Heat Transfer Characteristics of Rotating Ceramic Regenerators—Numerical Solution Using a Hybrid Finite Difference/Laplace Transform Scheme

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ABSTRACT

A hybrid numerical method, combining finite differences with respect to space and a Laplace transform with respect to time, is proposed to determine the heat transfer in a rotary heat exchanger used as a rotating ceramic regenerator for automotive gas turbines. The temperature distributions of the core and of the working fluids are solved for given boundary and initial conditions of a rotary regenerator using this method. An advantage of the present method is that it can be applied when the core and the working fluids have dissimilar temperature distributions.

The temperature change in the ceramic honeycomb core was determined from start up to periodic steady state operation. The heat exchanger effectiveness was obtained for an extruded ceramic core used in automotive gas turbine applications.

NOMENCLATURE

\( A_s \): heat transfer area (m\(^2\))
\( C \): heat capacity of fluid flow (kW/K) = \( G c_v \)
\( c_v \): specific heat of the core (kJ/kgK)
\( c_f \): specific heat of the fluids (kJ/kgK)
\( D_s \): hydraulic diameter of the matrix (m)
\( F \): defined by Eq. (19)
\( G \): flow rate of the working fluid (kg/s)
\( h \): heat transfer coefficient (W/m\(^2\)K)
\( J \): defined by Eq. (24), or the Colburn J factor (-)
\( L \): length of the regenerator core (m)
\( k \): thickness of the cell wall (m)
\( M \): mass of the regenerative core (kg)

\( m \): number of nodal points \([=1/\Delta X]\)
\( Nu \): Nusselt number (-)
\( N_r \): \([=M c_v R / C]\)
\( NTU \): number of heat transfer units (-)
\( P \): \([=NTU/2NR]\), see Eq. (13)
\( Pr \): Prandtl number of the fluids (-)
\( Q \): defined by Eq. (25)
\( R \): rotational speed of the regenerator core (rps)
\( Re \): Reynolds number (-)
\( S \): defined by Eq. (20)
\( T \): defined by Eq. (12)
\( t \): time (s)
\( X \): dimensionless position \([=x/L]\)
\( x \): longitudinal distance from the entrance of the core (m)
\( \Delta t \): cycle time (s)
\( \varepsilon \): heat exchanger effectiveness (%) 
\( \Theta \): core temperature (K)
\( \beta \): fluid temperature (K)
\( \Lambda \): \([=\Lambda_f A_s/CL]\) (-)
\( \Lambda_f \): thermal conductivity of the fluid (W/mK)
\( \Lambda_m \): thermal conductivity of the matrix (W/mK)
\( \tau \): dimensionless time (-) \([=tR]\)
\( \Omega \): dimensionless cycle time (-)

DEFINED BY Eqs. (40) and (41)

SUBSCRIPTS

c : cooling cycle
h : heating cycle
k : nodal number in matrix

INTRODUCTION

The thermal efficiency of gas turbine engines can be improved by regenerating the thermal energy of the exhaust gas. Rotary
regenerators are currently being developed for this purpose in automotive applications. Since the heat transfer area per unit volume of a rotary regenerator is two to ten times larger than that of a typical recuperator, regenerators are relatively compact and have a high heat exchanger effectiveness of over 90%.

The core of a rotary regenerator is exposed to hot and cold fluids alternately, which causes cyclic thermal stresses due to thermal expansion and contraction during the heating and cooling cycles. In gas turbine applications, a fine honeycomb structure made of such ceramics as MAS (Cordierite), LAS (Lithium-Aluminum-Silicate), and AS (Aluminous-Keatite) is used for the core. Since the inlet gas temperature of the regenerator can be over 1000°C, the ceramic honeycomb structure must have a low thermal expansion coefficient and high thermal stability. The manufacturing process of the ceramic matrix is composed of extrusion, corrugation, and calendaring processes. The extrusion process enables the manufacture of various cell forms with uniform quality, yielding small pressure drops and high heat transfer coefficients. Due to recent improvements in performance and compactness, honeycomb ceramics have obtained the following specifications: wall thickness 0.1mm, specific heat transfer area 5000m²/m³, and cell density 190 x 10⁴ holes/m². Figures 1 and 2 illustrate a honeycomb core manufactured by the extrusion process, along with details of the matrix. Table 1 shows the dimensions and physical properties of the matrix.

Under severe conditions when the inlet gas temperature exceeds 1000°C and the temperature difference between the two fluids can be greater than 800°C, thermal stress relief of the regenerator core is important as well as the thermal stability of the material. It is necessary to know both the periodic steady state and the transient state temperature changes in order to calculate the stress distribution of the core. This paper contributes to the analysis of the temperature distribution in the core during periodic steady state and transient state operation.

Since the temperature in the core and in the fluids changes with time, various methods of analysis have been suggested by many authors [1-11]. It is still difficult, however, to obtain the characteristics of the solution without a lengthy calculation, especially for cyclic heating and cooling. In reference[10], the working equations derived from the basic equations are similar to those of the present method. The solutions of the fundamental energy equation are expressed by two independent variables. In this paper, the independent variable x in space is discrete, but the time t is treated as a continuous variable. This decreases the calculation time and makes the solution easier to understand. We have determined the temperature change for each position of the core, and the heat exchanger effectiveness, from start up to periodic steady state operation. Furthermore, we have obtained the heat transfer characteristics of the matrix by comparing the results of the present analysis with measured data of the core outlet temperature. Details
are described in Appendix 1.

THEORETICAL ANALYSIS

Fundamental Equations

The following assumptions were made for the calculation:

(1) Since the wall is thin, the temperature distribution in the cell wall is assumed to be uniform across the thickness of the wall, and thermal conduction in the flow direction, in both the core and the working fluid, is neglected. (see Appendix 2.)

(2) The heat transfer coefficient between the fluids and the core is constant along the flow direction.

(3) Since the width of the flow channel is very small, the axial line-of-sight from a given point in the channel is limited. Hence, provided the axial temperature gradient is not excessive, radiation originating from upstream approximately balances that from downstream, meaning that radiative transfer can be neglected.

(4) The thermophysical properties are independent of temperature and are equal to the values given in Table 1.

Considering an energy balance for the control volume shown in Fig. 3, the following two partial differential equations are obtained [13]. A heat balance for the core, yields

\[ \text{Mo} \frac{\partial \theta}{\partial t} = A \frac{\partial}{\partial x} (\theta - \Theta) \]  

While a heat balance for the working fluid gives

\[ \text{Cl} \frac{\partial \theta}{\partial x} = A \frac{\partial}{\partial x} (\theta - \Theta) \]  

By solving these equations, the fluid and the core temperatures \( \theta \) and \( \Theta \) are obtained respectively as functions of time \( t \) and position \( x \). Introducing the nondimensional variables of \( x/L \equiv X, A \text{h}/C = \text{NTU}, \text{Moc} \text{R}/C = \text{Nr}, \) and \( \text{Tr} = \tau \), where \( R \) is the rotating speed, Eqs. (1) and (2) are expressed as follows:

\[ \frac{\partial \Theta}{\partial \tau} = \frac{\text{NTU}}{\text{Nr}} (\theta - \Theta) \]  

\[ \frac{\partial \Theta}{\partial X} = -\text{NTU}(\theta - \Theta) \]  

When the finite difference approximation with respect to \( X \) is applied to Eq. (4), the following equation is obtained:

\[ \Delta \Theta \Delta X = \frac{\text{NTU}}{2} \left( (\theta - \Theta)_{k-1} + (\theta - \Theta)_{k} \right) \]  

where \( \theta_k \) and \( \Theta_k \) are functions of time. The error incurred by using the finite difference approximation is discussed in Appendix 3. Then, for \( \theta_k \), we have

\[ \theta_k = \frac{1 - \text{NTU} \Delta X}{2} \theta_{k-1} \]  

\[ + \frac{\text{NTU} \Delta X}{2} (\Theta_{k-1} + \Theta_k) \]  

Equation (5) shows that the maximum value of \( \Delta X \) required for numerical stability is determined from the numerator of the first term, or

\[ \frac{1}{\Delta X} = m = \frac{\text{NTU}}{2} \]  

Therefore, for \( m = \text{NTU}/2 \), \( \theta_k \) is given as:

\[ \theta_k = \frac{1}{2} (\Theta_{k-1} + \Theta_k) \]  

The arrangement of the calculation nodes of \( \Theta \) and \( \Theta \) are shown in Fig. 4. Substituting the relationship in Eq. (7) into Eq. (3) results in

\[ \frac{d \Theta_k}{d \tau} = \frac{\text{NTU}}{\text{Nr}} \left[ \frac{1}{2} (\Theta_{k-1} + \Theta_k) - \Theta_k \right] \]  

\[ = \frac{\text{NTU}}{2 \text{Nr}} (\Theta_{k-1} - \Theta_k) \]  

Similarly, for \( \theta_k \), we have

\[ \frac{d \theta_k}{d \tau} + \frac{\text{NTU}}{2 \text{Nr}} \left( \theta_k - \frac{\text{NTU}}{2 \text{Nr}} \theta_{k-1} \right) = 0 \]  

Fig. 3 Calculation model

Fig. 4 Arrangement of \( \Theta \) and \( \Theta \) nodes

Finally, the following equations are obtained:
\[
\frac{d\theta_k}{dT} + \theta_k - \theta_{k-1} = 0
\]  
(10)
\[
\frac{d\theta_0}{dT} + \theta_0 - \theta_{-1} = 0
\]  
(11)
where
\[
T = P \tau = \frac{NTU}{2N_r} \cdot t \quad R = \frac{A_{th}}{2Mc_m} \cdot t
\]
(12)
\[
P = \frac{NTU}{2N_r}
\]
(13)

Analytical Method

By assuming a stepwise change in working temperature, the Laplace transform of Eq.(11) is
\[
Y(T) = E(T)Y_{0,c} + \theta_{0,c}F(T)
\]  
(14)
where
\[
E(T) = \begin{pmatrix}
F_1 & 0 & \cdots & 0 & 0 & 0 \\
F_2 & F_1 & \cdots & 0 & 0 & 0 \\
\vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\
F_{m-1} & F_{m-2} & \cdots & F_1 & 0 & 0 \\
F_m & F_{m-1} & \cdots & F_2 & F_1 & 0 \\
\end{pmatrix}
\]  
(15)

\[
Y_c(T) = \begin{pmatrix}
\theta_1 \\
\theta_2 \\
\vdots \\
\theta_{m-1} \\
\theta_m \\
\end{pmatrix}
\]
\[
Y_{0,c} = \begin{pmatrix}
\theta_0 \\
\theta_{0,1} \\
\vdots \\
\theta_{0,m-1} \\
\theta_{0,m} \\
\end{pmatrix}
\]
\[
F(T) = \begin{pmatrix}
S_1 \\
S_2 \\
\vdots \\
S_{m-1} \\
S_m \\
\end{pmatrix}
\]
(16)  
(17)  
(18)

On the other hand, for $k=0$, Eq.(10) is written as follows:
\[
\frac{d\theta_0}{dT} + \theta_0 = \theta_{-1}
\]  
(21)

On the right-hand side of the above equation, $\theta_{-1} = 2\theta_0 - \theta_1$ (from Eq.(7)). From that we get
\[
\frac{d\theta_0}{dT} + 2\theta_0 = 2\theta_0
\]  
(22)

The solution of Eq.(22) is
\[
\theta_0(T) = \theta_0 e^{-2T} + \theta_0 \left(1 - e^{-2T}\right)
\]  
(23)

From Eq.(7), $\theta_1(T)$ can be expressed as
\[
\theta_1(T) = 2\theta_0 \left(J_1(T)ight) - \theta_1(T)
\]  
(24)

Substituting Eqs.(14) and (23) into the right-hand side of the above equation, we get
\[
\theta_1(T) = 2 \left(\theta_0 J_0(T) + \theta_0 \left(1 - \theta_0 e^{-2T}\right)\right)
\]  
(25)

Also from Eq.(7), $\theta_{m-1}$ can be expressed as
\[
2\theta_{m-1} = \theta_{m-1} + \theta_m
\]

Thus, the formula for $\theta_1(T)$ is rearranged as follows:
\[
\theta_1(T) = \theta_0 \left(2\theta_1(T) - J_1(T)\right) + \theta_0 J_0(T)
\]  
(26)

In the same way, $\theta_{k}(T)$ for $k=0$ to $m$ during the cooling period is obtained:
\[
X_c(T) = A(T)X_{0,c} + \theta_{0,c}B(T)
\]  
(27)
where

\[
A(T) = \begin{pmatrix}
\begin{array}{ccccccc}
J_1 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 \\
J_2 & F_1 & 0 & 0 & \cdots & 0 & 0 & 0 \\
J_3 & F_2 & F_1 & 0 & \cdots & 0 & 0 & 0 \\
& & & & & \ddots & & \\
J_{n-1} & F_{n-2} & F_{n-3} & F_{n-4} & \cdots & F_2 & F_1 & 0 \\
J_n & F_{n-1} & F_{n-2} & F_{n-3} & F_{n-4} & \cdots & F_2 & F_1 & 0 \\
\end{array}
\end{pmatrix}
\]

\[ (28) \]

\[
X_{\ell}(T) = \begin{pmatrix}
\Theta_0 \\
\Theta_1 \\
\Theta_2 \\
\vdots \\
\Theta_n \\
\end{pmatrix} \quad \begin{pmatrix}
X_{\ell}(T) \\
X_{\ell+1}(T) \\
X_{\ell+2}(T) \\
\vdots \\
X_{\ell+n}(T) \\
\end{pmatrix} = \begin{pmatrix}
Q_0 \\
Q_1 \\
Q_2 \\
\vdots \\
Q_n \\
\end{pmatrix}
\]

\[ (29) \quad (30) \]

\[
B(T) = \begin{pmatrix}
\Theta_0 \\
\Theta_1 \\
\Theta_2 \\
\vdots \\
\Theta_n \\
\end{pmatrix} \quad \begin{pmatrix}
Q_0 \\
Q_1 \\
Q_2 \\
\vdots \\
Q_n \\
\end{pmatrix}
\]

\[ (31) \]

During the heating period, \( \Theta_{\ell}(T) \) is also calculated as follows:

\[
X_{h}(T) = C(T)X_{\ell,h} + 0_{b,h}D(T)
\]

\[ (32) \]

As shown in Fig.5, the cooling and heating fluids flow counter to each other in the core, and thus \( C(T) \), \( X_{\ell,h} \), \( D(T) \) are inversely rearranged rowwise to the elements of \( A(T) \), \( \Theta_{\ell,c} \), \( B(T) \).

Even if an arbitrary initial temperature distribution of the core is assumed, after sufficient time from the start, the temperature of each point of the rotating core changes periodically (see Fig.6). In the periodic steady state, the temperature distribution does not depend on the initial core temperature distribution, but rather on the inlet temperature of the heating and cooling fluids, the rotational speed, and the ratio of the duration of the cooling and heating periods.

**Simplified Calculation of Periodic Steady State**

When considering the core temperature \( \Theta_{\ell} \) in the periodic steady state, the initial core temperature \( X_0 \) in each period can be calculated as given below.

Taking the durations of cooling and heating periods to be \( T_h \) and \( T_c \), respectively, \( X_{\ell}(T_0) \) equals the initial value of the heating period in Eq. (32). Considering that \( A(T), X_{\ell,c}, \) and \( D(T) \)
are inversely rearranged rowwise to \( C(T) \), \( X_{0,\text{c}} \), and \( D(T) \), we get
\[
X_c(T_c) = A(T_c)X_{0,\text{c}} + \theta_{0,h}B(T_c)
\]  
whose rowwise inverse matrix \( X_c^{-1}(T_c) \) is
\[
X_c^{-1}(T_c) = C(T_c)X_{0,\text{c}} + \theta_{0,h}D(T_c) = X_{0,h}
\]  
Substituting Eq. (37) for Eq. (32) gives
\[
X_h(T_h) = C(T_h)[C(T_c)X_{0,\text{c}} + \theta_{0,c}D(T_c)]
\]
\[+ \theta_{0,h}D(T_h) = X_{0,c} \]  
Finally, \( X_{0,c} \), which represents the initial core temperature in the cooling period, is derived from Eq. (39):
\[
[I-C(T_h)C(T_c)]X_{0,c} = \theta_{0,c}C(T_c)D(T_c)+\theta_{0,h}D(T_h)
\]
where \( I \) is the unit matrix.

By substituting the solution for \( X_{0,c} \) in Eq. (39) into Eq. (27), the core temperature distribution in the cooling period is determined. The values of the core temperature when \( T=T_c \) in Eq. (27) are the initial core temperatures during the next heating period. On the other hand, the initial temperature distributions of the working fluids can be calculated by Eq. (7).

The temperature of each point of the core at the end of the heating period is equal to that at the beginning of the cooling period, and vice versa. When the inlet fluid temperatures \( \theta_{\text{in}} \) and \( \theta_{\text{out}} \) and the cooling and heating time periods \( \Delta t_c \) and \( \Delta t_h \) are given, the heat transfer performance of the regenerator is determined after a sufficient time from the start. In this condition, the total amount of heat exchanged during the cooling period equals that applied during the heating period.

An additional advantage of the present hybrid method is the ability to calculate the periodic steady state temperature distribution without having to calculate the initial transients. This differs from ordinary numerical methods which calculate the solution as it evolves from the initial condition. Thus, it is possible to achieve significant computational savings through the application of this hybrid finite difference / Laplace transform approach. The use of this simplified calculation, and further exploration of the actual computational savings, will be presented in a future publication.

Results and Discussion

The above procedure is applied to predict the performance of a regenerator mounted in an automotive gas turbine.

The calculations were carried out for the following conditions:

Core size: outer diameter 480 mm  
thickness 85 mm

Rotating speed of the core: 20 rpm

Quantity of flow: \( G_c = G_h = 0.60 \text{ kg/s} (\text{NTU}=10) \)
\( = 0.43 \text{ kg/s} (\text{NTU}=14) \)

Dimensionless inlet air temperature
at the hot side: \( \theta_{\text{in}} = 1.0 \)
at the cold side: \( \theta_{\text{in}} = 0 \)

Dimensionless initial core temperature: \( \theta_{0} = 0.5 \)

Matrix data: refer to Table 1

The nondimensional heating and cooling periods are given by
\[
\Omega_c = P_c \cdot \Delta t_c = P_c \cdot R \cdot \Delta t_c
\]
\[
\Omega_h = P_h \cdot \Delta t_h = P_h \cdot R \cdot \Delta t_h
\]

The following values of NTU and \( \Omega \) are adopted for the calculation:
\[ \text{NTU} = \text{NTU}_c = \text{NTU}_h = 10 (m=5), 14 (m=7) \]
\[
\Omega = \Omega_c = \Omega_h = 0.5, 1.25
\]

Figure 7 shows the temperature change at each position in the core (\( \theta_c - \theta_c') \), the compressed air temperature (\( \theta_{\text{in}} - \theta_{\text{in}}') \) and the exhaust gas temperature (\( \theta_c - \theta_c' \)), from start up to periodic steady state. When \( \Omega \) is small, changes in the temperature are small and more cycles are required until periodic steady state is established. Table 2 shows the heat exchanger effectiveness \( \varepsilon \) for periodic steady state and the number of cycles \( N_{\text{cy}} \) required for periodic steady state to be established. In the calculation, periodic steady state is considered to be established when the core temperature at the start of heating or cooling is within 0.1\% of the temperature in the previous cycle. The effectiveness \( \varepsilon \) is calculated from the following equation:
\[
\varepsilon = \frac{\theta_{0,h}-\overline{\theta}_h}{\theta_{0,h}-\theta_{0,c}} = \frac{\overline{\theta}_c-\theta_{0,c}}{\theta_{0,h}-\theta_{0,c}} \left( \frac{\Omega_c}{\Omega_h} \right) \left( \frac{C_c}{C_h} \right)
\]  
where \( \overline{\theta}_c \) and \( \overline{\theta}_h \) are the average exit temperatures of the fluids:
\[
\overline{\theta}_c = \frac{1}{\Omega_c} \int_0^\Omega \theta_{\text{m,c}}(T) dT
\]
\[
\overline{\theta}_h = \frac{1}{\Omega_h} \int_0^\Omega \theta_{\text{m,h}}(T) dT
\]
As shown in Fig. 8 and Table 2, the number of cycles required for the core to attain a periodic steady state temperature distribution increases as \( NTU \) increases and \( \Omega \) decreases. The physical explanation is that when \( NTU \) increases, which means that \( A_h \) increases or \( C \) becomes smaller, the amount of heat exchanged between the fluid and the core becomes smaller. Therefore, the heat exchanged between the fluid and the core is accomplished in a shorter distance from the entrance of the core, meaning that a longer time is required for the influence of the cooling or heating to reach the center of the core. Similarly, when \( NTU \) is small, the regions affected by heating and cooling during one cycle become larger, and the transient phenomena are shortened. As shown in Fig. 8, when \( \Omega \) becomes smaller, \( \varepsilon \) increases, and it becomes maximum when \( \Omega = 0 \). However, if the rotating speed is increased \((\Omega > 0)\), the carryover of pressurized air to exhaust gas also increases, and the total efficiency of the gas turbine is reduced. For this situation, an optimized rotating speed exists. The effect of carryover on rotary heat exchanger effectiveness is outside our discussion in this paper, and therefore this problem should be treated in the future.

Figure 8 also shows a comparison between the present calculated values of \( \varepsilon \) and previous experimental data [9,10,14].

As the next step of our research, experiments will be carried out using actual cores, and the core and fluid temperature changes will be measured. These results will be compared with the analytical solution presented in this paper. We will also analyze the effects of the initial core temperature distribution, and of unequal \( NTU \) values of the cooling and the heating sides.

**CONCLUSIONS**

After finite differentiating the fundamental heat transfer equations of a rotating regenerator with respect to space, and taking
the Laplace transform of the simultaneous differential equations with respect to time, the equations were solved to analyze the performance of a rotating ceramic regenerator suitable for automotive gas turbines. This analytical method can be applied when the core and working fluids do not have a uniform initial temperature distribution. An approximate periodic steady state solution is obtained, and the transient temperature distribution of the core and of the working fluid during the operation of the regenerator are illustrated. Using the given characteristic values of the regenerator (Ash/Mc, Mc, and NTU), the core and fluid temperatures at each point during the cooling and heating periods are calculated.

The effectiveness obtained from the numerical calculation for typical operating conditions for an automotive gas turbine are obtained. The accuracy of the present numerical method is confirmed by comparison with a previous analysis and with previous experimental results.

REFERENCES

APPENDIX

APPENDIX 1.

COMPARISON OF THIS METHOD WITH THE MEASURED HEAT TRANSFER CHARACTERISTICS OF THE MAS REGENERATOR CORE [15]

This Appendix presents experimental results relevant to our own regenerator core, i.e., that shown in Figs. 1 and 2.
EXPERIMENTAL EQUIPMENT AND PROCEDURE

(1) Heat transfer in the core may be evaluated by the dimensionless Colburn J factor \( J \). The parameter \( J \) is obtained by measuring the changes in the exit air temperature, when laminar flow at constant temperature is introduced into the core and the inlet air flow is heated or cooled in a stepwise fashion. The methods available for measuring \( J \) includes the shuttle rig measurement\[16\] and the transient rig measurement\[17\]. The experiment was carried out on the test piece whose characteristics are shown in Table 1 in the paper. The experimental procedure was similar to that of Reference \[18\] which also adopted the single-blow transient method.

(2) Calculation of the heat transfer coefficient

Temperature data are taken every second and are processed by a computer; NTU is obtained from the best-fit theoretical value with the experimental result.

The following Eqs. (45) and (46) were used to nondimensionalize time and temperature. Since the core was stationary in the experiment, the definition of dimensionless time \( \tau \) is different from that in the nomenclature, as shown in Eq. (46):

\[
\tau = \frac{1}{M \cdot c_m/(G \cdot c_p)} 
\]

(46)

Here, \( \theta_0 \) and \( \theta_1 \) are the air temperature before and after heating, \( \theta^* \) is the exit air temperature of the test piece, and \( t \) is the measurement time. In this experiment, \( m=10 \) is assumed. The heat transfer coefficient \( h \), the Nusselt number \( Nu \), and the \( J \) factor are calculated using Eqs. (47) to (49) from the obtained NTU:

\[
h = NTU \cdot G \cdot c_s/ A_s 
\]

(47)

\[
Nu = h D_s / \lambda_s 
\]

(48)

\[
J = Nu \cdot (Re \cdot Pr^{1/3}) 
\]

(49)

where \( \lambda_s \) is the thermal conductivity of the passing air (W/mK) and \( Pr=0.71 \) at room temperature.

In a previous report, NTU was calculated from the maximum slope of the temperature rise curve \[12\]. In this experiment, NTU was obtained by both the maximum slope method and the present method described above, and the results were compared.

TEST RESULT

(1) Comparison of the measurement data with the \( \tau-\theta \) curve

Figure 9 shows an example of the exit air temperature rise curve at the exit of the core. The straight line in the figure shows the maximum slope of the temperature curve for calculating NTU. The temperature of the exit air is read from the obtained temperature rise curve, and then compared with the \( \tau-\theta \) curve used in present paper. Figure 10 shows an example of the computer-processed \( \tau-\theta \) curve. By selecting the most appropriate value of NTU, the experimental temperature variation can be described completely by the analytical results.

(2) Calculation of NTU and comparison with the maximum slope method

The parameter NTU was obtained from the data measured at various Re numbers. Table 3