we get

$$X_{dest} = Q\,[1 - (T_a/T_b)\,] + G(\rho u^2/2)\ [k_{ce} + f\ (L/D_h)] \quad (11)$$

$$= X_{dest,h} + X_{dest,f} \quad (12)$$

Where $X_{dest,h}$ and $X_{dest,f}$ show the exergy destruction rates due to heat transfer and fluid friction, respectively.
The second law efficiency for a steady flow device can be given by its general definition [5],

$$\eta_{II} = [\text{exergy recovered / exergy supplied}] \quad (13)$$

for an microchannel evaporator with two unmixed fluid streams, the exergy supplied is the decrease in the exergy of the hot stream, and the exergy recovered is the increase in the exergy of the cold stream, the expression for second law efficiency can also be written as:

$$\eta_{II} = 1 - [\text{exergy destroyed / exergy supplied}] \quad (14)$$

where exergy destroyed =exergy supplied – exergy recovered.

Hence, second law efficiency for microchannel evaporator is given by the expression:

$$\boldsymbol{\eta_{II} = 1 - [Q\{1 - (T_a/T_b)\} + G(\rho u^2/2)\{k_{ce} + f\ (L/D_h)\}]/[\dot{m}\ (h_2 - h_1)]} \quad (15)$$

2.1Analysis

A micro-channel of size 50 X 50 mm equals to active cooling surface area of electronic cooling chip (assumed) having micro-channels machined on one side and heat dissipating device on the other side, as show in fig (1) is considered. This analysis is based on a practical microelectronics application dissipating a total power of 200 W, chip interface temperature (T_b) is at 45°C.A microchannel of height 0.008 m. and width 0.0005 m. is selected as base design.And by varying the height of microchannel while keeping width fix and vice versa, A total number of twenty micro-channel evaporator structures have been evaluated to identify the variation of second law efficiency with aspect ratio. Two refrigerants R-134A and R-22 are used for analysis of each case.

Table (3) Assumed parameter values for analysis

Parameter	Assumed value
Channel height, H_c (m)	0.008
Channel width, $2w_c$ (m)	0.0005
Fin thickness, $2w_w$ (m)	0.00014
Total heat dissipated or load (W)	200
Length of heat sink ,L (m)	0.05
Width of heat sink ,W (m)	0.05
Evaporator base temperature (k)	318
Ambient temperature	298
Surface Roughness factor, (ε)	1.5*10^(-6)

3 Results

3.1Evaluation study Results

By varying height of base deign and keeping its width fixed, the results of analysis are summarized for each design in table (1) & (2). The sizes of channel, Exergy destruction rate, aspect ratio, second law efficiency and pressure drop across the channel are reported for each case.

TABLE 1
RESULTS OF DESIGN CASES (WIDTH FIXED):R-22

s.no.	H_c (m)	α_c	Δp (Pa)	X_{dest} (w)	η_{II}
1	0.004	0.125	13263	112.8	0.4358
2	0.005	0.1	8303	75.35	0.6233
3	0.006	0.08333	5682	55.53	0.7223
4	0.007	0.07143	4131	43.81	0.7809
5	0.008	0.0625	3139	36.31	0.8185
6	0.009	0.05556	2466	31.22	0.8439
7	0.01	0.05	1988	27.61	0.862
8	0.011	0.04545	1637	24.96	0.8752
9	0.012	0.04167	1372	22.95	0.8853
10	0.013	0.03846	1166	21.39	0.893

TABLE 2
RESULTS OF DESIGN CASES (WIDTH FIXED):R-134A

s.no.	H_c (m)	α_c	Δp (Pa)	X_{dest} (w)	η_{II}
1	0.004	0.125	10638	93	0.535
2	0.005	0.1	6810	64.06	0.6797
3	0.006	0.08333	4756	48.53	0.7573
4	0.007	0.07143	3524	39.22	0.8039
5	0.008	0.0625	2724	33.17	0.8341
6	0.009	0.05556	2175	29.02	0.8549
7	0.01	0.05	1780	26.04	0.8698
8	0.011	0.04545	1487	23.82	0.8809
9	0.012	0.04167	1263	22.13	0.8894
10	0.013	0.03846	1087	20.8	0.896

Variation of second law efficiency with different parameters are as follows :

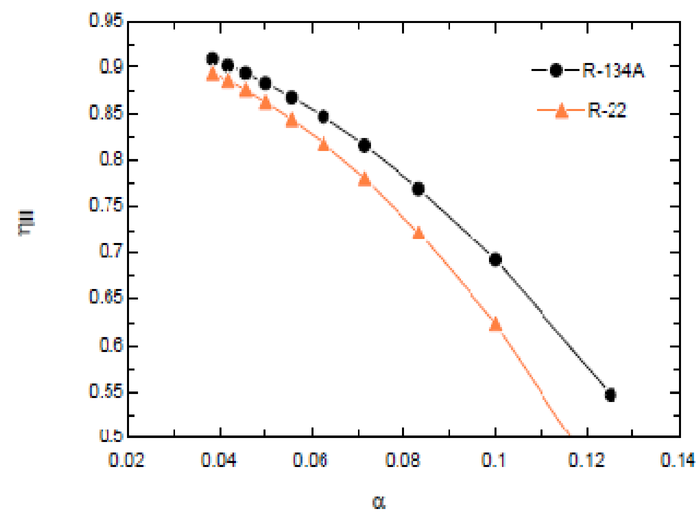

Fig. 3. **By** varying height of channel :Effect of aspect ratio on second law efficiency

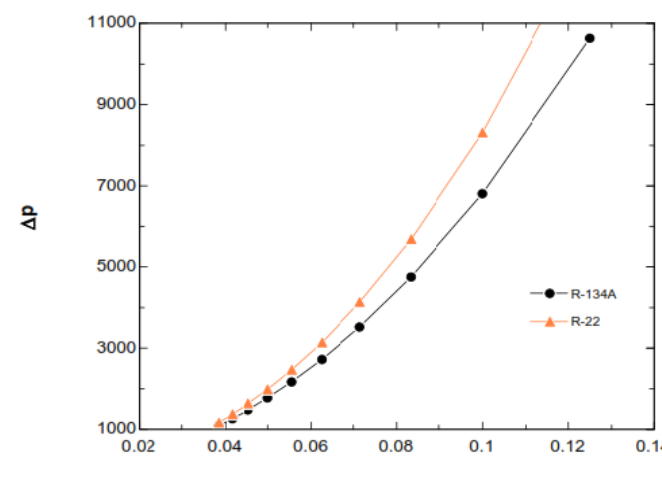

Fig. 4. **By** varying height of channel :Effect of aspect ratio on pressure loss

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Now ,by varying width of base deign and keeping its height fixed, the results of analysis are summarized for each design in table (3) & (4). The sizes of channel, Exergy destruction rate, aspect ratio, second law efficiency and pressure drop across the channel are reported for each case.

TABLE 3

RESULTS OF DESIGN CASES (HEIGHT FIXED) :R-22

s.no.	w_c (m)	α_c	Δp (Pa)	X_{dest} (w)	η_{II}
1	0.0001	0.025	15797	132	0.34
2	0.00015	0.0375	7345	68.11	0.6595
3	0.0002	0.05	4485	46.49	0.7676
4	0.00025	0.0625	3139	36.31	0.8185
5	0.0003	0.075	2381	30.58	0.8471
6	0.00035	0.0875	1904	26.97	0.8651
7	0.0004	0.1	1579	24.52	0.8774
8	0.00045	0.1125	1346	22.75	0.8862
9	0.0005	0.125	1171	21.43	0.8928
10	0.00055	0.1375	1036	20.41	0.898

TABLE4

RESULTS OF DESIGN CASES (HEIGHT FIXED) :R-134A

s.no.	w_c (m)	α_c	Δp (Pa)	X_{dest} (w)	η_{II}
1	0.0001	0.025	13520	114.8	0.4261
2	0.00015	0.0375	6343	60.53	0.6974
3	0.0002	0.05	3887	41.96	0.7902
4	0.00025	0.0625	2724	33.17	0.8341
5	0.0003	0.075	2067	28.21	0.859
6	0.00035	0.0875	1653	25.08	0.8746
7	0.0004	0.1	1371	22.94	0.8853
8	0.00045	0.1125	1168	21.41	0.8929
9	0.0005	0.125	1016	20.26	0.8987
10	0.00055	0.1375	898.5	19.37	0.9031

Variation of second law efficiency with different parameters are as follows :

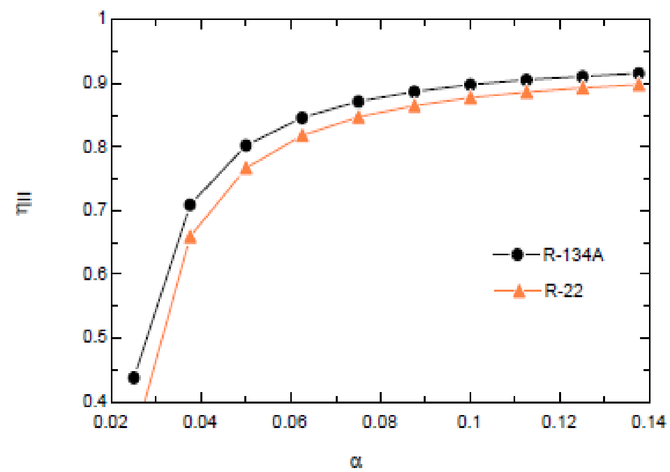

Fig. 5 **By** varying width of channel: Effect of aspect ratio on second law efficiency

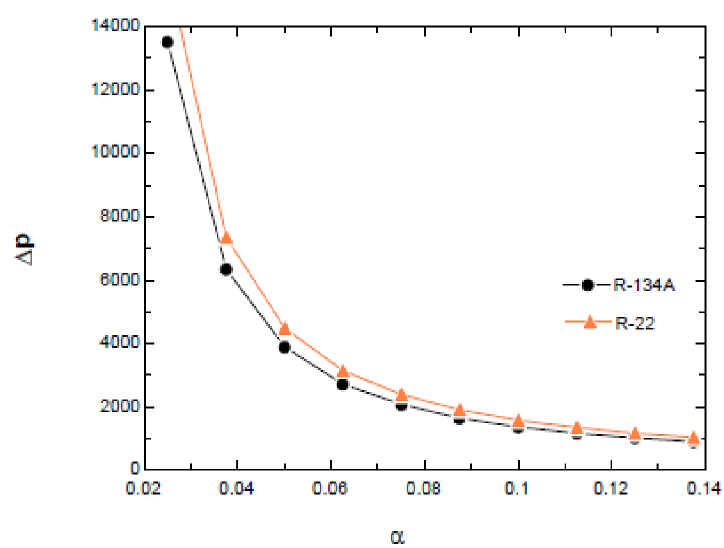

Fig. 6. **By** varying width of channel: Effect of aspect ratio on pressure loss

3.2 Parametric Study Results:

On base design parametric study is carried out by varying volume flow rate of refrigerant and heat load from source.Results are shown graphically as follows

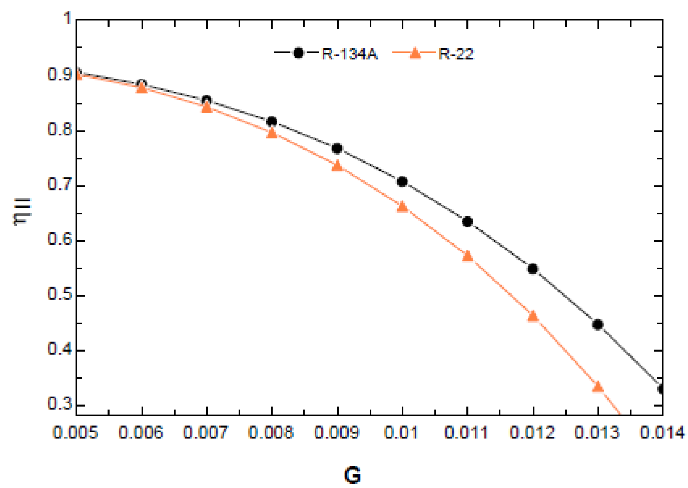

Fig. 7. Effect of volume flow rate on second law efficiency

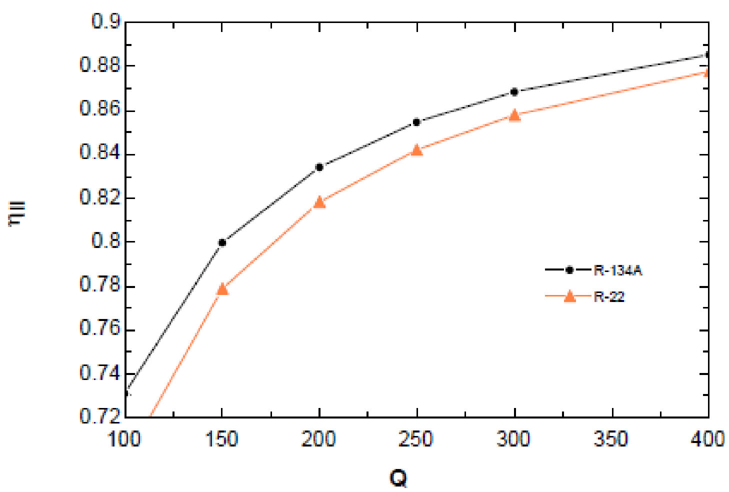

Fig. 8. Effect of heat load on second law efficiency

4 CONCLUSION

Based on evaluation study of analytical model of microchannel evaporator using R-22 & R-134A refrigerants we have following conclusions:

1. By keeping width of channel constant while varying it's height: second law efficiency of evaporator decreases with increase in aspect ratio.

2. By keeping width of channel constant while varying it's height: Pressure loss across the channel of evaporator increases with increase in aspect ratio.

3. By keeping height of channel constant while varying it's width: second law efficiency of evaporator increases with increase in aspect ratio

4. By keeping height of channel constant while varying it's width: Pressure loss across the channel of evaporator decreases with increase in aspect ratio

5 For base design parametric study shows that:

a) With increase in volume flow rate of refrigerant across the microchannel second law efficiency decreases,because of more exergy is destroyed across the channel.

b) With increase in heat load second law efficiency improves because lesser exergy is destroyed because of heat transfer and more exergy is recovered during heat transfer.

6 .R-134A shows better thermal performance than R-22 .

NOMENCLATURE

D_h Hydraulic diameter [m]

f Friction factor

H_c Channel height [m]

k_{ce} Sum of entrance and exit losses

m Mass flow rate [kg/s]

N Total number of microchannels

Q Heat transfer rate [W]

Re_{dh} Reynolds number based on hydraulic diameter = $D_h.U_{av}/\nu$

S_{gen} Total entropy generation rate [W/K]

T Absolute temperature [K]

u Average velocity in channels [m/s]

W Width of heat evaporator [m]

w_c Half of channel width [m]

w_w Half of fin thickness [m]

X_{dest} Exergy destroyed [W]

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GREEK SYMBOLS

α_c Channel aspect ratio $=2 w_c / H_c$

ν Kinematic viscosity of fluid [m^2/s]

ρ Fluid density [kg/m^3]

η_{II} Second law efficiency

ε Surface Roughness factor

SUBSCRIPTS

a Ambient

b base surface

c Channel

1 inlet

2 outlet

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