Int. J. Heat Mass Transfer. Vol. 24, No. 11, pp. 1793–1800, 1981
Printed in Great Britain

0017-9310/81/111793-08 $02.00/0
Pergamon Press Ltd.

THERMAL PERFORMANCE OF REGENERATORS AND WASTE HEAT RECOVERY

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(Received 20 November 1979 and in revised form 9 April 1981)

Abstract—A mathematical model to predict the thermal performance of regenerators with steam addition into the combustion air has been presented. The governing performance equations have been solved by Alternating Direction Approximation Method. On account of 15% steam addition to combustion air, overall heat transfer coefficient during cooling period is doubled, and is increased by 15% during heating period. The savings in fuel costs have been calculated by considering the depreciation, maintenance and operating costs of a waste heat boiler. It has been predicted that the steam injection is more beneficial for long and low capacity regenerators. The net optimum savings in fuel costs range between 1.3 and 4.2 million rupees annually for the various cases worked out.

NOMENCLATURE

a, semi-thickness of channel walls [m];
A_f, cross-sectional area of a channel passage [m^2];
Bi, Biot modulus, ha/k;
C, specific heat of solid [J/Kg K];
h, total heat transfer coefficient between solid surface and gas bulk [W/m^2 K];
h_c, convective heat transfer coefficient [W/m^2 K];
h_r, radiant heat transfer coefficient [W/m^2 K];
k, thermal conductivity of solid [W/m K];
L, regenerator height [m];
m, number of steps of integration in x-direction;
n, number of steps of integration in y-direction;
P, perimeter of channel cross-section [m];
S, specific heat of gas [J/m^3 K];
t, gas temperature [K];
t^*, dimensionless gas temperature, $\dfrac{t - t''_{in}}{t'_{in} - t''_{in}}$;
T, matrix temperature [K];
T^*, dimensionless matrix temperature $\dfrac{T - t''_{in}}{t'_{in} - t''_{in}}$;
v, volume of fluid entrained in a single channel [m^3];
V, volume flow rate of fluid at N.T.P. [m^3/s];
W_1, W_2, regenerator channel sides [m];
x, distance perpendicular to wall surface measured into wall thickness [m];
x^*, dimensionless distance, $\dfrac{x}{a}$;
XS, cubic metre of steam mixed per 100 cubic metres of air;
y, distance in the flow direction measured from the inlet end [m].

Greek Symbols
α, thermal diffusivity of solid [m^2/s];
ε, thermal ratio $\dfrac{t'_{in} - t'_o}{t'_{in} - t''_{in}}$ or $\dfrac{t''_o - t''_{in}}{t'_{in} - t''_{in}}$;
ρ, density of solid [Kg/m^3];
η, dimensionless time;
Λ, dimensionless length of regenerator;
ξ, dimensionless distance;
Π, dimensionless period;
τ, time [s];
τ_p, duration of a period [s].

Subscripts
in, inlet to regenerator;
o, outlet to regenerator;
i, position in x-direction;
j, position in y-direction;
k, position in time;
max, maximum value;
min, minimum value.

Superscripts
$'$, heating period;
$''$, cooling period.

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1. INTRODUCTION

THE RATE at which energy requirements are increasing day-by-day has made it necessary to conserve the fossil fuel reserves. Amongst the various measures being taken to this effect, is one to devise methods for improving the performance and efficiency of the existing equipment and appliances. Thermal regenerators are being increasingly used with furnaces in a large number of plants as fuel economising devices. Such furnaces utilise a large volume of air, and any idea which may promise to improve regenerator thermal ratios by even 0.5% is worthy of attention.

Air is transparent to thermal radiation. During the cooling period when the combustion air passes through the regenerator, heat is transferred from chequer surface to combustion air by convection alone. By introducing a radiating gas or vapour into the combustion air heat will also be transferred by radiation. During the heating period in the regenerator, the rate of heat transfer by radiation will be increased by the addition of radiative gas into the combustion air. The net effect of the increased heat transfer rates is a higher pre-heat temperature or lower fuel consumption. There is hardly any published literature available on the use of any radiating gas for improving regenerator performance. Recently, the authors [1] have proposed the use of steam for the purpose. They have predicted that by adding up to 15% steam to the combustion air through the regenerator, the cooling cycle thermal ratio is increased by over 25% in some representative cases. However, the work of the authors had the following limitations:

(1) The investigations were restricted to the regenerators of medium reduced lengths only;

(2) The overall economical gains on account of steam addition to the combustion air were not worked out. Only improvement in thermal ratios were predicted;

(3) The numerical integration scheme used was not self-starting and some other method had to be used for starting the solution.

(4) The heat capacity of the fluids entrapped in the regenerator channels was neglected as its effect is small [2].

The present work is the extension of the previous work of the authors to overcome the above shortcomings. The cost of steam generation has been considered, and the additional heating required for the injected steam has been included in the energy balance. The Alternating Direction Approximation Scheme has been used for numerical integration because of its self-starting property, unconditional stability and lesser computer time required as compared to many other methods [3].

2. MATHEMATICAL MODEL

The mathematical model used to investigate the scope of "waste heat recovery" in this paper is the same as earlier used by the authors [1]. The 2-D model has been selected for obtaining better accuracy of the predicted results. With steam addition into the combustion air, heat transfer coefficients are improved considerably. Hence, the values of Biot modulus also increases appreciably. It has been shown in [4] that with increase in the value of Biot modulus, the 1-D model becomes less accurate as compared to 2-D model to predict the effectiveness of regenerators. The difference between the values of thermal ratio predicted by the two models may be as high as 5% [4]. This difference is quite significant in terms of fuel economy. Therefore, for the present work the 2-D model has been selected.

In the mathematical model, the regenerator is assumed to consist of a number of rectangular passages of sides W_1 and W_2. The thickness of wall constituting the passages is $2a$, and P is the wall perimeter. The following are, in brief, the assumptions made:

(1) The thermal properties of the matrix material are constant and do not vary with temperature or position.

(2) The fluid flow rates are constant during both the periods. They may be different during the heating and the cooling periods.

(3) The thermal conductivity of the matrix is zero in the direction of fluid flow, i.e. in y-direction. The conductivity in the direction perpendicular to fluid flow, in the x-direction, is finite.

(4) The values of heat transfer coefficients in cooling and heating periods differ from each other, but for a particular period these values remain unchanged. These have been calculated on the basis of time–space average temperatures during the period. Similarly, the gas and air specific heats are assumed constant during the respective periods, and are calculated at the time–space mean temperatures for the periods.

(5) The fluid temperature is uniform across a cross-section.

(6) Axial conduction within the fluid is negligible.

(7) In between the periods, the hot/cold fluid is completely driven off and the space within the solid matrix is instantly filled with cold/hot fluid, entering from the opposite end. This is assumed to be accomplished without a change in the matrix temperature.

The x-direction is measured from the matrix heat transfer surface into the wall thickness. The y-direction is measured parallel to the flow, and always from the inlet end. The dimension in the third direction is taken as unity.

The centre plane of the wall thickness, at $x = a$, may be assumed to be taken as adiabatic, as all the channels making up the regenerator are identical. Therefore, for the purpose of mathematical analysis, it is sufficient to consider only half the matrix wall thickness of a single rectangular passage. The corner effect is also ignored.

The matrix wall semi-thickness is divided into rectangular elements each of sides Δx and Δy, such that $m\,\Delta x = a$ and $n\,\Delta y = L$. The next step is to write the heat balance equations for each of these elements. This gives the following set of 1st-order differential equations for the elements at any height level j:

(i) For the surface element, $i = 1$,

$$\frac{\partial T_1}{\partial \tau} = \frac{k}{\rho c(\Delta x)^2}(T_2 - T_1) + \frac{h}{\rho c \Delta x}(t - T_1). \quad (1)$$

(ii) For the inner elements, $i = 2$ to $i = m-1$,

$$\frac{\partial T_i}{\partial \tau} = \frac{k}{\rho c(\Delta x)^2}(T_{i+1} - T_i) + \frac{k}{\rho c(\Delta x)^2}(T_{i-1} - T_i). \quad (2)$$

(iii) For the innermost element having one surface adiabatic, $i = m$,

$$\frac{\partial T_m}{\partial \tau} = \frac{k}{\rho c(\Delta x)^2}(T_{m-1} - T_m). \quad (3)$$

An energy balance between the surface element, $i = 1$, and the gas bulk in the channel gives:

$$(T_1 - t) = \frac{VS}{hp}\frac{\partial t}{\partial y} + \frac{A_f S}{hp}\frac{\partial t}{\partial \tau}. \quad (4)$$

In the above equations (1)–(4), the suffix on T represents the location of the element in the x-direction. The condition of constant fluid inlet temperatures are expressed as

$$t'(0, \tau') = t'_{in}, \quad (5a)$$

$$t''(0, \tau'') = t''_{in}. \quad (5b)$$

As the y-direction is always measured from the inlet end, the conditions of flow reversal in between the periods are given by:

$$T'(x, y, \tau'_p) = T''(x, L-y, 0) \quad (6a)$$

$$T''(x, y, \tau''_p) = T'(x, L-y, 0) \quad (6b)$$

Equations (1)–(6) are transformed into non-dimensional form by making the following substitutions:

$x^* = x/a$, dimensionless distance along x-axis;

$\xi = \frac{hp}{VS} y$, dimensionless distance along y-axis;

$\eta = \frac{\alpha}{a^2}\left(\tau - \frac{v}{V}\frac{y}{L}\right)$, dimensionless time;

$B_i = \frac{ha}{k}$, Biot number;

$\Lambda = \frac{hpL}{VS}$, regenerator reduced length;

$\Pi = \frac{\alpha}{a^2}\left(\tau_p - \frac{v}{V}\right)$, regenerator reduced period.

Thus equations (7)–(12) are obtained:

$$\frac{\partial T_1^*}{\partial \eta} = \frac{(T_2^* - T_1^*)}{(\Delta x^*)^2} + \frac{Bi(t^* - T_1^*)}{\Delta x^*}, \quad (7)$$

$$\frac{\partial T_i^*}{\partial \eta} = \frac{T_{i+1}^* - T_i^*}{(\Delta x^*)^2} + \frac{T_{i-1}^* - T_i^*}{(\Delta x^*)^2}, \quad i = 2, 3, \ldots, m-1 \quad (8)$$

$$\frac{\partial T_m^*}{\partial \eta} = \frac{T_{m-1}^* - T_m^*}{(\Delta x^*)^2}, \quad (9)$$

$$\frac{\partial t^*}{\partial \xi} = T_1^* - t^*, \quad (10)$$

$$t^{*\prime}(0, \eta') = 1, \quad (11a)$$

$$t^{*\prime\prime}(0, \eta'') = 0, \quad (11b)$$

$$T^{*\prime}(x^*, \xi', \Pi') = T^{*\prime\prime}(x^*, \Lambda''[1 - \xi'/\Lambda'], 0), \quad (12a)$$

$$T^{*\prime\prime}(x^*, \xi'', \Pi'') = T^{*\prime}(x^* \Lambda'[1 - \xi''/\Lambda''], 0). \quad (12b)$$

It may be seen that at the same position in the matrix, $\xi'/\Lambda' = 1 - \xi''/\Lambda''$.

3. FINITE DIFFERENCE REPRESENTATION AND NUMERICAL METHOD

The numerical integration of the 1st-order differential equations (7)–(10) can be carried out by replacing them with their finite difference representations. The Alternating Direction Approximation Method is selected for equations (7)–(9). This relatively new method has an accuracy of the order of $(\Delta\tau)^2$. It is self-starting, unconditionally stable and requires much less computer time than many other methods available for integration, like Crank–Nicolson or Mitchell–Pearce [3]. The Alternating Direction Scheme as described in [2] gives two distinct formulae, one for the odd and other for the even time steps. The corresponding finite difference representations of equations (7)–(9) are:

For $j = 1, 2, \ldots, n$

(1) At odd time steps, $k = 2, 4, 6, \ldots$:

$$T_{1,j,k+1}^* = Y_1 T_{1,j,k}^* + Y_2 t_{j,k+1}^* + Y_3 T_{2,j,k}^*, \quad (13)$$

$$T_{i,j,k+1}^* = Y_4 T_{i,j,k}^* + Y_5(T_{i-1,j,k+1}^* + T_{i+1,j,k}^*), \quad i = 2, 3, \ldots, m-1, \quad (14)$$

and

$$T_{m,j,k+1}^* = Y_6 T_{m,j,k}^* + Y_5 T_{m-1,j,k+1}^*. \quad (15)$$

(2) At even time steps, $k = 1, 3, 5, \ldots$:

$$T_{1,j,k+1}^* = Y_8 T_{1,j,k}^* + Y_9 t_{j,k}^* + Y_5 T_{2,j,k+1}^*, \quad (16)$$

$$T_{i,j,k+1}^* = Y_4 T_{i,j,k}^* + Y_5(T_{i-1,j,k}^* + T_{i+1,j,k+1}^*), \quad i = m-1, m-2, \ldots, 3, 2, \quad (17)$$

and

$$T_{m,j,k+1}^* = Y_7 T_{m,j,k}^* + A_1 T_{m-1,j,k}^*. \quad (18)$$

The coefficients Y_i in these equations are:

$Y_1 = (1-A_1)/(1+A_2)$; $Y_2 = A_2/(1+A_2)$;
$Y_3 = A_1/(1+A_2)$; $Y_4 = (1-A_1)/(1+A_1)$;
$Y_5 = A_1/(1+A_1)$; $Y_6 = 1/(1+A_1)$;
$Y_7 = 1-A_1$; $Y_8 = (1-A_2)/(1+A_1)$

and

$$Y_9 = A_2/(1+A_1),$$

where $A_1 = \Delta\eta/(\Delta x^*)^2$ and $A_2 = \Delta\eta Bi/\Delta x^*$.

Equation (9) can be approximated by any formula used for integration of 1st-order differential equation. In the present work, the trapezoidal formula [5] is used for the purpose. Thus equation (9) is equivalent to

$$t^*_{j,k+1} = Z_1 t^*_{j-1,k+1} + Z_2(T^*_{1,j,k+1} + T^*_{1,j-1,k+1}) \quad j=2, 3, \ldots, n, \quad (19)$$

where

$$Z_1 = (1-0.5\Delta\xi)/(1+0.5\Delta\xi)$$

and

$$Z_2 = (0.5\Delta\xi)/(1+0.5\Delta\xi).$$

Equation (19) can be used to calculate the gas temperature at even time steps only. At odd time steps, the gas temperature at any height j has to be calculated before the matrix temperatures at the same height and so the equation (19) cannot be used. For this purpose a combination of equations (13) and (19) is used, as given in [2]. The step-by-step simulation process with a flow chart is also given in [2].

Based on the numerical method developed, the computer programme is prepared. The convective heat transfer coefficient, h_c, is calculated by using the graphs presented by Razelos and Paschkis [6]. The radiative heat flux in each period between the chequer surface and the fluid, q_r, is calculated as given in [7]. A mean surface temperature, T_s and a mean gas temperature, T_g are evaluated as discussed in earlier paper of the authors [9]. Corresponding to these mean temperatures, the values of emissivity of CO_2 and water vapour and their appropriate correction factors are found from graphs [7] or [8]. The radiative heat transfer is calculated from the expression: $q_r = h_r(T_g - T_s)$. The total heat transfer is the sum of h_c and h_r.

4. RESULTS AND DISCUSSION

To illustrate and establish the beneficial effects of steam addition to combustion air, the authors have taken a large number of examples for analysis. The results of some of the typical cases are reported here. The following parameters of the regenerator have been kept constant throughout the analysis:

air inlet temperature: 323 K,
hot gas inlet temperature: 1723 K,
thermal conductivity of solid: 1.581 W/m K,
thermal diffusivity of solid: 4×10^{-7} m^2/s,
emissivity of matrix surface: 0.8,
ratio of gas to air flow: 1.045,
heating or cooling period: 1200 s.

Gas composition by volume percentage when no steam is added to combustion air:

H_2O vapour: 11.36,
CO_2: 12.33,
O_2: 1.80,
N_2: 74.51.

Channel dimensions were chosen as:

$W_1 = 0.1524$ m,
$W_2 = 0.2286$ m,
$a = 0.0381$ m.

Air flow rate in each channel was taken as 0.01527 scms and regenerator height as 6.5 m.

The computer programme was run with the above data by injecting various percentages of steam into the combustion air through the regenerator. In the present work, steam has been assumed to be dry, saturated and at atmospheric pressure. The predicted results are displayed in Fig. 1. Figure 1(a) shows the variation of heat transfer coefficients with steam addition to the combustion air in both the heating and the cooling periods. The convective heat transfer coefficients during both the periods increase slightly on account of increased velocity with steam addition. The radiation heat transfer coefficient during cooling period increases from zero to 7.38 W/m^2 K with 15% addition of steam into the combustion air. It is observed that during the cooling period, the overall heat transfer coefficient is almost doubled by injecting 15% steam into the combustion air. However, during heating period radiative heat transfer increases from 25.14 W/m^2 K with no steam to 30.50 W/m^2 K with 15% steam addition to combustion air or it is increased by 20%. The overall heat transfer coefficient is increased by about 15%.

Addition of steam to combustion air also changes

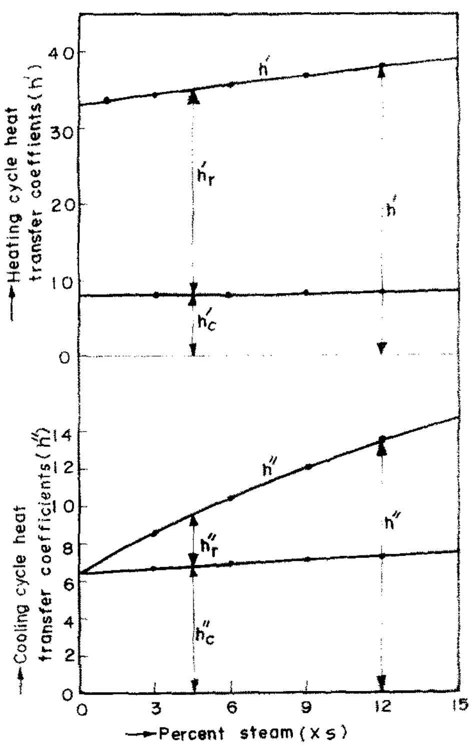

FIG. 1(a). Effect of steam addition to combustion air on heat transfer coefficients.

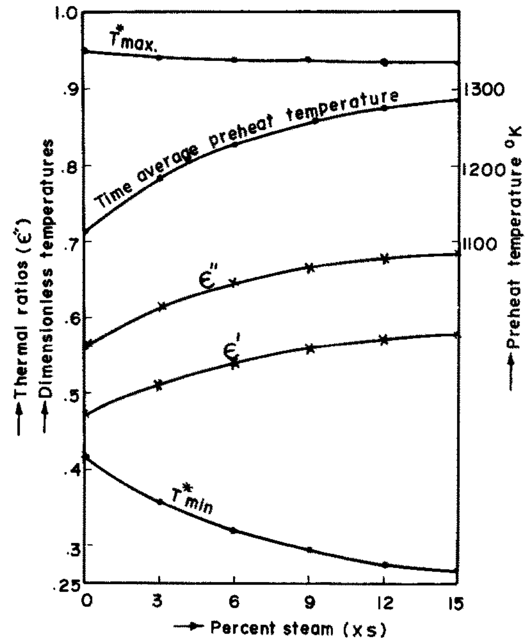

FIG. 1(b). Effect of steam addition to combustion air on regenerator performance.

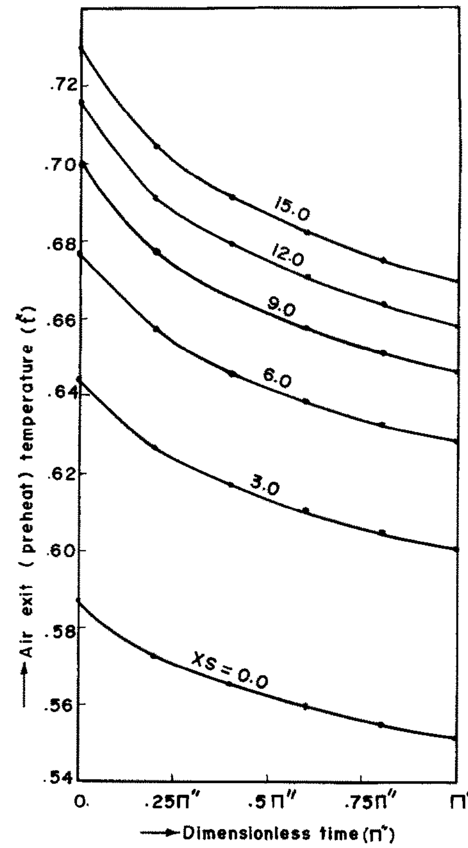

FIG. 1(c). Effect of steam addition to combustion air on time variation of air-exit temperature.

the gas thermal properties. The only gas thermal property that is to be dealt with quite frequently in the mathematical analysis of the thermal generator is the specific heat. Addition of 15% steam to combustion air increases the fluid specific heat by about 5% in the cooling period and about 3% in the heating period. As discussed in last paragraph, the overall heat transfer coefficient is almost doubled in the cooling period and increases by about 15% in the heating period. Therefore, the influence of overall heat transfer coefficient on the thermal performance of regenerator is very significant as compared to the influence of change in the thermal properties of the gas. Therefore, authors have not studied separately the effects of steam injection on the regenerator behaviour with change in thermal properties of gas. However, in the entire work, the specific heats of air and gas are interpolated from the tabulated values of specific heats of air and gas for the temperature range between 0° and 1500°C.

Figure 1(b) depicts that non-dimensional pre-heat temperature and the thermal ratios in cooling and heating periods increase and the non-dimensional matrix extreme temperature decrease with steam addition into the combustion air. Maximum matrix temperature T^*_{max} decreases slightly. The minimum matrix temperature T^*_{min} drops from 0.4127 to 0.2711 when 15% steam is mixed with combustion air, i.e. it drops by about 30%. The non-dimensional pre-heat temperature and the two thermal ratios increase sharply with up to 7% steam injection. With further increase in steam injection beyond 7%, these values increase slightly.

The time-variation of the pre-heat temperature is plotted in Fig. 1(c) for steam quantities from 0 to 15%. It may be seen that without steam the air outlet dimensionless temperature during the cooling period drops from 0.5876 to 0.5522, or by 6.27% of the time-average pre-heat temperature. With 15% steam, the drop is from 0.7296 to 0.6699 or 8.67% of the corresponding time average. It may be observed that:

(i) As the steam proportion is increased, the influence on the air pre-heat temperature becomes less marked. This is also evident from Fig. 1(b).

(ii) The curves for different steam quantities are almost parallel, except at the inlet end, which have greater slopes for higher steam percentages. This shows that the pattern in which the pre-heat temperature varies during the cooling period is little affected by the use of steam.

As the temperature distribution becomes more non-linear with the use of steam in combustion air, the use of the 2-space-dimensional model for the present work is fully justified [10].

A number of other examples considered have given similar results as presented above. In a set of examples with the above data the regenerator height was varied from 2.5 to 9 m. The effects of mixing steam with combustion air on the thermal ratios of heating and cooling periods are plotted in Figs. 2 and 3 respectively. Here again it may be observed that beyond about 10% steam there is relatively less improvement in the thermal ratios.

The increase in thermal ratios and the consequent higher pre-heat temperature, however, do not nec-

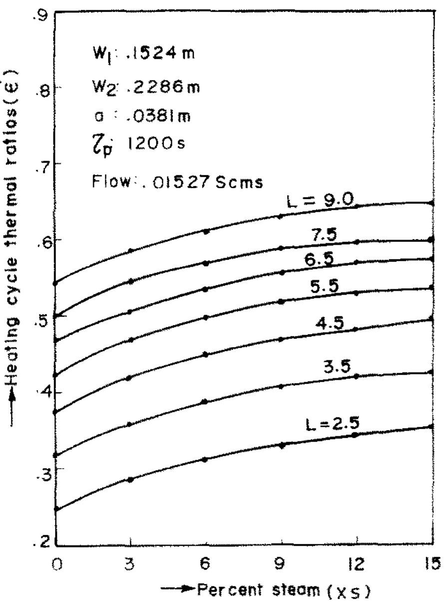

FIG. 2. Effect of stdam injection on heating cycle thermal ratios.

essarily reflect the beneficial effects of the use of steam with combustion air. The mixing of steam to combustion air requires additional fuel in the furnace to heat the steam from the pre-heat temperature to the temperature of the hot gases. The net savings in the heat requirements of the furnace, ΔH is obtained by making energy balances for the furnace with and without steam. It is also equal to the difference between increase in enthalpy of pre-heat air because of steam addition and the extra heat input required to increase the temperature of the injected steam, from pre-heat temperature to the hot gases temperature. This has been plotted for regenerator heights from 2.5 to 7.5 m in Fig. 4 and for four different flow rates in Fig. 5. The

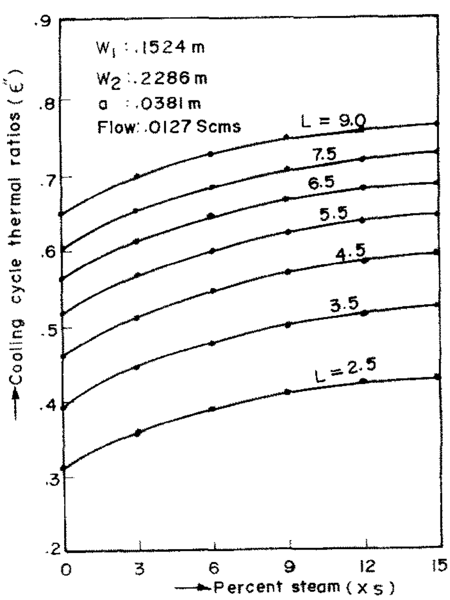

FIG. 3. Effect of steam injection on cooling cycle thermal ratios.

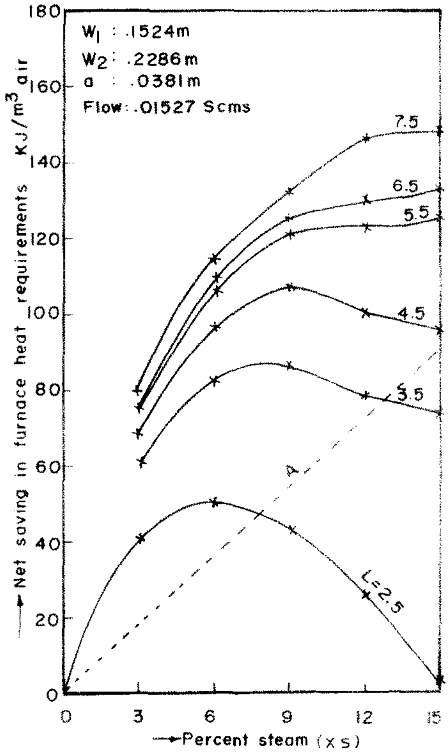

FIG. 4. Net saving in heat requirement in the furnace for different regenerator heights.

benefit of using steam would be maximum when $d(\Delta H)/d(xS)$ is zero. These graphs illustrate that:

(i) The steam addition is most beneficial for regenerators with relatively greater height.

(ii) After a certain steam addition the curves of ΔH vs XS start sloping down, and hence the steam addition becomes less beneficial. For regenerators of 3.5 and

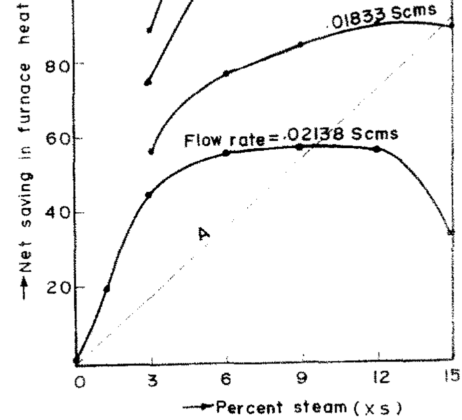

FIG. 5. Net saving in heat requirement for different flow rates.

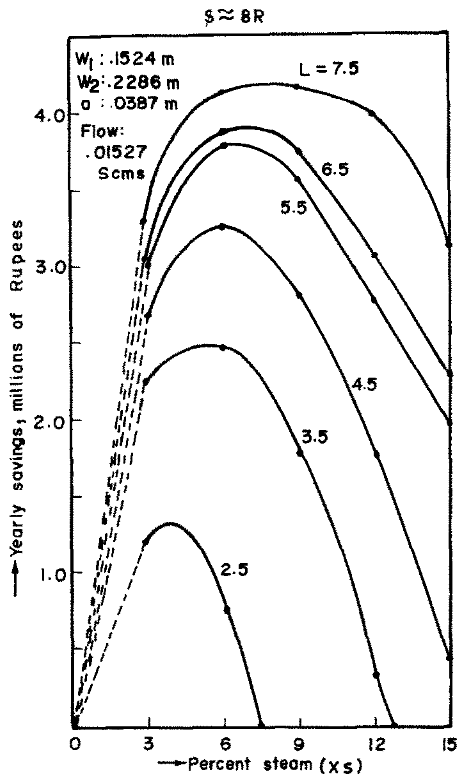

FIG. 6. Yearly savings for different regenerator heights.‡

4.5 m height in the present example, the maximum values of ΔH is observed with about 9% steam addition. For 5.5 and 6.5 m it occurs at about 12% steam addition.

(iii) Steam addition is more beneficial at lower flow rates as compared to the higher flow rates (Fig. 5).

The above conclusions are valid when the steam is available entirely free of cost, from process plants. The steam available from process industries is mostly used for space heating in cold countries. In tropical countries like India, this steam is generally unutilised and available free of cost.

When steam is not available free of cost, from process plants, it can be generated in waste heat boilers. In the present analysis, the depreciation cost, maintenance and operating cost of the waste heat boiler has been considered. To generate 1 scm of steam about 2000 KJ of heat input is required from waste gases coming out of regenerator. An experience from boiler operation suggest that the depreciation, maintenance and operating cost of the waste heat boiler may be taken as about 30% of the total cost of fuel required to generate it. So to generate 1 m^3 of steam in waste heat boiler, an expenditure equivalent to about 600 kJ of heat input is required. This has been shown as dotted line A in Figs. 4 and 5. An overall net saving in the fuel heat in the furnace is the vertical distance between the line A and any curve above it. The portion of the curves lying below the line A is not at all economically feasible.

Figure 6 displays the net savings in the fuel cost after considering depreciation, maintenance and operating cost of waste heat boiler as a function of steam addition to combustion air for various chequer heights. The results shown in Fig. 6 are based on the following data:

(i) The plant is using 60 scms air. This value is quite normal for a moderate size unit.

(ii) The calorific value of the fuel is 35,600 kJ/l.

(iii) The cost of fuel is Rs 1.00‡ per litre.

It is observed from Fig. 6 that the yearly savings in fuel costs work out to be in millions of rupees. It is also shown that as regenerator cheques heights are increased, the maximum savings in fuel costs are obtained with higher steam percentage. Thus for L = 3.5 with 6% steam addition, maximum yearly saving of about 2.4 million rupees is obtained. For L = 6.5, the corresponding figure is 4.8 million rupees with about 7.5% steam addition. The yearly savings in fuel costs, as a result of steam addition to combustion air, decreases rapidly by injecting steam in excess of the optimum value for the particular case.

‡ One U.S. $ is approx. equivalent to Rs. 8.15 in 1981.

5. CONCLUSIONS

(1) With 15% steam addition to the combustion air, the overall heat transfer coefficient is almost doubled during cooling cycle and increases by about 15% during heating cycle.

(2) With the 15% steam addition, maximum matrix temperature drops slightly and minimum matrix temperature drops by 30%.

(3) Non-dimensional pre-heat temperatures and the thermal ratios in both the periods increase with increase in steam addition.

(4) The pattern of variation of non-dimensional pre-heat temperature does not change with the steam injection into the combustion air.

(5) With up to 10% increase in steam addition to combustion air, the thermal ratios, during the cooling and heating periods, increase with the increase in regenerator height from 2.5 to 9 m. For all the cases considered, further addition of steam beyond 10% does not significantly improve the thermal ratios.

(6) For maximum net saving in fuel cost, the volume of steam to be injected into the combustion air increases with increase in chequer height of the regenerator.

(7) Some of gains in fuel savings are offset by considering the depreciation, maintenance and operating costs of the waste heat boiler.

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PERFORMANCE THERMIQUE DES REGENERATEURS ET RECUPERATION DE CHALEUR PERDUE

Résumé—On présente un modèle mathématique pour évaluer les performances thermiques de régénérateurs avec addition de vapeur dans l'air de combustion. Les équations de base sont résolues par la méthode des directions alternées. On estime que pour l'addition de 15 pour cent de vapeur à l'air de combustion, le coefficient global de transfert thermique est doublé pendant la période de refroidissement et qu'il est augmenté de 15 pour cent pendant la période de chauffage. On calcule l'économie en considérant les coûts de fonctionnement, d'entretien et la dépréciation de la chaudière. On montre que l'injection de vapeur est plus rentable pour des régénérateurs de longue et faible capacité. L'optimum d'économie dans les dépenses de fuel se situe entre 1,3 et 4,2 millions de roupies par an pour les différents cas étudiés.

DAS THERMISCHE VERHALTEN VON REGENERATOREN UND WÄRMERÜCKGEWINNUNGSANLAGEN

Zusammenfassung — Es wird ein mathematisches Modell vorgestellt, das zur Vorausberechnung des thermischen Verhaltens von Regeneratoren dient, bei denen Dampf zur Verbrennungsluft hinzugefügt wird. Die beschreibenden Gleichungen werden mit Hilfe des Approximations-Verfahrens der alternierenden Richtungen gelöst. Durch Zufuhr von 15% Dampf zur Verbrennungsluft verdoppelt sich der Gesamt-Wärmeübergangs-Koeffizient während der Kühlperiode und erhöht sich während der Heizperiode um 15%. Die Einsparungen an Brennstoffkosten wurden unter Berücksichtigung der Abschreibung, der Instandhaltungs- und Betriebskosten eines Abhitzekessels berechnet. Es zeigte sich, daß Dampfeinspritzung besonders für lange Regeneratoren und für solche mit geringer Speicherkapazität vorteilhaft ist. Die optimalen Nettoeinsparungen bei den Brennstoffkosten bewegen sich für die verschiedenen behandelten Fälle zwischen 1,3 und 4,2 Millionen Rupien jährlich.

ТЕПЛОВОЙ РЕЖИМ РЕГЕНЕРАТОРОВ И РЕГЕНЕРАЦИЯ ОТРАБОТАННОГО ТЕПЛА

Аннотация — Предложена математическая модель для расчета теплового режима регенераторов при добавлении пара к воздуху, поступающему в зону горения. Основные уравнения, описывающие данный режим, решаются приближенным методом переменных направлений. Если количество добавляемого к воздуху пара составляет 15%, величина суммарного коэффициента теплопереноса удваивается при охлаждении и возрастает на 15% при нагревании. Снижение себестоимости топлива рассчитывается из анализа амортизационных расходов, стоимости технического обслуживания и эксплуатации котла. Расчеты показывают, что более выгодно использовать пар в регенераторах с малой емкостью. Оптимальное снижение себестоимости топлива составляет от 1,3 до 4,2 миллионов рупий в год для различных разработанных вариантов.