Dynamics of Spiral Plate Heat Exchanger Subject to the Change of Inlet Fluid Temperature

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Abstract

Dynamic behavior of the fluid temperature for spiral plate heat exchanger is studied. The fundamental equations for unsteady state and the algorithm for numerical solution are proposed utilizing the overall heat transfer coefficient and temperature profile estimation method reported previously. Response to the chage of influx temperature is discussed in frequency domain. Time constants of the apparatus are evaluated from the first order cumulant for several operating conditions.

1. Introduction

The spiral plate heat exchanger consists of two plates wound round each other maintaining a constant distance in spiral to form the concentric channels. Except for the center and periphery, heat is exchanged through the bilateral sides of the channel. Winding a conduit into spiral from, the efficiency of the equipment increases approximately twice as high as that of a straight structure[1]. The cross section of the channel is rectangular with a high aspect ratio. It is reported that the flow is turbulent in effect due to a number of spacer studs inserted in the passage[2]. The spiral plate heat exchanger offers following advantages[3),4)] : it is compact with high temperature efficiency; high velocity of the fluid avoids a fouling problem; distribution is good because of the single flow passage; counter flow and long channel make possible to control temperature precisely.

In this paper, dynamic behavior of spiral plate heat exchanger subject to the change of inlet temperature of cold fluid is discussed. Fundamental equations and their algorithm for the solutions are proposed. In the calculation procedure, estimation method for overall heat transfer coefficient is used to evaluate the initial temperature profille. The change of the output temperature of the hot fluid is calculated through the equations as the cold influx temperature varies. Characteristics are discussed in the frequency region. There are several ways to express the dynamics quantitatively for heat exchanger[5]. Inditial response is intuitively clear to understand. However, the equation will be complicated. Pulse method and statistical method are difficult to realize the input to the equipment experimentally. Frequency response is excellent method to treat the unit as a distributed parameter system. It is also easy to obtain other responses approximately from the results. Time constants of the unit are estimated from the cumulant for several operating conditions.

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2. Fundamental Equations

Four combinations of the cold and hot fluids are possible for the counter flow of this type heat exchanger. For a general heat exchanger, countercurrent shows less phase shift and higher gain characteristics than co−current[6]. Since all through the equipment each passage is adjoined by an ascending and a descending turn of the other passage, it is argued that the structure of the unit is not exactly counter−flow type. The flow configuration discussed in this paper is the conventional one, as shown in Figure 1, because it is confirmed to have the highest transfer coefficient. Angle of difference between entrance and exit at the center and periphery are assumed to be π and $\pi/4$ respectively. The geometric structure and the flow pattern adopted in this paper are based on the previous discussion[7]. An arbitrary point on the channel is represented in polar coordinate (r, θ). The origin (r_{min}, 0) is taken at the inner exit of the cold fluid.

The fundamental equations of spiral plate heat exchanger are obtained based on following assumptions.

(1) fluid temperature is uniform in the channel cross section perpendicular to the flow direction

(2) conduction is negligible in the direction of flow

(3) heat transfer coefficient is independent of the curvature of the channel and fluid temperature

(4) physical properties of the fluid are constant in the apparatus

According to the direction of centrifugal force and the location of hot and cold fluid, two different heat transfer coefficients are introduced in the analysis.

Cold fluid ;

$0 \leqq \theta \leqq \pi$

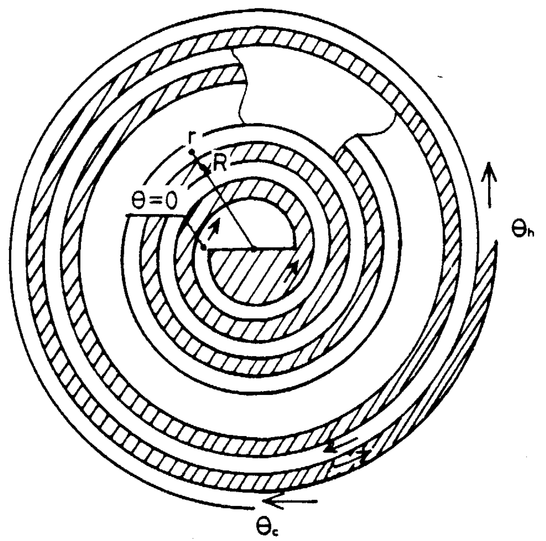

Fig. 1 Flow configuration and coordinate for spiral plate heat exchanger.

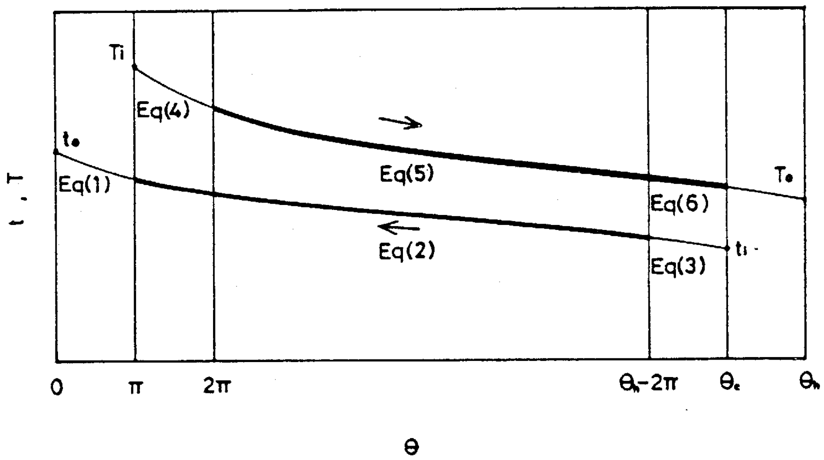

Fig. 2 Temperature profile and the equations.

$$r(\theta)\frac{\partial t(\theta,\xi)}{\partial \xi}=\frac{w}{\rho\varepsilon b}\frac{\partial t(\theta,\xi)}{\partial \theta}+\frac{h_1}{c\rho\varepsilon}\left\{r(\theta)+\frac{\varepsilon}{2}\right\}\left\{T(\theta+2\pi,\xi)-t(\theta,\xi)\right\} \quad (1)$$

$\pi\leqq\theta\leqq(\theta_h-2\pi)$

$$r(\theta)\frac{\partial t(\theta,\xi)}{\partial \xi}=\frac{w}{\rho\varepsilon b}\frac{\partial t(\theta,\xi)}{\partial \theta}+\frac{h_1}{c\rho\varepsilon}\left\{r(\theta)+\frac{\varepsilon}{2}\right\}\left\{T(\theta+2\pi,\xi)-t(\theta,\xi)\right\}$$
$$+\frac{h_2}{c\rho\varepsilon}\left\{r(\theta)-\frac{\varepsilon}{2}\right\}\left\{T(\theta,\xi)-t(\theta,\xi)\right\} \quad (2)$$

$(\theta_h-2\pi)\leqq\theta\leqq\theta_c$

$$r(\theta)\frac{\partial t(\theta,\xi)}{\partial \xi}=\frac{w}{\rho\varepsilon b}\frac{\partial t(\theta,\xi)}{\partial \theta}+\frac{h_2}{c\rho\varepsilon}\left\{r(\theta)-\frac{\varepsilon}{2}\right\}\left\{T(\theta,\xi)-t(\theta,\xi)\right\} \quad (3)$$

Hot fluid ;

$0\leqq\theta\leqq2\pi$

$$R(\theta)\frac{\partial T(\theta,\ \xi)}{\partial \xi}=\frac{W}{\rho\varepsilon b}\frac{\partial T(\theta,\ \xi)}{\partial \theta}+\frac{h_2}{C\rho\varepsilon}\left\{R(\theta)+\frac{\varepsilon}{2}\right\}\left\{t(\theta,\ \xi)-T(\theta,\xi)\right\} \quad (4)$$

$2\pi\leqq\theta\leqq\theta_c$

$$R(\theta)\frac{\partial T(\theta,\xi)}{\partial \xi}=\frac{W}{\rho\varepsilon b}\frac{\partial T(\theta,\xi)}{\partial \theta}+\frac{h_1}{C\rho\varepsilon}\left\{R(\theta)-\frac{\varepsilon}{2}\right\}$$
$$\left\{t(\theta-2\pi,\ \xi)-T(\theta,\xi)\right\}+\frac{h_2}{C\rho\varepsilon}\left\{R(\theta)+\frac{\varepsilon}{2}\right\}\left\{t(\theta,\xi)-T(\theta,\xi)\right\} \quad (5)$$

$\theta_c\leqq\theta\leqq\theta_h$

$$R(\theta)\frac{\partial T(\theta,\xi)}{\partial \xi}=\frac{W}{\rho\varepsilon b}\frac{\partial T(\theta,\xi)}{\partial \theta}+\frac{h_1}{C\rho\varepsilon}\left\{R(\theta)-\frac{\varepsilon}{2}\right\}\left\{t(\theta-2\pi,\xi)-T(\theta,\xi)\right\} \quad (6)$$

Figure 2 shows the temperature distribution in the apparatus and the channel range where correspondiong equatons are applied.

3. Algorithm

Flow chart of the algorithm is shown in Figure 3. Overall heat transfer coefficient U is estimated by author's method where steady state equations are solved numeri-

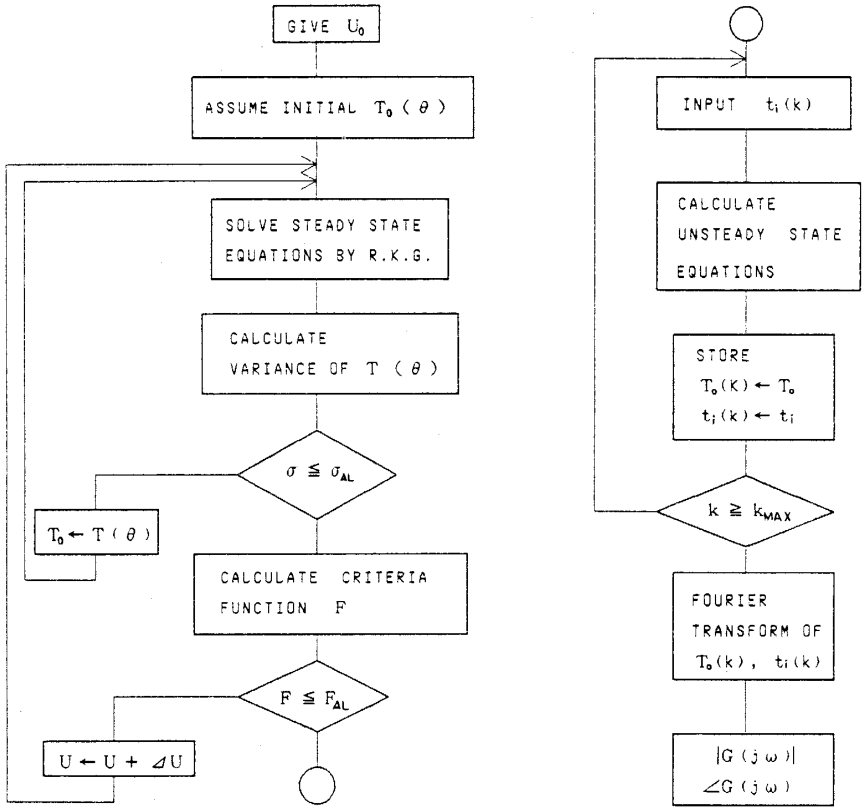

Fig. 3 Flow chart of the algorithm.

cally. The boundary conditions for the equations are obtained from the characteristic chart for effectiveness η and number of transfer units N_A if the operating conditions such as the temperatures of inlet fluids and flowrates are given. The approximate value of U could be also evaluated from the relationship describing the mean temperature difference defined for the heat exchanger of this type. Temperatures are calculated, starting at time zero, towards the direction of θ positive. At the first stage, a steady state temperature distribution is substituted into the unknown temperature term $T(\theta+2\pi)$.

A triangle shape function is used as a input for the inlet temperature of cold fluid t_i. The vertex angle of the function should be small to obtain the accurate results in a high frequency region. Sampled value of this function is substituted into the corresponding term in the equation (4).

Calculated values t_o and T_o are stored into the dimemsions in order of time. Input and output are transformed separately into frequency domain by Fourier transformation to obtain the transfer function of the system.

Accuracy of the estimated value of U affects much to the analysis of dynamic performance. Therefore convergence of the solution of hot fluid temperature in steady state condition is important. Any form of a function can be assumed for the initial temperature profile. In this report, the distribution of a general counter-flow type heat exchanger is used to promote the calculation speed.

Table 1 Heat transfer condition

No.	w	W	t_i	t_o	T_i	T_o	η	N_A	C_R
	(ton/hr)			(℃)				(−)	
1	1.10	1.10	11.68	32.79	42.04	22.49	0.70	2.03	0.99
2	0.87	0.87	11.41	37.27	48.17	23.54	0.70	2.20	1.00
3	0.71	0.71	11.57	42.23	54.97	24.64	0.71	2.37	1.00
4	1.30	0.88	11.37	25.40	39.41	18.20	0.76	2.10	0.67
5	0.54	0.87	11.47	54.76	62.38	36.62	0.85	2.88	0.62
6	1.10	1.64	11.27	31.00	36.88	24.31	0.77	2.14	0.67
7	1.10	0.74	11.41	30.79	47.44	19.21	0.78	2.44	0.67

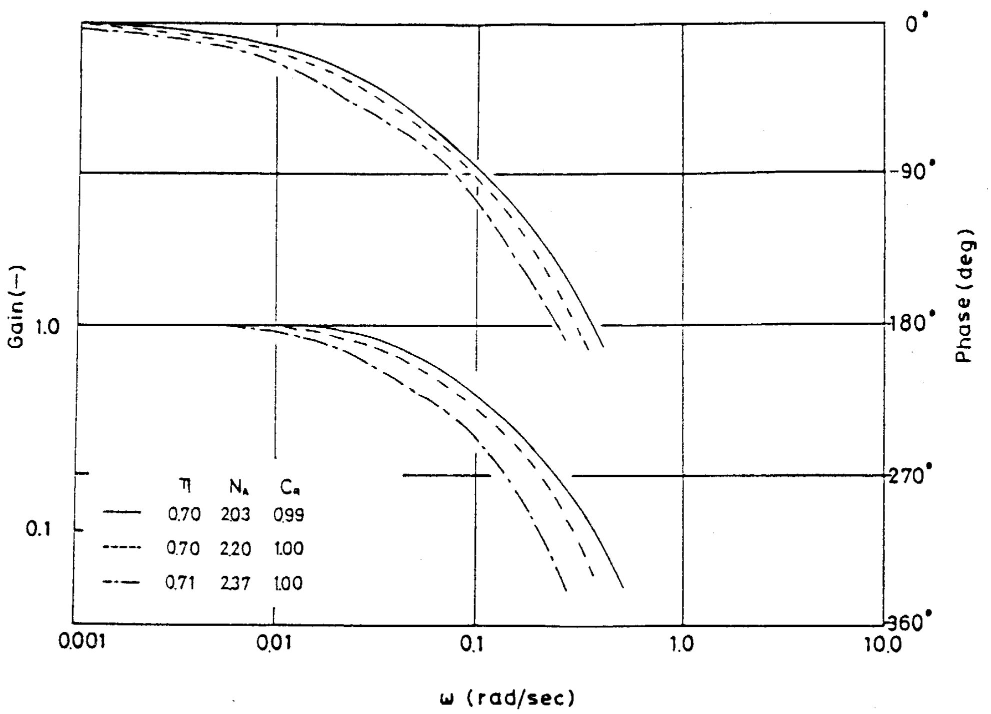

Fig. 4 Response to the change of inlet temperature (1).

4. Results

Flow conditions used for the calculations are shown in Table 1. Figure 4 shows the responses whose flow rate ratios are equal. The equipment responses fast in the high flow rate. System delay increases above the frequency $\omega = 0.1$. Time constant is large at lower flowrate. It indicates that the convection is dominant in the response of the system. Effect of the variation of cold fluid flow rate is compared in Figure 5. In this case, it is clear that the flow rate much affects the behavior of the unit. Phase shift shows same tendency as shown in the previous result. Figure 6 shows the responses where flow rates of cold fluid are constant. Comparing the previous behavior of the system, rapid response is attained as the absolute value of the flow rate is high.

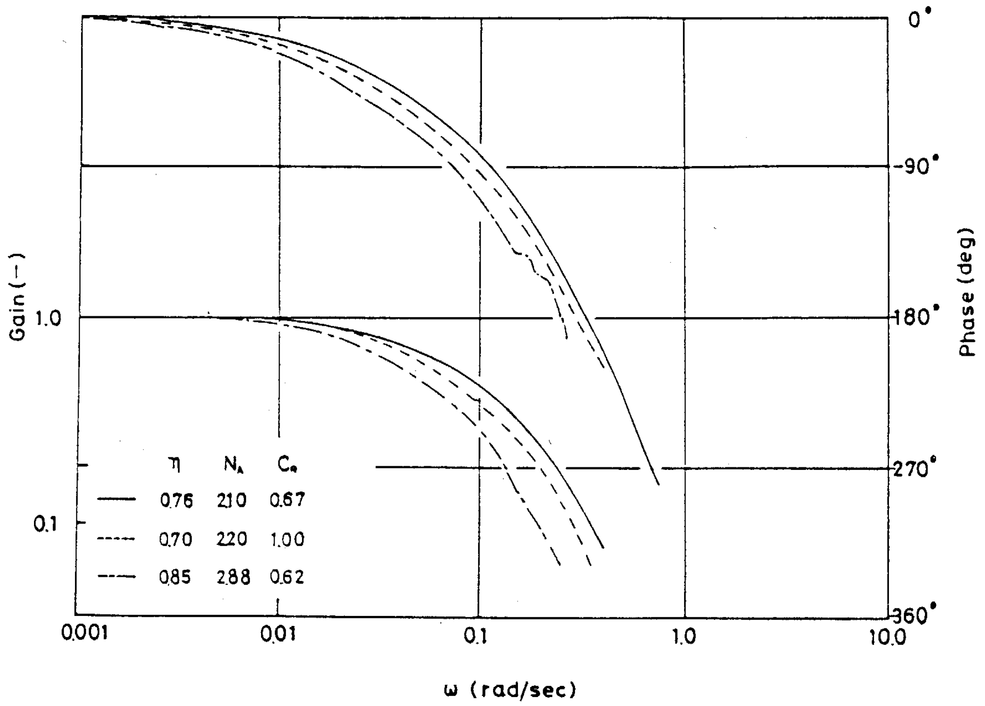

Fig. 5 Response to the change of inlet temperature (2).

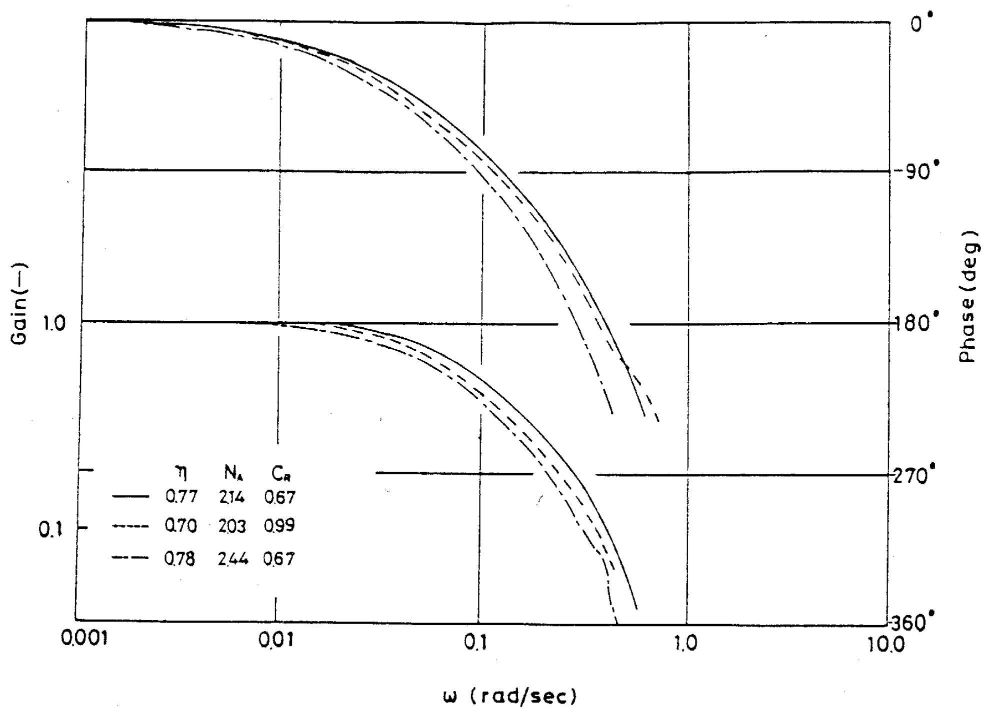

Fig. 6 Response to the change of inlet temperature (3).

Table 2 Residense time and cumulant

No.	τ_c	τ_h [sec]	c_1
1	14.7	14.6	25.6
2	18.5	18.5	31.3
3	22.8	22.9	44.7
4	12.4	18.4	27.5
5	29.7	18.5	39.5
6	14.7	9.9	21.3
7	14.7	21.8	31.1

Responses in the region more than $\omega=0.5$ are not shown because the solutions were unstable due to the property of the input function. Resonance would be observed at the high frequency value for a conventional heat exchanger of counter−flow. For a plate heat exchanger whose heat transfer structure is relatively similar to spiral plate type, it reported that resonance was observed at $\omega=2.0$[8].

As for a practical apparatus, it is obvious that heat capacity of the shell and heat capacity of the passage wall affect the dynamics of the unit[9]. Both of the capacities tend to decrease the resonance at high frequency region. For spiral plate heat exchanger, it seems to show some periodic phenomena owing to the passage structure in which ascending fluid and descending fluid meet every turn. It is predicted that the period will be some function of the flow rate ratio.

Mean delay constant of the system is estimated by the cumulant. Plotting the data q versus ω^2 in a low frequency region, intersection of the function provides the first order cumulant. Based on the results obtained above, as summarized in Table 2, the cumulant is correlated with the residence time τ and overall heat transfer coefficient U to offer a practical measure for the estimation of the value.

$$c_1=57-2.8\tau_c-2.0\tau_h+0.19\tau_c\tau_h-0.0019U \tag{7}$$

5. Concluding Remarks

Unsteady state equations were proposed for spiral plate heat exchanger of the standard type specification of the conformation and flow pattern. Dynamic behavior of the equipment was studied on the basis of the method developed for the estimation of effectiveness. Response to the variation of influx temperature for cold fluid were calculated through the equations directly. The characteristics were discussed in frequency domain. As a first step of the analysis, capacities of wall and shell were neglected. However the effect term should be considered in the fundamental equations directly to improve the model. According to the additional approximate calculation so far, the results did not show difference clearly between them. The range of the data on flow ratio processed was relatively wide. As for the heat transfer unit the values were limited in a narrow region. The rage may be necessary to be spread to assess the properties of the unit more comprehensively.

Nomenclature

A	heat transfer area
b	channel height
C_R	capacity ratio $\equiv \min(\Gamma/\gamma, \gamma/\Gamma)$
c	specific heat
c_1	the first order cumulant
h	heat transfer coefficient
N_A	number of heat transfer unit $\equiv \max(\gamma, \Gamma)$
q	$= -\angle G(j\omega)/\omega$
R	radius of channel for hot fluid
r	radius of channel for cold fluid
T	hot fluid temperature
t	cold fluid temperature
ΔT_{lm}	logarithmic mean temperature difference
U	overall heat transfer coefficient
W	flow rate for hot fluid
w	flow rate for cold fluid
Γ	$\equiv AU/CW$
γ	$\equiv AU/cw$
ε	channel width
η	temperature efficincy
θ	angle
θ_c	maximum angle of turns for cold fluid
θ_h	maximum angle of turns for hot fluid
ζ	time
τ	residence time
ω	frequency

Suffix

i	entrance
o	exit

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