Energy Vol. 15, No. 5, pp. 459–465, 1990
Printed in Great Britain

0360-5442/90 $3.00 + 0.00
Pergamon Press plc

SELECTION OF THE OPTIMAL EXTRACTION PRESSURE FOR STEAM FROM A CONDENSATION-EXTRACTION TURBINE

VOJIN GRKOVIĆ
CHP Kolubara B, 14210 Ub, Yugoslavia

(*Received* 18 *May* 1988; *received for publication* 15 *September* 1989)

Abstract—The minimum exergy loss is a suitable criterion for defining a turbine design for district heating which is optimally adjusted to heat-load variations over a period of time. An analysis of four different turbine versions has yielded optimal turbine compliance with load variations.

1. THERMODYNAMIC BACKGROUND

The exergy is equal to that part of the system energy which can be converted, in a reversible manner, into useful work or electrical energy.[1] The remaining part of the energy has been termed anergy.[2] The part of exergy that is converted into anergy is lost.

The specific exergy of unit mass flow is defined by

$$e = h - h_e - T_e \times (s - s_e), \tag{1}$$

where the subscript e refers to environmental conditions. Kinetic, potential and chemical energy will not be considered. Equation (1) is derived from the first and second laws of thermodynamics. The system exergy loss due to irreversible processes is determined from an exergy balance. The system must be limited by a fixed control boundary through which mass, heat and work enter and leave.

2. CONDENSATION-EXTRACTION TURBINES

Condensation-extraction turbines are used for the combined generation of electrical and heat energy. Electrical energy is transferred to the electrical grid while heat energy is transferred through appropriate heat exchangers to the hot-water network of the district-heating system. The loads for the two systems vary greatly with time. The district-heating system load is determined by the water flow and the water temperatures [in front of and behind the heater H in Fig. 1(a)]. On the steam side of the heater, the saturation pressure must be established for out-flow heating to the required temperature. The pressure at the steam-extraction point must be sufficiently great to compensate for the pressure drop between the turbine and heat exchanger. Independent control of the electric and heat loads is implemented by using the element R in the turbine immediately behind the steam-extraction point.

We assume that the boiler steam flow (BSF) is constant for economic reasons. If variations of BSF are needed, our analysis must be extended. If we assume that the water flow in the district-heating system does not change with time, then the turbine heat load will be proportional to the temperature difference $t_{ow} - t_{rw}$ (Fig. 2). For maximum boiler steam flow and the control element R in the fully open position, the turbine steam-extraction pressure P_{td} allows water heating to the temperature t_{owd} through the temperature difference $\Delta t_d = t_{owd} - t_{rwd}$ (Fig. 2). This pressure P_{td} is called the design-extraction pressure at the loss-free point. A higher outflow water temperature corresponds to a greater heat load and is obtained by closing the control element R. Simultaneously, the pressure at the extraction point is

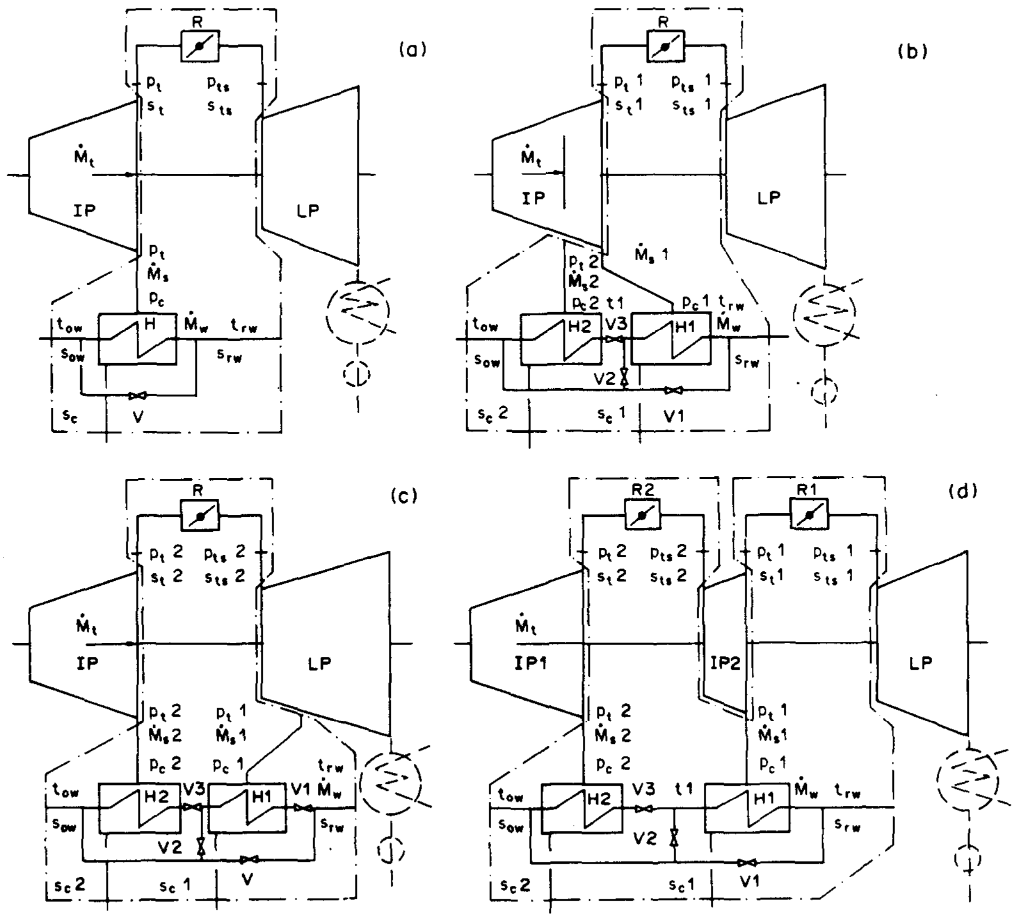

Fig. 1. Schematic representation of optimization procedures for condensation-extraction turbines.

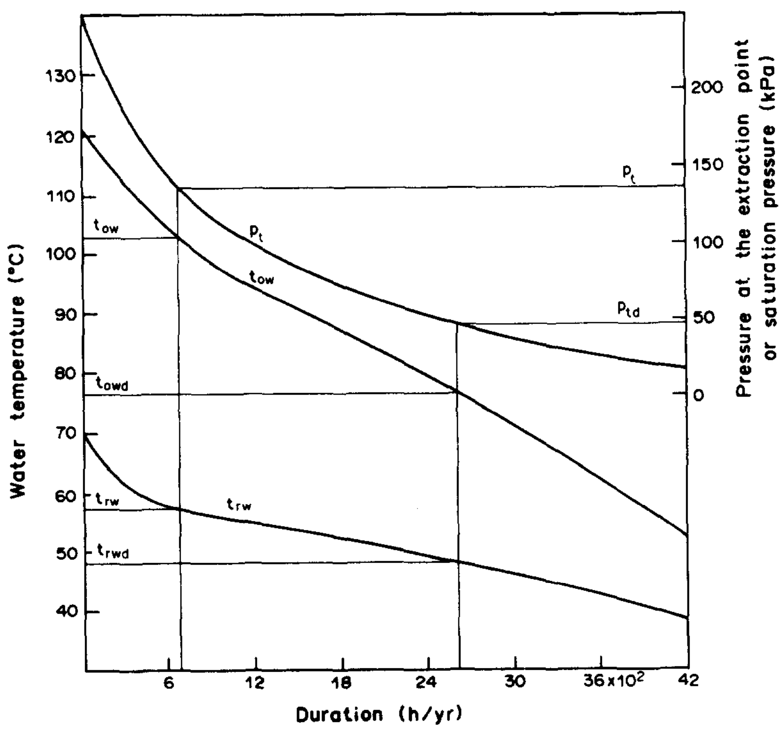

Fig. 2. Outlet and return-water temperature-duration curves at the heater outlet and inlet, respectively.

increased to the needed value p_t and the nonextracted steam flow is throttled by the element R. Due to irreversibility of the throttling process, a part of the steam exergy is lost.

At heat loads lower than the design load, control is obtained by opening the valve V [Fig. 1(a)]. Due to irreversibility of the mixing process, an additional thermodynamic loss is created. The integral over time for both losses depends on the design-extraction pressure. To determine the optimal turbine design, it is necessary to calculate the most favorable design-extraction pressure.

Condensation-extraction turbines with two steam extractions are often used when one or both extraction systems have appropriate control elements [Figs. 1(b), (c), and (d)]. For this arrangements, the optimal design-extraction pressures can be also determined from thermodynamic analysis. Müller[3] has given an exmple in which the gains in electric power are calculated for the turbine arrangement in Fig. 1(a). Hammer[4] has published analyses that are based on calculations of entropy changes and are directly connected with exergy losses for the turbine arrangements in Figs. 1(a) and (b).

3. APPLICATION OF EXERGETIC ANALYSIS TO DISTRICT-HEATING TURBINES

A T-100-130 (U.S.S.R.) turbine was selected for reference in the numerical application. The following basic data apply: live steam conditions refer to 12.75 MPa/550°C/135 kg/sec and no reheat, the water temperature for cooling is 20°C. There are seven stages of feedwater heating to a temperature of 232°C. The turbine has one high-pressure section with a single-row governing stage, one intermediate pressure, as well as one double-flow, low-pressure section in which the last-stage buckets are 550 mm in length. It has a maximum electric capacity for operation with condensation of 120 MW and a maximum district-heating capacity of 200 MJ/sec.[5,6] Design modifications were made in order to determine all variants from Fig. 1 and designs for loss-free points.

By establishing the exergy balance for fluids entering and leaving the control boundary for the turbine with a single controlled steam extraction [Fig. 1(a)], we obtain the following exergy losses in a small time interval:

$$\Delta E(p_{td}) = T_{tc} \times \eta_t \times [(M_t - M_s) \times (s_{ts} - s_t) + M_s \times (s_c - s_t) + M_w \times (s_{ow} - s_{rw})]. \tag{2a}$$

In Eq. (2a), we have omitted the terms which define exergy losses caused by friction in the piping between the turbine and heater and by heat transfer from the heater to the environment. The factor η_t represents the efficiency of the turbine part between the steam-extraction point and the condenser. It accounts for the exergy losses that could, in principle, be converted into useful work by the actual expansion process. It is also assumed that the environmental temperature equals the saturated steam temperature in the condenser.

Equation (2a) may be rewritten as the sum of Eqs. (2b) and (2c), viz.

$$\Delta E_t(p_{td}) = \mathrm{T}_{tc} \times \eta_t \times (M_t - M_s) \times (s_{ts} - s_t), \tag{2b}$$

$$\Delta E_{ht}(P_{td}) = T_{tc} \times \eta_t \times [M_s \times (s_c - s_t) + M_w \times (s_{ow} - s_{rw})]. \tag{2c}$$

Equation (2b) represents the exergy loss due to steam throttling of control element R. Its value depends on the heat load (i.e., on the actual pressure value in the turbine at the extraction point) and on the design-extraction pressure (i.e., on the turbine design at the loss-free point, see Fig. 3).

Equation (2c) represents the total exergy loss due to heat-transfer irreversibility in the heater including contributions from the mixtures of cold-return and hot-outlet water if any. This quantity also depends on the actual pressure value in the turbine at the extraction point and on the design-extraction pressure (Fig. 4).

The total exergy losses during an average heating season or during an average year are obtained by integrating Eq. (2a) over time. For convenient numerical integration, we have

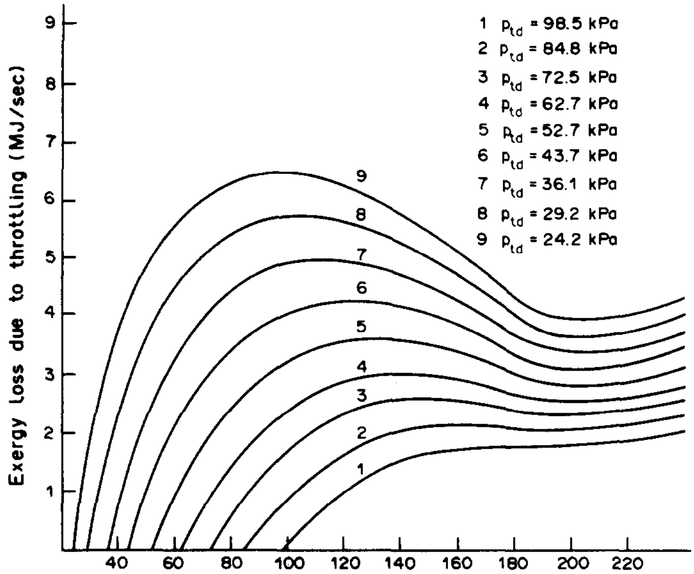

Fig. 3. Exergy losses due to steam throttling as functions of the pressure at the extraction point and the design-extraction pressure.

rewritten Eq. (2a) as

$$\Delta E(p_{td}) = T_{tc} \times \eta_t \times M_w \times (s_{ow} - s_{rw}) + T_{tc} \times \eta t \times [M_s \times (s_c - s_t) + (M_t - M_s) \times (s_{ts} - s_t)]. \tag{2d}$$

The first term represents a change in the exergy of the circulating water. Its value depends on the heat load, maximum heat load (Q_{max}) and share of the turbine heat load in the district heating-systems heat-load (parameter k), as is shown in Fig. 5.

The integral over time of this term does not depend on the design-extraction pressure. The second term in Eq. (2d) represents a change in the steam-flows exergy. Its value is shown in Fig. 6. The integral over time of this term depends on the value of the turbine design-extraction pressure.

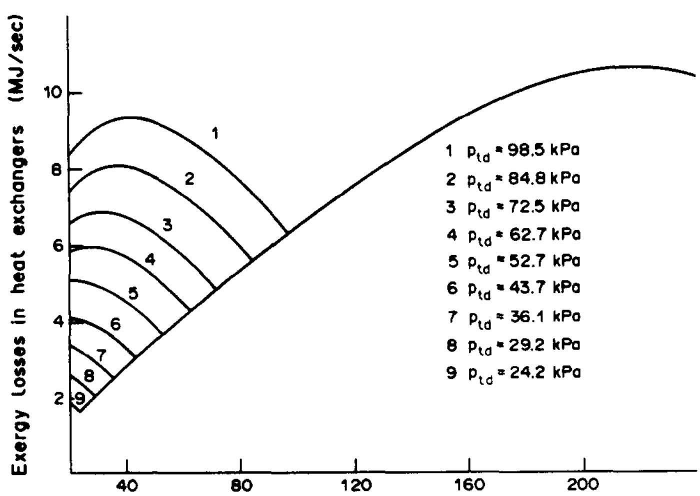

Fig. 4. Exergy losses due to heat transfer as functions of the pressure at the extraction point and the design-extraction pressure.

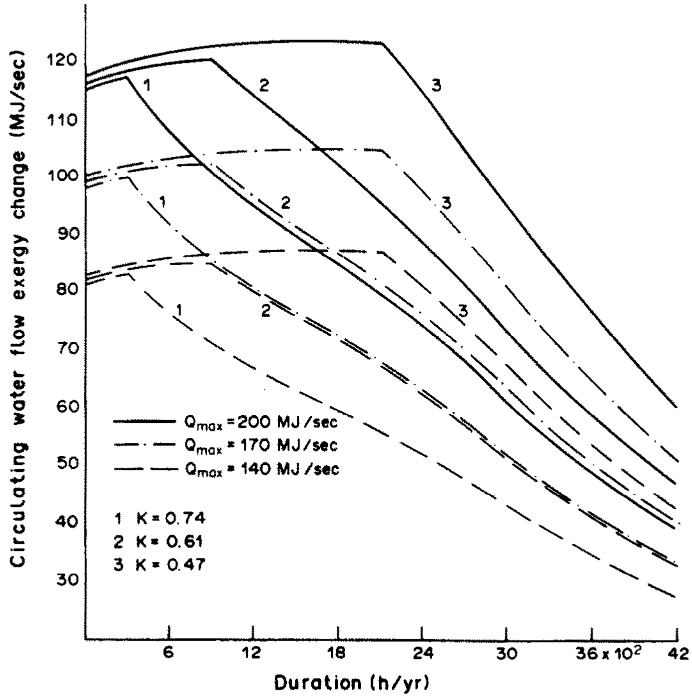

Fig. 5. Change in the exergy of the circulating water as a function of the duration of the heat load.

For the system shown in Fig. 1(b), the following formula is obtained for exergy losses:

$$\begin{aligned}\Delta E(p_{td1}, p_{td2}) = T_{tc} \times \eta_t \times [&(M_t - M_{s1} - M_{s2}) \times (s_{ts1} - s_{t1}) \\ &+ M_{s1} \times (s_{c1} - s_{t1}) + M_{s2} \times (s_{c2} - s_{t2}) \\ &+ M_w \times (s_{ow} - s_{rw})]. \end{aligned} \tag{3}$$

We have allowed for exergy losses due to throttling by the control element R and for irreversible heat transfers.

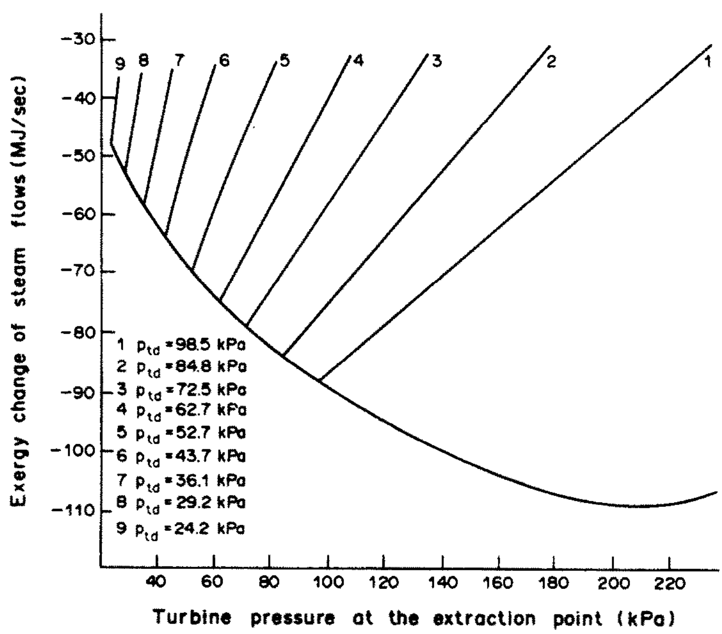

Fig. 6. Extracted steam exergy change as a function of the pressure at the extraction point and the design extraction pressure for version a (see Fig. 1).

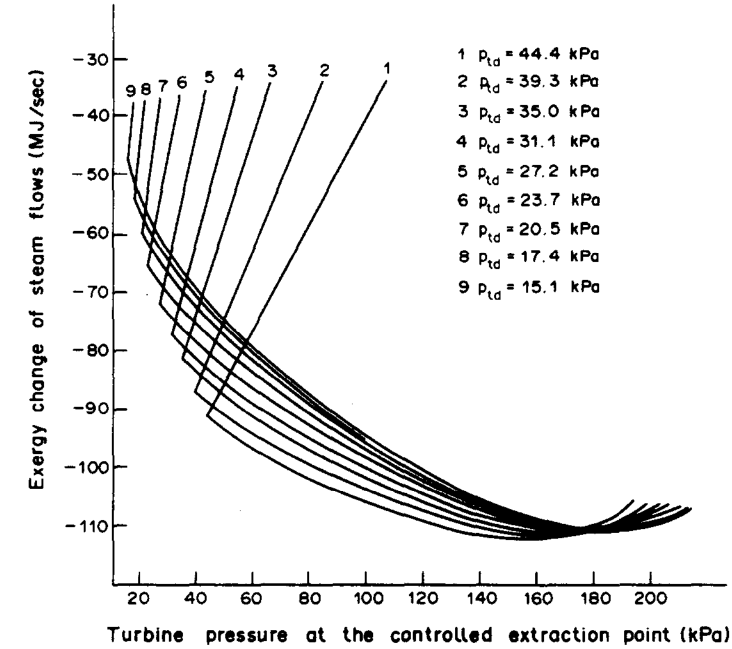

Fig. 7. Changes in the exergy of the steam flows as function of the pressures at the controlled extraction point and the design-extraction pressures for version b.

Using Eq. (3), we may distinguish between terms which correspond to changes in the exergy of the circulating water (for which Fig. 5 is valid) and terms which indicate exergy changes for steam flows. The values of the latter are shown in Fig. 7. The extraction ports on the turbine are so located that both heaters are equally loaded at the loss-free point.

Analogus to the previous cases, we may calculate the steam-flow exergy changes for the turbine versions of Figs. 1(c) and (d).

The graphs in Figs. 3–8 have been obtained from numerically-calculated values.

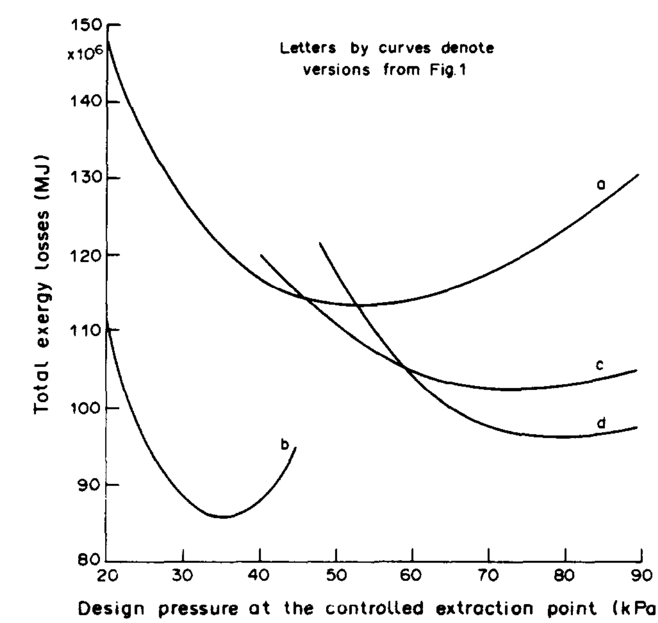

Fig. 8. Dependence of total exergy losses for the heating season on the design-extraction pressure.

4. THE OPTIMUM DESIGN

Our analysis may be used to select the optimum location for bleeding steam used in district heating. The optimum location yields minimal total exergy losses. It is, therefore, necessary to calculate the total exergy losses as an integral over time of the equation for exergy losses. For this purpose, the extraction parameters (extraction pressure, temperature, enthalpy, entropy, etc.) must be calculated at off-design heat-load conditions for assumed time steps. A specially-designed computer program is used.[7] It was developed by using system simulation[8] and steam-turbine theory. Corresponding values for the temperature of the water that is used to cool the condenser are obtained by applying a correlation using statistical meteorological data. The exergy losses, as defined by Eqs. (2) or (3), must be calculated. The total value of the exergy losses is obtained from the trapezoidal-rule routine.

Calculated curves of the total exergy losses during the observation period vs design steam-extraction pressures for all four versions (Fig. 1) are presented in Fig. 8. For versions b and c, the curves refer to the design pressure at the controlled extraction point; for version d, they refer to the warmer extraction point. The design-extraction pressure for minimum exergy loss is the optimal pressure.

It is apparent that version b yields minimal total exergy losses. The exergy-loss reductions correspond to gains of 7.5×10^6 and 3×10^6 kWh/yr for version b, as compared to versions a and d, respectively. Each version has its own optimal extraction-pressure value at the loss-free point. Furthermore, each turbine version exhibits different sensitivities to the design-extraction pressure, with version b being most and version a least sensitive.

REFERENCES

1. Z. Rant, *Fortschr. Ing. Wiss.* **22,** 36 (1956).
2. Z. Rant, *Strojinski Vest.* **8,** 1 (in Slovenian) (1962).
3. H. Müller, *Wärme* **76,** 78 (1970).
4. H. Hammer, *Mitt. VGB* **49,** 210 (1969).
5. E. J. Beneson and L. S. Joffe, *District Heating Steam Turbines* (in Russian), Energy, Moskva (1976).
6. A. V. Sheglaev, *Steam Turbines* (in Russian), Energy, Moskva, (1976).
7. V. R. Grkovic, *BWK* **39,** 349 (1987).
8. W. F. Stocker, *Design of Thermal Systems,* McGraw–Hill, New York, NY (1971).

NOMENCLATURE

e = Exergy (kJ/kg)
M_s = Extracted steam flow (kg/sec)
M_t = Turbine steam flow before extraction (kg/sec)
M_w = Water flow (kg/sec)
p = Pressure (Pa)
s_c = Entropy of condensate (kJ/kg K)
s_{ow} = Entropy of outflow water (kJ/kg K)
s_{rw} = Entropy of return water (kJ/kg K)
s_t = Steam entropy at the extraction point (kJ/kg K)
s_{ts} = Steam entropy after throttling (kJ/kg K)
T_{tc} = Saturation temperature in the turbine condensor (K)

Greek letters

η_t = Turbine officiency

Subscripts

1 = Lower heat
2 = Upper heater
d = Design
t = Turbine