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Thermodynamic optimization of reheat regenerative thermal-power plants

M.A. Habib*, S.A.M. Said, I. Al-Zaharna

Mechanical Engineering Department King Fahd University of Petroleum and Minerals, Dhahran 31261, Saudi Arabia

Abstract

A first- and second-law procedure for the optimization of the reheat pressure level in reheat regeneration thermal-power plants is presented. The procedure is general in form and is applied for a thermal-power plant having two reheat pressure levels and two open-type feed-water heaters. The second-law efficiency of the steam generator, turbine cycle and plant are evaluated and optimized. The irreversibilities in the different components of the steam generator and turbine cycle sections are evaluated and discussed. Additional constraints such as the steam qualities at the exits of the different turbine stages are considered.

1. Introduction

Steam reheating is an important feature in steam-power plants. The main objective of reheat is to increase the power output and, under certain conditions, the thermal efficiency of the plant, thus improving plant performance. There is a wide range over which reheat pressures can be varied. Hence, for every set of steam conditions, an optimum value of reheat pressure exists that will yield an optimum steam turbine-boiler reheat-cycle. Second-law analysis [1–3] was considered for optimizing conventional plants [4]. Huang [5] and Habib [6] provided a second-law analysis for the optimization of cogeneration steam plants. Some constraints exist and present a limiting factor for the range of choice of reheat pressures. Sciubba and Su [7] analyzed the influence of reheat temperature and pressure on regeneration-cycle performance.

* Corresponding author. Tel: +966-3-860-4467; fax: +996-3-860-2949.
E-mail address: mahabib@kfupm.edu.sa (M.A. Habib)

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Nomenclature

$\dot{A}$	rate of flow availability, kW
C_{pg}	specific heat of exhaust gases at constant pressure, kJ/kg
$\dot{E}$	exergy rate, kW
H	feedwater specific enthalpy, kJ/kg
HHV	higher heating value, kJ/kg
h	Specific enthalpy, kJ/kg
$\dot{I}$	irreversibility rate, kW
L	the total number of components of the exhaust gases
M	molecular weight, kg/kmol, the number of extractions between the boiler exit and a specific turbine
m	the number of extractions in the turbine, the number of feed-water heaters
$\dot{m}$	mass flow rate, kg/s
N	the number of turbines
n	the number of reheats
$\dot{n}$	rate of molar flow, kmol/s
P	power output, kW
P_1	low reheat-pressure, kPa
P_2	high reheat-pressure, kPa
$\dot{Q}$	rate of heat transfer, kW
q	heat transfer per unit mass, kJ/kg
S	specific entropy of feed-water, kJ/kgK
s	specific entropy of steam, kJ/kgK
T	temperature, °C
X_1	volumetric concentration, dimensionless
Y	mass fraction of extracted steam, dimensionless
y	enthalpy difference ratio, Eq. (3)
ΔH	increase in the enthalpy of water in the boiler, kJ/kg
Δh	increase in the steam enthalpy in the reheater, kJ/kg
η	efficiency
η_I	first-law thermal efficiency
η_{II}	second-law thermal efficiency
λ	excess-air factor
Ψ	flow availability per unit mass, kJ/kg
ψ_1	the flow availability of the steam at exit of the boiler, kJ/kg
ψ	rate of flow availability per unit mass, kJ/kg

Subscripts

b	boiler (heat exchanger)unit
C	turbine-cycle unit

c	condenser
cc	combustion chamber
ch	chemical
e	exit
exh	exhaust gases
f	fuel
g	steam-generator
i	inlet
o	ambient conditions
Q	heat
s	steam
t	turbine
th	thermo-mechanical
w	water
x_1	quality of exit of high-pressure turbine
x_2	quality of exit of intermediate-pressure turbine

Their results indicated that the most economical gain would occur when the reheat temperature increases no more than 30 K from the saturation temperature corresponding to the steam pressure from a high-pressure turbine-exhaust. Silvestri et al. [8] concluded that both first- and second-reheat pressures can be varied over an appreciable range only with a limited effect on the heat rate. Equipment designs and operating concerns that place limits on reheat-pressure selection were also noted in both studies.

Recent studies indicate the importance of reheat temperature control [9] and numerical modeling of reheat regenerative furnaces [10]. The most recent analyses indicate the possibility of attaining high plant-efficiencies, over 45%, as a result of using efficient steam turbines [11], even reaching 67% with multiple Rankine topping cycles [12]. However, improving the performance of existing plant configurations through optimization of reheat pressures remains a desirable objective for the next decade or so.

The present paper is an extension of a previous paper by Habib et al. [13] which applied first-and second-law analysis for optimizing the reheat pressures of non-regenerative power plants. The analysis in the present paper is extended to include feedwater heating in the plant.

2. Mathematical formulation and solution procedure

The layout of the thermal-power plant considered in the present work is shown in Fig. 1. The plant utilizes the reheat regeneration-cycle and comprises a boiler with two reheats, multistage turbines, a condenser and open-type feed-water heaters.

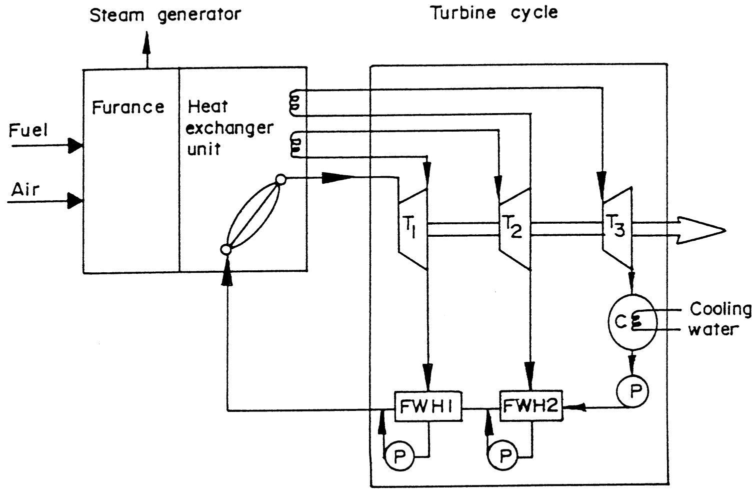


Fig. 1. Schematic of the steam-power plant.

2.1. First-law analysis

For a fixed rate of power output of the plant, and n turbines, the steam mass flow rate can be evaluated as follows:

$$\dot{m}_s = P/\left[\sum_{k=1}^{n}\Delta h_k - \sum_{L=1}^{n}\left(\sum_{j=1}^{L-1}Y_j\right)\Delta h_L\right] \tag{1}$$

where Y_j is the mass flow rate fraction of the steam extracted at the exit of turbine stage j normalized by the steam mass flow rate at the exit of the boiler and is given by:

$$\left[Y_j = \left(1-\sum_{i=1}^{j-1}Y_i\right)y_j\right]_{j\neq 1} \qquad \text{with } Y_1 = y_1. \tag{2}$$

where y_j is the enthalpy difference ratio and is given by

$$y_j = \left[\frac{H_{\mathrm{W_e}} - H_{\mathrm{W_i}}}{h_{\mathrm{T_e}} - H_{\mathrm{W_i}}}\right]_j \tag{3}$$

The fuel mass flow rate can be obtained from:

$$\dot{m}_f = \dot{m}_S\left\{\left(h_{\mathrm{b_e}} - H_{\mathrm{b_i}}\right) + \sum_{k=2}^{N}\left[\left(1-\sum_{j=1}^{m}Y_j\right)\Delta H_k\right]\right\}/(\eta_{\mathrm{b}}xHHV) \tag{4}$$

where m is the number of extractions in the turbine and N is the number of reheats. The plant's thermal-efficiency is given by

$$\eta_I = P/(\dot{m}_{\mathrm{f}} HHV) \tag{5}$$

2.2. Exergy and irreversibility analysis

The thermal power plant can be divided into two main units, the steam generator which consists of the furnace and the heat exchanger unit and the turbine cycle which consists of the high-pressure turbine, intermediate-pressure turbine, low-pressure turbine, condenser and feed-water heaters. The exergy inlet to the steam generator can be calculated as:

$$\dot{E}_g = \dot{m}_{\mathrm{f}} HHV(1 - T_o/T_b) \tag{6}$$

The exergy destruction in the steam generator can be calculated as described in the following paragraphs.

The exergy destruction in the furnace occurs as exergy losses from the boiler in the exhaust gases, thermo-mechanical loss and chemical loss. The availability destruction via the stack losses can be presented by:

$$\dot{A}_Q = \dot{m}_{\mathrm{f}} HHV(1 - \eta_{\mathrm{b}})(1 - T_{\mathrm{o}}/T_{\mathrm{exh}}) \tag{7}$$

To evaluate the thermo-mechanical contribution to availability destruction in the stack gases, the combustion equation of a mixture of L gases having volumetric concentration can be presented as:

$$\begin{aligned} &\sum_{1}^{L} X_i \left\{ \mathrm{C_a H_b} + \lambda\left(a + \frac{b}{4}\right)\mathrm{O_2} + 3.76\left(a + \frac{b}{4}\right)\mathrm{N_2} \right) \rightarrow \\ &\Sigma X_i \left\{ a\mathrm{CO_2} + \frac{b}{2}\mathrm{H_2O} + (\lambda - 1)\left(a + \frac{b}{4}\right)\mathrm{O_2} + 3.76\lambda\left(a + \frac{b}{4}\right)\mathrm{N_2} \right\} \end{aligned} \tag{8}$$

where λ is the excess-air factor and L is the number of fuel constituents. The availability destruction in the stack gases can, therefore, be expressed as

$$\begin{aligned} \dot{A}_{\mathrm{th}} = \dot{n}_{\mathrm{f}} \sum_{1}^{L} x_{\mathrm{i}} \Big\{ & a\psi_{\mathrm{CO_2}} + (b/2)\psi_{\mathrm{H_2O}} \\ & + (\lambda - 1)\left(a + \frac{b}{4}\right)\psi\mathrm{O_2} + \lambda\left(a + \frac{b}{4}\right)3.76\psi\mathrm{N_2} \Big\} \end{aligned} \tag{9}$$

where $\dot{n}_f$ is the fuel flow rate in kmol/s and is given by:

$$\dot{n}_{\mathrm{f}} = \dot{m}_{\mathrm{f}}/M_{\mathrm{f}} \tag{10}$$

and ψ is the flow availability rate, given by

$$\psi = (h - h_{\mathrm{o}}) - T_{\mathrm{o}}(S - S_{\mathrm{o}}) \tag{11}$$

The chemical availability destruction is given by

$$\dot{A}_{\mathrm{ch}} = \dot{n}_{\mathrm{f}}\overline{R}T_{\mathrm{o}}\sum_{i=1}^{M} ln\frac{Y_{\mathrm{i}}}{Y_{\mathrm{i}}^{e}} \tag{12}$$

where Y_{i} and Y_{i}^{e} are the molar fractions of components i in the stack gas mixture at T_{o} and P_{o} and in the environment, respectively, and M is the total number of exhaust gas constituents in the stack.

2.3. Irreversibility analysis

The irreversibility rate in the heat exchanger unit can be expressed as

$$\dot{I}_{\mathrm{b}} = T_{\mathrm{o}}\left\{\dot{m}_{\mathrm{s}}\left[\left(s_{\mathrm{b_c}} - S_{\mathrm{w_i}}\right) + \sum_{K=1}^{N}\left(1 - \sum_{J=1}^{K} Y_J\right)\left(s_{k_{\mathrm{e}}} - s_{k_1}\right)\right] - \frac{\dot{Q}_{\mathrm{o}}}{T}\right\} \tag{13}$$

Thus, the irreversibility losses in the steam generator are given by

$$\dot{I}_{\mathrm{g}} = \dot{A}_{\mathrm{Q}} + \dot{A}_{\mathrm{th}} + \dot{A}_{\mathrm{ch}} + \dot{I}_{\mathrm{b}} \tag{14}$$

The availability rate at the exit of the steam generator (inlet to the turbine cycle) is given by

$$\dot{E}_{\mathrm{c}} = \dot{m}_{\mathrm{s}}\left\{\psi_1 + \sum_{k=2}^{N}\left(1 - \sum_{j=1}^{k-1} Y_j\right)\Delta\psi_k\right\} \tag{15}$$

The second-law efficiency of the steam generator is

$$\eta_{\Pi,\mathrm{b}} = \frac{\dot{E}_{\mathrm{c}}}{\dot{E}_{\mathrm{g}}} \tag{16}$$

and the second law efficiency of the turbine cycle is

$$\eta_{\Pi,C} = P/\dot{E}_{\mathrm{c}} \tag{17}$$

The irreversibility rates of the components of the turbine cycle are

$$\dot{I}_{\mathrm{t_i}} = T_{\mathrm{o}}\dot{m}_{\mathrm{s}}\sum_{k=1}^{J}\left(1 - \sum_{L=K}^{K+J} Y_L\right)s_{\mathrm{e},k} - \left(1 - \sum_{L=1}^{M} Y_L\right)S_{\mathrm{i}} \tag{18}$$

where J is the number of extractions per turbine and K is the extraction order. M is the number of previous extractions between the exit of boiler and inlet to turbine i.

The irreversibility rate in the condenser is given by:

$$\dot{I}_{\mathrm{c}} = T_{\mathrm{o}}\left\{\dot{m}_{\mathrm{s}}\left[1-\sum_{1}^{L}Y_k\right](S_{\mathrm{e}}-S_{\mathrm{i}})-\frac{\dot{Q}_{\mathrm{c}}}{T_{\mathrm{o}}}\right\} \tag{19}$$

where

$$\dot{Q}_{\mathrm{c}} = \dot{m}_{\mathrm{s}}\left[1-\sum_{k=0}^{L}Y_k\right](H_{\mathrm{e}}-h_{\mathrm{i}})$$

and L is the total number of extractions.

$$\dot{I}_{H,K} = \dot{m}_{\mathrm{s}}\left[\left(1-\sum_{l=1}^{N-1}Y_1\right)S_{\mathrm{e}}-Y_{\mathrm{i}}s_{\mathrm{i}}-\left(1-\sum_{l=1}^{N}Y_1\right)S_{\mathrm{i}}\right] \tag{20}$$

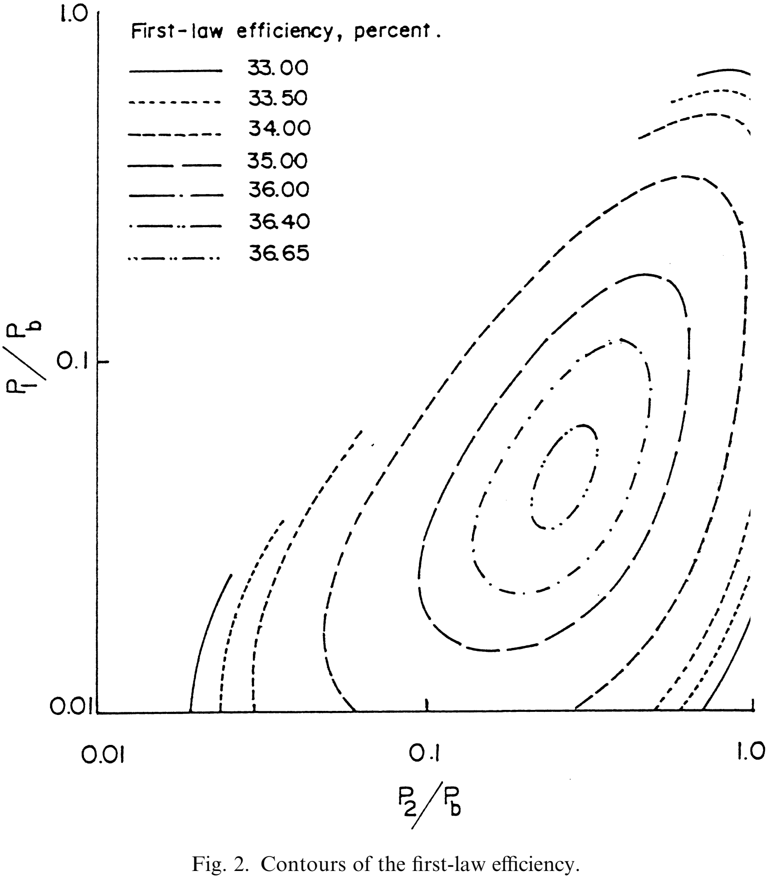


Fig. 2. Contours of the first-law efficiency.

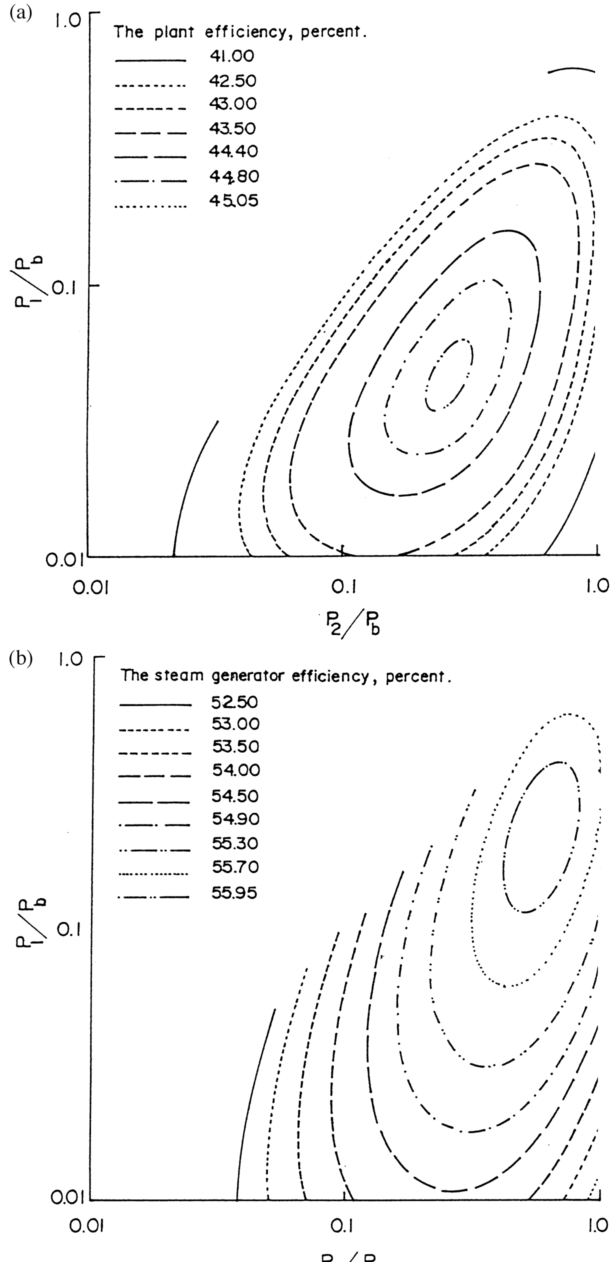


Fig. 3.(a) Contours of the second-law efficiency; (b) contours of the second-law efficiency of the steam generator.

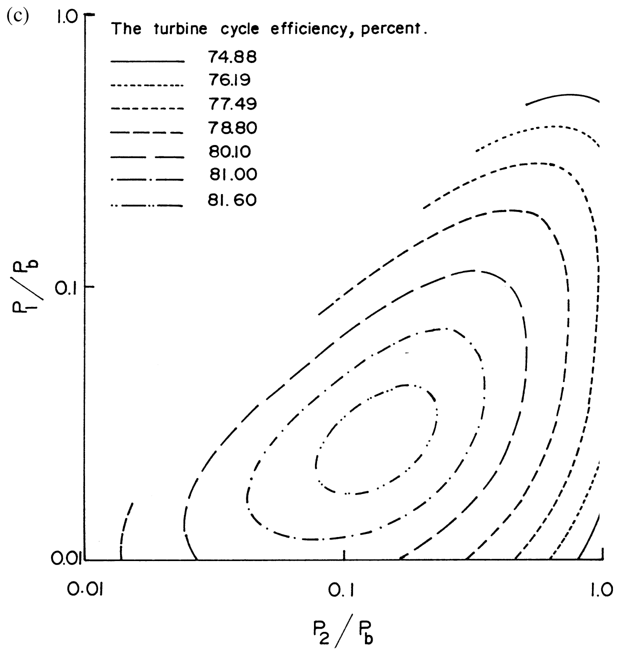


Fig. 3.(c) Contours of the second-law efficiency of the turbine-cycle unit.

Thus, the irreversibility of the turbine cycle is given by

$$\dot{I}_C = \sum_{k=1}^{n} \dot{I}_{t_k} + \sum_{k=1}^{m} \dot{I}_{H_k} + \dot{I}_c \tag{21}$$

where m is the number of feed-water heaters.

In the present study, the boiler efficiency is taken as 88% and the exhaust gas temperature is calculated from

$$\dot{m}_f HHV(1 - \eta_b) = \dot{m}_g C_{pg}(T_{exh} - T_o) \tag{22}$$

where

$$\dot{m}_g = \dot{m}_f\left[1 + \lambda(A/F)_{th}\right]$$

with λ taken as 1.15 and $C_{pg} = 1.26$.

3. Results and discussion

The contours of the first-law thermal efficiency are shown in Fig. 2. The figure indicates a maximum efficiency close to high reheat-pressure P_1 of 25% of the boiler pressure and at low reheat-pressure P_2 of 4.4% of the boiler pressure. The diagonal of the figure with $P_2 = P_1$ presents the single reheat case. For this case, the maximum

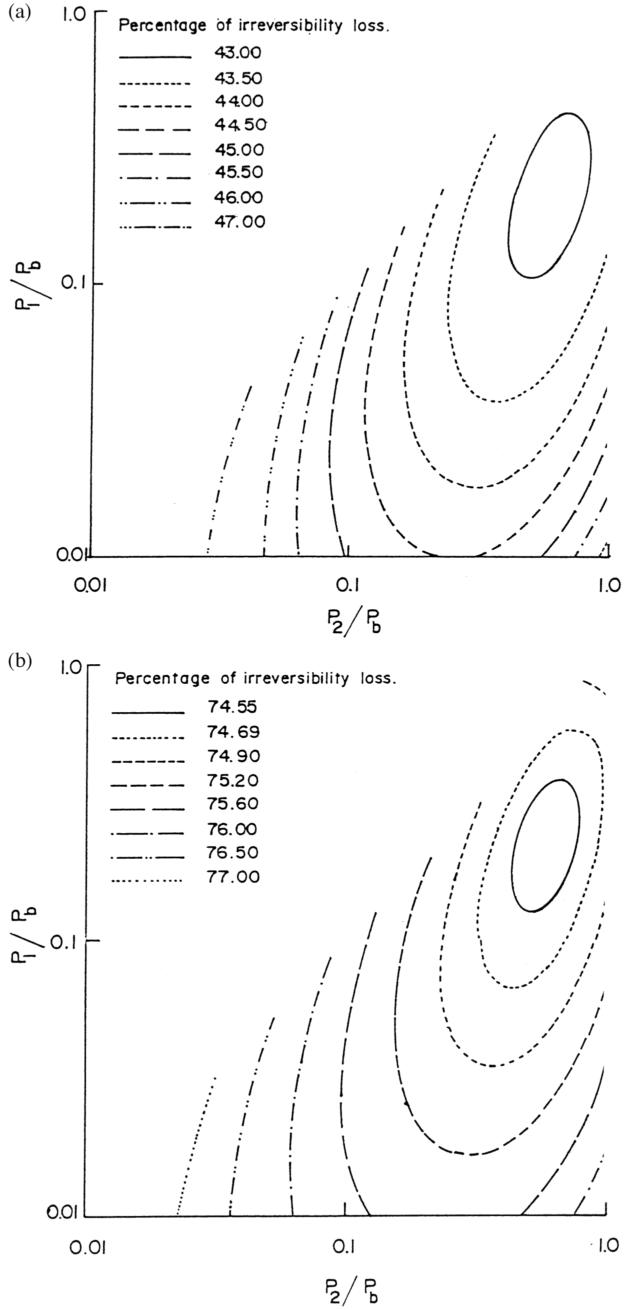


Fig. 4.(a) Contours of the percentage irreversibility losses in the steam generator; (b) contours of the percentage irreversibility losses in the heat-exchanger unit.

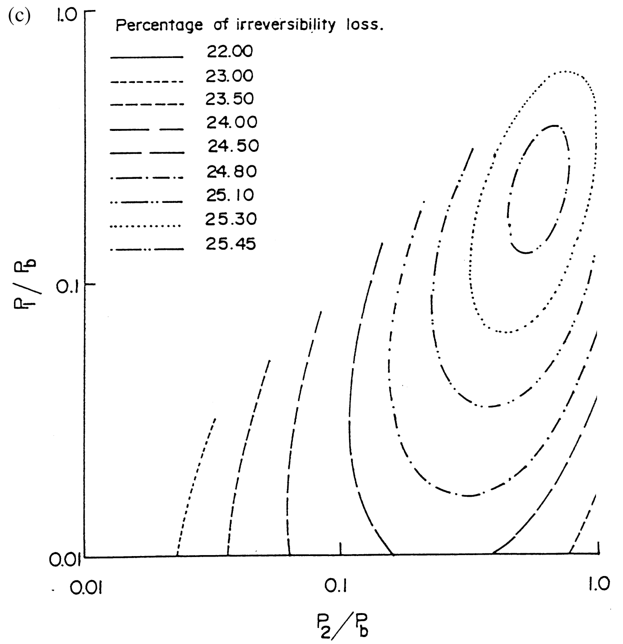


Fig. 4.(c) Contours of the percentage irreversibility losses in the furnace.

efficiency is almost 0.357. Thus the improvement in η_I due to incorporating the second reheat is approximately 0.02 or 5.6%. The contours provide a spectrum for the selection of optimum thermodynamic reheat-level for a specific application. For the same improvement in efficiency, a selection of reheat pressures is available to suit different temperatures at the inlet to the boiler. These temperatures are usually limited by the boiler design. It is anticipated that low temperatures at the inlet to the reheaters at the second reheat-pressure will result in a high temperature at the exit of the low-pressure turbine.

The second-law efficiencies of the plant, steam generator and turbine cycle are shown in Fig. 3a, b and c respectively. Fig. 3(a) indicates an improvement of.0261 or 6.3% in the second-law efficiency due to incorporating the second reheat pressure. Compared with Fig. 2, Fig. 3(a) indicates that the maximum second-law efficiency occurs at the same location as the maximum of the first-law efficiency. Comparison of the three figures [3(a, b, and c)] indicates different optimum conditions of reheat pressures for the plant, the steam generator and the cycle. The optimum steam generator efficiency occurs at $P_2 = 56\%$ and $P_1 = 20\%$ of the boiler pressure. Corresponding values for the cycle are 12.6 and 2.8%. The figure also indicates that the steam generator efficiency is more sensitive to changes in P_2 than in P_1. Cycle efficiency is more sensitive to P_1 than P_2.

At high values of reheat pressures, the temperature difference through which heat is transferred from hot gases to the steam is lower and therefore irreversibilities are expected to be lower. The irreversibility rates in the steam generator are shown in

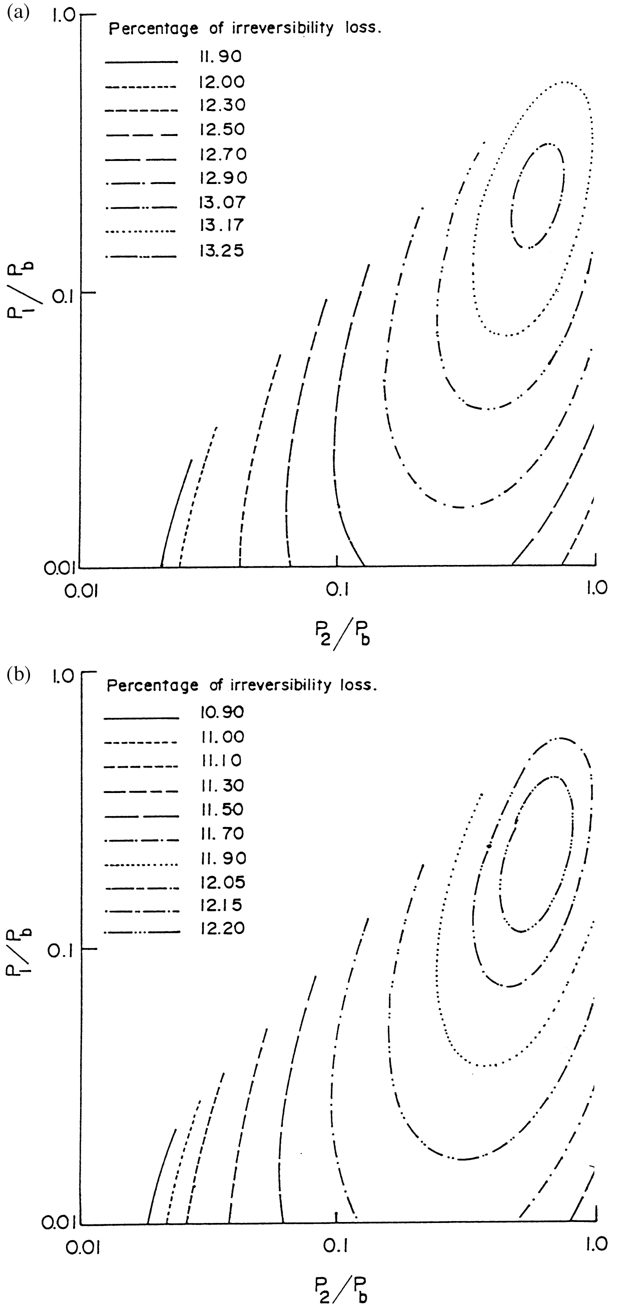


Fig. 5.(a) Contours of the percentage irreversibility losses in the exhaust gases; (b) contours of the percentage thermomechanical/chemical irreversibility losses.

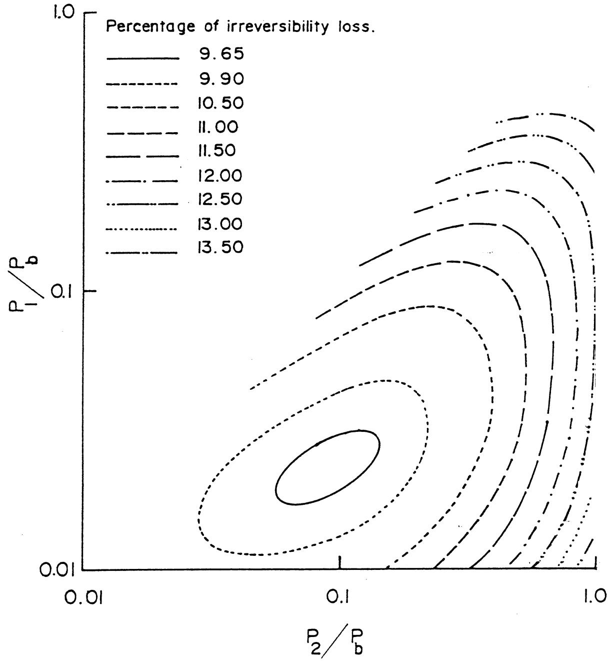


Fig. 6. Contours of the percentage irreversibility losses in the turbine-cycle unit.

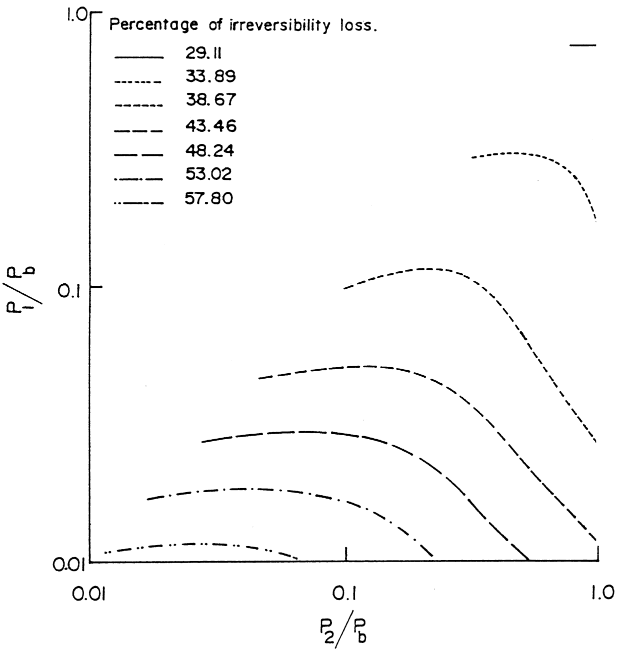
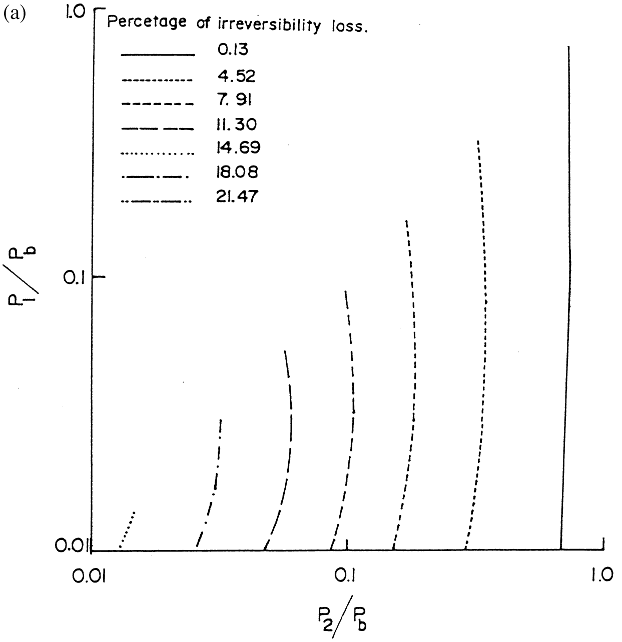


Fig. 7. Contours of the percentage irreversibility losses in the condenser.

(a)



(b)

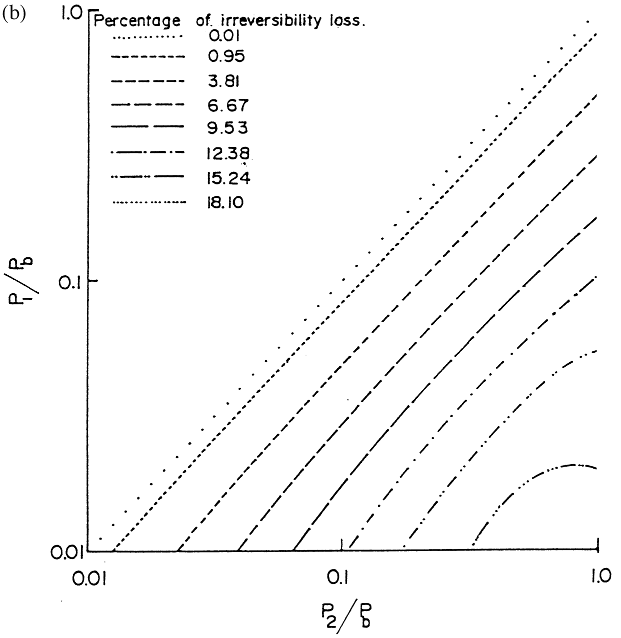


Fig. 8.(a) Contours of the percentage irreversibility losses in the high-pressure turbine; (b) contours of the percentage irreversibility losses in the intermediate-pressure turbine.

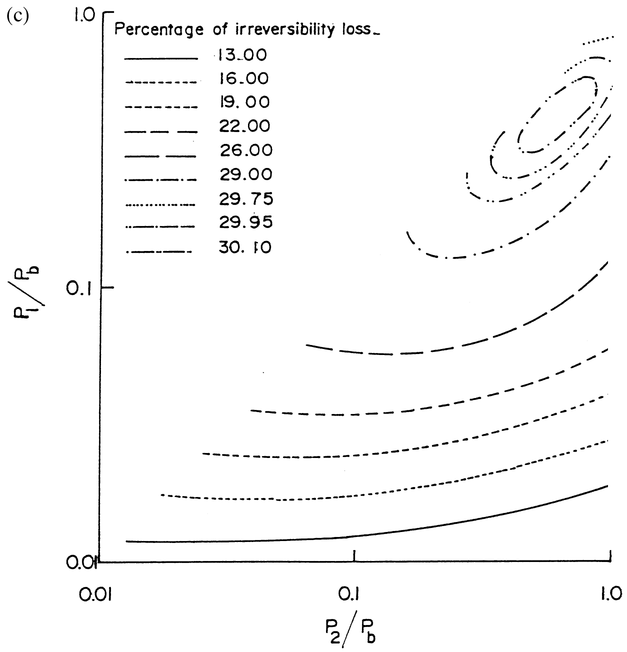


Fig. 8.(c) Contours of the percentage irreversibility losses in the low-pressure turbine.

Figs. 4 and 5. It is indicated by Fig. 4(a) that the minimum irreversibility rates occur at $P_2 = 56.2\%$ and $P_1 = 20\%$ of the boiler pressure. The irreversibility in the steam generator is more sensitive to changes in P_2 than in P_1. The irreversibility losses in the steam generator are due to availability destruction in the furnace and boiler heat-exchanger sections. The irreversibilities in these two sections are shown in Fig. 4(b and c). These figures indicate that the heat-transfer irreversibility losses are more than 3 times greater than the furnace irreversibility losses. The furnace irreversibility losses are due to availability destruction as thermo-mechaincal, chemical and exergy destruction in stack gases. These are presented in Fig. 5(a and b).

The irreversibilities in the cycle and its components are shown in Figs. 6–9. The cycle irreversibility losses are given in Fig. 6 and account for about 12% of the exergy input to the power plant. The cycle irreversibility losses are more sensitive to the low reheat-pressure P_1 than the high reheat-pressure P_2. The irreversibility losses occur at $P_2 = 10\%$ and $P_1 = 2.5\%$ of the boiler pressure. The irreversibility losses due to the cycle components, the condenser, the three turbines and the two feed-water heaters are given in Figs. 7 to 9. The losses in these components at the optimum reheat pressures are 49, 7, 18, 11, 3 and 12%, respectively.

The choice of reheat pressures which may be available to the plant designer to achieve optimum efficiencies is limited by some specific constraints. These are the quality of the steam at exit of each turbine. A typical presentation of these constraints is given in Fig. 10 where the quality values at the exit of the high pressure turbine, x_1,

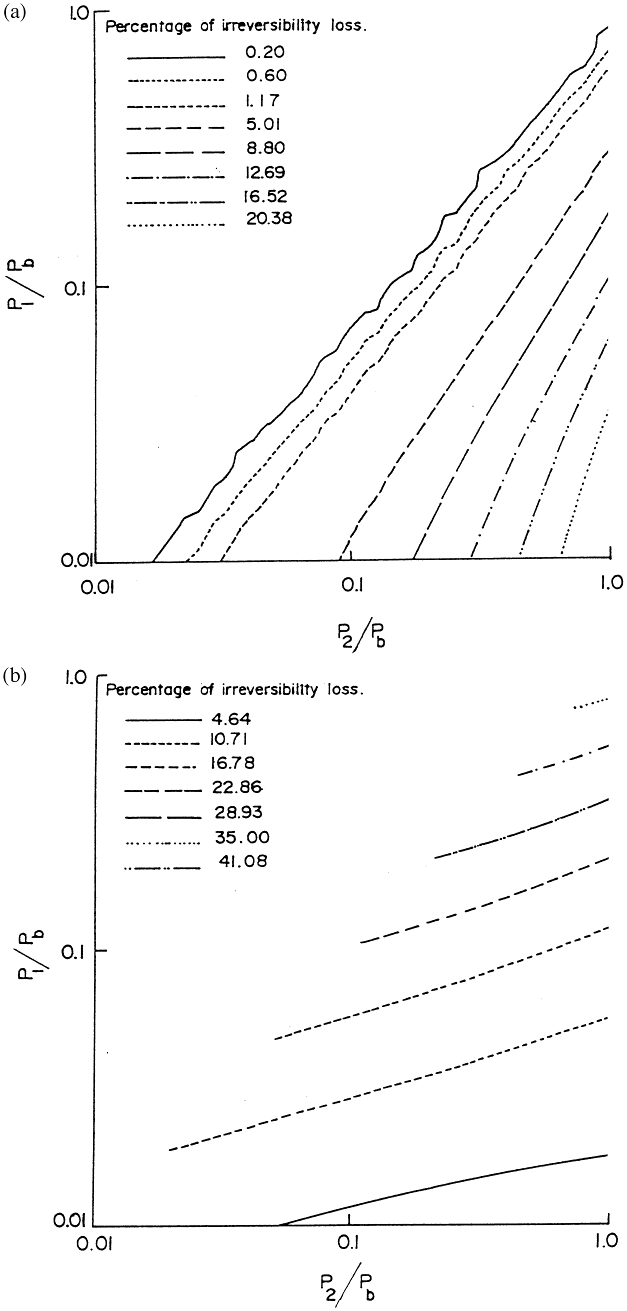
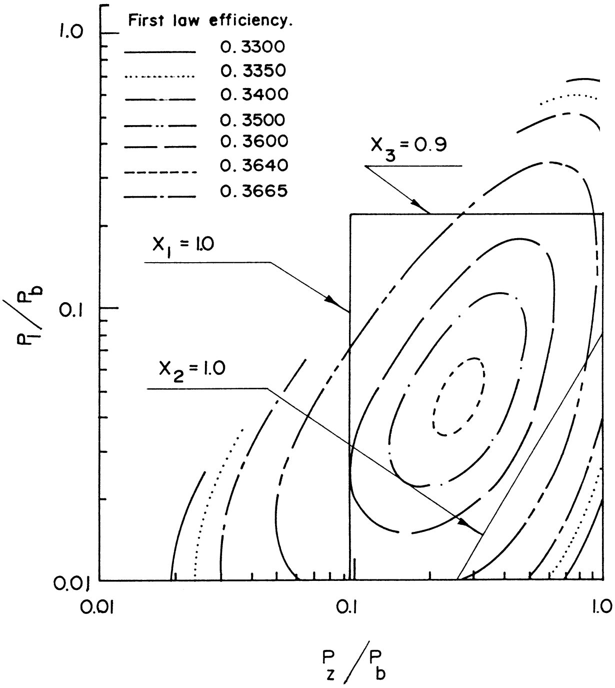


Fig. 9.(a) Contours of the percentage irreversibility losses in the high-pressure feed-water heater; (b) contours of the irreversibility losses in the low-pressure feed-water heater.



10. Effect of constaints due to steam quality at the exits of turbines on the contours of first-law efficiency.

and the intermediate pressure turbine, x_2, were kept above 1.0. In the case of the turbine preceding the condenser, the quality value x_3 was maintained above 0.9. Considering the constraints on quality at exit of each turbine, the choice of the two reheat pressures will be limited to the region bounded by $x_1 = 1.0, x_2 = 1.0$ and $x_3 = 0.9$ as indicated by Fig. 10.

4. Conclusions

A thermodynamic optimization procedure of reheat pressures for the reheat regeneration thermal-plant is presented. The results provide the different optimum low- and high-reheat pressures for each of the steam generator, turbine-cycle unit and the overall plant. It is concluded that the first and second law efficiency are more strongly influenced by the high reheat-pressure than the low reheat-pressure.

Acknowledgements

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