# EVALUATION OF PERFORMANCES OF AN OIL-FIRED PLANT CONVERTED TO COMBINED CYCLE PLANT

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ABSTRACT

Repowering of fossil units by means of natural-gas fueled combustion turbines plays a major role in the emergency plan defined and actuated by ENEL (Italian National Electricity Board) to reduce the expected shortage of generating capacity in the next years.

CISE uses PEPSE to support ENEL (Italian National Electricity Board) in studying the performances and technical limits of repowering configurations, besides solving ordinary thermal balances for fossil power plants.

This paper describes the most important aspects of PEPSE code application to the pre-assessment study for conversion of existing 70 MWe oil-fired power plants, in service for several years, to 165 MWe combined cycle power plants.

This application was aimed to test the ability of PEPSE model to evaluate the operating characteristics of modified plants and, particularly, of dual pressure heat recovery steam generators (HRSG). This component was simulated by means of the techniques presented in the first paper.

Three different plant configurations were analized on the basis of both thermodynamic and operating criteria.

The HRSG was simulated in design mode using correlations for heat transfer with finned tubes. Therefore it was possible to extend the

study also to plant partial loads in order to define the operating load range and to evaluate the influence of the different operating procedures on plant performances.

## 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 supports ENEL in plant performance analysis for repowering of existing plants. To this aim PEPSE was initially used to evaluate different repowering solutions for fossil units ranging from 110 MW up to 660 MW.
Natural gas fueled combustion turbines were used to repower fossil units by means of an added low pressure HRSG. Due to the satisfactory results obtained, it was decided to extended PEPSE application to the preliminary study on the convenience and feasibility of converting existing conventional units, the rate of which is about 70 MW, to combined cycles by means of gas turbines.

In this paper the most important application aspects which were encountered in pre-assessment study are described.
The application activity is performed developing the following steps:

- evaluation of the different technical alternatives for retrofitting the existing units considering also, as one of the goals, the possibility of an extensive re-use of existing components;

- determination of the HRSG parameters which best match the existing plants (in particular the turbine);

- analysis of the operating conditions of the converted plants at part loads;

- analysis of the converted plant performances when gas turbines of roughly the same size but by different vendors are used.

A typical existing plant, considered as reference in this study, is simulated on the basis of ENEL balances at different loads (see Fig.1); the HRSG is simulated following the procedure presented in a previous paper[1].

## 2. COMPARISON BETWEEN DIFFERENT COMBINED CYCLE SCHEMES

The basic concept in the conversion of the existing plant to a combined cycle plant is the substitution of the fossil fired boiler with a Heat Recovery Steam Generator located on the exhaust gas stream from a gas turbine which, in this particular case, are sized 110 to 120 MW depending on the vendor.
Existing feedwater heater are not used in this scheme.
For deaeration of feedwater different alternatives are considered.
Use of a deaerating condenser is considered not convenient both for the implications about a complete modification of the existing one and for the decrease in deaerating performances at low loads.
Two possible alternatives would be:

a) use of the existing deaerator and associated turbine steam extraction with possible re-use of feedwater pump for the HP steam level (Fig. 2)
b) installation of a deaerating tower on the top of the low pressure boiler drum with a new feed pump (Fig. 3)

Both alternatives considered for the new arrangement are modeled by PEPSE (see Figs.4-5).

The following sequence is considered for the tube bundles inside the HRSG:

- HP Superheater in parallel with IP Reheater
- HP Evaporator
- HP Economizer
- LP Superheater
- LP Evaporator
- LP Economizer

The turbine is operated in sliding pressure on the basis of the vendor thermal balances for the original plant. It is assumed that its behaviour doesn't substantially change for the new steam conditions. The results have been later verified with the manufacturer.

At first the HRSG is simulated in performance mode assuming preliminary values[2],[3] for its thermal design parameters and setting the drum pressure to the value determined by steam turbine characteristics as illustrated in ref.(1).

In particular it is assumed:

$$\Delta T_{approach,SH,HP} = 20.°C \quad ( 36.°F )$$
$$\Delta T_{approach,RH} = 40.°C \quad ( 72.°F )$$
$$\Delta T_{pinch-point,HP} = 15.°C \quad ( 27.°F )$$
$$\Delta T_{approach,ECO,HP} = 20.°C \quad ( 36.°F )$$
$$\Delta T_{approach,SH,LP} = 10.°C \quad ( 18.°F )$$
$$\Delta T_{pinch-point,LP} = 25.°C \quad ( 45.°F )$$
$$\Delta T_{approach,ECO,LP} = 15.°C \quad ( 27.°F )$$

In Tables 1 and 2 the most interesting parameters of the plant are summarized respectively for both alternatives of re-use of existing deaerator and of a deaerating steam drum.

No significant difference exist, from a thermodynamic point of view, between the two alternatives.

For the more detailed analysis the only alternative of re-use of the existing deaerator is considered.

## 3. CHOICE OF THE HRSG PARAMETERS

The HRSG parameters are chosen on the basis of the following criteria:

- overload of the L.P. turbine not greater than 10% in flow;
- steam temperature at the cross-over close to the design value of the existing plant (214°C or 417.2°F);
- erosion at LP last stage buckets limited to the minimum;
- avoidance of steaming in the economizer tube bundles at part loads.

PEPSE code is then used to perform some thermal balances to evaluate the thermal parameters which best match the previous criteria.

### 3.1. $\Delta T$ approach SH - HP

The value of this parameter is chosen on the basis of a previous work[3] because, according to the previous criteria, it has a secondary importance. It is assumed:

$$\Delta T_{approach,SH,HP} = 20°C \quad (36°F)$$

### 3.2. $\Delta T$ approach RH

This parameter has an appreciable effect on the temperature at the discharge of I.P. turbine section. In order to limit the increase of steam temperature at the cross over with respect to the design value, it is taken:

$$\Delta T_{approach,RH} = 40°C \quad (72°F)$$

## 3.3. ΔT pinch point - HP

The value of $T_{pp,HP}$ greatly affects the value of the HP steam flow rate but also the required heat transfer effective area. The choice has been made on the basis of an economical optimization performed by means of a PC program. The following value is assumed:

$$\Delta T_{pinch\ point,HP} = 15°C \quad (27°F)$$

## 3.4. ΔT approach SH - LP

This parameter determines the LP inlet steam temperature. The best mixing condition of steam at the cross-over is obtained when superheating of steam from HRSG is high. By consequence the following value is taken:

$$\Delta T_{approach,SH\ LP} = 10°C \quad (18°F)$$

## 3.5. ΔT pinch point LP

The value of LP pinch point greatly affects the LP steam flow rate. In order to limit the LP turbine overload to 10% of the design value without reducing the HP and IP steam flow rate the following value is taken:

$$\Delta T_{pinch\ point\ LP} = 40°C \quad (72°F)$$

It should be pointed out that, from an economic point of view, lower values would be preferable.

In Tab. 3 plant parameters for the considered solution are summarized.

## 4. OPERATING CONDITIONS AT PART LOADS

The analysis at part loads is aimed to evaluate plant performances, and particularly efficiency, and to verify the compliance with technical constraints.
This analysis is performed using a design mode PEPSE model of the HRSG (Fig. 4) including finned tube calculations.
Techincal constraints are expressed using the following parameters:

- water subcooling at the inlet of the HP and LP steam drums,
- steam temperature at the outlet of HP/LP SH and IP RH,
- HP/LP steam mass flow rates,
- governing stage (GS) shell temperature,
- moisture in the LP turbine section,
- exhaust gas temperature at the stack inlet,

### 4.1. Plant operation

Part loads are obtained by reducing gas turbine load and operating the IGV so that the temperature of the flue gas is constant in the range 70% - 100% of the full load (see Fig. 6). In such a way thermal cycling of the steam turbine is minimized.

### 4.2. Water temperature at the inlet of LP ECO

To avoid cold-end corrosion on the LP ECO tube bundles, the water temperature at the inlet of the economizer is maintained to a value of 50°C (122°F). This is obtained by recirculating the required flow rate of water from the LP steam drum to the ECO inlet.
Control of the water temperature at the ECO inlet provides a further advantage, that is increase of the water subcooling at the LP ECO outlet.

### 4.3. HP and LP pressure control

Pressure at steam turbine inlet, and therefore in the HP steam drum, is not controlled at a constant value but varies with the steam flow rate as dictated by the gas turbine.
For the preliminary thermodynamic analysis with PEPSE, the following control scheme is considered, while a review of the associated operational and control aspects is in process in order to verify its applicability.
The deaerator pressure is driven by the pressure at the steam extraction in the IP turbine section; the LP steam drum is kept at constant pressure by the regulating valve on the steam line. The LP drum level is controlled through a regulating valve and a circulating pump on the feedwater line.
It should be pointed out that the changes in pressure have a sensible effect on the fluid velocity in the HRSG and, in particular, in the steam drum moisture separators.

## 4.4. Analysis of some performance and operational parameters

In Figs. 7 to 13 the trends of the performance and operational parameters considered during the study are drawn. In particular, Fig. 7 shows the net electric power generated by the whole plant and by the steam turbine alone, while Fig. 8 shows plant net efficiency versus gas turbine load.

Figure 9 has a considerable importance in evaluating the minimum plant load because it shows the water subcooling at the outlet of economizer tube bundles. At about 50% of full load, steaming in the HP economizer may occur. Therefore, for lower loads, economizer protection devices are required.

Governing stage shell temperature, which is another important parameter, is maintained as constant as possible (Fig. 10) thus reducing the thermal stresses of turbine rotor.

PEPSE calculations show values of moisture in the last LP turbine stage well below the maximum allowable values suggested by vendors. Consequently it is defined that no blade erosion problems would arise by the modification of the original plant to a combined cycle plant.

## 4.5. Ambience air temperature

This parameter heavily affects gas turbine and, therefore, plant performances.

Assuming a location in Southern Italy, two ambience conditions are considered. The first, the reference condition, is based on a value $T_{amb}$=15.°C (59.°F). The second is based on $T_{amb}$=40.°C (104.°F). Gas turbine exhaust characteristics at different ambience temperatures are known from the vendor.

The condition with $T_{amb}$=-10.°C (14.°F), usually considered for other plant sites in Northern Italy, is neglected here.

A comparison between plant performances for both ambience conditions is given in Tab. 4.

## 5. OPERATION WITH DIFFERENT VENDOR GAS TURBINES

The analyses previously described are based on the characteristics of a 110 MW gas turbine MS 9001 E by Nuovo Pignone. Further analyses are performed to evaluate plant performances when a 122 MW gas turbine 50 D5 by FIAT is used keeping the characteristic of the other equipment (HRSG and steam turbine) constant.

In Tab. 5 plant performances at full load are given for both gas turbines.

The performances of the steam section (boiler and turbine) are substantially the same for both gas turbines. In particular the thermal power released to condenser changes only of 3%.

Also the plant performances don't change in a considerable way. In fact the net efficiency of the plant is the same in both cases (47.%) while the net electric power generated varies of about 7%.

## 6. SUMMARY

PEPSE is used to perform a pre-assessment and feasibility analysis of conversion of small sized fossil fired plants to combined cycles.

A peculiar aspect of this study is the use of a PEPSE model for finned tube HRSG both in performance mode and in design mode.

Choice of the best thermodynamic design parameters of the HRSG is performed by means of a performance mode model.

Design mode model allows the evaluation of the behaviour of converted plants at part loads.

It should be pointed out that the analysis is preliminary, not considering operating / control procedures and detailed economic aspects. Nevertheless it is very interesting to compare the efficiencies of the original plant and of the plant after the conversion (Fig. 14).

## 7. FURTHER DEVELOPMENTS

This preliminary study has shown the usefulness of PEPSE in the analysis of conversion of existing plants.
On the basis of the work done, the set-up of a plant model considering the operating problems in a more detailed way is now in progress. In particular, condenser cooling (tower or sea water), pressure drops in streams and control procedures will be taken into account.

## REFERENCES


1) R.Colombo,M.de Carli,G.Aquilanti: "PEPSE modeling of finned tube heat exchangers for combined cycle plants", 1989 PEPSE Users'Meeting.

2) M.Mirone,E.Morgani,M.Pisanti,R.Senis,R.Colombo: "ENEL activities for the project of a 300 MW combined cycle plant" (in Italian), 1989 ATI Meeting on Combined Cycles and Cogeneration Perspectives, Bologna (I), May 1989

3) H.Hagen,M.J.J.Linnemeijer: "Combined cycle operating experience of a converted steam power plant", 1988 ASME COGEN-TURBO, Montreux (CH), August 1988

Tab. 1 - Plant parameters for the solution with the existing deaerator

| HRSG | | | STEAM TURBINE | | |
|---|---|---|---|---|---|
| $P_{HP}$ | (kPa) | 7284.4 | $P_{IN,HP}$ | (kPa) | 6919.6 |
| $P_{LP}$ | (kPa) | 430.5 | $P_{IN,LP}$ | (kPa) | 248.1 |
| $\Delta T_{SH\ HP}$ | (°C) | 20.0 | $T_{IN,HP}$ | (°C) | 506.0 |
| $\Delta T_{SH\ LP}$ | (°C) | 10.0 | $T_{IN,IP}$ | (°C) | 483.8 |
| $\Delta T_{PP\ HP}$ | (°C) | 15.0 | $T_{OUT,IP}$ | (°C) | 236.6 |
| $\Delta T_{app,ECO,HP}$ | (oC) | 25.0 | $T_{IN,LP}$ | (°C) | 237.4 |
| $\Delta T_{PP\ LP}$ | (°C) | 20.0 | $T_{GS}$ | (°C) | 477.5 |
| $\Delta T_{app,ECO,LP}$ | (°C) | 15.0 | $\dot{M}_{IN,HP}$ | (kg/s) | 40.91 |
| $\Delta T_{RM,IP}$ | (°C) | 40.0 | $\dot{M}_{IN,LP}$ | (kg/s) | 54.67 |
| $\dot{M}_{STEAM,HP}$ | (kg/s) | 41.030 | Overload LP turbine | (%) | 17.26 |
| $\dot{M}_{STEAM,LP}$ | (kg/s) | 15.19 | | | |
| $P_{DEA}$ | (kPa) | 430.51 | $W_{therm,COND}$ | (MW) | 124.87 |

$W_{el,NGT}$ =108.95 MW
$W_{el,N,ST}$ = 55.74 MW
$W_{el,N,TOT}$ = 164.69 MW
$\eta_{N,TOT}$ = 47.29 %

Tab. 2 - Plant parameters for the solution with deaerating L.P. steam drum

| HRSG | | | STEAM TURBINE | | |
|---|---|---|---|---|---|
| $P_{HP}$ | (kPa) | 7294.2 | $P_{IN,HP}$ | (kPa) | 6929.4 |
| $P_{LP}$ | (kPa) | 280.5 | $P_{IN,LP}$ | (kPa) | 254.0 |
| $\Delta T_{SH\ HP}$ | (°C) | 20.0 | $T_{IN,HP}$ | (°C) | 506.0 |
| $\Delta T_{SH\ LP}$ | (°C) | 10.0 | $T_{IN,IP}$ | (°C) | 483.8 |
| $\Delta T_{PP,HP}$ | (°C) | 15.0 | $T_{OUT,IP}$ | (°C) | 238.9 |
| $\Delta T_{PP,LP}$ | (°C) | 25.0 | $T_{IN,LP}$ | (°C) | 238.2 |
| $\Delta T_{app,ECO,HP}$ | (°C) | 20.0 | $T_{GS}$ | (°C) | 477.3 |
| $\Delta T_{app,ECO,LP}$ | (°C) | 15.0 | $\dot{M}_{IN,HP}$ | (kg/s) | 40.89 |
| $\Delta T_{RM,IP}$ | (°C) | 40.0 | $\dot{M}_{IN,LP}$ | (kg/s) | 55.84 |
| $\dot{M}_{STEAM,HP}$ | (kg/s) | 41.02 | Overload LP turbine | (%) | 19.8 |
| $\dot{M}_{STEAM,LP}$ | (kg/s) | 14.94 | | | |
| $P_{DEA}$ | (kPa) | 280.5 | $W_{therm,COND}$ | (MW) | 127.67 |

$W_{el,NGT}$ = 108.95 MW

$W_{el,N,ST}$ = 56.44 MW

$W_{el,N,TOT}$ = 165.39 MW

$\eta_{N,TOT}$ = 47.49 %

Tab. 3 - Plant parameters for the analized solution

| HRSG | | STEAM TURBINE | |
|---|---|---|---|
| $P_{HP}$ (kPa) | 7276.5 | $P_{IN,HP}$ (kPa) | 6913.7 |
| $P_{LP}$ (kPa) | 421.7 | $P_{IN,LP}$ (kPa) | 231.4 |
| $\Delta T_{SH,HP}$ (°C) | 20.0 | $T_{IN,HP}$ (°C) | 506.0 |
| $\Delta T_{SH,LP}$ (°C) | 10.0 | $T_{IN,IP}$ (°C) | 484.0 |
| $\Delta T_{PP,HP}$ (°C) | 15.0 | $T_{OUT,IP}$ (°C) | 229.1 |
| $\Delta T_{PP,LP}$ (°C) | 40.0 | $T_{IN,LP}$ (°C) | 231.5 |
| $\Delta T_{app,ECO,HP}$ (°C) | 20.0 | $T_{GS}$ (°C) | 477.7 |
| $\Delta T_{app,ECO,LP}$ (°C) | 25.0 | $\dot{M}_{IN,HP}$ (kg/s) | 40.92 |
| $\Delta T_{RH,IP}$ (°C) | 40.0 | $\dot{M}_{IN,LP}$ (kg/s) | 51.17 |
| $\dot{M}_{STEAM,HP}$ (kg/s) | 41.04 | Overload LP turbine (%) | 9.8 |
| $\dot{M}_{STEAM,LP}$ (kg/s) | 12.48 | | |
| $P_{DEA}$ (kPa) | 421.7 | $W_{therm,COND}$ (MW) | 116.52 |

$W_{el,N,GT}$ = 108.95 MW

$W_{el,N,ST}$ = 54.55 MW

$W_{el,N,TOT}$ = 163.50 MW

$\eta_{N,TOT}$ = 46.95 %

Tab. 4 - Plant performances at different ambience temperatures

| | | | |
|---|---|---|---|
| Ambience temperature | (°C) | + 15° | + 40 |
| Condenser pressure | (kPa) | 4.903 | 4.903 |
| **Gas turbine** | | | |
| Net electric power | (MW) | 108.95 | 90.65 |
| Thermal power | (MW) | 348.23 | 303.50 |
| Flue gas mass flow rate | (kg/s) | 405.0 | 364.44 |
| Flue gas temperature | (°C) | 529 | 546 |
| **HRSG** | | | |
| HP Steam | | | |
| Mass flow rate | (kg/s) | 41.39 | 39.58 |
| Pressure | (kPa) | 7276.5 | 7140.8 |
| Temperature | (°C) | 509 | 524 |
| LP Steam | | | |
| Mass flow rate | (kg/s) | 12.28 | 10.98 |
| Pressure | (kPa) | 421.7 | 405.2 |
| Temperature | (°C) | 245 | 241 |
| Fluegas stack temp. | (°C) | 143.5 | 139.3 |
| **Steam turbine** | | | |
| G.S. temperature | (°C) | 506 | 521 |
| Gross electric power | (MW) | 56.999 | 54.941 |
| Thermal power at condenser | (MW) | 115.95 | 109.49 |
| **Plant** | | | |
| Natural gas flow rate | ($Nm^3/h$) | 36.300 | 31.637 |
| Net electric power | (MW) | 163.66 | 143.38 |
| Net efficiency | (%) | 47.00 | 47.24 |

Tab. 5 - Plant performances with gas turbines of different vendors

| | | MS9001E | 50D5S |
|---|---|---|---|
| Ambience temperature | (°C) | + 15 | + 15 |
| Condenser pressure | (kPa) | 4.903 | 4.903 |
| Gas turbine | | | |
| Net electric power | (MW) | 108.95 | 121.72 |
| Thermal power | (MW) | 348.23 | 375.09 |
| Flue gas mass flow rate | (kg/s) | 405.0 | 448. |
| Flue gas temperature | (°C) | 529 | 504 |
| HRSG | | | |
| HP Steam | | | |
| Mass flow rate | (kg/s) | 41.39 | 41.61 |
| Pressure | (kPa) | 7276.5 | 7286.9 |
| Temperature | (°C) | 509 | 487.4 |
| LP Steam | | | |
| Mass flow rate | (kg/s) | 12.28 | 13.75 |
| Pressure | (kPa) | 421.7 | 419.4 |
| Temperature | (°C) | 245 | 248 |
| Fluegas stack temp. | (°C) | 143.5 | 147.3 |
| Steam turbine | | | |
| G.S. temperature | (°C) | 506 | 484 |
| Gross electric power | (MW) | 56.999 | 54.992 |
| Thermal power at condenser | (MW) | 115.95 | 119.59 |
| Plant | | | |
| Natural gas flow rate | ($Nm^3/h$) | 36.300 | 39.100 |
| Net electric power | (MW) | 163.66 | 176.423 |
| Net efficiency | (%) | 47.00 | 47.03 |

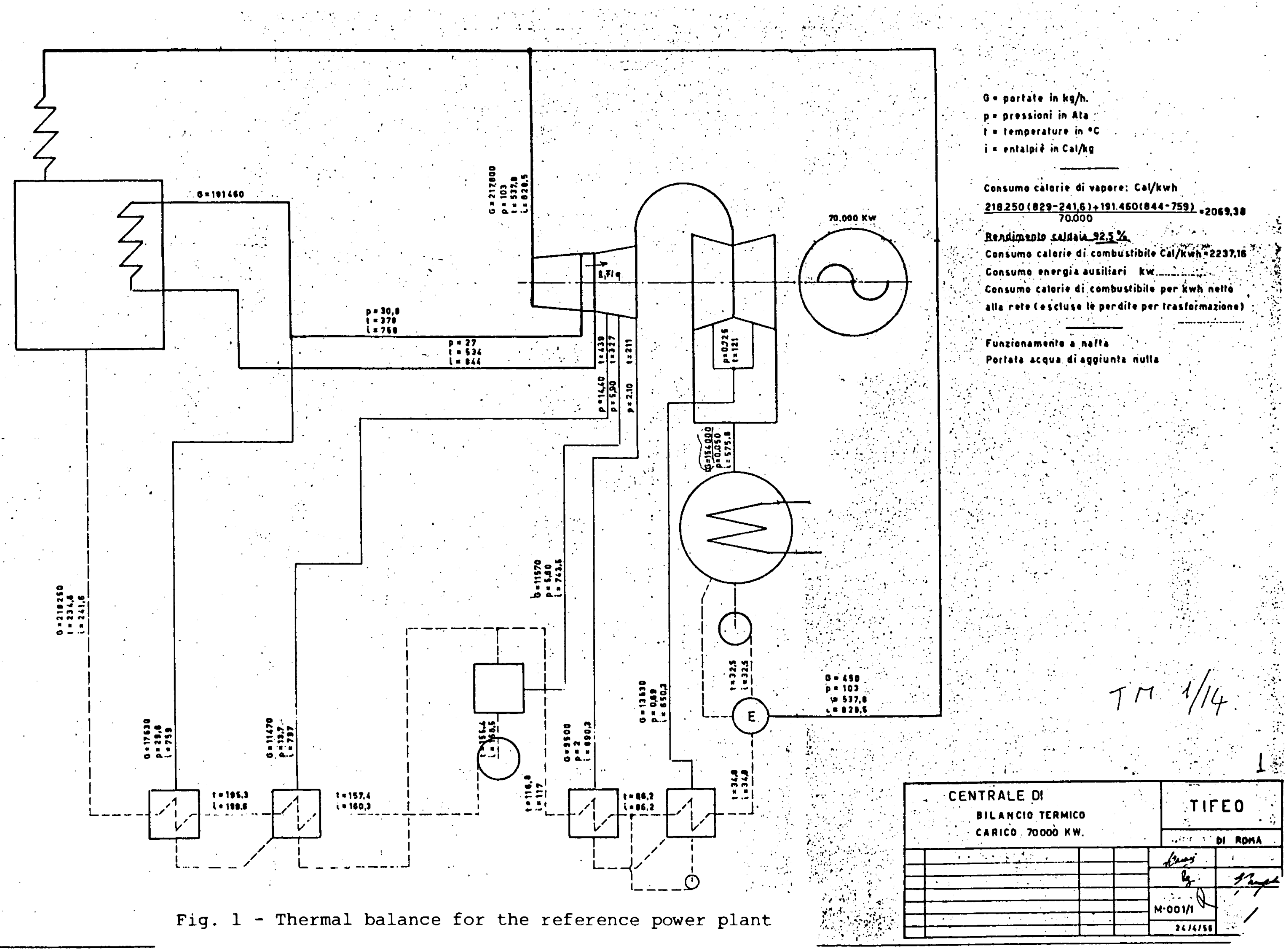


Fig. 1 - Thermal balance for the reference power plant

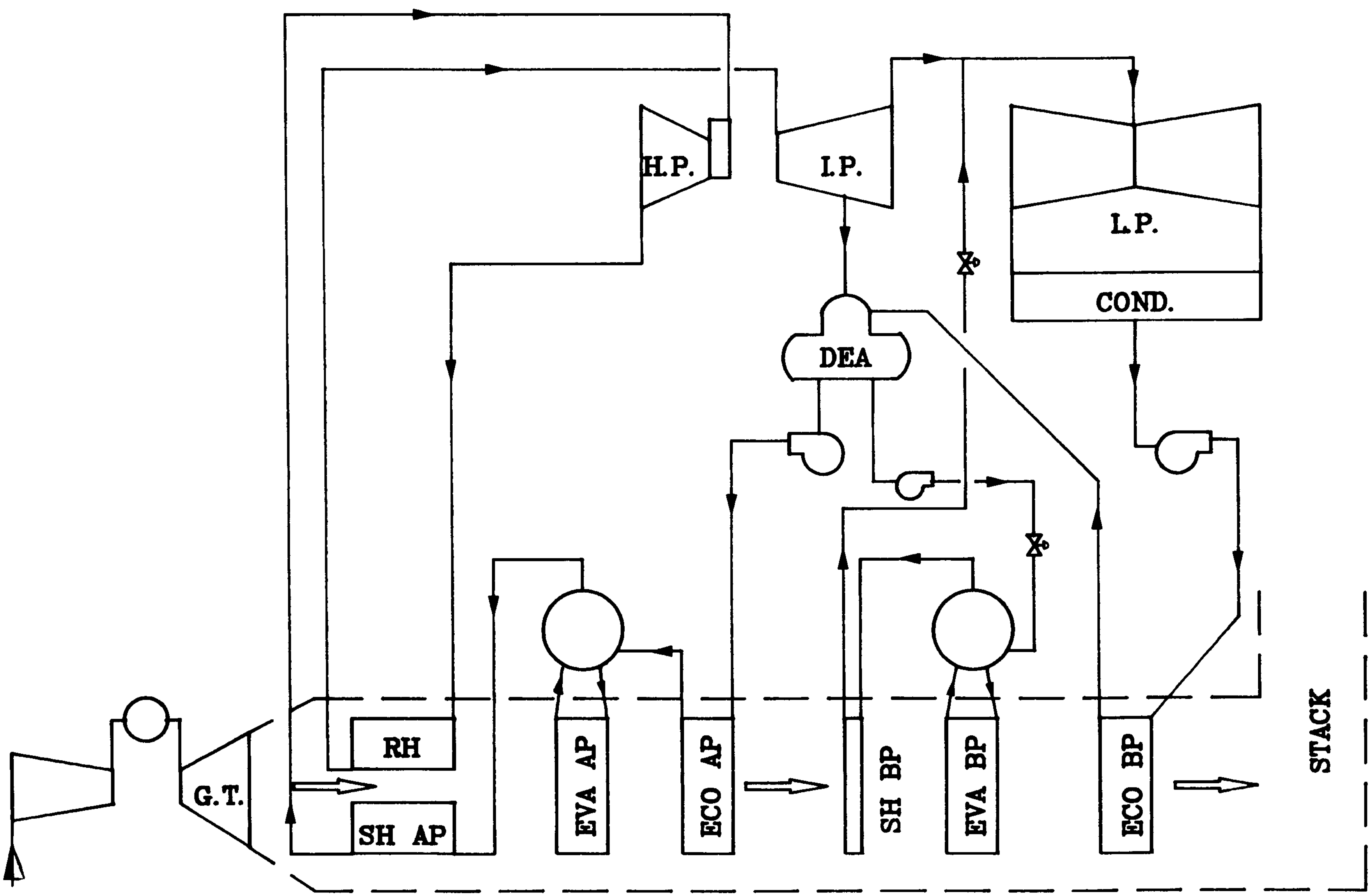


Fig. 2 - Solution maintaining the existing deaerator

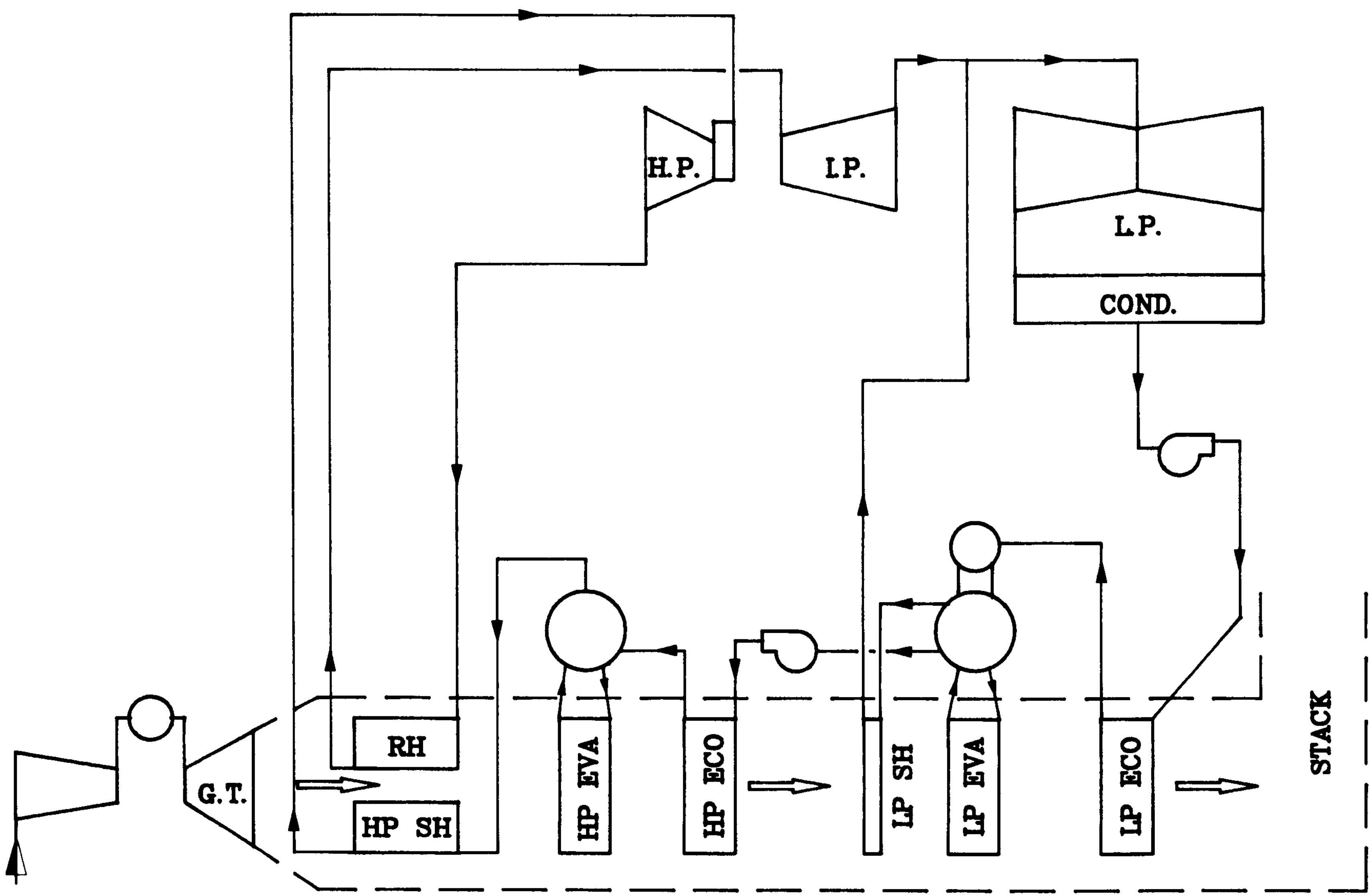


Fig. 3 - Solution with a new deaerating LP steam drum

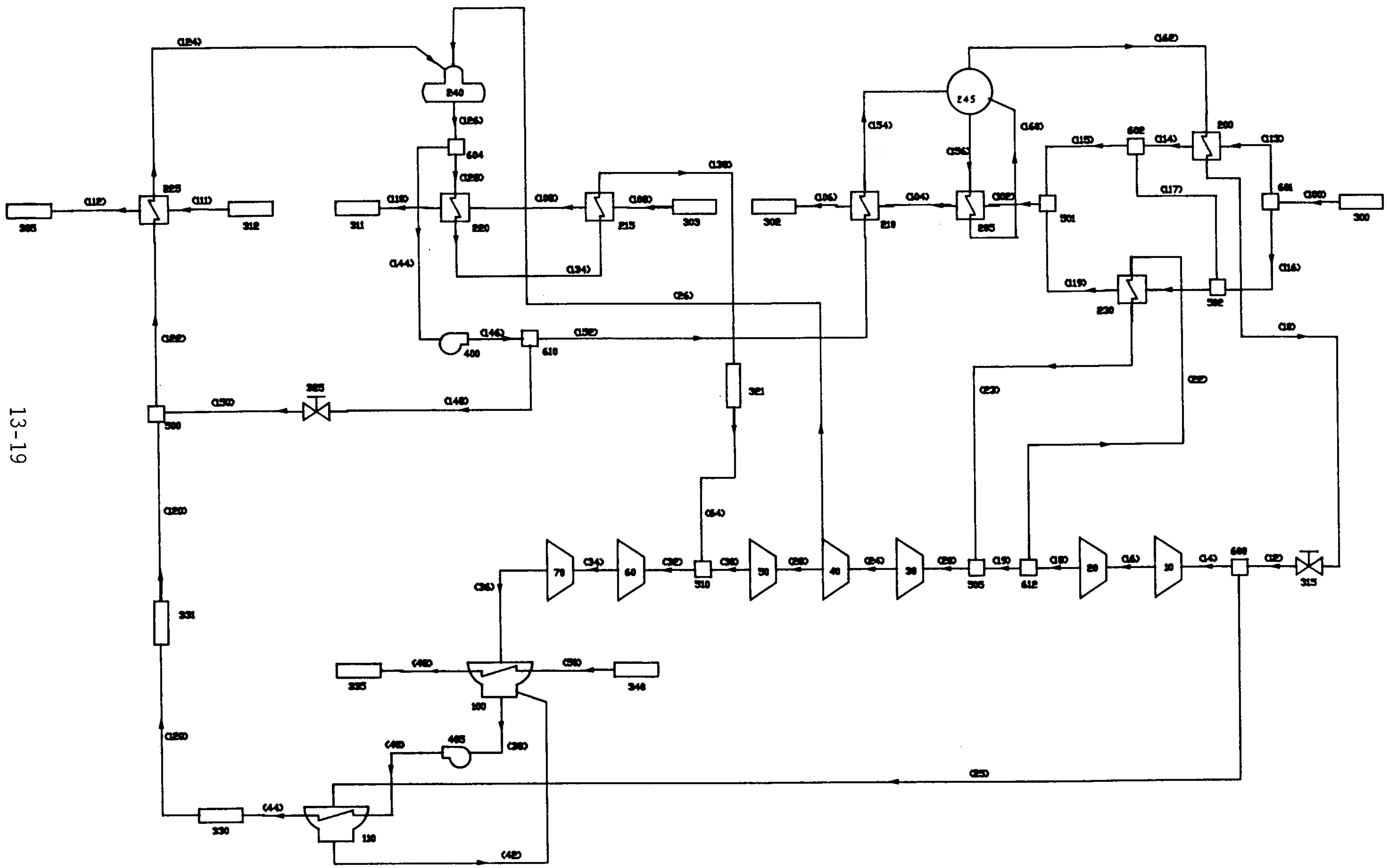

Fig. 4 - PEPSE model for the solution in Fig. 2

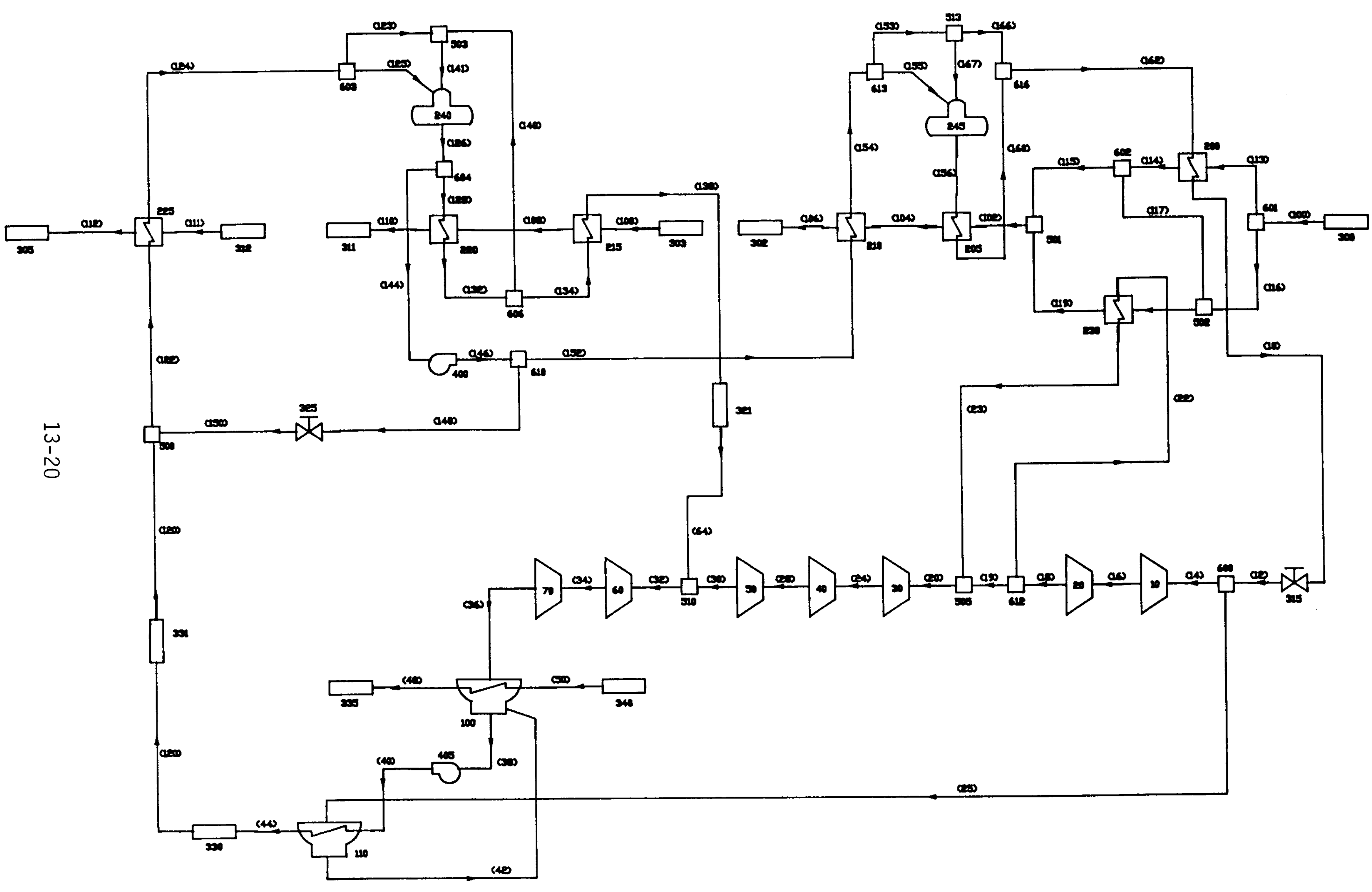

Fig. 5 - PEPSE model for the solution in Fig. 3

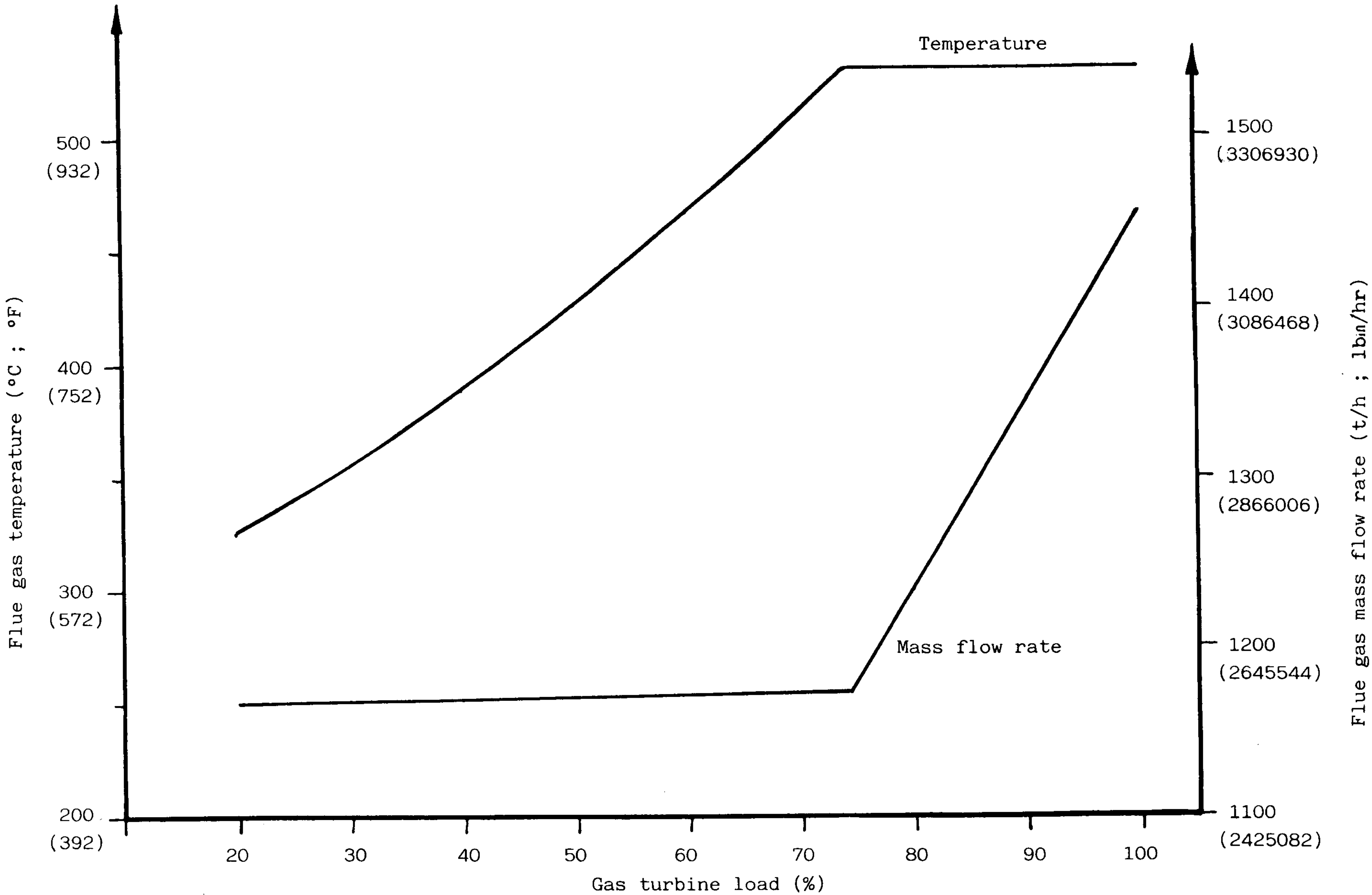


Fig. 6 - Gas turbine exahust flow rate and temperature vs. load

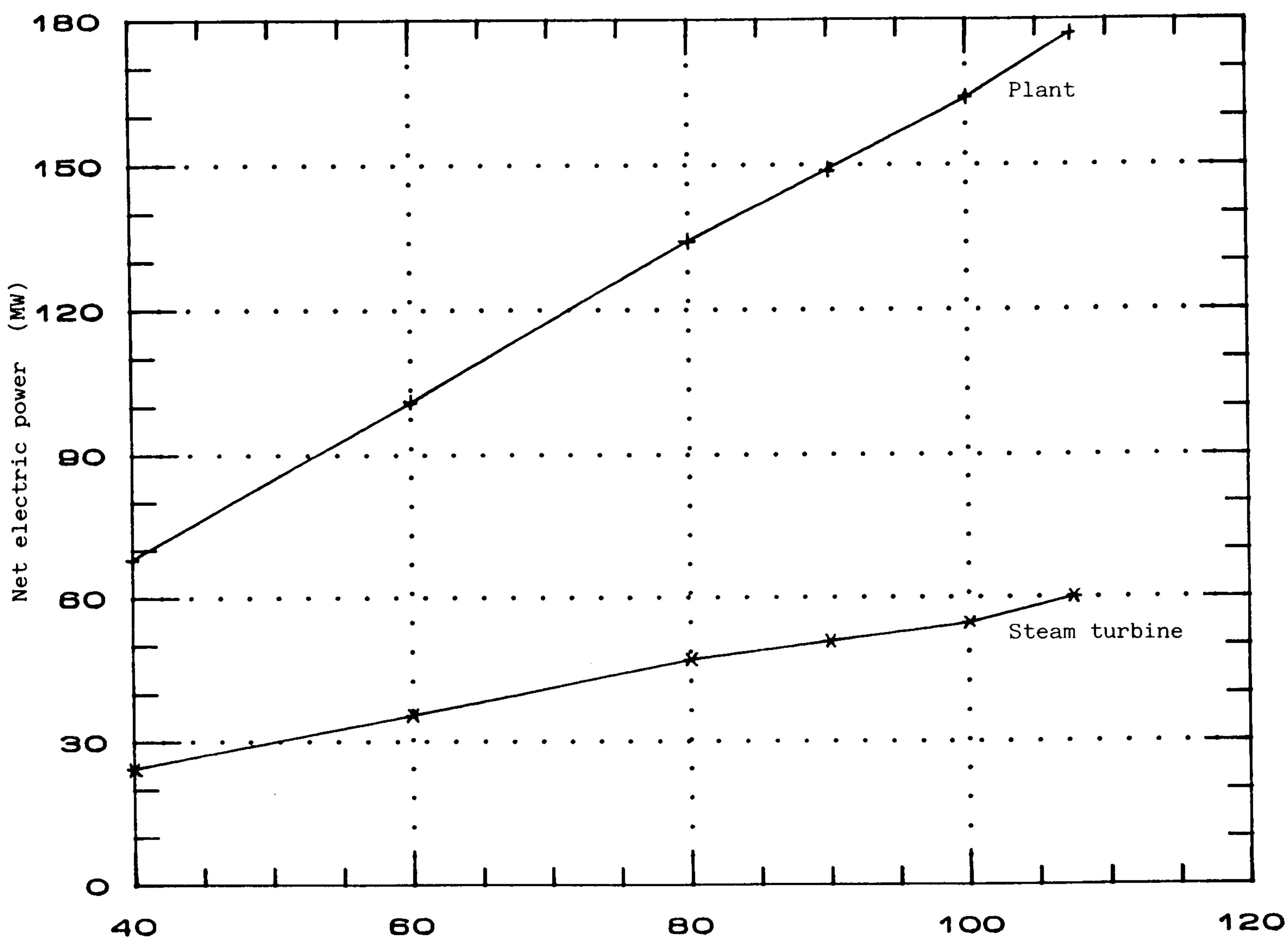


Fig. 7 - Gas turbine load (%)

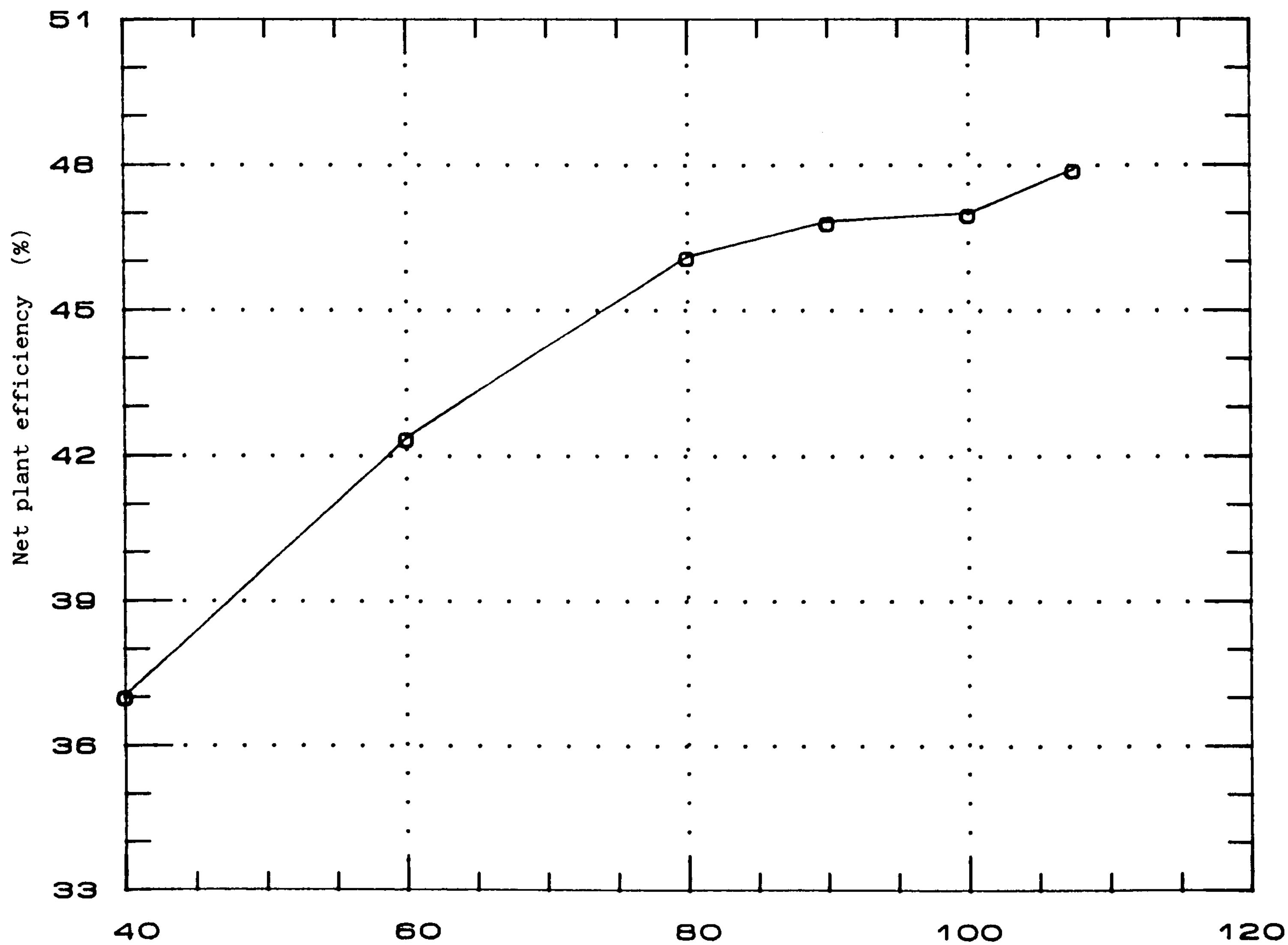


Fig. 8 - Gas turbine load (%)

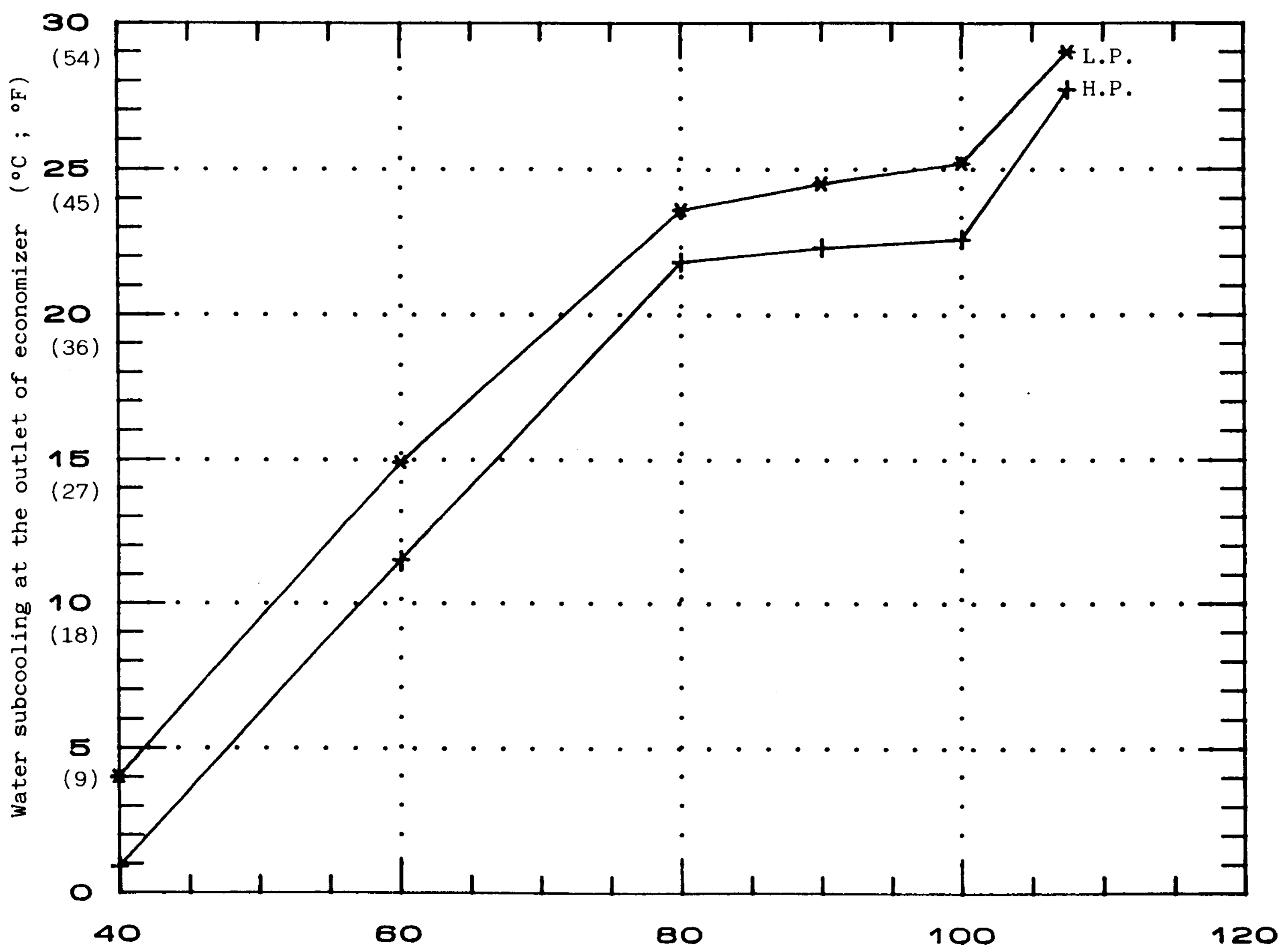


Fig. 9 - Gas turbine load (%)

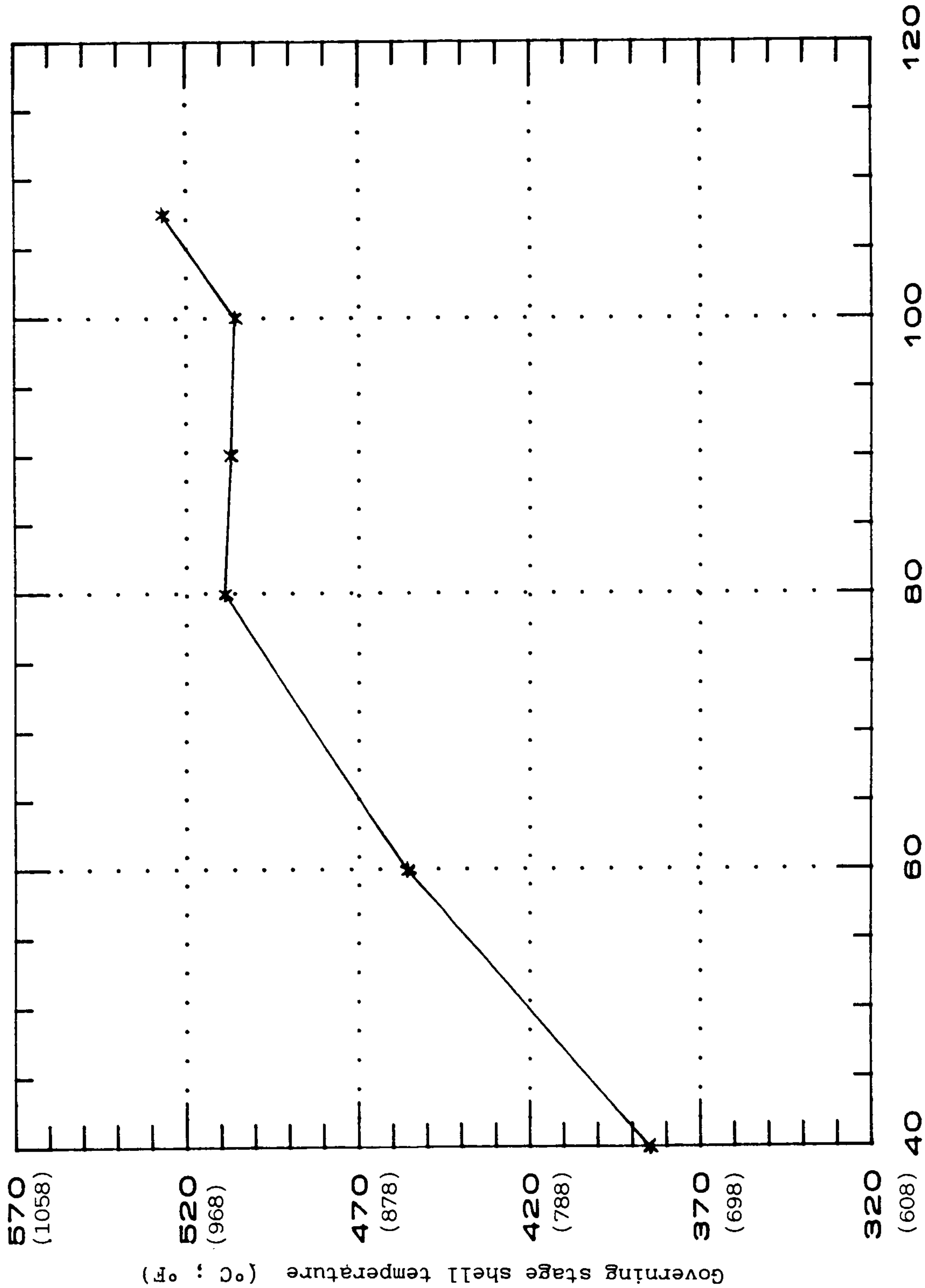


Fig. 10 - Gas turbine load (%)

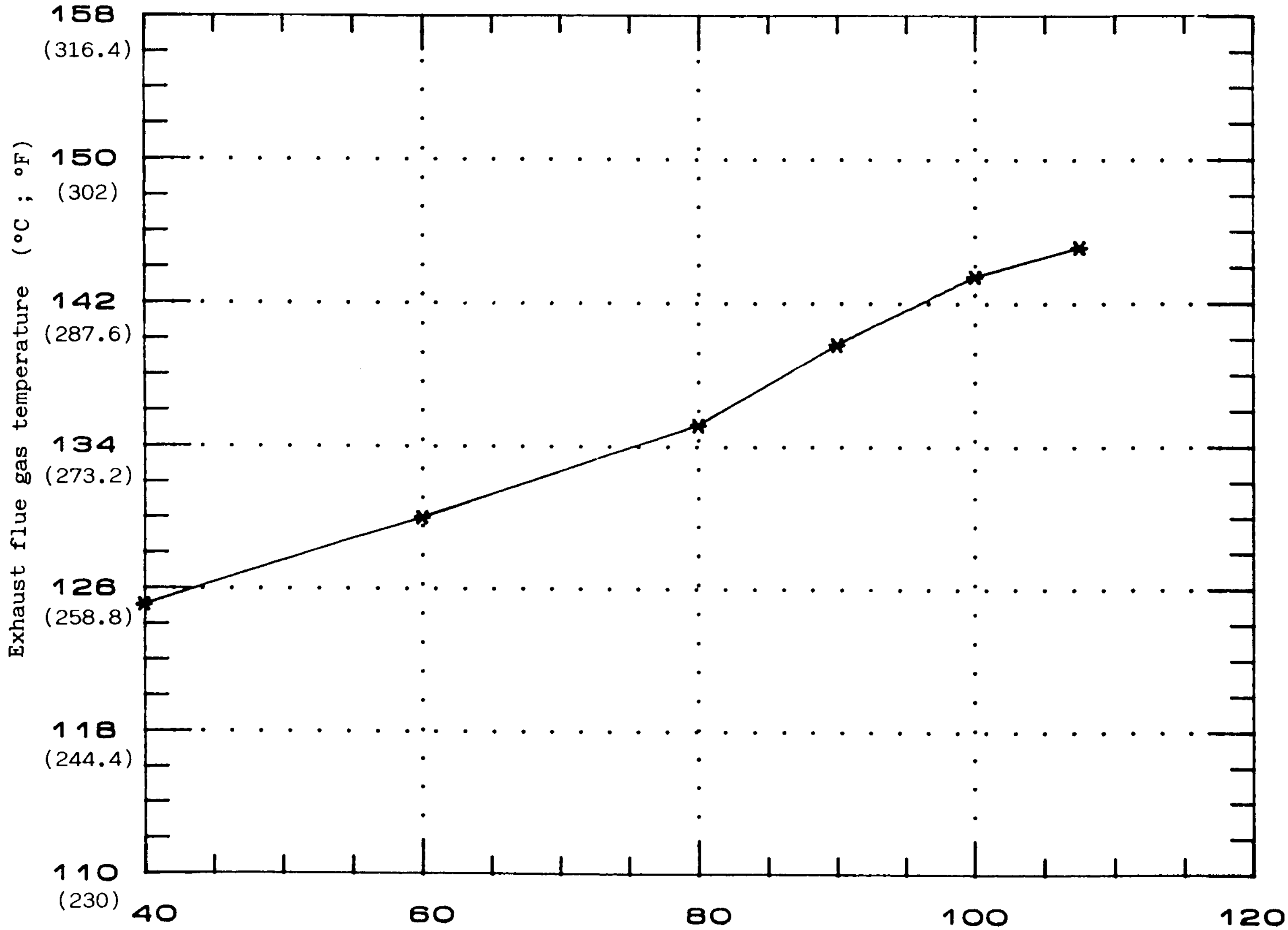


Fig. 11 - Gas turbine load (%)

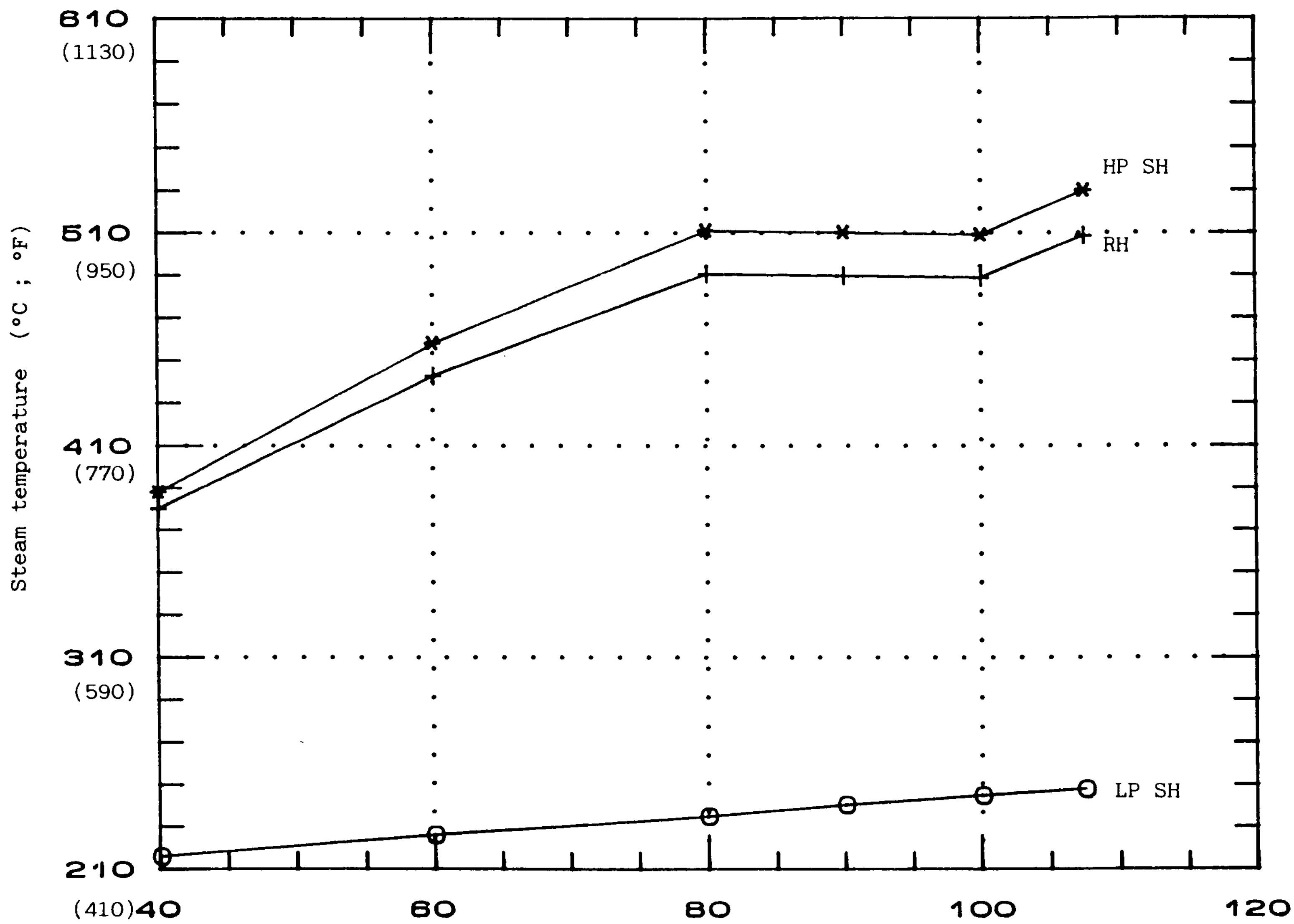


Fig. 12 - Gas turbine load (%)

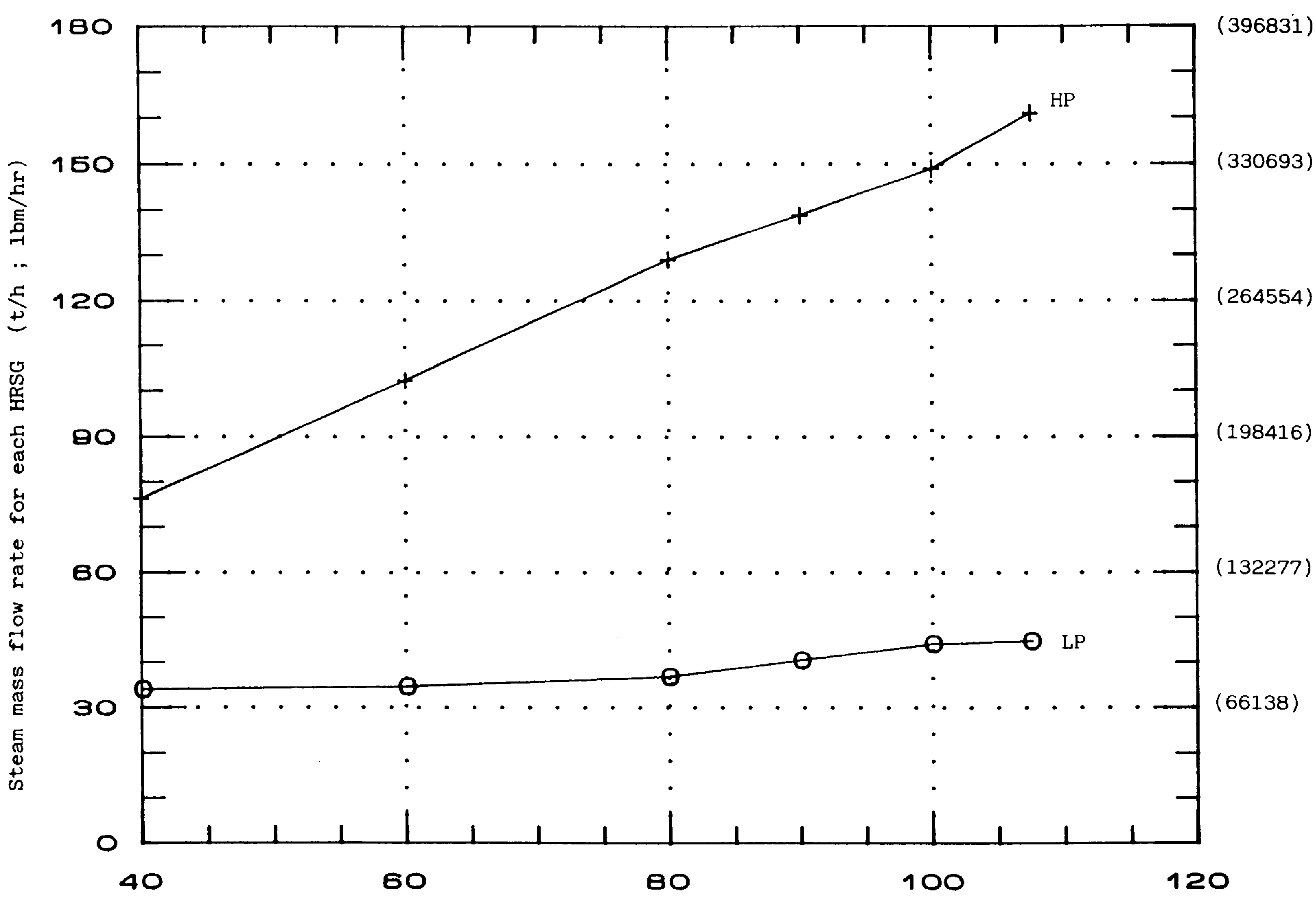


Fig. 13 - Gas turbine load (%)

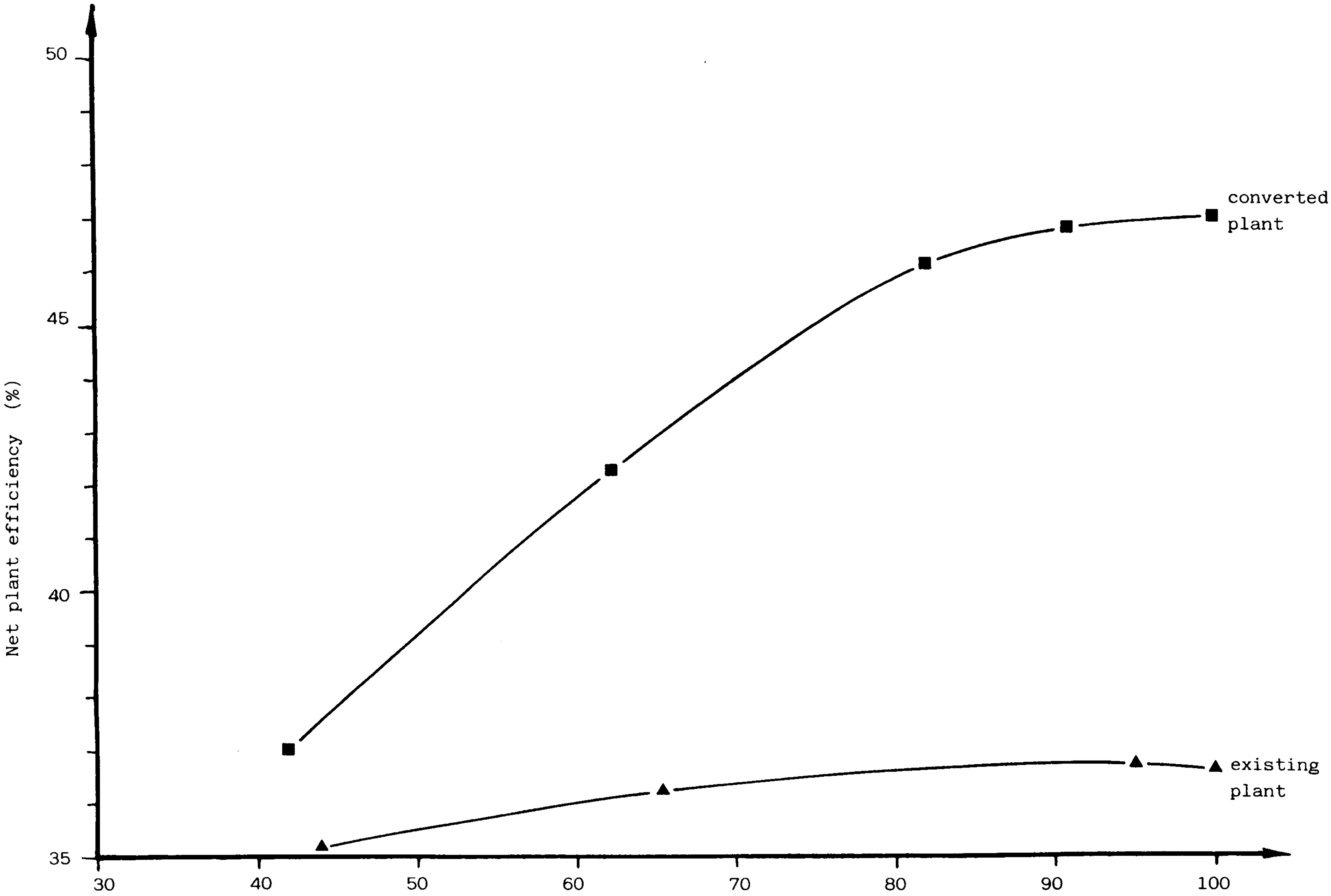


Fig. 14 - Full load percent