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Steam Power Station Design Criteria

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by

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Introduction

The trend for improving the efficiency of steam power stations resulted in a steady increase of both steam pressures and temperatures. The maximum possible live steam conditions are, however, limited by the characteristics of the materials available. Relevant diagrams are given for superheater tubes and live steam pipes. In selecting the live steam conditions it will have to be decided in each case whether the use of expensive austenitic materials is justified from an economical point of view or whether the live steam conditions should be selected so that all parts of the plant can be manufactured from ferritic materials. With non-reheat turbines the live steam conditions are limited by the steam wetness in the final turbine stages since with a view to protecting the turbine blades an exhaust wetness of approx. 12 to 13 % only is permissible.

Next to the application of elevated live steam conditions, reheating and regenerative feedwater heating are decisive

factors for improving the efficiency of steam power stations. Target values for the main design data are given in tabular form. As a result of these considerations, the average annual specific heat rate of the turbine plant, the entire plant unit as well as of the complete power station is discussed and represented in the form of diagrams as a basis for determination of the plant efficiency. The average fuel costs and the generating costs per net kWh are covered by further diagrams. Considerations on the capital investment required for steam power stations give a survey on the essential factors involved and are embodied in a diagram showing the mean range of the initial costs of steam power stations with pit coal firing and condensing plants for fresh water cooling. The general trend of some factors affecting the capital investment is also discussed.

Limitation of Live Steam Conditions by Material Strength Properties

In steam power plants, the maximum live steam conditions to be obtained are primarily limited by the heat resistance of the materials available. Since the strength properties of the various steel grades are strongly affected by temperature, steam tubes at the end section of the boiler superheater as well as the live steam pipes are subjected to the highest stresses and thus form the chief criterion for the selection of the live steam conditions.

The maximum live steam conditions at the turbine inlet obtainable with different superheater tube materials are given in Fig. 1. In preparing the curves, the following basic values were used:

Stressing of superheater tube material embodying a safety factor of 1.5 over the 100 000 h creep limit.

Steam pressure drop from superheater outlet to turbine inlet = 5 % of steam pressure at throttle.

Boiler design pressure = 1.1 x superheater outlet pressure (forced calculation boiler)

Temperature drop from superheater outlet to turbine inlet = 5°C

Tube wall temperature = 35°C above relevant steam temperature

Ratio O.D./I.D. of superheater tubes = 1.75.

The progress of the curves is chiefly determined by the 100 000 h creep limit of the material concerned. The horizontal branch of the curves is determined by the resistance limit to scaling of the material. (The resistance limit to scaling is defined by the temperature which must not be exceeded by the steel grade concerned - independent of the level of mechanical stressing - in continuous operation for preventing a shortening of the tube life due to scaling.)

The live steam conditions may be selected from Fig. 1 for the relevant operational requirements. The upper limit of the live steam conditions is given by the range of application of that superheater tube material the use of which is still economical.

The live steam conditions are further limited by the range of application of the steel grades used for the live steam piping. Apart from a few special designs, such as steel-tape armoured pipes, the maximum possible steam conditions are determined, next to the material strength properties, by the ratio between outer and inner diameter of the pipe.

The highest diameter ratio of the pipes available at present

is approx. 1.7. It is not possible, however, to provide this diameter ratio for all pipe sizes. In the case of plants with only one live steam line, a lower diameter ratio will restrict the output obtainable but not the live steam conditions.

Parallel runs of live steam pipes are employed for higher outputs. The maximum permissible live steam conditions for the different materials used for live steam pipes are shown in Fig. 2.

Fig. 3 gives, in tabular form, the strength properties and compositions of the steel grades used for pipes and tubes.

On the turbine side, limitations are imposed on the maximum permissible live steam conditions by the strength properties of the admission elements. These are, however, within the limits shown in Figs. 1 and 2.

Limitations of Live Steam Conditions by Exhaust Wetness

The turbine plays an essential part in the design of the steam cycle. Besides the problems already dealt with, the maximum possible live steam conditions are limited also by the so-called "cold end" of the turbine, i.e. the last turbine stages. For protection of the blades in this region, an exhaust wetness of approx. 12 to 13 % only can be permitted. It thus

follows that with non-reheat cycles operating at the usual live steam temperature, steam pressures of up to approx. 90 to 110 atg at the turbine inlet can be safely handled. The exact limit pressure for non-reheat turbines depends on the condenser pressure and, consequently, on the circulating water temperature as well as on the turbine efficiency and will have to be determined separately for each application.

When a higher live steam pressure is desired for reasons of plant economy, the reheat cycle will have to be applied for ensuring that the permissible limit of exhaust wetness is not exceeded.

We differentiate between two ranges for the maximum permissible steam conditions as shown in Figs. 1 and 2. These ranges are obtained as a result of the material characteristics:

Ferritic steels - with body-centered cube lattice space - comparatively low alloyed (e.g. 2.25 % Cr; 1.0 % Mo)

Austenitic steels - with face-centered cube lattice space - highly alloyed (e.g. 16.5 % Cr; 1.8 % Mo; 16.5 % Ni).

It is necessary, therefore, that next to price and steam wetness considerations also the differing properties of the two groups are taken into account for power station planning and

that the relative merits of each group are assessed on the basis of local conditions.

The trend for improving the efficiency of steam power stations resulted in a steady increase of both steam pressures and temperatures. The limitations imposed by the characteristics of the available materials have been outlined above.

Gain Obtained by Feedwater Heating

Next to elevated live steam conditions and reheating, regenerative feedwater heating is a decisive factor for improving the efficiency of steam power plants. The gain in heat rate achieved by regenerative feedwater heating is a direct result of the final feedwater temperature and the number of heater stages.

Diagrams showing the gain in heat rate are available so that the optimum conditions may be selected. Unfortunately, however, a combination of such diagrams for several live steam conditions results in curves which are difficult to follow. For this reason, the examples given herein are restricted to a few diagrams showing the gain in heat rate for some selected operating conditions.

Fig. 4 shows the effect of feedwater heating on the specific

heat rate as a function of the total heat rise in the heating plant for the following design data:

Economical rating	9 MW
Maximum rating	10 MW
Throttle steam conditions	28 ata, 448 C
Condenser pressure	0.035 ata
Circulating water temperature	15°C
Number of heaters	1 to 5

Fig. 4 indicates that with five feedwater heating stages a gain in heat rate up to 8.8 % is possible at economical load of the turboset as compared with a cycle arrangement not incorporating feedwater heating.

Figs. 5 and 6 give comparisons between a non-reheat and a reheat plant operating under otherwise identical conditions.

	Fig. 5	Fig. 6
Economical rating	50 MW	50 MW
Maximum rating	64 MW	64 MW
Throttle steam conditions	111 ata, 535°C	111 ata, 535°C
Reheat	none	at 26/23.5 ata to 540°C
Condenser pressure	0.06 ata	0.06 ata
Circulating water temperature	25°C	25°C
Number of heaters	1 to 10	1 to 10

While with a non-reheat plant a gain in heat rate up to approx. 12.8 % with ten-stage feedwater heating may be achieved in comparison with a cycle not incorporating feedwater heaters, approx. 10.7 % only are obtainable with a reheat plant under otherwise identical conditions.

It is thus evident from a comparison of Figs. 5 and 6 that the possible gain by feedwater heating is higher with non-reheat than with reheat cycles for the following reasons:

The specific heat rate of a steam power plant is improved by reheating as well as by feedwater heating. Either of these methods brings the steam cycle nearer to the ideal Carnot cycle. Since part of the practicable improvement is achieved already by the introduction of reheating, the extent of a possible improvement through feedwater heating is less with reheat plants. Altogether, the improvement of the specific heat rate by feedwater heating plus reheat will be greater than would be the case with feedwater heating only at otherwise identical operating conditions.

The introduction of reheating not only results in a fundamental modification of the cycle arrangement but also affects the overall design of the plant. Comparing reheat and non-reheat plants

operating under identical conditions and having the same generator outputs reveals the following salient points of reheat plants:

(1) Lower steam wetness in the last turbine stages, whereby the inner efficiency of the L.P. section is improved.

(2) Lower live steam flow, whereby the inner efficiency of the H.P. section may be reduced, while the steam requirements of the turbine-driven boiler feed pump decrease.

(3) Lower exhaust steam flow, thus lower c.w. and power requirements for circulating water and condensate pumps.

Owing to the many and varied factors affecting the cycle efficiency, the gain in the heat rate achieved by reheating is approx. 3.5 to 4.5 % depending on the prevailing conditions and will have to be determined separately for each application. It should be remembered, moreover, that the gain brought about by reheating can be determined properly only for a live steam range which is possible without reheat (exhaust wetness 12 to 13 %) as well as by application of the reheat cycle. At higher live steam pressures, which in view of the steam wetness in the last turbine stages are obtainable with reheating only, one can not speak of any gain by reheat.

Further examples of the gain achieved by feedwater heating are

given in Figs. 7 and 8 for high-pressure and extra high-pressure plants with single and double reheat.

Fig. 7 illustrates the gain in heat rate obtained by feedwater heating for a plant of a design usually employed in Germany for medium and large unit ratings from 64 to 150 MW at single reheat. The plant design data are as follows:

Economical rating	87 MW
Maximum rating	100 MW
Throttle steam conditions	181 ata, 535°C
Single reheat at	33/30.5 ata to 540°C
Condenser pressure	0.03 ata
Circulating water temperature	12°C
Number of heaters	1 to 10

Fig. 9 shows a 150 MW reheat turbine-generator set operating under similar design conditions. With this unit, all turbine components and pipes as well as the super-heater tubes can be manufactured from ferritic material (10 Cr Mo 910).

For a cycle operating at very high live steam conditions and double reheat, Fig. 8 depicts the gain achieved by feedwater heating at the following cycle conditions:

Economical rating	135 MW

Maximum rating	150 MW
Throttle steam conditions	300 ata, 605°C
First reheat at	120/111 ata to 555°C
Second reheat at	36/33 ata to 555°C
Condenser pressure	0.025 ata
Circulating water temperature	9°C
Number of heaters	1 to 10

These elevated high steam conditions necessitate the use of austenitic materials.

Fig. 10 is a section through the H.P. part of a 100 MW turbine for operation at 260 atg, 605°C and reheat to 530°C. Similar units of this type have been in successful operation for several years.

Fig. 11 shows a 250 MW unit rated for 250 atg, 540°C and reheat to 540°C.

Chief Design Data for Steam Power Plants

Next to the application of reheat and feedwater heating for improving the cycle efficiency (and thus the specific heat rate) other factors such as terminal temperature difference of the feed heaters, arrangement and rating of desuperheaters and drain coolers, use of extraction steam-driven turbines for boiler feed pumps and

other auxiliaries, recovery of the heat losses of hydrogen and oil coolers etc. will have to be taken into account. These factors are, however, not dealt with in detail since the facts outlined previously herein already give a survey on the chief design data for steam power plants.

Based on the usual live steam conditions in Germany and on the IEC recommendations, the data given in the form of tables in Figs. 12, 13 and 14 represent an evaluation of the design criteria mentioned previously.

Specific Heat Rate of Steam Turbine Plants

In the planning of steam power plants, the specific heat rate is of particular importance. It is not possible, unfortunately, to represent the specific heat rate for various design conditions in simple graphic form since the heat rate of steam turbine plants is affected by too many factors. When the investigations are, however, restricted to the differentiation of a few essential factors such as shown in Fig. 15, a general and clear outline of a relatively wide range of application of steam power plants is obtained. The diagram gives the mean specific heat rates of 56/64 MW to 130/150 MW units with and without reheat at economical load. The values covered by the diagram apply to turbine plants with six or seven-stage feedwater heating to the final

temperatures t_{sp} shown on the diagram, but excluding the condensate losses (make-up water flow) and without considering the spray water flow supplied to superheater and reheater for control of steam temperature.

The heat rate of the total plant is determined also by the steam generators and the station auxiliaries. The steam generator efficiency mainly depends on the fuels and the methods of firing used. With pit coal firing, the economical efficiency of the steam generator in large-size plants is between 88 and 93 % depending on type of firing and fuel quality.

The variable part of the station service demand is chiefly determined by the power requirements of the boiler feed pump, which may vary within wide limits (approx. 2 to 4 %) depending on the live steam conditions selected. With coal-fired stations, the remainder of the station service demand can be assumed to be approx. 3.0 to 4.5 %, referred to maximum load.

Average Annual Specific Heat Rate of Entire Plant Unit, Fuel Costs and Net Generating Costs per kWh.

For determining the efficiency of the entire plant the average annual specific heat rate is of importance. With the same plant, this heat rate may vary within wide limits in dependence of the number of operating hours. Fig. 16 shows the specific heat rate

for a 72/80 MW plant within a live steam range from 120 atg, 530°C to 180 atg, 565°C. The diagram gives the specific heat rate for economical and maximum load as well as the annual average for 4000 to 7000 operating hours per year separately for the turbine-generator set and for the entire unit. It is seen that the effect of the operating hours on the average annual specific heat rate may be far higher than the influence of the live steam conditions.

Fig. 17 illustrates the average annual fuel costs per net kWh plotted against the number of operating hours for 4 different live steam design conditions. The calculation of the fuel costs was based on the specific heat rate shown in Fig. 16 and on heat generating costs of 2.95 US$/Gcal (12.394 DM/Gcal) - corresponding to a price of coal of 20.-- US$/ton (84,-- DM/t) at a lower calorific value of 6500 kcal/kg[x]. For other generating costs, the fuel costs will have to be calculated accordingly.

Fig. 18 shows, for the same example, the average annual generating costs per net kWh at the H.V. bus plotted against the number of operatinghours for three rates of amortisation, i.e. 11.746 % (at a term of depreciation of 20 years and 10 % interest on capital), 10.185 % (at a term of depreciation of 20 years and 8 % interest on capital), and 8.718 % (at a term of depreciation of 20 years and 6 % interest on capital). The average annual net generating costs per kWh given herein do not include the building

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x) 1 US$ = 4.20 DM

interest as well as other factors.

Average Range of Capital Investment for Steam Power Stations

It is necessary for a determination of the average annual net generating costs per kWh that especially the capital investment, i.e. the initial cost, is known.

The graphic representation in Fig. 19 illustrates the most essential factors influencing the initial cost of steam power plants either directly or through planning considerations.

The individual factors, i.e. the conditions dictated by external circumstances and situations and the requirements resulting therefrom are combined in groups according to their interrelation and the sequence of the planning and projecting work, with the basic scheme being as follows:

Special Conditions and Requirements

(1) Initial planning relating to economics and plant efficiency considerations

Fuels, lubricants, etc.

System layout and load conditions

General economic considerations

Location

(2) Technical planning and projecting work

Thermal design

Water supply

Civil engineering design

(3) Construction planning

Plant machinery

Electrical equipment

Civil engineering

The factors outlined under 1 to 3 above cover the entire plant and the resulting overall costs.

The factors influencing the costs are indicated by a heavy line, whereas those factors which are of importance only for planning are shown by thin lines. An arrow indicates either an active relation between one element and the other or a purely pertinent connection.

A few notes are given outside the boxes either for information purposes or as examples.

Considering the variety of factors influencing the capital investment, the conclusion is reached that no differentiated outline of the capital investment for all possible cases but only a survey on the general trends can be achieved. The mean range of the

absolute power station costs in dependance of the installed capacity is shown on Fig. 20, which gives statistical values for condensing turbine-generator sets in unit connection with two turbines and two boilers with fresh water cooling. When very many power station costs are evaluated, the mean value curve, which represents a medium design and construction of the power station, may be regarded as that curve indicating the most probable trend in the capital investment. This appears not to be justified since the number of results evaluated cannot by any means be regarded to be large enough and as, moreover, a medium design and construction of a power station not based on concrete facts has no practical value. The "true" trend curve will hardly follow a steady course since the factors influencing the design are hardly "occasional errors" but largely "systematic errors". It thus follows that trend curves may be based only on subjective and, partly, even on arbitrary assumptions.

Although an attempt may be made, in line with experience with plants already planned completely, to compile a few differentiated target values, the application of such data for other cases will always be questionable. Under due consideration of this reservation, a few figures indicating the general trend in costs are given below as a supplementation to the diagram of Fig. 20.

Referred to a power station using pit coal as fuel the costs of a lignite-fired power station

with large-size units will be approx. 6 to 7 % higher,

with small-size units approx. 7 to 10 % higher.

The costs of a power station using oil as fuel will be approx. 10 % lower, while with small-size units and oil firing the costs will even be 15 % lower.

A power station employing mixed firing will be approx. 4 to 5 % higher in costs when the power station is to deliver its full output with straight oil firing as well as with straight coal firing.

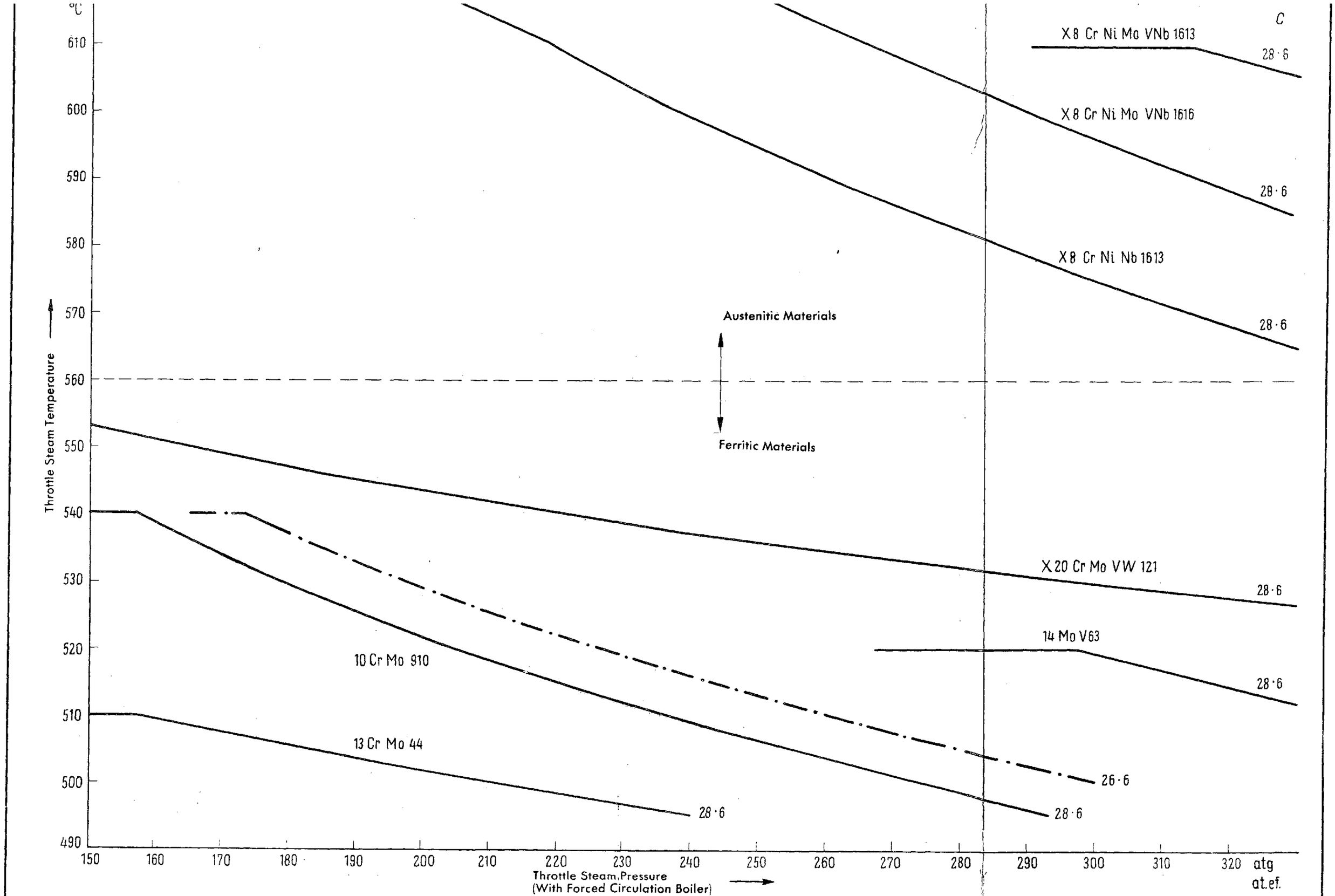
With a power station using high-quality natural gas (lower calorific value ≐ 8000 kcal/m^3 at N.T.P.) as fuel, the capital investment will be about the same as for an oil-fired plant since any increase in the boiler size is compensated by the omission of the oil system. A rise of the capital expenditure in comparison with oil-fired plants is to be expected when waste gas or a low-grade gas is used.

It should be noted, however, that power stations using pit coal mainly use fresh water for the condenser, while in view of the coal transport charges lignite-fired stations will have to be installed near the pit where fresh water is rarely available. For power stations using lignite firing the use of cooling towers may

thus be regarded practically as the normal case. There is a further difference in the capital expenditure inasmuch as a power plant with pit coal firing and cooling towers is approx. 3.5 to 4.0 % more expensive then a pit coal power station with fresh water cooling. In the case of lignite-fired stations the conveyors may even reach down into the pit so that even in this respect only there may be considerable differences in the plant design and in the capital expenditure. The latter may apply, however, also to mine-owned pit coal power stations, especially when non-saleable coal is to be fired. For such a power station, however, a higher capital expenditure than for plants with normal pit coal firing may likewise be expected.

Summary

An attempt was made to compile the design data for planning steam power stations as well as to state some salient points on plant efficiency. The attached diagrams and the other details used in this article refer to power stations equipped with condensing turbine-generator sets. A steam supply for space heating or industrial purposes may influence the design, the capital investment and particularely the efficiency of a power station to a considerable degree. These questions cannot, however, be discussed within the scope of this article.



LIMITS OF APPLICATION OF STEEL GRADES FOR SUPERHEATER TUBES

FIG. 1

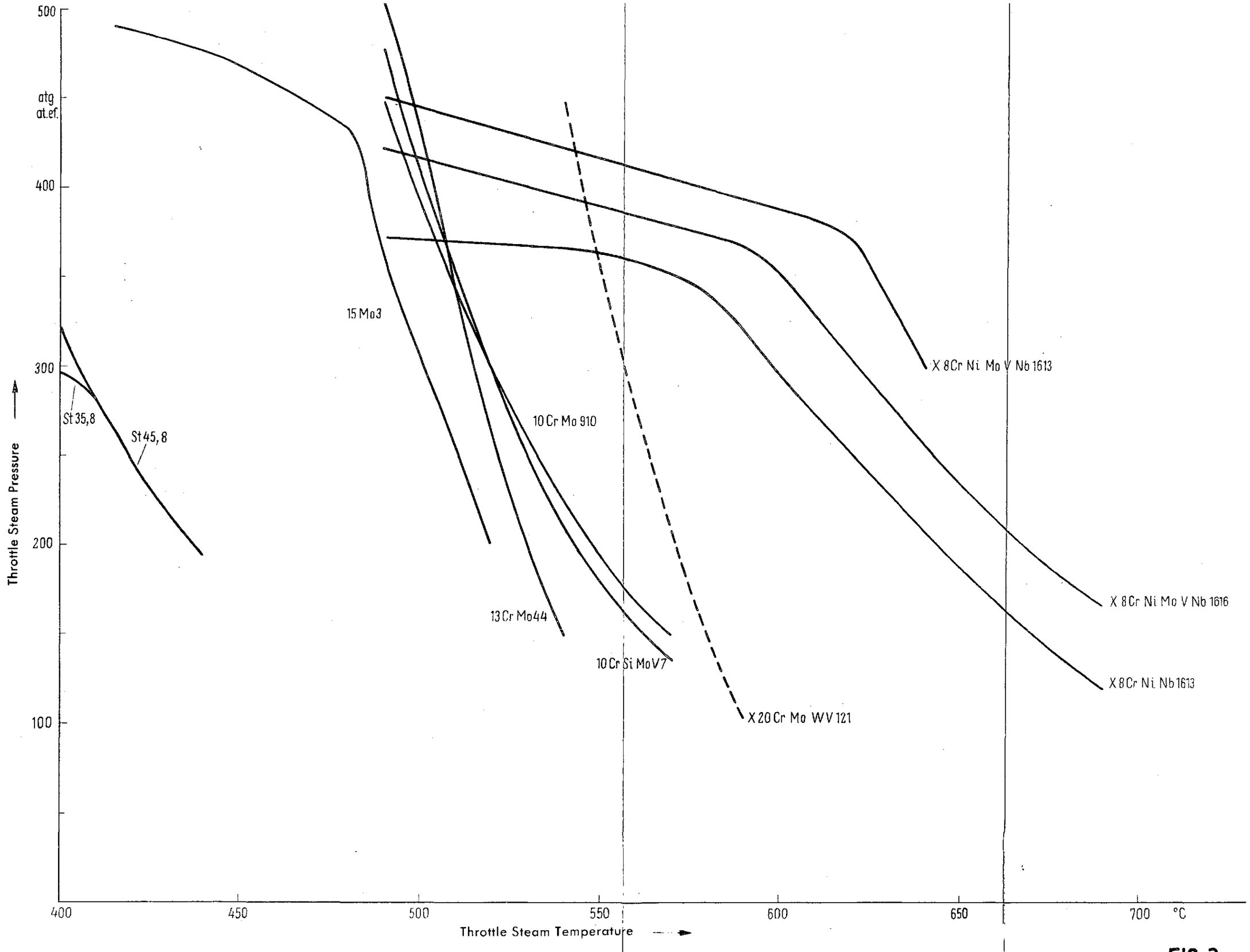


FIG. 2

LIMITS OF APPLICATION OF STEEL GRADES FOR LIVE STEAM PIPES

TUBE AND PIPE MATERIALS — MATERIALES PARA TUBOS

Type (Old type) Tipo (Tipo antiguo)	Material No. Nr. del Material	Chemical Composition Composición quimica %	Yield Point Punto cedente ata/a °C	Yield Point min. kg/mm²	Tensile strength Resistencia tensora min. kg/mm²	Ultimate elongation Alargamiento de ruptura L=5d min. %	Impact strength Resitencia a los impactos min. kgm/cm²	Test temperature Temperatura de prueba °C	1% Creep limit for: 1% limite para: 10.000 hs min. kg/mm²	1% Creep limit for: de flujo 100.000 hs min. kg/mm²	Endurance limit for: limite de 10.000 hs kg/mm²	Endurance limit for: continuación: 100.000 hs kg/mm²
13 CrMo 4 4 (TH 32, Marwe 17L)	7335 (472)	C .10/.18 Cr .70/1.0 Mo .40/.50 Si .15/.35 Mn .40/.70	20	30	45 to/hasta 58	22	6	500 510 520 530 540 550	17 14.9 12.8 10.9 9.1 7.5	12 10.2 8.3 6.6 5 3.7	24 21.3 18.5 15.8 13.3 11	17 14.3 11.4 8.9 6.7 5
10 CrMo 9 10 (3HK,5,Marwe 215E)	7380	C <.15 Cr 2.0/2.5 Mo.90/1.10 Si <.50 Mn.30/.60	20	27	50 to/hasta 65	20	6	500 520 540 550 (580)[1]	16 12.4 9.2 8.0 (5.6)[1]	10 7.4 5.3 4.5 (3.2)[1]	20 15.7 12.1 10.5 (7.0)[1]	14 10 7.4 6.5 (4.6)[1]
X20 CrMo VW 121		C .57-.25 Si.15-15 Cr 11-12.5 Mo .8-1.2 Ni .3-.8 Mn .4-.7 V .25-.35 W .4-.6	20	45	70 to/hasta 85	15		500 520 540 560 580 600	22 19.4 16.0 12 8.5 6	18 15.3 11.9 8.1 5 3	28 25.5 22 17.5 21.5 8.5	23 20.5 17 12.5 8 5.0
14 MoV 63	5515	C .14 Si.25 Cr.50 Mo.55 Mn.50 V .30 P+S <je .03	200 300 400 500	32 30 28 24	50 to/hasta 65	vertical/ a lo largo de 22 horizontal/ atraves 20		450 500 530 550 570		14 10 7.5 6.0 4.5		21 15 11.2 9.0 7.4
AUSTENITIC MATERIAL — MATERIALE AUSTENITA												
X8Cr NiNb 1613 (ATS)(BVT A 1)	4961	C .08 Cr 17.0 Ni 13.0 Nb 10x%C	20	22	55 to/hasta 70	35	>15	500 550 600 650 700	20 14 10 7 5	16 10 6 4 2.5	32 22 15 10 6	25 15 8 5 3.5
X8Cr NiMoVNb 1613 verg 65 (ATS 6) (BVTA 3)	4988	C <.08 Cr ≦17.0 Mo ≦ 1.4 Ni 13.5 V ≦.70 N_2 ≦.10 Nb >10x%C	20	50	65 to/hasta 85	17	>6	550 575 600 625 650	28 26 23 19 16	20 18 16 13 10	30 29 27 23 20	25 23 21 17 14
X8Cr NiMoNb 1616 (ATS 15)(BVT A 20)		C .08 Cr 17.0 Mo 1.8 Ni 16.5 Nb 10x%C	20	25	55 to/hasta 70	35	>15	500 550 600 650 700 750	20 15 11 8 6 4	17 11 7 5 3.5 2	32 23 16 12 8 6	26 16 8 6 4.5 2.5

FIG. 3

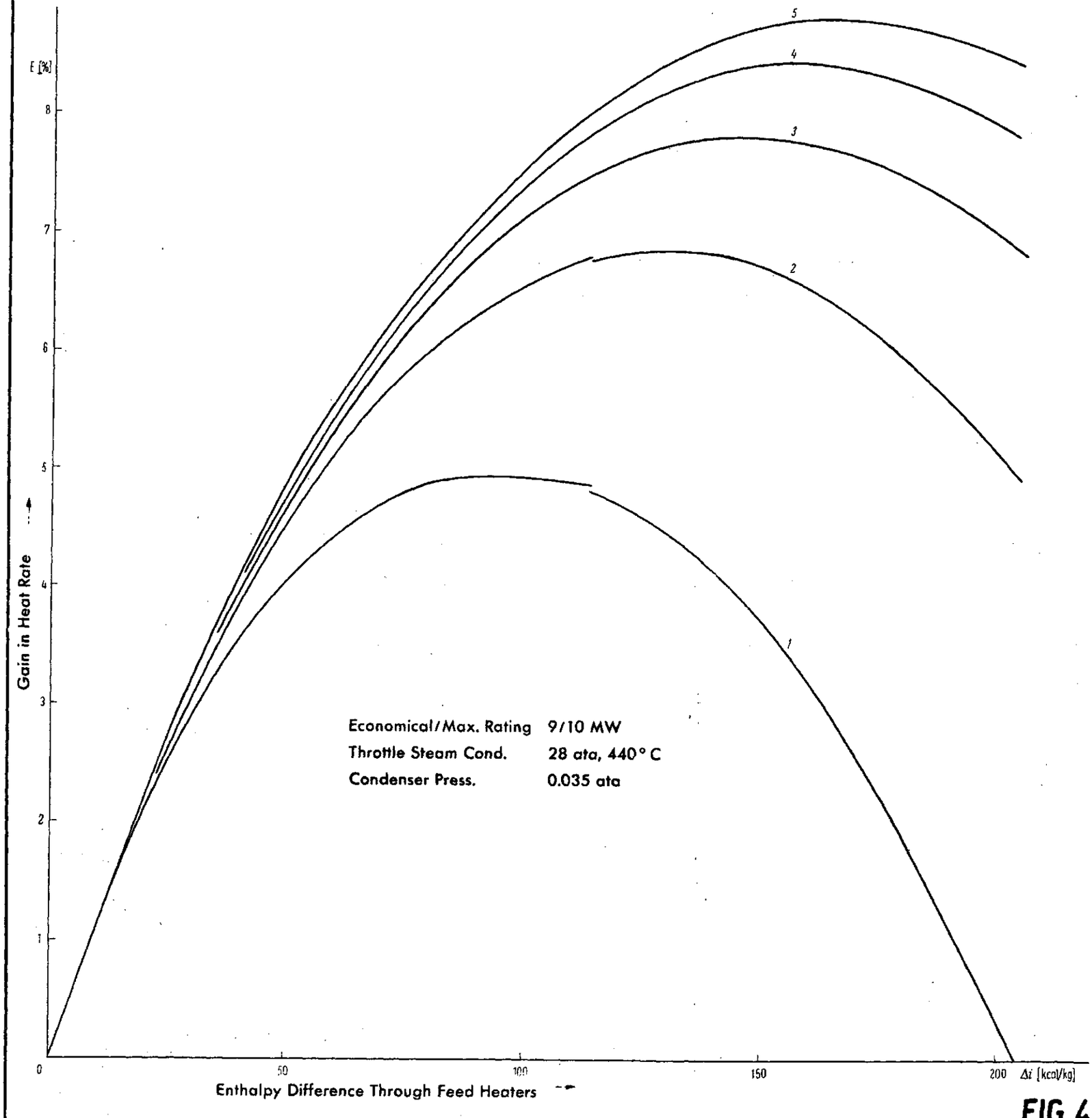


FIG. 4

EFFECT OF FEEDWATER HEATING ON SPECIFIC HEAT RATE OF NON-REHEAT PLANT

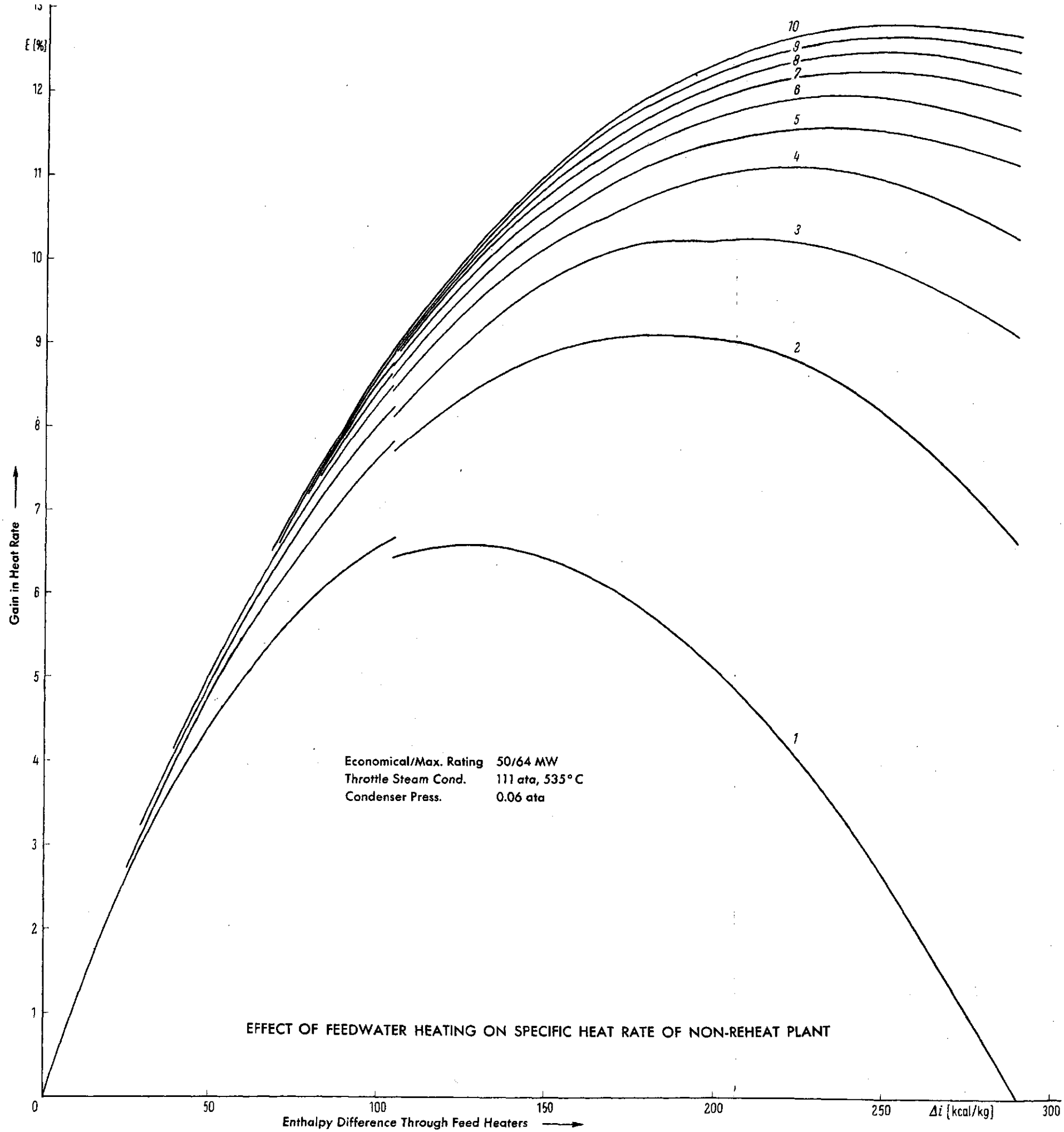
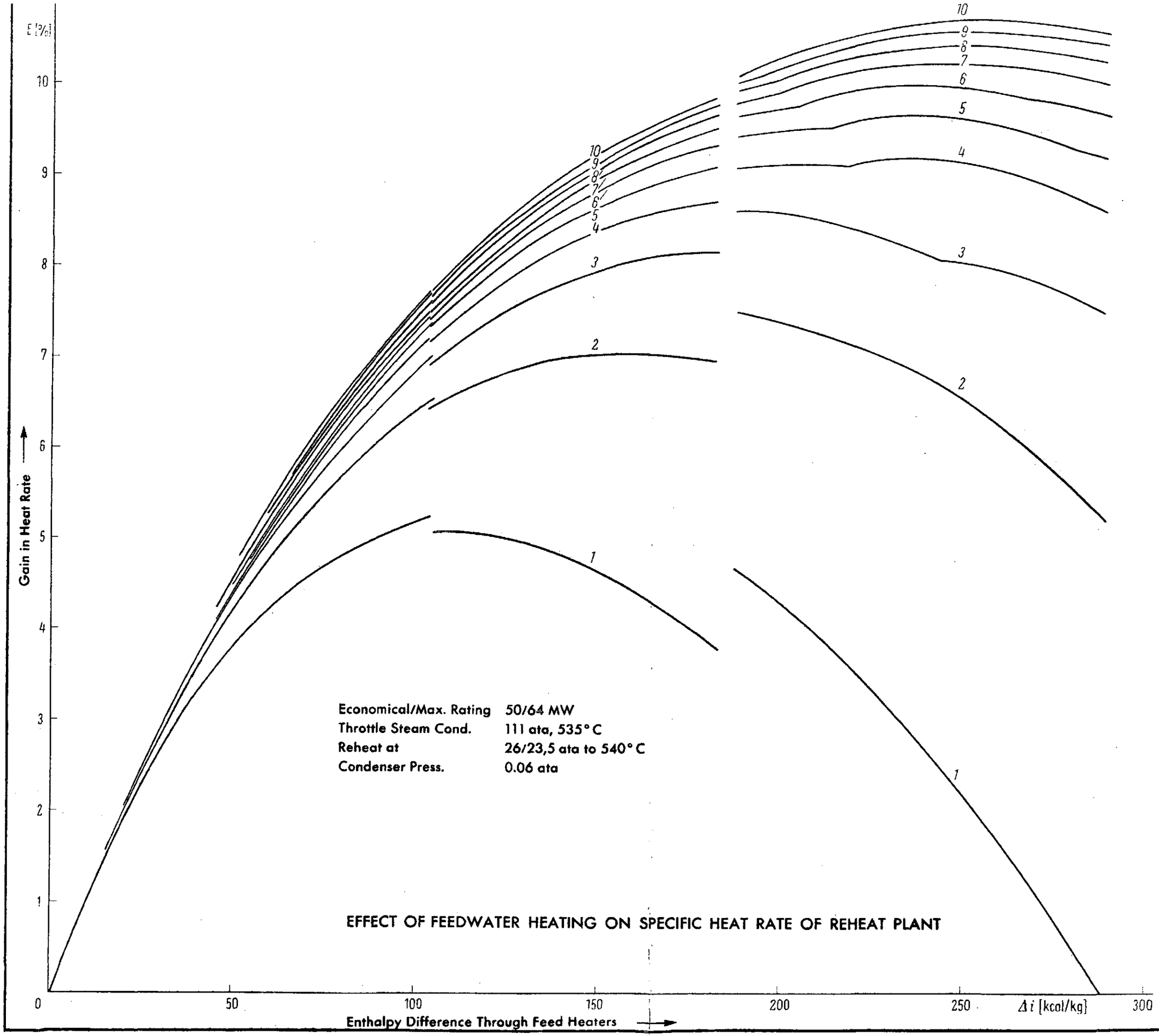
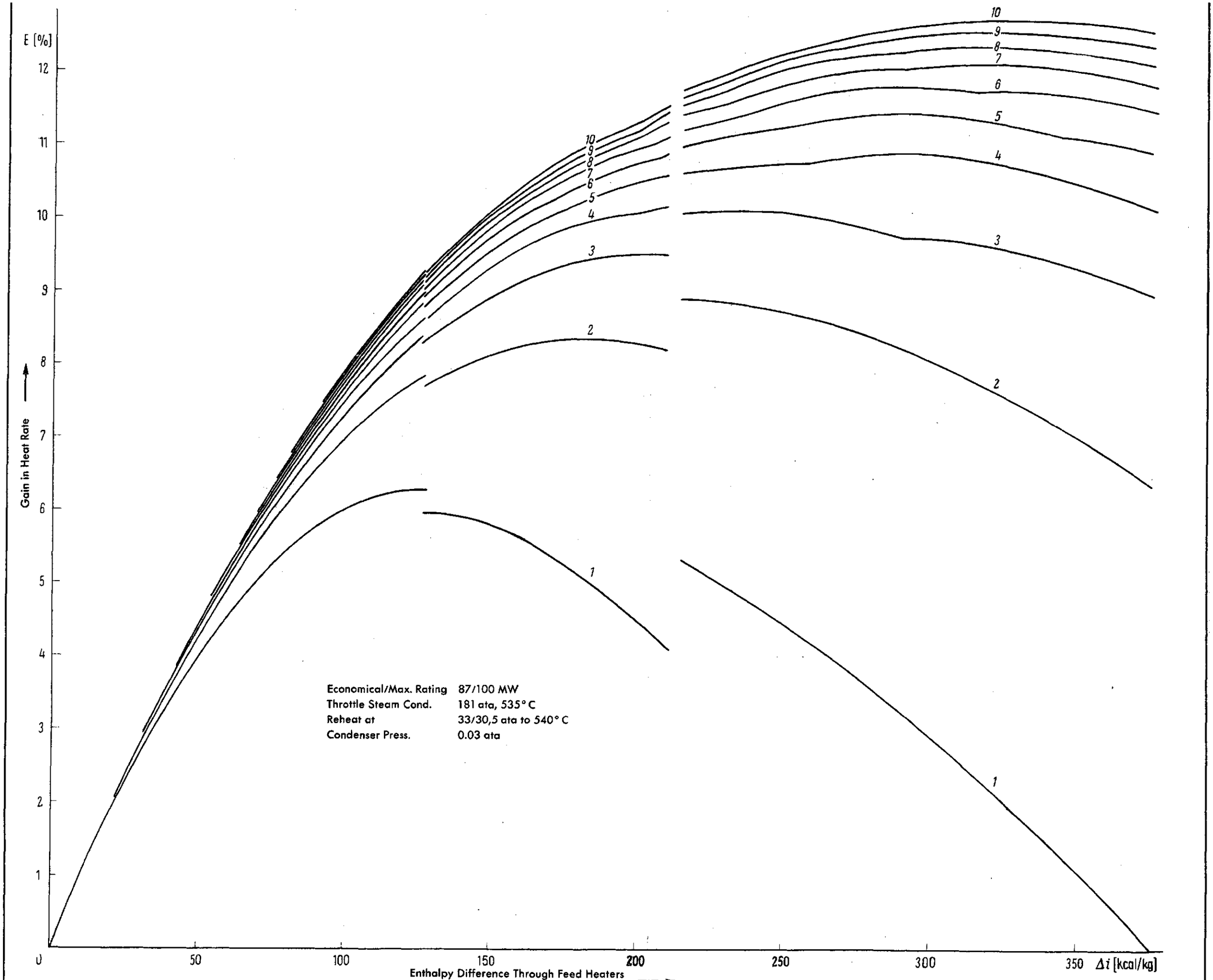

E [%]
Gain in Heat Rate
Economical/Max. Rating 50/64 MW
Throttle Steam Cond. 111 ata, 535° C
Condenser Press. 0.06 ata
EFFECT OF FEEDWATER HEATING ON SPECIFIC HEAT RATE OF NON-REHEAT PLANT
Enthalpy Difference Through Feed Heaters
Δi [kcal/kg]
0
50
100
150
200
250
300
1
2
3
4
5
6
7
8
9
10
11
12
13

FIG. 5



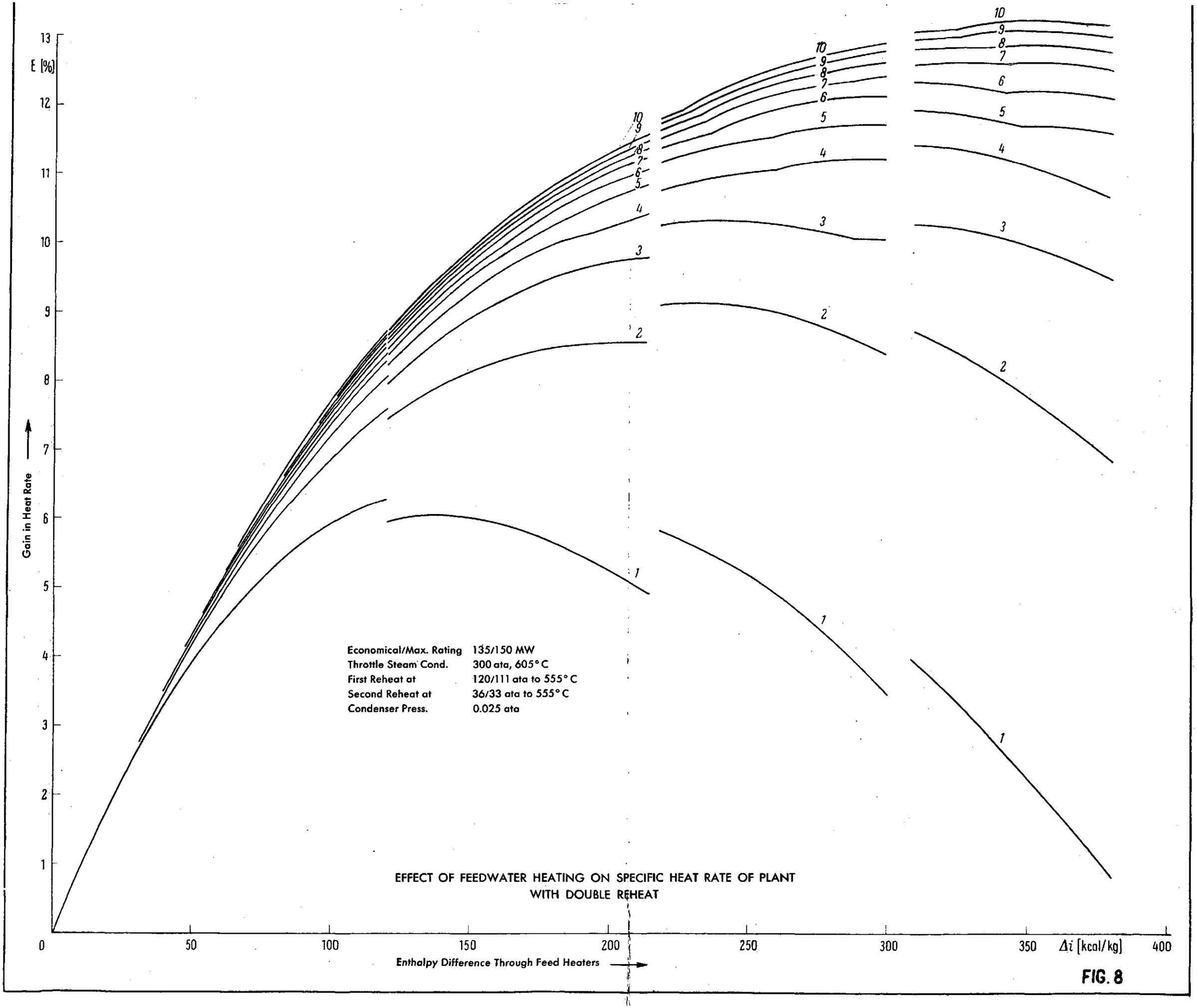
EFFECT OF FEEDWATER HEATING ON SPECIFIC HEAT RATE OF REHEAT PLANT

FIG. 6



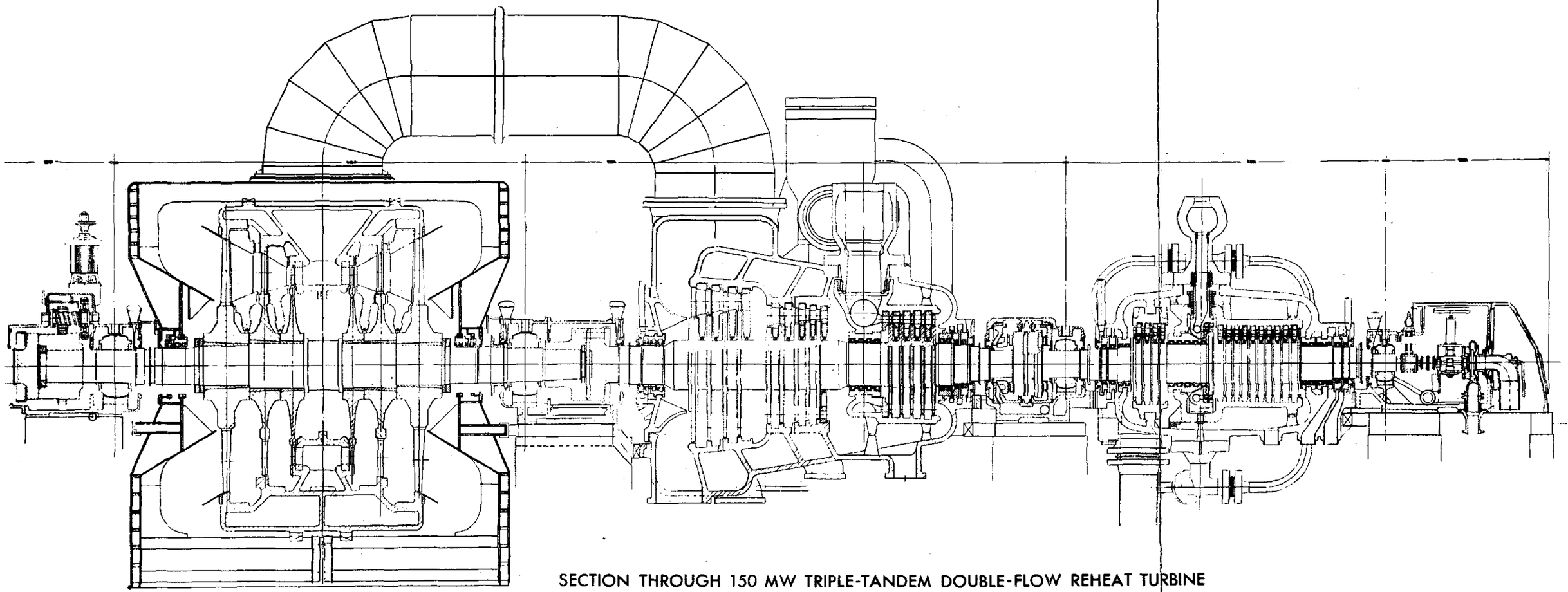
EFFECT OF FEEDWATER HEATING ON SPECIFIC HEAT RATE OF PLANT WITH SINGLE REHEAT

FIG. 7



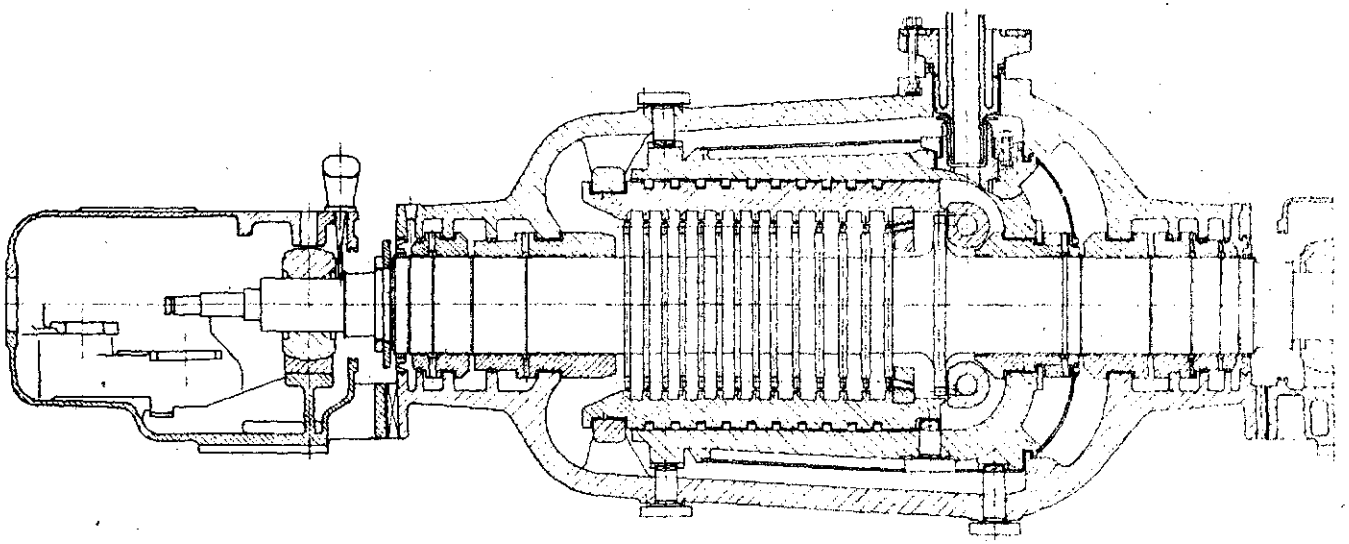
EFFECT OF FEEDWATER HEATING ON SPECIFIC HEAT RATE OF PLANT WITH DOUBLE REHEAT

FIG. 8



SECTION THROUGH 150 MW TRIPLE-TANDEM DOUBLE-FLOW REHEAT TURBINE

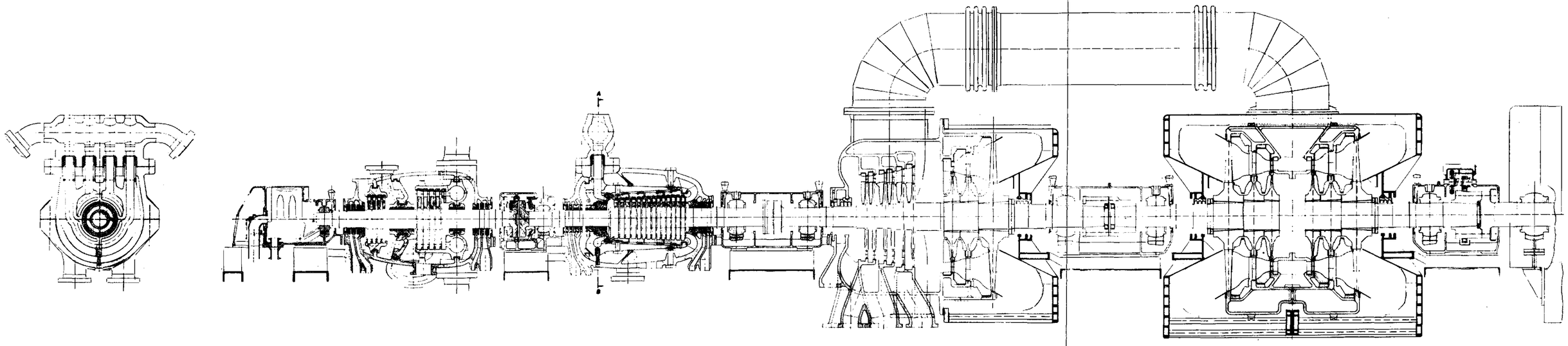
FIG.9



Barrel-Type Inner Casing

H. P. SECTION OF 150 MW EXTRA-HIGH PRESSURE TURBINE, STEAM AT 260 atg AND 605° C

FIG. 10



SECTION THROUGH 250 MW QUADRUPLE-TANDEM TRIPLE-FLOW REHEAT TURBINE

FIG. 11

General Design Date for Medium and High-Pressure Non-Reheat Plants

No.	Description	Unit	Dates												
			To German Regulations								To IEC Recommendations				
1.	Throttle steam pressure	atg	21	27	35	56	70		110		34	42	64	88	105
2.	Throttle steam temperature	°C	390 (385)	440 (435)	445 (440)	495 (490)	500	525 535	500	525 535 565	440 (435)	460 (455)	490 (485)	535 565	535 565
3.	Steam pressure at superheater outlet	atg	22,5	29	37	59	74	74	115	115	36	44	67	92	109
4.	Steam temperature at superheater outlet	°C	400	450	450	500	505	530 540	505	530 540 570	450	470	500	540 570	540 570
5.	Boiler drum pressure (natural circulation boiler)	atg	24	30,5	40	64	82	83	127	129	39	48,5	74	102	123
6.	Boiler design pressure 61) Natural circulation boiler	atg	25	32	42	67	87		136		41	51	78	109	130
	62) Forced circulation boiler	atg	-	-	-	-	-		127		-	-	-	101	120
7.	Target values for feedwater pressure required after boiler feed pump at maximum boiler output (referred to design pressure and operation of safety blowoff valve) 71) Natural circulation boiler	atg	28	45	56	84	103	104	163	165	55	66	94	130	155
	72) Forced circulation boiler	atg	-	-	-	-	-	-	183	185	-	-	-	155	180
8.	Final feedwater temperature and number of heaters	°C	105+ 140	105+ 160	105+ 180	140+ 180	160+	200	180+	230	105+ 180	140+ 190	160+ 200	160 220	180+ 230
		-	2+3	2+4	2+4	3+5	3+5	4+6	4+6	4+6	2+4	3+5	3+5	4+6	4+6

FIG. 12

General Design Data for High-Pressure Plants with Single Reheat

No.	Description	Unit	Data											
1.	Live steam pressure at turbine inlet	atg	105		110		126		140		165		180	
2.	Live steam temperature at turbine inlet 21) All parts of ferritic material	°C	535		535		535		535		535		530 (535)	
	22) Turbine and piping of ferritic material (10 CrMo 9 10) - superheater of austenitic material	°C	565		565		565		565		565		555	
3.	Steam pressure at superheater outlet	atg	109		115		132		147		173		189	
4.	Steam temperature at superheater outlet	°C	540	570	540	570	540	570	540	570	540	570	535 (540)	560
5.	Boiler drum pressure (natural circulation boiler)	atg	123		129		147		162		-		-	
6.	Boiler design pressure 61) Natural circulation boiler	atg	130		136		157		175		-		-	
	62) Forced circulation boiler	atg	120		127		146		162		191		208	
7.	Target values for feedwater pressure required after boiler feed pump at maximum boiler output (referred to design pressure and operation of safety blowoff valve) 71) Natural circulation boiler	atg	155		165		185		205		-		-	
	72) Forced circulation boiler	atg	180		185		210		230		270		290	
8.	Final feedwater temperature	°C	220+ 240	225+ 240	220+ 240	225+ 240	230+ 240	230+ 250	230+ 250	235+ 250	235+ 280	235+ 280	235+ 280 (300)	240+ 280 (300)
9.	Number of heaters	-	5+6	6+7	5+6	6+7	6+7	6+7	6+7	6+8	6+8	6+8	6+8 (8+10)	6+9 (8+10)
10.	Reheat steam temperature at turbine inlet (equal to or lower than values given)	°C	540	565	540	565	540	565	540	565	540	565	540	560
11.	Reheat steam temperature at reheater outlet	°C	545	570	545	570	545	570	545	570	545	570	545	565
12.	Reheat steam pressure		25 to 32 ata depending on local conditions											
13.	Reheat steam pressure at turbine inlet at approx. 8 % pressure drop		23 to 29.5 ata											

Note: The feed pump delivery pressure at closed shut-off valve is a criterion of the H.P. feed heater design.

FIG. 13

General Design Data for High-Pressure Plants with Single or Double Reheat

No.	Description	Unit	Data		
1.	Live steam pressure at turbine inlet	atg	225	250	300
2.	Live steam temperature at turbine inlet				
	21) All parts of ferritic material	°C	510 (515)	505 (510)	490 (495)
	22) Turbine and piping of ferritic material, superheater of austenitic material	°C	540	530	520
	23) Use of austenite X8CrNiNb 16 13	°C	570	560	545
	24) Use of austenite X8CrNiMoNb 16 16	°C	585	575	555
	25) Use of austenite X8CrCiMoVNB 16 13	°C	610	610	605
3.	Steam pressure at superheater outlet	atg	237	263	315
4.	Steam temperature at superheater outlet	°C	515 (520) 545 575 590 615	510 (515) 535 565 580 615	495 (500) 525 550 560 610
5.	Boiler design pressure	atg	261	290	347
6.	Target values for feedwater pressure required after boiler feed pump at maximum boiler output (referred to design pressure and operation of safety blowoff valve)		350	385	445
7.	Final feedwater temperatur	°C	260 - 300	280 - 320	280 - 340
8.	Number of heaters	-	7 - 10	7 - 10	8 - 10
9.	Reheat data		Depending on local conditions		

FIG. 14

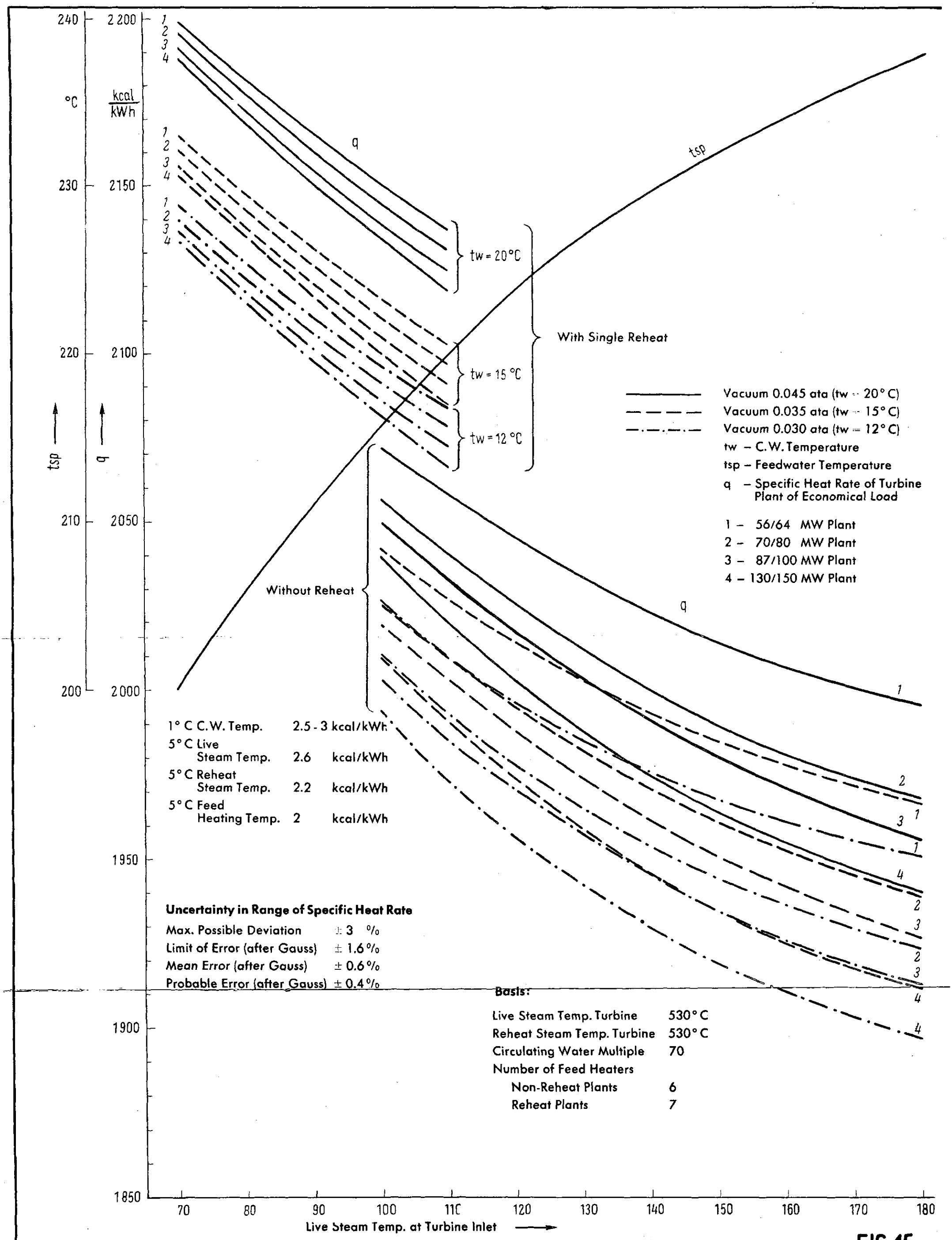


FIG. 15

MEAN SPECIFIC HEAT RATES OF LARGE-SIZE CONDENSING TURBINE PLANT AT ECONOMICAL LOAD (87 % OF MAX. LOAD)

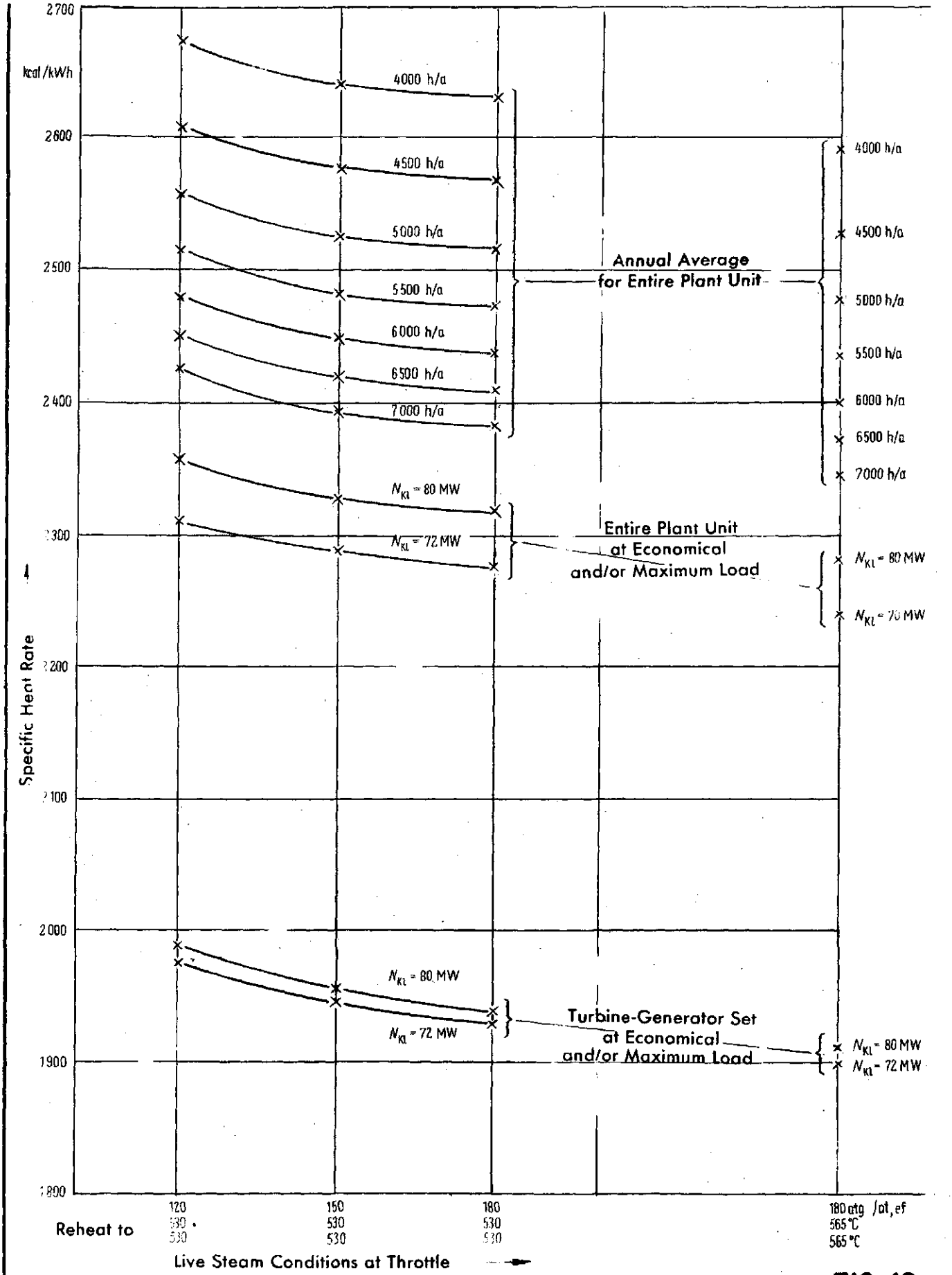
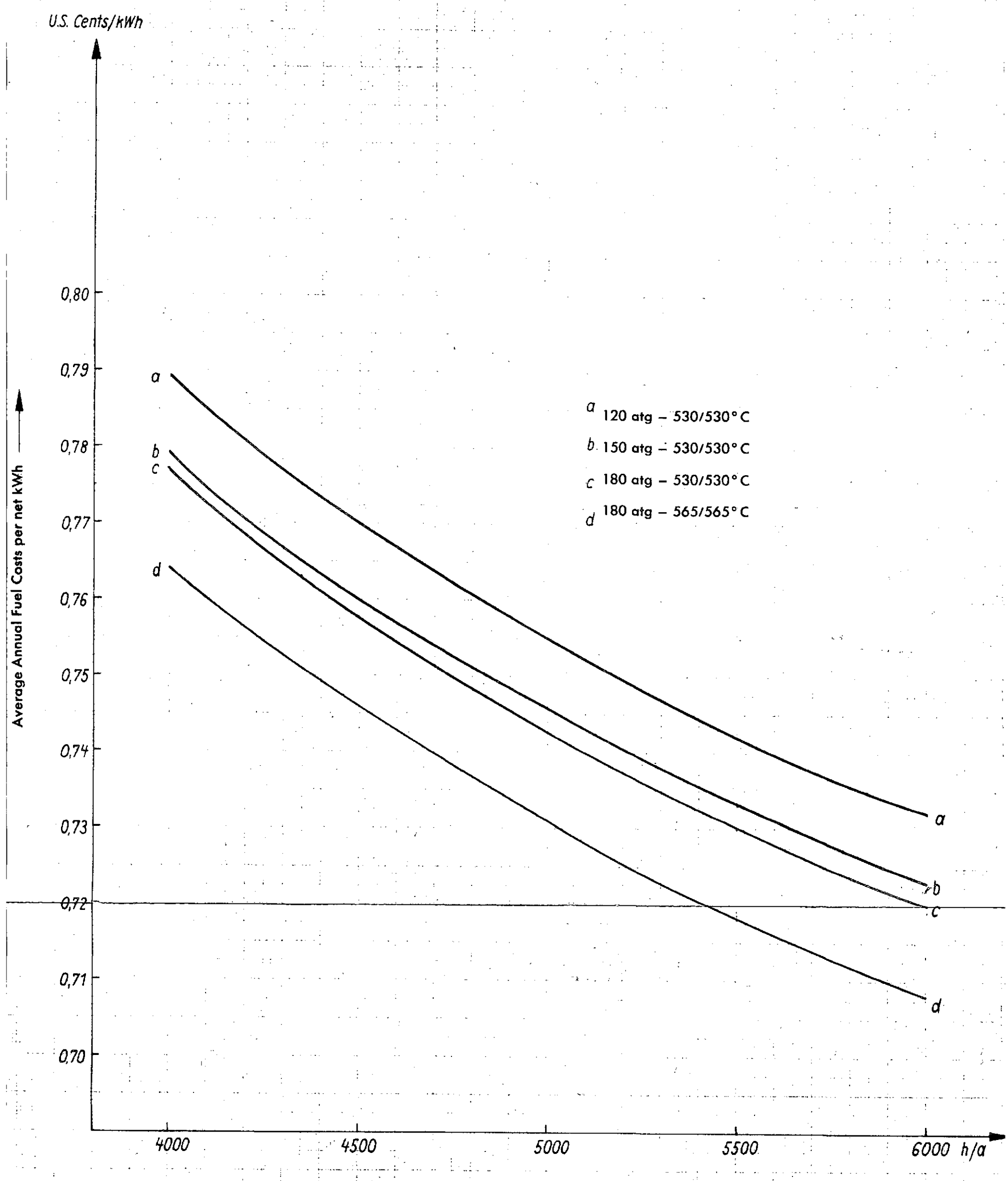


FIG. 16

AVERAGE ANNUAL SPECIFIC HEAT RATE



Number of Annual Operating Hours h/a
(Referred to Annual Elec. Energy and Max. Available Elec. Power at Straight Condensing Operation)

FIG. 17

AVERAGE ANNUAL FUEL COSTS PER NET kWh
FOR 72/80 MW POWER PLANT (Unit heat cost 2.95 $/Gcal; 12.394 DM/Gcal)

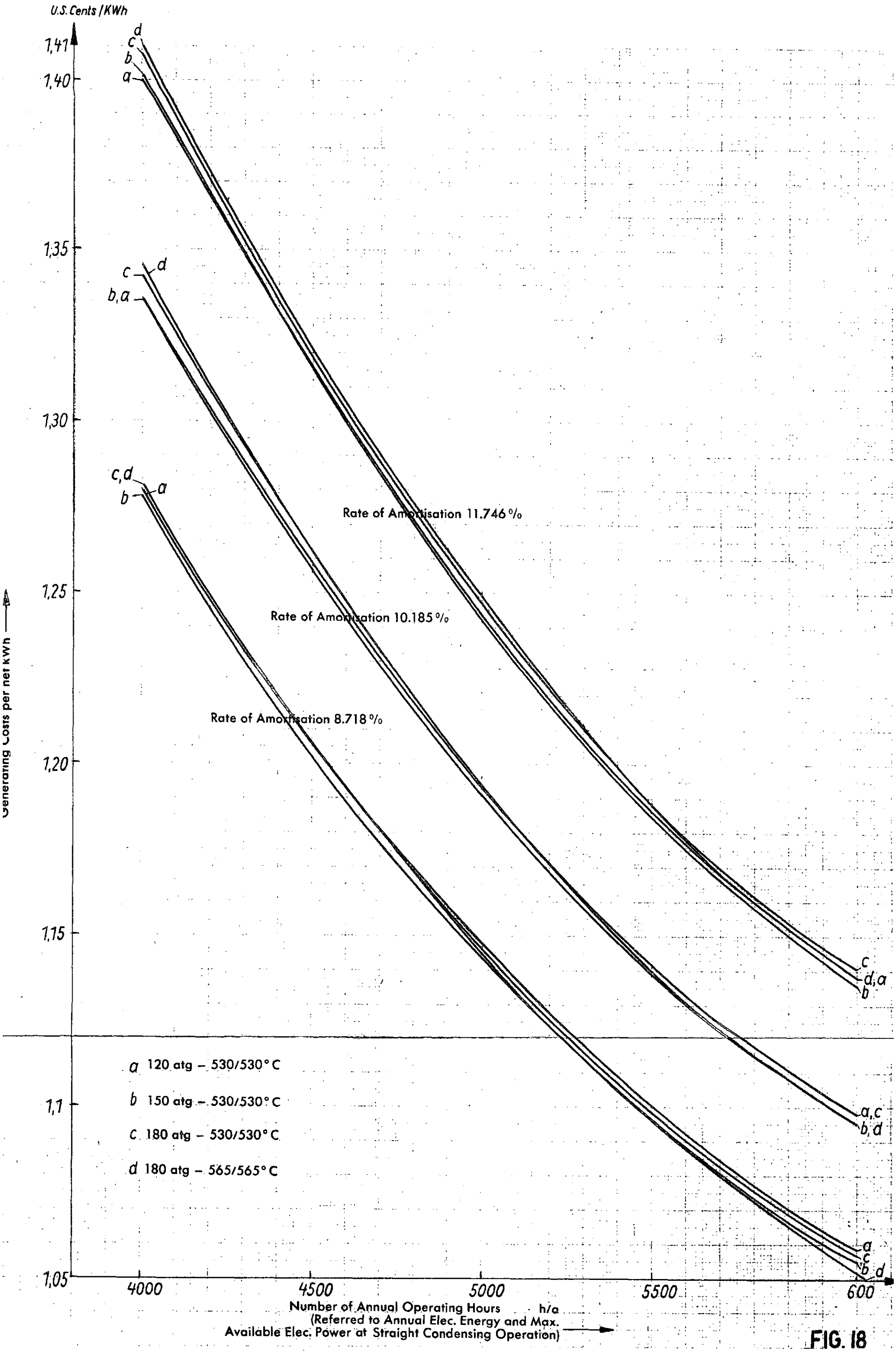
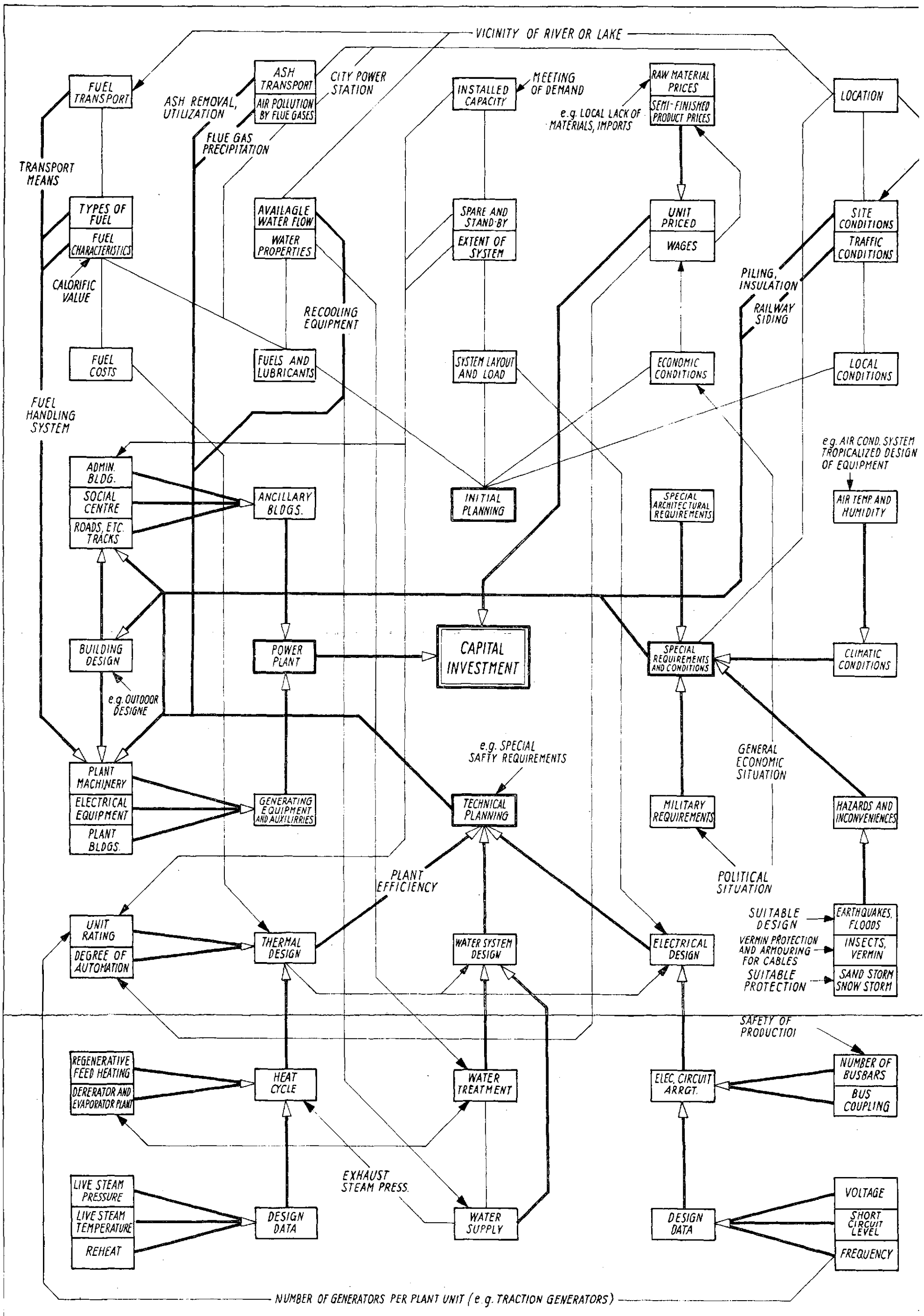


FIG. 18

AVERAGE ANNUAL GENERATING COSTS PER kWh



FACTORS AFFECTING THE CAPITAL INVESTMENT

FIG. 19

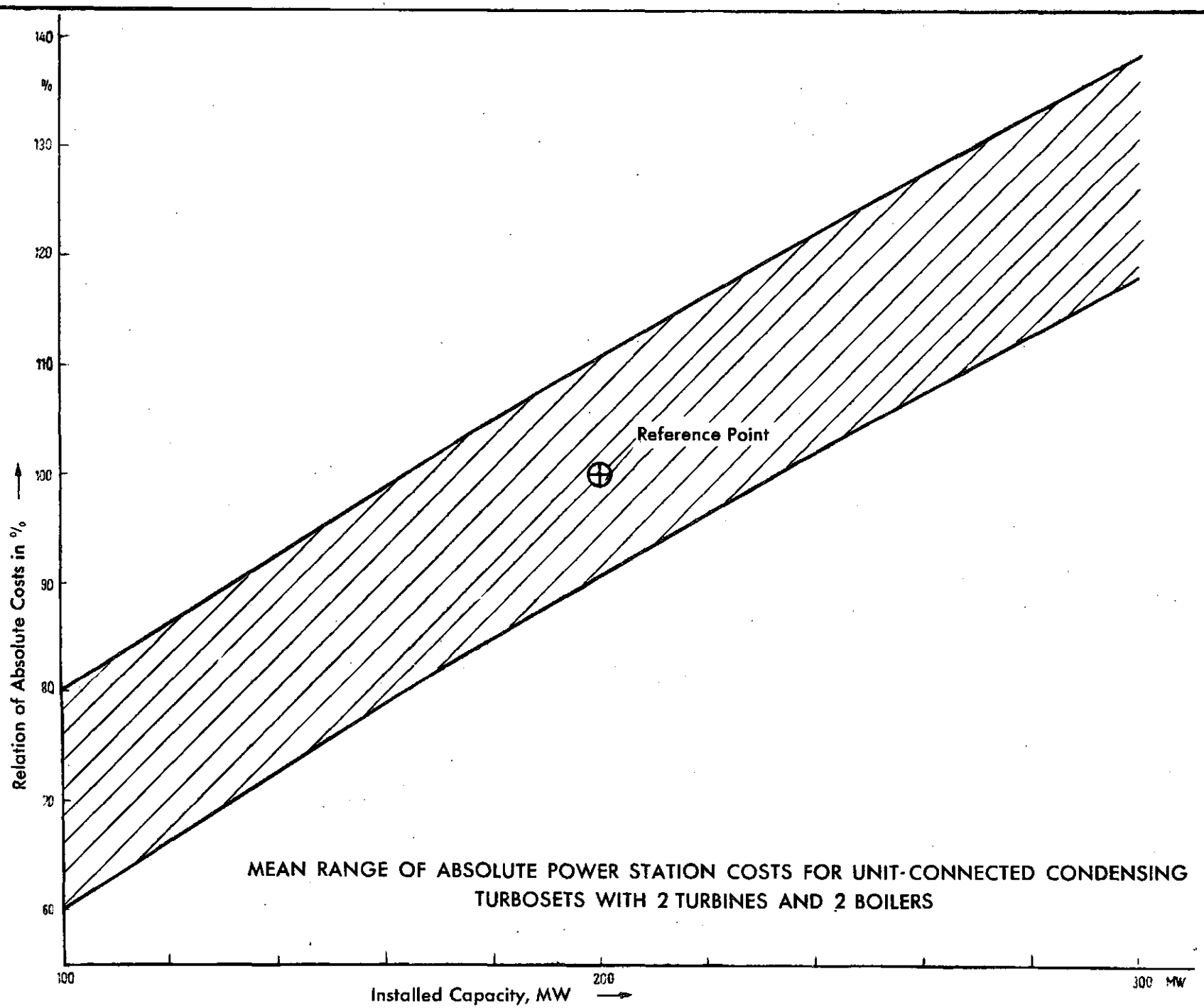

140
%
130
120
110
100
90
80
70
60
Relation of Absolute Costs in % →
Reference Point
MEAN RANGE OF ABSOLUTE POWER STATION COSTS FOR UNIT-CONNECTED CONDENSING TURBOSETS WITH 2 TURBINES AND 2 BOILERS
100
200
300 MW
Installed Capacity, MW →

FIG. 20