ENTROPY GENERATION ANALYSIS FOR IN-LINE TUBE BANKS

بحث تحليلى لمعيار إضمحلال الطاقة للمبادلات الحرارية ذات الوضع الهندسى ذات الصفوف الخطية للأنابيب

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الخلاصة :

يتناول البحث دراسة تحليليه لمعيار مقدار الإضمحلال فى الطاقة (اللإنعكاسية) التى لا يمكن إسترجاعها أو الإستفادة منها مرة أخرى فى شكل طاقة شغل حيث يعتبر مقدار الإضمحلال فى الطاقة أداه أساسية فى تقنين و تحليل عمليات الطاقة.

و البحث يتعرض لمناقشة إضمحلال الطاقة الناتجة عن إنتقال الحرارة بالإضافة إلى الإضمحلال الناتج عن طاقة الدفع الضغطية اللازمة لدفع المائع خلال المبادل الحرارى.

باستخدام الدراسة التحليلية للقانون الثانى أمكن الربط بين الطاقة المضمحلة الناتجة عن إنتقال الحرارة و الناتجة عن الفقد فى الضغط بعلاقة واحدة توضح معامل الأداء بالنسبة لأى مبادل حرارى من النوع ذو الوضع الهندسى ذات الصفوف الخطية للأنابيب.

أوضحت الدراسة التحليلية أنه عند حالات معينة لكل مبادل حرارى تكون قيمة الطاقة المضمحلة أقل مايمكن بمعنى أنه عند هذه الحالات يكون أكبر معامل أداء للمبادل الحرارى.

Abstract:

Entropy generation is an effective tool in energy process analysis and optimization. Problems as entropy generation in heat exchangers due to pressure losses and due to heat transfer are discussed. The entropy generation number Ns provides a meaning for evaluating the performance of a heat exchanger.

The performance of in-line tube banks are studied at constant relative transfer pitch with variable relative longitudinal pitch and vice-versa. The results of the performance showed that there is a minimum entropy generation occurs at particular Reynolds number and number of transfer units for each tube bank.

Introduction:

Heat transfer, as a way of thinking and formulating problems, is considerably older than thermodynamics. The fundamental engineering problem in heat transfer, the relationship between temperature difference and heat transfer rate was formulated more than 200 years ago by Newton, Biot, Furier et. al. Heat transfer processes and devices are inherently irreversible; in other words, heat transfer phenomena affects the one way destruction of available work. Nowadays, in which available work is increasing scare and expensive, it is necessary to describe in precise terms how wasteful heat transfer phenomena are, so that responsible decisions can be made about whether and how to curb such waste.

Consideration of available energy and irreversibility falls within the thermodynamic realm. Thermodynamics, in general, and the second law, in

particular, occupies a central place in the solution of heat transfer problems. Entropy generation should assume a central role in heat transfer, as central as the relationship between temperature difference and heat transfer rate or the relationship between pressure drop energy and flow through a duct or the combination between entropy generation caused by heat transfer rate as well as pressure drop energy.

A heat exchanger is an inherently irreversible device and, consequently the second law aspects of heat exchanger theory and design have been considered frequently [1, 2, 3].

It can be readily shown that for most flow passages that might be used for the heat transfer surfaces of an exchanger, the heat transfer rate per unit of surface area can be increased by increasing fluid-flow velocity. But, the friction-power expenditure is also increased with flow velocity. So that, the frictional pressure drop losses are said to be coupled in the sense that any design change aimed to reducing one type of loss is likely to have an opposite effect on the other. Due to this coupling, it is then difficult to determine a priori whether a proposed design modification will yield a net improvement in heat exchanger performance.

This paper presents the coupling between losses due to heat transfer across the fluid-to-fluid temperature difference and losses caused by fluid friction, for in-line tube banks with various relative longitudinal pitches at constant relative transverse pitch and vice-versa using the concept of heat exchanger irreversibility. Based on this concept, the entropy generation number is used as a basic parameter in describing heat exchanger performance. this dimensionless group was proposed by[1].

The entropy generation number is used as indicator for heat exchanger performance, namely the ratio of heat transfer energy to lost fluid bumping power as defined in [4, 5]. It is shown in what follows that increasing the ratio of heat transfer energy to pumping power is not sufficient for calming improvements in heat exchanger performance.

Further studies on compact cross flow heat exchangers[6] and on regenerators of gas turbine [7] were mainly concerned with the optimization through the choice of the minimum entropy production.

This paper deals with:

(a) Developing a general expression of the entropy generation.

(b) Defining two new correlations for optimum NTU and Reynolds number at which the heat exchanger operates more efficiently.

(c) Investigating which type of in-line tube heat exchanger operates with minimum losses at the range of operation.

Entropy Generation Analysis:

For the heat exchanger, the entropy generation rate can be given by

$$dS_{gen} = m_h\, ds_h + m_c\, ds_c \qquad (1)$$

where the heat transfer to the environment is neglected, and the subscripts c and h are for cold and hot streams respectively.

Expressing the entropy generation in a general way.

$$ds = C_p\, dT/T - (\partial\upsilon/\partial T)_p\, dP \qquad (2)$$

The integration between inlet and outlet gives

$$S_{gen} = m_h \left\{ C_{p_h} \ln(T_o / T_i)_h - \int_i^o (\partial \upsilon / \partial T)_{h,p}\, dP \right\} + m_c \left\{ C_{p_c} \ln(T_o / T_i)_c - \int_i^o (\partial \upsilon / \partial T)_{c,p}\, dP \right\} \quad (3)$$

with the assumption that C_{p_h} and C_{p_c} are averaged between T_i & T_o for each stream. There are two cases, the first when the capacity rate of the hot stream is the minimum capacity rate ($C_h = C_{min}$). The second case, when the capacity rate of the hot stream is the maximum capacity rate ($C_h = C_{max}$).

(I) *Entropy generation due to heat transfer with temperature difference:*

The heat exchanger effectiveness (ε) is defined as the ratio of the actual heat transfer (Q_{act}) to the maximum possible heat transfer (Q_{max}) [10] where,

$$Q_{act} = C_h(T_{hi} - T_{ho}) = C_C(T_{Co} - T_{Ci}) \quad (4)$$

$$Q_{max} = C_{min}(T_{hi} - T_{Ci}) \quad (5)$$

Combining equation (4) and (5) with the definition of the effectiveness leads to,

$$\varepsilon = \frac{Q_{act}}{Q_{max}} = \frac{C_h(T_{hi} - T_{ho})}{C_{min}(T_{hi} - T_{Ci})} = \frac{C_C(T_{Co} - T_{Ci})}{C_{min}(T_{hi} - T_{Ci})} \quad (6)$$

(i) Entropy generation at $C_h = C_{min}$

when $C_h = C_{min}$ equation (6) becomes,

$$\varepsilon = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{Ci}} = \frac{1 - (T_{ho} / T_{hi})}{1 - (T_{Ci} / T_{hi})} \quad (7)$$

Then the ratio of outlet to inlet temperature for the hot stream (T_{ho} / T_{hi}) can be put in this form

$$\frac{T_{ho}}{T_{hi}} = 1 - \varepsilon(1 - \tau) \quad (8)$$

where $\tau = (T_{Ci} / T_{hi})$ is the absolute inlet temperatures ratio. Also the ε can be put in the form

$$\varepsilon = \frac{C_{max}(T_{Co} - T_{Ci})}{C_{min}(T_{hi} - T_{Ci})} = \frac{1}{\omega}\left[\frac{1 - T_{ho} / T_{hi}}{1 - \tau}\right] \quad (9)$$

where ω is the ratio of the minimum to maximum capacity rates. Lets us take the absolute outlet to inlet temperature ratio for the clod stream, (T_{Co} / T_{Ci}) as,

$$\frac{T_{Co}}{T_{Ci}} = 1 + \varepsilon\,\omega\,(1/\tau - 1) \quad (10)$$

The entropy generation due to heat transfer with temperature difference is given by,

$$S_{gen,\Delta T} = (m\,C_P)_h \ln\left(\frac{T_{ho}}{T_{hi}}\right) + (m\,C_P)_c \ln\left(\frac{T_{Co}}{T_{Ci}}\right) \quad (11)$$

Combining equations (8) and (10) with the equation (11) gives,

$$S_{gen,\Delta T} = C_{min} \ln[1 - \varepsilon(1 - \tau)] + C_{max} \ln[1 + \varepsilon\omega\,(1/\tau - 1)] \quad (12)$$

(ii) Entropy generation at $C_h = C_{max}$

Where $C_h = C_{max}$, the ratio of outlet to inlet temperature in terms of effectiveness becomes

$$\varepsilon = \frac{C_{max}(T_{hi} - T_{h0})}{C_{min}(T_{hi} - T_{Ci})} = \frac{1}{\omega}\left[\frac{(T_{h0}/T_{hi})}{1/\tau \ -1}\right]$$

So that,

$$\frac{T_{h0}}{T_{hi}} = 1 + \varepsilon\omega\,(1-\tau) \qquad (13)$$

Also, $\varepsilon = \dfrac{T_{C0} - T_{Ci}}{T_{hi} - T_{Ci}} = \dfrac{T_{C0}/T_{Ci} - 1}{1/\tau \ -1}$

So that,

$$\frac{T_{C0}}{T_{Ci}} = 1 + \varepsilon(1-\tau) \qquad (14)$$

Combining equations (13) and (14) with equation (11) gives,

$$S_{gen,\Delta T} = C_{max}\left\{\ln[1 - \varepsilon\omega(1-\tau)] + \omega\ln[1 + \varepsilon(1/\tau - 1)]\right\} \qquad (15)$$

(II) *Entropy generation due to pressure drop*

The entropy generation due to pressure drop for a heat exchanger is given from equation (3),

$$S_{gen,\Delta P} = -m_h\int_i^o (\partial\upsilon/\partial T)_{h,p}\, dP - m_C\int_i^o (\partial\upsilon/\partial T)_{c,p} dP \qquad (16)$$

for a liquid it is assumed that

$(\partial\upsilon/\partial T)_p = \beta\upsilon = \text{constant}$

where β is volumetric expansion coefficient

While for a perfect gas

$$\int_i^o (\partial\upsilon/\partial T)_p\, dP = \int_i^o R/P\, dP = R\ln\left(\frac{P_o}{P_i}\right) = R\ln\left(\frac{P_i + \Delta P}{P_i}\right)$$

$$= R\ln\left(1 + \frac{\Delta P}{P_i}\right) \qquad (17)$$

with the hypothesis $\Delta P/P_i << 1$ one gets

$$\int_i^o (\partial\upsilon/\partial T)_p\, dP = R\ln\left(1 + \frac{\Delta P}{P_i}\right) \approx R\,\Delta P/P_i \qquad (18)$$

In general it is possible to express

$$\int_i^o (\partial\upsilon/\partial T)_p\, dP = \gamma\,\Delta P \qquad (19)$$

where $\gamma = \beta v$ (20)

for a liquid, and

$$\gamma = R/P_i \qquad (21)$$

for a perfect gas

(i) Entropy generation at $C_h = C_{min}$

When $C_h = C_{min}$, the entropy generation due to pressure drop is given by combining equations (16) and (19) to give

$$S_{gen,\Delta P} = -(C_{min}/C_{p_h})\,\gamma_h\,\Delta P_h - (C_{max}/C_{p_c})\,\gamma_c\,\Delta P_c \qquad (22)$$

(ii) Entropy generation at $C_h = C_{max}$

In this case, when $C_h = C_{max}$, the $S_{gen,\Delta P}$ becomes,

$$S_{gen,\Delta P} = -(C_{max}/C_{p_h})\,\gamma_h\,\Delta P_h - (C_{min}/C_{p_c})\,\gamma_c\,\Delta P_c \qquad (23)$$

Total Entropy Generation:

The total entropy generation due to heat transfer with temperature difference and pressure drop is given by:

(1) At $C_h = C_{min}$, adding equations (12) and (22) leads to

$$S_{gen} = C_{max}\{\omega \ln[1-\varepsilon(1-\tau)] + \ln[1+\omega\varepsilon(1/\tau - 1)] - (\omega / C_{p_h})\, \gamma_h\, \Delta P_h - (1/C_{p_c})\, \gamma_c\, \Delta P_c\} \quad (24)$$

(2) At $C_h = C_{max}$, adding equations (15) and (23) to get

$$S_{gen} = C_{max}\{ \ln[1-\omega\varepsilon(1-\tau)] + \omega \ln[1+\varepsilon(1/\tau - 1)] - (1/C_{p_h})\, \gamma_h\, \Delta P_h - (\omega / C_{p_c})\, \gamma_c\, \Delta P_c\} \quad (25)$$

Since the momentum transfer information is condensed in dimensionless group such as friction factor, skin friction coefficient or drag coefficient, and since heat transfer results are expressed similarly in the form of Nusselt or Stanton number, it is appropriate to define a dimensionless group for second-law analysis in heat transfer, the entropy generation number Ns [1]. This group is defined in a manner similar to friction factor and Nusselt number

$$N_S = \frac{\text{Actual entropy generation rate}}{\text{Characteristic entropy genration rate}}$$

i.e. $$N_S = \frac{S_{gen}}{C_{max}} \quad (26)$$

Therefore with the help of definition of Ns, equation (24) can be put in the form, when $C_h = C_{min}$

$$N_S = \omega \ln[1-\varepsilon(1-\tau)] + \ln[1+\omega\varepsilon(1/\tau - 1)] - (\omega / C_{p_h})\, \gamma_h\, \Delta P_h - (1/C_{p_c})\, \gamma_c\, \Delta P_c \quad (27)$$

Also, when $C_h = C_{max}$ equation (25) can be take the form,

$$N_S = \ln[1-\omega\varepsilon(1-\tau)] + \omega \ln[1+\varepsilon(1/\tau - 1)] - (1/C_{p_h})\, \gamma_h\, \Delta P_h - (\omega / C_{p_c})\, \gamma_c\, \Delta P_c \quad (28)$$

Equations (27) and (28) can be put in the form

$$N_S = N_{S,T} + N_{S,P_h} + N_{S,P_c} \quad (29)$$

where

$N_{S,T}$ is the entropy generation number due to heat transfer with temperature difference,

N_{S,P_h} is the entropy generation number due to pressure drop of a hot stream,

N_{S,P_c} is the entropy generation number due to pressure drop of a cold stream.

(1) At $C_h = C_{min}$

$$N_{S,T} = \omega \ln[1-\varepsilon(1-\tau)] + \ln[1+\omega\varepsilon(1/\tau - 1)] \quad (30)$$

$$N_{S,P_h} = -(\omega / C_{p_h})\, \gamma_h\, \Delta P_h \quad (31)$$

$$N_{S,P_c} = -(1/C_{p_c})\, \gamma_c\, \Delta P_C \quad (32)$$

(1) At $C_h = C_{max}$

$$N_{S,T} = \ln[1-\omega\varepsilon(1-\tau)] + \omega \ln[1+\varepsilon(1/\tau - 1)] \quad (33)$$

$$N_{s,P_h} = -(1/C_{p_h})\,\gamma_h\,\Delta P_h \tag{34}$$

$$N_{s,P_c} = -(\omega/C_{p_c})\,\gamma_c\,\Delta P_C \tag{35}$$

It can be shown that $N_{s,T}$ has an extremum at $\partial N_{s,T}/\partial\varepsilon = 0$, simplifying this equation leads to $\varepsilon = \dfrac{1}{1+\omega}$ (36)

The effectiveness which given by above equation is the value at which maximum irreversibility contribution from heat transfer with temperature difference.

Application of Second Law Analysis For Crossflow of In-Line Tube Banks

Heat transfer of tube banks is governed mainly by the flow velocity, tube arrangement, fluid physical properties, thermal load, and heat flux distribution. In the dimensionless from,

$$Nu = f(R_e, P_r, K/K_w, \mu/\mu_w, C_p/C_{p_w}, \rho/\rho_w, S_t/D, S_l/D) \tag{37}$$

where

S_t= Transfer pitch , S_l = Longitudinal pitch , D = Tube diameter

and Re is based on the minimum flow area. Experimental results are interpreted by functional relation in the exponential form [4]. Where C and m in a definite region of Re are governed by the tube arrangement, and the exponent n and p denotes the effect of fluid physical properties on heat transfer. An experimental investigation of the heat transfer characteristics was performed by [8, 9] for in-line banks.

The Stanton number which is defined by

$$St = Nu / Re\,.\,Pr \tag{38}$$

and the number of transfer unit NTU is defined [10]

$$NTU = (4L/D_h)\,S_t\,(C/C_{min})\,\eta_o \tag{39}$$

where,

- D_h is the hydraulic diameter
- L is the length of heat exchanger flow path
- C is the capacity rate of the stream
- C_{min} is the minimum capacity rate
- η_o is the surface efficiency parameter

The effectiveness ε when the stronger fluid is mixed is given by [10]

$$\varepsilon = \frac{1}{\omega}\left\{1-\exp\left[-\omega\left(1-\exp(-NTU)\right)\right]\right\} \tag{40}$$

but when the weaker fluid is mixed

$$\varepsilon = 1-\exp\left[-\left(1-\exp(-NTU\cdot\omega)\right)/\omega\right] \tag{41}$$

Form equations (37) to (41) the entropy generation number due to transfer can be evaluated.

Pressure losses across banks of tube is governed by the flow dynamics in the intertube spaces. The existence of considerable acceleration and deceleration over tubes develops separated flow that absorb large amounts of kinetic energy of fluid motion. Consequently, tube arrangement must be a determining factor for the pressure drop, ΔP for flow through a bank of tubes is a function of the geometry (expressed in terms of S_t , S_l and D), the number of tube rows in the bank Z, the flow velocity u, and the physical properties of the fluid

$$\Delta P = f(u, S_t, S_l, D, Z, \mu, \rho) \tag{42}$$

It is convenient, in what follows, to define u as the maximum velocity attained by the fluid as it passes through the varies gaps between the tubes. Relation (42) may be written in dimensionless form,

$$Eu = f(R_e, a, b, Z) \qquad (43)$$

The appropriate equations for Eu was performed by [9] for in-line tube banks. The pressure drop ΔP across the tube banks is then given by,

$$\Delta P = Eu\,(\rho u^2/2)\,Z \qquad (44)$$

So that, the entropy generation due to pressure drop can be accounted.

Results And Discussion:

The entropy generation calculations have been carried out according to the equations formulated in entropy generation analysis as before at τ=0.5 and ω=1.0. As an input, the data reported in [8, 9] for heat transfer and drag in tube bundle of in-line tube banks in the form of correlations for Nusselt number and Euler number.

The results are presented in the following form

(i) $N_{S,P}$ versus Re (ii) $N_{S,T}$ versus Re

(iii) N_S versus Re (iv) N_S versus NTU

(v) ϕ, Irreversibility distribution. Ratio versus Re

Fig. (1) presents the results of $N_{S,P}$ as a function of Reynolds number for values of relative transfer pitch "a". The monotonic increase in $N_{S,P}$ with Reynolds number for all values of "a" reflects the effect of the greater power expended to overcome drag as Reynolds number increases. This effect may also be recognized by considering the fact that in the limit as Reynolds number goes to zero, drag goes to zero.

The effect of narrowing the transfer pitch, i.e., reducing the value of "a" will be to increase velocity gradient between adjacent tubes and hence lead to greater flow resistance. This is reflected in Fig. (1) as the lowest $N_{S,P}$ for a=2.5, increasing progressively as "a" goes to 1.25.

In Fig. (2) $N_{S,T}$ is plotted against Reynolds number, indicating that entropy generation due to heat transfer differs from entropy generation due to flow drag (pressure drop) in the essential way, that as Reynolds number increases entropy generation decreases. For in-line tube banks, all transverse spacings have the same heat transfer behaviour but not the same friction drag behaviour [8],this means that the entropy generation due to heat transfer is the same for all transfer spacings. This is reflected in Fig. (2), where all the lines for different "a" converge to a single line.

Both of heat transfer coefficient and pressure drop are increasing fluid-flow velocity. The losses of heat transfer and pressure drop are said to be coupled in the sense that any design change aimed to reduce one type of loss is likely to has an opposite effect on the other, as shown in Fig. (3). This figure illustrates that the entropy generation due to pressure drop increases by increasing Reynolds number and vice-versa for the entropy generation due to heat transfer. Due to this coupling there is optimum point at particular Reynolds number at which minimum entropy generation occures.

The total entropy generation lines due to heat transfer and drag of external flow for in-line tube banks are not intersecting as shown in Fig. (4). The value of a=1.25 has higher Ns throughout. Widder spacing are better for lower entropy generation. The final choice will of course depend on a cost estimate for different designs. For largest transverse pitch a=2.5, the minimum entropy generation occures at about

Reynolds number 64000. The range of the Re from 28000 to 64000 is the range for minimum entropy generation for all investigated transverse pitches .

A clear and well-defined minimum Ns for in-line tube banks is shown in Fig (5). In this Figure the Ns due to the interaction of the two entropy generation mechanisms is plotted against NTU. The locus of the minimum Ns being in a straight line. We may even fit an equation for the straight line to interpolate or extrapolate mininma for different values of "a" while "b" is kept constant b=2.0.

The occurrence of the minima Ns has the important connotation that, if a bundle geometry is chosen suitably and operated at appropriate Reynolds number, it will have the best entropy generation characteristics. Such a result is an additional criterion for design and operation of heat exchangers.

The results of irreversibility distribution ratio (Irr, Dist. Ration, ϕ) against the Re for various value of transverse pitch are presented in Fig (6). The entropy generation Ns, has been arranged to show the special forms assumed by Ns in the two extremes, $\phi \rightarrow 0$ and $\phi \rightarrow \infty$. As expected, in a situation dominated by heat transfer irreversibility ($\phi \rightarrow 0$) the entropy generation number will be proportional among to the heat transfer coefficient. Conversely, when the irreversibility is dominated by ΔP effects ($\phi \rightarrow \infty$), Ns will vary as the friction factor. Fig (6) reflect that $\phi<1$, this means that, the entropy generation caused by heat transfer is grater than that caused by pressure drop. Thus, from the viewpoint of thermal design optimization, the systematic elimination of irreversibility sources in the system is the direct route towards to minimize the entropy generation caused by heat transfer irreversibility by increasing the heat transfer coefficient.

At constant transverse spacing, a=2.0, the total entropy generation due to heat transfer and drag of external fluid Ns is plotted against Re as shown in Fig. (7) for in-line tube banks. The effect of narrowing the longitudinal pitch, i.e., reducing the value of "b" will be reduced Ns. As the lowest Ns for b=1.25 increasing progressively as "b" goes to 2.0. For all longitudinal pitches investigated there is a minimum Ns indicated in Fig. (7). The minimum Ns is also indicated in Fig. (8), where Ns is plotted against NTU.

Conclusions:

The results showed that, the entropy generation number is generally non-monotonic with respect to changing design parameters. Consequently, the use of design rules such as minimizing the fluid-to-fluid temperature difference or maximizing the ratio of beat transfer rate to fluid bumping power is not sufficient for seeking improved thermal performance. The Ns criterion is a more adequate measure of thermodynamic imperfection and provides a more complete picture of how various design variable influence of thermal performance. The second law is used to evaluate the performance of crossflow heat exchanger for in-line tube banks at ω=1.0 and τ=0.5. The tube banks are analysed for tube banks of constant relative transverse pitch (a=2.0) and variable relative longitudinal pitch and vice-versa. The coupling of entropy generation due to heat transfer and pressure drop of external flow are as a qualitative results for the performance of these heat exchangers. the results indicated that, the entropy generation number Ns decreases with the increase of Re up to optimum Re while above that, Ns increases for all the tube banks. At b=2.0 and a=1.25, 1.5, 2.0 & 2.5, the higher performance for tube bank of a=2.5. But at a=2.0 and b=1.25, 1.5 & 2.0, the higher performance for tube bank of b=1.25.

Nomenclature:

a relative transverse pitch
b relative longitudinal pitch
C capacity rate m C_p (W / K)
C_p specific heat at constant pressure (J / Kg K)
Eu Euler number
G exchanger mass velocity based on the minimum free-flow area in the core (Kg / m^2 s)
h convective heat transfer coefficient (W / m^2 K)
m mass flow rate (Kg / s)
Ns entropy generation number due to ΔP and ΔT
$N_{s,t}$ entropy generation number due to heat transfer with temperature difference, ΔT
$N_{s,p}$ entropy generation number due to pressure drop, ΔP
Nu Nusselt number based on actual heat transfer area
NTU number of transfer unit
P pressure (N / m^2)
ΔP pressure drop (N / m^2)
Pr Prandtl number
Q heat transfer interaction (W)
Re Reynolds number
$S_{gen,\Delta P}$ entropy generation due to fluid friction (W / K)
$S_{gen,\Delta T}$ entropy generation due to heat transfer (W / K)
St Stanton number
u velocity in minimum free flow area (m / s)
v specific volume (m^3/ Kg)
Z total number of rows in the heat exchanger

Greek Letter Symbols

β volumetric expansion coefficient
τ absolute inlet temperature ratio (Tci / Thi)
ω capacity ratio (C_{min} / C_{max})
ε heat exchanger effectiveness (Q_{act} / Q_{max})
φ irreversibility distribution ratio ($N_{s,p}$ / $N_{s,t}$)
μ dynamic viscosity (N·S / m^2)
γ kinamatic viscosity (m^2/ s)
ρ fluid density (Kg / m^3)

Subcribts

a air
c cold stream
h hot stream
i inner, inlet
o outside, outlet

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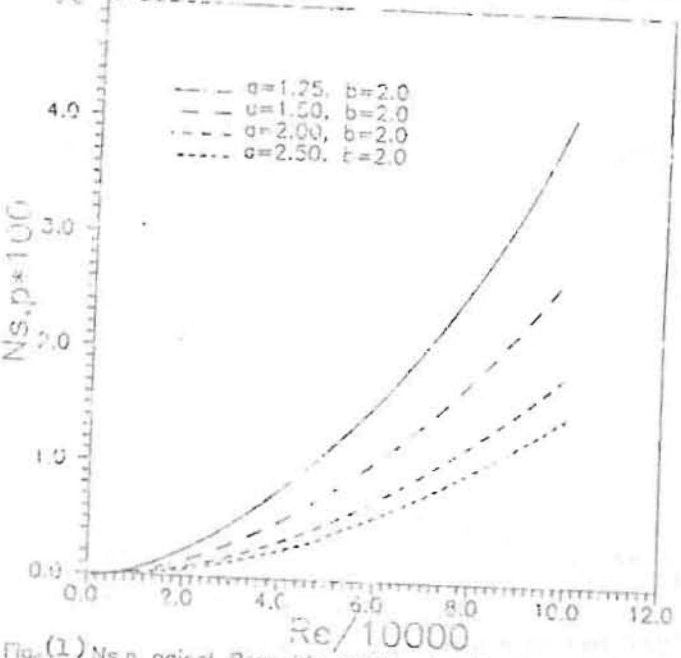

Fig. (1) Ns,p against Reynolds number for in-line tube banks (b=2.0)

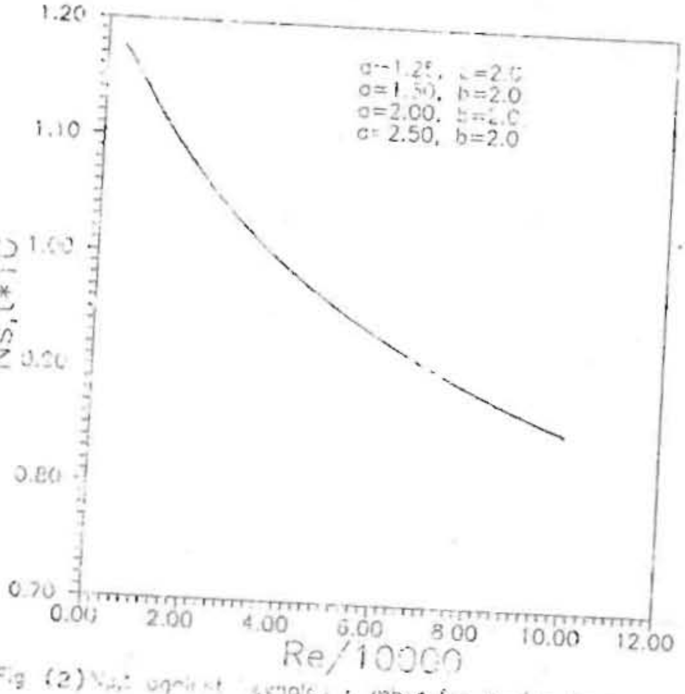

Fig. (2) Ns,t against Reynolds number for in-line tube banks (b=2.0)

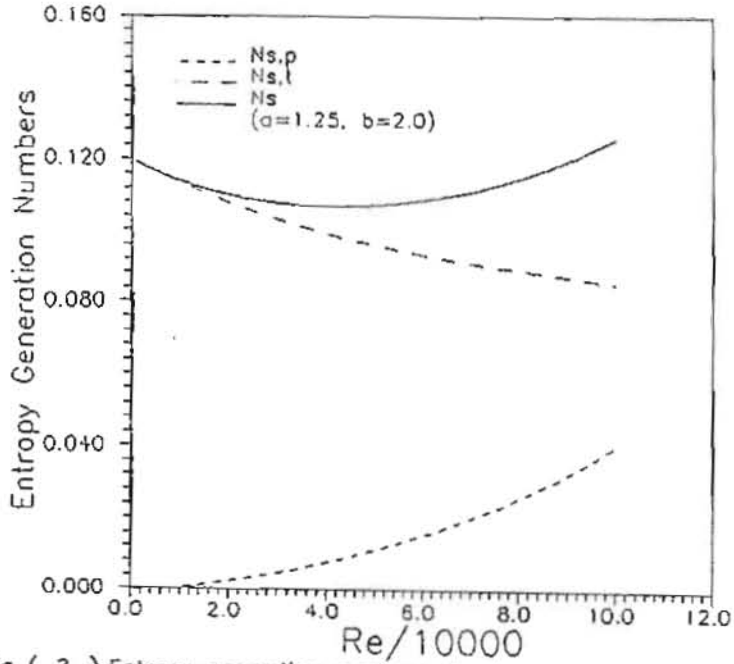

Fig. (3) Entropy generation numbers due to pressure drop, heat transfer & total Ns for in-line tube bank

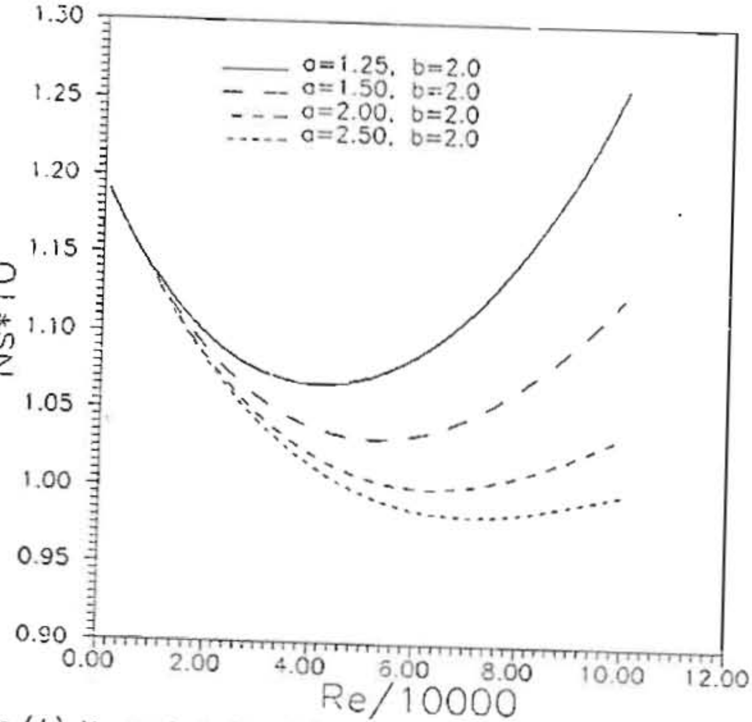

Fig. (4) Ns against Reynolds number for in-line tube banks (b=2.0)

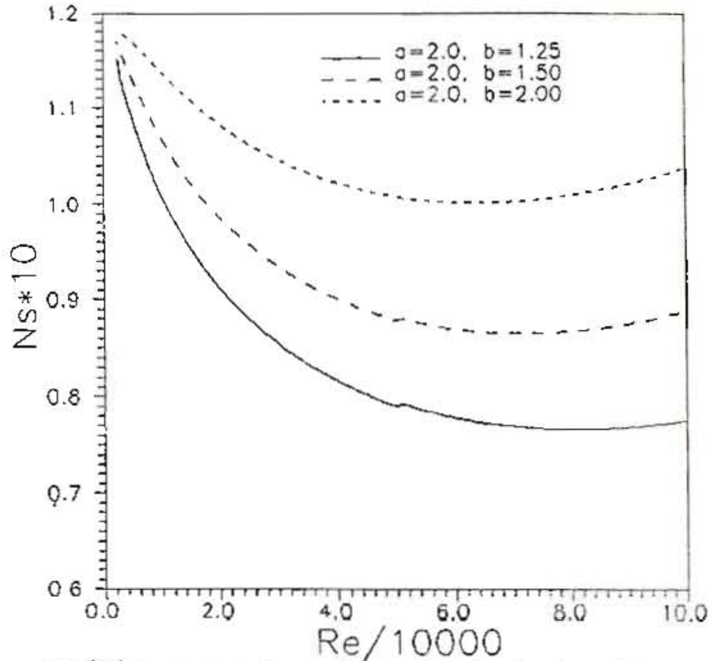

Fig.(7) Ns against Reynolds number for in-line plain tube banks (a= 2.0)

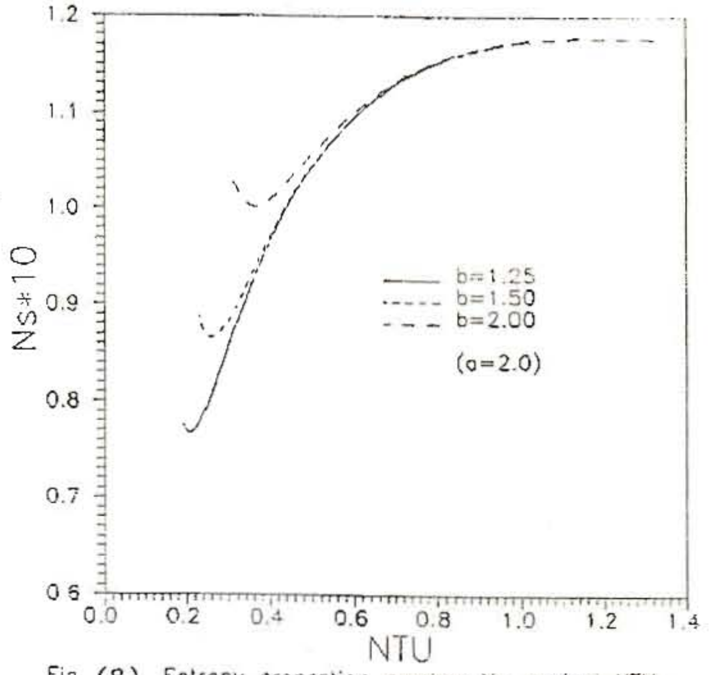

Fig. (8) Entropy grenration number Ns against NTU for in-line tube banks at a=2.0