가스터빈 리제너레이터 내부유동에 관한 수치해석적 연구

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Numerical Investigation of Flowing Process for Regenerative Heat Exchanger of a Gas Turbine Engine

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ABSTRACT

A distributed nonlinear mathematical model for investigation of regenerative heat exchangers of both a continuous and periodic operation is described in the paper. The non-iterative numerical integration scheme for conjugate unsteady heat exchange problem of one dimensional flows and two dimensional matrix wall conductivity is developed. Case study of a regenerative heat exchanger with a rotary ceramic matrix is presented. The range of optimum rotation rates of the regenerator providing the greatest calorific efficiency is determined.

초 록

본 논문에는 열교환기로서의 재색기내부에서 발생하는 주기적 및 연속적 유동을 조사하기 위한 분산, 비선형 수학적 모델이 제시되어 있다. 일차원 유동 및 2차원 격벽 내 비정상 열교환 문제를 해결하기 위한 비 반복적 수치적분 방법이 개발되었다. 제안된 방법에 대한 검증을 위하여 회전 세라믹 매트릭스 재생기에 대한 예가 제시되었으며 재생기가 최고의 열역학적 효율을 각기 위한 최적의 회전수를 계산하였다.

Key Words: Gas Turbine Engine(가스터빈엔진), Regenerator(재생기), Matrix(열), Mathematical Model(수학적모델), Convective Heat Transfer(대류전열), Conductivity(전도)

1. INTRODUCTION

Heat released from one heat carrier (combustion product) during the process of heating up of a regene- rator system will be absorbed by channel wall matrix or be transferred to other heat carrier such as air during the cooling process. The most optimum heat transferring process can be perceived by two basic types of regenerators such as (a) and (b) of Fig. 1. A continuous and periodic

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c	specific heat (J/(kg K))
D	rotor diameter (m)
d	hydraulic channel diameter $=4S/\Pi$ (m)
G	mass flow rate (kg/s)
Gz	Graetz number $=Re\ Pr\ d/L$
h	convective heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/mK
L	channel length (m) or Coefficient of TDMA
Nu	Nusselt number $=\alpha d/\lambda$
n	unit vector, normal to the surface
Pr	Prandtl number $=\nu/a=\mu c_p/\lambda$
p	pressure (Pa)
Q	heat flow rate (W)
q	heat flux, heat transfer rate/unit surface area (W/m^2)
R	coefficients of TDMA
Re	Reynolds number $=wd/\nu$
S	channel cross sectional area (m^2)
t	temperature of air (°C)
w	z-direction velocity (m/s)
y	Cartesian coordinate perpendicular to flow
z	Cartesian coordinate parallel to flow

Greek

Δ	difference and grid step
Δp	pressure drop (loss) (Pa)
Δy	y-direction unit width of matrix (m)
Δz	z-direction unit length of matrix and channel (m)
$\Delta \tau$	time step (s)
δ	channel wall thickness(m) or relative mean
ε	relative cross section of matrix
ζ	distributed total friction
η	heat exchanger effectiveness
θ	temperature of matrix wall (°C)
ϑ	temperature of gas (°C)
μ	dynamic viscosity coefficient of the fluid (Pa-s)
ν	kinematic viscosity, μ/ρ (m^2/s)
Π	perimeter of channel (m)
ρ	density (kg/m^3)
σ	weighting factor
τ	time variable (s)
ω	rotational speed of regenerator (rev/min)

Subscripts

a	air, time of air flow
b	Beginning
c	period of cycle
g	gas, time of gas flow
in	Inlet
I	total number of nodes in z-direction
i	grid node numbers in z-direction
J	total number of nodes in y-direction
j	grid node numbers in y-direction
m	matrix material
out	outlet
p	for specific heat at constant pressure
s	for time under sealing or switching
1	for grid weight on time
2-4	for grid weight in directions of y and z
±1/2	half-integer index

Superscripts

max	maximum
n	time steps
*	intermediate value of θ
1	for segregate channel

Abbreviation

MM	mathematical model
TDM	Tri Diagonal Matrix Algorithm

operation. In the former case, the cylindrical matrix and flow distributing and collecting part rotates with a constant speed with respect to each other. Heat carriers (gas and air) flow through them exchanging heat continuously. The typical example of these type of regenerators are those of Ljungstrom with a rotating matrix (fixed sectors) and regenerator of Rothemulle with a fixed matrix (rotating sectors). In the latter type, regenerator matrix (reversible heat accumulator) is motionless and fills the casing volume through which the heat-carriers flow periodically. Very high performance characteristic and self-cleaning capability belong to these type of regenerators. The matrix volume of these regenerative heat exchangers is much less than that occupied by heat surfaces of recuperative heat exchanger[1,3] if other conditions are maintained the same. As matrix materials for regenerators, ceramics and steel alloy are used under high temperatures or severe operating mediums, low-alloy and carbon steels are used at mean and low temperatures, whereas plastic, paper, wool and other nonmetallic materials are normally used at temperatures lower than 65 ℃.

The regenerators with continuous operation have found broad applications such as air heaters of transport gas turbine units or boilers, heat exchangers for air conditioning and ventilation for buildings, or heat exchange cycle for refrigerator, heat pump and as drying mechanism for plants[1,4]. They ensure constancy in outlet gas and air temperatures. Disadvantages of this type of regenerators as energy unit are the

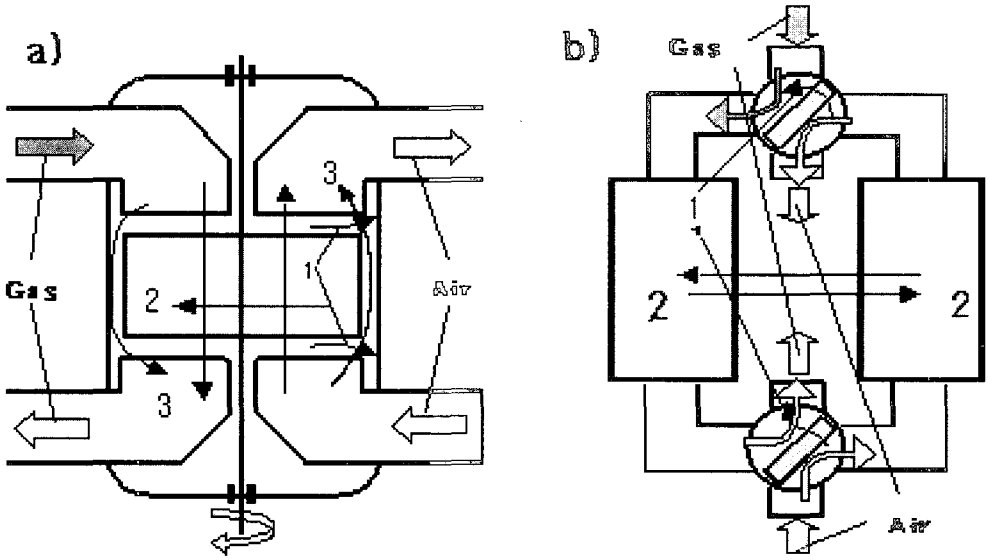

Fig. 1 Schemes of (a) continuous and (b) periodic operation regenerators

significant air leakages, complexity of the rotor driving mechanism and aggravation in performances with low quality fuels. In order to reduce the heat carrier leakage during rotation, radial, peripheral and axial seals are used. The relative rotating speeds vary from 0.5 - 3.0 rev/min for regenerators of large boilers with a rotary diameter 10m to 18m whereas vehicular gas turbine engine regenerators have rotary diameter 0.6 m with small rotary speeds. The periodic operation regenerators are more often used in air heating for different industrial furnaces and metallurgical industry[1-7]. The main advantage of this mechanism is the simplicity in construction of matrix and seals implemented by separate offset elements. It is also necessary to note reduced leakages and lenient requirements for fuel grade. The main disadvantage of such regenerators is the cyclic change in temperature of exit gas and air.

Traditionally regenerators are calculated on quasi-static relations with the use of semiempirical unsteady coefficient[1-7]. For most of the approximated analytical mathematical models (MM) [8-13], assumptions of a constant heat transfer coefficient and thermal properties for heat carriers and matrix, neglection of heat conduction in matrix in the direction of flow and infinite heat conduction in the direction perpendicular to flow are generally accepted. The concepts introduced can be used conveniently during initial design stage of regenerator. The present MM, in general, is oriented to resolve research calculations and, accordingly, differs from the conventional methods. Previous works of Bachnke and Shah et al. [14,15] took into account the importance of longitudinal heat conduction, and in Willmot and Heggs [16,17] transverse heat conduction is considered. Chen et al[18] took into account the finite value of heat conduction in both directions. In the work of Willmot and Zepei, a conventional approach to account for the matrix heat conduction with heat accumulated in heat carrier streams[19,20] is considered. Such an accounting is necessary, for example, in modelling the regenerator of Stirling engines[13]. The classification of regenerator leakages and method of its calculation are given in the work of Shah et al[21]. Nevertheless, followings are not taken into account in the literatures introduced here such as:

1) heat conduction in matrix parallel or perpendicular to streams directions;

2) distribution of local heat transfer coefficients though the channel length;

3) temperature dependence of physical characteristics of both heat carriers and matrix material;

4) losses of carryover in matrix and heat carrier leakages in seals or switches:

5) heat accumulation in heat carriers streams;

6) accounting the time for the channel under seals or switching of channels.

The numerical MM introduced in the paper takes into account the indicated effects and allows to model directly the processes for both types of regenerators. It is possible to use the present MM in the analysis of more complicated technical task of energy system.

2. PHYSICAL MODEL

In Fig. 1(a), a regenerator scheme of continuous operation is introduced. The arrows mark possible places of leakages or carryover of heat-carriers:

1- air in gas due to a large difference in pressures,

2- air in gas and gas in air, considering acquisition of heat carrier in the channels of a rotating matrix under seals, so-called carryover losses,

3- leakages of air and gas near the matrix.

In Fig. 1(b), regenerator scheme of periodic operation is presented.

1- Leakages of the dispenser in switching of streams

2- carryover at the time of heat carrier changing.

Tip and collector effects in distribution and collecting of heat carriers, dissimilarity in flow sections and thickness of matrix walls, defects and contamination in channels and dividing walls, heat conduction in circumferential direction in matrix material (only for regenerator with continuous operation) were neglected. It is also assumed that the considered types of the regenerator can be investigated on an example of the single elementary channel of constant cross-section S with perimeter Π, equivalent diameter d, length L and thickness of wall δ (see Fig. 2(a)). The outer surface of this wall (by virtue of a symmetry) is thermally isolated. Thermal capacity c_m and heat conduction λ_m of the channel wall depend on local temperature $\theta(y, z)$, where y is the direction for matrix thickness, and z is the length of channel contiguous to the direction of gas stream. During time τ_g, a constant gas flow G_g in passes through the regenerator. The gas enters the channel with flow rate $G^1_{g\,in}$, temperature ϑ_{in} and pressure $p_{g,in}$. Local heat transfer coefficient h_g from gas to wall (and the gas pressure losses $d\Delta p_g$) on an element dz of the channel is determined by local temperature ϑ and speed w_g of the gas. Speed w_g varies along the channel length. Heat conductivity k_g, heat capacity c_{pg} , density ρ_g and kinematic viscosity ν_g are determined by of parameters gas flow in the channel. After the

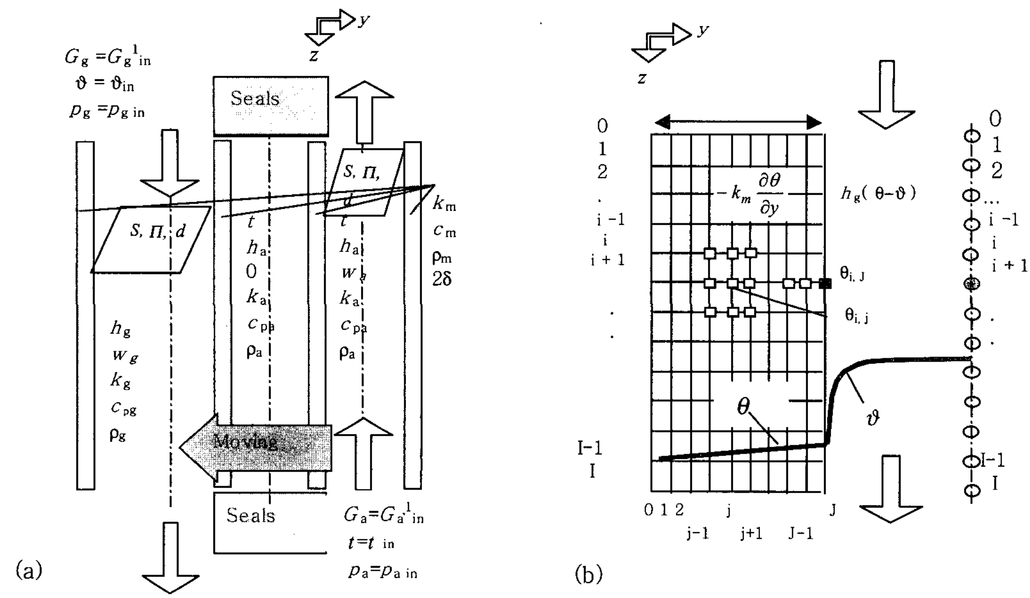

Fig. 2 Channel element (a) and integration grid area (b) for matrix and stream

time τ_g from the beginning of process the gas in the channel does not move during the course of time τ_s. The time τ_s corresponds to a period when channel is placed under a seal (for continuous operation regenerators) or the time of switching the valves (for periodic-flow regenerators). The channel wall temperature happens to be equal by this time.

In time $\tau_g+\tau_s$ the channel heads in the direction opposite to the motion of gas, and air with inlet parameters of $G^1_{a\ in}$, t_{in} , $p_{a\ in}$ will be distributed along the length with flow properties h_a, k_a, c_{pa}, ρ_a. This time the channel wall will be cooled and gives heat back to the air. In time $\tau_g+\tau_s+\tau_a$, the flow of air stops at time τ_s and at this time the channel sits again under the seal or switching of the valve. After the cycle with period of $\tau_c = \tau_g+\tau_s+\tau_a+\tau_s$ is completed then all the process repeats again.

3. MATHEMATICAL MODEL

The process of a heat exchange in a regenerator can be described by a set of energy equations for flow stream and heat conduction equation for wall with appropriate initial and boundary conditions, interface and closing relationships. The equations of energy for both gas and air along the channel, when the flow rate being constant in time, for most cases of a reverse flow, will look like:

$$\begin{cases} S\rho_g c_{pg}\dfrac{\partial \vartheta}{\partial \tau} + G_g c_{pg}\dfrac{\partial \vartheta}{\partial Z} = \Pi \cdot h_g(\theta - \vartheta); \\ S\rho_a c_{pa}\dfrac{\partial t}{\partial \tau} + G_a c_{pa}\dfrac{\partial t}{\partial Z} = \Pi \cdot h_a(\theta - t) \end{cases} \quad (1)$$

where, for the cycle period c,

$$Z = \begin{cases} z, & by \quad 0 < \tau \le \tau_g + \tau_s \ ; \\ L - z, & by \quad \tau_g + \tau_s < \tau \le \tau_c \end{cases}$$

$$\begin{cases} h_g(\theta - \vartheta); \\ h_a(\theta - t). \end{cases}$$

The expression for a specific heat flux q is written here for future use in the formulation of boundary conditions. Heat conduction equation for matrix will be,

$$\rho_m c_m \frac{\partial \theta}{\partial \tau} = div(k_m grad\, \theta). \qquad (2)$$

The differential operator on the right side of the equation (2) is recorded in cartesian or cylindrical coordinates if continuous operation process of regenerator is to be described, or in spherical coordinates if the calculation is conducted for a sphere filling of the periodic operation regenerator. The integration area of a heat conduction equation represents a rectangle in the former case, whose width is half the wall thickness, and the height is same of matrix (length of the channel). In the latter case, this area consists of a row sequentially located sub- domains representing single spheres. The intensity of a local heat exchange of gas or air and matrix walls is determined by a heat flux term, q, which is included in right side of energy equations. It is simultaneously taken into account in the heat conduction equation with a third kind boundary conditions for internal surface of the channel or for surface of spheres:

$$-k_m \frac{\partial \theta}{\partial y} \Big|_{y=0} = q \qquad (3)$$

The influence of edge effects on the temperature field along the length of matrix is considered to be small. For upper and lower boundaries of the area, mid line of wall thickness of matrix or axes of spheres filled is regarded as adiabatic boundary conditions and are written with respect to the appropriate normal n,

$$-k_m \frac{\partial \theta}{\partial n} \Big|_{z=0,L}^{y=0} = 0 . \qquad (4)$$

The set of equations (1), (2), in addition to interface conditions (3)with external boundary condition (4), will close the equations. For a local heat exchange coefficient of gas (or air) and wall,

$$h = h\,(Nu,\ Gz,\ z/d,\ \text{geometry of } S), \qquad (5)$$

where, Nu is the Nusselt number and $Gz = Re\cdot Pr\cdot d/L$ represents the Graetz number. The conditions for heat carriers are,

$$\rho_g = \rho_g(p_g, \vartheta), \quad \rho_a = \rho_a(p_a, t), \qquad (6)$$

and physical properties of matrix will be as,

$$c_m = c_m(\theta), \quad k_m = k_m(\theta). \qquad (7)$$

for physical of properties of heat carriers are

$$c_{pg} = c_{pg}(\vartheta),\ k_g = k_g(\vartheta),\ \nu_g = \nu_g(\vartheta),$$
$$c_{pa} = c_{pa}(t),\ k_a = k_a(t),\ \nu_a = \nu_a(t) \qquad (8)$$

The local values of heat transfer coefficients are calculated based on channels cross-section (circular, rectangular, triangular) or diameters of sphere and its packing relations as recommended in works [1,2,5, 22] for laminar, transient and turbulent flow conditions. It is recommended for both thermally developing and fully developed flow with uniform heat flux. Aerodynamic drops are determined with the help of distributed friction factor along the length, which takes into account the availability of local resistances and flow acceleration.

$$\mathrm{d}\,\Delta p = \zeta G^2\, dZ/(2\rho S^2). \qquad (9)$$

The initial condition for the equation set (1) and (2) in a quasi-stationary state is determined using a method called installation of the solution. Initially before the calculation, the heating from a cold state is considered constant:
-for equation of energy,

$$\vartheta(0,z) = \vartheta_{\mathrm{beg}}, \quad t(0,z) = t_{\mathrm{beg}}; \qquad (10)$$

- for heat conduction equation,

$$\theta(0,y,z) = \theta_{\mathrm{beg}}, \qquad (11)$$

and then it is updated by calculation results of several turnovers.
Boundary conditions are:

$$G_g\,|_{z=0} = G_g{}^1{}_{in}, \quad \vartheta\,|_{z=0} = \vartheta_{in} \text{ - for gas;}$$
$$G_a\,|_{z=L} = G_a{}^1{}_{in}, \quad t\,|_{z=L} = t_{in} \text{ - for air.} \qquad (12)$$

Here $G_g{}^1{}_{in}$, $G_a{}^1{}_{in}$ are the flow rate of the heat carrier entering a gas or air channel matrix sector. The definition of inlet flow in the channel is not quite correct. Heating or cooling of the heat carrier results in change of hydraulic resistance of channel elements and, as a consequence, redistribution of the heat carrier rates will follow. According to MM, exit temperature from a single channel will vary constantly. It corresponds to a case of a periodic operation regenerator. For definition of an outlet temperature for a continuous operation regenerator, it is necessary to evaluate mean integral value of the output temperature of heat-carrier from a single channel. The simplified integration procedure can be written as:

$$\bar{\vartheta}_{out} = \frac{1}{\tau_g}\int_0^{\tau_g} \vartheta_{out\,t}(\tau)d\tau$$

$$\bar{t}_{out} = \frac{1}{\tau_a}\int_{\tau_g+\tau_s}^{\tau_g+\tau_s+\tau_a} t_{out}(\tau)d\tau \qquad (13)$$

Physically it means mixing of streams in the collecting sector leaving from all elementary channels for each heat-carrier at the given instant. MM from Eq. (1) to Eq. (12) describes processes not only for regenerators but can also describe the simultaneous account of two streams processes of heat carriers in recuperative heat exchangers.

4. ALGORITHM OF NUMERICAL CALCULATION

The finite difference scheme is used for the solution of equation set (1) to (12) of the mathematical model. Major steps of the numerical solution are described briefly: 1) obtaining of grid analog area for integration 2) construction of finite difference analog equations; 3) construction of non-iterative algorithm for the solution of algebraic set of equations concerning the required values at the selected grid point. Approximation of all equations will be made in uniform, one-dimensional z axis for streams and two-dimensional grids for heat conduction equation, as shown in Fig. 2b. For finite difference analog of an energy equation (for gas only), a partial implicit scheme with weight factors σ_1 and σ_2 [23] is constructed. For brevity purpose, hereinafter, superscript n

+ 1 for time layer and subscripts g, a, m will be omitted with a designation $\sigma_k^- = 1 - \sigma_k$, for k=1,2,3,4.

$$S\rho_i^n\{[\sigma_1\vartheta_i + \sigma_1^-\vartheta_{i-1}] - [\sigma_1\vartheta_i^n + \sigma_1^-\vartheta_{i-1}^n]\}/\Delta\tau + G[\sigma_2(\vartheta_i - \vartheta_{i-1}) + \sigma_2^-(\vartheta_i^n - \vartheta_{i-1}^n)]/\Delta z = \Pi(h_i^n/c_{pi}^n)\{\sigma_2[\sigma_1(\theta_i - \vartheta_i) + \sigma_1^-(\theta_{i-1} - \vartheta_{i-1})] + \sigma_2^-[\sigma_1(\theta_i^n - \vartheta_i^n) + \sigma_1^-(\theta_{i-1}^n - \vartheta_{i-1}^n)]\} \quad (14)$$

$$i = 1, 2, , I.$$

The gas properties ρ_i^n, c_n^{pi} and heat transfer coefficient h_i^n are evaluated based on weight mean temperature of the previous time layer.

$$\bar{\vartheta}_i^n = \sigma_1 \vartheta_i^n + \sigma_1^- \vartheta_{i-1}^n \quad (15)$$

The finite difference analog for an equation (2), like in equation (1), is constructed with use of partial implicit scheme with weight factors σ_3 and σ_4 for a local one dimensional (or ADI- Alternated Direction Implicit) method. As a simplest case : a matrix with a straight channel in cartesian coordinates will take the form:

- for execution on z axis with intermediate temperature field * will be obtained, not as a solution,

$$c_{i,j}^n \rho_{i,j}^n (\theta_{i,j}^* - \theta_{i,j}^n)/\Delta\tau = \{\sigma_3[k_{i+1/2,j}^n(\theta_{i+1,j}^* - \theta_{i,j}^*)/\Delta z - k_{i-1/2,j}^n(\theta_{i,j}^* - \theta_{i-1,j}^*)/\Delta z]/\Delta z + \{\sigma_3^-[k_{i+1/2,j}^{n}(\theta_{i+1,j}^n - \theta_{i,j}^n)/z - k_{i-1/2,j}^n(\theta_{i,j}^n - \theta_{i-1,j}^n)/\Delta z]/\Delta z,$$

$$i=1,2,,I-1, \quad j=0,1,2,, J;$$

- for execution in y axis with a resultant field of temperatures will be obtained, becoming a solution,

$$c_{i,j}^n \rho_{i,j}^n (\theta_{i,j} - \theta_{i,j}^*)/\Delta\tau = \{\sigma_4[k_{i,j+1/2}^n(\theta_{i,j+1} - \theta_{i,j})/\Delta y - k_{i,j-1/2}^n(\theta_{i,j} - \theta_{i,j-1})/y]/y + \{\sigma_4 - [k_{i,j+1/2}^n(\theta_{i,j+1}^* - \theta_{i,j}^*)/\Delta y - k_{i,j-1/2}^n(\theta_{i,j}^* - \theta_{i,j-1}^*)/\Delta y]/\Delta y,$$

$$i=0,1,, I, \quad j=1,2,, J-1. \quad (16)$$

The thermal conductivity is calculated from Eq. (15) on a mean temperature of the previous temporary layer n between the appropriate nodes, but the thermal capacity obtained directly on nodes. If the temperature of internal nodes of the channel wall and heat transfer coefficients are known, the difference analog in energy equation for heat carriers (14) can be solved directly with respect to the outlet temperature i. With temperatures and heat transfer coefficients from heat carriers to the channel are known, solution of difference analog for bi-variate heat conduction equation can be solved using a three-point execution method [24]. For a joint solution of equation set of a conjugate problem, a modified method to solve conjugated problem was offered by the authors. The idea for gas flow in the channel is explained below. Beginning of calculation of the energy equation with respect to the outlet gas temperature i is impossible because the temperature i,J in the channel wall is yet not known. The explicit solutional scheme, where the temperature of wall can be taken from a previous time step is not considered here. For this reason, solving of a heat conduction equation is carried out first. For regenerators of a continuous operation, calculation sweeps back and forth along the coordinate z for all stratums j = 0,1,2,..J of thickness. With the condition (4) applied to the back of the matrix, such runs in any way are not connected to the stream of heat carrier.

Further, a straight run in the direction of y axis perpendicular to the direction of gas flow is made from zero coordinate of thermally insulated boundary to the coordinate of channel wall washed by gas. Backward run, with known boundary conditions, starts with the recovered wall temperature [25] in the final nodes (i, J) with the formula,

$$\theta_{J}=\frac{[c_{i,J}^{n}\,\rho\Delta y/(2\Delta\tau)]\,\theta_{i,J}^{*}-\sigma_4 k_{i,J-1/2}^{n}R_J/\Delta y+\sigma_4 h_i^{n}\vartheta_i-\sigma_4'\,[k_{i,J-1/2}^{n}(\theta_{i,J}^{*}-\theta_{i,J-1}^{*})/y+h_i^{n}(\theta_{i,J}^{*}-\vartheta_i^{n})]}{c_{i,J}^{n'}\rho\Delta y/(2\Delta\tau)+\sigma_4 k_{i,J-1/2}^{n}(1-L_J)/\Delta y+\sigma_4 h_i^{n}} \quad (17)$$

where L_J and R_J are running factors calculated during a straight line run. The Eq. (17) defines only the relation between wall temperature $\theta_{i,J}$ and the gas temperature ϑ_i. Together with the energy equation (14) it is possible to define temperature i in the outlet node and the temperature of boundary node of a matrix $\theta_{i,J}$ as there exist two equations and two unknowns. After this, internal wall temperature $\theta_{i,j}$ on all i- line grid nodes can be restored with the help of backward run [25] along the direction of y axis. From this it is possible to pass onto the next grid layer i+1 in the direction of gas motion. Obtaining temperatures of gas and matrix wall on a time layer n + 1 in all grid area completes the calculation of each step of time . After reassignment of parameters of a layer n+1 the parameters of a layer n can pass to next calculation step. This algorithm iterates within the limits of considered number of cycle turnovers c and for all given number of cycles. The regenerator effectiveness is usually determined in approximated models with the equation:

$$\eta=\frac{Q_g}{Q_g^{\max}}=\frac{\vartheta_{in}-\bar{\vartheta}_{out}}{\vartheta_{in}-t_{in}}=\frac{Q_a}{Q_a^{\max}}=\frac{\bar{t}_{out}-t_{in}}{\vartheta_{in}-t_{in}}$$

In this case effectiveness is determined after a termination of each cycle τ_c on two equations:

$$\eta_g=\frac{Q_g}{Q_g^{\max}}=\frac{(c_{pg}\vartheta)_{in}-(c_{pg}\bar{\vartheta})_{out}}{(c_{pg}\vartheta)_{in}-(c_{pg}t)_{in}} \quad (18)$$

$$\eta_a=\frac{Q_a}{Q_a^{\max}}=\frac{(c_{pa}t)_{out}-(c_{pa}t)_{in}}{(c_{pa}\vartheta)_{in}-(c_{pa}t)_{in}}$$

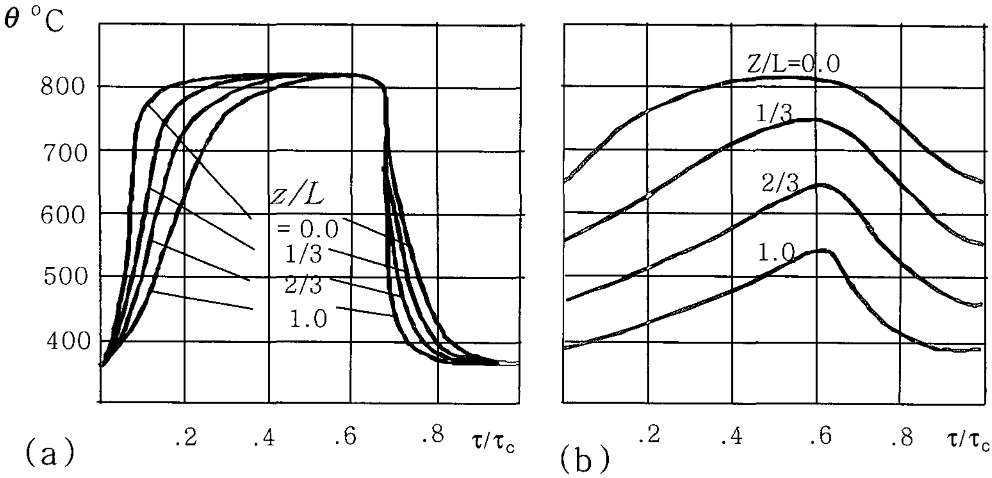

Fig. 3 A time history of channel wall temperatures of a matrix with rotor frequencies variation (a) 0.25 rev/min and (b) 2.0 rev/min

which are expressing relation given (perceived) in a heat cycle (considering the dependency of thermal capacity on temperature) to maximum possible quantity of heat at chilling (heat) of this heat carrier to other heat carrier's entering temperature. For a long, quasi-steady state operation and with heat losses being neglected, η_g will be same as η_a thus exactitude of the energy balance in finite difference scheme can be obtained.

A FORTRAN program using the aforementioned algorithm of calculation process is made. Initial data for calculation consists of :

- type and design parameter of the regenerator: height L, diameter of the matrix D, area S, perimeter Π, wall thickness 2δ, shape of the channel, period of the cycle τ_c, number of cycles; the relative gas and air sector areas ε_g, ε_a, and seal areas ε_s ratio of seal leakage to entire flow late etc.
- inlet parameters of heat carriers, such as temperature, flow rates, pressure, direction of the flow;
- parameters of the numerical scheme: number of nodes, approximation coefficients and property of heat carriers;
- auxiliary parameters, calculation control logic and presentation of the results.

5. RESULTS AND DISCUSSION

In order to illustrate the capabilities of the developed algorithm, calculation result of the main parameters of the intermediate regenerator with rotating ceramic matrix versus rotation frequency variation is presented. During calculation, following initial data are used: L = 0.81m, D = 0.694m, S = 1.56mm^2, Π= 5.0mm, 2δ =0.3mm, ε_g =0.62, ε_a =0.31, $2\varepsilon_s$ =0.07, ϑ_{in} = 921^{0}C, t_{in} = 467^{0}C, G_g =4.05kg/s, G_a =4.0kg/s, $p_{g.in}$ =0.415 MPa, $p_{a.in}$ =1.71MPa, I=11, J=11. The calculation results of matrix wall temperatures are shown in Fig. 3. Thermal discrepancy with the specified parameters, such as geometry and spatial coarse grid discrete numbers were less than 1%. At a rotor speed of 0.25 r/min (see Fig. 3a), gas has time to heat and the air accordingly has time to cool the channel wall temperature close to the inlet temperature. The channel in gas and air streams is extended over a long distance. The mean temperature for period of heating gas from channel matrix to exit is high, and was about 800 ℃. With a same reason, the mean outlet air temperature after a period of cooling was 590 ℃

With a rotor frequency of 2 rev/min (see Fig. 3b) saturation of temperature appears only in small part of the inlet channel length. The effective temperature head of the heat-carrier grows and the heating of air in the regenerator increases. Further increasing of rotor frequency (see Fig. 4) practically does not influence the regenerator operational characteristic and parameter values almost do not change. The results are presented in the Fig. 4. In the high speed region, there exist carryover loss and air outlet temperature begins to decrease. In Fig. 4a, changes of output temperatures of air 1, gas 2 and mean temperatures of a matrix 3 with speed variation are shown. The outlet temperature of heat carrier strongly depend on rotor speed in the beginning. At speed equal to 8 rev/min, air temperature at exit from regenerator reaches the maximum (865 ℃). Maximum value of 83.6% of regeneration effectiveness is also reached as shown in Fig. 4b. The total

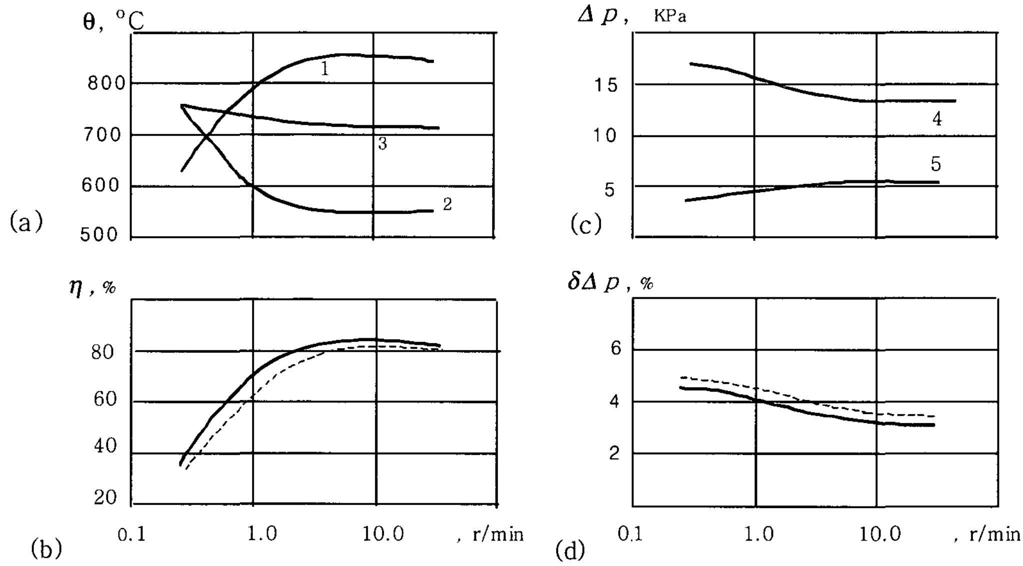

Fig. 4. Relations of (a) temperatures, (b) effectiveness, (c) pressure losses and (d) total relative aerodynamic drop of the regenerator from rotational speed of rotor matrix at absence leakages in seals (continuous lines) and at 3 %-s' air leakages on an input and exit

resistance in the regenerator (see Fig. 4d) also decreases with increasing the rotational speed. In fact, decrease in gas temperature at exit from channel and increase in outlet air temperature is possible only at similar variation of average temperature for all channel length. The pressure loss in the gas channel decreases whereas pressure loss in air grows as shown in Fig. 4c. For the considered design, due to a large difference in inlet pressures, the change of pressure losses in gas duct has more effect on pressure drop in general. The reduced results of numerical investigation illustrate the relationship between the main parameters of the regenerator with rotor frequency change. At the same time, investigation of other parameters is possible using the mathematical model presented.

6. CONCLUSION

A distributed mathematical model based on a set of energy equations for heat carriers and matrix is developed. The model is universal and useful for analysis of regenerators for both a periodic and continuous operation. A convenient, non-iterative calculation method for conjugate problem based on sequential execution of both longitudinal and cross-sectional stream directions is proposed. A calculation process of rotating regenerator is presented using an example and existence of an optimum rotor frequency for regenerator effectiveness is confirmed. A future development of the model will be in the direction of refinement of boundary conditions, calculation techniques of leakage in seals, nonuniform distribution of the flow rates along parallel channels.

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