PEPSE* MODELING OF FINNED TUBE HEAT EXCHANGERS FOR A COMBINED CYCLE PLANT

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ABSTRACT

Since last year a lot of work on combined and repowered cycle plants has been done in CISE to support ENEL, the Italian National Electricity Board, in his thermodynamic evaluations.
This report shows the theoretic and code problems arisen and the solutions adopted during the simulation of a dual pressure heat recovery steam generator (HRSG) with finned tubes applied to a combined cycle power plant.
The aim of the work was to evaluate the behaviour of the plant at different loads and conditions. It required design mode calculations, for which a set of about three hundred operations was defined.
The complex geometry and the difficulties in the convergence of calculation required particular and sophisticated techniques in the simulation.
The obtained model was extensively used for a complete load range analysis of several plants. An example is described in a second paper.

FOREWORD

CISE is a research Company developing innovation technologies and transferring them to industry. CISE's activities cover the fields of energy production, environmental protection and manufacturing industry. In the field of energy production CISE can contribute in solving design, control and diagnostic problems in the field of heat transfer, fluid dynamics, structural engineering and nuclear engineering. Its major customers are ENEL (the Italian national electricity board), domestic and foreign industry, national and international research agencies: ENEA, EEC, ESA.

ENEL is the Italian national electricity board which operates about 45000 MW of power generating plants. In 1988 ENEL produced more than 150 TWh of electricity. ENEL is also responsible for design and construction of its power plants.

1. INTRODUCTION

CISE - Tecnologie Innovative is involved in the ENEL retro-fitting plan of some italian power stations. More precisely, two sections of CISE - Thermo-fluid-dynamic Processes Section and Thermo-fluid-dynamic Models Section - together with ENEL-DCO, are giving particular attention to the first stage of this study, that regards the analysis of thermodynamic advantages of combined or repowered cycles.

To this aim, PEPSE is extensively used to evaluate different repowering solutions for fossil units ranging from 75 MW to 660 MW. Natural gas-fueled combustion turbines are used to repower fossil units by means of an added Heat Recovery Steam Generator (HRSG).

All analyses concern existing plants that have been in commercial operation for many years. Two reasons condition the choice of the repowered configuration, besides efficiency considerations:

- in most cases there isn't space enough to install new turbo-machinery close to steam cycle. Since gas-turbine have to be positioned some hundreds meters far away, very high pressure pipelines costs become very high, conditioning the choice of the repowered configuration;
- the need of increasing as much as possible the electric power production.

Therefore the tested solutions are the following:

- direct feedwater repowered configuration: figure 1 shows a schematic diagram with two heaters on exhaust gas line. Also solutions with only one heater (the high pressure one) have been considered. This scheme has been adopted for existing 320 MW size plants, while it proved not to be suitable for lower size units. In fact, as the feedwater flow rate becomes lower, the heat recovery from from gas turbine exhaust, and thus the overall efficiency, decreases. Even if its efficiency is not the highest reachable (fig. 2), selection of this scheme for existing 320 MW unit is due to its operational flexibility and minimum impact with existing components and layout;

- supplemental boiler repowered configuration (fig. 3): a fraction of feedwater from the deaerator output is pumped to a supplemental boiler at intermediate pressure. The generated steam is mixed with reheat steam and conveyed to the IP turbine. This scheme has been chosen for a conventional 660 MW units plant under construction. These units have hyper-critical steam generators, therefore the design pressure of HP feedwater lines is very high (350 bar - 5080 psig). If the direct feedwater repowered configuration had been chosen, this would have led to a very high wall thickness (105 mm - 4" 1/3) of the long pipelines (about 1600 m - 5250 feet) which connect the heater to the steam cycle, with disadvantages in terms of costs, construction time and reliability. Figure 4 shows efficiency curves of this scheme;
- full repowered configuration: (fig. 5) given the standard size (110 MW) of gas turbines to be used, it is the most interesting scheme for plants sized 150 MW or lower, since it allows a significant increase of both generating capacity and efficiencies (see diagram of fig. 6).

A PEPSE data set is prepared for each plant to be analyzed, in order to fit its design performances. In most cases, different loads are tested, and also because of the many plants to be analyzed, the models are required to be able to fit performances at different loads only changing very few data (for instance, exhaust gas inlet conditions). Successively a HRSG is added, substituing or in addition to the existent boiler.

Even if different schemes are investigated with the same methodology, this report mainly describes full repowered configurations, called "combined cycle" cases. For all of these, a dual pressure HRSG is chosen, according to EPRI analysis[1]. Therefore, each HRSG is generally composed of six finned tube heat

exchangers, i.e. an economizer, an evaporator with boiler drum and a superheater for each pressure level.
For the simulation, the number, size and configuration of tubes are calculated with another program on a personal computer.
This report shows the methods applied to model the plants and assist the convergence of calculations. The application of these methods to a particular case and an evaluation of results are illustrated on another paper.

2. FULL REPOWERED CONFIGURATIONS

Generally, two gas turbine groups with two Heat Recovery Steam Generators (HRSGs) are considered. According to EPRI analysis[1], most cases have a dual pressure HRSG. Few plants have an HRSG with three pressure levels (fig. 7). This report mainly describes the work on dual pressure configuration.
The combustion gas turbines are not considered in the model. A source component is used to generate the exhaust gas flow to each HRSG, with temperature and mass flow rate given as functions of gas turbine generated power, as stated by the manufacturer.
Depending on the aim of the analysis, different schemes are used:
- one that allows tests with two HRSGs working at different conditions (fig. 5);
- one for the detailed analysis of the plant with symmetric HRSGs (that is the normal working condition). An example is shown in fig. 8. In this case only one HRSG is represented. The other is simulated with a sink and two source components: these feed the cycle with steam whose characteristics and flow rates are imposed to be equal to those of the represented HRSG. With a model based on this scheme a by-pass of the economizers is allowed, to prevent cold-end corrosion when burning liquid fuel in gas turbine. Besides, a recirculating line is included to avoid a too low

temperature of water at inlet of the LP economizer when burning natural gas.

Usually the following sequence of heat exchangers in series along the exhaust gas flow path is adopted:

- . Superheater - high pressure (SH-HP)
- . evaporator - high pressure (EVA-HP)
- . superheater - low pressure (SH-LP)
- . economizer - high pressure (ECO-HP)
- . evaporator - low pressure (EVA-LP)
- . economizer - low pressure (ECO-LP)

When a reheater is present, its tube rows are physically alternated with the SH-HP ones. Since this configuration is not possible with PEPSE code, unless a simulation of each tube row with a type 28 component is performed, the tested solution have RH and SH-HP in parallel (fig. 9). The exhaust gas flow rates through these two heat exchangers are evaluated in order to have the same temperature at their outlets.

Since the analyses began when boiler drum component was not available on PEPSE code, other components had to be used to impose saturated conditions for evaporator inlet fluid. Initially a mixer and a moisture separator were used (figure 10/A), but soon this schematic was left because at each iteration quality and mass flow rate in the line connecting the mixer and the moisture separator were highly unstable. Therefore a contact heater was used (fig. 10/B). For a correct simulation, a dummy line without flow was used to have the contact heater pressure equal to feedwater inlet one. With such a model, the evaporator flow rate was smaller than the real one (about five times the feedwater flow rate) and outlet quality was one instead of about 0.2. But that was negligible because saturation temperature and heat transfer coefficient did not change. Since when PEPSE last version became available, boiler drum component has been used, except for all those cases in which the drum pressure is controlled by an extraction from a turbine (fig. 9).

3. CALCULATION PROCEDURES

The methodology used in the study of a typical plant is the following:

step 1) a reference case is obtained simulating the existing plant at full load by means of an input file easily adaptable to other different loads (usually 70% and 40% of full load). For almost all cases type 4 to 7 turbine components are used, while all heat exchangers (condenser, feedwater heaters, etc.) are calculated in performance mode. The results have to be reasonably similar to design performances provided by ENEL;

step 2) an HRSG is added to the reference model at full load using type 28 components calculated in performance mode. Pinch point, approach and subcooling temperature differences (see diagram of fig. 11) are input data stored in operational variables. Then, a set of operations (88NNN0 cards)[2] is prepared for the following PEPSE calculations:

- pressure P_{HP} for steam going to the throttle valve is controlled to have the valve wide open. Thus, high pressure is function of steam flow rate:

$$P_{HP} = P_{HP\ previous\ iteration} * (EQTFR)^n$$

where: EQTFR = current equivalent throttle flow ratio
n = relaxation exponent (usually 0.17 - 0.20)

- pressure P_{LP} for steam to intermediate (or low) pressure turbine section is controlled to be equal to HP (or IP) turbine shell pressure (P_{shell}), in order to avoid thermal dissipations:

$$P_{LP} = P_{LP\ \text{previous iteration}} * \left[\frac{P_{shell}}{P_{LP}}\right]^{n}_{\text{previous iteration}}$$

- superheater and economizer tube side outlet temperatures (TTTORH variables[2]) are obtained the former subtracting the approach temperature difference to the exhaust gas inlet temperature, the latter subtracting the subcooling temperature difference to the saturation temperature in the boiler drum;

- mass flow rate through each boiler drum is controlled by the pinch point temperature difference. A method for that is described below;

<u>step 3)</u> the obtained thermodynamic inlet and outlet conditions of exhaust gas and steam or water are elaborated by a program running on a personal computer to calculate the geometric characteristics of each HRSG tube bundle. Some hypothesis were defined:

- exhaust gas flow cross section have to be rectangular (1.25 is the ratio of the sides), constant through the whole HRSG and dimensioned to have a pressure drop of 230 mm (9") of water gage on gas flow;

- finned tube dimensions are fixed;

- no parallel tube rows in superheater and economizer (i.e. just one tube row leaves from the headers).

At this step an economic optimization is done, analizing fuel savings and increments in energy productions and plant costs;

step 4) the obtained geometric characteristics are used to modify PEPSE input files to allow design mode calculations of type 28 components. This step is necessary to verify the behaviour of the adopted configurations and solutions at low loads, especially in order to avoid steaming conditions in the economizers or high moisture in the last LP stage of turbine. To this aim, heat transfer equations are specified through a set of operations (see next section) and controls on pinch point, subcooling and approach temperature differences are disactivated. Pressures are still calculated as described for step 2, while mass flow rates, when not calculated directly by PEPSE (see section 5.) are evaluated in the following way:

$$\Gamma_{HP} = \Gamma_{HP\ \text{previous iter.}} * \left[\frac{X_{out\ EVA\ HP}}{EQTFR}\right]^{n}_{\text{previous iteration}}$$

$$\Gamma_{LP} = \Gamma_{LP\ \text{previous iter.}} * (X_{out\ EVA\ LP})^{n}$$

where: $X_{out\ EVA}$ = quality at boiler drum outlet
n = relaxation exponent (usually 0.17 - 0.20);

step 5) at this point, a complete comparison between standard and repowered configurations is possible, either at full or partial load. In many cases, also different gas turbines outlet conditions are tested, either supposing to adopt a different turbine or simply changing load or air conditions (i.e. modifying exhaust temperature and mass flow rate).

4. FINNED TUBE HEAT TRANSFER COEFFICIENT

PEPSE does not contain finned tube calculations. A heat transfer coefficient multiplier or a corrected heat transfer area could be used, but that would give inaccurate results at partial loads. This way could be followed if there is the possibility to specify a correction factor only for outside tube heat transfer coefficients. Therefore the application of external heat transfer equations[1] become necessary. Thus, a set of about 250 operations has been prepared to calculate finned tube inside and outside heat transfer coefficients. For that, PEPSE code has been modified to allow up to 400 operations. Actually, a dedicated routine in the code could be written, as indicated by EI International. Initially the use of operations was choosen because it appeared to allow a better debug and control of calculations.

The used heat transfer equations, written with the operations, are the following:

outside heat transfer coefficient:

$$\alpha_{out} = 0.183 * \frac{k'}{D'} * (Re')^{0.63} * FF$$

SH inside heat transfer coefficient (PEPSE uses Dittus Boelter equation):

$$\alpha_{in} = 0.01087 * \frac{k}{D} * (Re)^{0.861} * Pr$$

[1] J.Vampola - Generalization of the laws governing heat transfer and pressure drop during tranverse flow of gases in finned tube banks - Heat and mass transfer - 1965 - Vol. 1

ECO inside heat transfer coefficient (Dittus-Boelter equation, same as PEPSE):

$$\alpha_{in} = 0.023 * \frac{k}{D} * (Re)^{0.8} * (Pr)^{0.4}$$

The meanings of symbols are:

α = heat transfer coefficient (W/m^2 °C)

k' = exhaust gas conductivity (W/m °C)

D' = average outside finned tube diameter (m)

Re'= exhaust gas Reynolds Number (referred to D')

FF = fouling factor

k = water or steam conductivity (W/m^2 °C)

D = outside tube diameter (m)

Re = inside tube Reynolds Number (referred to outside diameter)

Pr = inside tube Prandtl Number

EVA inside coefficient is assumed to be constant and equal to 10,000 W/m^2 °C

Global heat transfer coefficient for each tube bundle, referred to external finned tube area, is then calculated as follows:

$$\alpha = \frac{1}{\frac{1}{\alpha_{in}} * \frac{A3}{A1} + \frac{1}{FTA * \alpha_{out}} + \frac{S}{k_T} * \frac{A3}{A1}}$$

where A1 = outside tube area (fins area excluded) per unit length (m^2/m)

A3 = outside finned tube total area per unit length (m^2/m)

S = corrected tube thickness (m) = $D/2 * \log (D/D_{in})$

D_{in} = inside tube diameter (m)

k_T = tube matherial conductivity (W/m °C)

$$FTA = \text{fin efficiency} = \frac{\exp(UB) - \exp(-UB)}{\exp(UB) + \exp(-UB)} * \frac{1}{UB} - 2 * .06 \frac{HAL}{D} * UB^{.585}$$

$$UB = HAL * \sqrt{(2\, \alpha_{out} / (k_T * Y))}$$

HAL = fin width (m)

Y = fin thickness (m)

Global heat transfer coefficient is calculated with operations, assuming fin efficiency FTA constant through all calculations (it was determined manually). The result is automatically stored in HTTIRH variable. Consistently in AATIRH variable the finned tube external area must be written.

5. CONVERGENCE PROBLEMS AND ADOPTED SOLUTIONS

All the problems discussed above lead to a model for which calculations have great difficulties in converging. Main problems are connected with the following situations:

1) exhaust gas and steam flows in the whole HRSG are counter-current. That means that a solution can be found only with several iterations;

2) pressures in HRSGs are controlled as described before;

3) steam at SH inlets (i.e. boiler drum outlets) must have quality equal to one (saturated steam). That requires a control on high

pressure and low pressure mass flow rates. It is done in the following ways:

- if a boiler drum component is used and the evaporator is simulated with a type 28 component in design mode, PEPSE is able to correct flow rates by means of a demand splitter or a source component;
- if the evaporator is simulated with a type 28 component in performance mode, other operations are required. The reason is evident if the schematic diagram is without boiler drum component (contact heater, splitter and mixer components cannot control feedwater flow rates). If a boiler drum component is used, the feedwater flow rate control through PEPSE options becomes impossible (if drum circulation ratio is inconsistent with evaporator outlet quality) or undefined (if drum circulation ratio is consistent with evaporator outlet quality).

Every case and almost every run requires particular modifications on input data set because of calculation problems. In general the following methods work quite successfully:

- source components are used when steam or exhaust gas conditions and/or flow rates are known or easily achievable with some operations. More in details:
  - when pinch point temperature differences are input data, exhaust gas temperatures at evaporator outlets are calculated from steam saturation temperature;
  - at superheater inlets steam quality is fixed to one and mass flow rate has to be equal to the boiler drum inlet one;
  - HRSG inlet flow rate must be equal to the sum of low pressure and high pressure steam flow rates;

- when pinch point and superheater approach temperature differences at high pressure are input data, high pressure steam flow rate is calculated with an energy balance. Therefore for each saturation pressure the steam flow rate is immediately obtained, without the need of multiple iterations. In some cases, this method is applied also to the low pressure HRSG section, but that is a little more difficult because beetween the EVA-LP and the SH-LP there is the ECO-HP.

6. SUMMARY

Efficiency improvement and power increase of many existing fossil plants are the aim of the analysis described in this report and in a second one.

The main problems in these studies are connected with the complexity of the proposed configurations. Since many plants and configurations were to be tested, manual interventions has been reduced as much as possible.

Submodels for the boiler drum and for the HRSG have been developed and included in the global plant model. The introduction of some source/sink components and the use of operations to control a number of flow rates and pressures are some of the solutions that had to be adopted to assist calculation convergence.

A method for finned tube calculations of heat transfer coefficient has been developed based on a set of operations (cards 88NNN0).

An example of the application of this analysis is shown in an other report.

REFERENCES

(1) EPRI - Practical Feasibility of Advanced Steam Systems for Combined-Cycle Power Plants - May 1988

(2) G.L. Minner, E.J. Hansen, W.C. Kettenacker, P.H. Klink - PEPSE Manual. User Input Description - EI International

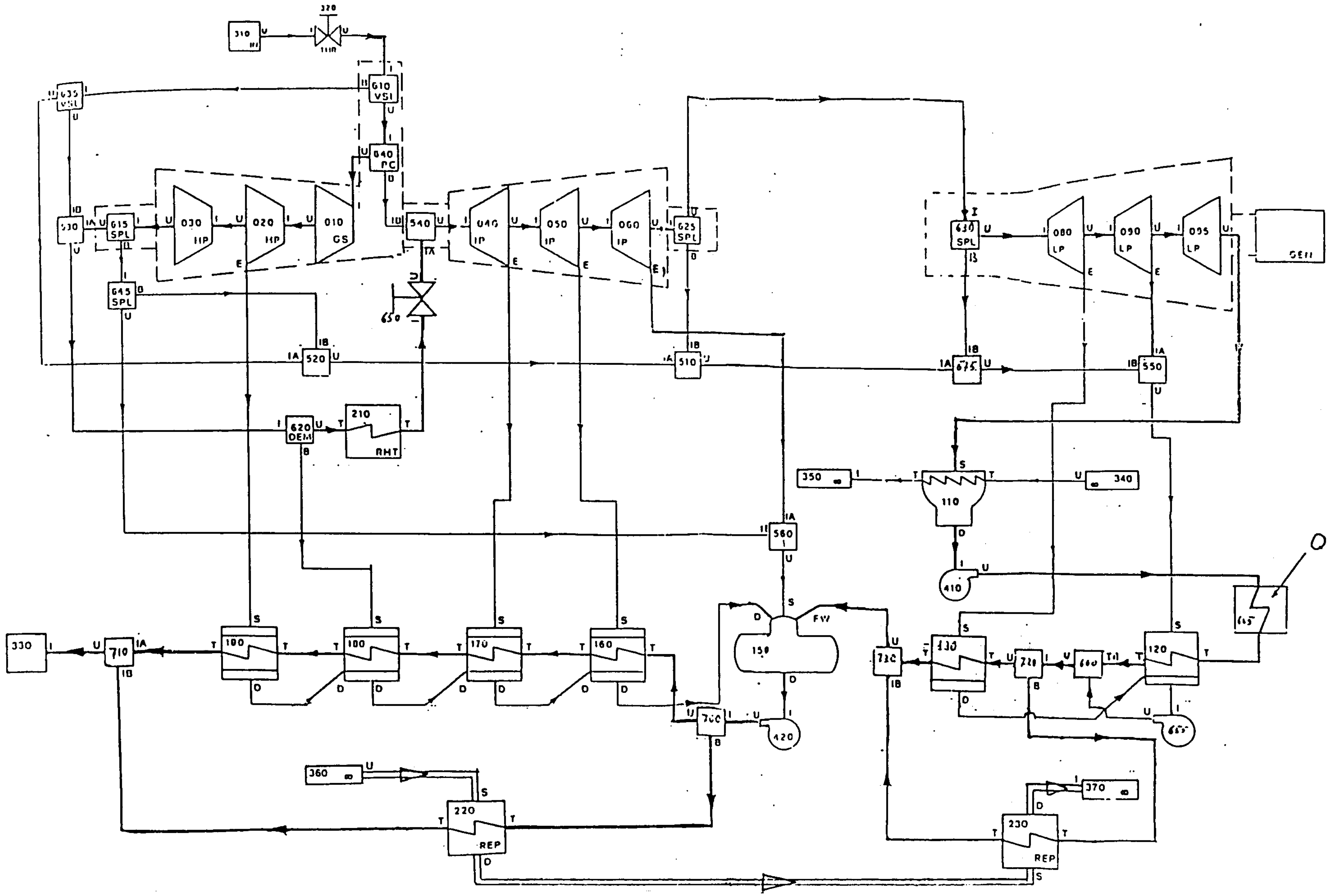


Fig. 1 - Schematic diagram of a direct feedwater repowered cycle

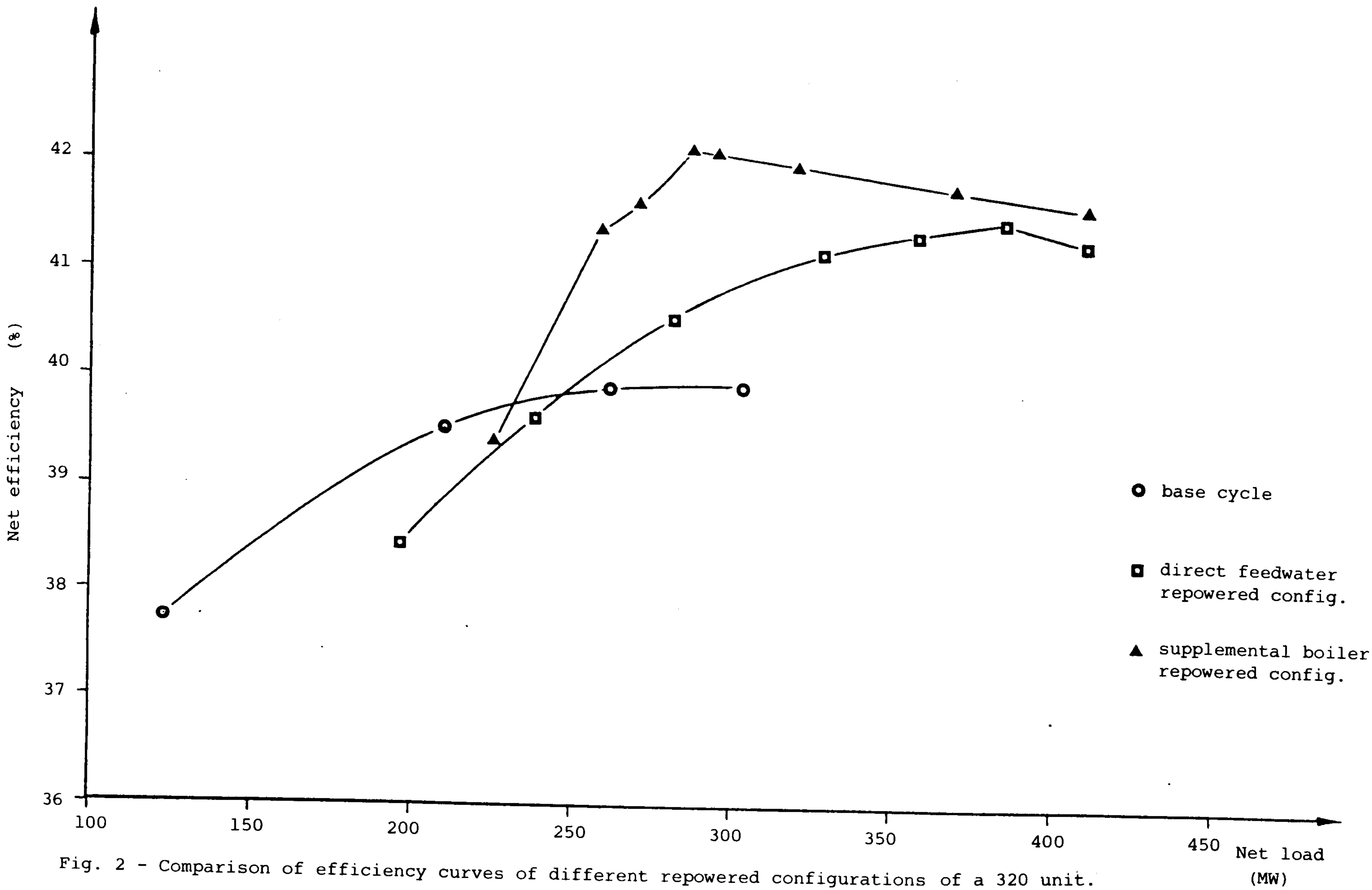


Fig. 2 - Comparison of efficiency curves of different repowered configurations of a 320 unit.

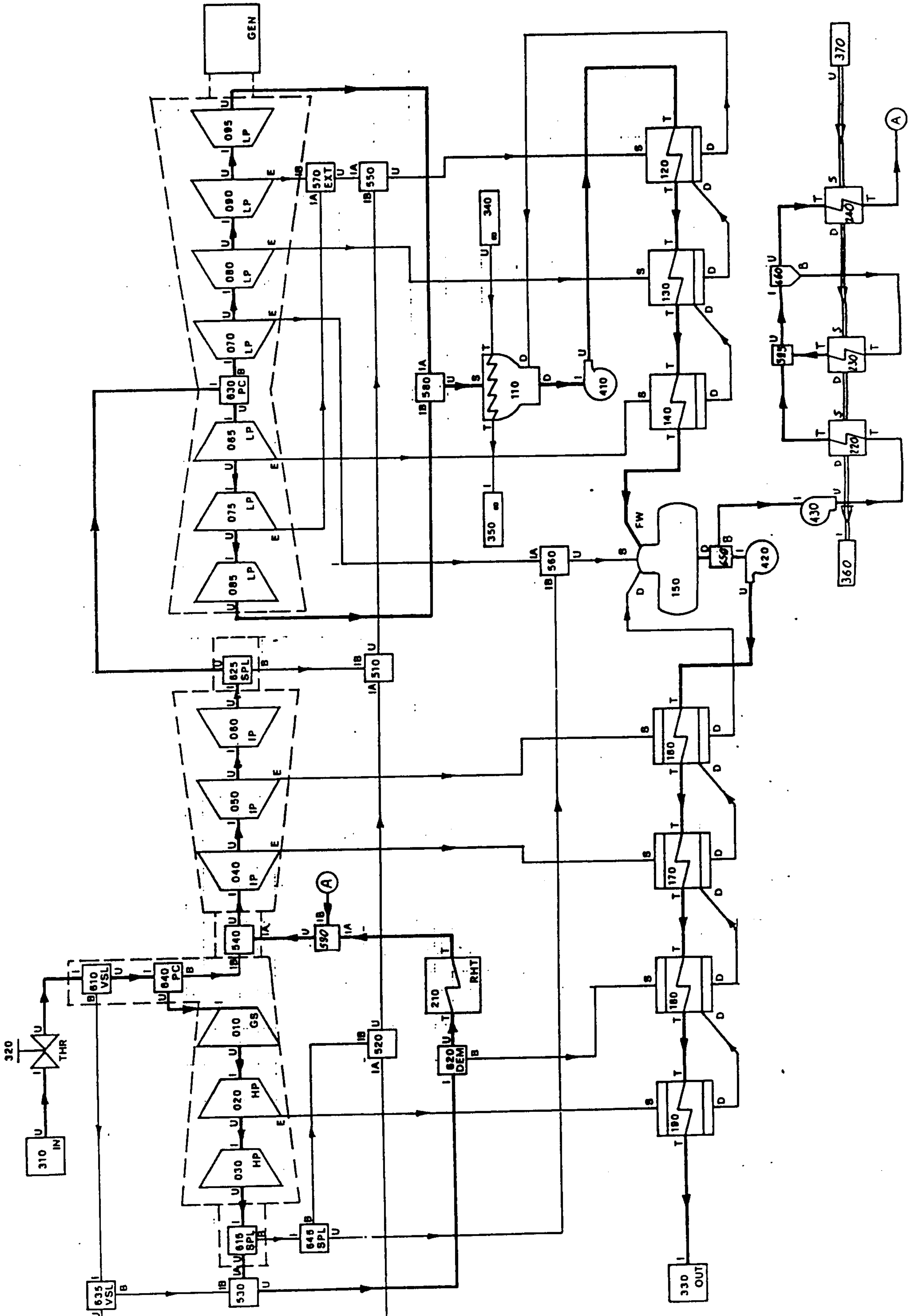


Fig. 3 - Schematic diagram of a supplemental boiler repowered cycle.

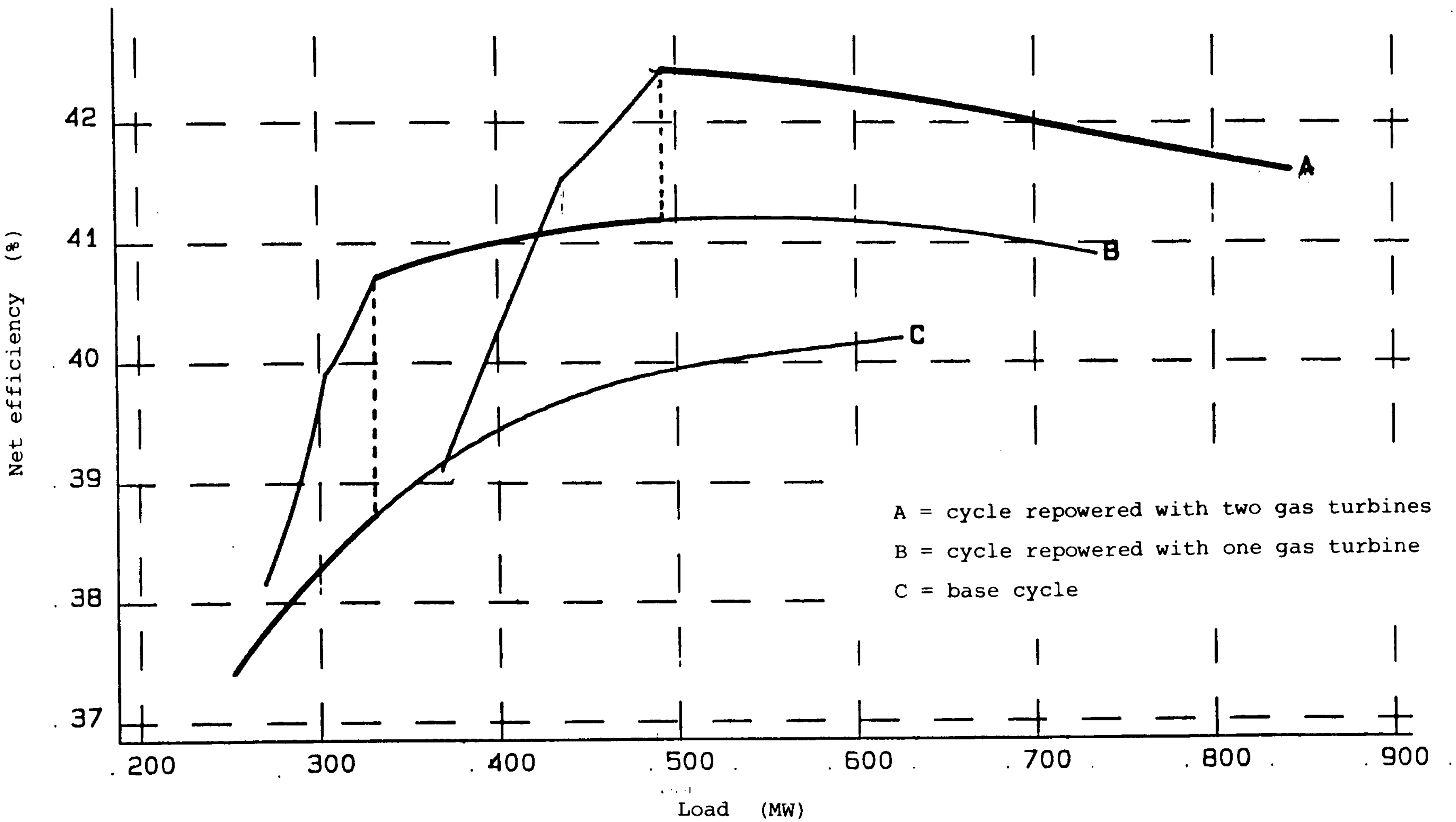


Fig. 4 - Comparison of efficiency curves of different repowered configurations with supplemental boiler of a 660 MW unit.

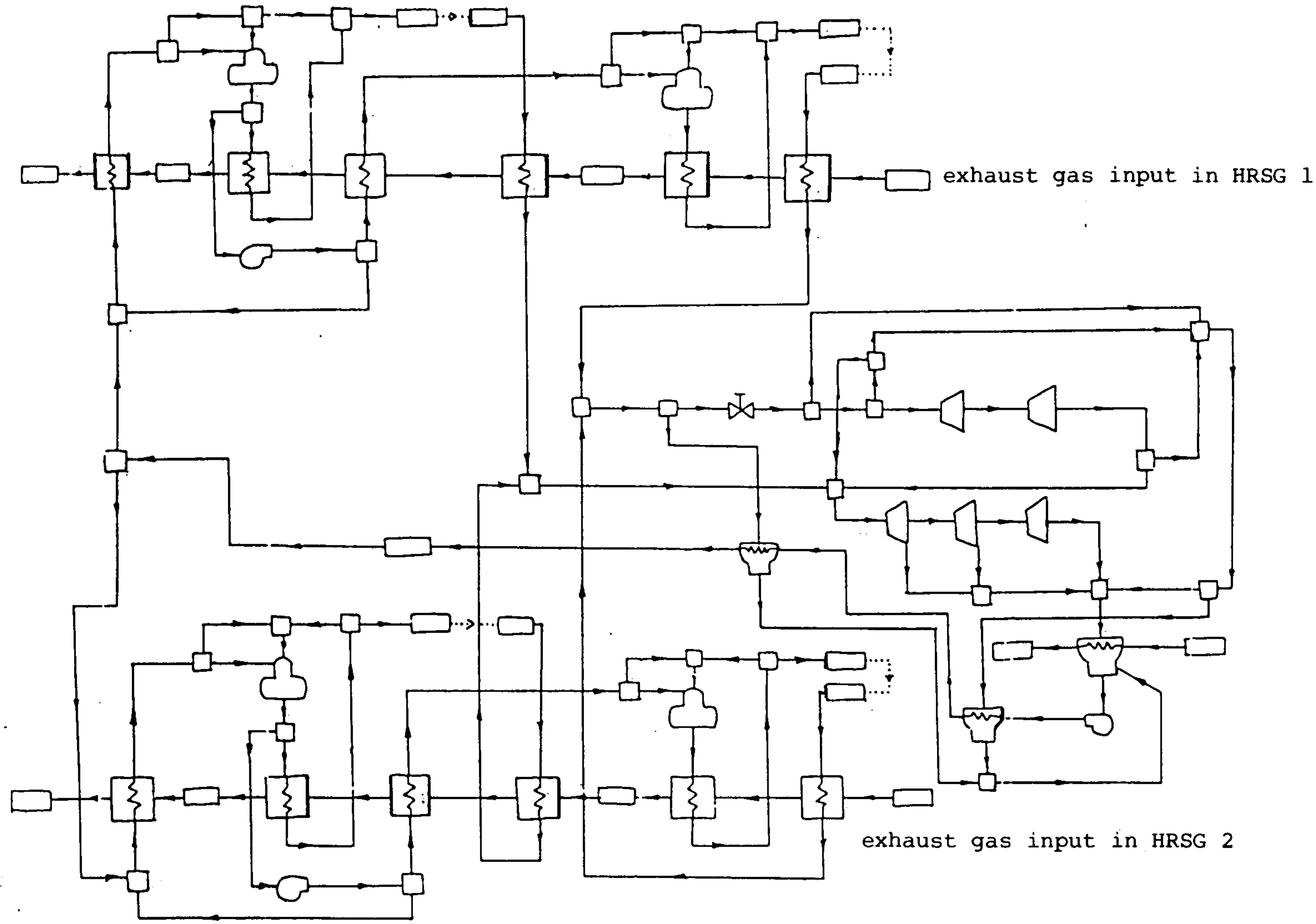


Fig. 5 - Schematic diagram of a full repowered cycle with two HRSGs.

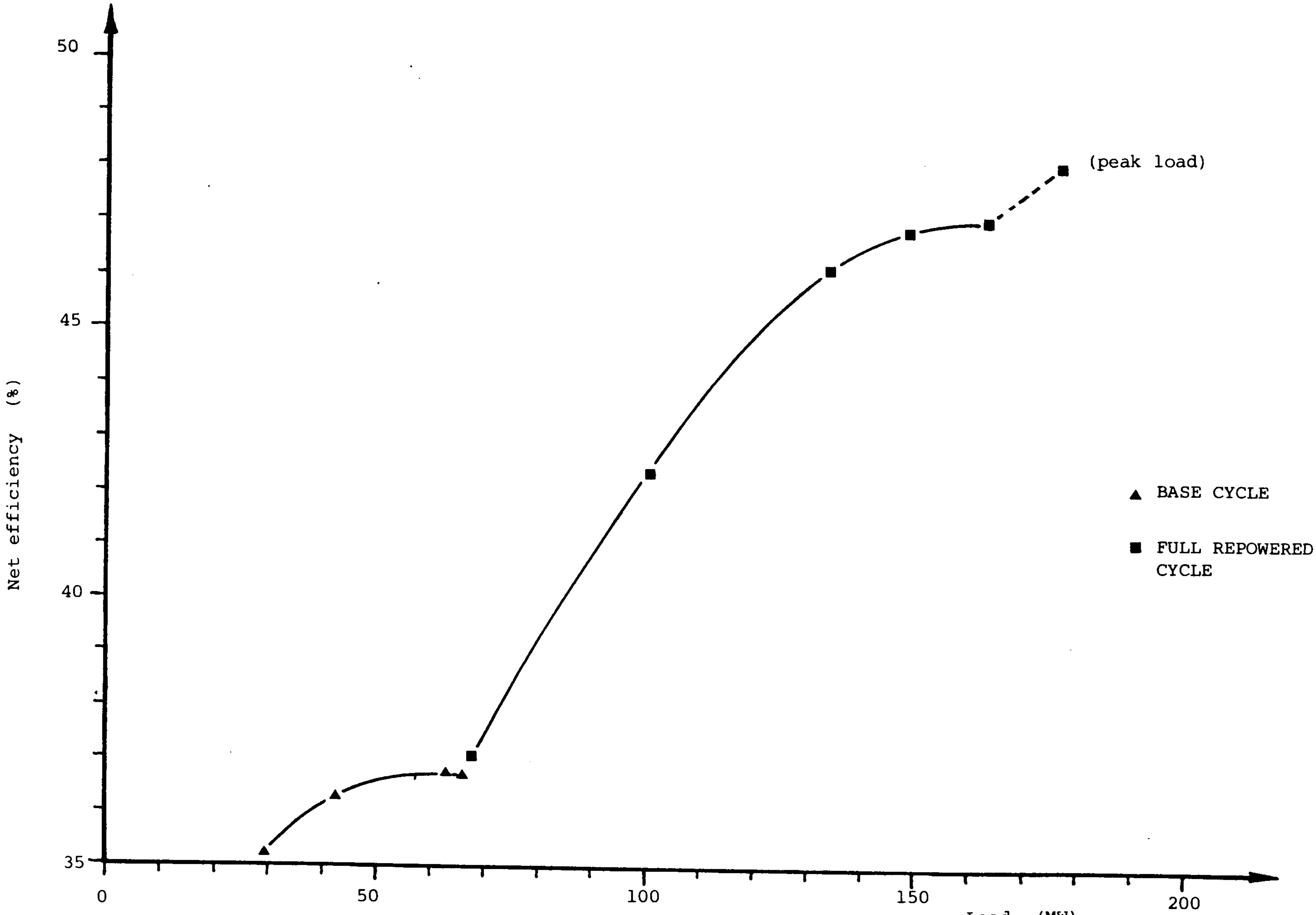


Fig. 6 - Curve of calculated net efficiency of a full repowered cycle

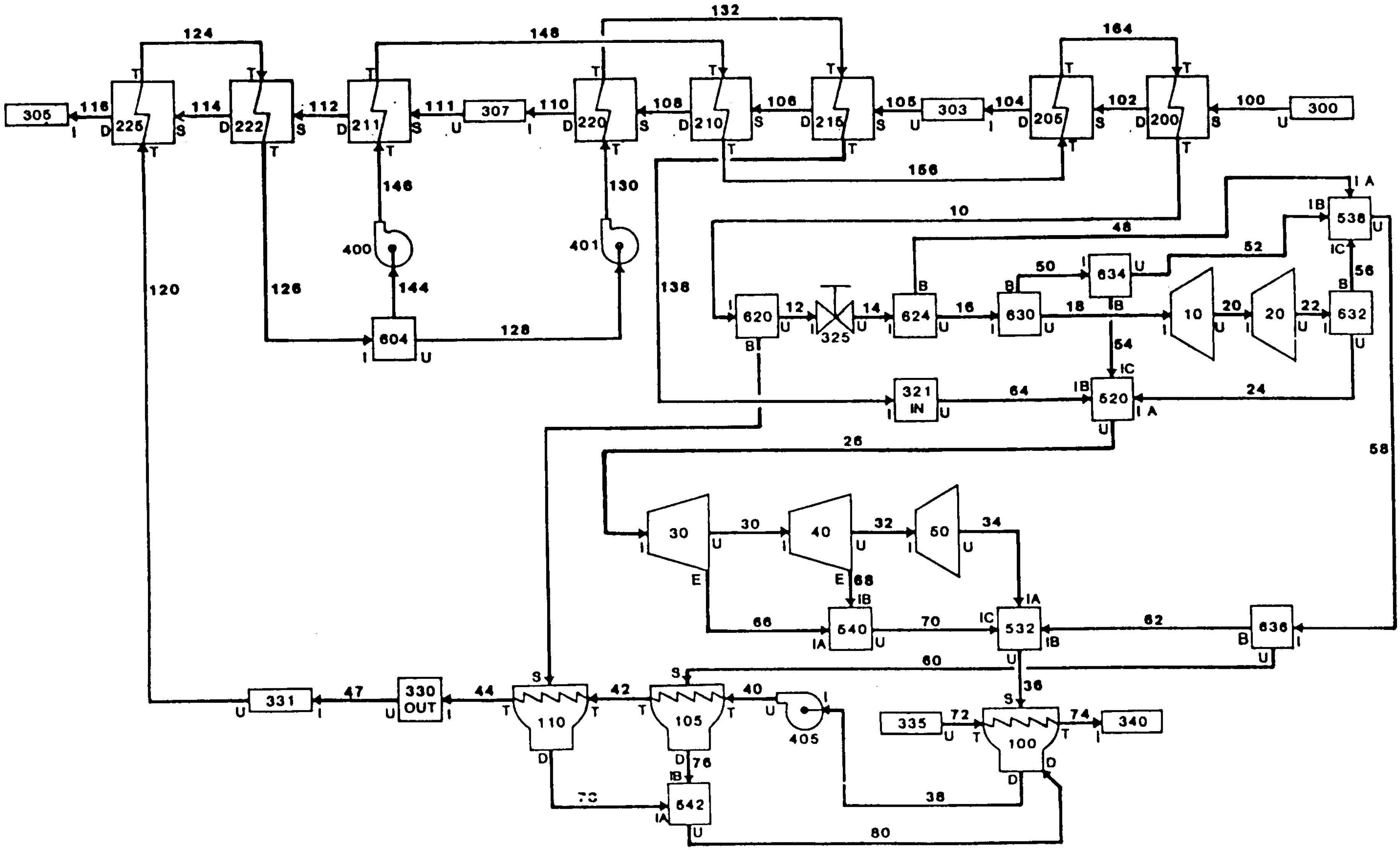


Fig. 7 - Schematic diagram of a full repowered cycle with three pressure levels in the HRSG.

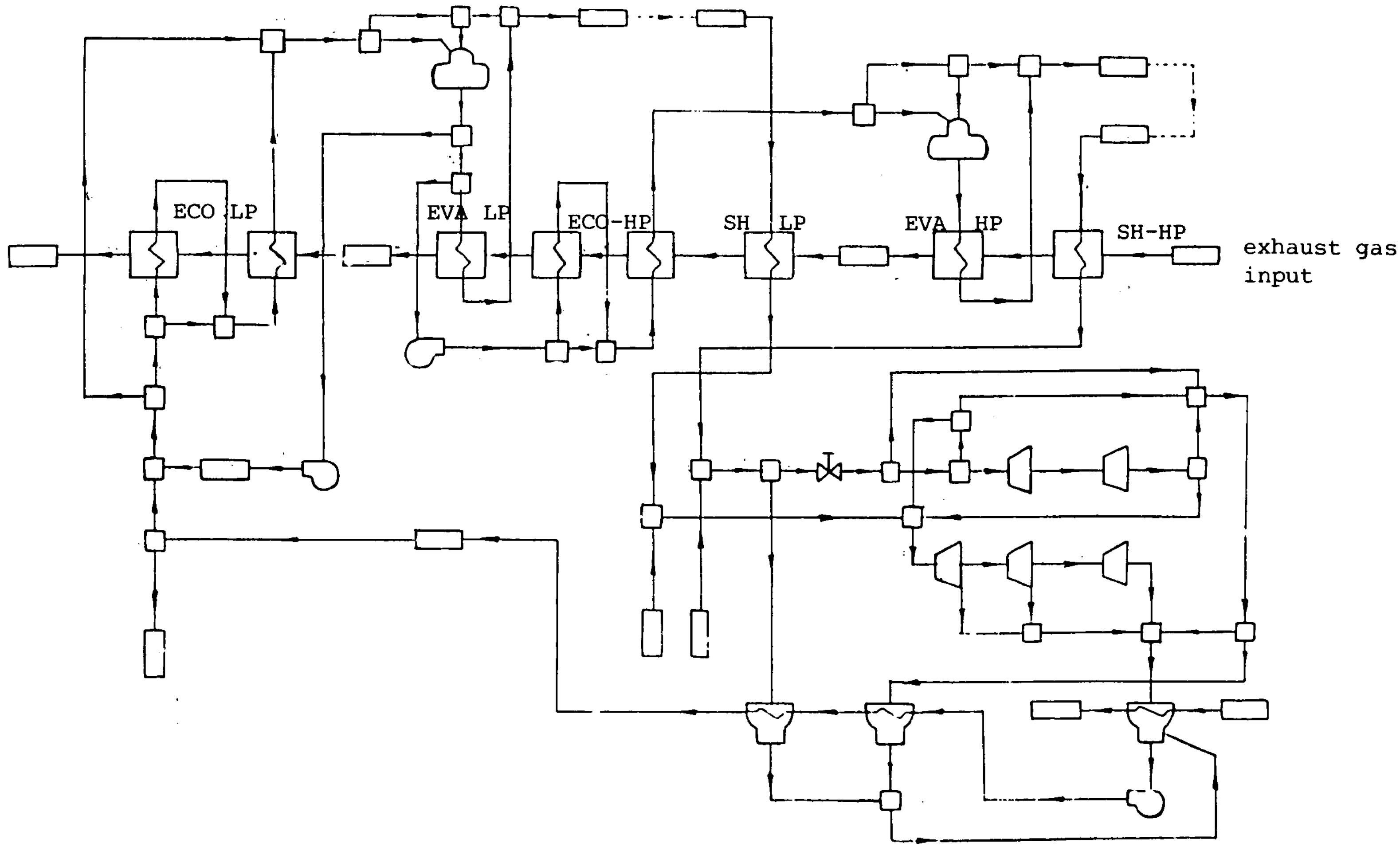


Fig. 8- Schematic diagram of a full repowered cycle (only one HRSG is represented).

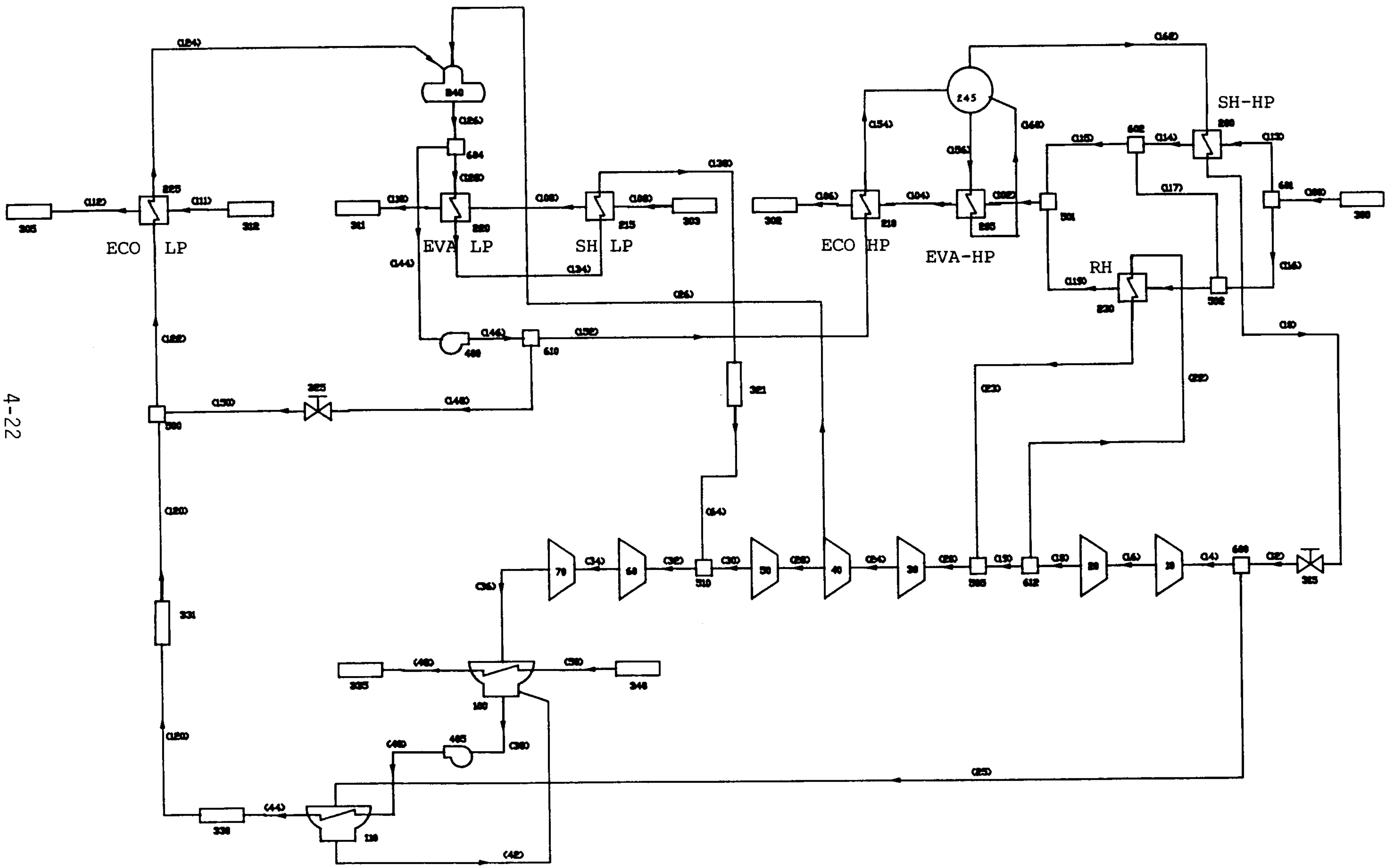
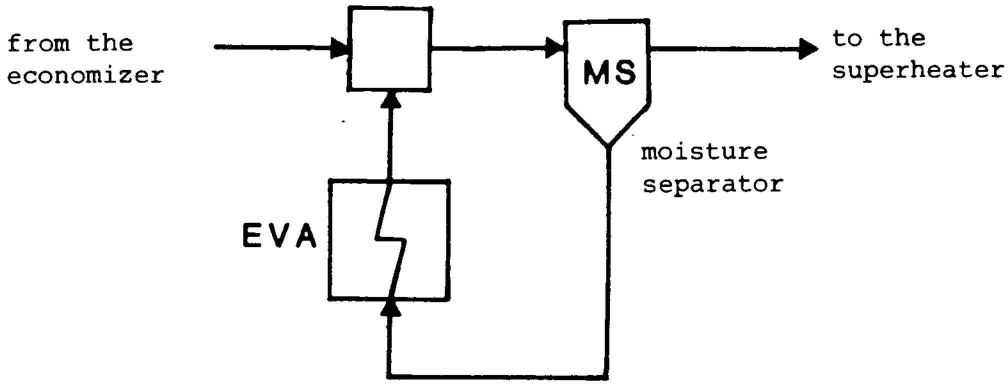
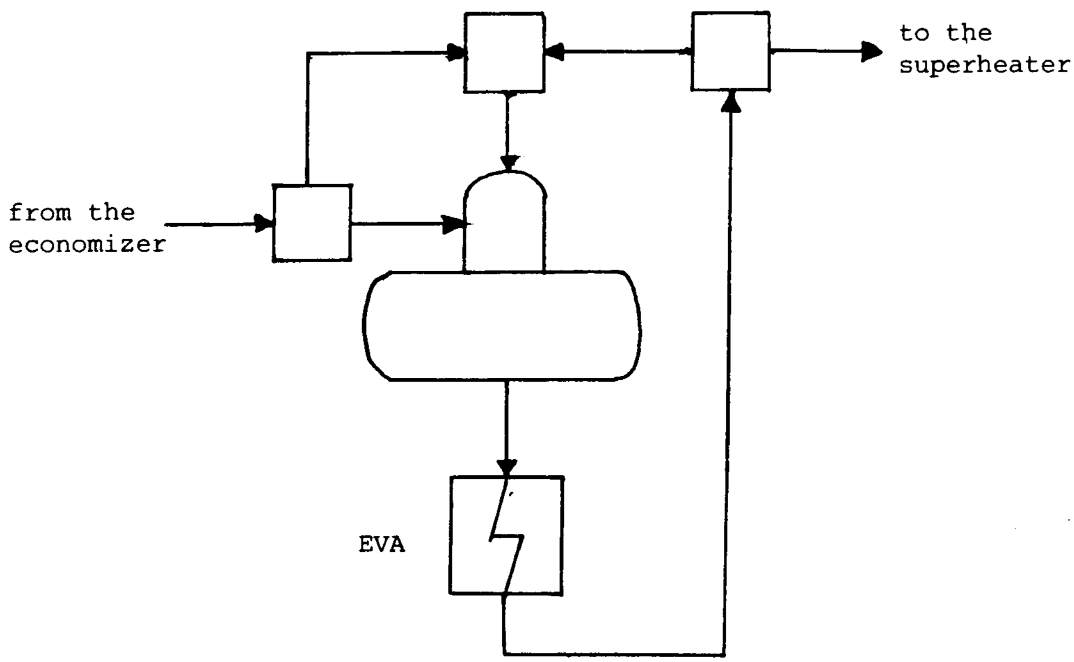


Fig. 9 - Schematic diagram of a full repowered cycle with a steam extraction from a turbine and a reheater.



A)



B)

Fig. 10 - Submodels used to simulate a boiler drum

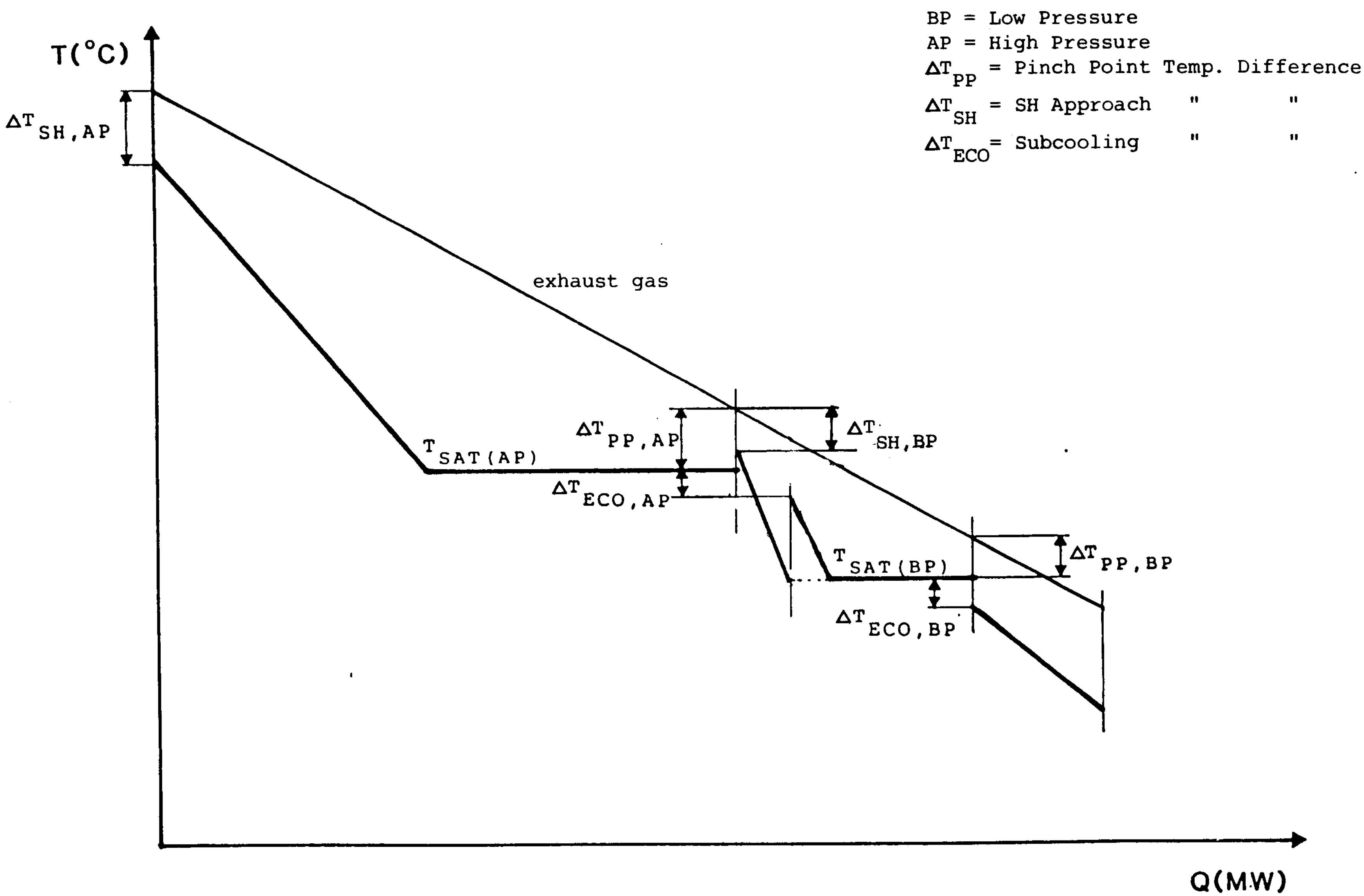


Fig. 11 - Temperatures through a six tube bundle HRSG.