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SUMMARIZATION OF STUDIES ON NUCLEAR HEAT ENGINE SYSTEMS

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1.0 INTRODUCTION

Knowledge of the parameters affecting heat engine systems is required in order to arrive at optimum nuclear powerplant designs for given applications. It is necessary to know the interrelation of these parameters with the nuclear parameters in order to establish realistically optimum designs for the overall plant. Work conducted at the University of Michigan under the sponsorship of the Chrysler Corporation relating the heat engine and nuclear parameters into overall systems analyses is summarized in this report.

The development of relatively lightweight, compact, mobile nuclear powerplants is a field offering considerable promise for the future, but one which has received proportionately little attention. This report considers primarily plants of this type. A preliminary investigation has been conducted of the various conceivable heat engine types as components of nuclear plants. Comparative efficiency data under conditions of similar inlet and outlet temperature have been developed. Some preliminary evaluations of the probable costs and weights of the various types have been made. In addition, economic optimizations under the special operating conditions of a nuclear powerplant have been made for several of the major components under varying conditions of working fluid, temperatures, pressures, and power outputs.

To bring the work to a more significant conclusion than is presently possible, it would be necessary that the various optimized components be considered in overall powerplant systems, so that cost and weight analyses of the systems could be computed and comparisons afforded between the various heat engine arrangements, each under its optimized conditions.

2.0 HEAT ENGINE SYSTEMS CONSIDERED

In a nuclear powerplant, the nuclear reactor represents a heat source, similar as far as the heat engine system is concerned, to a boiler or combustor. To provide a complete powerplant, it is necessary that the reactor be accompanied by a heat engine cycle comprising the prime-mover, components and a heat sink. Commercial practice has indicated the suitability of certain types of heat engine systems for certain specific applications and power ranges. For example, the superheated steam turbine plant has been found advantageous for cases of high power output where weight is not a primary factor. For smaller outputs where weight is of only moderate importance, the diesel engine is indicated because of its high thermal efficiency. For cases where weight is of prime importance, as in aircraft, the gas turbine cycle or jet engine seems particularly advantageous. It is not necessarily true that under the special conditions of a nuclear plant the same limitations will apply. It is necessary that the various competing types of heat engine systems be considered without prior prejudice along with systems which do not presently show commercial feasibility. Such a basic preliminary evaluation has been made as the initial phase of this investigation.

Some of the factors in which the nuclear and conventional fossil-fueled plant differ are:

1. In a nuclear plant it is necessary to prevent the escape of fluids which may have become radioactive through proximity to the fissioning fuel.

2. In nuclear plants it is necessary to consider the effects on nuclear reactivity and stability of the fluid which extracts the heat of fission from the core. This fluid may be either the heat engine working fluid or an intermediate heat transfer medium.
3. The heat source in a nuclear plant does not impart combustion products to heat exchange or working fluids, thereby eliminating this source of chemical corrosion. In some proposed systems, however, the working fluid might contain fission products.

The emphasis of the present investigation is upon the suitability of the various heat engine systems themselves for nuclear powerplants. Hence, no attempt was made to consider in detail the transfer of the fission heat to the heat engine working fluid. Under these conditions the various types of nuclear steam plants, for example, are considered similar insofar as their limiting temperatures and required power outputs are similar.

The heat engine types which have been investigated are:

1. Steam systems - saturated, superheated, and supercritical.
2. Rankine cycles using fluids other than water, including combination cycle employing two or more fluids.
3. Gas turbine cycles - open and closed, regenerative, non-regenerative, air and other gases as working fluids.
4. Combination vapor-gas cycles.

After the completion of the initial phases of the investigation, it appeared that the gas turbine cycles, utilizing the closed-cycle arrangement, were of especial importance from the viewpoint of compact, lightweight, high performance plants. For this reason, special emphasis was placed upon the investigation of plants of this type.

3.0 GAS TURBINE CYCLES

3.1 General Limitations

Following a preliminary investigation the gas turbine system appeared especially promising from the viewpoint of an economically feasible nuclear powerplant of moderate output and minimum size and weight. It appears possible to delineate some of the conceivable gas turbine arrangements which are particularly favorable in the nuclear application.

For minimum size and weight and maximum efficiency, the gaseous working fluid should be used directly as the heat extractant fluid in the reactor. This would make an intermediate coolant loop unnecessary. However, in this case, as well as in some of the cases using an intermediate loop, the gas turbine working fluid would be exposed to high level radiation. Thus, a gas exhibiting minimum induced radioactivity is desirable, and in most applications it is essential that the system be closed. Consequently, the investigation has been concentrated on the closed-cycle systems. With such systems gases other than air are possible. Three gases exhibiting a wide range of molecular weight and ratio of specific heats have been included: helium, air, and carbon-dioxide. The trends exhibited by these gases can be applied to others if consideration is given to the differences in molecular weight and ratio of specific heats.

In general, the possible gas turbine cycles can be divided according to those which utilize regeneration and those which do not. It is possible to obtain a relatively high thermal efficiency with either type of cycle, and in fact either is used in certain conventional applications. The non-regenerative cycle is commonly employed, for example, in jet and turboprop aircraft engines while the regenerative arrangement tends to be favored for moderate power applications where weight is not of such overriding importance. To achieve high thermal efficiency with the non-regenerative cycle it is necessary that the pressure ratio be high. Hence, turbine and compressor efficiencies must be excellent for satisfactory performance. For this reason, the range of feasible operation may be limited. By additions of weight, it is possible to achieve equal or superior efficiency with the regenerative cycle at low pressure ratio. This arrangement is less sensitive to turbine and compressor efficiency so that a broader operating range becomes possible.

The low pressure ratio of the regenerative cycle is particularly advantageous for a closed-cycle in that it allows a high mean fluid density, considering a practical limitation to the maximum pressure and temperature. Thus, the machinery sizes and weights for a given output are minimized. The investigations have been concentrated on the regenerative cycles.

3.2 Scope of Optimizations Conducted

Preliminary operational optimizations of the various components comprising the complete nuclear gas turbine plant have been completed, as well as complete plant optimizations in some cases. It would be most desirable if these various component optimizations could be integrated into complete plant designs so that the overall effects on weight, size, and cost of power from a variation of the different independent parameters could be evaluated.

The studies completed to the present time have included the following:

1. Effects upon thermal efficiency of variations in system arrangement, temperature limits, pressure limits, working fluid, and component efficiencies and effectivenesses.
2. Approximate determination of economically attainable turbomachinery component efficiencies as affected by power level, pressure level, working fluid, and temperature limitations.
3. Approximate economic optimization of the regenerator component for specific assumptions regarding type of surface. The cost of heat from a nuclear reactor was compared with the capital cost amortization and costs of pumping work for the regenerator to determine the optimum design point for the regenerator, for various conditions of temperature, pressure, working fluid selection, and power level.
4. Approximate economic optimization for a single type of gas-cooled reactor, the reactor core considered as a heat exchanger. The optimum balance between the cost of heat, the cost of pump work to force the fluid through the reactor, and the amortization of capital cost was considered, as affected by the maximum allowable reactor temperature, the pressure level, the power output, and the selection of working fluid.

3.3 Recommendations for Additional Studies

Rather modest additional studies to bring the work accomplished to date to more significant conclusions would include the integration of the various optimized components into overall powerplants so that meaningful statements regarding the effect on power cost and plant size and weight of various changes in the dependent variables of pressure and temperature limitations, power level, and selection of working fluid could be made. This would not require a major additional effort but has not been accomplished to date.

3.4 Optimizations Accomplished

1. Thermal Efficiency for Arbitrary Component Efficiencies

The thermal efficiency of gas turbine cycles as it is affected by the ratio of specific heats of the working fluid, the cycle temperature limits, the pressure ratio, the system arrangement, the parasitic pressure drops, and the component efficiencies and effectivenesses has been evaluated and is presented in Reference 1. Thermal efficiency is plotted against pressure ratio in a series of curve sheets, while the additional parameters listed above are varied separately. From these data, it is possible to estimate the thermal efficiency of a gas turbine system for any combination of the independent parameters. The applicable curve sheets (Figures 1 through 14) are reproduced in this report for convenience. Some additional curves not included in Reference 1 have been added. The assumed component efficiencies are given in Table I.

If the component efficiencies and effectivenesses are considered as arbitrary assumptions, as in this case, the thermal efficiency is not a function of the pressure level or of the molecular weight of the working fluid. It is, however, a function of the ratio of specific heats. It is shown in Reference 1 that the thermal efficiency at the optimum pressure ratio decreases slightly for higher values of the ratio of specific heat. Since such values are characteristic of monatomic gases, there is a slight disadvantage to helium, for example, with respect to air if equal component efficiencies, temperature limitations, and cycle arrangement are assumed. The disadvantage becomes less for higher temperature ratio cycles.

It is also shown in Reference 1 that the optimum pressure ratio, from the viewpoint of thermal efficiency, decreases markedly with an increase in the ratio of specific heats. Thus, for a highly regenerative cycle, it is of the order of 3 for air and 2 for helium. For this reason, even though the pressure ratio per stage attainable with a low molecular weight gas such as helium is considerably less than it is for air or carbon-dioxide, the discrepancy in required number of stages is reduced by the reduced overall pressure ratio requirement. If, as would be indicated from purely fluid dynamic considerations, it were possible to maintain a constant Mach Number design between air and helium, for instance, then the number of turbomachinery stages for the helium unit would be less than for the air machine because of the reduced overall ratio requirement. Also, as shown in Reference 1, (see also Figure 16 of this report) the flow areas would be less and hence the helium machines would be substantially smaller. Such a design may be possible for the compressor. However, it is unlikely for the turbine because of the limitation on wheel speed due to centrifugal stress.

2. Turbomachinery Component Efficiencies

The efficiency levels which may be reasonably anticipated for the turbomachinery components of a gas turbine cycle are a function of the power level, the pressure level and ratio, the temperatures, the working fluid, and the degree to which cost and weight may be sacrificed in the interest of higher efficiency. With regard to the latter consideration, a small change in the turbomachinery component efficiencies is reflected by a rather large change in the overall plant thermal efficiency. Also, in most cases the turbomachinery units of a closed-cycle nuclear plant comprise only a small portion of the overall plant cost and weight. Hence, very little sacrifice of component efficiency can be justified on this basis.

The attainable turbomachinery efficiencies can be estimated by a knowledge of the type of machine involved (axial, centrifugal, or positive displacement), the volume flow rate and required pressure ratio, the flow Reynold's Numbers within the machine, and the Mach Numbers (or required pressure ratio per stage). To obtain an approximate estimate of efficiency, it is then necessary to compare with the known efficiency of air-handling machines of comparable size, and consider correction factors based upon the changes in Reynolds' Numbers and Mach Numbers from the conventional air machine. Such a procedure has been carried through over a large range of power outputs, temperature levels, and working fluid selections. The details of the procedure are given in Reference 1.

The resulting turbomachinery efficiency estimates for the working fluids helium, air, and carbon-dioxide, over a wide range of turbine power (not plant output) and inlet temperature to the turbine, are included in Figures 17 through 25 of this report. It is assumed that the given efficiencies are mean values to be applied to compressor and turbine alike in the calculation of the overall plant efficiency. Since the turbine efficiency is usually slightly greater than that of the compressor for the same plant, it is presumed that the turbine efficiency would be generally in excess of the value given and the compressor efficiency somewhat less. It is noted that the efficiency decreases for an increase in pressure and/or a decrease in power. These factors are a result of the reduced flow path dimensions which more than overcome the increased Reynolds' Number at higher pressure.

Figures 26 through 34 show the estimated turbine wheel tip diameters for the various cases.

3. Regenerator Optimizations

Of the components which together comprise a closed-cycle gas turbine plant those associated with heat transfer include the major portion of the cost and weight for the entire plant. The most important of these components in a highly regenerative cycle are the regenerator and the nuclear reactor, or the heat-source exchanger if there is an intermediate coolant loop. Consequently, preliminary attempts have been made to locate under certain specialized assumptions the economically optimum design points for each of these units. A wholly comprehensive study of this sort would be extremely tedious. However, the preliminary studies reported here do illustrate the interrelations of the various significant parameters and delineate the significant trends.

The results of the regenerator optimizations are given in Reference 2 and the controlling assumptions explained in some detail. A heat exchange surface of a specific type was assumed as a basis for the calculations. This was the "all-prime" surface of the Griscom Russell Company, which has been developed specifically for the closed-cycle gas turbine application. Preliminary evaluations have indicated that an extremely compact surface is desirable from the viewpoints of both cost and weight. It is not meant to imply that the cost of the first unit utilizing a special surface of this type would be less than that of a competitive shell-and-tube design. However, the eventual cost, once the developmental cost has been assimilated, seems very favorable. The significant features of the surface arrangement selected for this evaluation are the very small passages accompanied by a sufficient degree of ruggedness to withstand high temperature, high pressure gases, and the suitability to eventual low cost production. With the closed-cycle plant it is possible to use small passages since the gas is clean in that no combustion products are involved. A schematic representation of the surface arrangement selected for this study is shown in Figure 35.

The process of economic optimization of the regenerator of the closed-cycle nuclear gas turbine powerplant is one of delineating that design configuration, considering the assumed surface arrangement which will minimize the operating cost of the unit. The operating cost is considered to include amortization of capital cost (at a rate consistent in this case with utilities' practice), pump work to force the gas through the unit (considering the cost of heat and the thermal efficiency of the nuclear plant), and heat saved as a negative cost. The effect of increased compressor size required for increased pump work was also considered as a capital cost item. The thermal efficiency estimations for the optimization were based on the regenerative cycle shown in Figure 1. The turbomachinery component efficiencies used were estimated as previously described, and are shown in Figures 17 through 25 .

The resulting designs for both helium and air over a wide range of plant outputs and temperature limitations are shown in terms of regenerator cross-sectional area and volume in Figure 36 through 47. The areas shown are the total areas, including both hot and cold passages and the separating metal. It should be noted that in some cases the optimization leads to impractical pancake-shaped designs in which the length to diameter ratio of the unit is so small that the inlet manifolding would require considerably greater investment than the heat transfer area itself. This effect was not considered in the optimizations. It is noted in general that the size of the regenerator decreases markedly for increased pressure level at a given output.

Figures 48 through 71 show the film coefficients and fluid velocities for both the hot and cold sides. Some of these are repeated from Reference 2 for convenience. It is noted that the economically justifiable film coefficients increase considerably with pressure level although the values are relatively modest: up to about 100 BTU/hr-ft^2-°F for air and only about 20 for helium. This somewhat surprising result is because the regenerators are considerably smaller for helium than for air so that the saving in capital cost effected by increasing the film coefficient is relatively not great. The velocities for both helium and air are low--less than 100 feet/second in all cases. Neither optimized film coefficients nor velocities are substantially affected by the power level.

Figures 72 through 74 show the Reynold's Numbers for the gas flow in the regenerator. It is noted that in all cases the Reynolds' Number for the hot side is less than the turbulent-laminar transition value. However, air for the cold side shows Reynold's Numbers in the transition region and beyond.

The regenerator effectivenesses for air and helium over a large range of output power, pressure level, and turbine inlet temperature are plotted in Figures 75 through 80. These curves were also shown in Reference 2 but are presented here for convenience. It is shown that the highest effectivenesses are economically justified for the higher pressure levels and power outputs. The variation with power output is not great. Effectiveness variation with pressure level is considerably more substantial, varying from a maximum of near 0.90 for either air or helium at 1000 psia maximum pressure to approximately 0.55 for air at a pressure corresponding to an open regenerative cycle. The variation with pressure for helium is not so great but is still significant.

4. Overall Plant Thermal Efficiencies

It is possible to evaluate the overall plant thermal efficiency based upon the turbomachinery component efficiency estimations and the regenerator optimizations. Such estimates can be made for either air or helium from the available data over a wide range of power outputs, cycle temperature limitations, and pressure levels. Since the regenerator optimizations did not include carbon-dioxide, it is not possible to consider this gas in the overall efficiency estimates. Preliminary evaluations have shown, however, that the results would be quite close to those for air.

For the thermal efficiency estimations for air and helium, it is necessary to include an arbitrary assumption regarding the parasitic pressure losses in those cycle components which have not been investigated in detail, i.e., ducting, intercooler, precooler, and, most important, the reactor. Preliminary evaluations of all these have shown that the assumed pressure ratio of 1.07 times the turbine expansion ratio was probably conservative. Since this ratio is a fixed assumption for all the cases investigated, the results do not reflect adequately the relative advantages of some of the gases with respect to low frictional pressure losses.

The calculations were all based on a single overall cycle pressure ratio (3.0) which was selected as a reasonable compromise between the optima for the various cases. Corrections for the individual cases have been made. Thus, the plotted thermal efficiencies apply to that overall cycle pressure ratio which is optimum for the particular case.

The resulting curves are shown in Figures 81 through 86. As is well recognized, the efficiency increases very strongly with an increase in the overall cycle temperature ratio. Also, it increases with the power output, principally because of the improved turbomachinery component efficiencies at greater outputs. The effect of pressure level is not great. For air, the maximum efficiencies occur for intermediate pressure levels, with the lowest and also the highest maximum cycle pressures investigated (45 and 100 psia respectively) showing a decrease. For helium, the efficiency decreases continuously for an increase in pressure level. These facts are the results of conflicting trends: the turbomachinery component efficiencies are adversely affected by high pressure because of the reduced flow path dimensions, and the economically optimized regenerator effectiveness increases with increased pressure because of the reduced size and capital investment necessary to achieve a given effectiveness. In general, the overall effect of pressure on the thermal efficiency is only a matter of a few percentage points.

In general, the helium thermal efficiencies are slightly less than those for air. Again, it is only a question of a few percentage points.

Even though these results show some variation in thermal efficiency between pressure levels and gases, it is not possible to state that there is any positive advantage of one case over another. In order to make such a statement, it would be necessary to make preliminary cost estimates of the overall plants and then optimize, balancing the amortization of capital cost against the fissile material burn-up cost. On such a basis, it is quite likely that the highest thermal efficiency would not prove to be the economic optimum.

Table II presents a summarization of the applicable data for 20,000 horsepower plants for both air and helium as working fluid.

5. Gas-Cooled Reactor Optimization

It is to be expected that the economic optimization of a reactor core as a heat exchanger will show optimum heat transfer coefficients differing considerably from those of the regenerator, for example, because of the large capital cost variations associated with changes in the fissile material inventory as well as in the other significant reactor parameters. For instance, there is a substantial feed-back into the critical mass requirement inherent in a change of core dimensions. In general, it would be expected that the effective cost of heat transfer surface within the reactor would be much greater than that in a conventional heat exchanger so thatlarger film coefficients and pumping losses would be justified for the reactor than for the regenerator. This would be the case only if heat transfer area rather than criticality requirements were controlling. A general investigation of the situation would be extremely difficult because of the many types of reactor cores to be considered and the unknown magnitude of many of the most important cost factors. A preliminary investigation has been completed for a specific configuration to attempt to delineate to some extent the significant trends.

The reactor investigated is of the liquid-metal-fuel type employing solutions or slurries of highly enriched uranium in bismuth. No attempt to investigate the mechanical feasibility of the system has been made or to determine its relative advantages or disadvantages as compared with other systems. The system has merely been assumed to provide a basis for the optimization of the core from the viewpoint of heat transfer. A cylindrical graphite core of height equal to diameter has been assumed. It is fitted with vertical wells for the liquid metal fuel and holes for the passage of the coolant gas, which is also the gas turbine working fluid. Steel liners were assumed for the gas tubes and provision was taken in the nuclear calculations for the presence of the steel. The very approximate criticality estimates were based on the work of J. Chernick, Reference 3. The procedure and results are related in some detail in Reference 2. The significant features are repeated here for convenience.

It has been assumed that the temperature limitation for the overall plant is in the reactor rather than the gas turbine equipment. Thus, a maximum reactor temperature (1700°F) was assumed. This value would seem to be within the realm of possibility within the next few years and would provide gases to the turbine at a temperature level not too excessive. The question is one of the economically optimum degree of approach of the gas temperature leaving the reactor to the maximum assumed reactor temperature. If the approach is close, the required heat transfer area will be large and the capital cost increased. On the other hand, due to the large dimensions, the uranium inventory may be decreased because of a smaller critical mass. Also, the thermal efficiency of the plant will be maximized, so that the uranium burn-up cost will be minimum. If the degree of temperature approach to the reactor maximum is not close, the reverse is the case. On this general basis it is possible to optimize the core.

As with the regenerator, the optimization attempted to delineate the minimum operating cost. This cost is a function of the following factors:

i. Uranium Burn-up (assumed cost of $20/gram of U-235). This is affected by the plant thermal efficiency which is in turn a function of degree of approach of maximum gas temperature to reactor temperature, pumping loss in the reactor, and the remaining cycle components. These were evaluated according to the optimizations already described.

ii. Uranium Inventory. This is affected by criticality considerations which are a function of the core dimensions. For larger cores, up to a point, the critical mass is decreased so that the inventory costs are reduced.

iii. Capital Cost Amortization. Very rough estimates of the reactor capital cost were made. It seemed reasonable that the cost should increase both with an increase in core volume and power level. An increase in core size at a given power level would increase the required shielding as would an increase in power level for a given volume of core. It was assumed for estimating that the reactor cost was proportional to the square root of the product of power level and volume of core. In addition, it was assumed that the cost for a given power and core volume would be greater for higher gas pressures. An increase of the order of 20% was assumed between 400 and 1000 psia.

In addition to the above, an instrumentation cost was assumed proportional to the power level.

All reactor costs were based upon rough estimates for the Brookhaven National Laboratory's LMFR designs (Reference 4).

The reactor core optimization calculations to the present time have considered only air as the working fluid.

The results of the calculations are shown in Figures 87 through 91, showing turbine inlet temperature, reactor core log mean temperature difference, reactor core heat transfer gas film coefficient, gas temperature increase in the reactor, and reactor core diameter.

It is noted from an examination of any of the curves that for small outputs the reactor criticality and uranium inventory considerations are controlling and prevent a reduction of the core dimensions to the point indicated from a pure heat exchange viewpoint. For this reason, the approach between gas and reactor temperature is very close and the film coefficient small. For large outputs, the heat transport and transfer problems along with thermal efficiency become controlling so that it is necessary to increase the reactor size considerably (more so for low pressure than high). Even so, the velocities and film coefficients become large (up to the order of 500 BTU/hr.ft.^{2}°F). As expected, these are much larger than the optimized coefficients for the regenerator because the capital cost penalty of increased flow and heat transfer area is much greater.

The frictional pressure ratios through the reactor are listed in Table III. It is noted that even for 60,000 horsepower plant output, the ratio ranges only between 1 and 3% as the pressure is decreased from 1000 to 400 psia. For large outputs of this range, the use of the open cycle pressures would be prohibitive from this viewpoint. However, for a high-pressure, closed cycle it is noted that outputs of this order are feasible for a direct-cooled core. The plant thermal efficiencies are not given. However, they would be in excess of 40% since the turbine inlet temperatures are in all cases greater than 1600°F and the parasitic pressure drops within the thermal efficiency range estimations previously given in this report. For these maximum power cases, the core diameter is only of the order of 6 feet.

In the intermediate power range the heat transfer and transport capabilities of the reactor are not limiting. Since the capital cost was assumed to increase with increases of both power and size, the economic optimum shows a decreased core size to alleviate the effect of increased power. The increased inventory cost is not sufficient to overcome this trend.

4.0 HEAT ENGINE SYSTEMS OTHER THAN GAS TURBINE

In order to make a realistic appraisal of the suitability of various heat engine systems for a given nuclear application it is necessary that the optimum arrangements for each cycle be compared on a common basis. If the final optimization of the various cycles is to be upon a basis of power cost, then the minimum power costs possible for each of the various heat engine systems must be compared. A similar procedure must be followed if it is desired to optimize instead with respect to weight or size. The results will differ, of course, depending upon the objectives of the optimization.

After a preliminary evaluation of the various possibilities, it appeared that a directly-cooled reactor with a closed-cycle gas turbine system offered low power costs as well as compactness and light weight, particularly in the power output range up to about 30 megawatts. The possibility of low-cost power appears to exist with this combination once the developmental costs have been assimilated since the required machinery is of small size and does not appear excessively complex. A very preliminary cost estimate, bearing out this conclusion, is presented in Reference 1.

To arrive at these preliminary conclusions regarding the suitability of the closed-cycle gas turbine systems in the small to moderate power range, it was necessary to evaluate in an approximate manner the alternative systems. It is not meant to imply as a result of these preliminary evaluations that the gas turbine systems are necessarily superior in all respects to the alternatives even in the low power range. An attempt has been made, however, to delineate some of the applicable parameters necessary to form a preliminary judgment. A further comparison between the overall optimum systems of the various types would be necessary.

4.1 Steam Powerplants

1. General Limitations

The steam plant, for given temperature limitations, is restricted in its pressure levels by the saturation properties of water. In the case of the closed-cycle gas turbine there is no such restriction since the working fluid is restricted to the gaseous phase. Thus, the working fluid density at any point of the cycle is a dependent variable with a steam plant so that the volumetric flow rates, for small outputs, may become inconveniently small. For this reason, it does not appear possible to design efficient steam plants for small output and high temperature (which automatically implies high pressure). With the closed-cycle gas turbine the design pressure level can be reduced if desired so that a reasonable flow path design can be achieved for any output. It appears, however, that the steam system definitely is competitive for outputs in excess of 5000 to 10,000 horsepower.

The steam cycle, at least for saturated conditions and conditions of moderate superheat, approaches the ideal Carnot cycle quite closely. Hence, the thermal efficiencies attainable, if the flow rates are large enough to allow reasonable component efficiencies, are high even at low inlet temperatures. The anticipated thermal efficiencies for large nuclear steam plants as a function of maximum cycle temperature, based upon operating performance of existing fossil-fueled plants, is shown in Figure 92. This figure is repeated from Reference 1 for convenience. Also included is the anticipated thermal efficiency for gas turbine plants of similar size, extrapolated from the previously discussed gas turbine cycle optimizations. At any temperature up to 1500°F the steam cycle appears superior in thermal efficiency although the proportionate difference becomes relatively slight for larger temperatures. It is usually considered in commercial practice that gas turbine plants become economically competitive with steam plants if the temperature available to the gas turbine is at least 1200°F. Commercially, steam plants to the present time have been limited to about 1150°F. As the maximum design steam temperature is increased, the pressure level must also be increased to allow an efficient cycle. As temperatures greatly in excess of the critical water temperature of 705°F are considered, the degree of similarity between the steam cycle and the Carnot cycle decreases, so that the proportionate gain in efficiency is less. For this reason, it would appear that if temperatures on the order of 1500°F could be made available from a nuclear heat source, the commercially feasible efficiency of a gas turbine plant would exceed that of a steam plant, particularly for moderate to low outputs. It is obvious from an examination of Figure 92, however, that a gas turbine temperature of about 1500°F is necessary to equal the thermal efficiencies for large steam plants at about 900°F. However, since the capital cost amortization represents a very large portion of the cost of power from nuclear plants, it is necessary to compare the capital costs of optimum gas turbine and steam plants to determine their relative economic suitabilities in a given application. This has not been done in detail, although a preliminary estimate as reported in Reference 1 did show a probable future advantage to the closed-cycle gas turbine in this respect. The present economic advantage appears to be with the steam plant, although there is substantial cost and weight disadvantage.

2. Cost, Weight, and Efficiency Estimates

As in the case of gas turbines, it would be expected that the efficiency of steam plants would decrease with reduced temperature ratio across the cycle and also with lower power output because of the adverse effect of reduced flow path dimensions on component efficiency. An approximate estimate of these effects is given in Figure 93. The sources of the data are previous estimates and studies by manufacturers as stated on the curves.

Figure 94 shows estimated costs of equipment for steam powerplants in the 5000 to 200,000 horsepower range as a function of maximum cycle steam temperature. Each power level shows a temperature corresponding to minimum equipment cost. This temperature represents a balance between two opposing trends: on one hand, increased costs of material and other refinements required by higher temperature and pressure, and on the other hand, increased costs resulting from greater size required to handle a low density fluid at very low temperature and pressure. The temperature which represents the minimum cost of equipment increases as the power output is increased. Thus, a higher temperature could presumably be utilized economically for large output plants. The sources of the data are stated on the curve sheet. If power production cost is considered, the optimum temperature will be higher. Higher temperatures mean higher thermal efficiencies, which would reduce uranium burn-up costs.

4.2 Rankine Cycles for Fluids Other Than Water

1. Thermodynamic Characteristics

A pure Rankine cycle represents a very close approach to the ideal Carnot cycle. However, because of the physical properties of water it is not possible to consider a steam Rankine cycle except for maximum temperature considerably below 705°F, the critical temperature of water. Hence, the use of fluids with critical temperature higher than that of water is suggested in order to take full advantage of the thermal efficiency capabilities of the Rankine cycle when a source of high temperature is available. The thermodynamically ideal fluid in this respect would be one possessing a high critical temperature, not inconveniently high pressure in the range of maximum cycle temperature, and still substantial pressure in the range of the heat sink temperature where condensation would occur. No single fluid possessing these capabilities is available.

Various of the liquid metals possess suitable high temperature properties. Mercury has a convenient saturation pressure in the range of 1200-1300°F although it rises precipitously for higher temperature. Sodium has a saturation pressure of the order of 15 to 30 psia in the range between 1500 and 2000°F and might thus be desirable if a temperature source in excess of 1500°F were available. Intermediate between mercury and sodium with respect to the pressure-temperature relation are potassium and rubidium. Zinc is quite similar to sodium with a slightly reduced vapor pressure at a given temperature. Sodium, potassium, and zinc appear desirable in their large latent heat of vaporization which exceeds that of water. A list of some of the significant physical properties is included in Table IV.

If these fluids were to be used as reactor coolant fluids in a liquid metal boiling reactor, the thermal neutron absorption cross-section would be of importance. It is noted from an examination of the table that the cross-sections are fairly reasonable with the exception of mercury which has a thermal cross-section of 380 barns. Hence, a mercury-cooled reactor would necessarily be of the fast rather than thermal type.

None of the fluids discussed exhibits sufficient vapor density at heat sink temperature to allow a feasible turbine design for large power output. For this reason, it is necessary to consider a binary cycle wherein the liquid metal would be condensed at a relatively high temperature so that a reasonable turbine design would be possible. The condensation heat from the liquid metal would be used as the heat source for a steam plant operating between the liquid metal condensing temperature, which would be suitable to a steam Rankine cycle, and the steam condenser. Such binary plants using water and mercury as working fluids have been in operation as fossil-fueled central station units for some years and have exhibited extremely high thermal efficiency.

If the high temperature working fluid were to be sodium, it might be necessary to consider a trinary cycle of perhaps sodium-mercury-water. This arrangement is necessitated by the fact that a minimum feasible condensing temperature for the sodium appears to be in the range of 1000-1100°F. Such a temperature is inconveniently high for a Rankine-type steam cycle and would necessitate large thermodynamic irreversibilities. A further possibility would be a sodium-air cycle wherein the condensing sodium would be used to power a gas turbine cycle.

Figures 95 through 99 are the approximate temperature-entropy diagrams for cycles of these types. The pertinent characteristics are summarized in Table V. Calculations with ideal and also realistic component efficiencies are included in the table. For comparison, estimated efficiencies for a gas turbine plant and a supercritical steam plant operating with the same maximum (1500°F) and minimum temperatures are shown. It is noted that the binary and trinary cycle efficiencies show an improvement of about 10 percentage points (about 20%) over the steam cycle and 15 percentage points over the gas turbine.

There is some doubt regarding the thermodynamic performance of sodium because of dimerization. It seems possible that equilibrium conditions will not be attained on the passage through the turbine. Estimates have shown that the likely variation of thermodynamic properties on this account are not sufficient to change greatly the expected thermal efficiency.

Preliminary investigations have also been conducted to determine the feasibility of a sodium turbine. It was found that the fluid properties were not particularly unfavorable.

2. <u>Economic Feasibility of Binary Cycles</u>

The economic feasibility of nuclear binary or trinary cycle plants is a question not only of thermal efficiency, in which they excel, but also of operating expenses and capital cost amortization. It should be mentioned that little development of this type of cycle for fossil-fueled plants has been evident in recent years. Apparently the operational difficulties and the high capital cost more than overcome the thermal efficiency advantage, at least for conventional-fueled plants.

With respect to nuclear plants at the present time and in the future, the situation may be changed in the following particulars:

<u>i.</u> If the temperatures in excess of 1500°F become possible the steam cycle, which is of necessity supercritical, becomes especially

unattractive, particularly in the range of low to moderate output. This is because of the very small volumetric flows and excessive pressures. The high temperature liquid metal cycle could be designed for such a temperature to require only moderate pressure and reasonable flow rates.

ii. There have been great improvements in the technology associated with the handling of liquid metals in recent years due to the research and development efforts in connection with their use as reactor coolants.

For these reasons, it appears that a further examination of the possibilities of such plants is warranted.

A very approximate investigation for a plant of about 30,000 horsepower outpu has been conducted. It was indicated that the likely capital equipment cost for a sodium-boiler reactor combined with a sodium-mercury-steam plant would be too large to be overcome by the thermal efficiency advantage of such a cycle. This is due to the requirements for special materials and the fact that the density of the working fluid in the sodium portion of the cycle is very low. Thus, the equipment is similar in size to the low pressure stages of a steam turbine unit. These results are reported in Reference. 1.

5.0 CONCLUSIONS FROM STUDIES

An investigation has been made on a very broad front covering various conceivable heat engine systems, working fluids, temperature and pressure ranges, and power outputs from a few hundred to approximately 60,000 horsepower. Much detailed information regarding the various systems has resulted. However, it has not been possible as yet to carry the investigation far enough to compare the various systems on a basis of overall cost, overall size, or overall weight.

A comparison between the anticipated thermal efficiencies of the various systems for relatively large output plants is presented in Figure 92. It is shown that the binary liquid-metal-water Rankine cycles allow the greatest thermal efficiency values for a fixed temperature. They are followed by steam and gas turbine systems in that order.

Cost estimates for steam plants for various temperature and power levels are included in the report although no comparable data for gas turbine or binary plant systems are given.

Considerable detailed information is included regarding the optimum design points for the components which together comprise a nuclear gas turbine system. The effects of the selection of working fluid, pressure level, temperature level, pressure ratio, and power output are shown. However, no final conclusions are possible regarding the most suitable overall selections. It is still necessary to integrate the optimized components for various fluids and temperature and pressure levels into overall plants, and then to determine the relative power costs, weights, and sizes.

TABLE I. ASSUMED COMPONENT EFFICIENCIES FOR THE BASIC GAS TURBINE CYCLE

Turbine Efficiency	85%
Compressor Efficiency	85%
Regenerator Effectiveness	93%
Ratio of Compressor Pressure Ratio to Turbine Expansion Ratio	1.07
Cooling Medium Temperature	70 F
Minimum Fluid Temperature	90 F

TABLE II

PERFORMANCE DATA FOR TYPICAL OPTIMIZED NUCLEAR POWERED,
CLOSED-CYCLE GAS TURBINE POWER PLANTS

	AIR PLANT			HELIUM PLANT		
Max. Pressure, psia	1000	400	45	1000	400	45
Max Temperature, °F	1200	1200	1200	1200	1200	1200
Output Power, HP	20,000	20,000	20,000	20,000	20,000	20,000
Thermal Efficiency	0.3348	0.3417	0.2954	0.2734	0.2897	0.3130
Regenerative Effectiveness	0.90	0.87	0.66	0.85	0.83	0.78
Regenerator Volume, cu ft	328.53	385.25	674.68	328.24	354.04	448.77
Regenerator Cross-Sectional Area, sq ft	59.69	105.25	1008.8	176.32	233.41	466.88
Regenerator Length, ft	5.504	3.662	0.669	1.862	1.517	0.957
No. of Tubes in Regenerator	7.324×10^5	1.291×10^6	1.238×10^7	2.163×10^6	2.864×10^6	5.753×10^6
Regenerator Reynold's No.	7,000	4,000	400	400	300	150
Regenerator film Coefficient, Btu per ft^2-hr-°F	135.44	84.53	4.828	18.26	17.71	16.06
Adiabatic Turbine and Compressor Efficiency	0.862	0.876	0.900	0.876	0.885	0.892

TABLE III

BOILING CYCLE FLUIDS

Fluid	M.P. °F	B.P. °F	Sp.G.	Latent Heat of Vaporization Btu/lb	Thermal Neutron Abs. Cross-Section barns
Mercury	-36	675	13.5	125	380 ± 20
Sodium	208	1621	0.8	1810	0.5
Zinc	787	1663	7.14	1645	1.06
Rubidium	102	1270	1.53	880	0.7
Potassium	144	1400	0.87	1790	2.0
Bismuth*	509	2842	9.8	--	.032
Lead*	620	2952	11.35	365	0.17

*Included only for comparative purposes.

TABLE IV. TABULATED EFFICIENCIES OF BINARY AND TRINARY VAPOR CYCLES AT 1500 F INLET, 70 F COOLING WATER (1" Hg cond.)

Cycle Description	T-S Diagram	Efficiency
1) Mercury-Steam, Extraction Mercury Turbine Eff. - .80 Steam Turbine Eff. - .85 Steam Superheated		.588
2) Mercury-Steam, Non-Extraction Mercury Turbine Eff. - .80 Steam Turbine Eff. - .85 Steam Superheated		.550
3) Sodium-Mercury-Steam, Extraction Sodium Turbine Eff. - .80 Mercury Turbine Eff. - .83 Steam Turbine Eff. - .85 Mercury and Steam Superheated	Fig. 95	.600
4) Sodium-Mercury-Steam, Extraction Sodium Turbine Eff. - .75 Mercury Trubine Eff. - .80 Steam Turbine Eff. - .85 Mercury and Steam Superheated	Fig. 96	.588
5) Sodium-Mercury-Steam, Extraction Sodium Turbine Eff. - 1.00 Mercury Turbine Eff. - 1.00 Steam Turbine Eff. - 1.00 No Superheat	Fig. 97	.671
6) Sodium-Steam, Extraction Sodium Turbine Eff. - .80 Steam Turbine Eff. - .85 Steam Superheated	Fig. 98	.518
7) Sodium-Air, Extraction Sodium Turbine Eff. - .80 Air Turbine Eff. - .89 Air Compressor Eff. - .89 Regenerator Effectiveness - .93 Precooler Terminal Δt - 20 F Compression to Expansion Ratio - 1.07	Fig. 99	.486
8) Supercritical Steam Cycle 7000 psig - 1500 F Reference 4		.485
9) Gas Turbine Cycle - 1500 F Regenerator, Reheat, Intercooler		.448

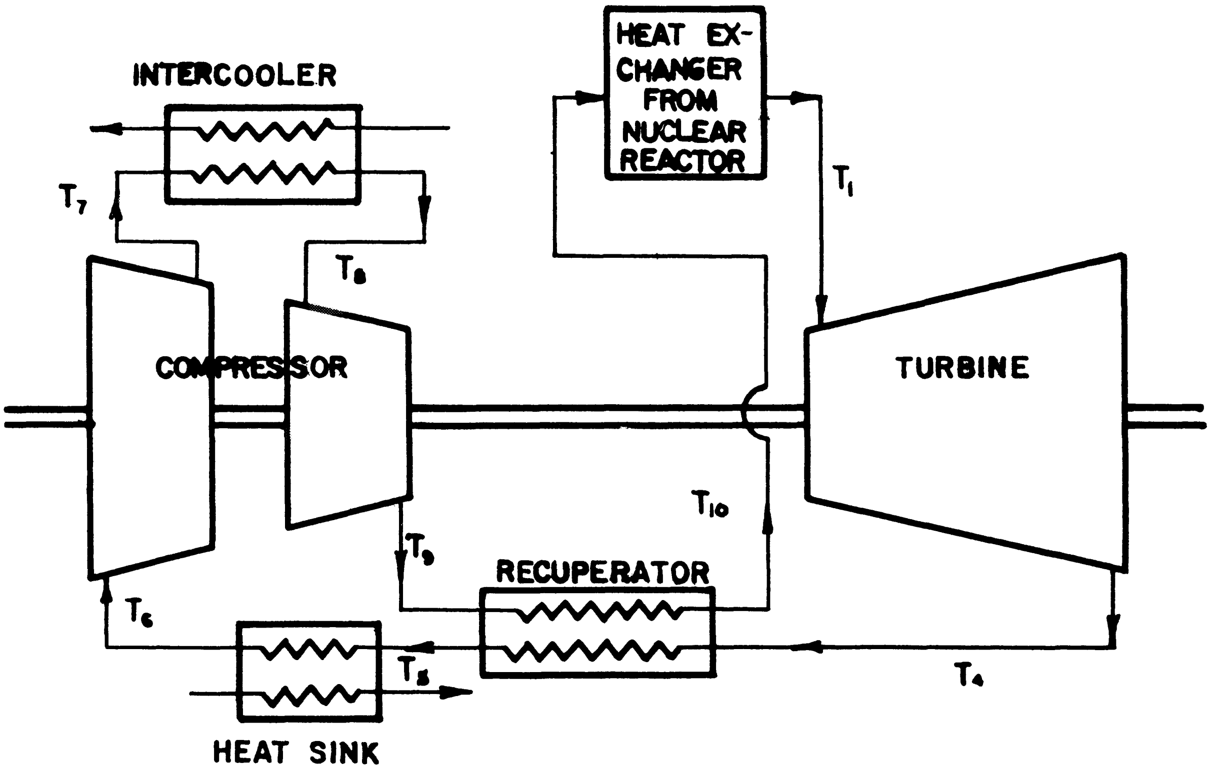


Figure 1. **SCHEMATIC FLOW DIAGRAM OF THE BASIC GAS TURBINE CYCLE**

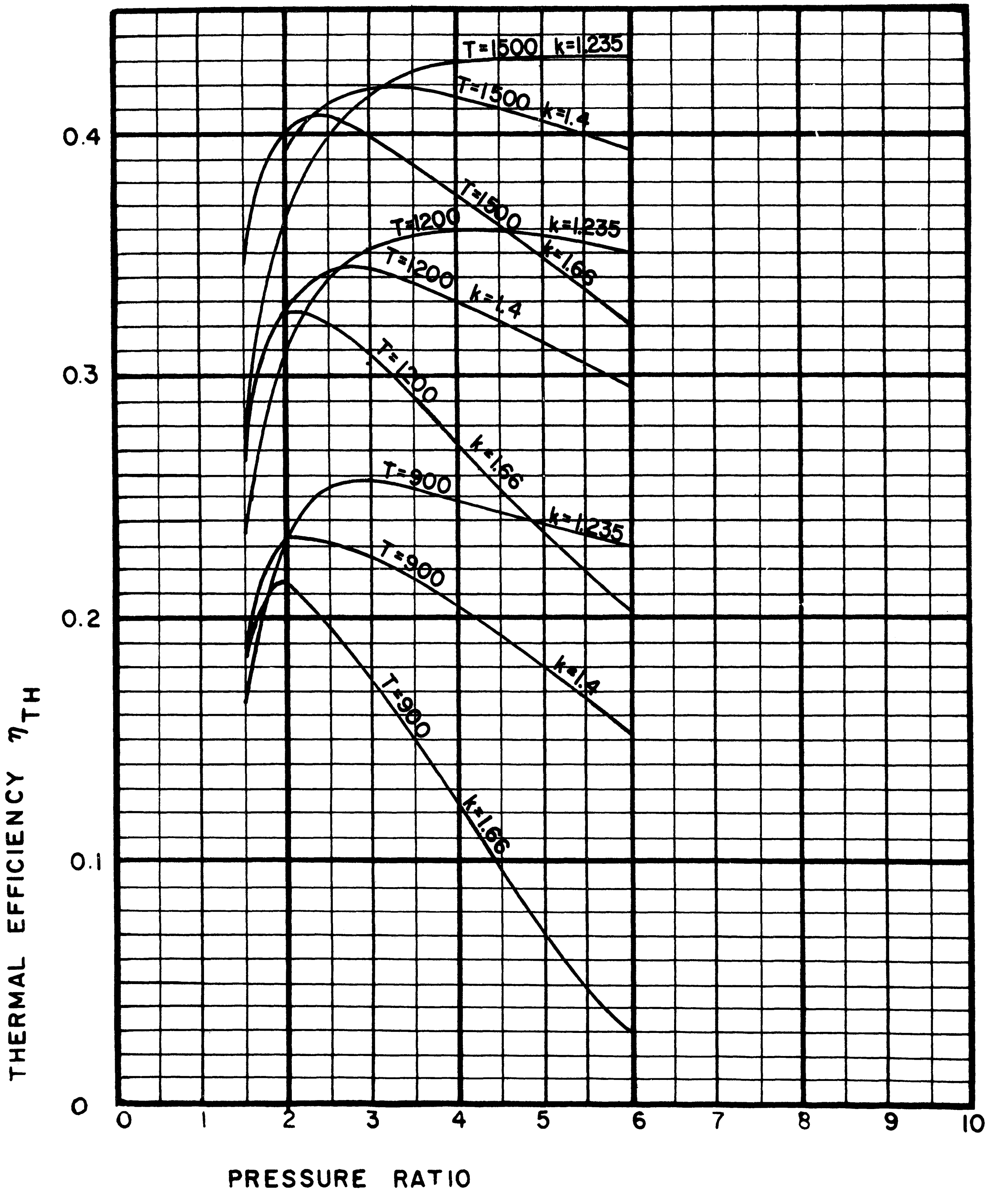


Figure 2. THERMAL EFFICIENCY OF A GAS TURBINE CYCLE WITH VARIOUS CYCLE ARRANGEMENTS

"BASIC" CYCLE

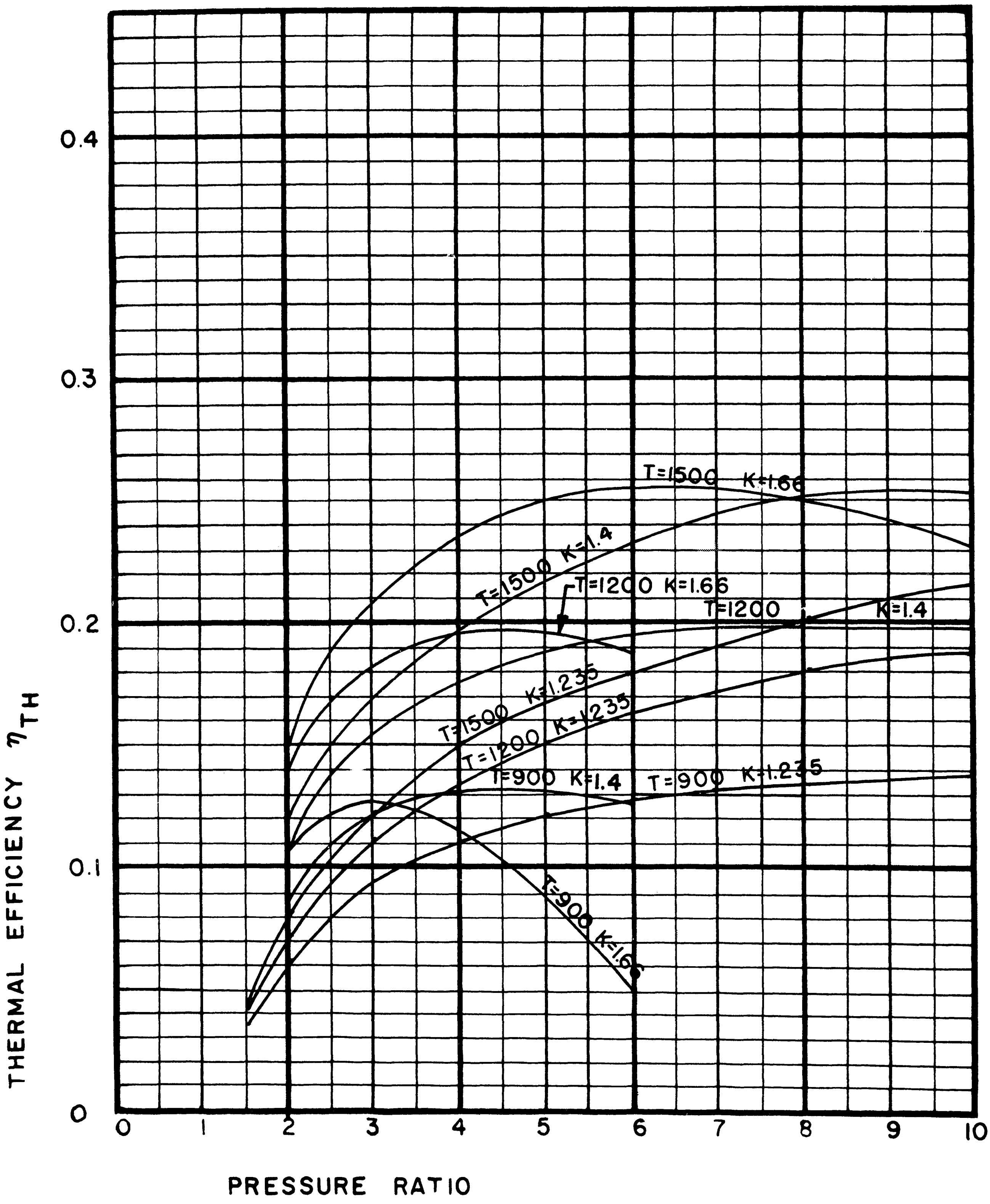


Figure 3. THERMAL EFFICIENCY OF A GAS TURBINE CYCLE WITH VARIOUS CYCLE ARRANGEMENTS

"BASIC" CYCLE WITHOUT RECUPERATOR

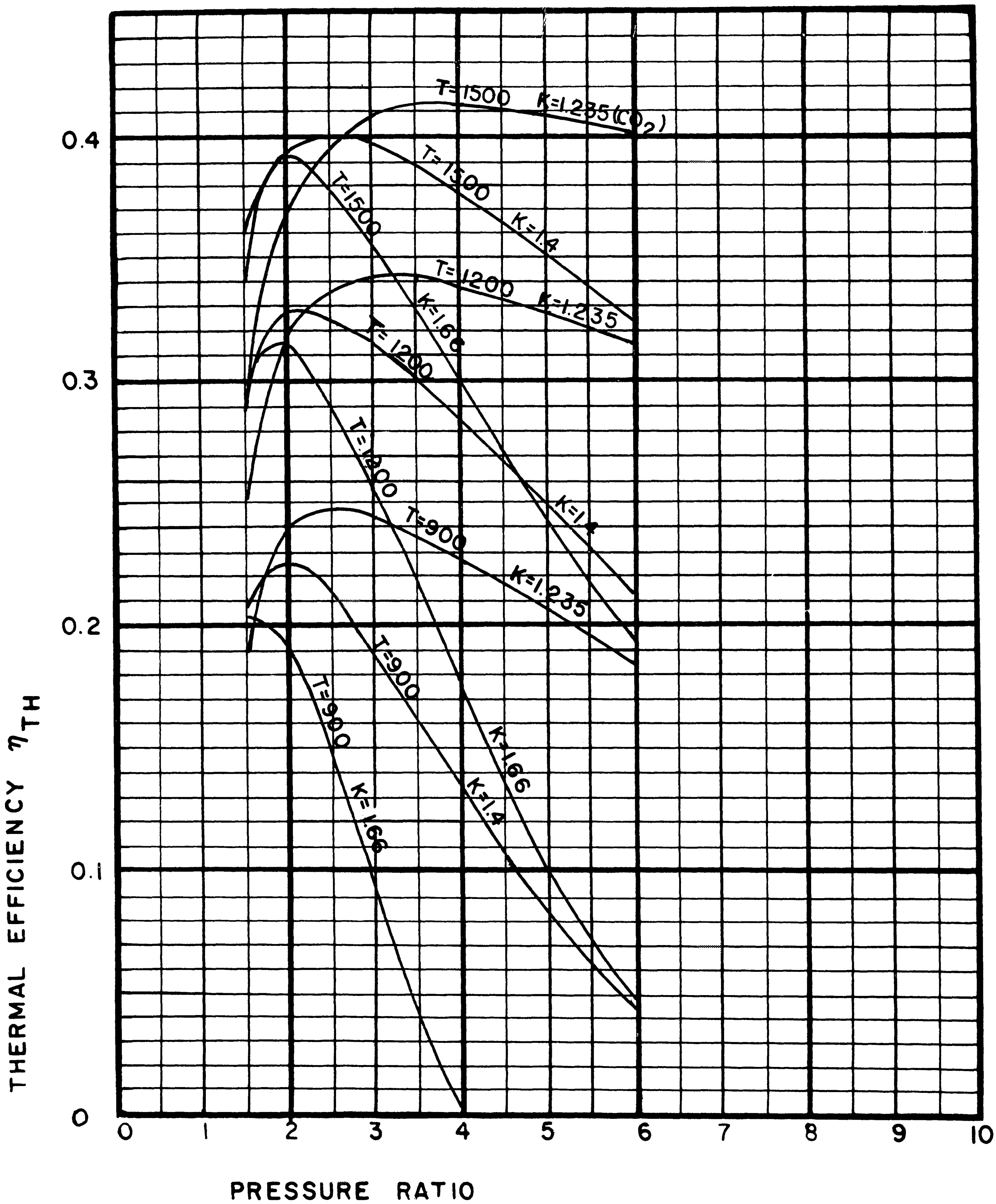


Figure 4. THERMAL EFFICIENCY OF A GAS TURBINE CYCLE WITH VARIOUS CYCLE ARRANGEMENTS

"BASIC" CYCLE WITHOUT INTERCOOLER

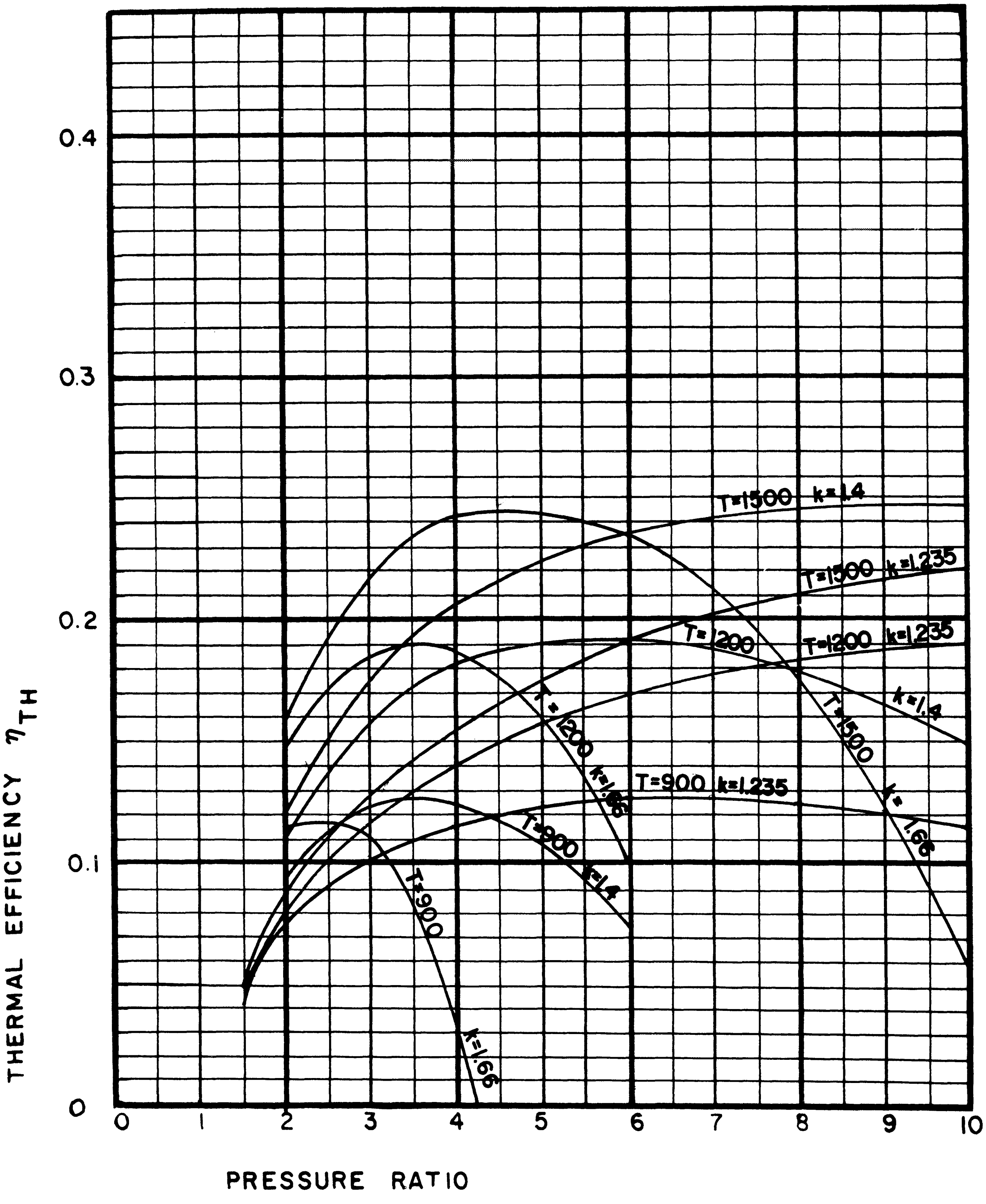


Figure 5. THERMAL EFFICIENCY OF A GAS TURBINE CYCLE WITH VARIOUS CYCLE ARRANGEMENTS

"BASIC" CYCLE WITHOUT RECUPERATOR AND INTERCOOLER

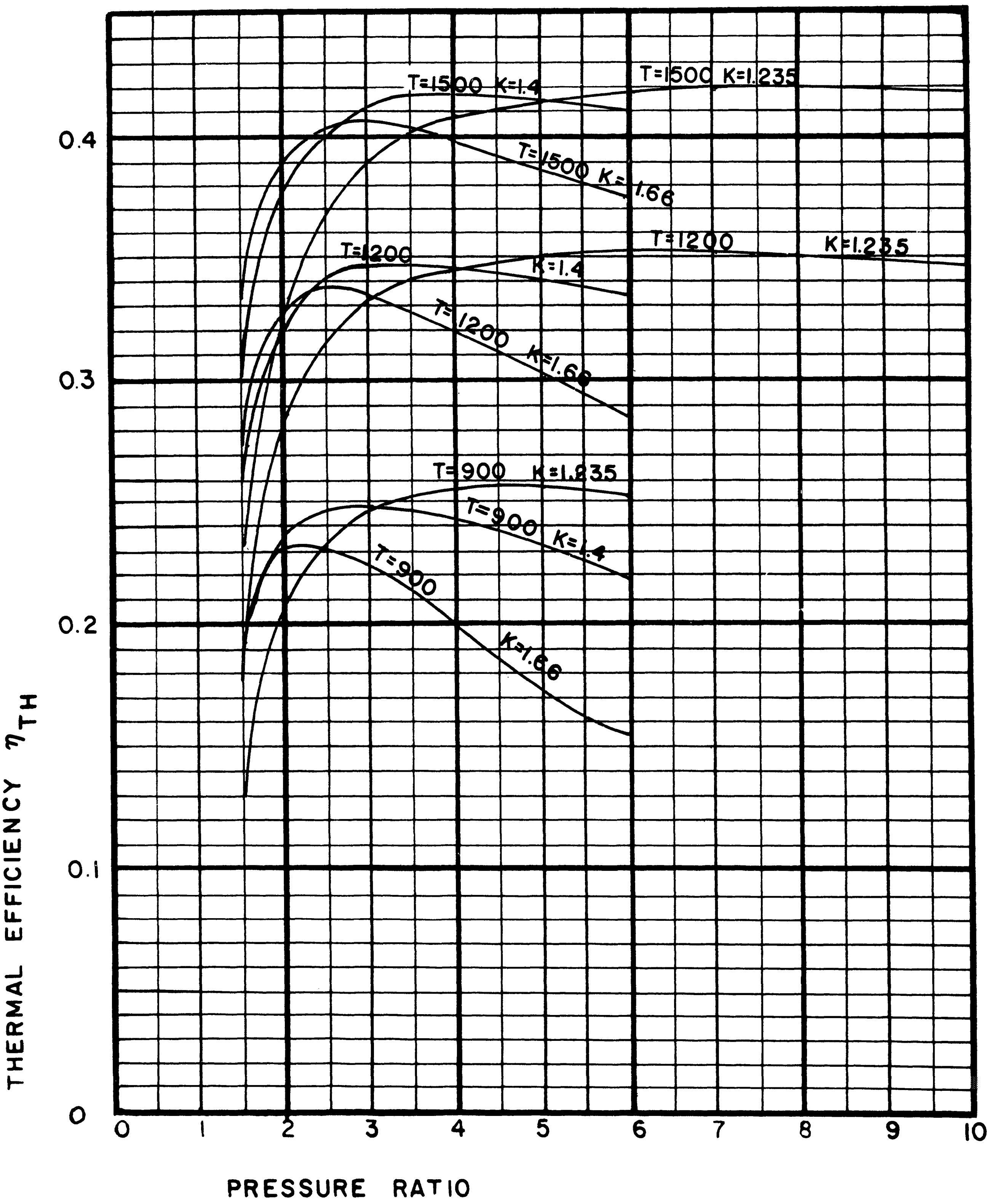


Figure 6. THERMAL EFFICIENCY OF A GAS TURBINE CYCLE WITH VARIOUS CYCLE ARRANGEMENTS.

"BASIC" CYCLE WITH REHEATER

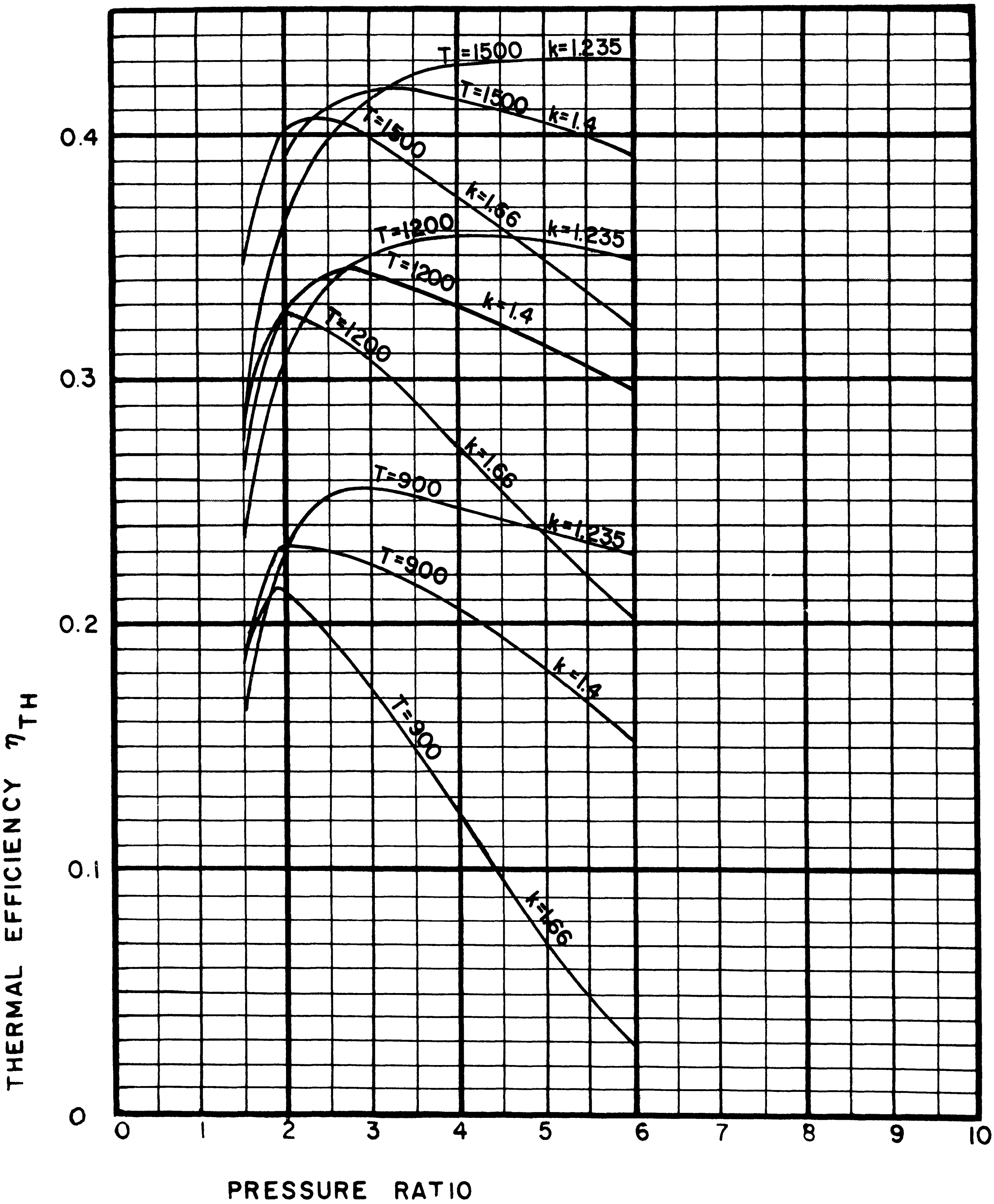


Figure 7. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH RECUPERATOR EFFECTIVENESS = 0.93

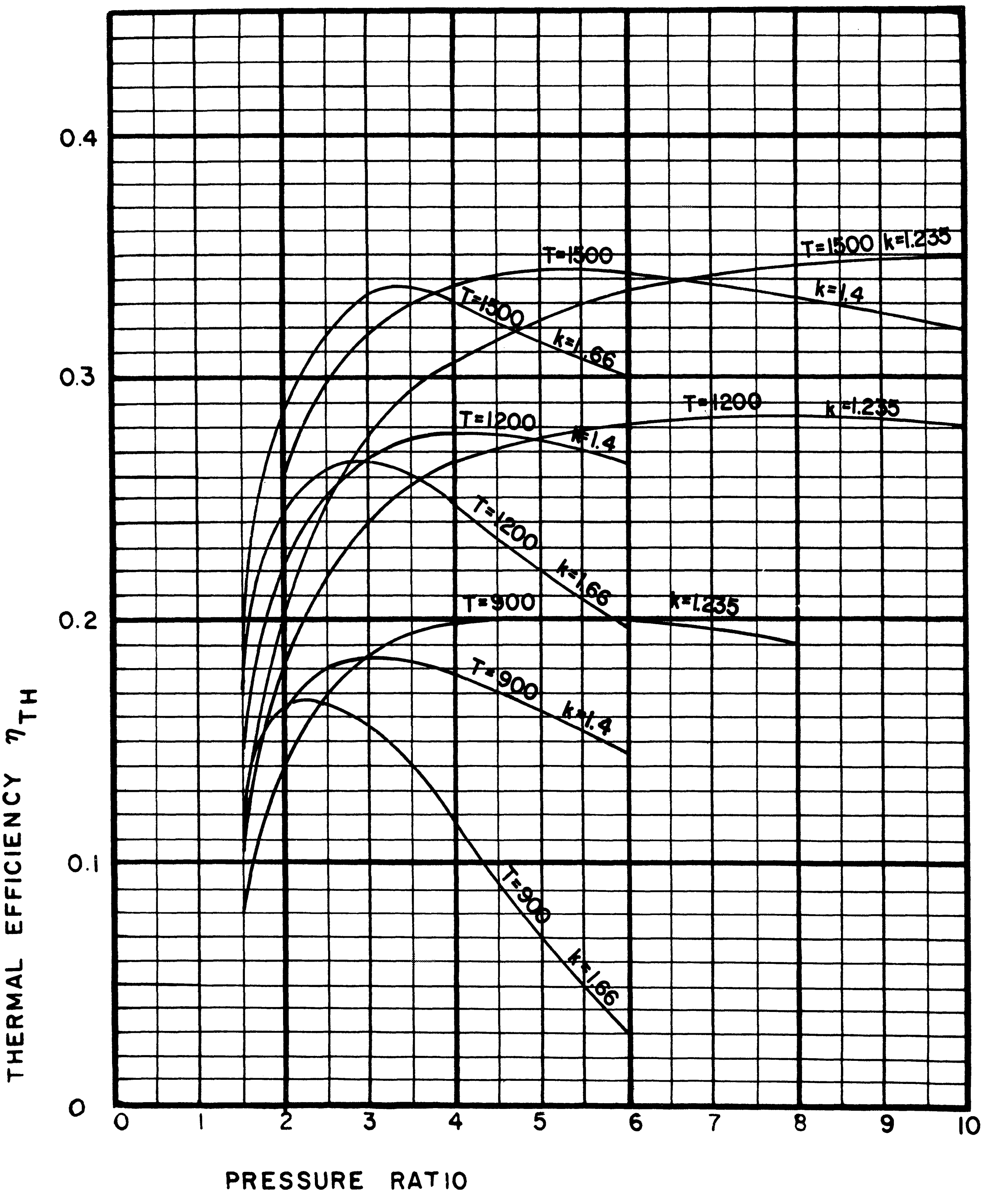


Figure 8. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH RECUPERATOR EFFECTIVENESS = 0.75

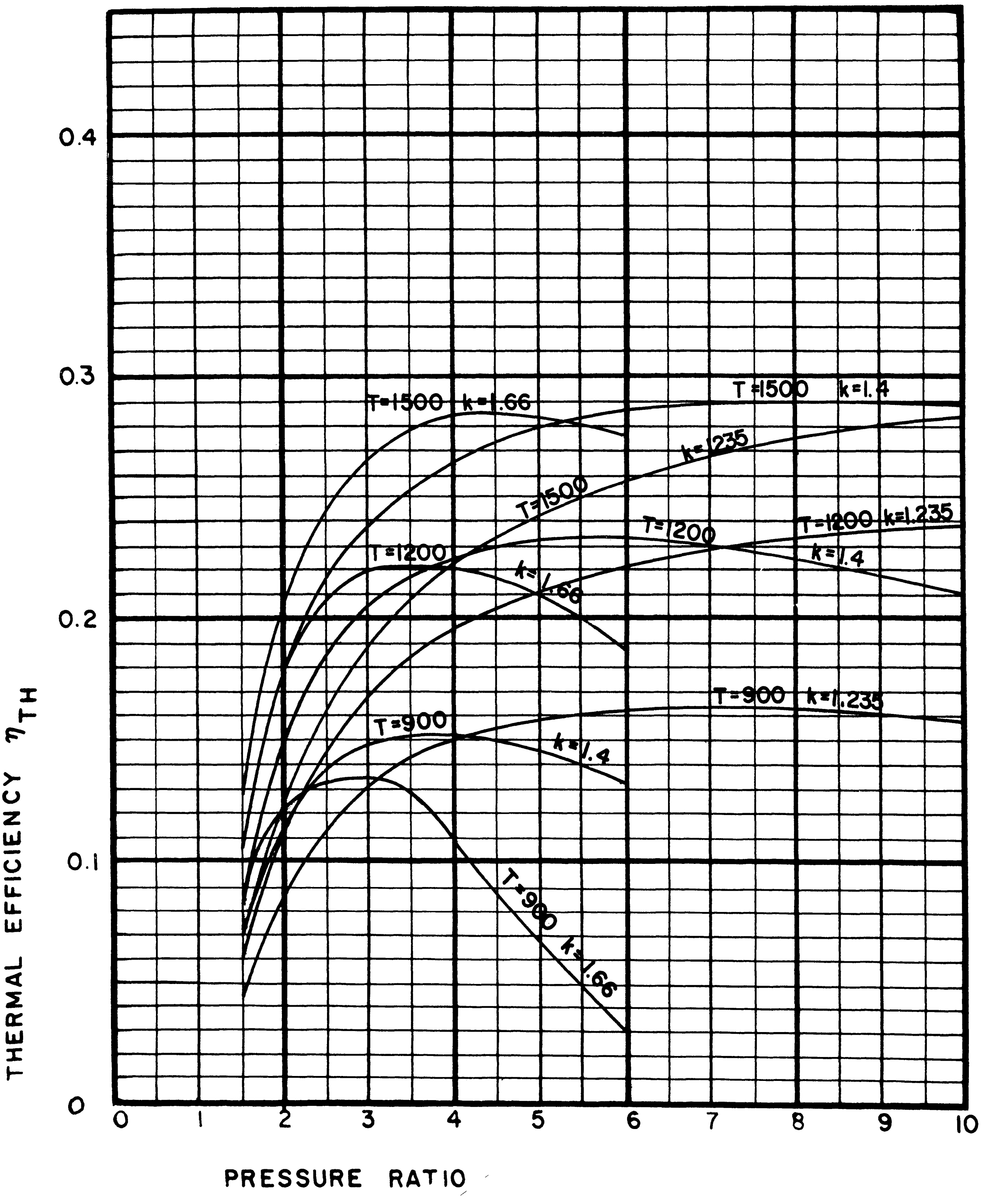


Figure 9. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH RECUPERATOR EFFECTIVENESS = 0.50

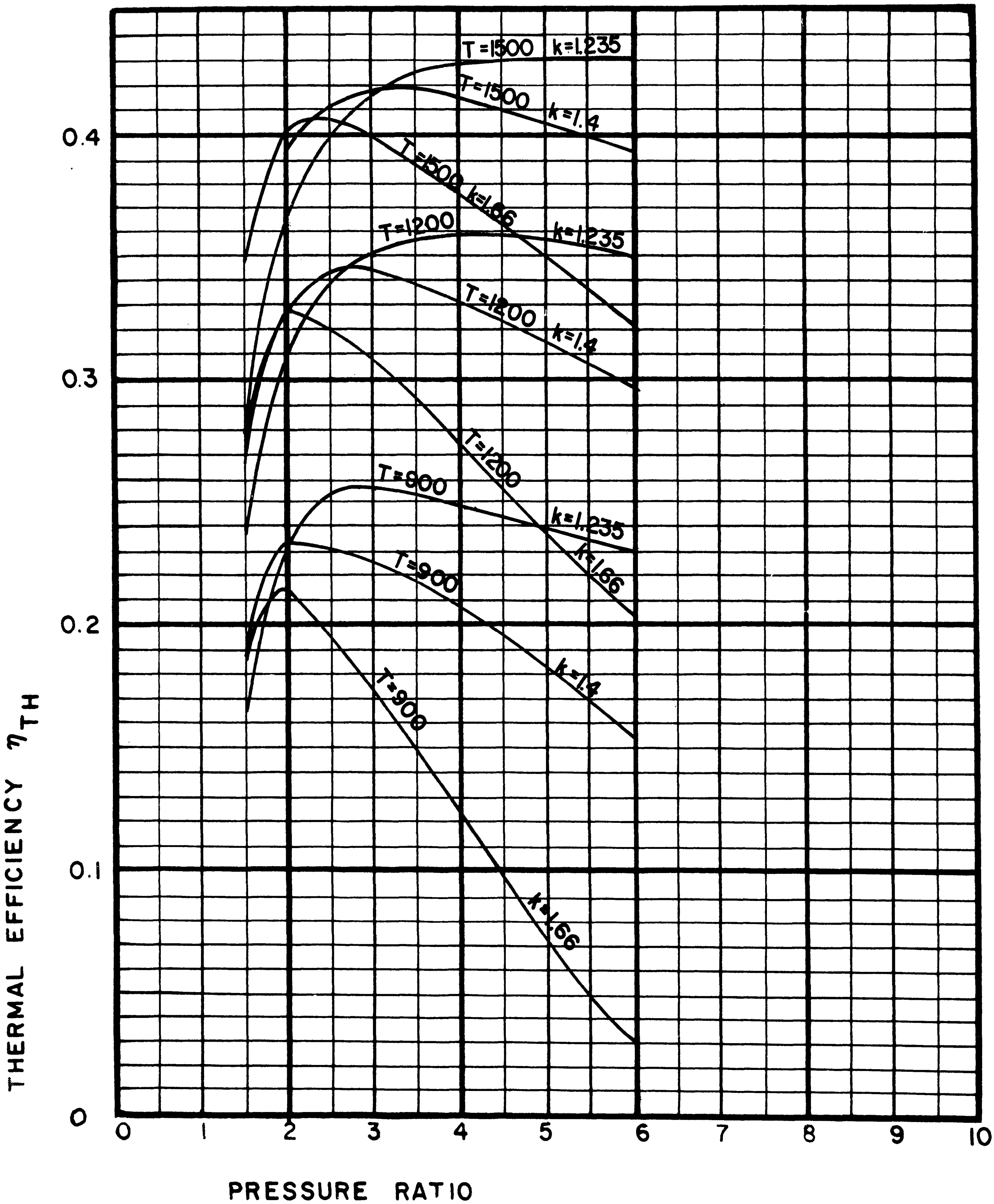


Figure 10. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH FRICTIONAL PRESSURE LOSSES=0.07

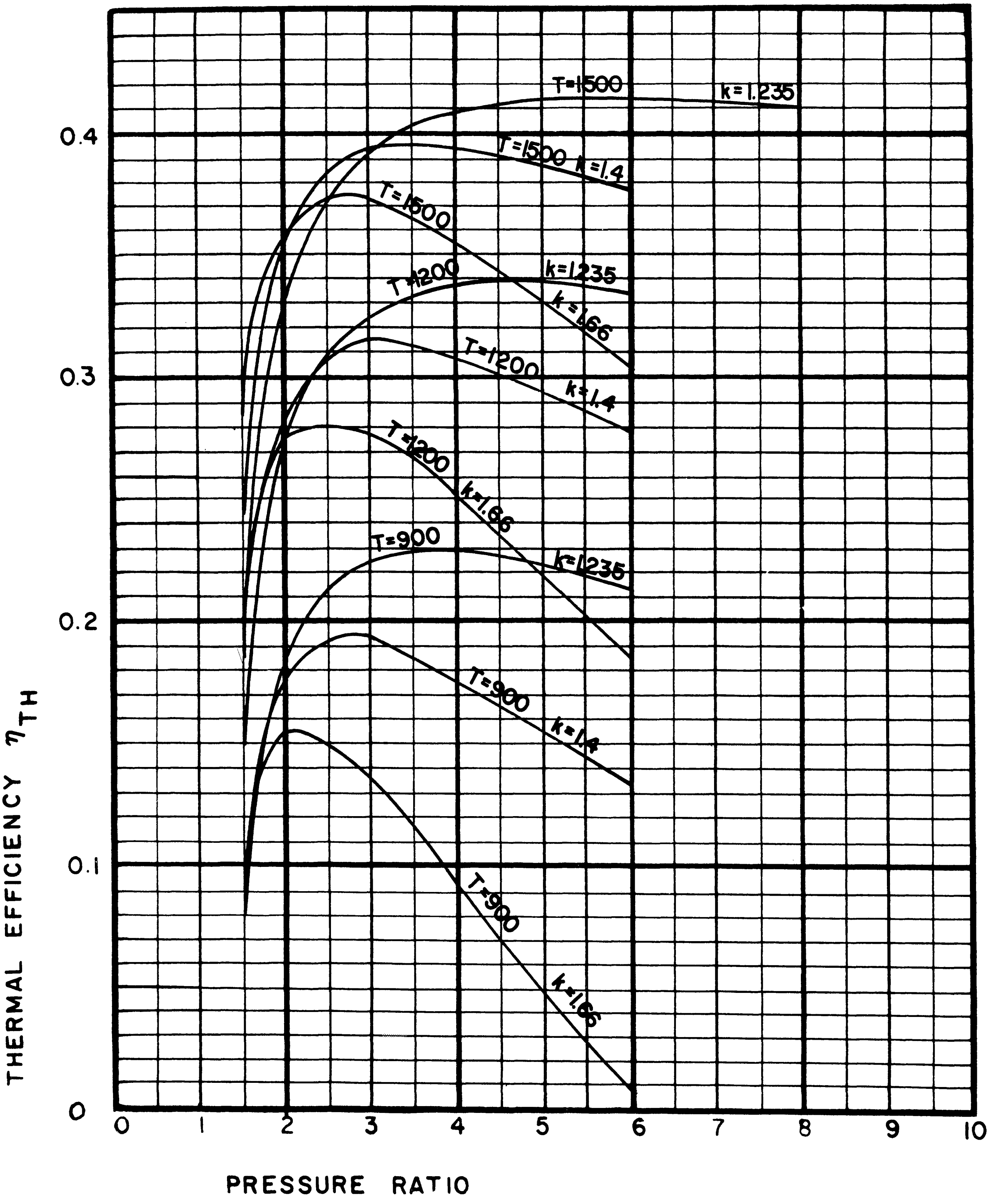


Figure 11. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH FRICTIONAL PRESSURE LOSSES=0.12

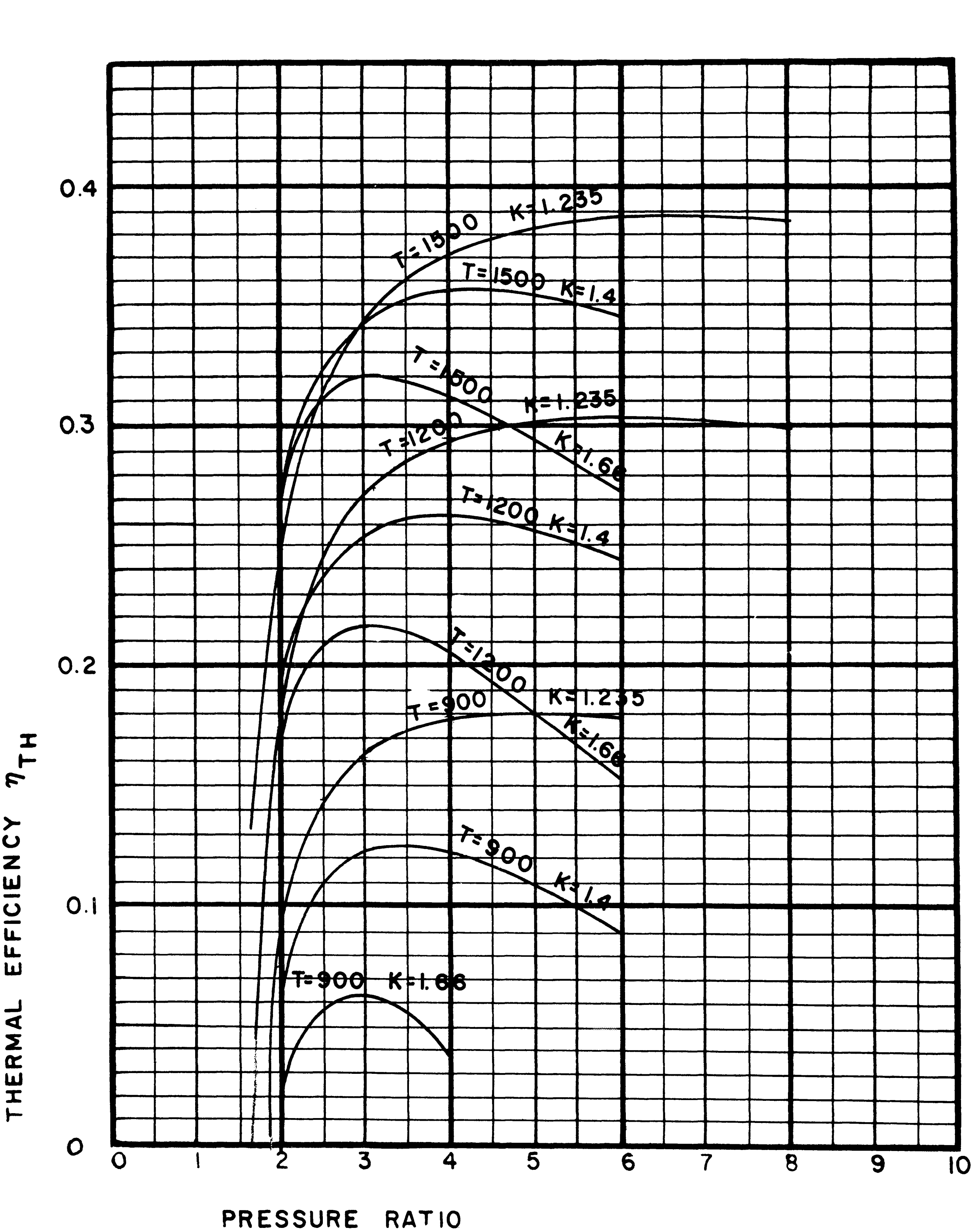


Figure 12. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH FRICTIONAL PRESSURE LOSSES= 0.20

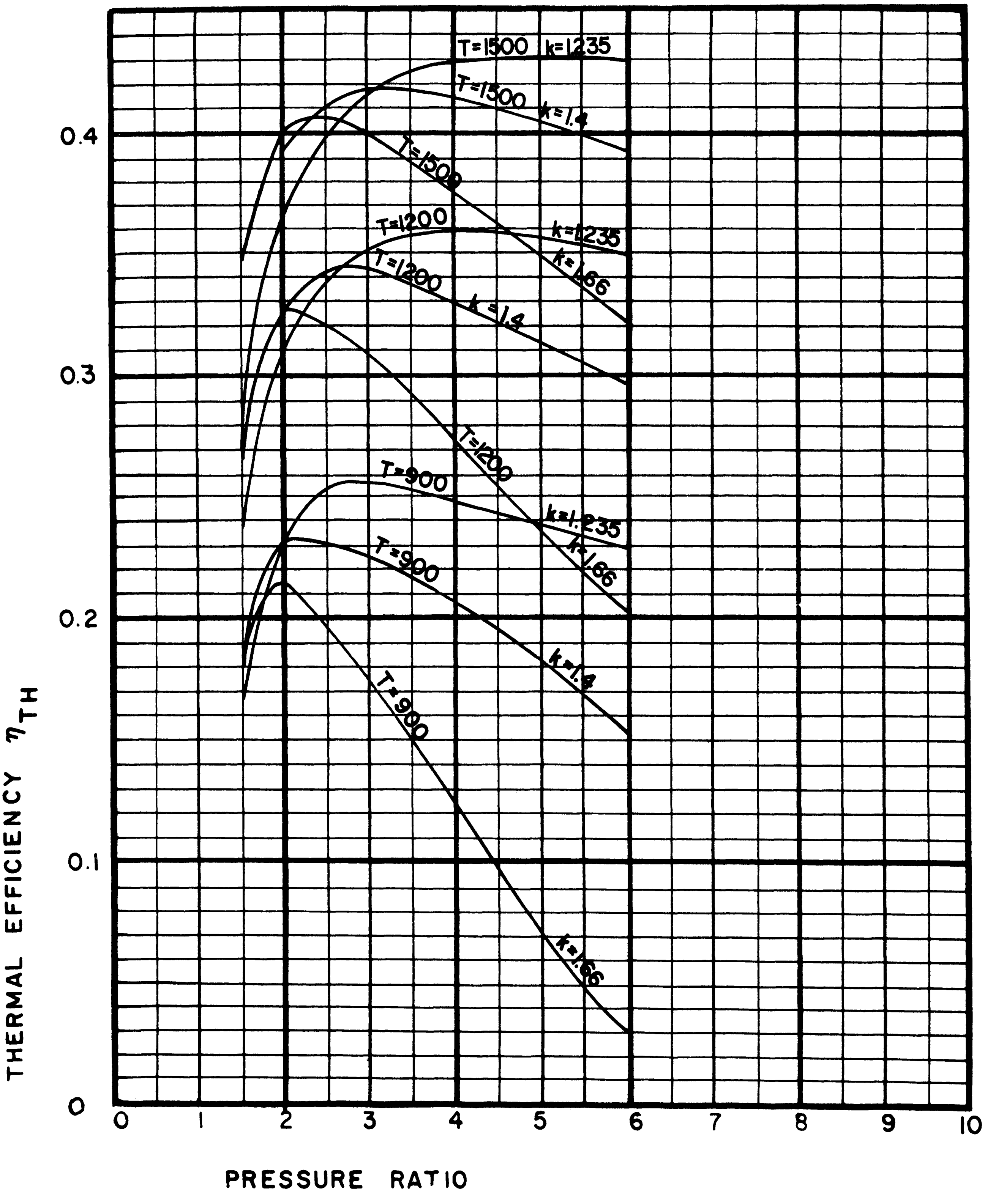


Figure 13. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH TURBINE EFFICIENCY = COMPRESSOR EFFICIENCY = 0.85

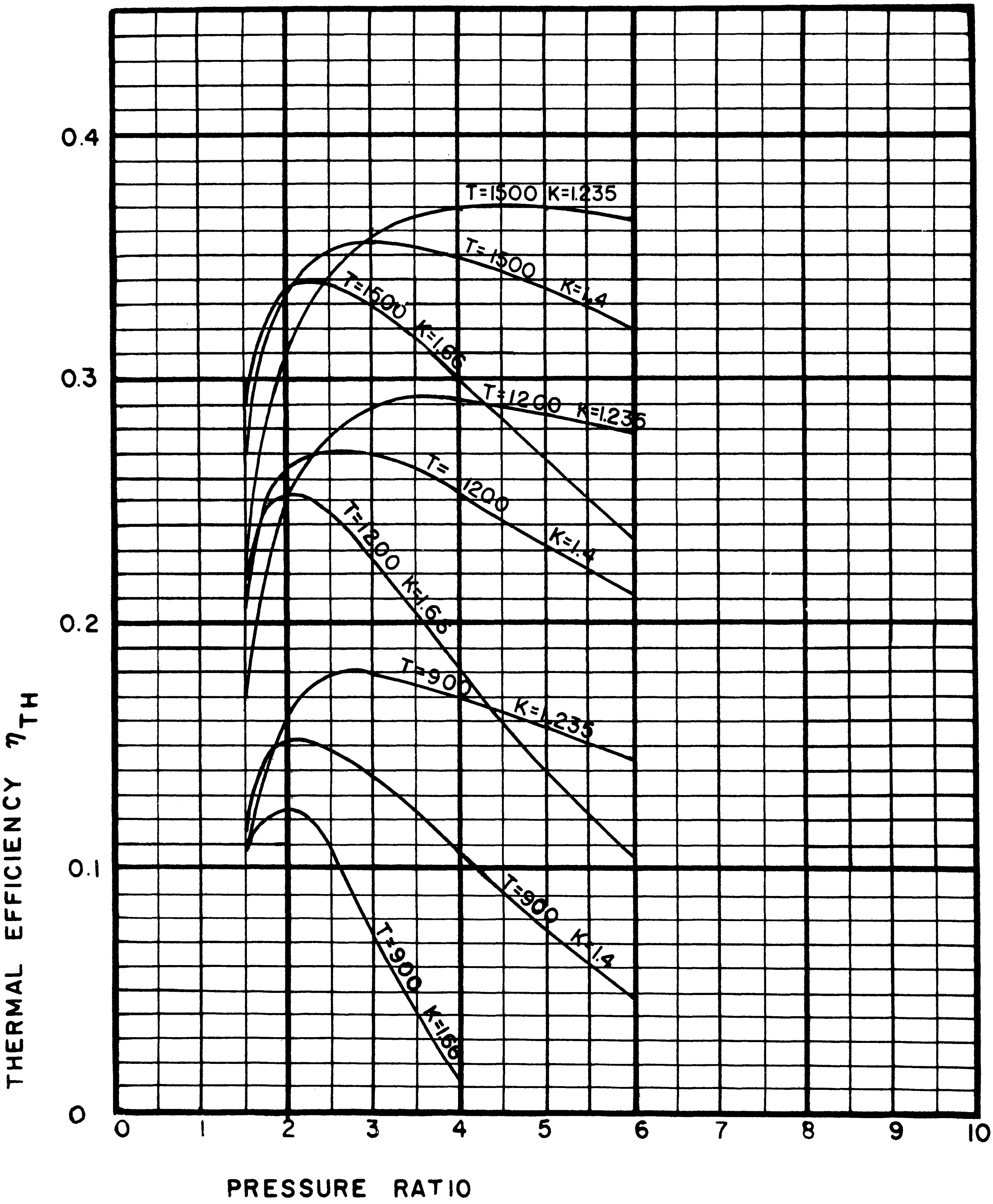


Figure 14. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH TURBINE EFFICIENCY=COMPRESSOR EFFICIENCY=0.80

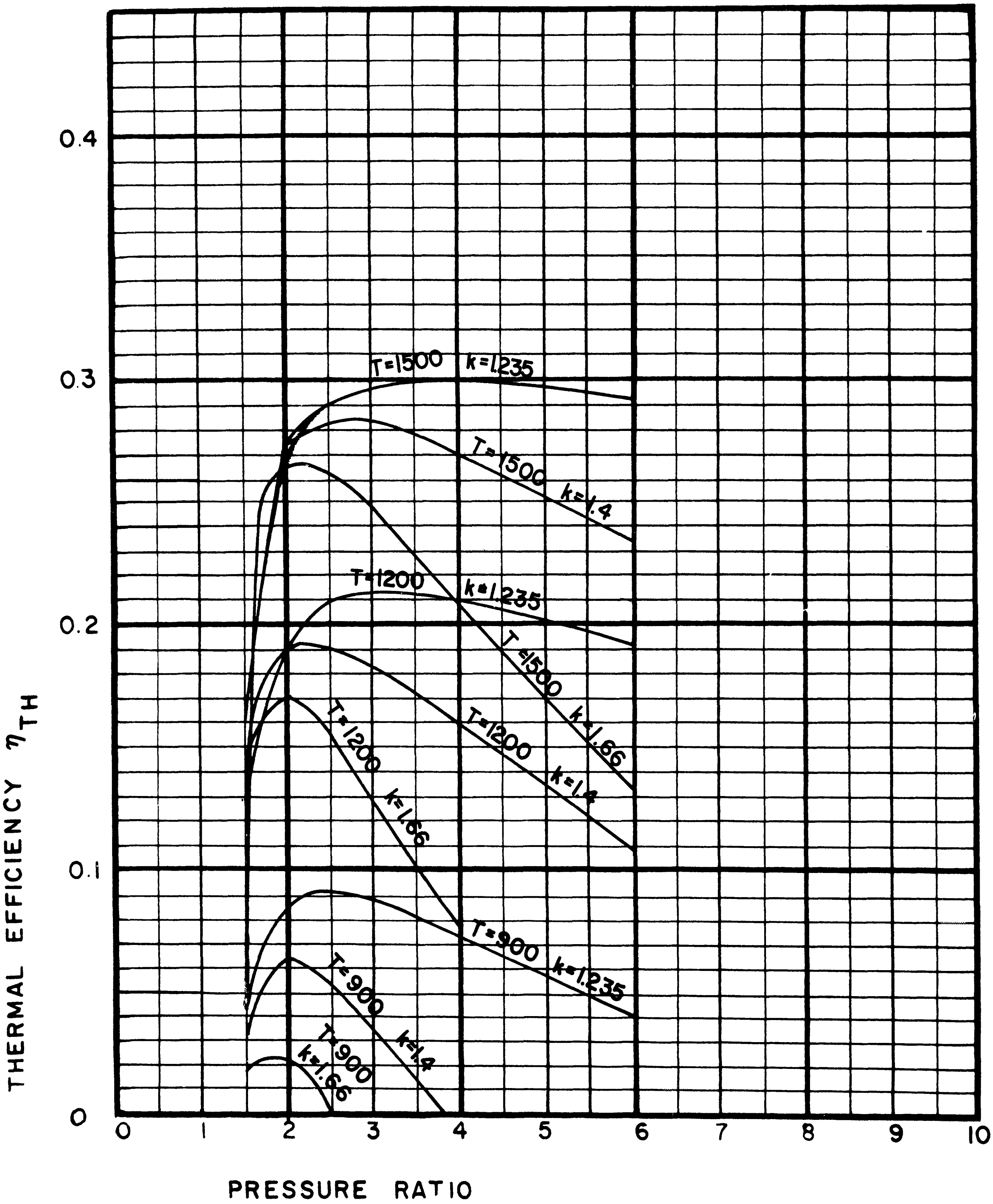


Figure 15. THERMAL EFFICIENCY OF A "BASIC" GAS TURBINE CYCLE WITH TURBINE EFFICIENCY = COMPRESSOR EFFICIENCY = 0.75

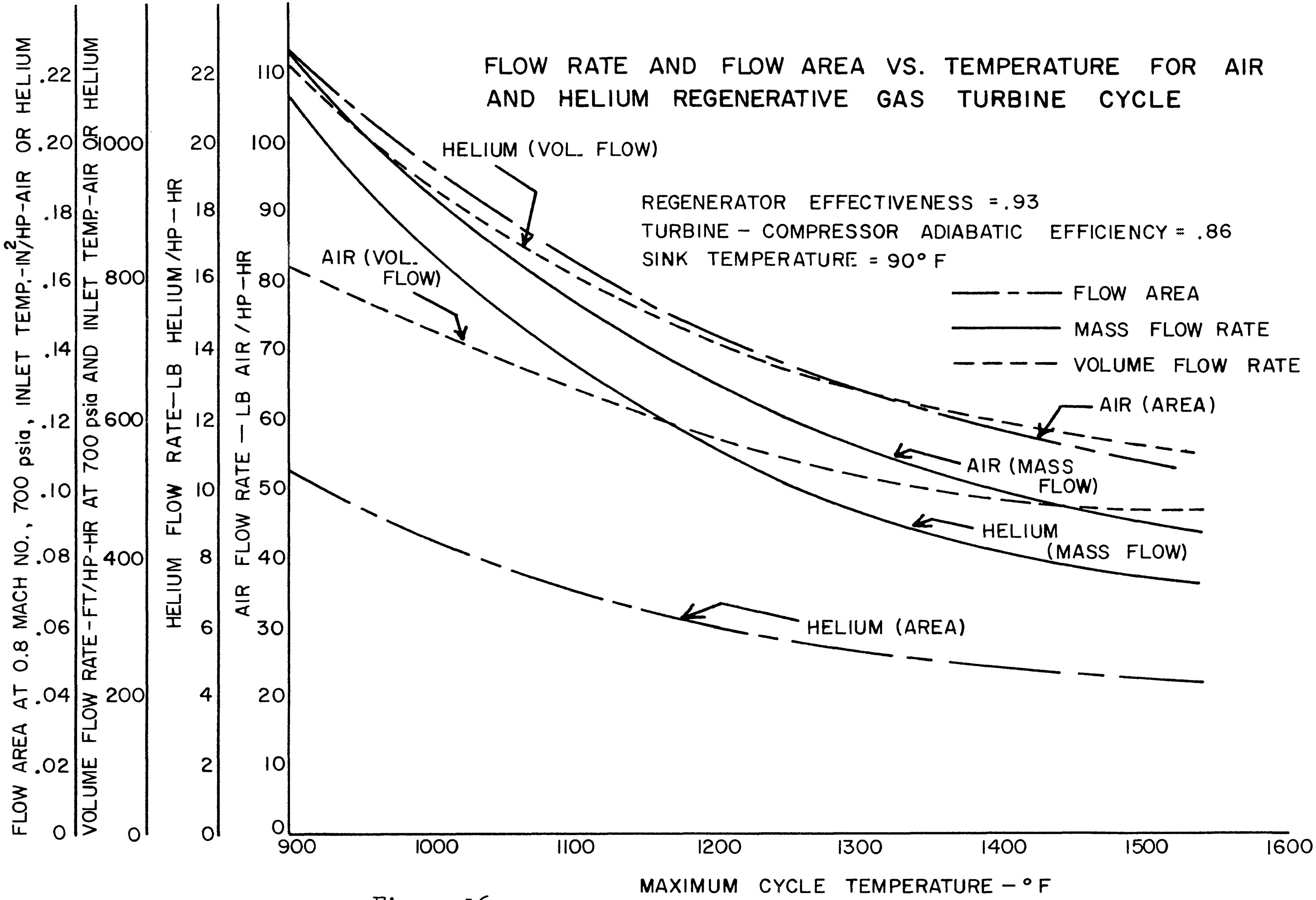
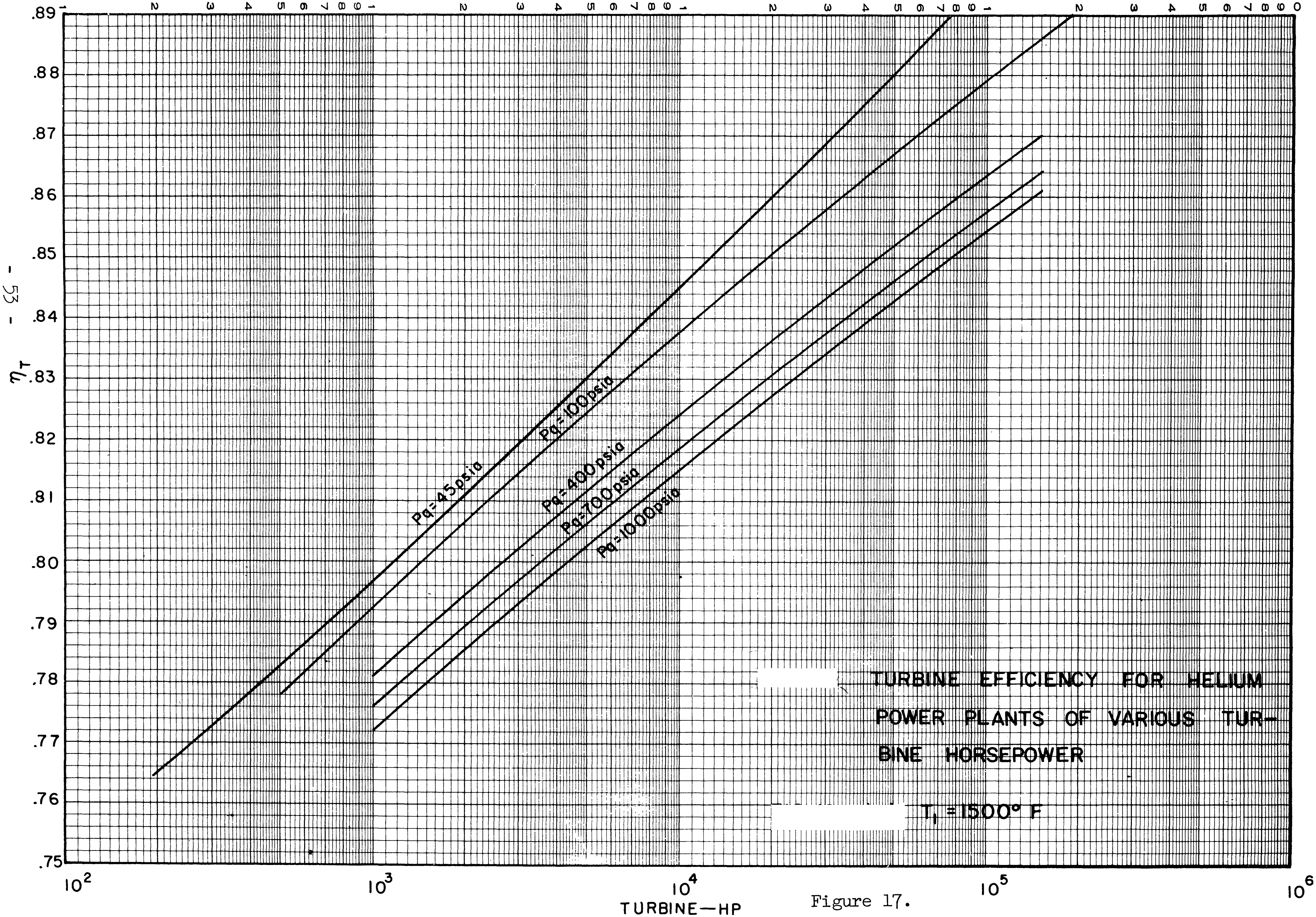
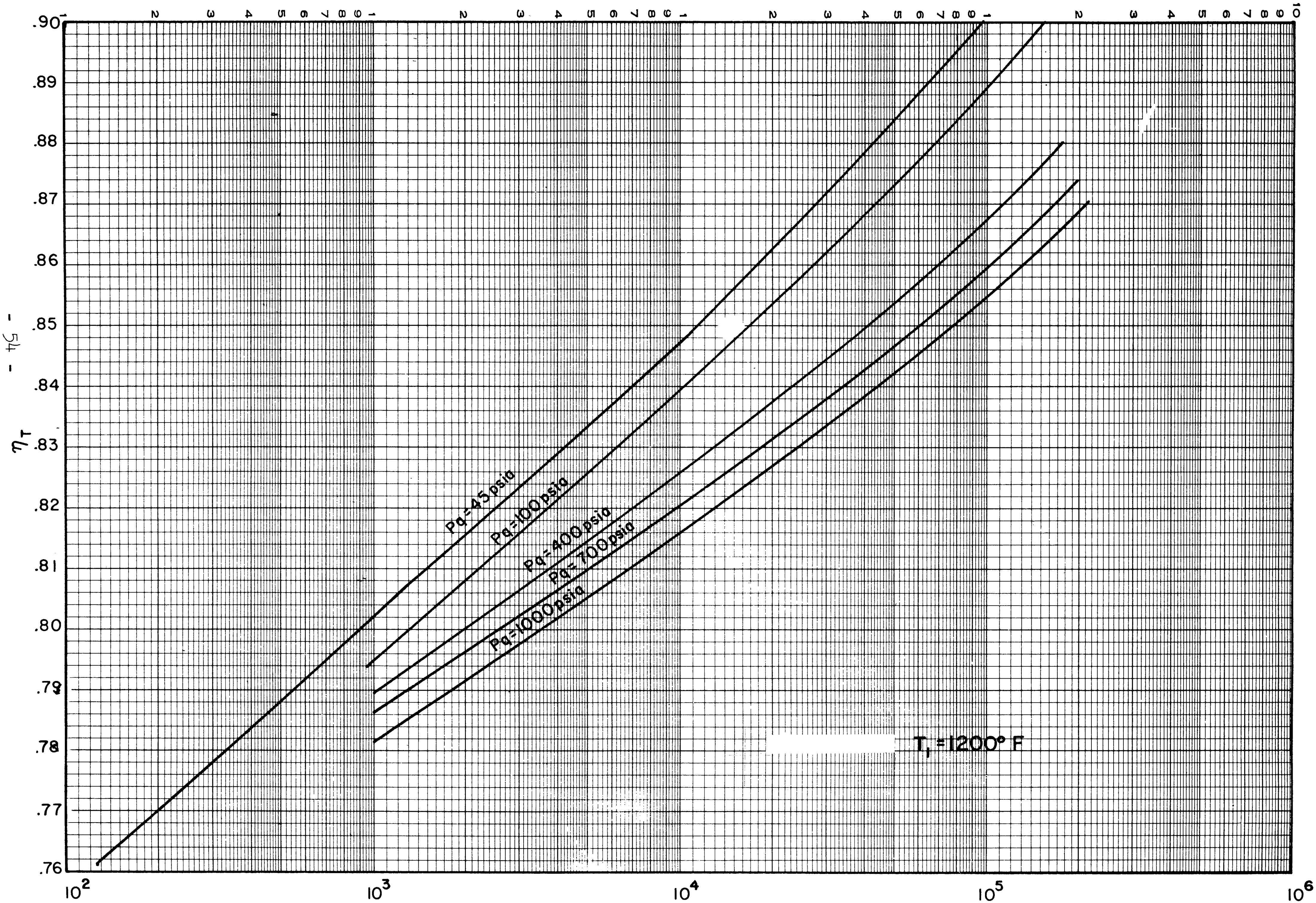

FLOW RATE AND FLOW AREA VS. TEMPERATURE FOR AIR AND HELIUM REGENERATIVE GAS TURBINE CYCLE
REGENERATOR EFFECTIVENESS = .93
TURBINE - COMPRESSOR ADIABATIC EFFICIENCY = .86
SINK TEMPERATURE = 90° F
FLOW AREA
MASS FLOW RATE
VOLUME FLOW RATE
HELIUM (VOL. FLOW)
AIR (VOL. FLOW)
AIR (AREA)
AIR (MASS FLOW)
HELIUM (MASS FLOW)
HELIUM (AREA)
FLOW AREA AT 0.8 MACH NO., 700 psia, INLET TEMP.-IN²/HP-AIR OR HELIUM
VOLUME FLOW RATE-FT/HP-HR AT 700 psia AND INLET TEMP.-AIR OR HELIUM
HELIUM FLOW RATE—LB HELIUM/HP—HR
AIR FLOW RATE —LB AIR/HP—HR
MAXIMUM CYCLE TEMPERATURE — °F

Figure 16.


TURBINE EFFICIENCY FOR HELIUM POWER PLANTS OF VARIOUS TURBINE HORSEPOWER
T_1 =1500° F
Pa=45psia
Pa=100psia
Pa=400psia
Pa=700psia
Pa=1000psia
η_T
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.77
.76
.75
10^2
10^3
10^4
10^5
10^6
TURBINE—HP
Figure 17.



η_T
TURBINE — HP
$T_1 = 1200°$ F
Pq = 45 psia
Pq = 100 psia
Pq = 400 psia
Pq = 700 psia
Pq = 1000 psia

Figure 18.

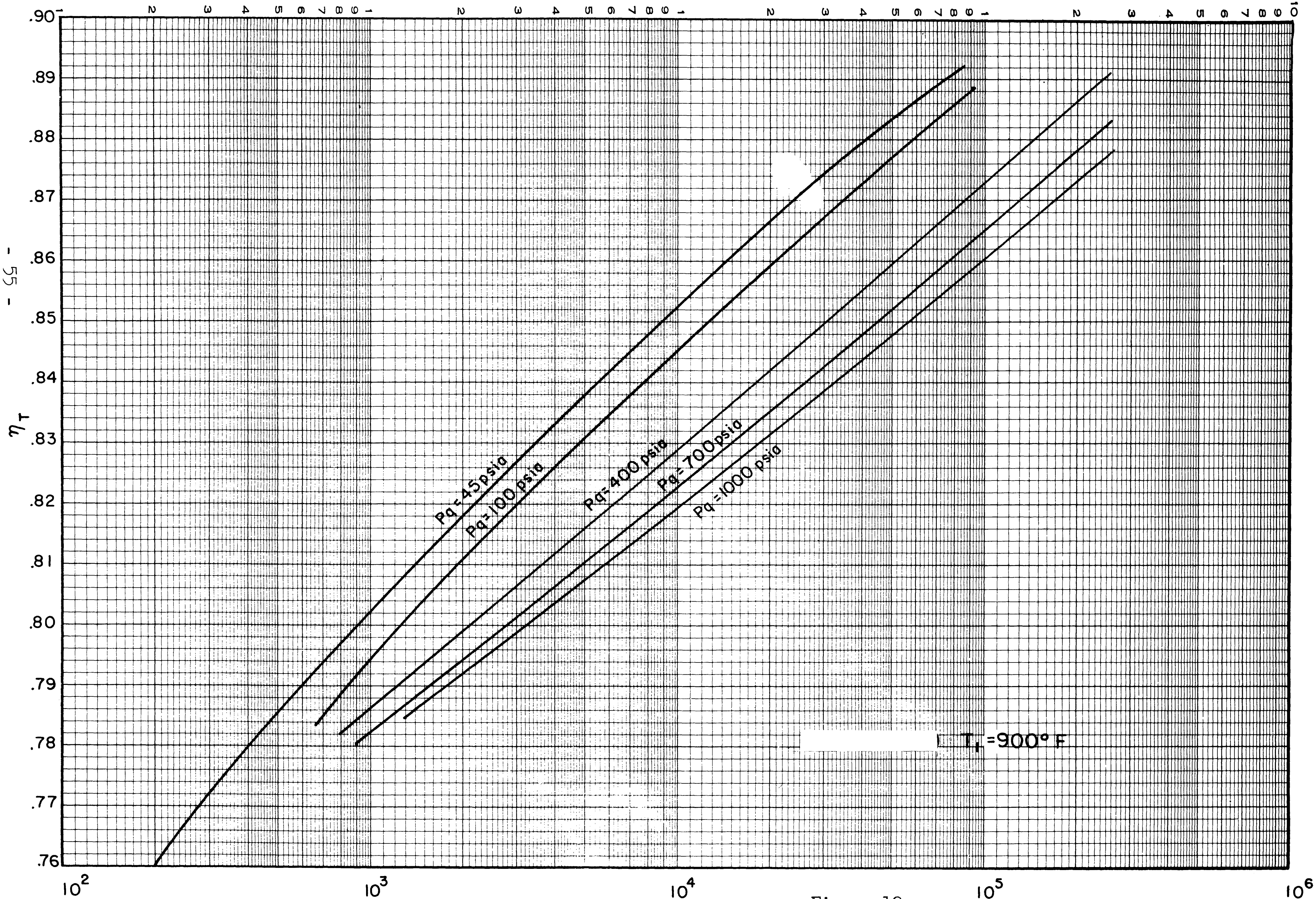

Pq=45psia
Pq=100 psia
Pq=400 psia
Pq=700psia
Pq=1000 psia
T1 =900° F
TURBINE — HP
ηT

Figure 19.

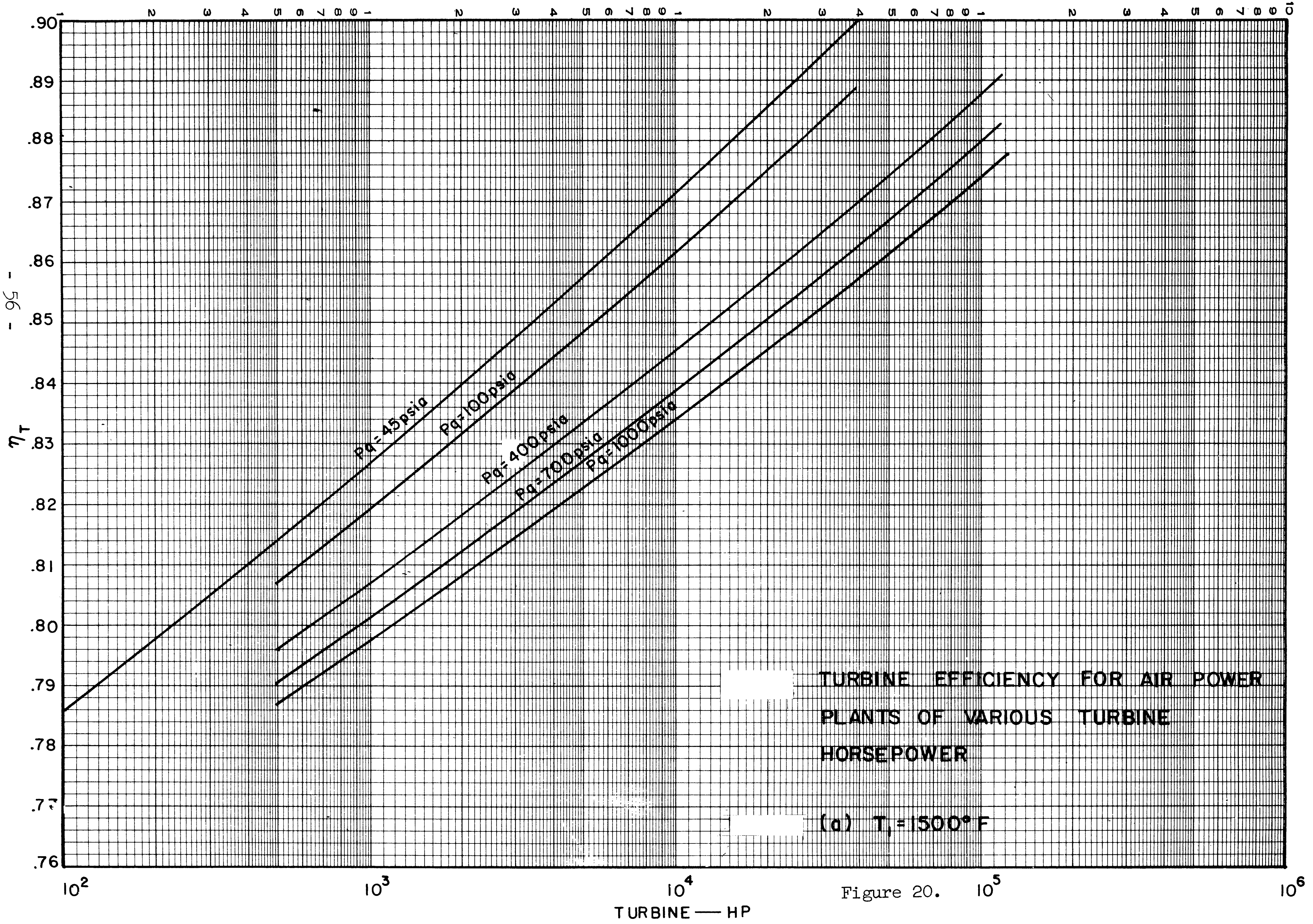
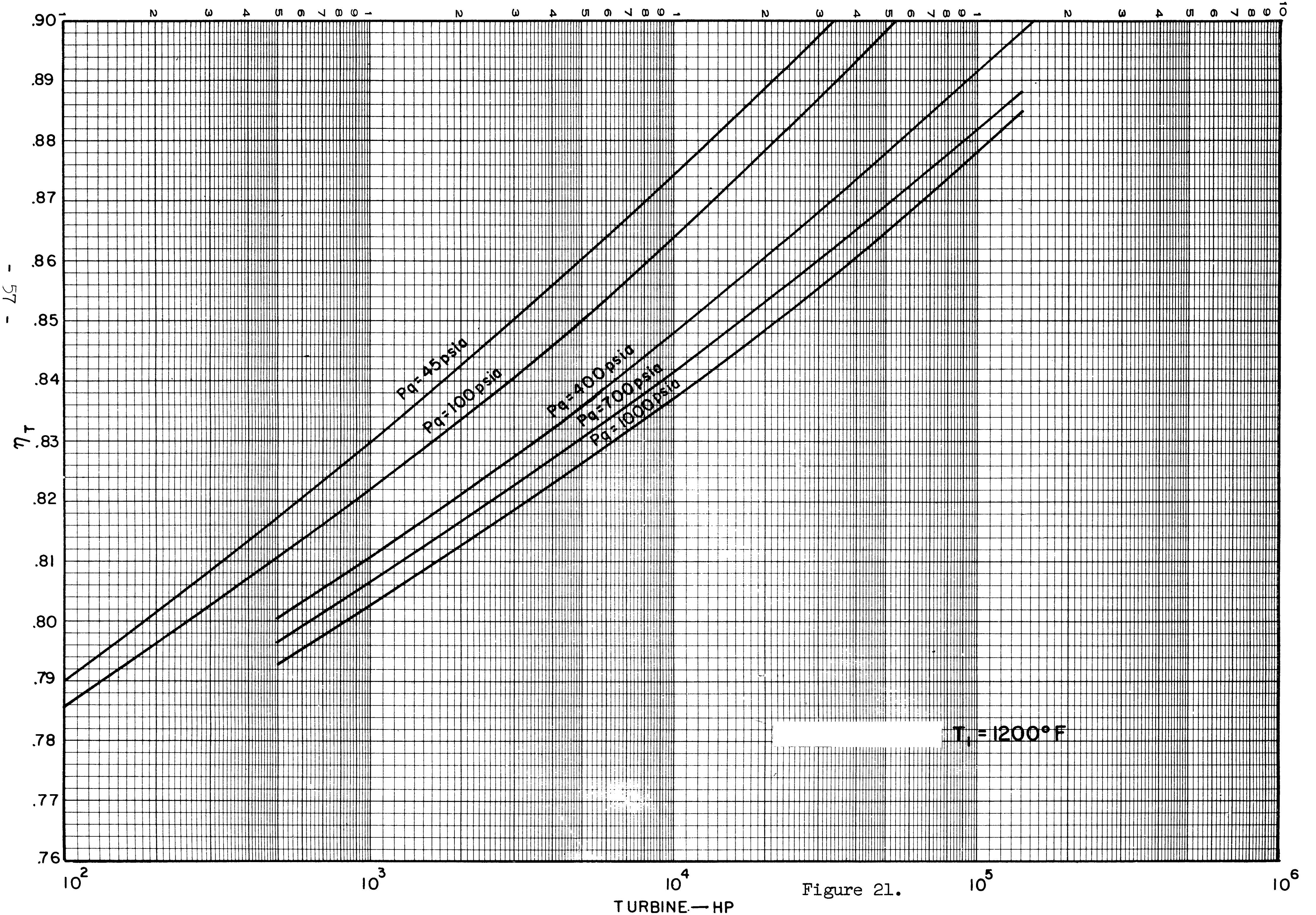

TURBINE EFFICIENCY FOR AIR POWER PLANTS OF VARIOUS TURBINE HORSEPOWER
(a) T_1=1500° F
Pa=45psia
Pa=100psia
Pa=400psia
Pa=700psia
Pa=1000psia
η_T
TURBINE — HP
.90
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.76
10^2
10^3
10^4
10^5
10^6

Figure 20.



.90
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.77
.76
η_T
10^2
10^3
10^4
10^5
10^6
TURBINE — HP
Pa = 45 psia
Pa = 100 psia
Pa = 400 psia
Pa = 700 psia
Pa = 1000 psia
T_1 = 1200° F

Figure 21.

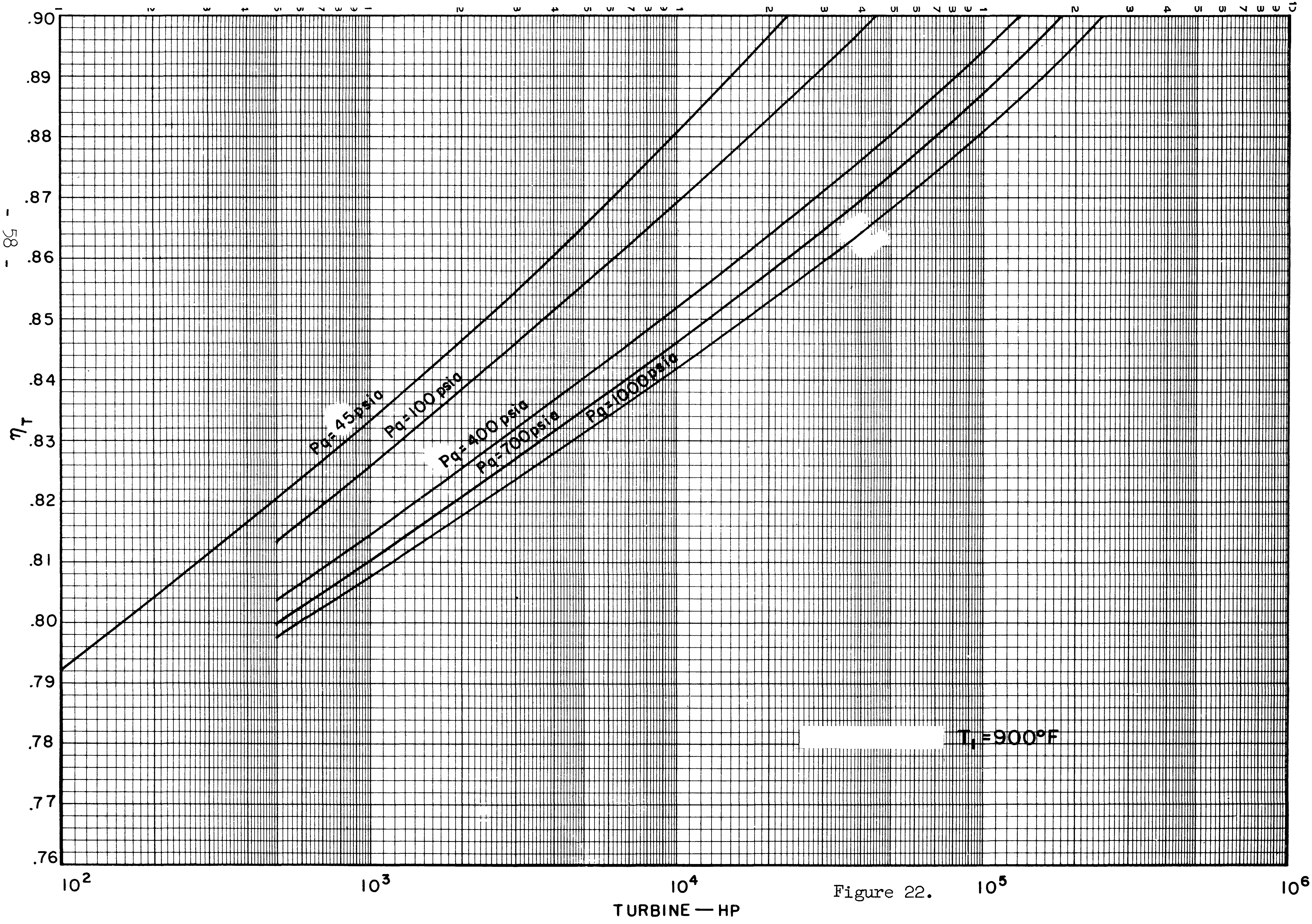
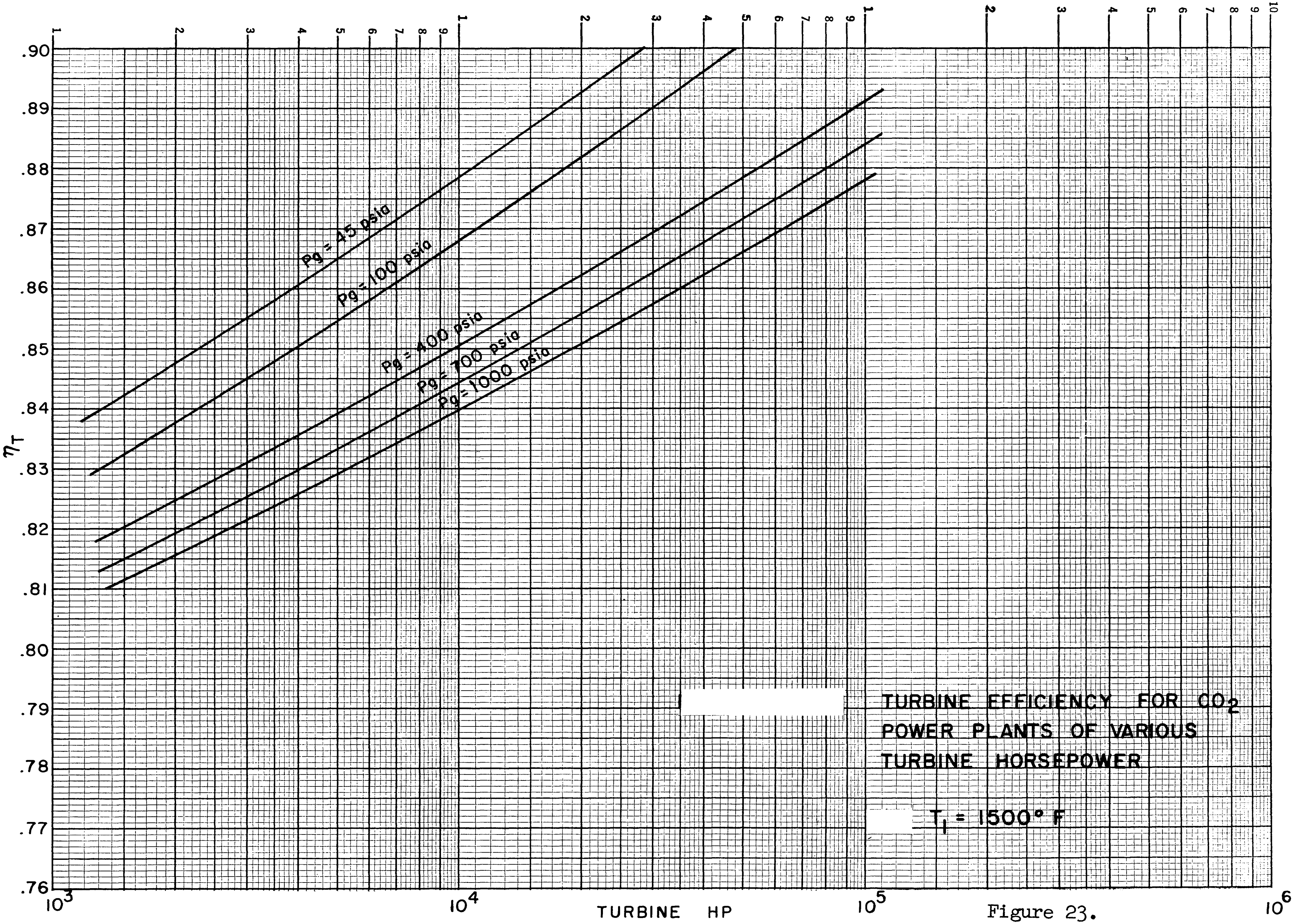
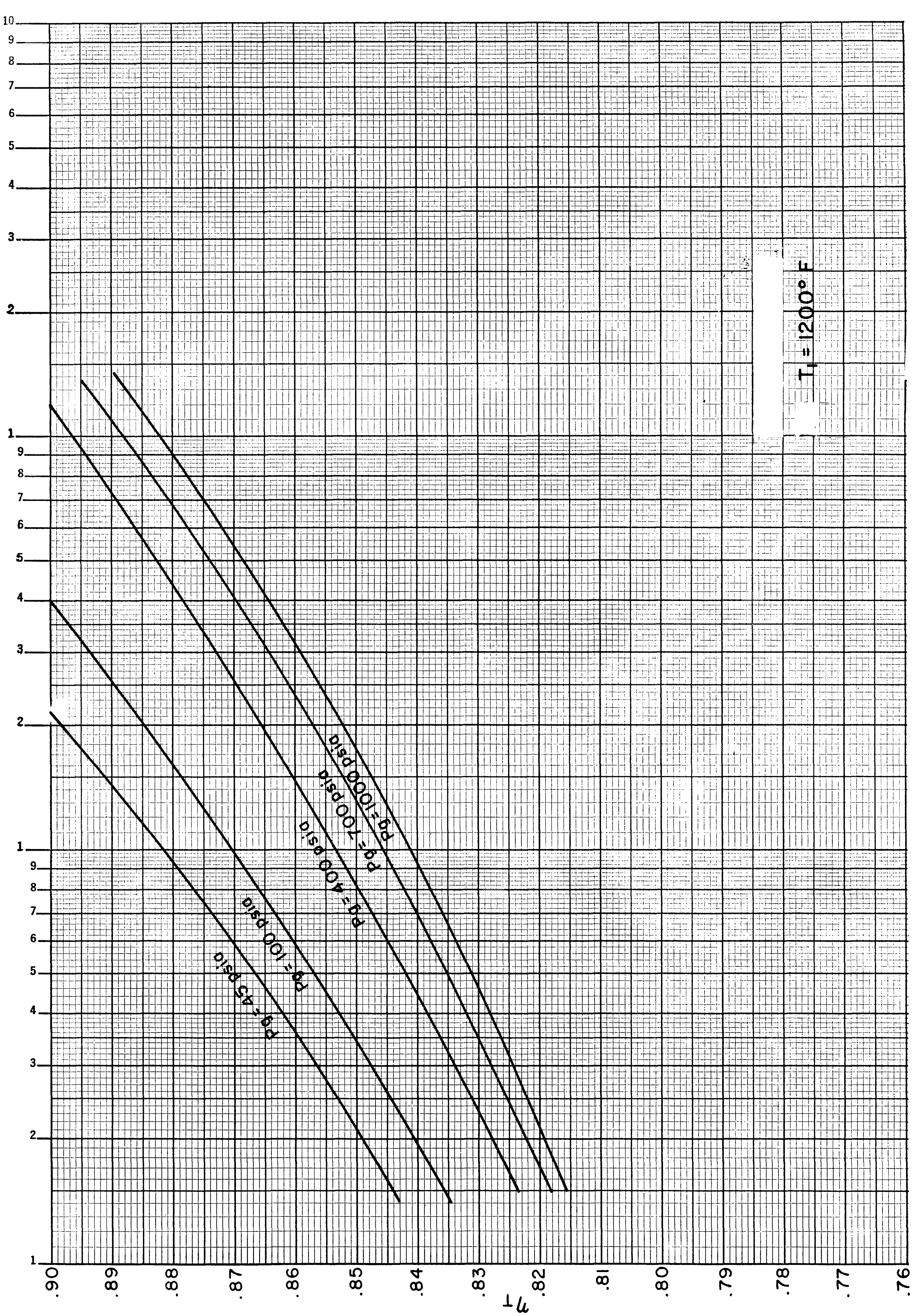

.90
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.77
.76
η_T
Pq= 45psia
Pq=100 psia
Pq=400 psia
Pq=700psia
Pq=1000psia
T_1=900°F
10^2
10^3
10^4
10^5
10^6
TURBINE — HP

Figure 22.



TURBINE EFFICIENCY FOR CO_2
POWER PLANTS OF VARIOUS
TURBINE HORSEPOWER
T_1 = 1500° F
Pg = 45 psia
Pg = 100 psia
Pg = 400 psia
Pg = 700 psia
Pg = 1000 psia
η_T
.90
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.77
.76
10^3
10^4
10^5
10^6
TURBINE HP

Figure 23.



$T_1 = 1200°$ F
Pg = 1000 psia
Pg = 700 psia
Pg = 400 psia
Pg = 100 psia
Pg = 45 psia
TURBINE HP
η_T

Figure 24.

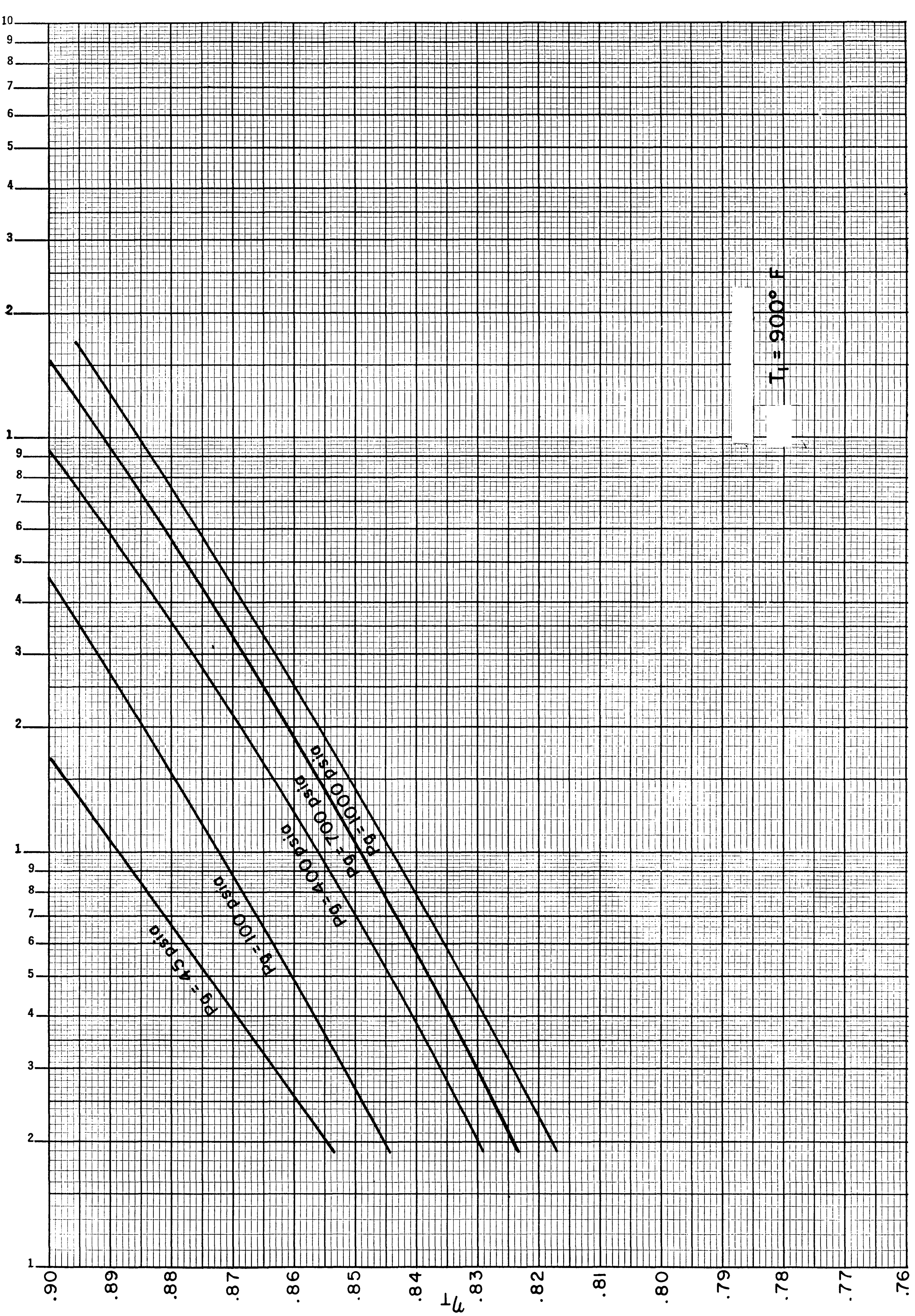
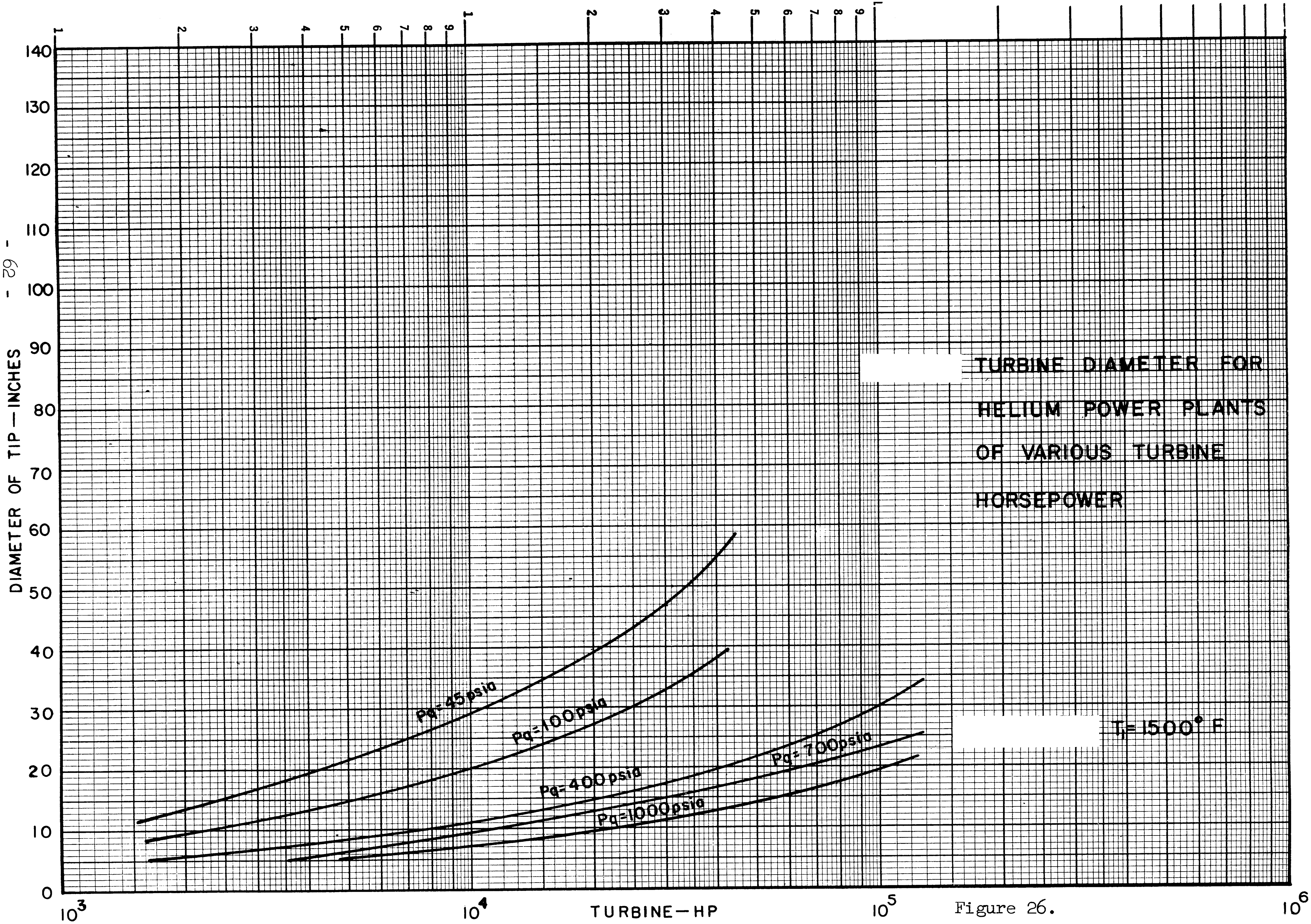
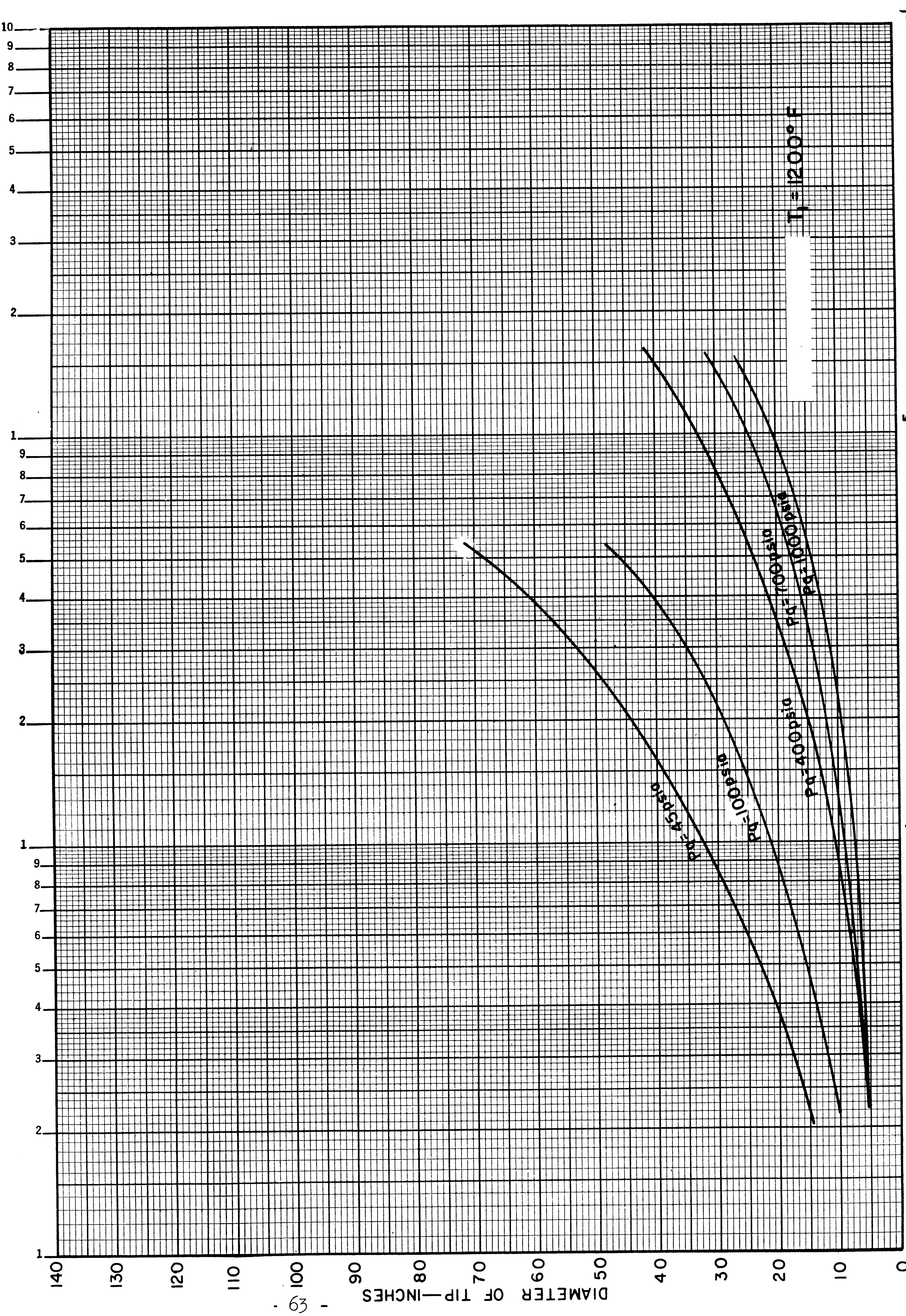

$T_1 = 900° F$
Pg = 1000 psia
Pg = 700 psia
Pg = 400 psia
Pg = 100 psia
Pg = 45 psia
TURBINE HP
10^3
10^4
10^5
10^6
η_T
.90
.89
.88
.87
.86
.85
.84
.83
.82
.81
.80
.79
.78
.77
.76

Figure 25.



TURBINE DIAMETER FOR
HELIUM POWER PLANTS
OF VARIOUS TURBINE
HORSEPOWER
T1=1500° F
Pq=45psia
Pq=100psia
Pq=400psia
Pq=700psia
Pq=1000psia
DIAMETER OF TIP—INCHES
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
TURBINE—HP
10^3
10^4
10^5
10^6

Figure 26.



$T_1 = 1200°F$
Pa=45psia
Pa=100psia
Pa=400psia
Pa=700psia
Pa=1000psia
TURBINE — HP
DIAMETER OF TIP—INCHES
10^3
10^4
10^5
10^6
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0

Figure 27.

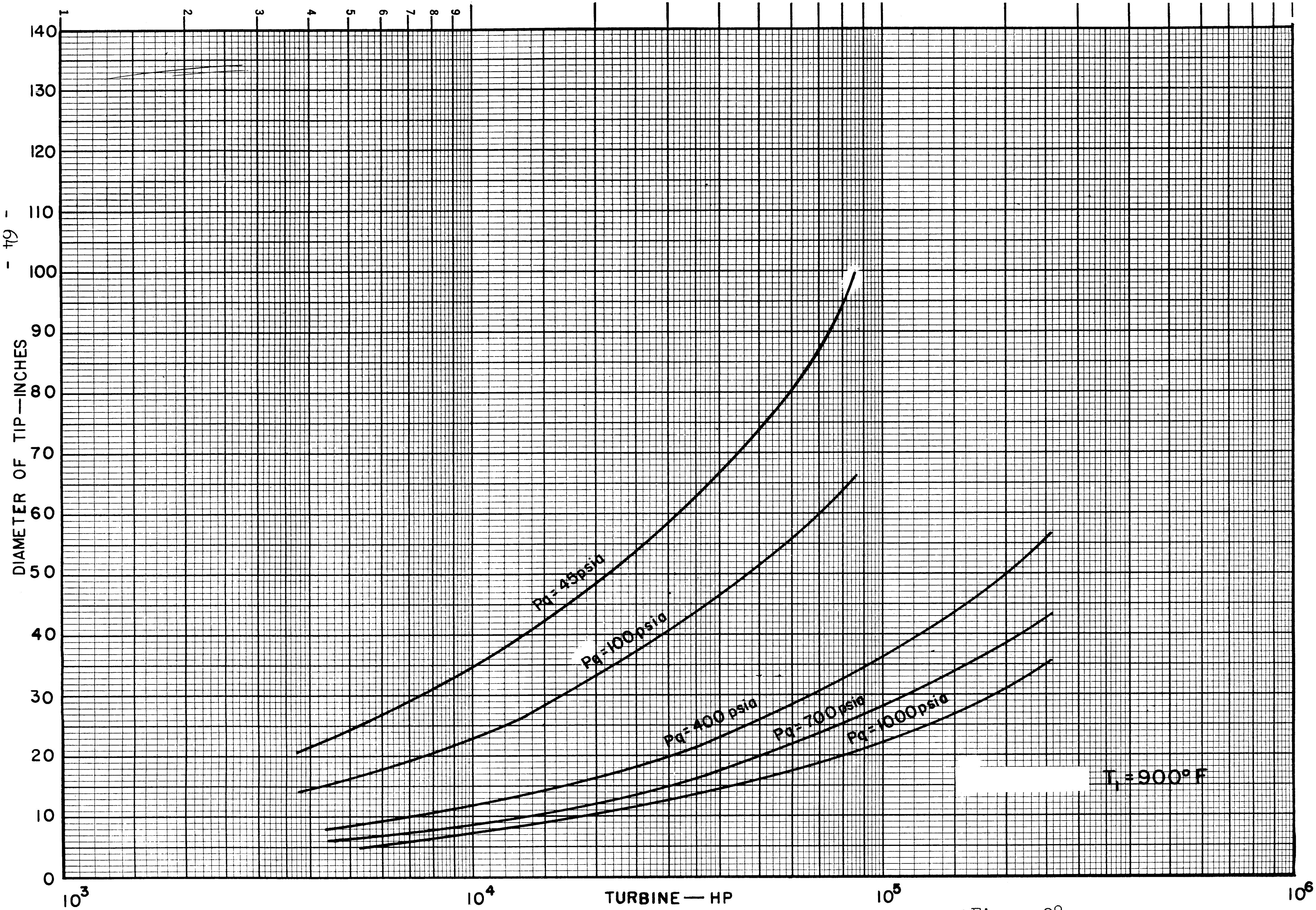

DIAMETER OF TIP—INCHES
TURBINE — HP
Pa = 45 psia
Pa = 100 psia
Pa = 400 psia
Pa = 700 psia
Pa = 1000 psia
$T_1 = 900°F$

Figure 28.

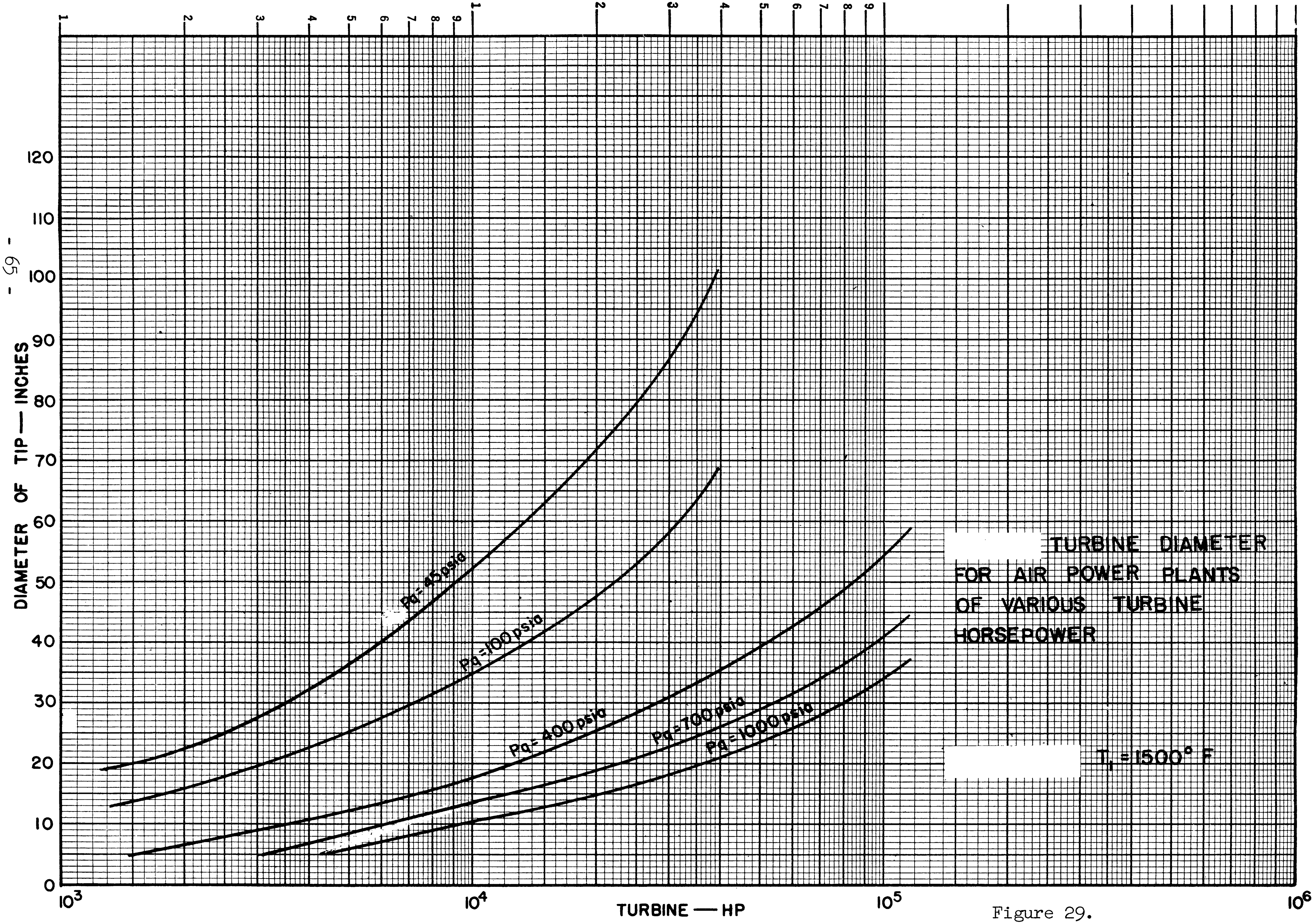
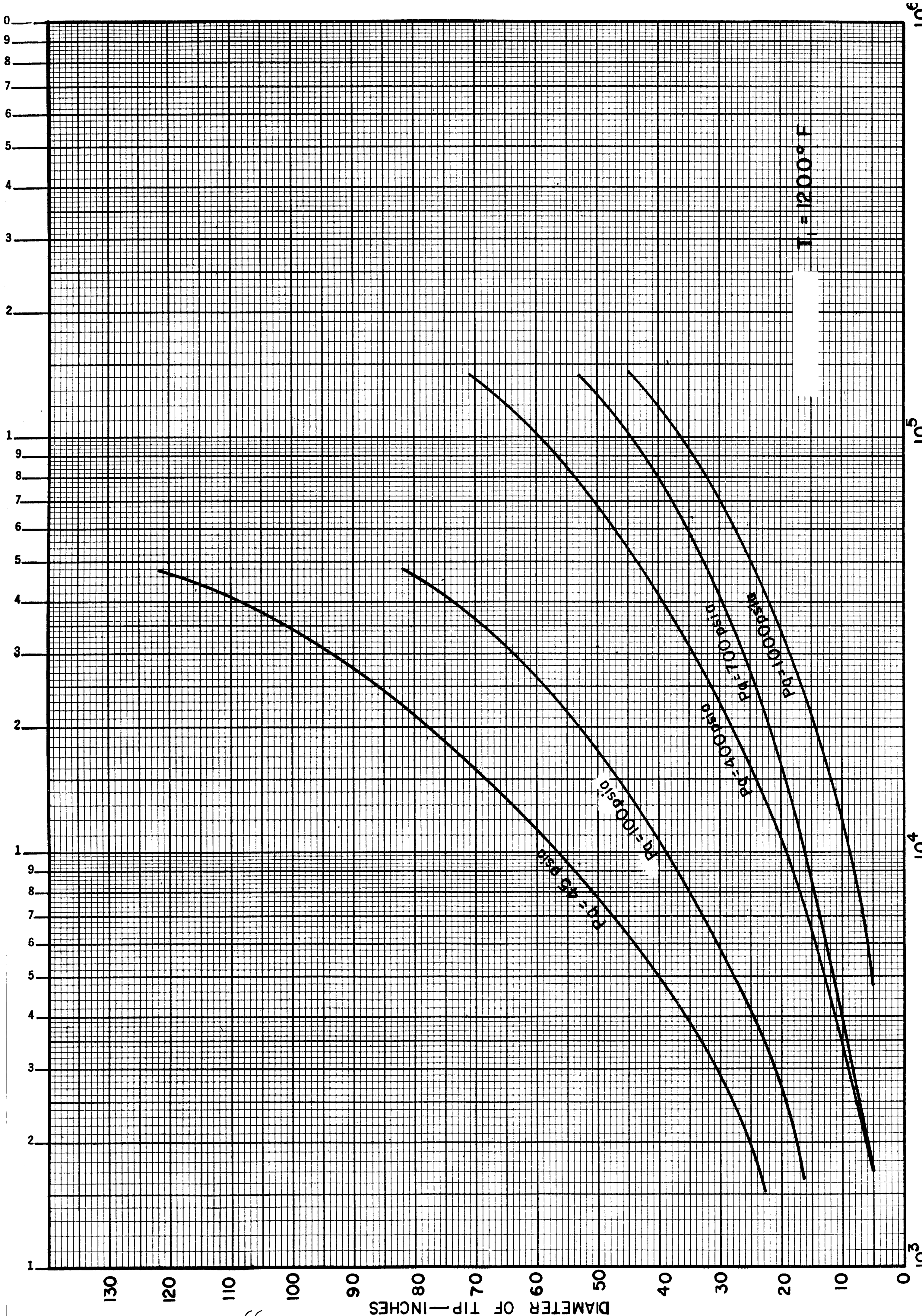

TURBINE DIAMETER
FOR AIR POWER PLANTS
OF VARIOUS TURBINE
HORSEPOWER
T1 = 1500° F
Pq = 45 psia
Pq = 100 psia
Pq = 400 psia
Pq = 700 psia
Pq = 1000 psia
DIAMETER OF TIP — INCHES
TURBINE — HP
0
10
20
30
40
50
60
70
80
90
100
110
120
10³
10⁴
10⁵
10⁶

Figure 29.



T_1 = 1200° F
Pd = 1000 psia
Pd = 700 psia
Pd = 400 psia
Pd = 100 psia
Pd = 45 psia
TURBINE—HP
DIAMETER OF TIP—INCHES
10^3
10^4
10^5
10^6
0
10
20
30
40
50
60
70
80
90
100
110
120
130

Figure 30.

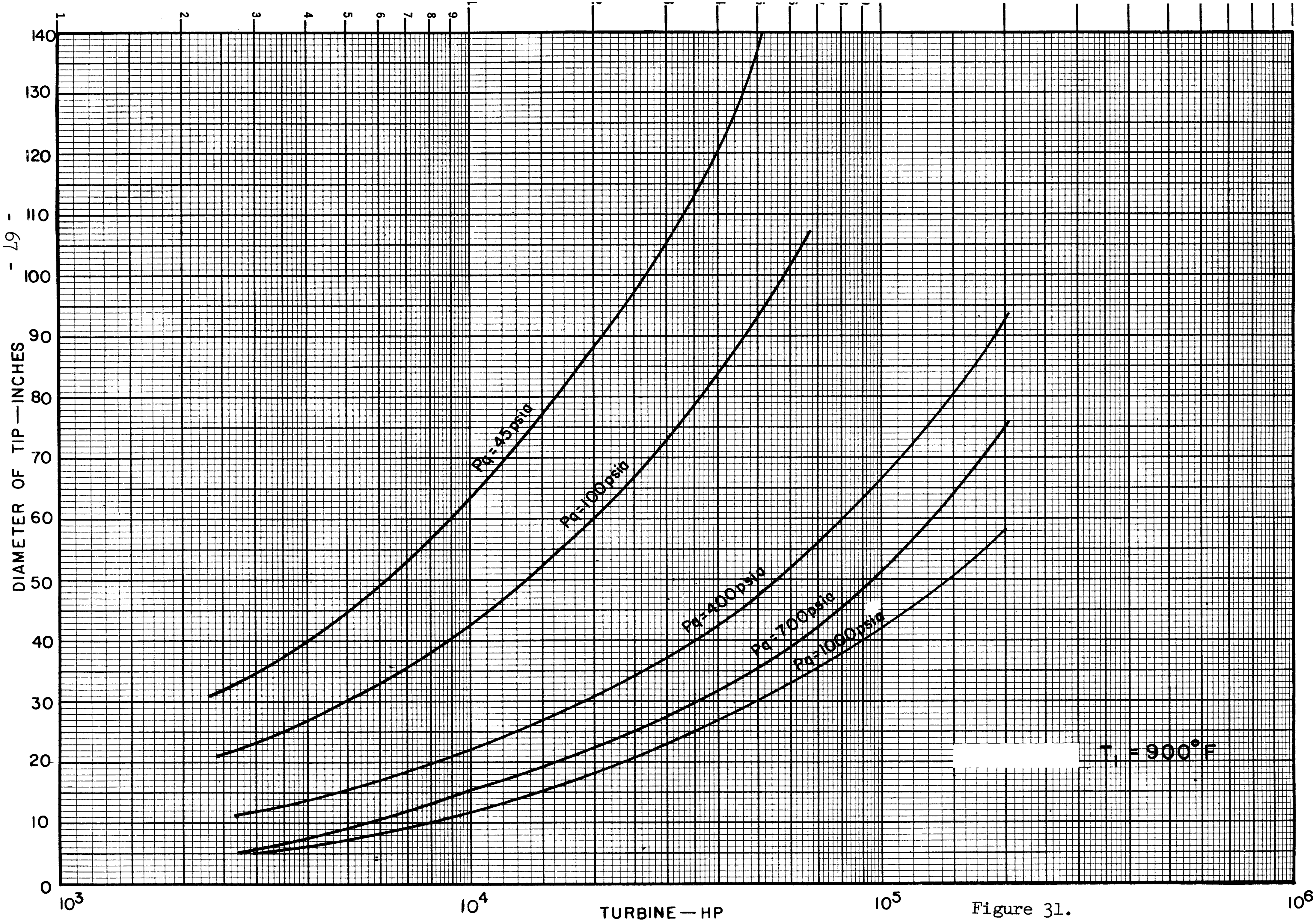
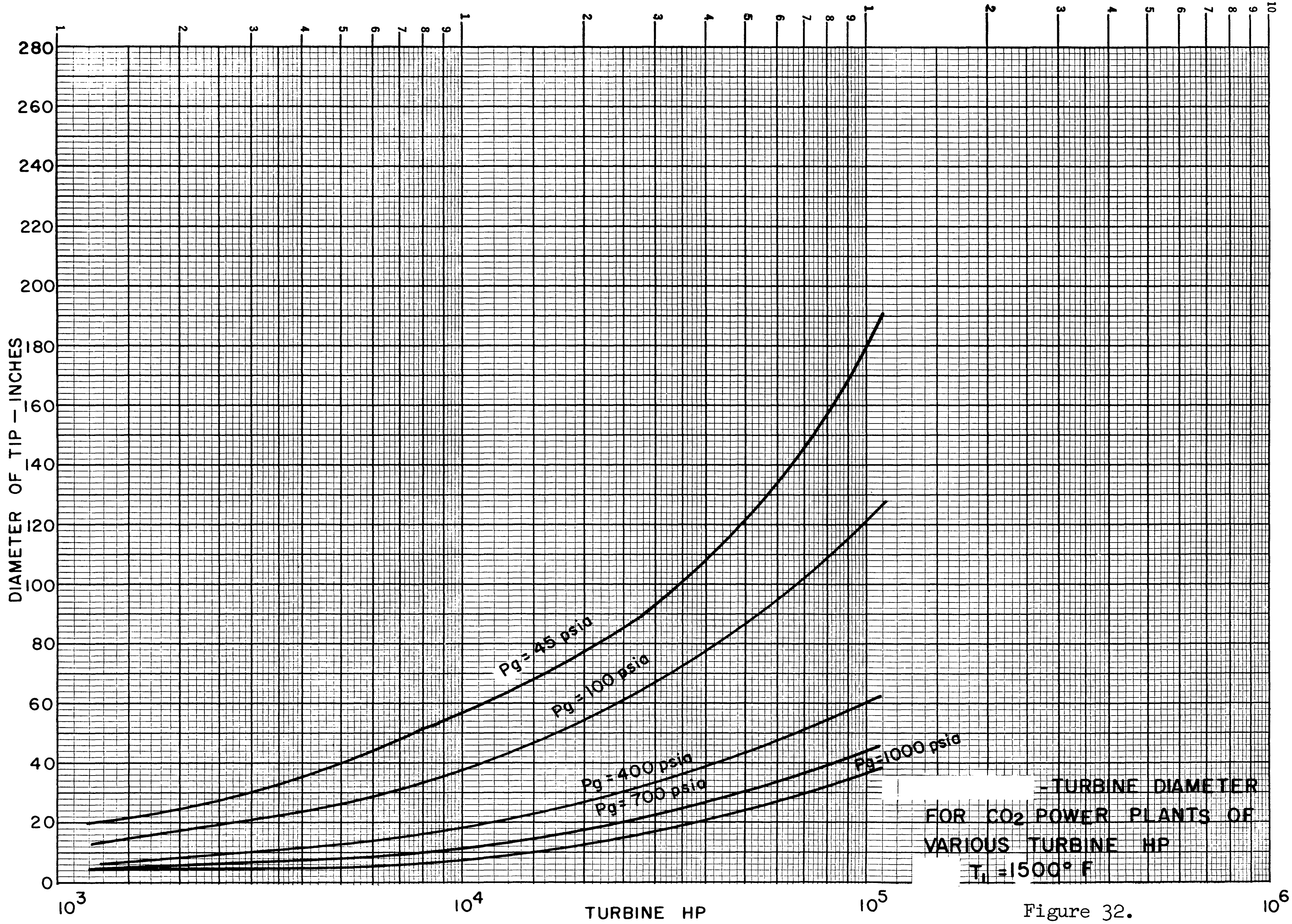

DIAMETER OF TIP—INCHES
TURBINE—HP
10^3
10^4
10^5
10^6
Pa=45psia
Pa=100psia
Pa=400psia
Pa=700psia
Pa=1000psia
T_1=900°F

Figure 31.



DIAMETER OF TIP — INCHES
TURBINE HP
Pg = 45 psia
Pg = 100 psia
Pg = 400 psia
Pg = 700 psia
Pg = 1000 psia
-TURBINE DIAMETER
FOR CO_2 POWER PLANTS OF
VARIOUS TURBINE HP
T_1 = 1500° F

Figure 32.

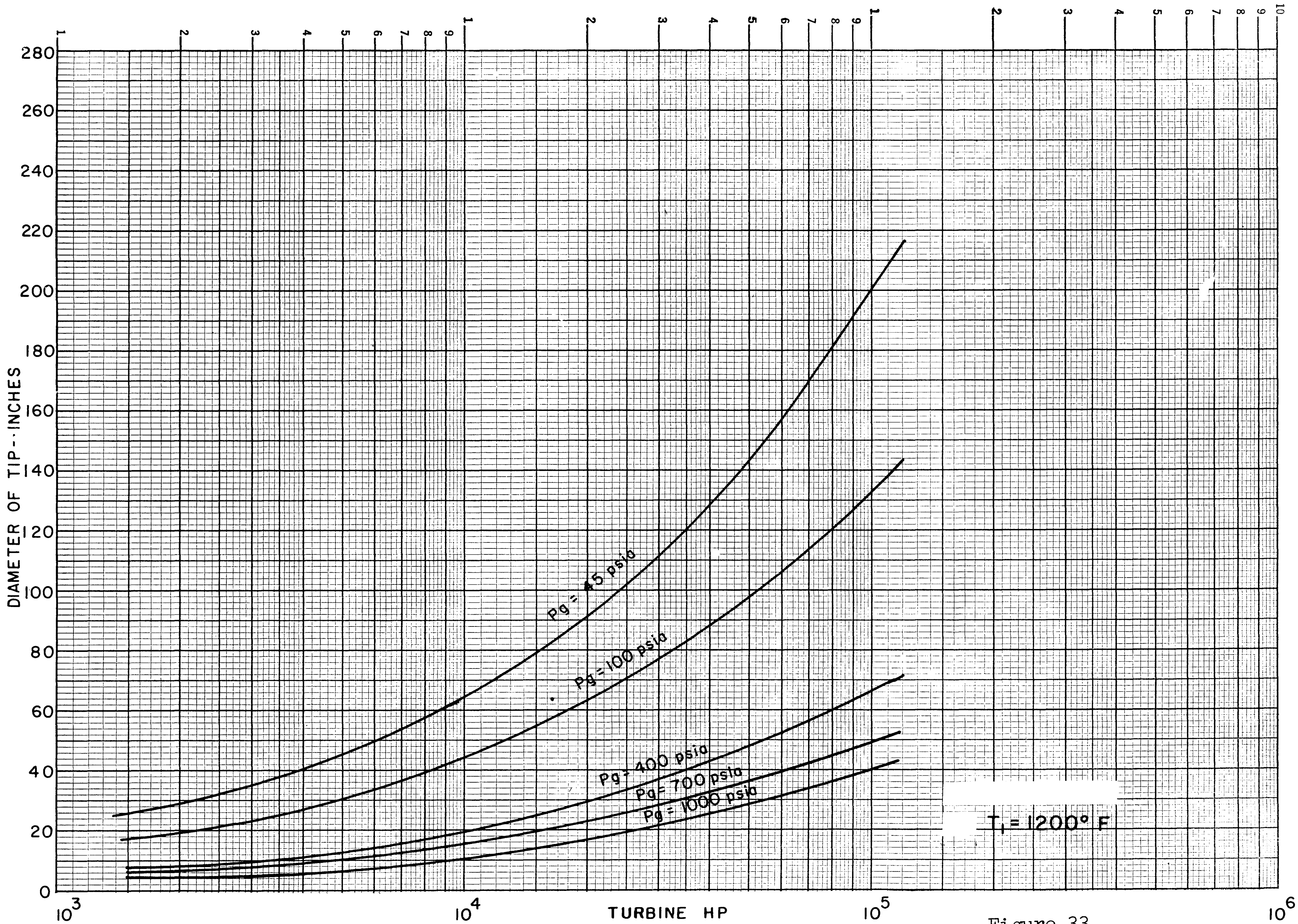

DIAMETER OF TIP--INCHES
TURBINE HP
Pg = 45 psia
Pg = 100 psia
Pg = 400 psia
Pg = 700 psia
Pg = 1000 psia
T1 = 1200° F

Figure 33.

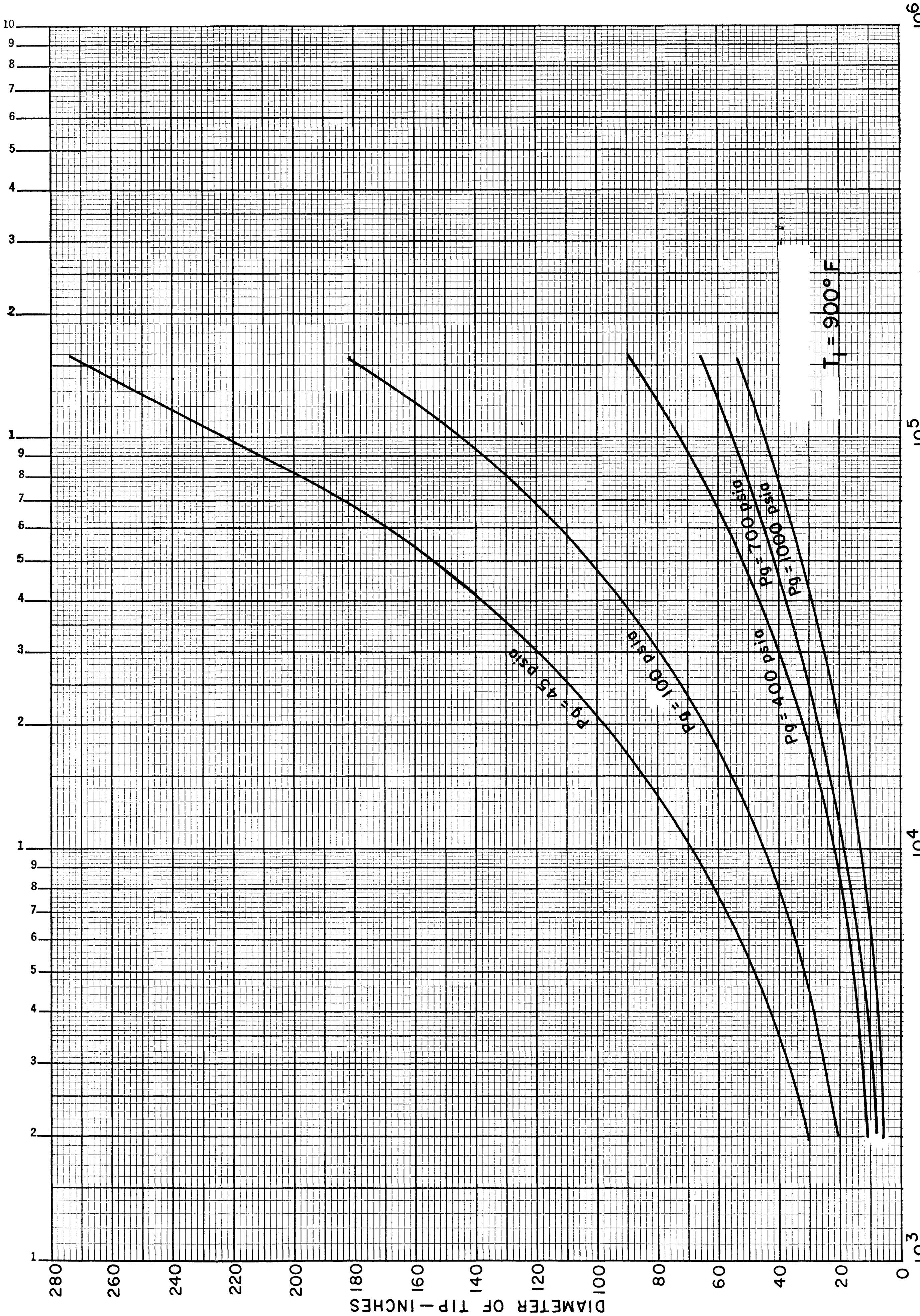
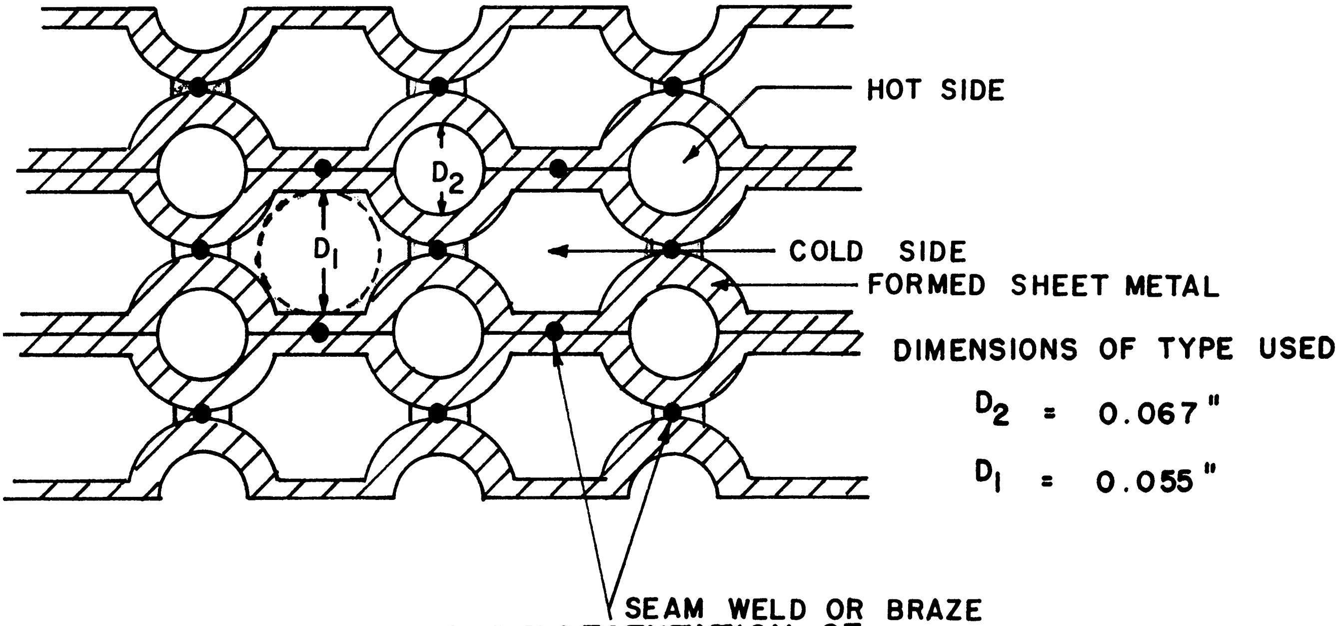

T_1 = 900°F
Pg = 45 psia
Pg = 100 psia
Pg = 400 psia
Pg = 700 psia
Pg = 1000 psia
TURBINE HP
DIAMETER OF TIP—INCHES
10^3
10^4
10^5
10^6
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0

Figure 34.



SCHEMATIC REPRESENTATION OF GRISCOM-RUSSELL "ALL-PRIME SURFACE BRAZED LATTICE" FOR CLOSED-CYCLE GAS TURBINE REGENERATOR.

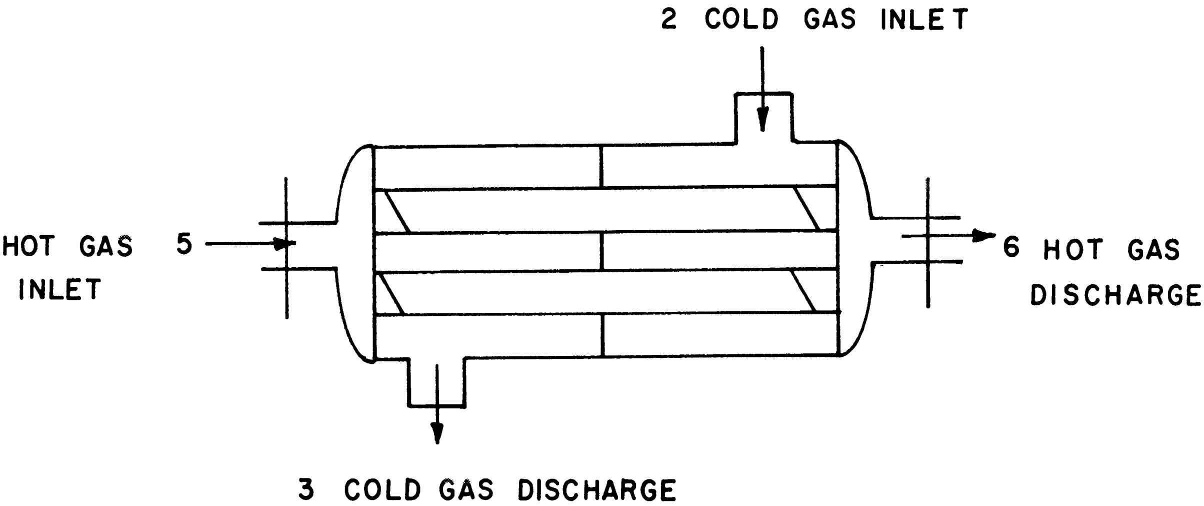


Figure 35. SCHEMATIC DIAGRAM FOR HEAT EXCHANGER

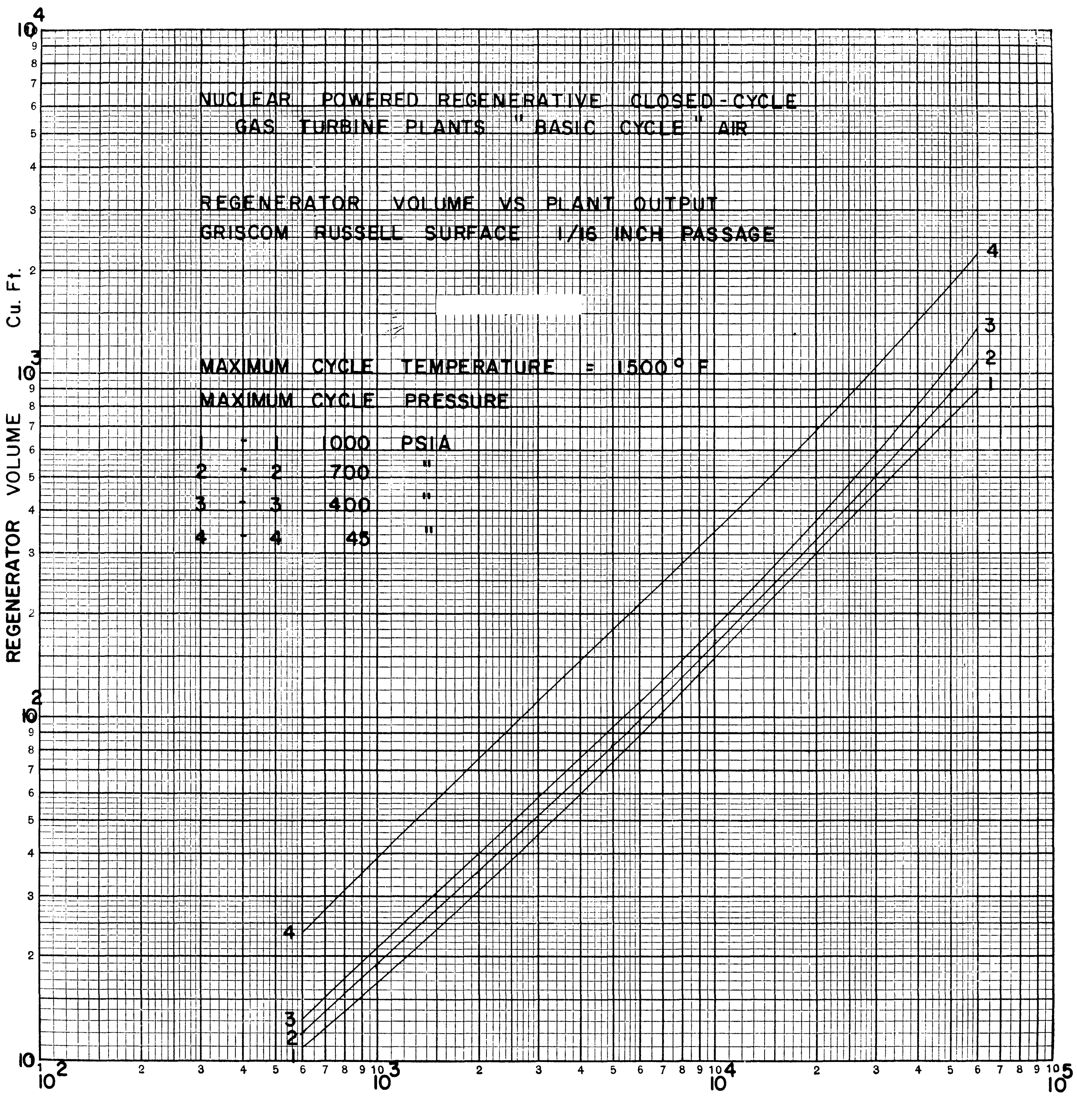


Figure 36. PLANT OUTPUT, HP

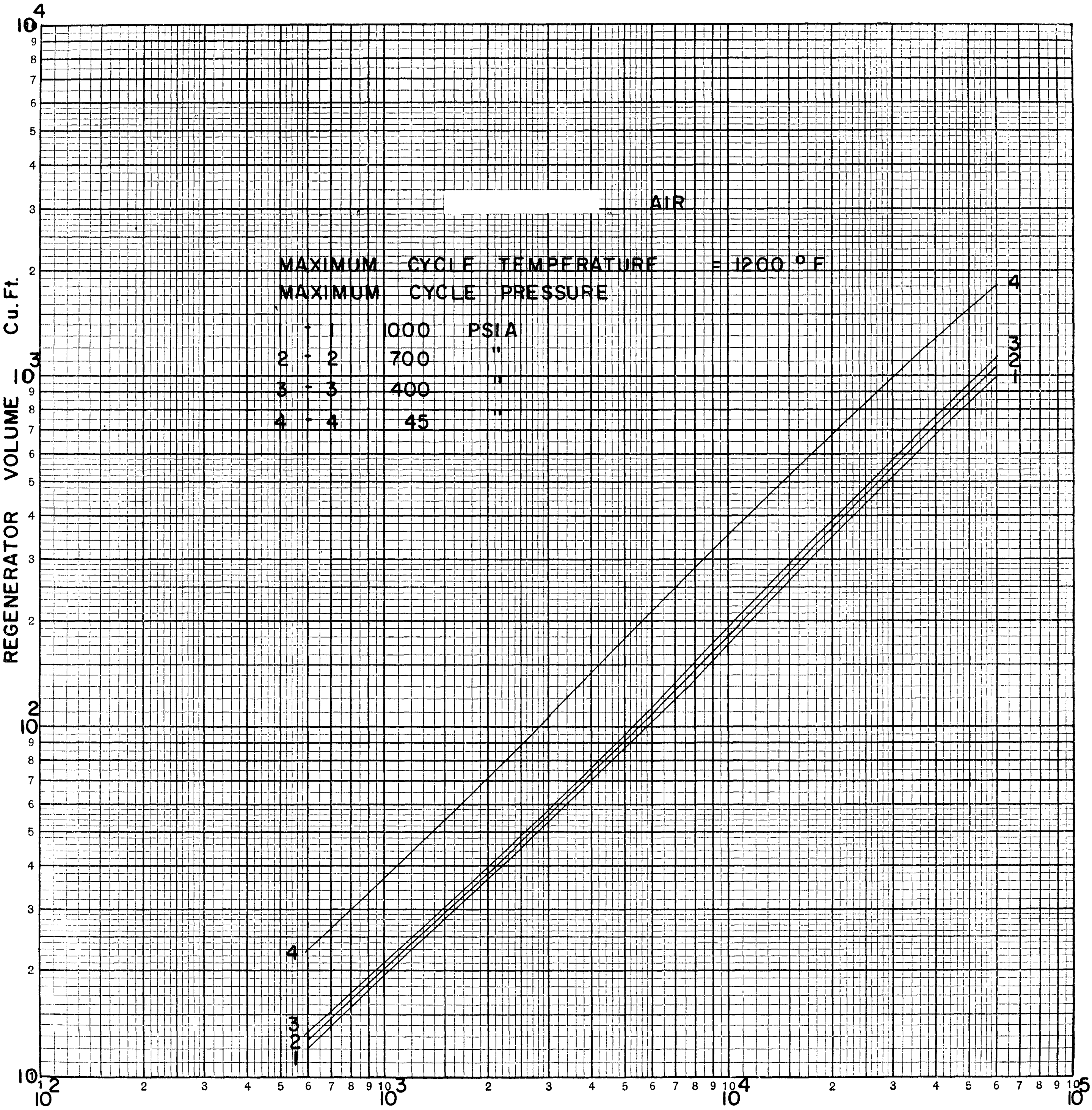


Figure 37. PLANT OUTPUT, HP

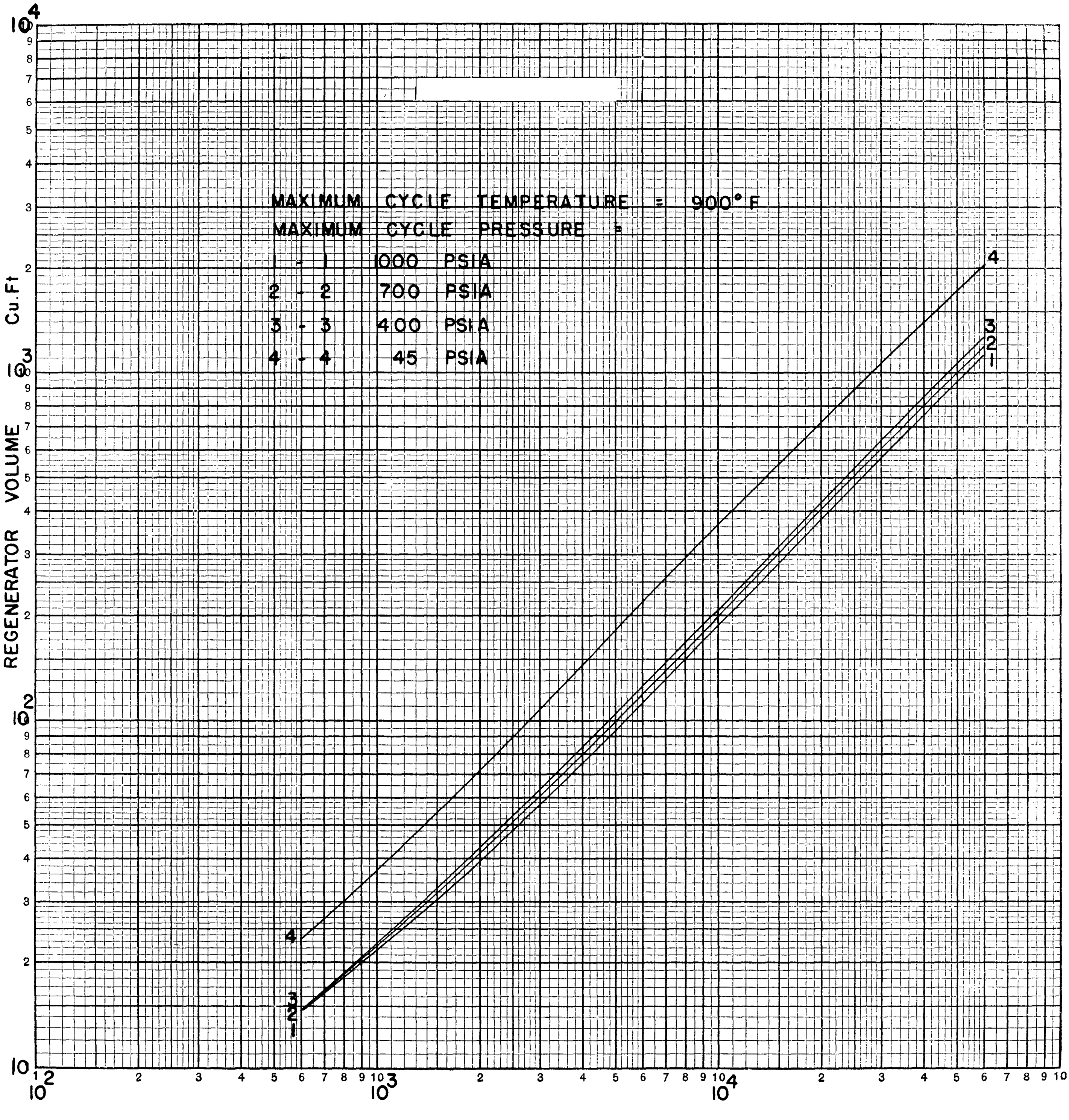

MAXIMUM CYCLE TEMPERATURE = 900° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR VOLUME Cu. Ft
PLANT OUTPUT, HP

Figure 38.

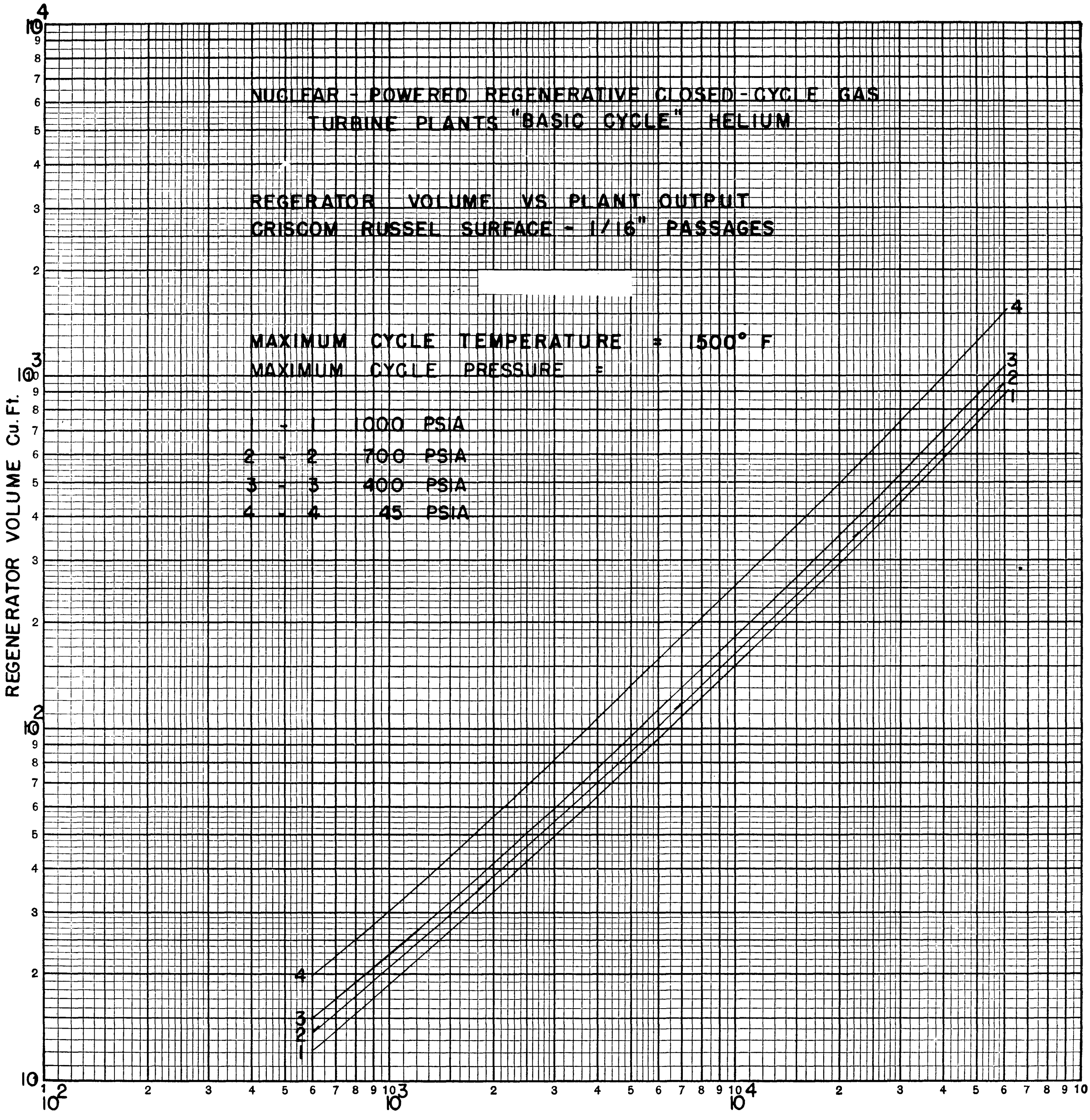

NUCLEAR - POWERED REGENERATIVE CLOSED-CYCLE GAS
TURBINE PLANTS "BASIC CYCLE" HELIUM
REGERATOR VOLUME VS PLANT OUTPUT
CRISCOM RUSSEL SURFACE - 1/16" PASSAGES
MAXIMUM CYCLE TEMPERATURE = 1500° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR VOLUME Cu. Ft.
PLANT OUTPUT , HP

Figure 39.

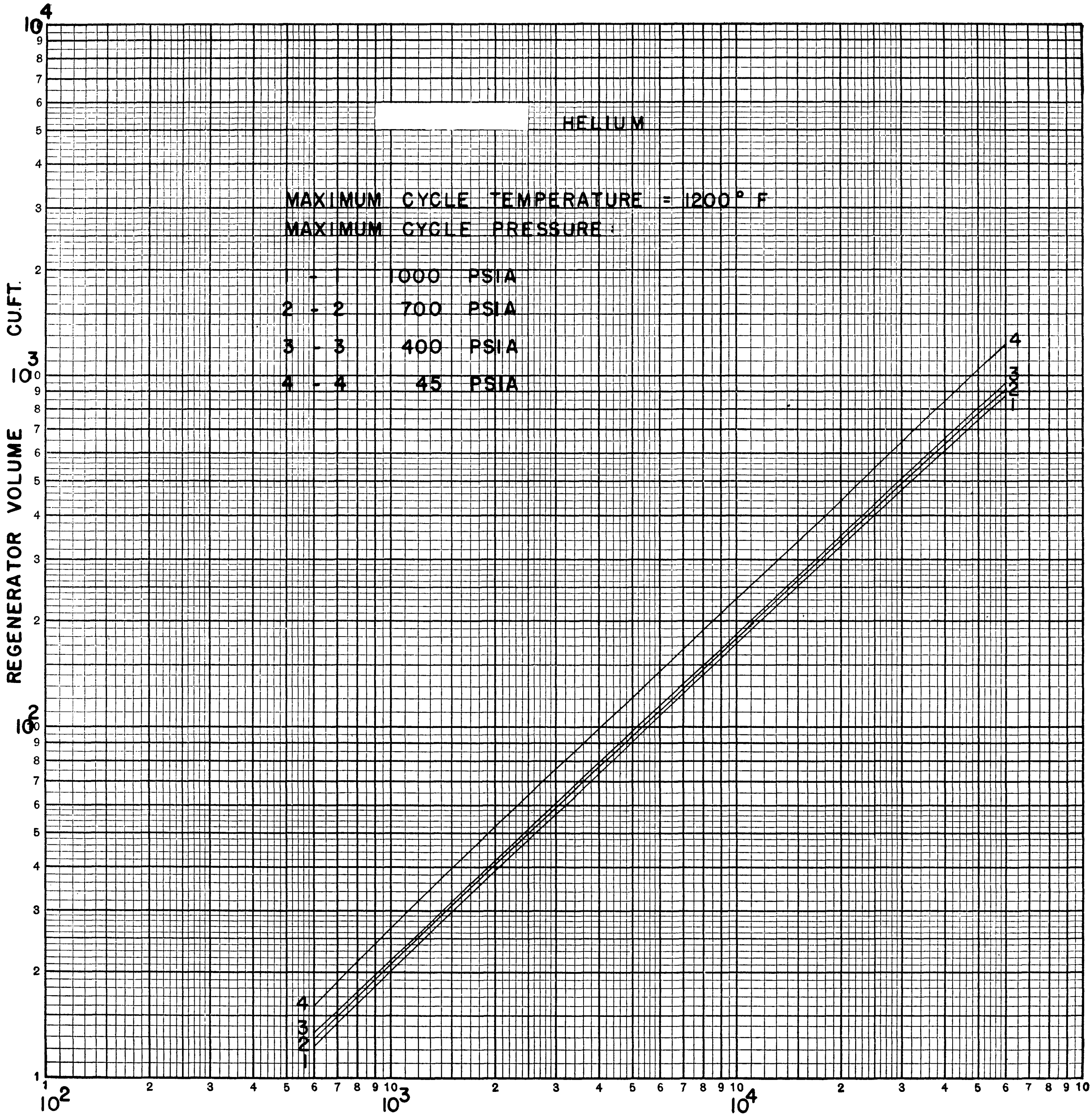


Figure 40. PLANT OUTPUT, HP

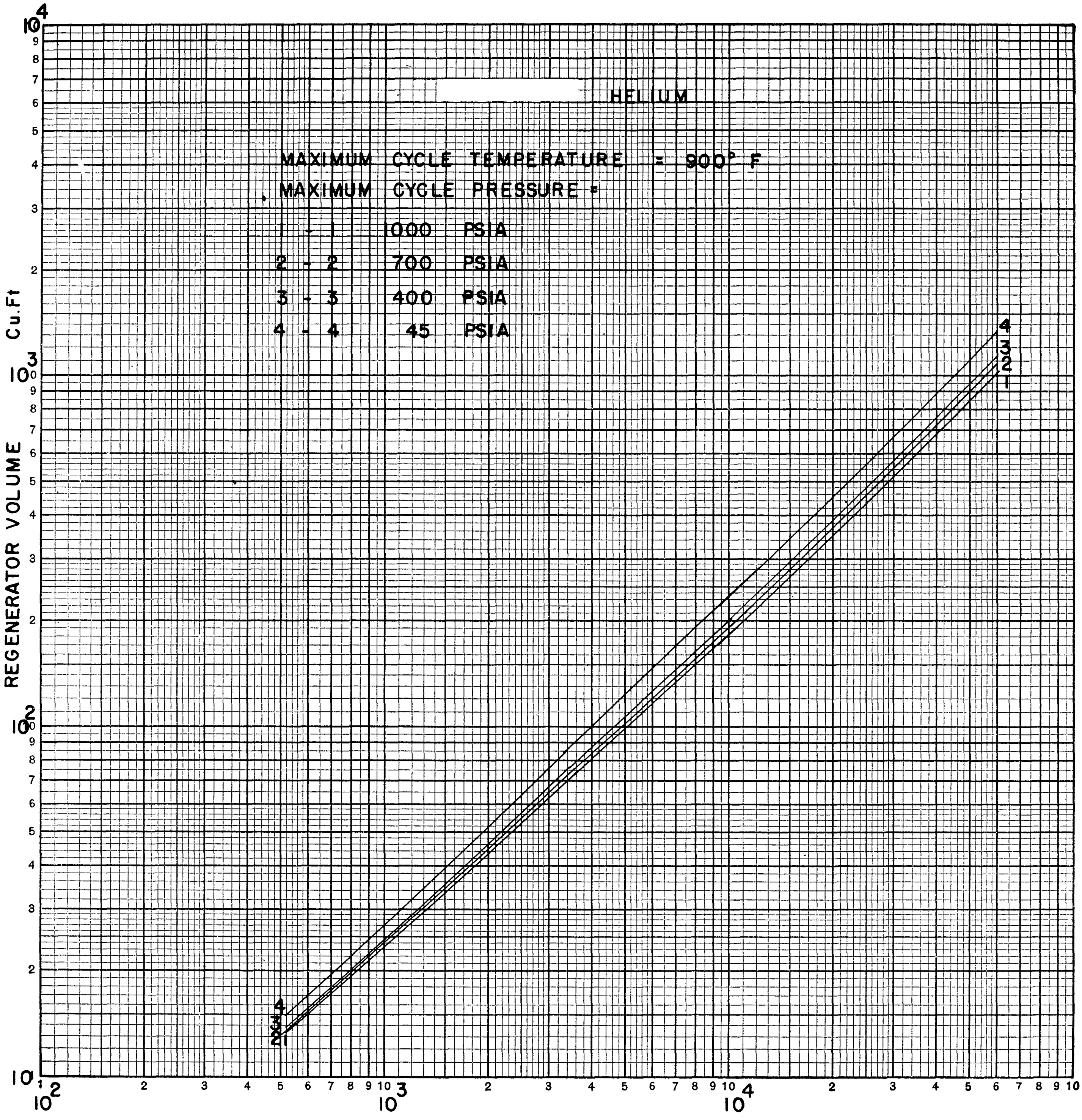


Figure 41. PLANT OUTPUT , HP

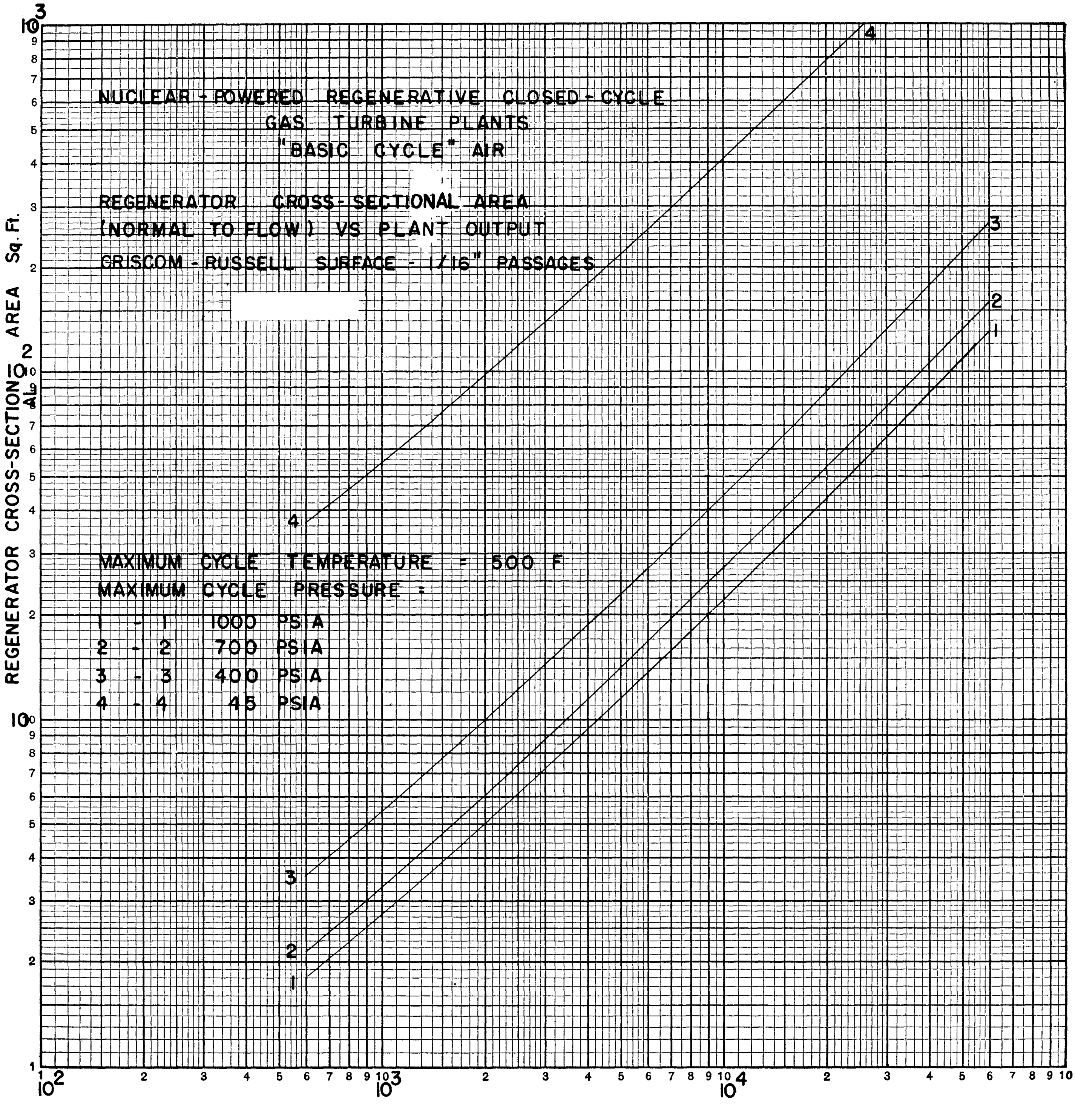

NUCLEAR - POWERED REGENERATIVE CLOSED - CYCLE
GAS TURBINE PLANTS
"BASIC CYCLE" AIR
REGENERATOR CROSS-SECTIONAL AREA
(NORMAL TO FLOW) VS PLANT OUTPUT
GRISCOM - RUSSELL SURFACE - 1/16" PASSAGES
MAXIMUM CYCLE TEMPERATURE = 1500 F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR CROSS-SECTIONAL AREA Sq. Ft.
PLANT OUTPUT, HP

Figure 42.

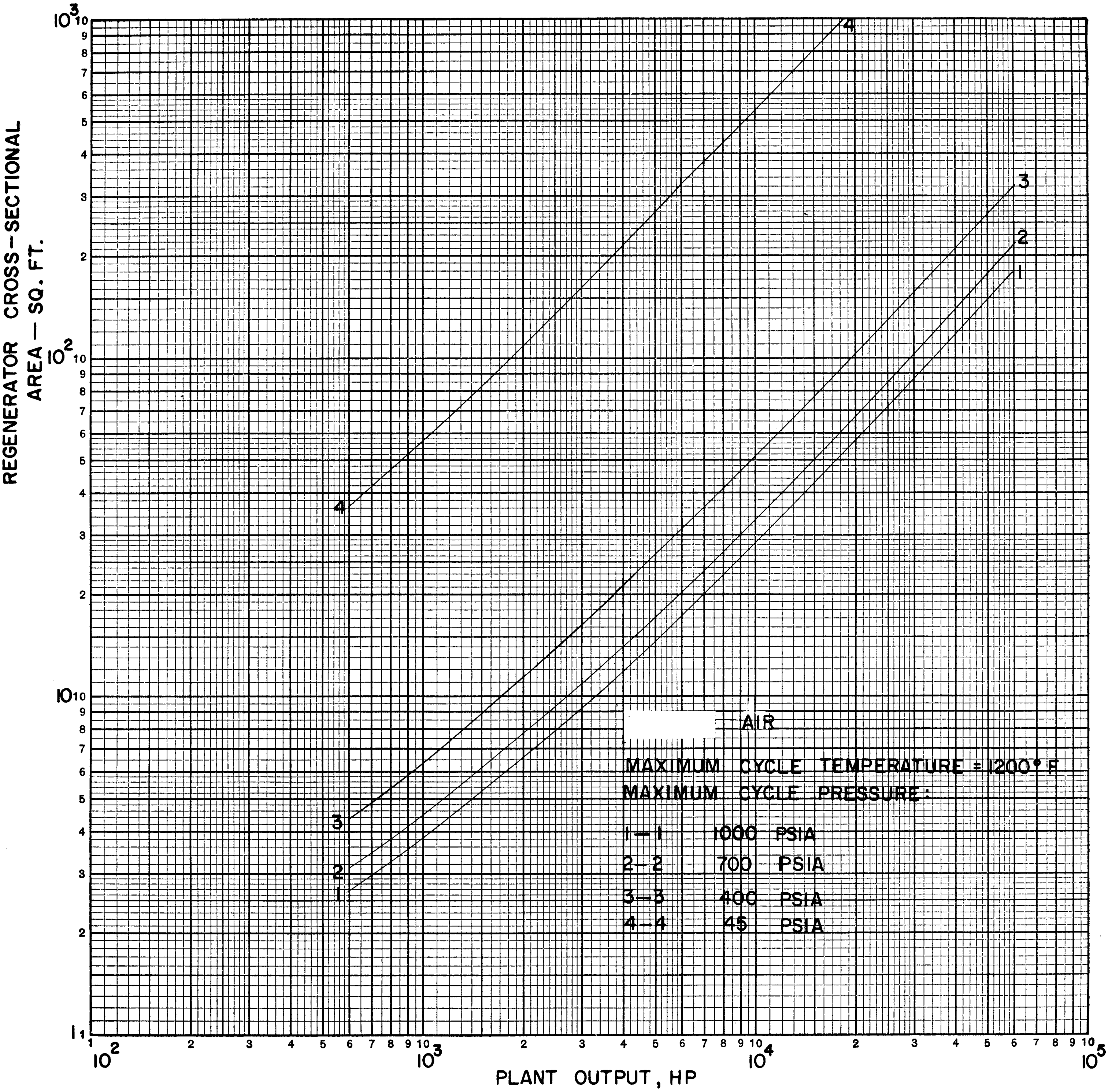

REGENERATOR CROSS-SECTIONAL AREA — SQ. FT.
PLANT OUTPUT, HP
AIR
MAXIMUM CYCLE TEMPERATURE = 1200° F
MAXIMUM CYCLE PRESSURE:
1—1 1000 PSIA
2—2 700 PSIA
3—3 400 PSIA
4—4 45 PSIA

Figure 43.

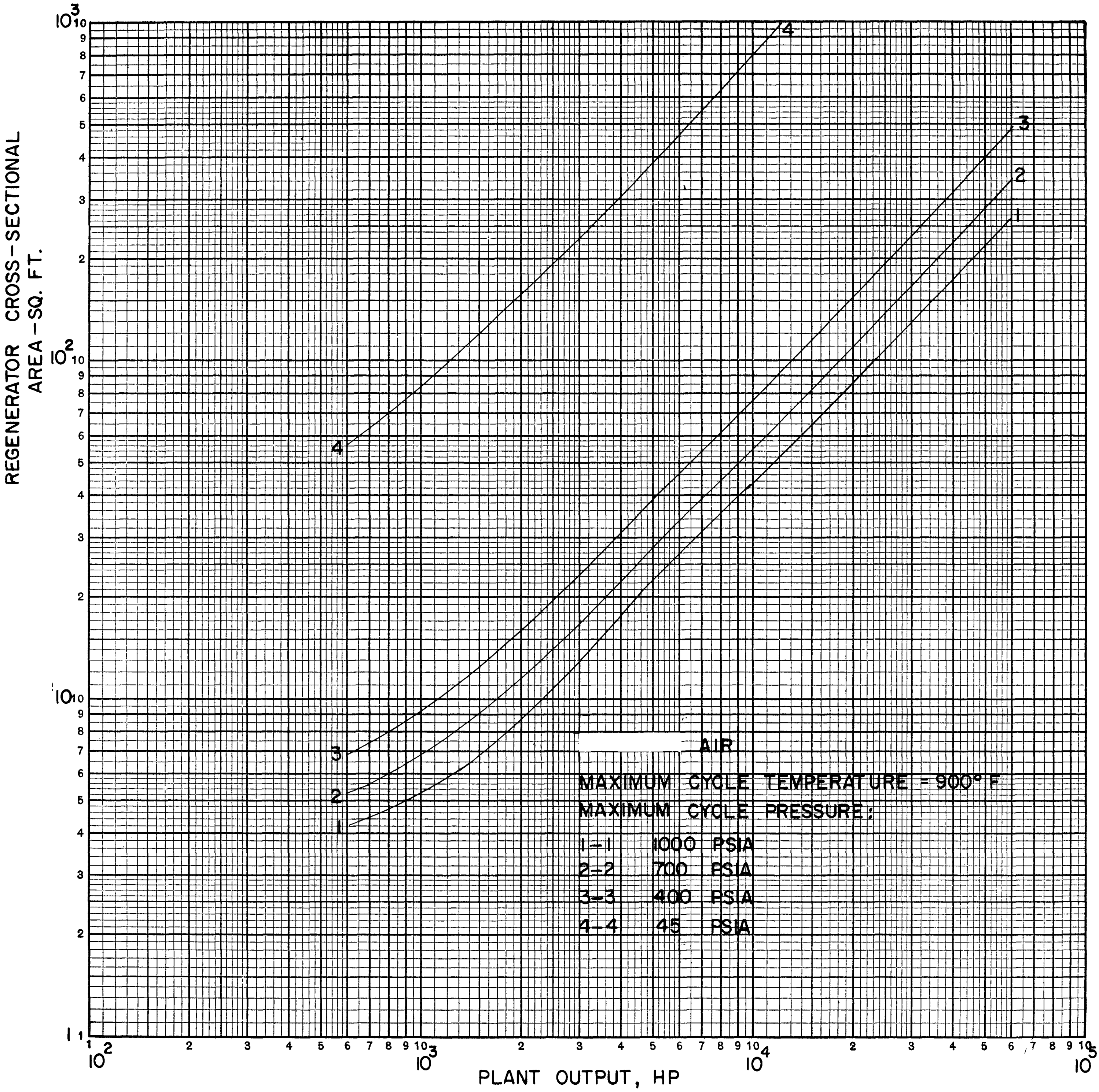

REGENERATOR CROSS-SECTIONAL AREA-SQ. FT.
PLANT OUTPUT, HP
AIR
MAXIMUM CYCLE TEMPERATURE = 900° F
MAXIMUM CYCLE PRESSURE:
1-1 1000 PSIA
2-2 700 PSIA
3-3 400 PSIA
4-4 45 PSIA

Figure 44.

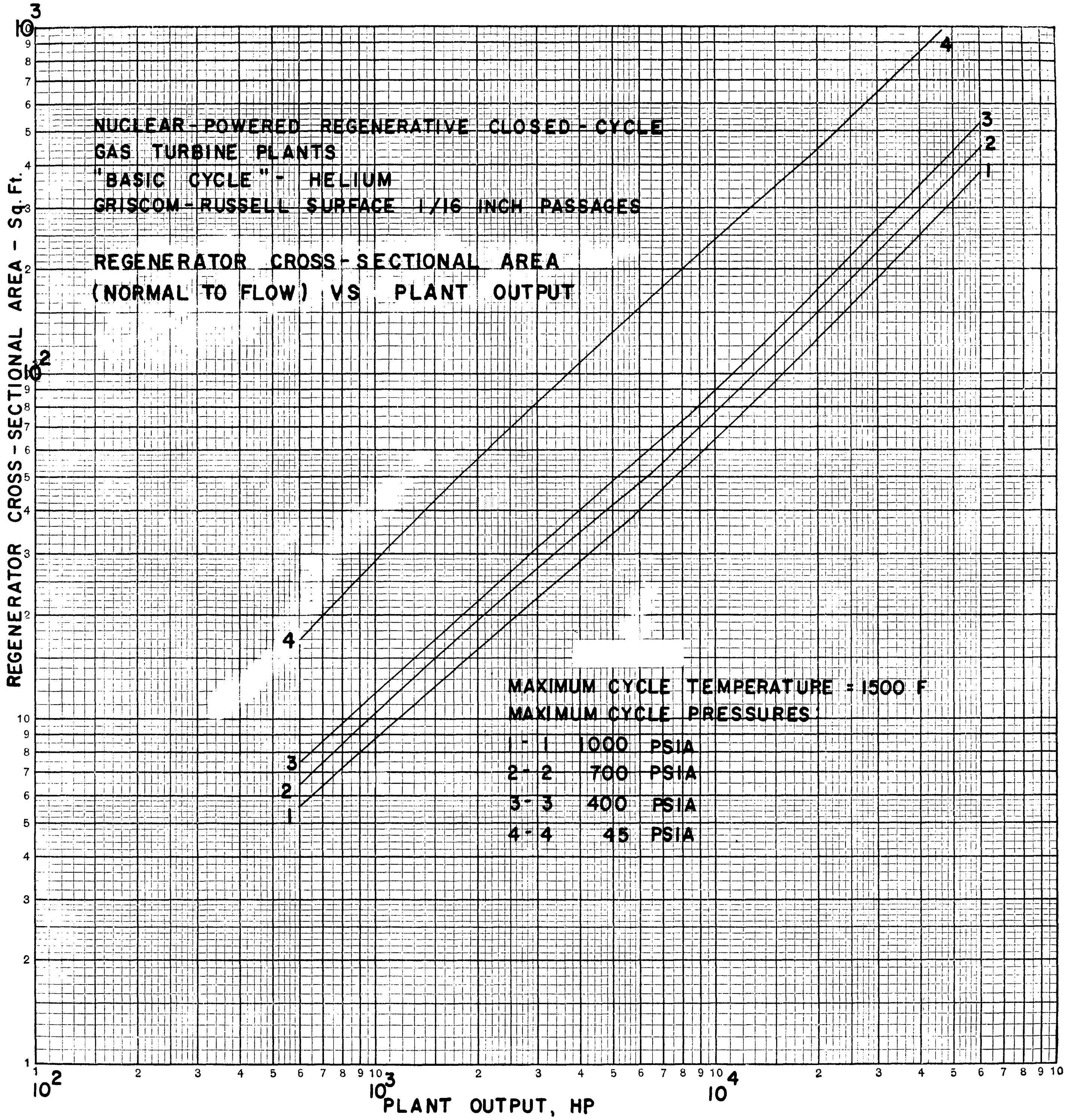

NUCLEAR-POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS
"BASIC CYCLE" - HELIUM
GRISCOM-RUSSELL SURFACE 1/16 INCH PASSAGES
REGENERATOR CROSS-SECTIONAL AREA
(NORMAL TO FLOW) VS PLANT OUTPUT
MAXIMUM CYCLE TEMPERATURE = 1500 F
MAXIMUM CYCLE PRESSURES:
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR CROSS-SECTIONAL AREA - Sq. Ft.
PLANT OUTPUT, HP

Figure 45.

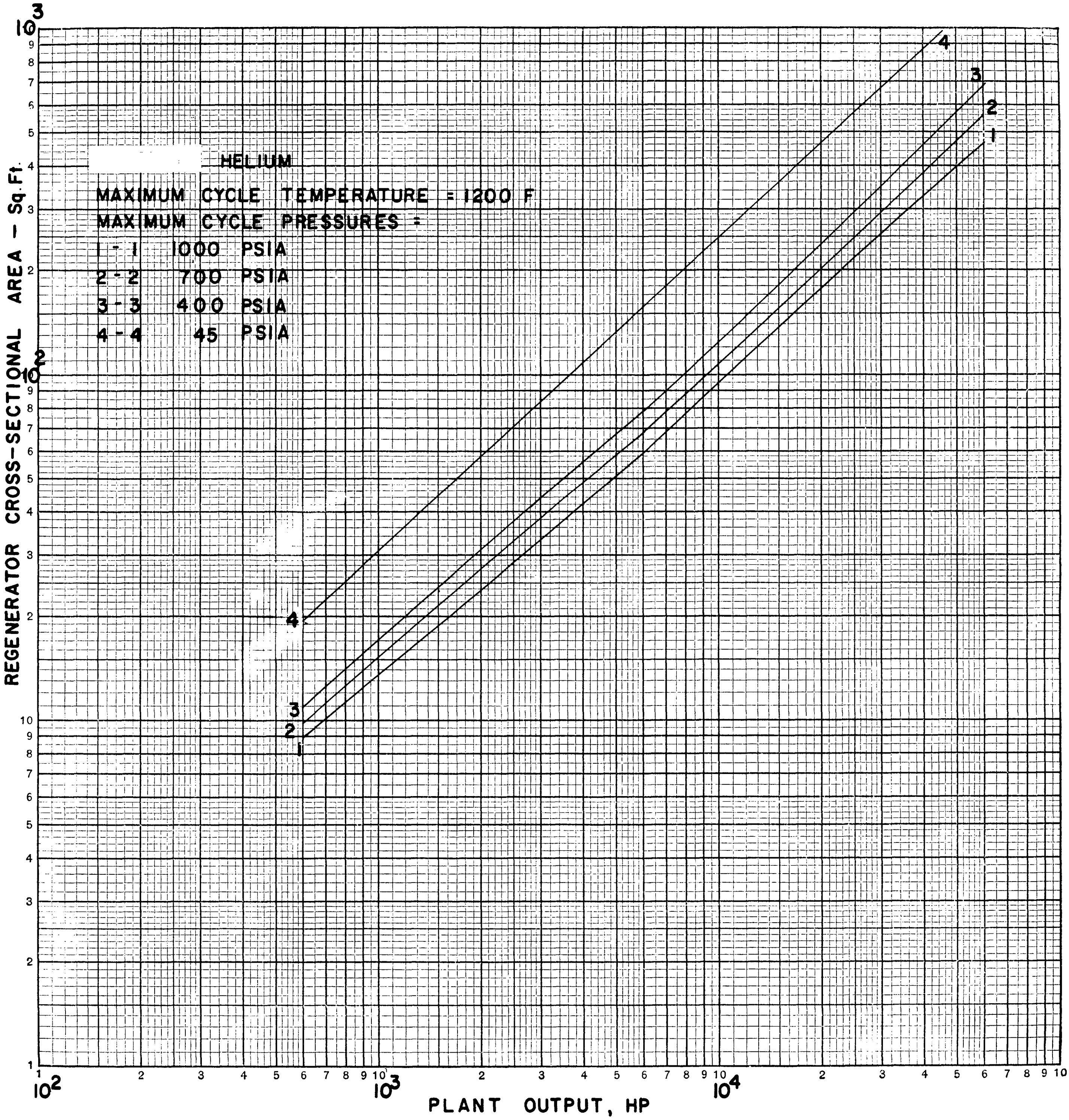
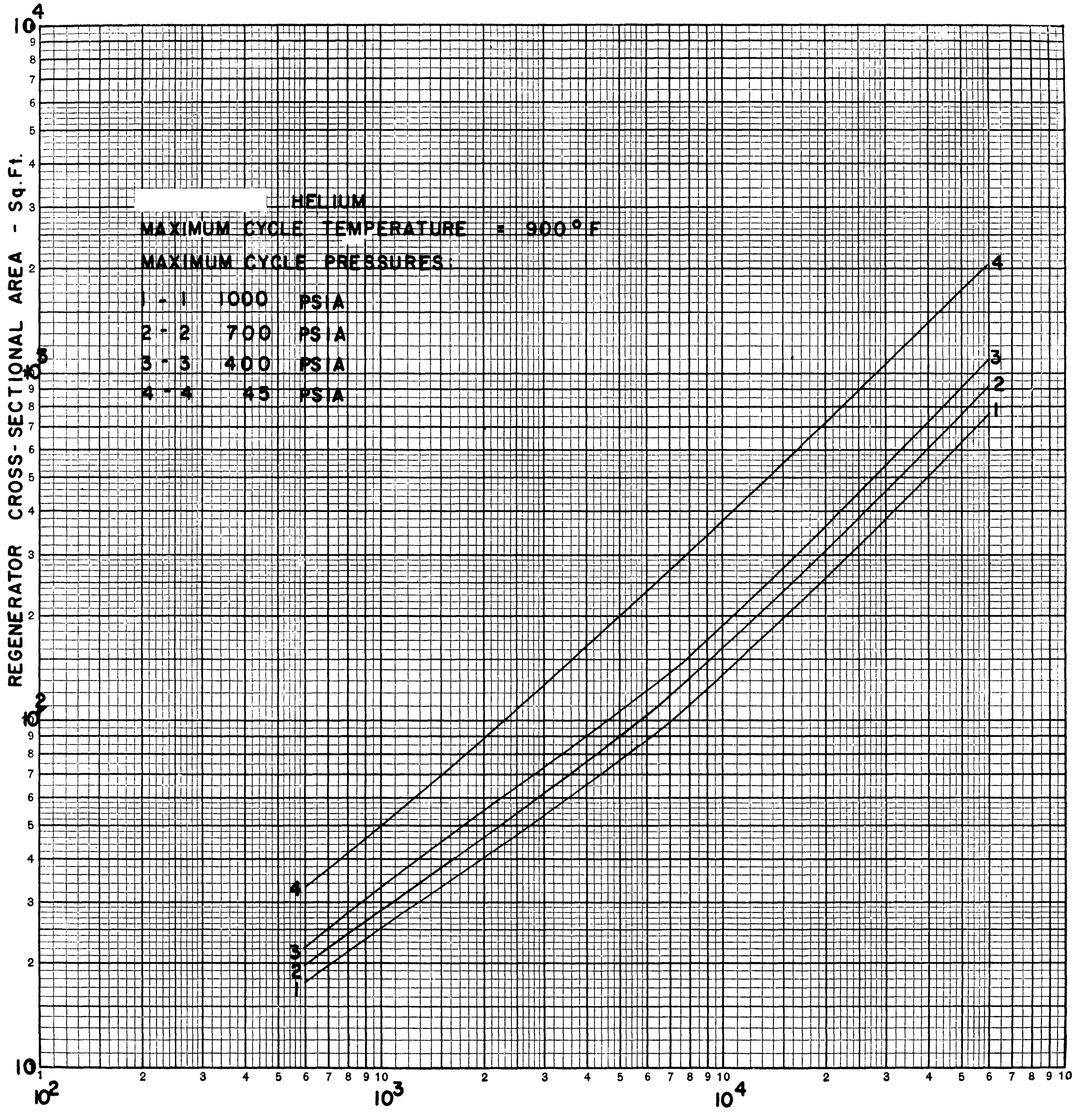

HELIUM
MAXIMUM CYCLE TEMPERATURE = 1200 F
MAXIMUM CYCLE PRESSURES =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR CROSS-SECTIONAL AREA - Sq. Ft.
PLANT OUTPUT, HP
10^2
10^3
10^4

Figure 46.



HELIUM
MAXIMUM CYCLE TEMPERATURE = 900°F
MAXIMUM CYCLE PRESSURES:
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR CROSS-SECTIONAL AREA - Sq. Ft.
10
10^2
10^3
10^4
1
2
3
4
PLANT OUTPUT, HP

Figure 47.

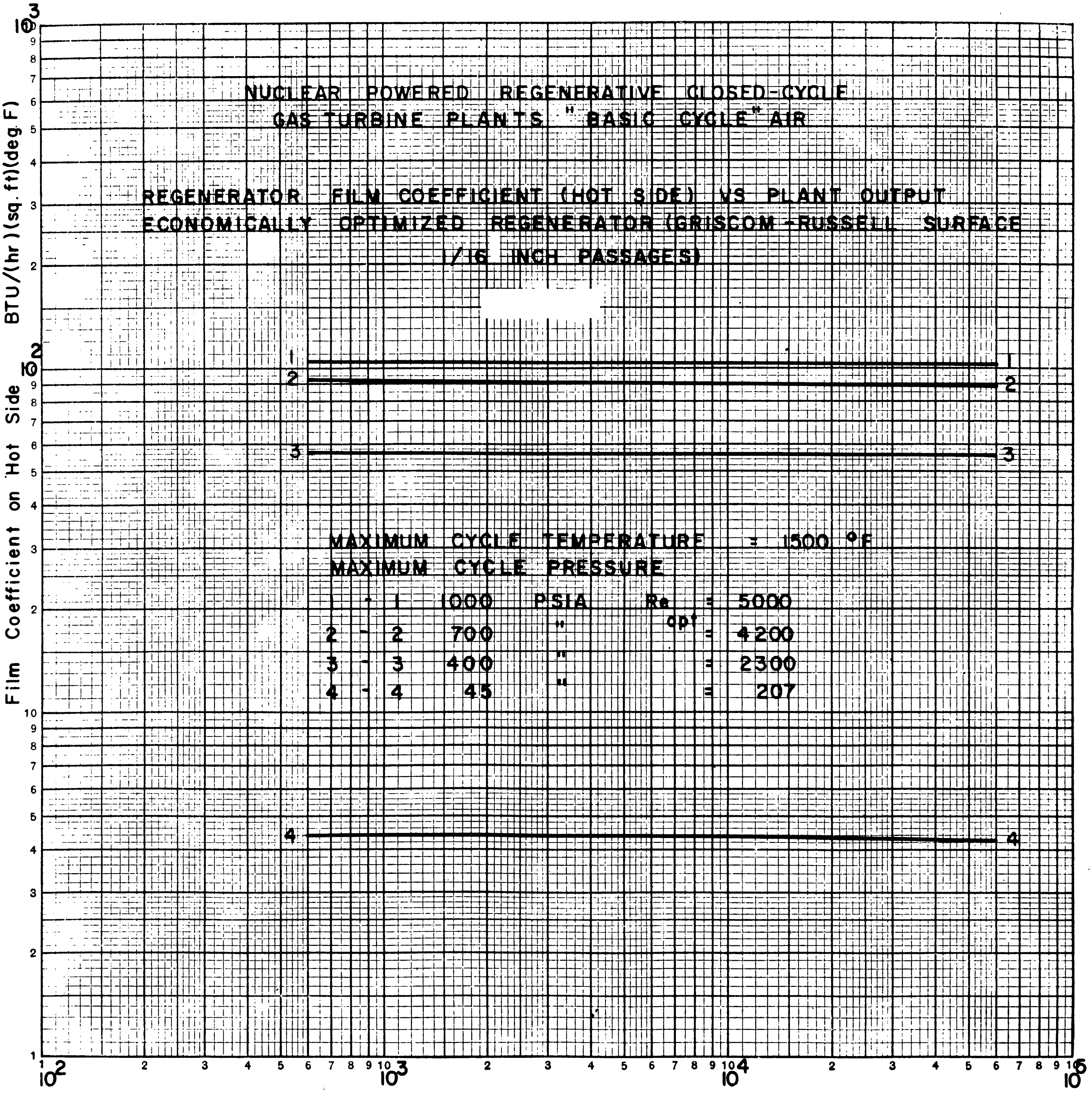

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" AIR
REGENERATOR FILM COEFFICIENT (HOT SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM-RUSSELL SURFACE
1/16 INCH PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 5000
2 - 2 700 " = 4200
3 - 3 400 " = 2300
4 - 4 45 " = 207
Film Coefficient on Hot Side BTU/(hr)(sq. ft)(deg. F)
PLANT OUTPUT, HP

Figure 48.

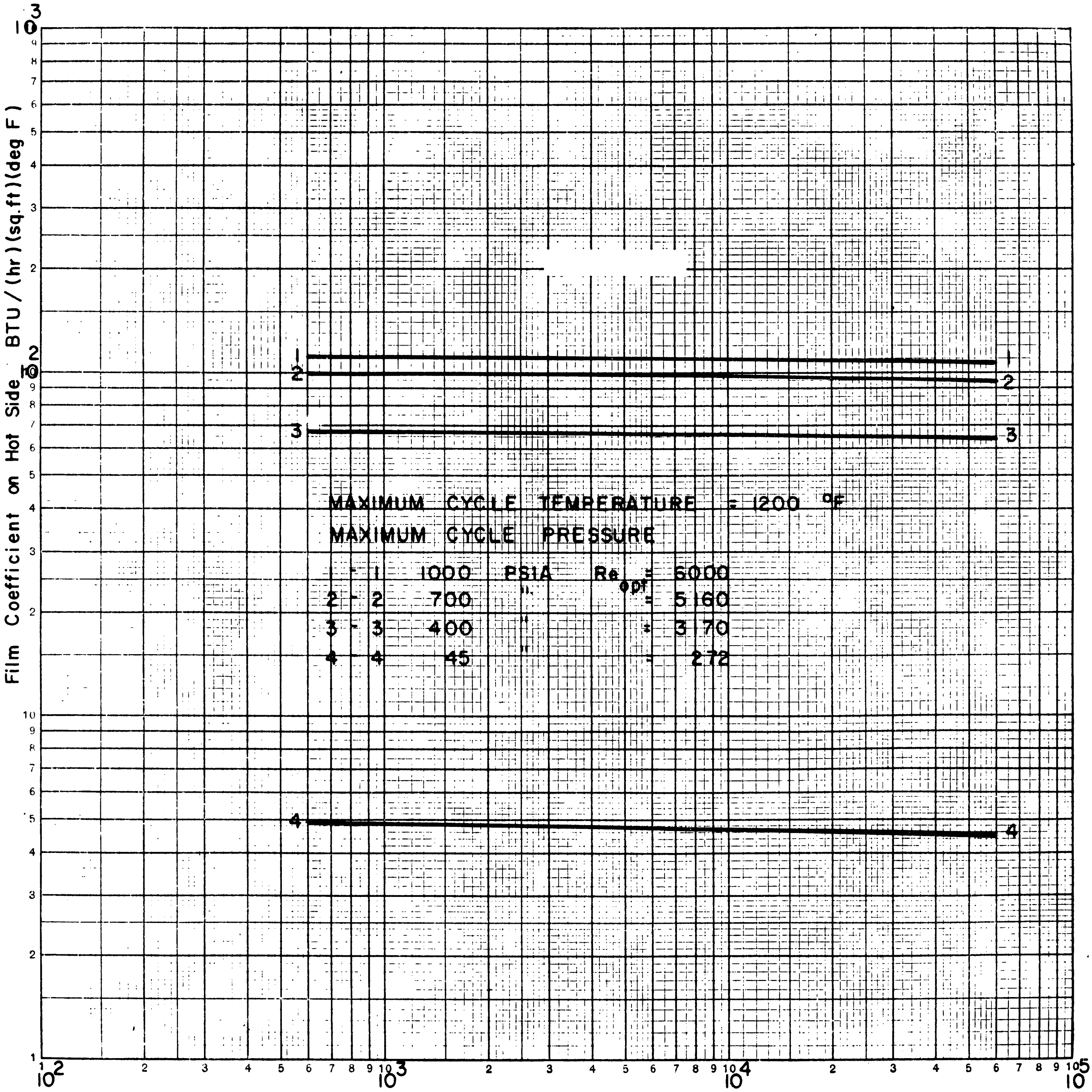

Film Coefficient on Hot Side BTU / (hr) (sq.ft) (deg F)
MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re opt = 6000
2 - 2 700 " = 5160
3 - 3 400 " = 3170
4 - 4 45 " = 272
1
2
3
4
10^2
10^3
10^4
10^5
PLANT OUTPUT, HP

Figure 49.

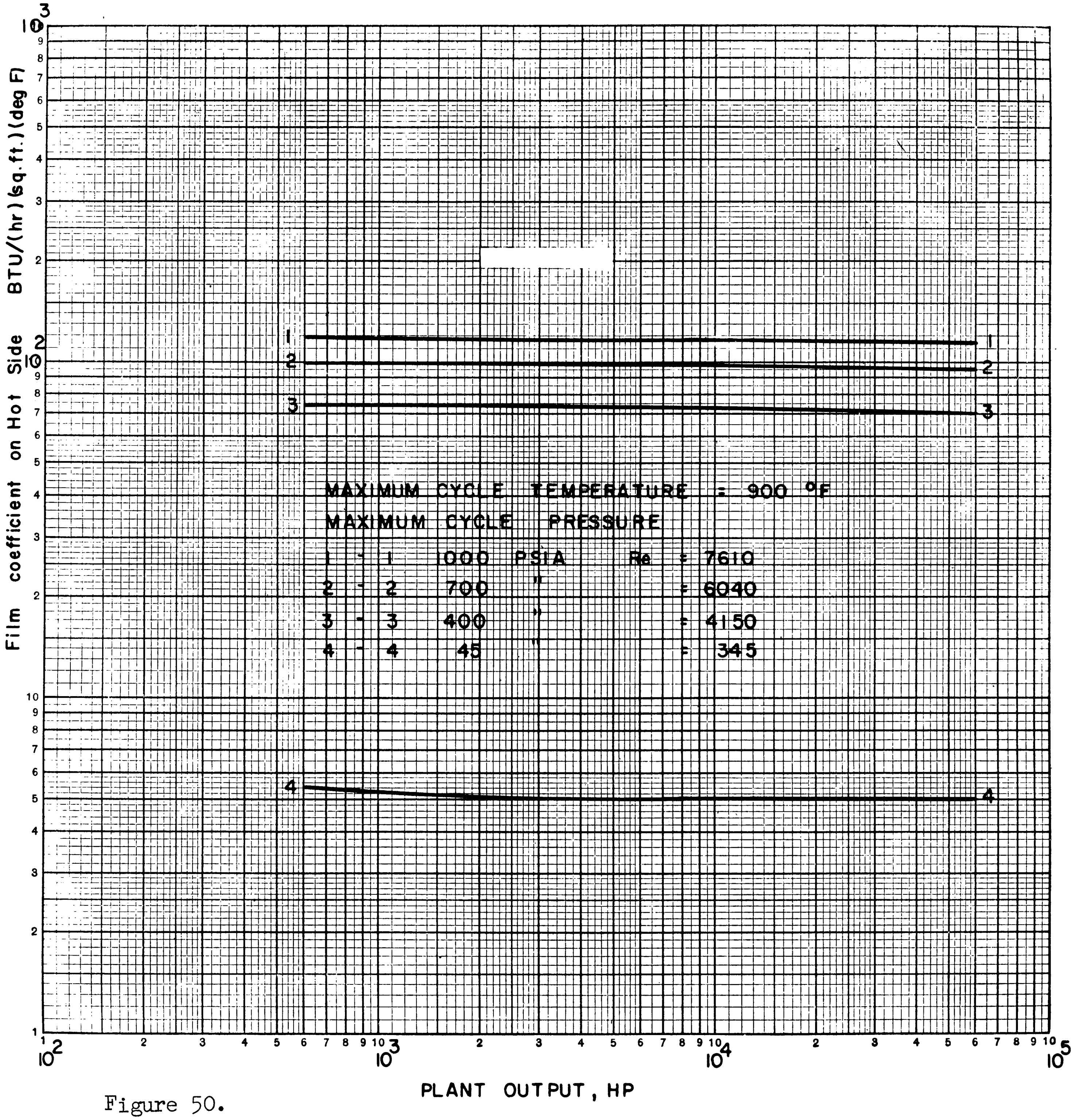

MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re = 7610
2 - 2 700 " = 6040
3 - 3 400 " = 4150
4 - 4 45 " = 345
Film coefficient on Hot Side BTU/(hr)(sq.ft.)(deg F)
PLANT OUTPUT, HP

Figure 50.

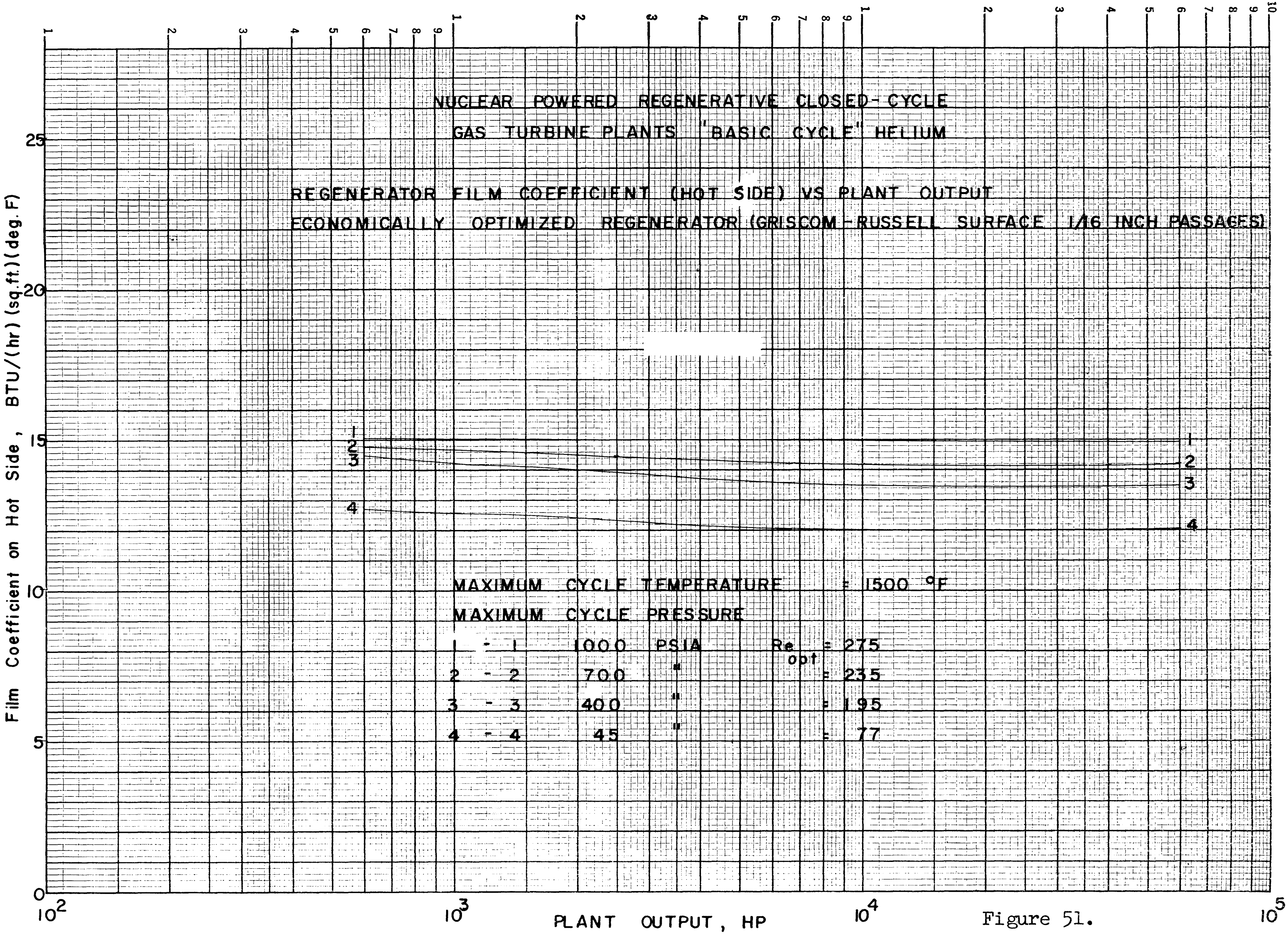

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" HELIUM
REGENERATOR FILM COEFFICIENT (HOT SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM-RUSSELL SURFACE 1/16 INCH PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re opt = 275
2 - 2 700 " = 235
3 - 3 400 " = 195
4 - 4 45 " = 77
Film Coefficient on Hot Side, BTU/(hr)(sq.ft.)(deg.F)
PLANT OUTPUT, HP

Figure 51.

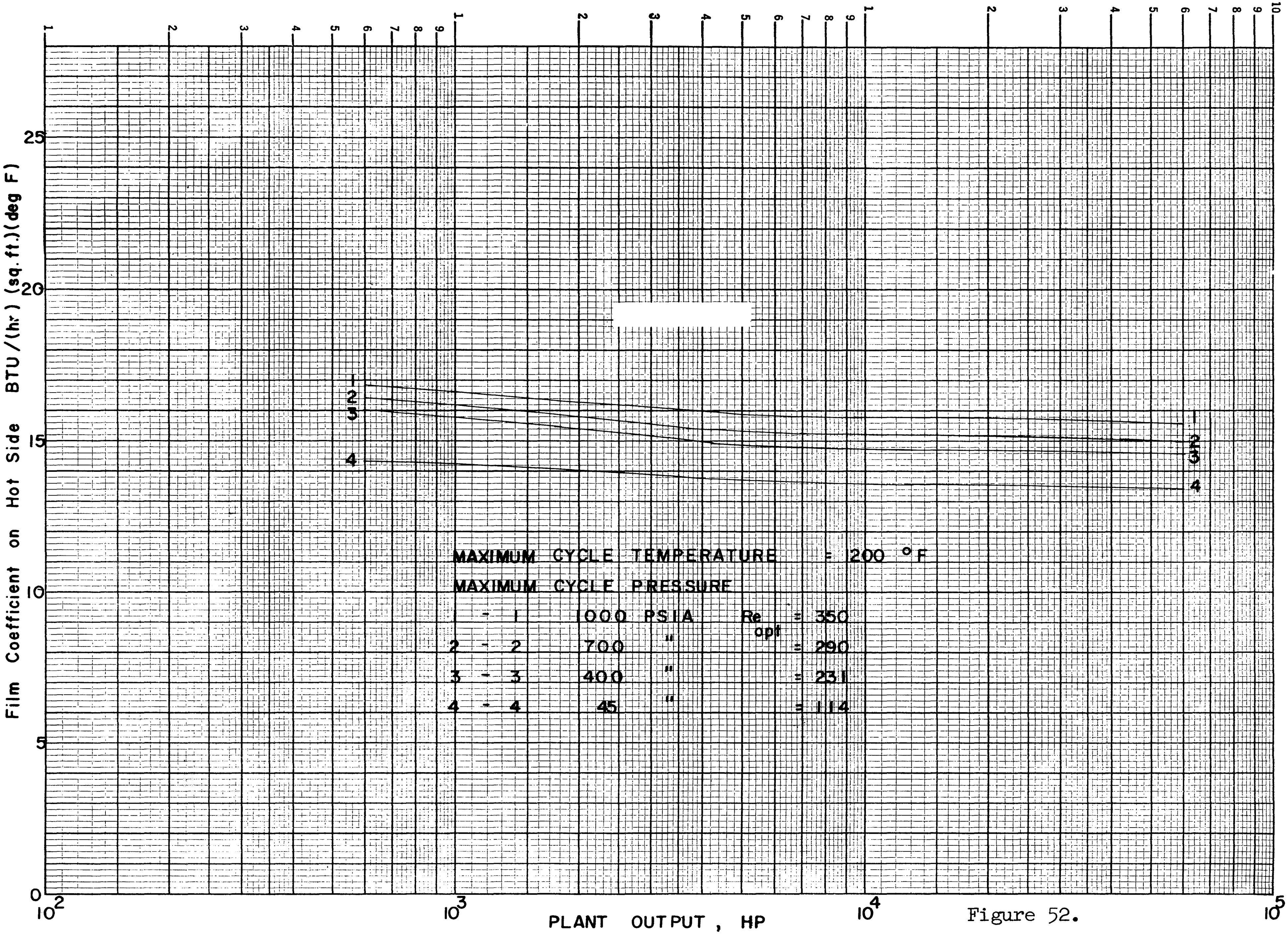
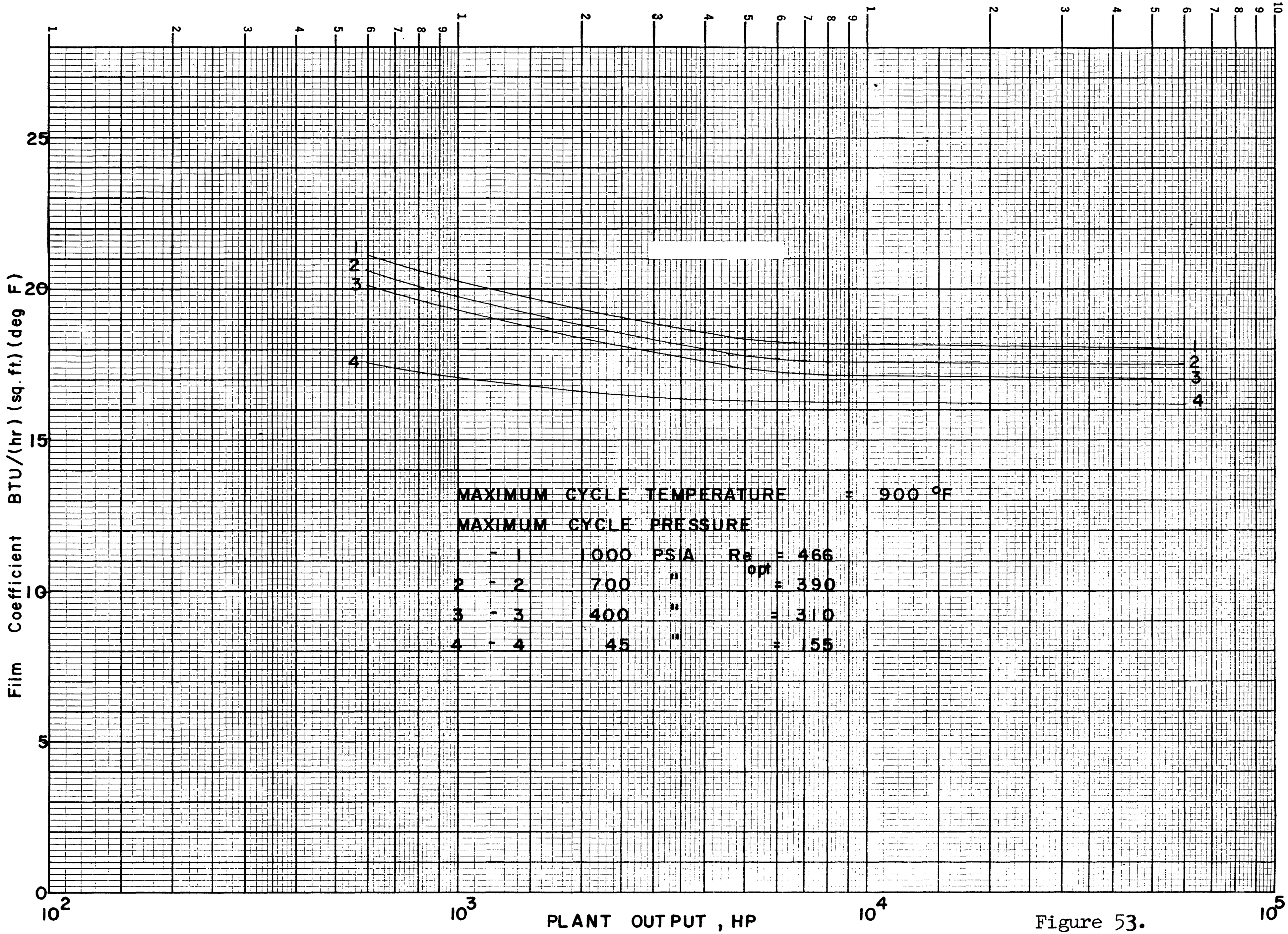

MAXIMUM CYCLE TEMPERATURE = 200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 350
2 - 2 700 " = 290
3 - 3 400 " = 231
4 - 4 45 " = 114
Film Coefficient on Hot Side BTU/(hr) (sq.ft.)(deg F)
PLANT OUTPUT, HP

Figure 52.



MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 466
2 - 2 700 " = 390
3 - 3 400 " = 310
4 - 4 45 " = 155
Film Coefficient BTU/(hr)(sq.ft.)(deg F)
PLANT OUTPUT, HP

Figure 53.

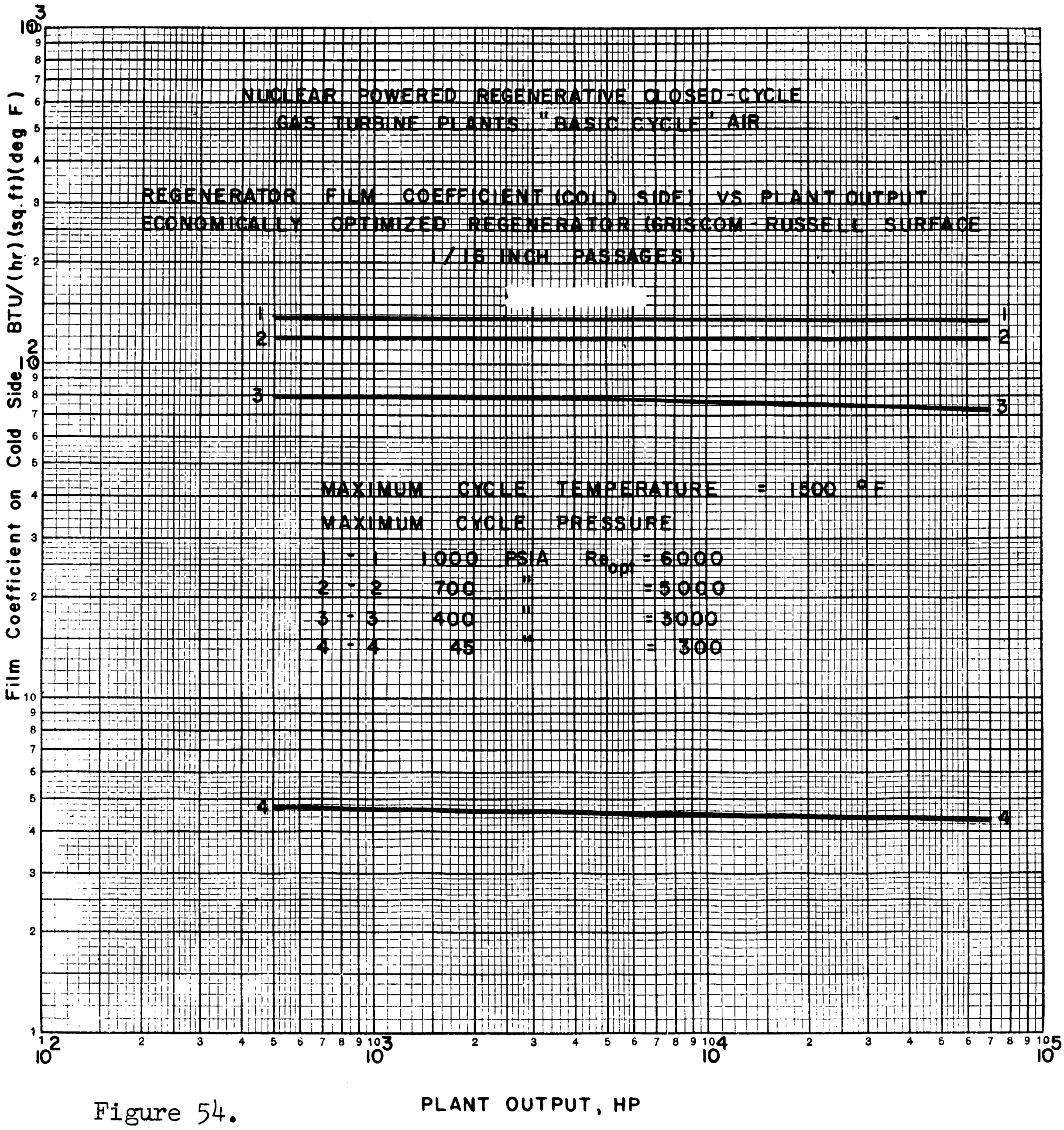

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" AIR
REGENERATOR FILM COEFFICIENT (COLD SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM-RUSSELL SURFACE
1/16 INCH PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re opt = 6000
2 - 2 700 " = 5000
3 - 3 400 " = 3000
4 - 4 45 " = 300
Film Coefficient on Cold Side_ BTU/(hr)(sq.ft)(deg F)
PLANT OUTPUT, HP

Figure 54.

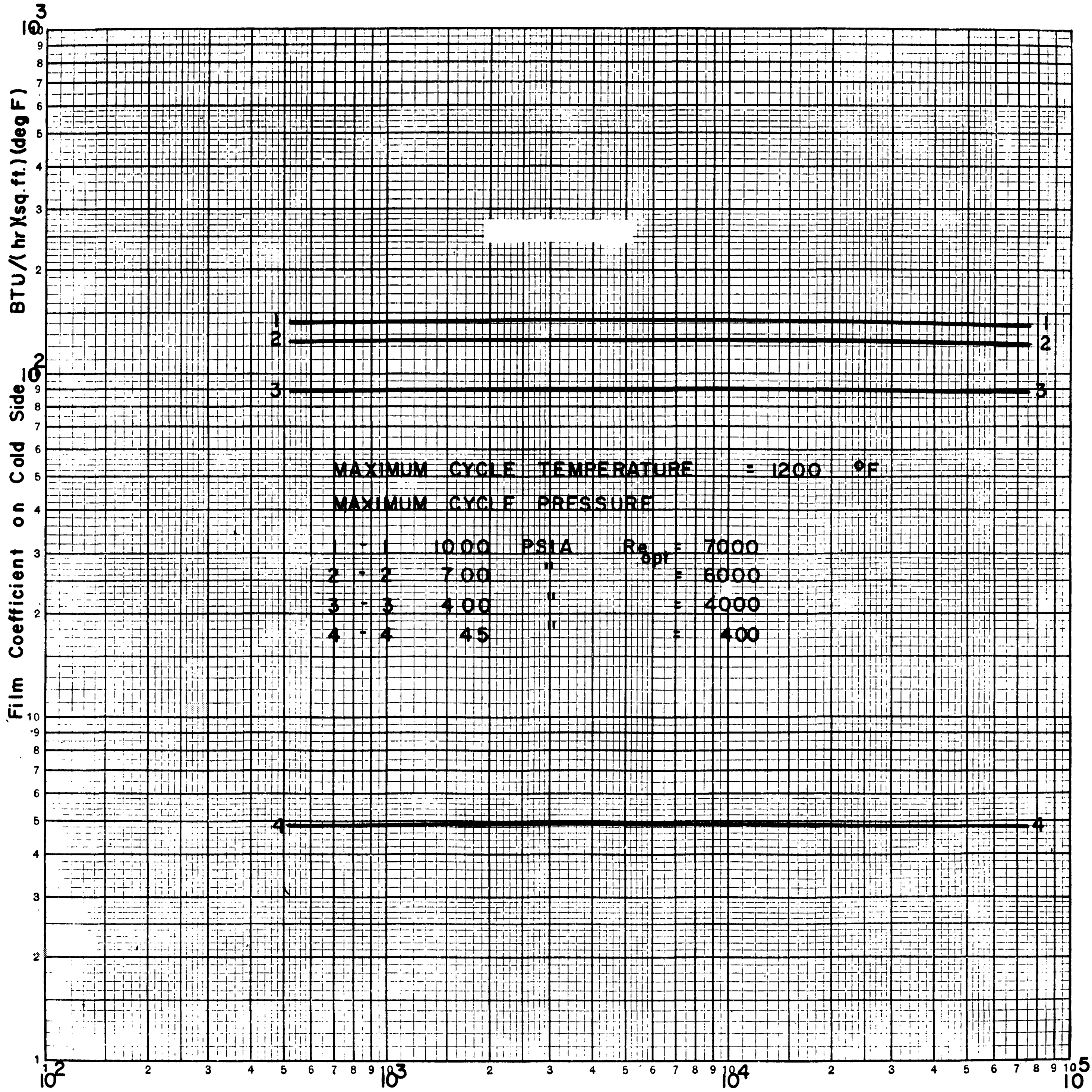

Film Coefficient on Cold Side BTU/(hr)(sq.ft.)(deg F)
PLANT OUTPUT, HP
MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 7000
2 - 2 700 " = 6000
3 - 3 400 " = 4000
4 - 4 45 " = 400

Figure 55.

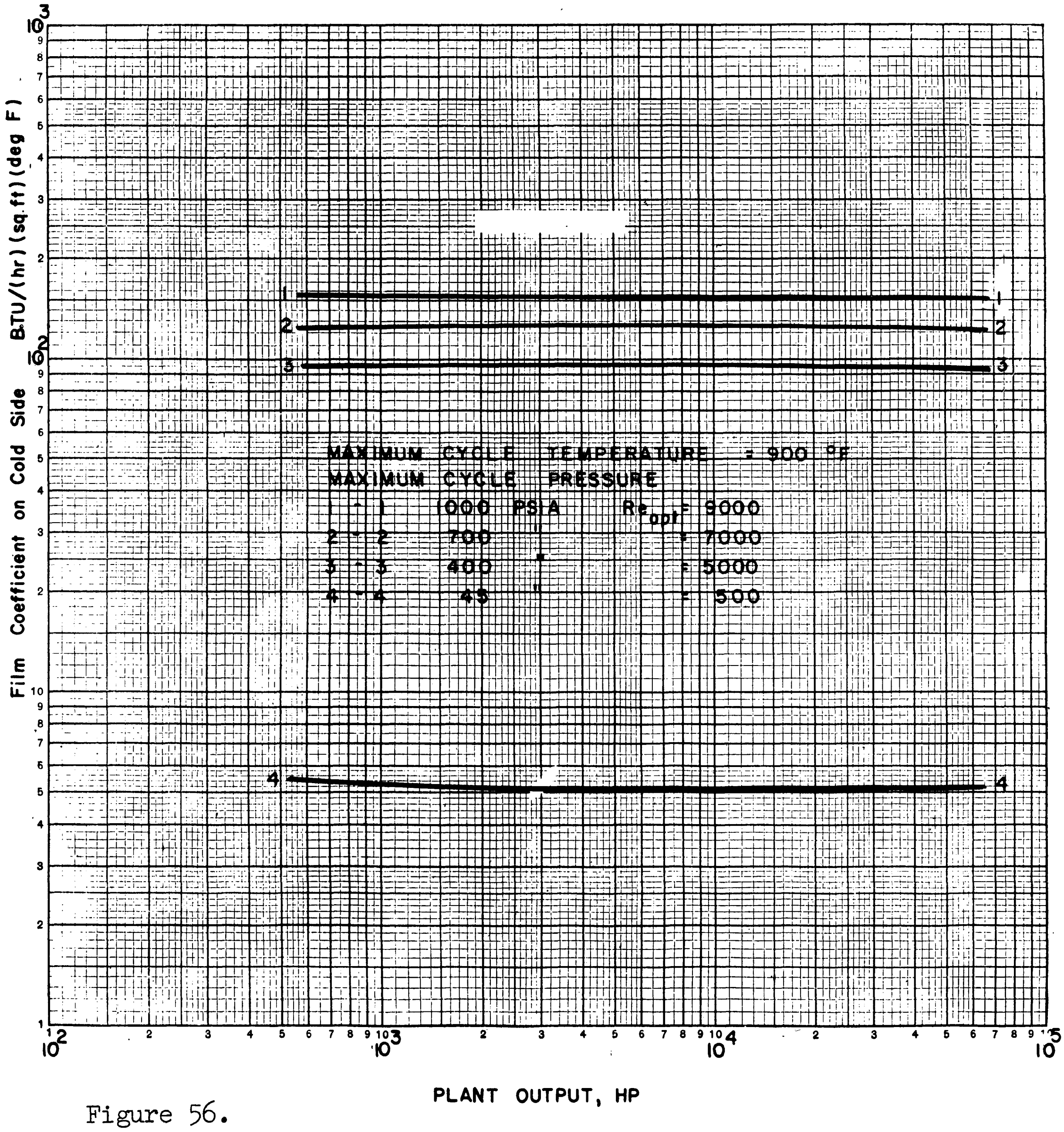
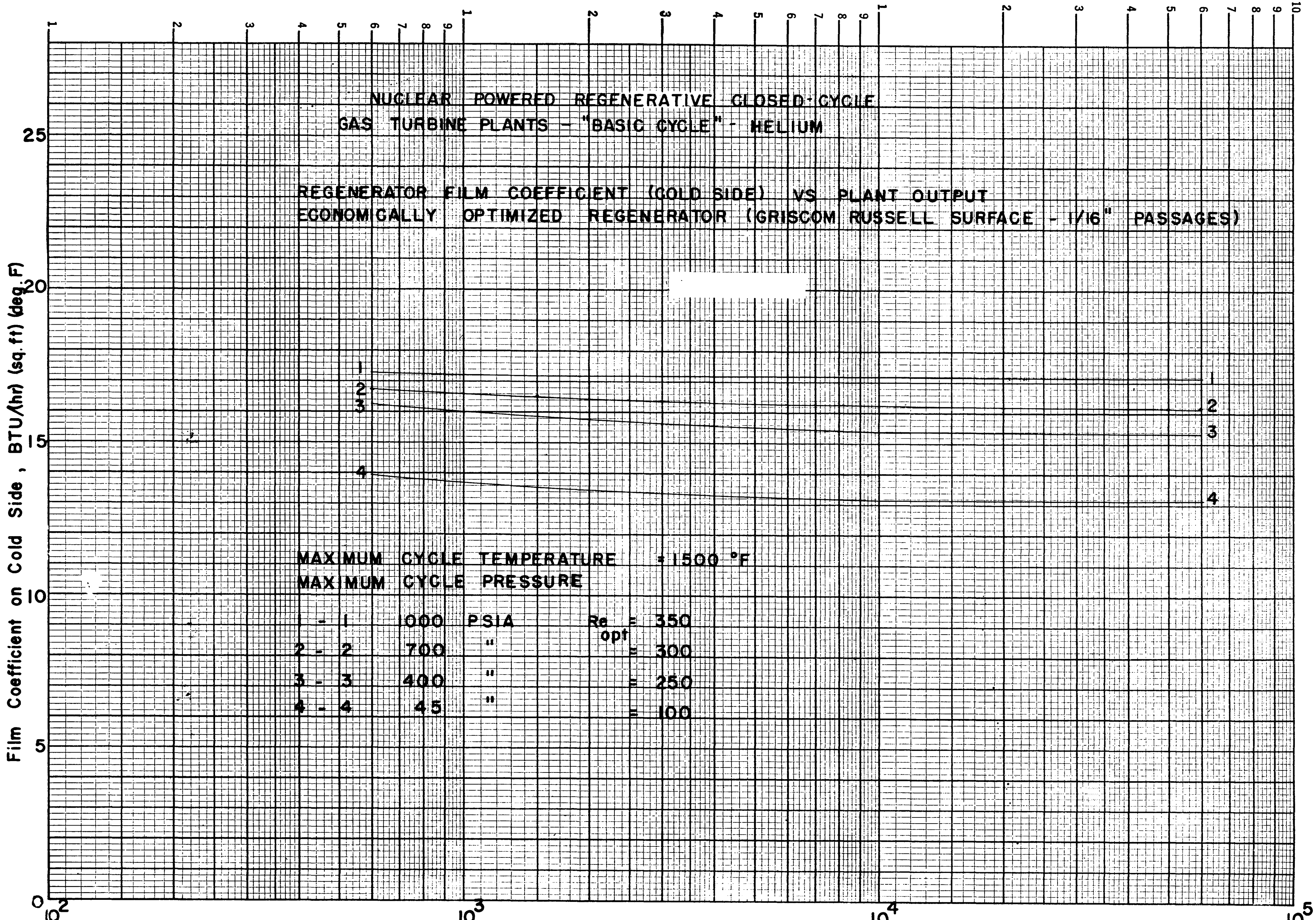

Film Coefficient on Cold Side BTU/(hr)(sq ft)(deg F)
MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 9000
2 - 2 700 " = 7000
3 - 3 400 " = 5000
4 - 4 45 " = 500
PLANT OUTPUT, HP

Figure 56.



NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS - "BASIC CYCLE" - HELIUM
REGENERATOR FILM COEFFICIENT (COLD SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM RUSSELL SURFACE - 1/16" PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 350
2 - 2 700 " = 300
3 - 3 400 " = 250
4 - 4 45 " = 100
Film Coefficient on Cold Side , BTU/(hr) (sq.ft) (deg. F)
PLANT OUTPUT, HP

Figure 57.

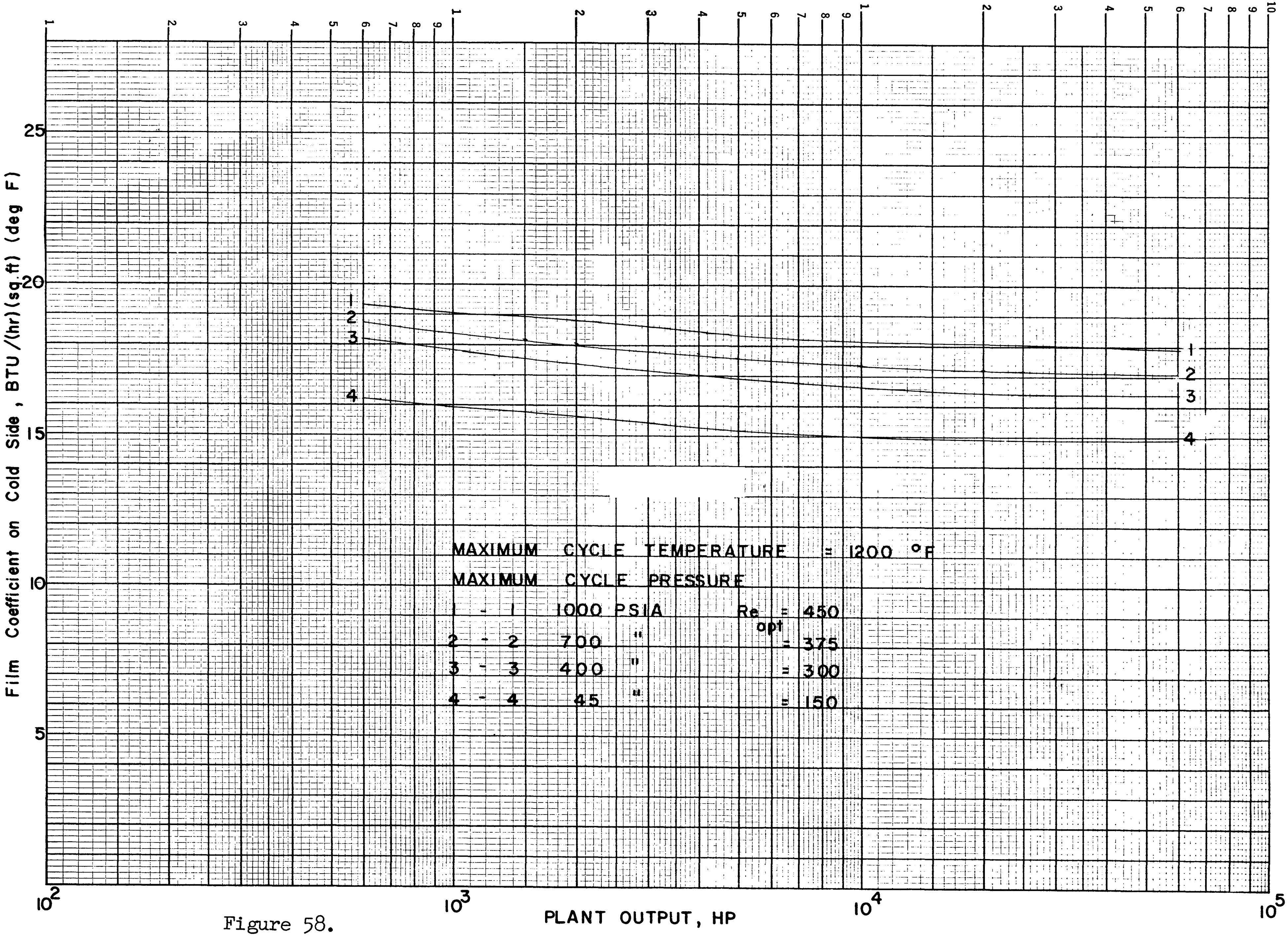

MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re opt = 450
2 - 2 700 " = 375
3 - 3 400 " = 300
4 - 4 45 " = 150
Film Coefficient on Cold Side, BTU/(hr)(sq.ft) (deg F)
PLANT OUTPUT, HP

Figure 58.

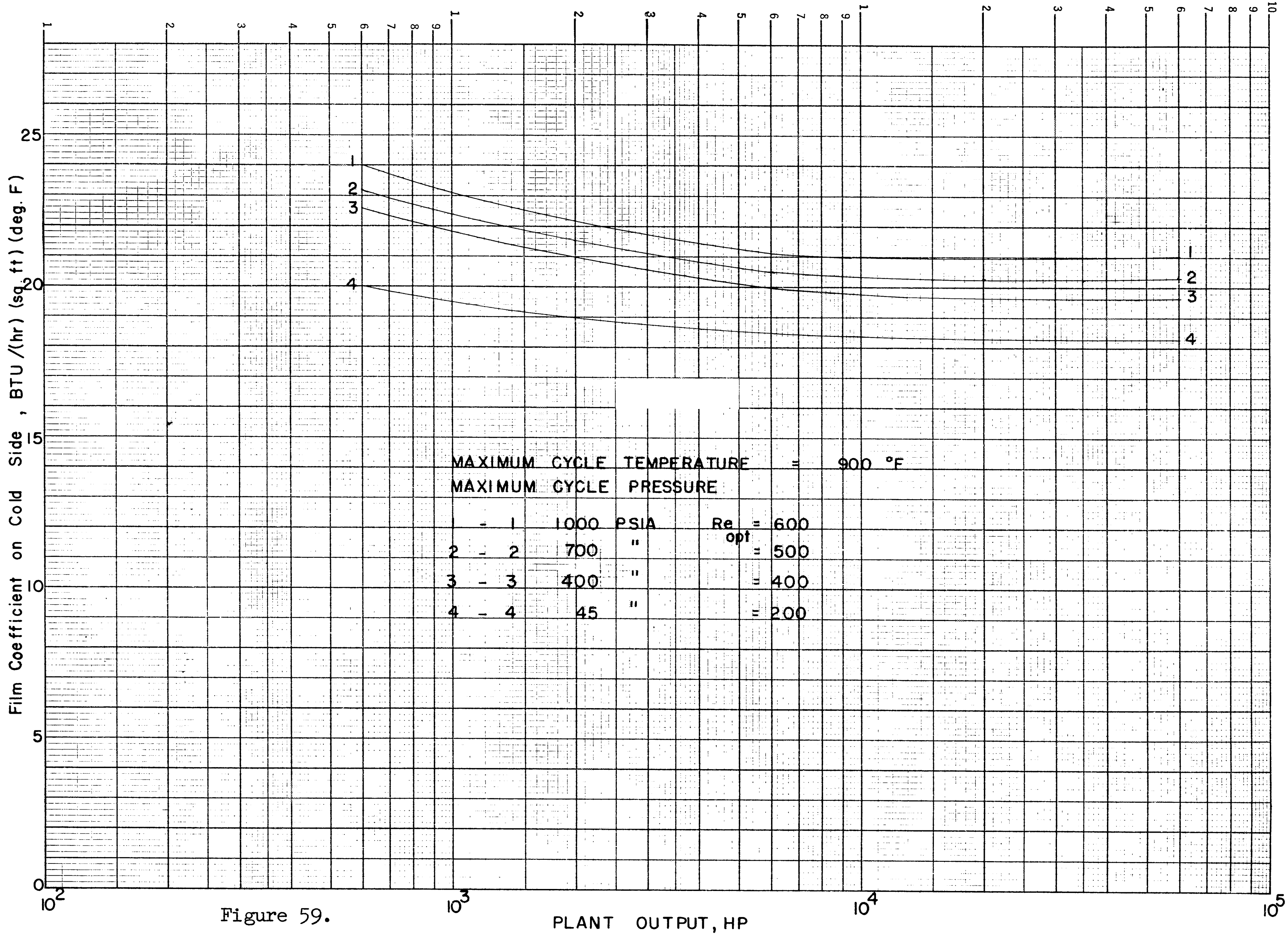

MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA Re_{opt} = 600
2 - 2 700 " = 500
3 - 3 400 " = 400
4 - 4 45 " = 200
Film Coefficient on Cold Side, BTU/(hr) (sq. ft) (deg. F)
PLANT OUTPUT, HP
10^2
10^3
10^4
10^5

Figure 59.

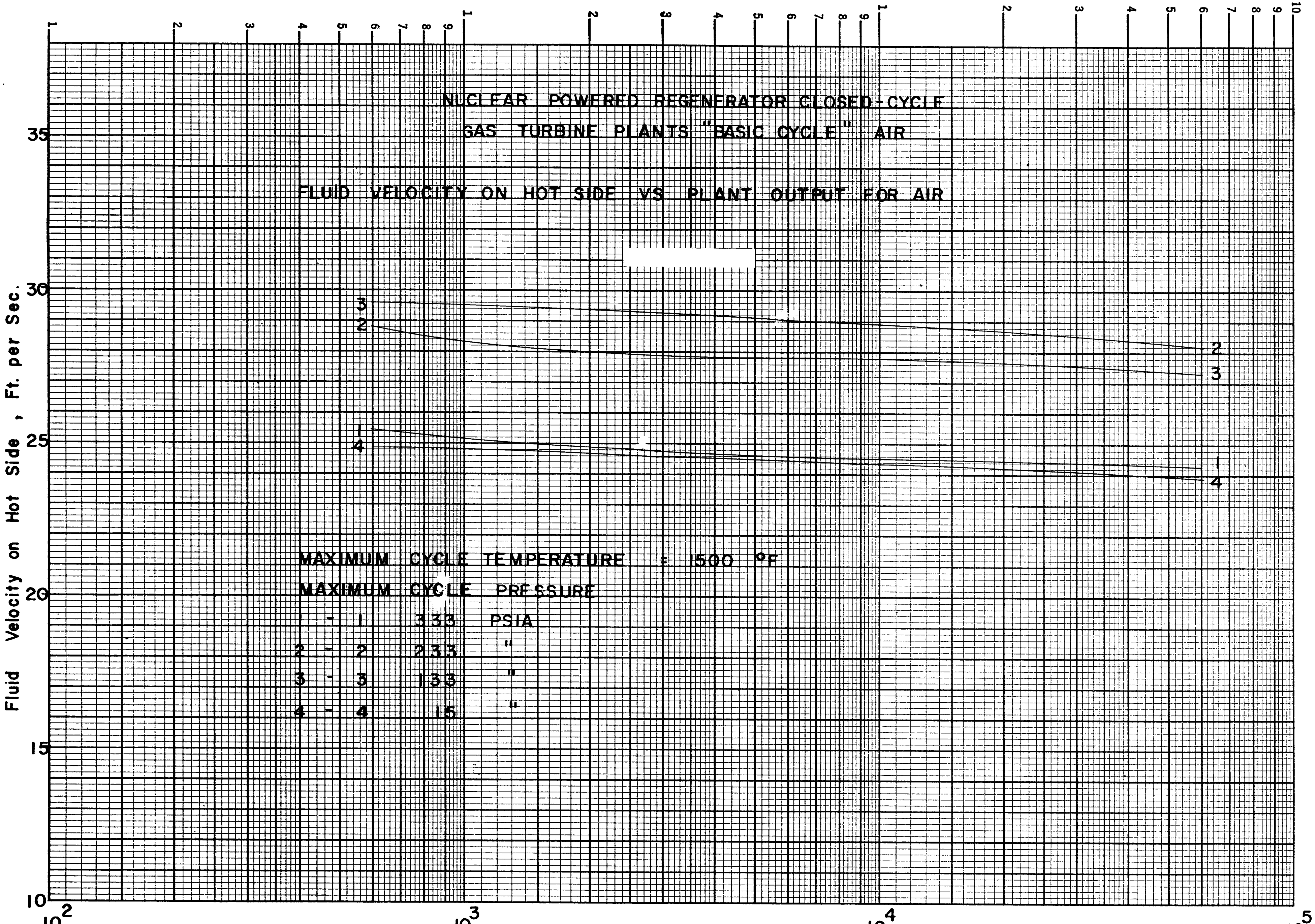

NUCLEAR POWERED REGENERATOR CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" AIR
FLUID VELOCITY ON HOT SIDE VS PLANT OUTPUT FOR AIR
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 333 PSIA
2 - 2 233 "
3 - 3 133 "
4 - 4 15 "
Fluid Velocity on Hot Side , Ft. per Sec.
PLANT OUTPUT , HP

Figure 60.

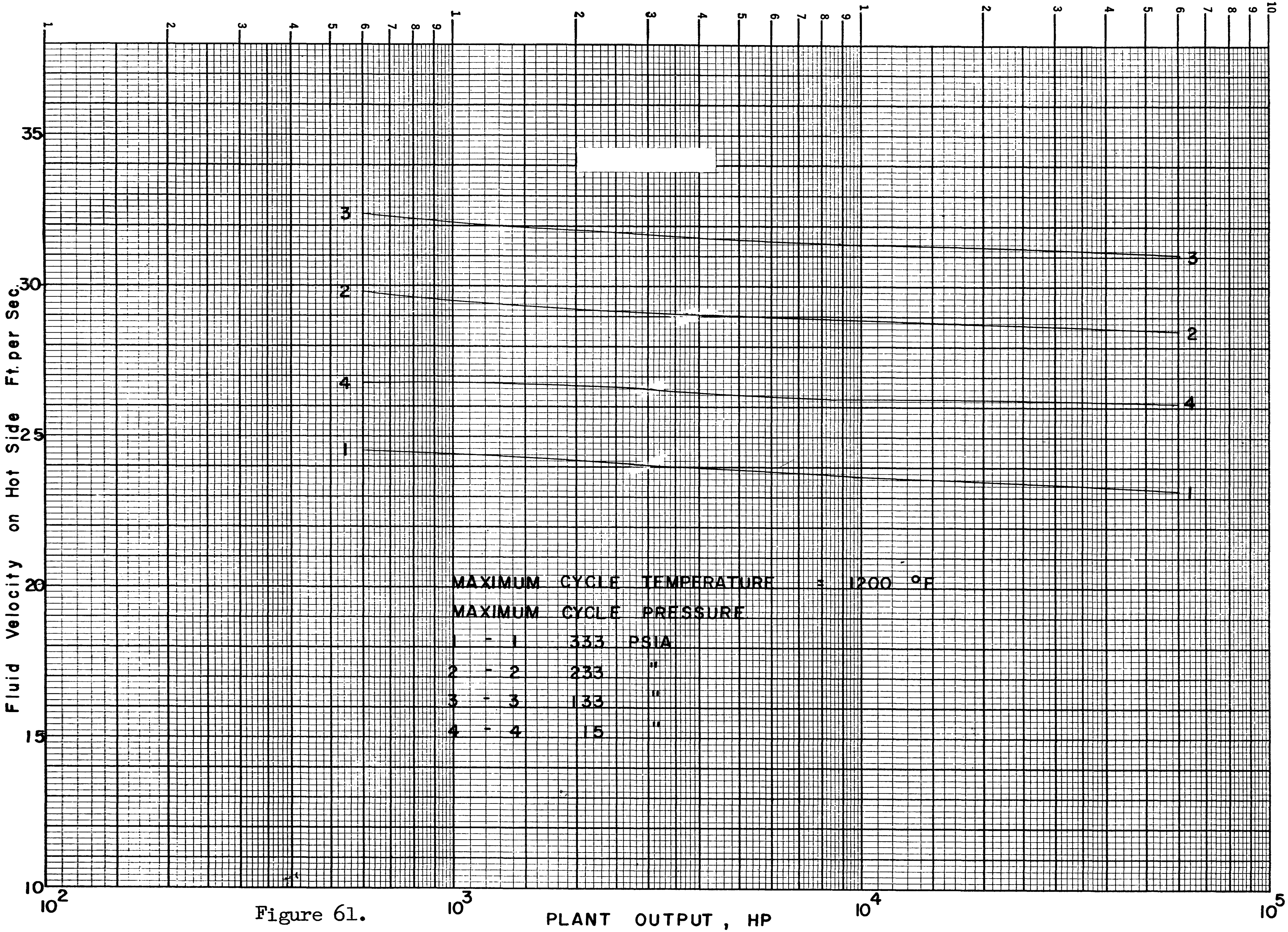
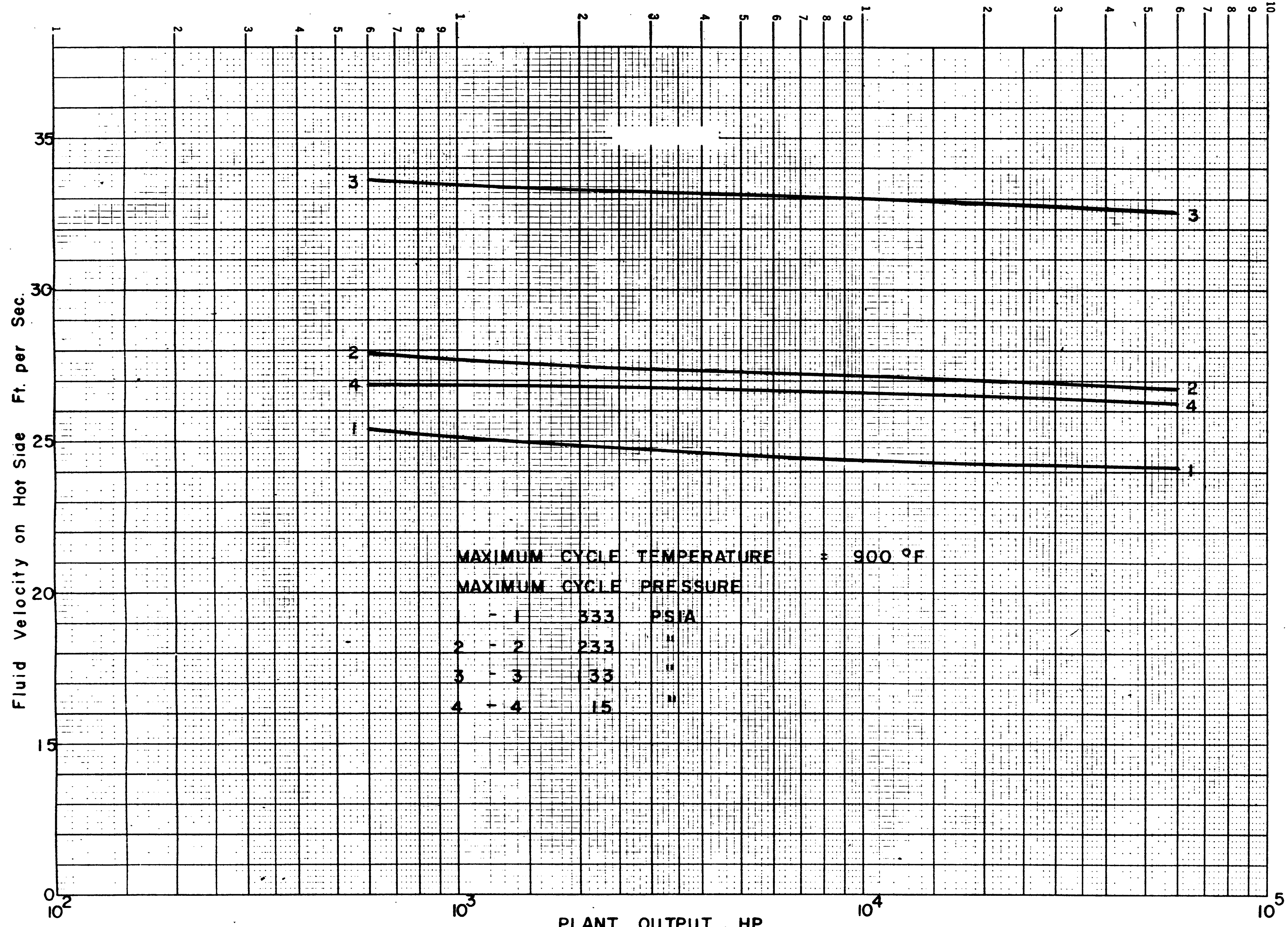

MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 333 PSIA
2 - 2 233 "
3 - 3 133 "
4 - 4 15 "
Fluid Velocity on Hot Side Ft. per Sec.
PLANT OUTPUT, HP

Figure 61.


MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 333 PSIA
2 - 2 233 "
3 - 3 133 "
4 - 4 15 "
Fluid Velocity on Hot Side Ft. per Sec.
PLANT OUTPUT, HP
Figure 62.

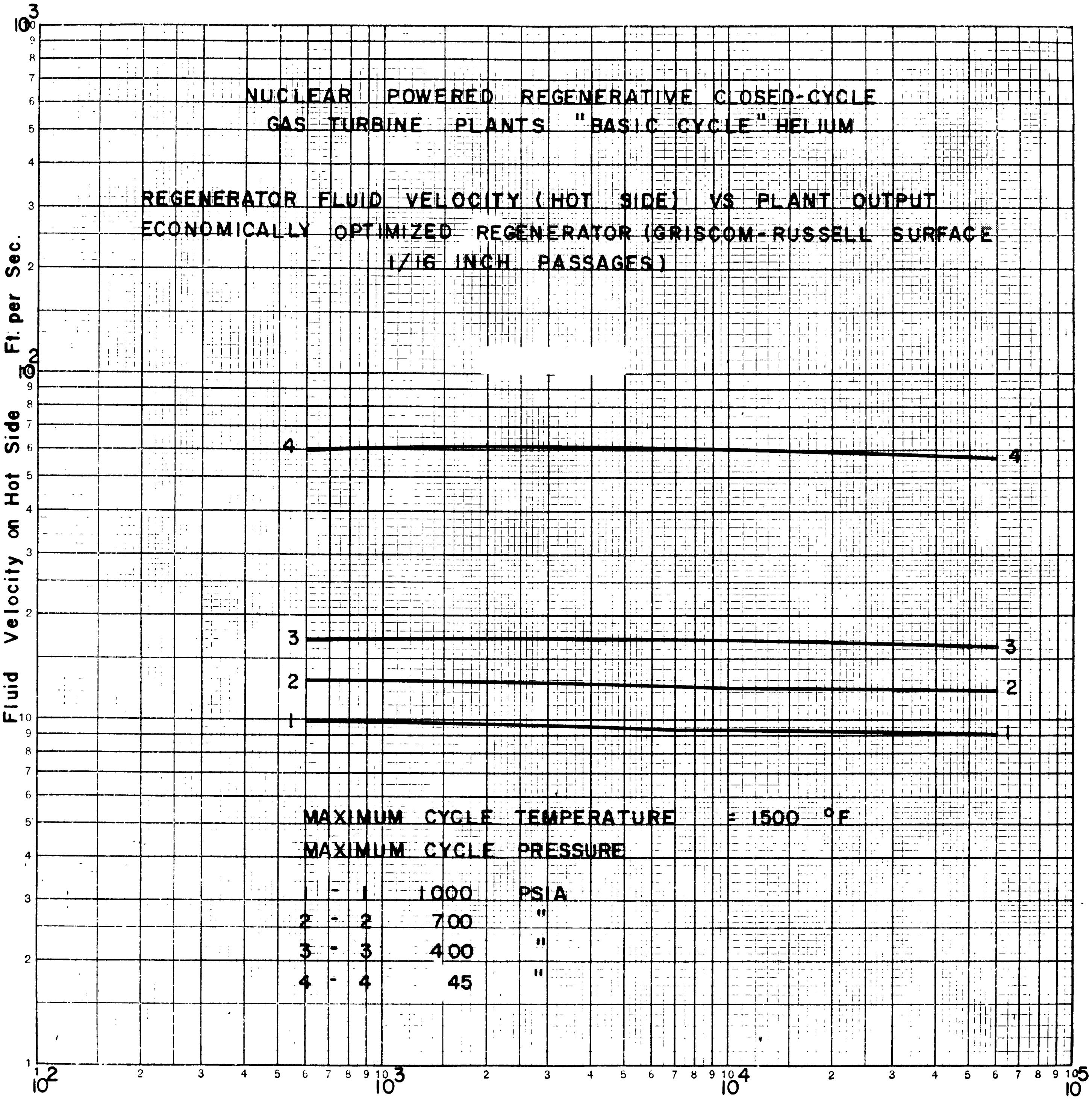
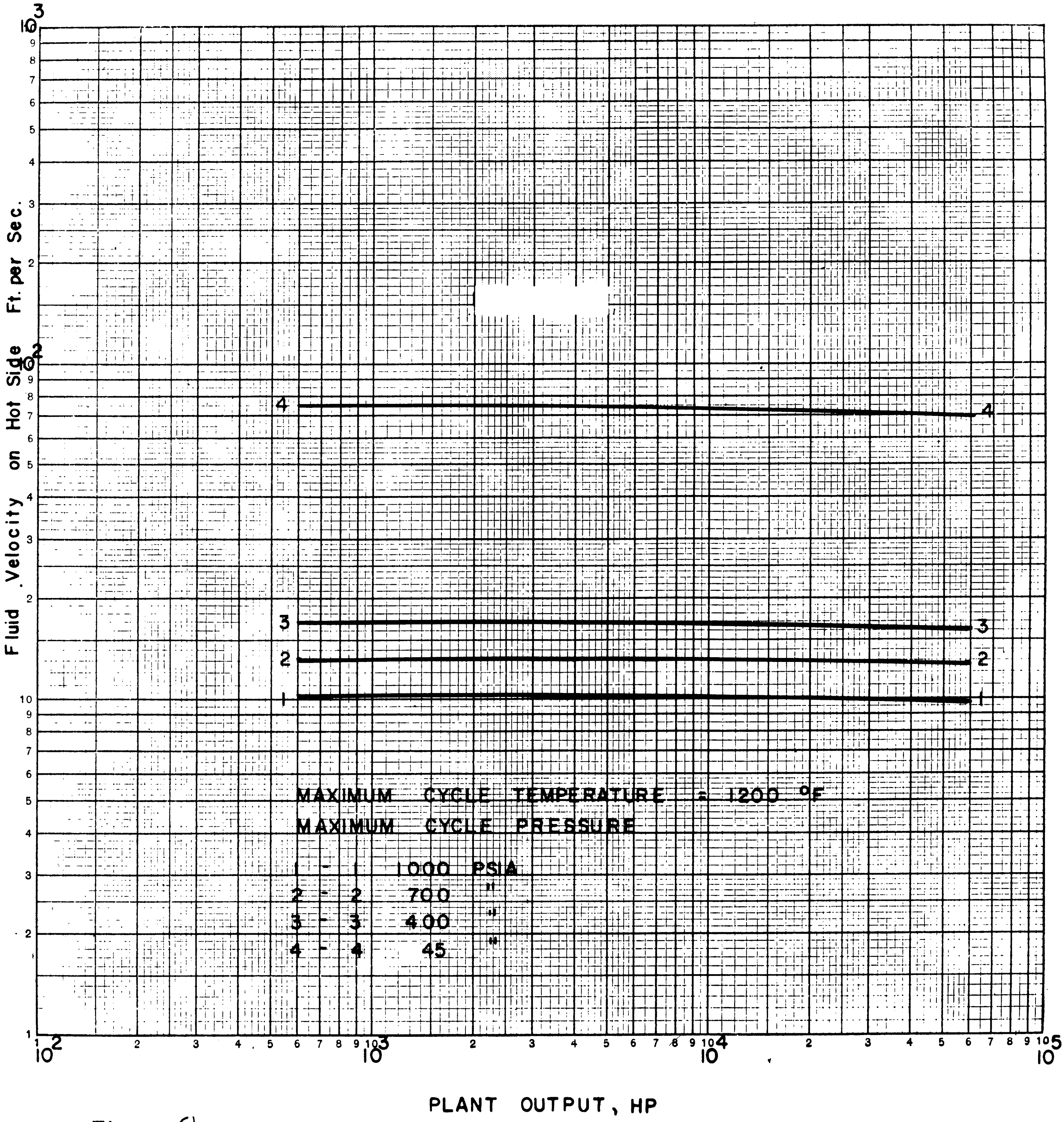

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" HELIUM
REGENERATOR FLUID VELOCITY (HOT SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM-RUSSELL SURFACE
1/16 INCH PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
Fluid Velocity on Hot Side Ft. per Sec.
PLANT OUTPUT, HP

Figure 63.



Fluid Velocity on Hot Side Ft. per Sec.
PLANT OUTPUT, HP
MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "

Figure 64.

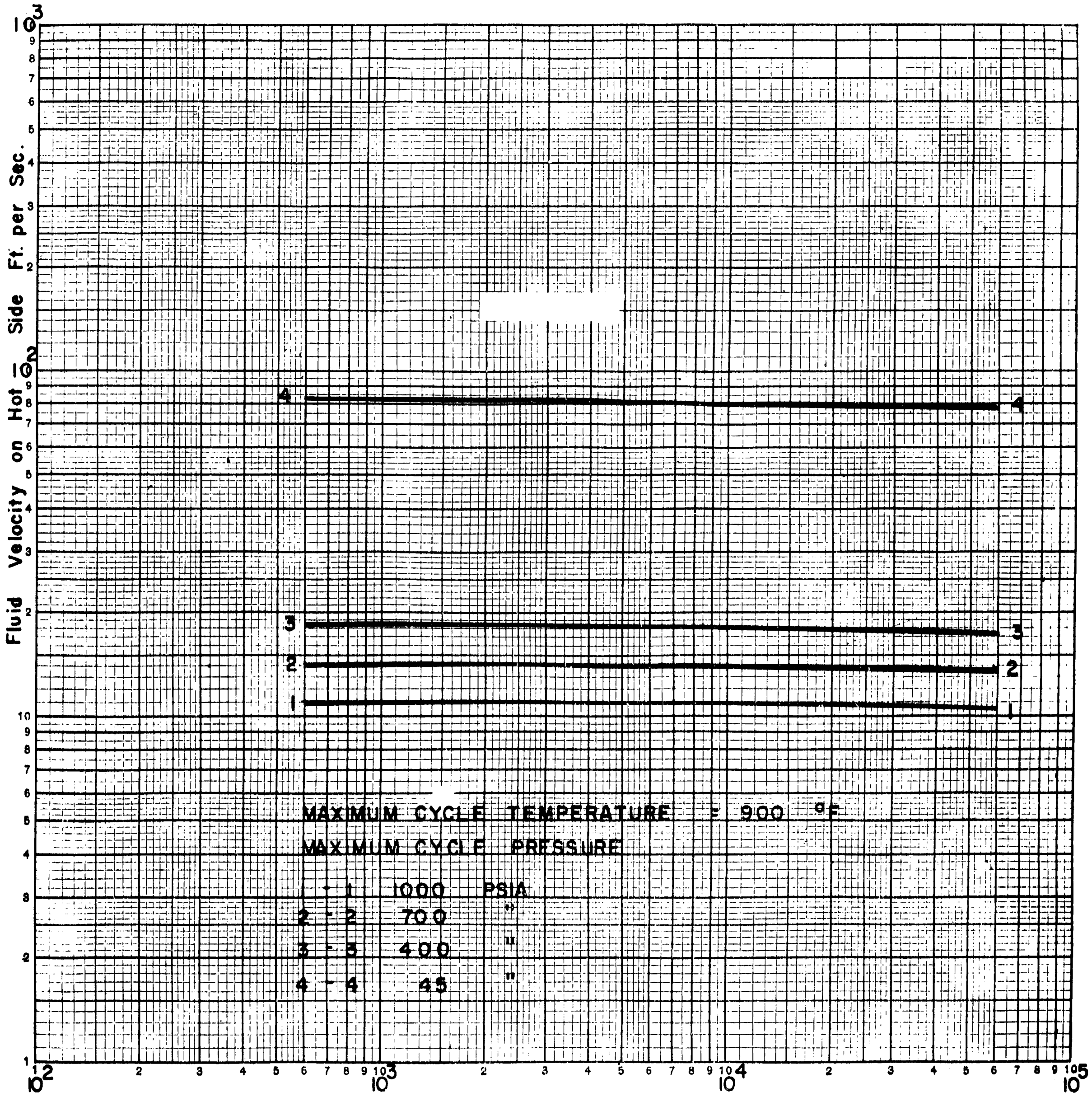
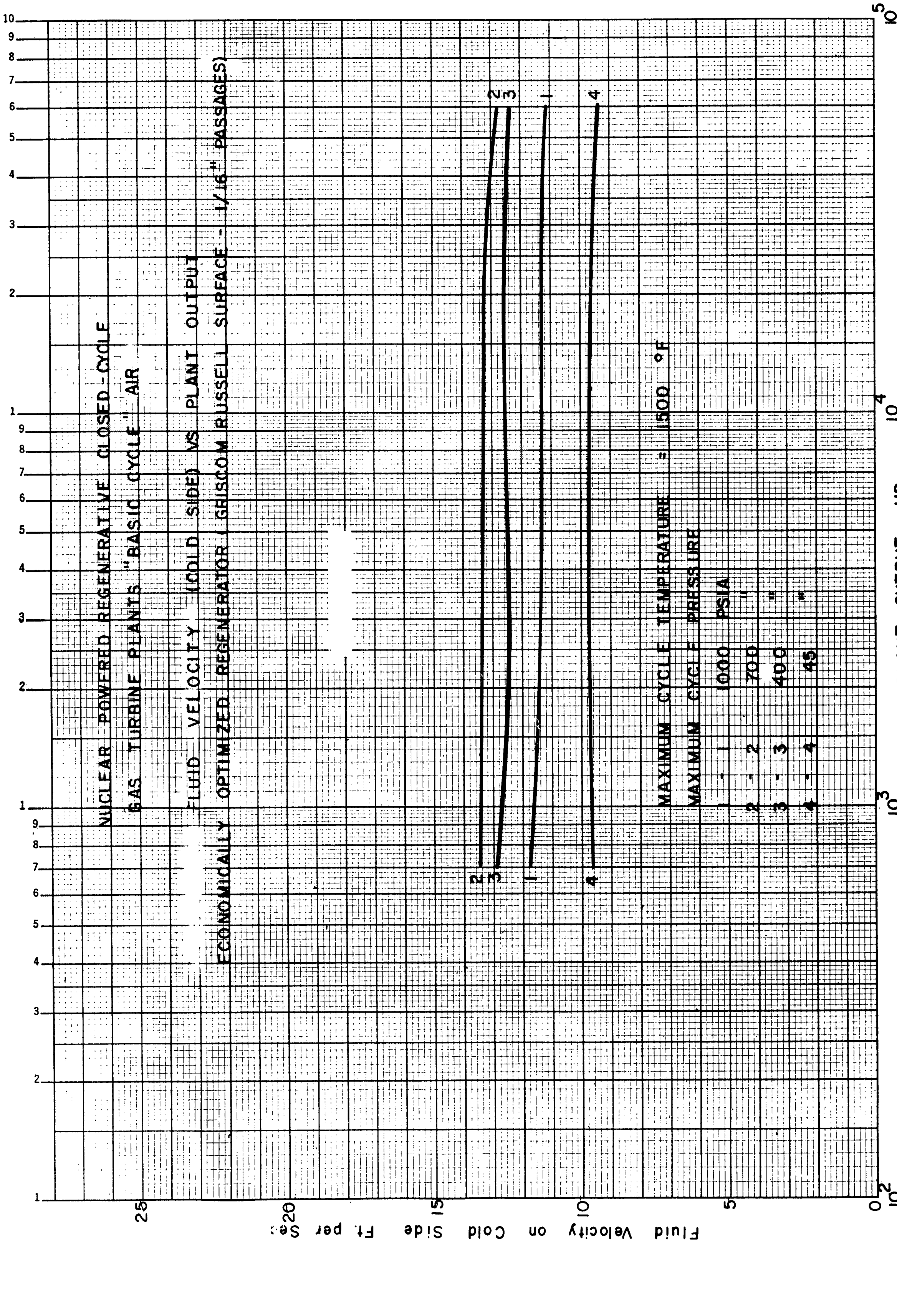


Figure 65. PLANT OUTPUT, HP



NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" AIR
FLUID VELOCITY (COLD SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM RUSSELL SURFACE - 1/16" PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
PLANT OUTPUT, HP
Fluid Velocity on Cold Side Ft. per Sec.

Figure 66.

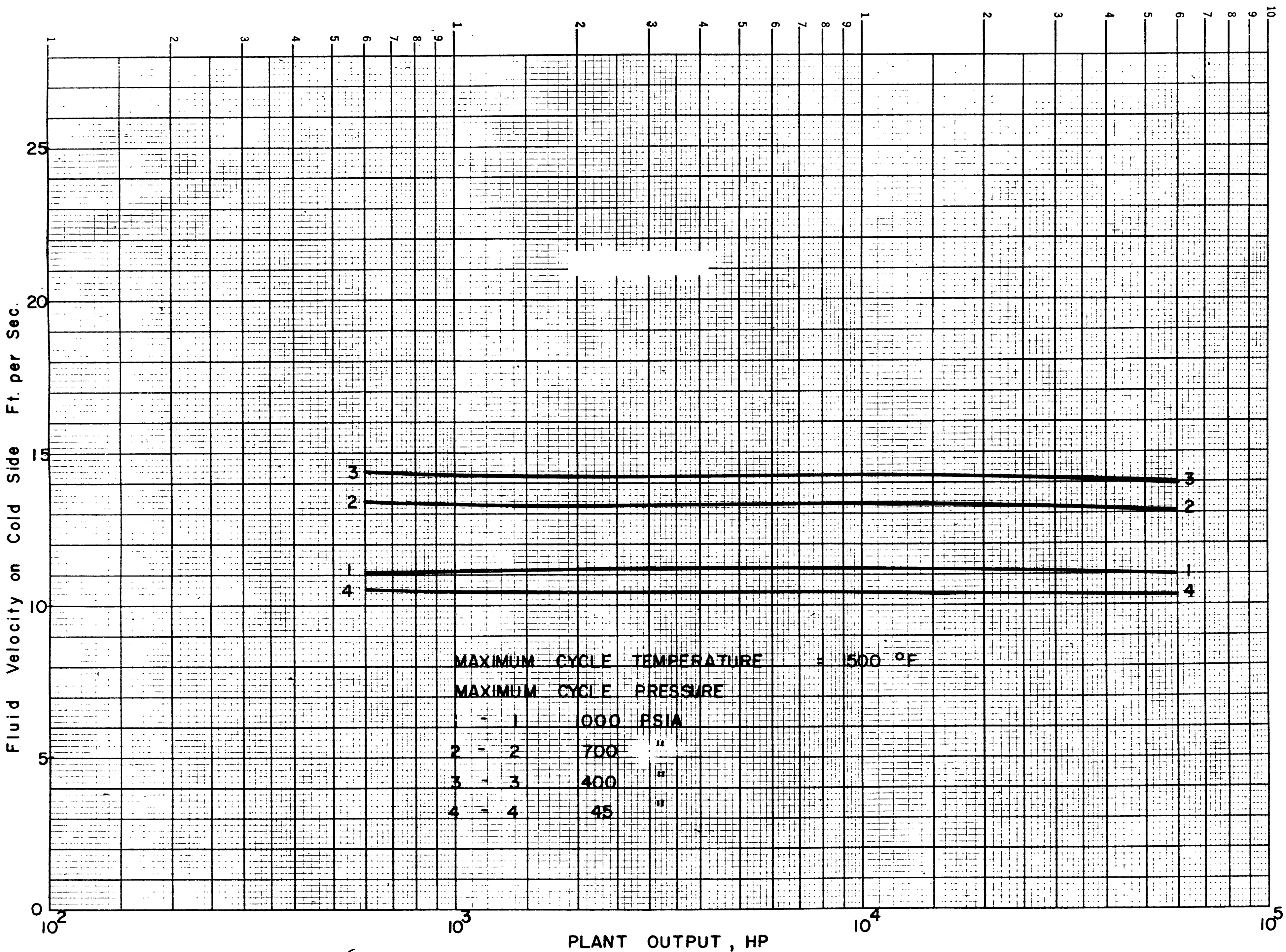

Fluid Velocity on Cold Side Ft. per Sec.
PLANT OUTPUT, HP
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "

Figure 67.

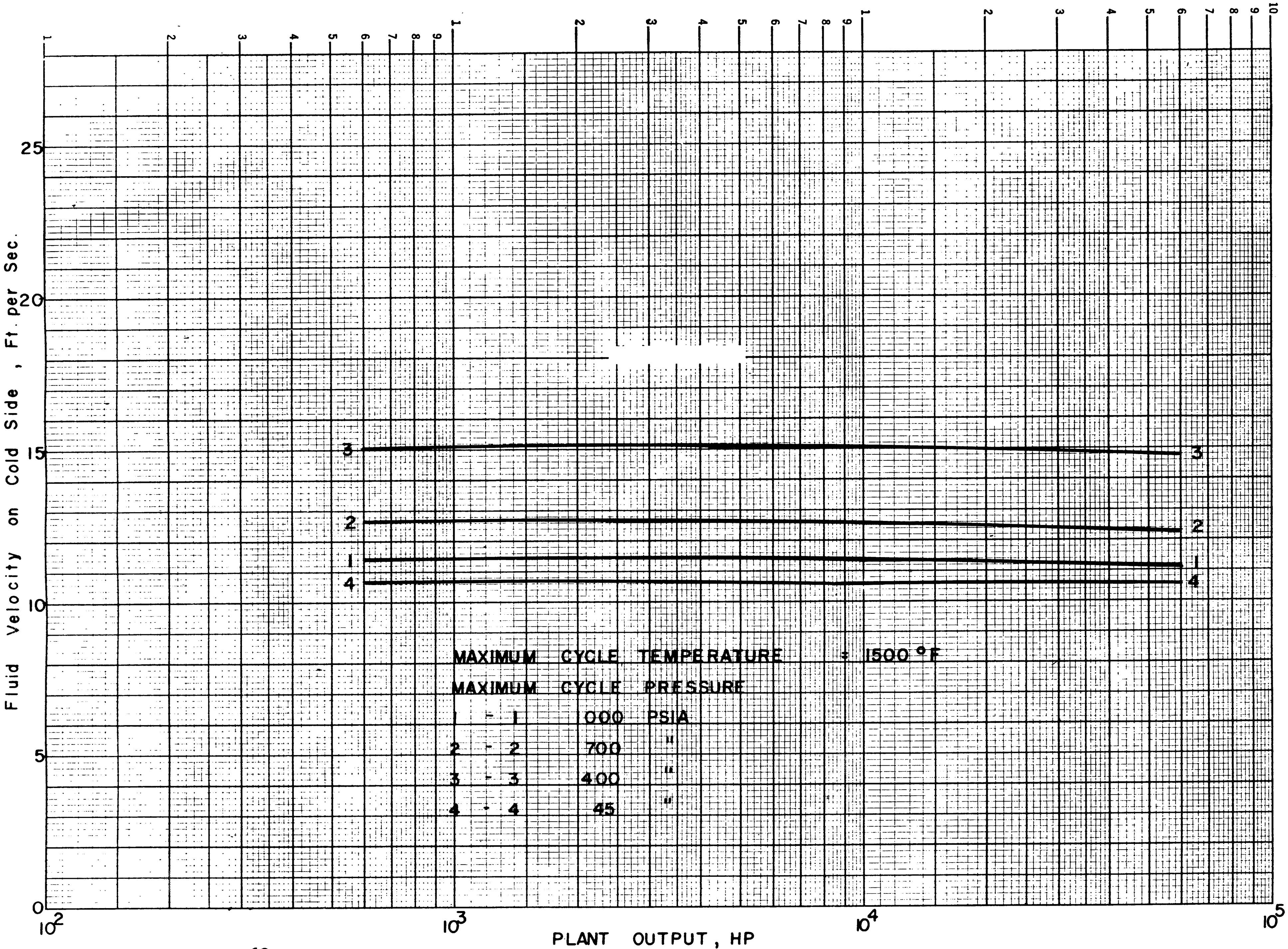

MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
Fluid Velocity on Cold Side , Ft. per Sec.
PLANT OUTPUT, HP

Figure 68.

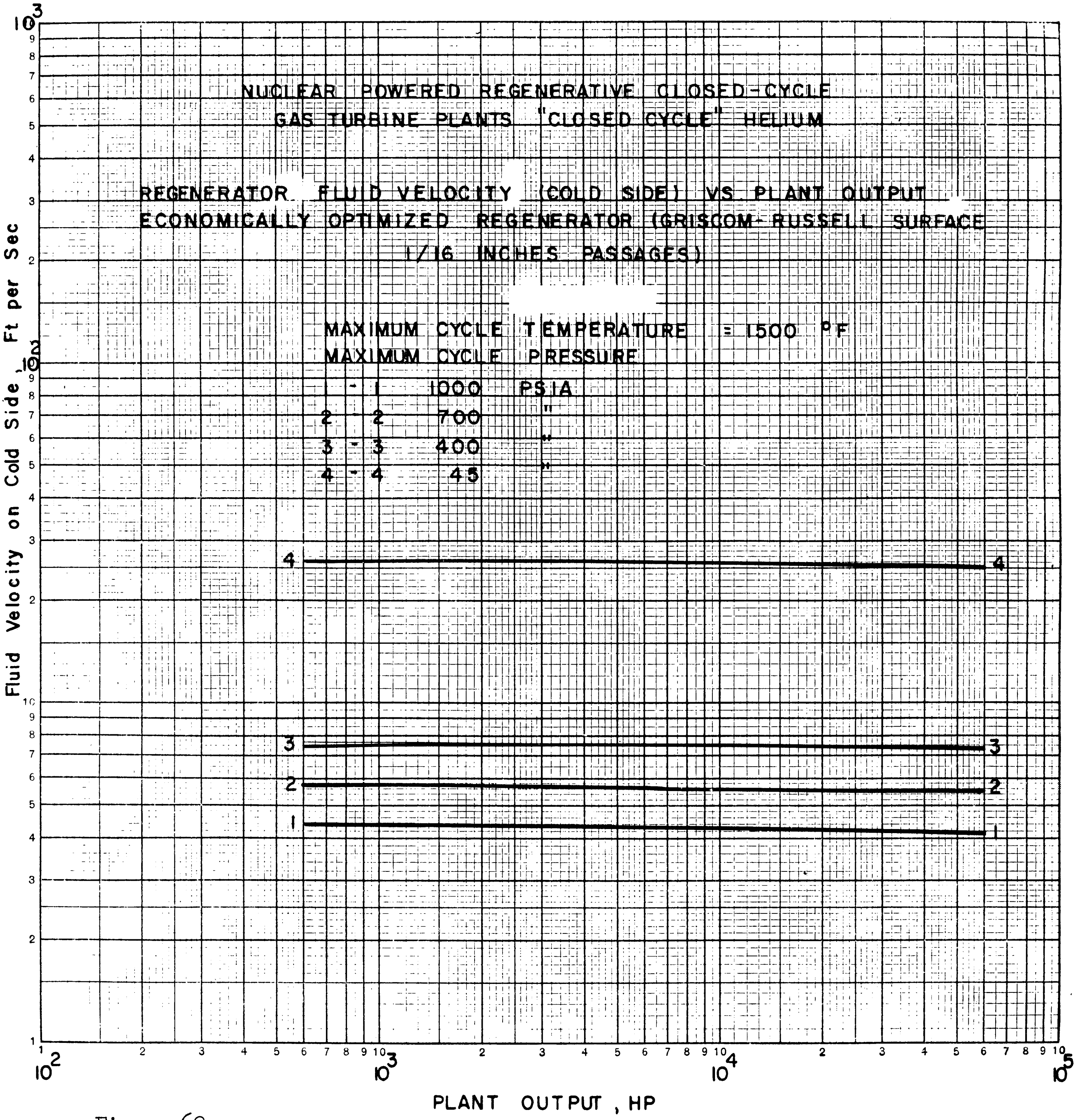

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "CLOSED CYCLE" HELIUM
REGENERATOR FLUID VELOCITY (COLD SIDE) VS PLANT OUTPUT
ECONOMICALLY OPTIMIZED REGENERATOR (GRISCOM-RUSSELL SURFACE
1/16 INCHES PASSAGES)
MAXIMUM CYCLE TEMPERATURE = 1500 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
Fluid Velocity on Cold Side, Ft per Sec
PLANT OUTPUT, HP

Figure 69.

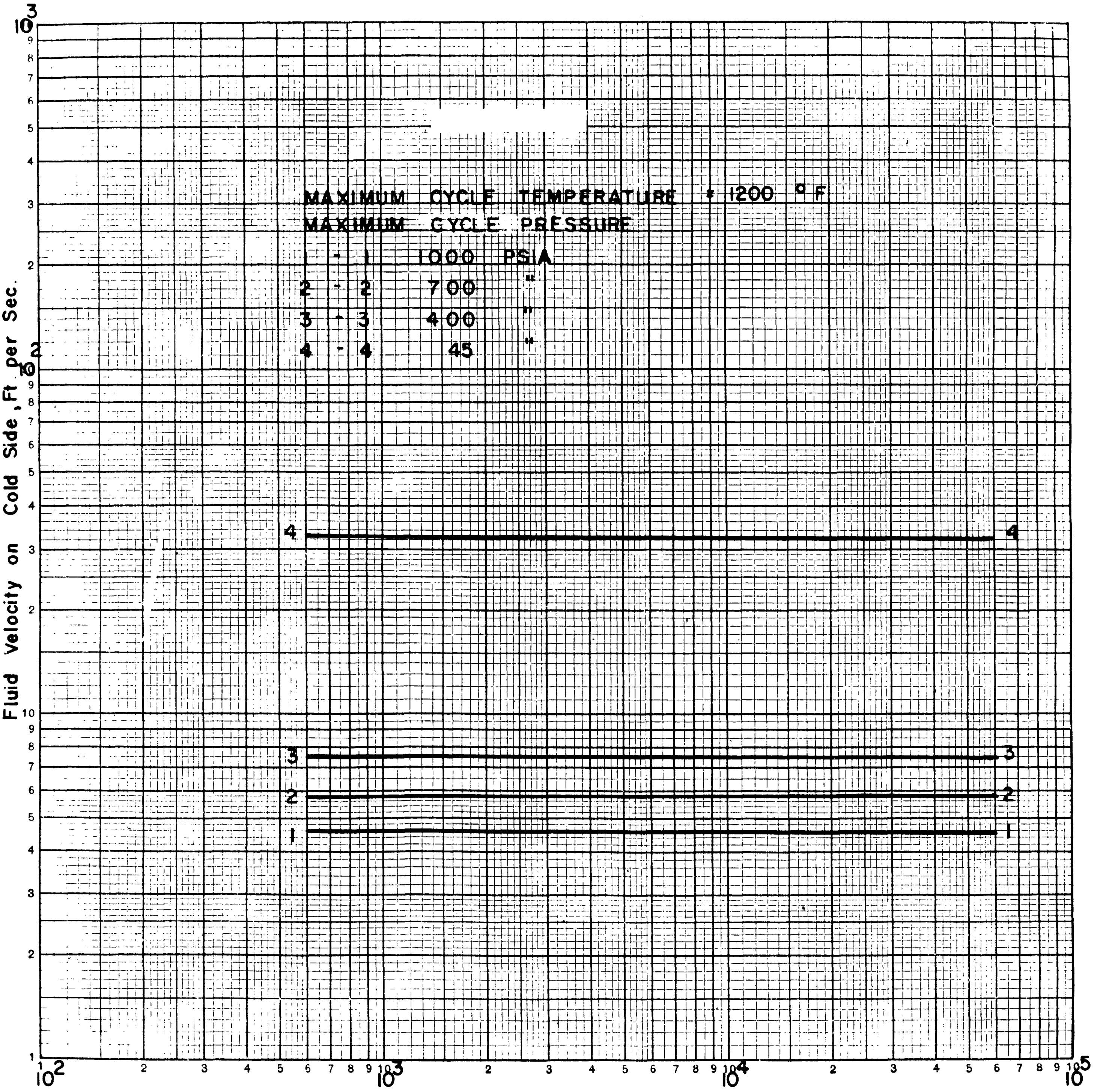
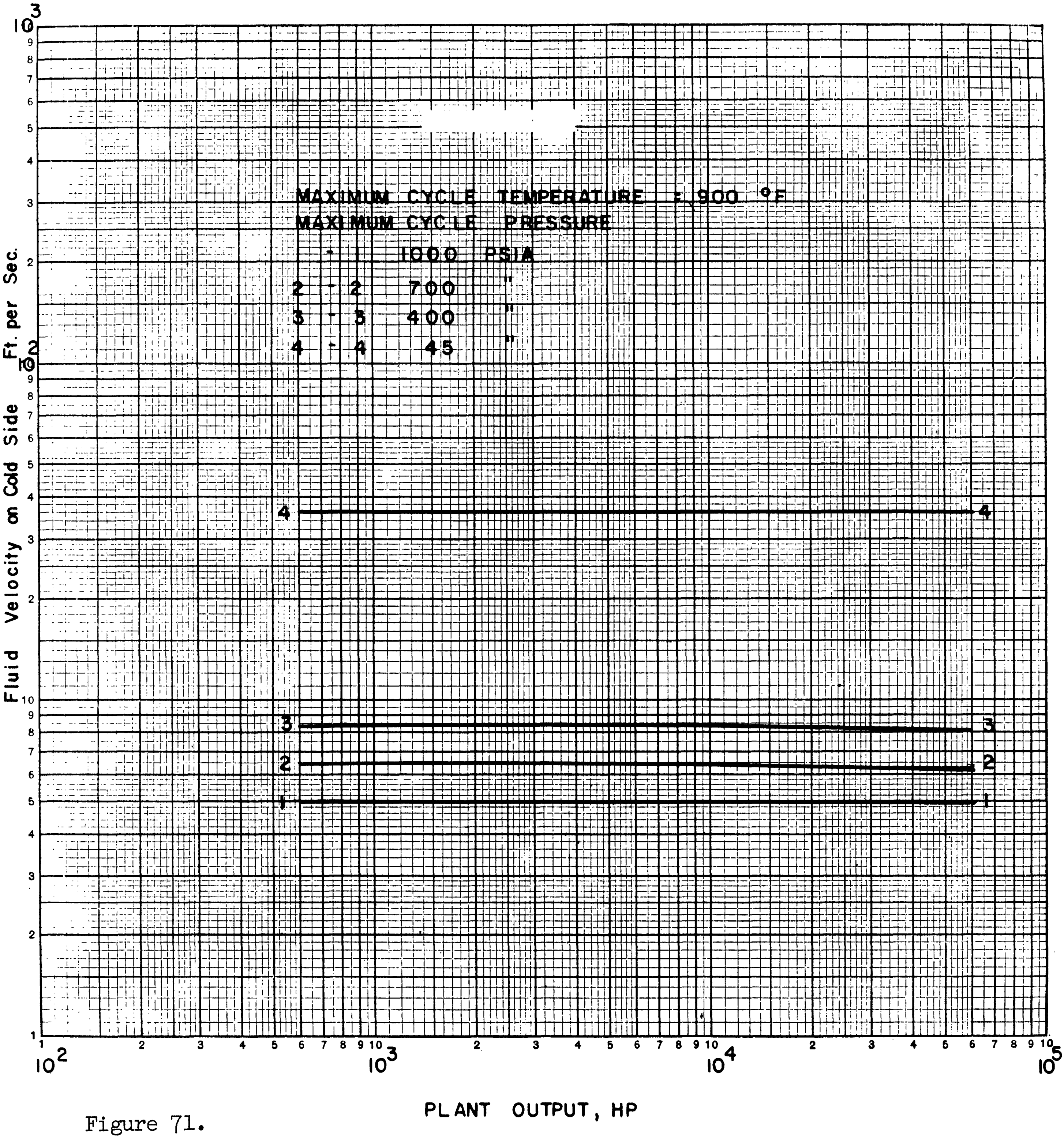

MAXIMUM CYCLE TEMPERATURE = 1200 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
Fluid Velocity on Cold Side, Ft per Sec.
PLANT OUTPUT, HP

Figure 70.



MAXIMUM CYCLE TEMPERATURE = 900 °F
MAXIMUM CYCLE PRESSURE
1 - 1 1000 PSIA
2 - 2 700 "
3 - 3 400 "
4 - 4 45 "
Fluid Velocity on Cold Side Ft. per Sec.
PLANT OUTPUT, HP

Figure 71.

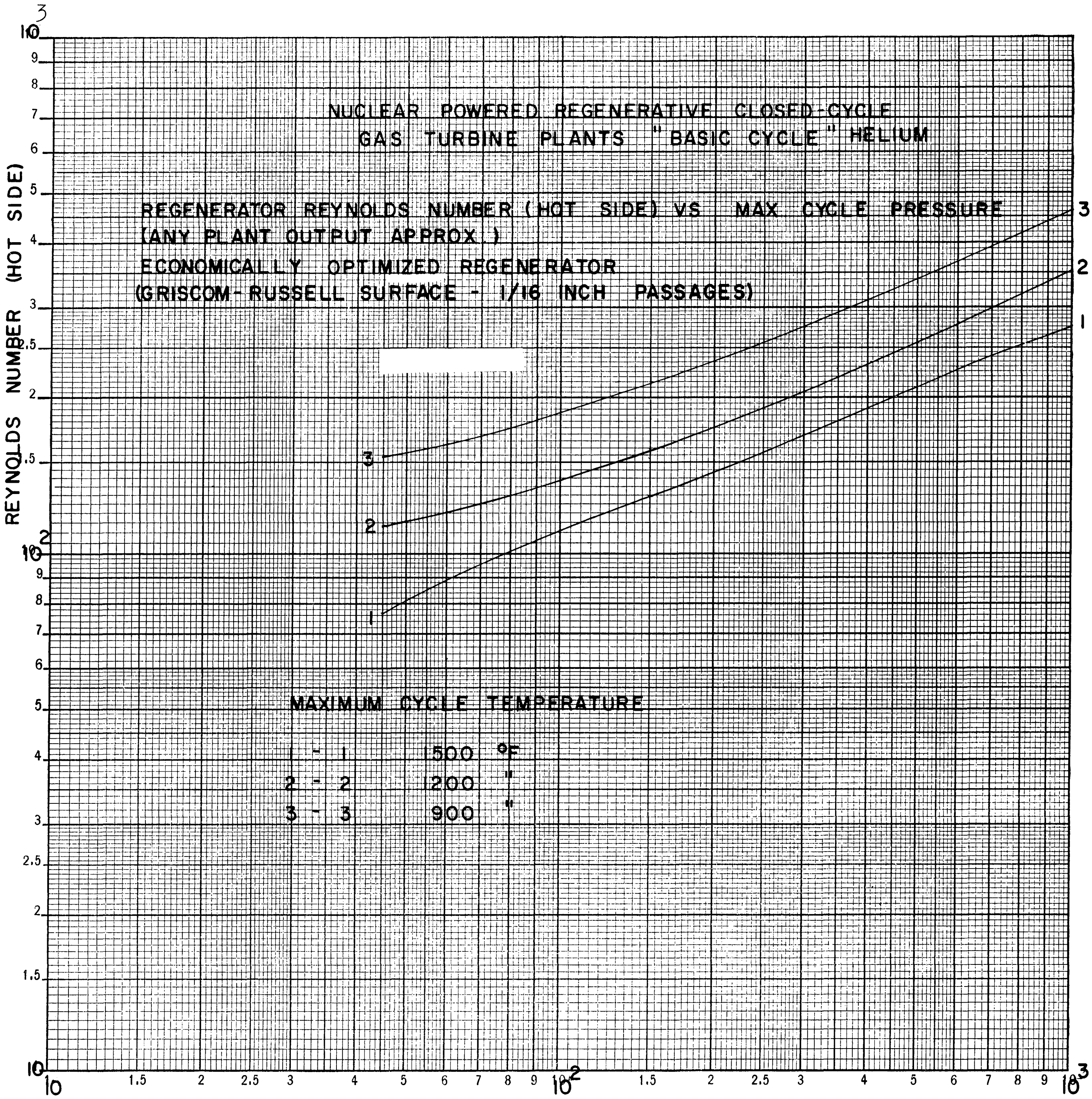


Figure 72. MAXIMUM CYCLE PRESSURE, PSIA

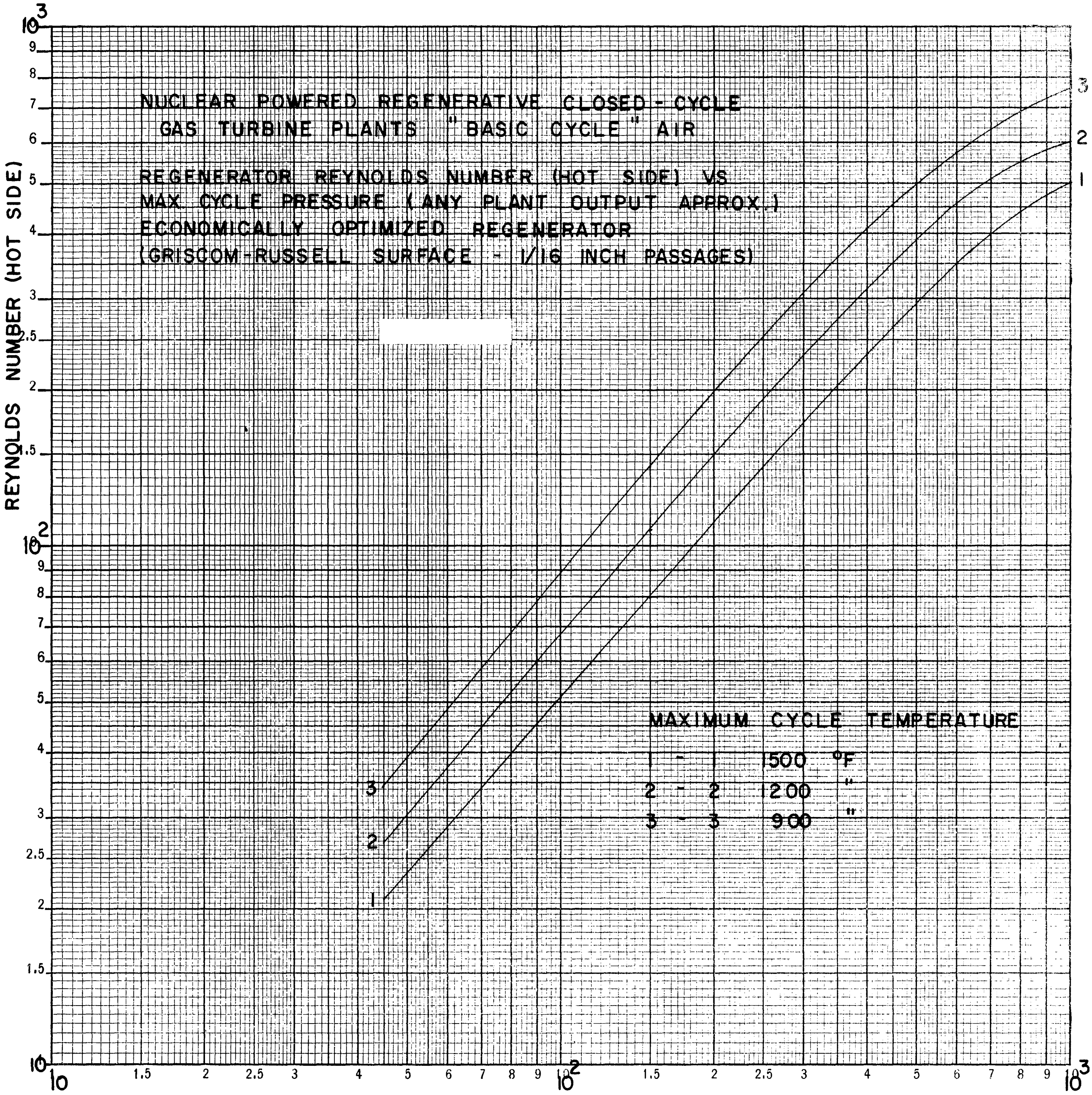

NUCLEAR POWERED REGENERATIVE CLOSED - CYCLE
GAS TURBINE PLANTS "BASIC CYCLE" AIR
REGENERATOR REYNOLDS NUMBER (HOT SIDE) VS
MAX CYCLE PRESSURE (ANY PLANT OUTPUT APPROX.)
ECONOMICALLY OPTIMIZED REGENERATOR
(GRISCOM-RUSSELL SURFACE - 1/16 INCH PASSAGES)
MAXIMUM CYCLE TEMPERATURE
1 - 1 1500 °F
2 - 2 1200 "
3 - 3 900 "
REYNOLDS NUMBER (HOT SIDE)
MAXIMUM CYCLE PRESSURE, PSIA

Figure 73.

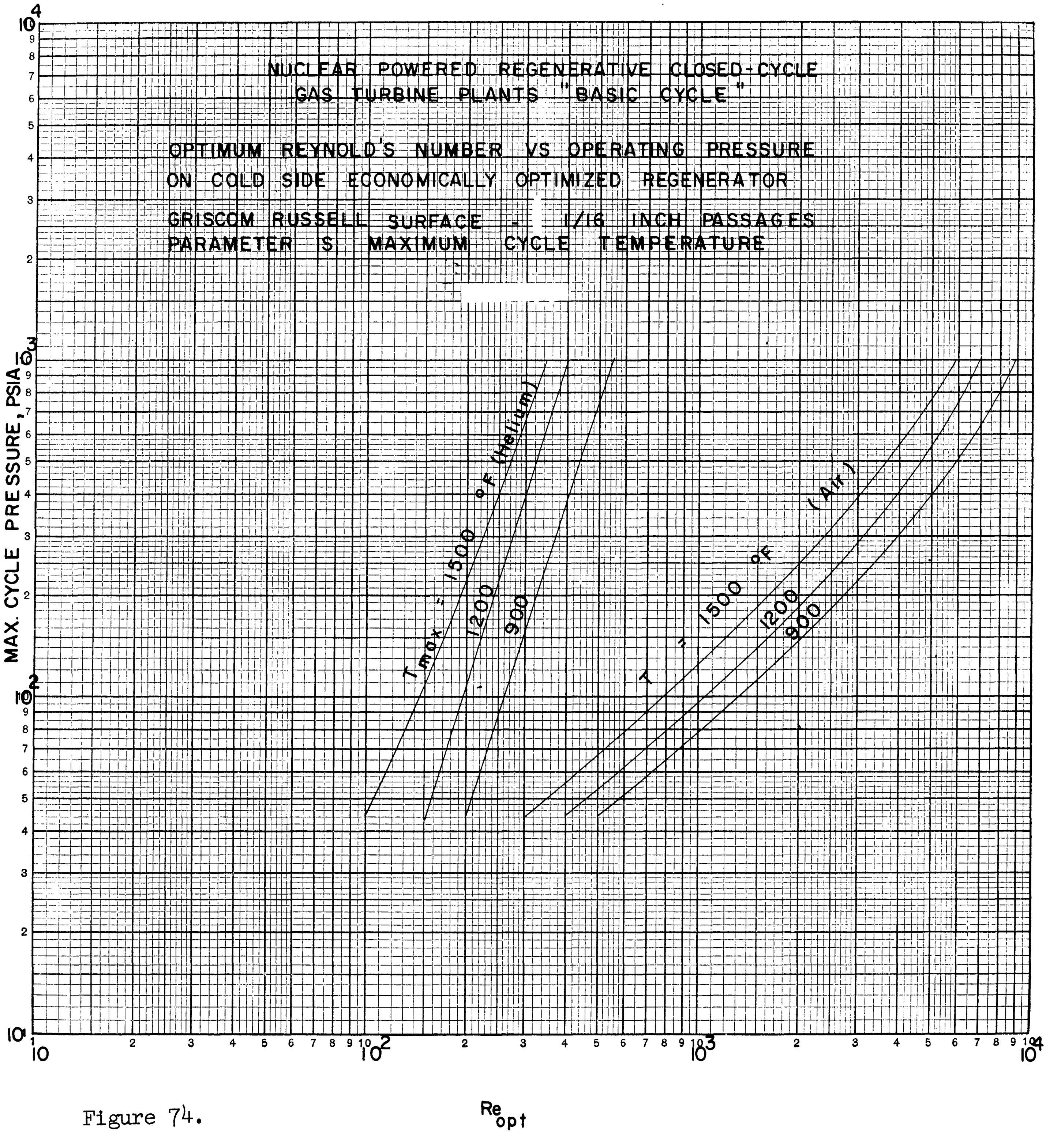

NUCLEAR POWERED REGENERATIVE CLOSED-CYCLE
GAS TURBINE PLANTS "BASIC CYCLE"
OPTIMUM REYNOLD'S NUMBER VS OPERATING PRESSURE
ON COLD SIDE ECONOMICALLY OPTIMIZED REGENERATOR
GRISCOM RUSSELL SURFACE - I 1/16 INCH PASSAGES
PARAMETER IS MAXIMUM CYCLE TEMPERATURE
MAX. CYCLE PRESSURE, PSIA
Tmax = 1500 °F (Helium)
1200
900
T = 1500 °F (Air)
1200
900
Re opt

Figure 74.

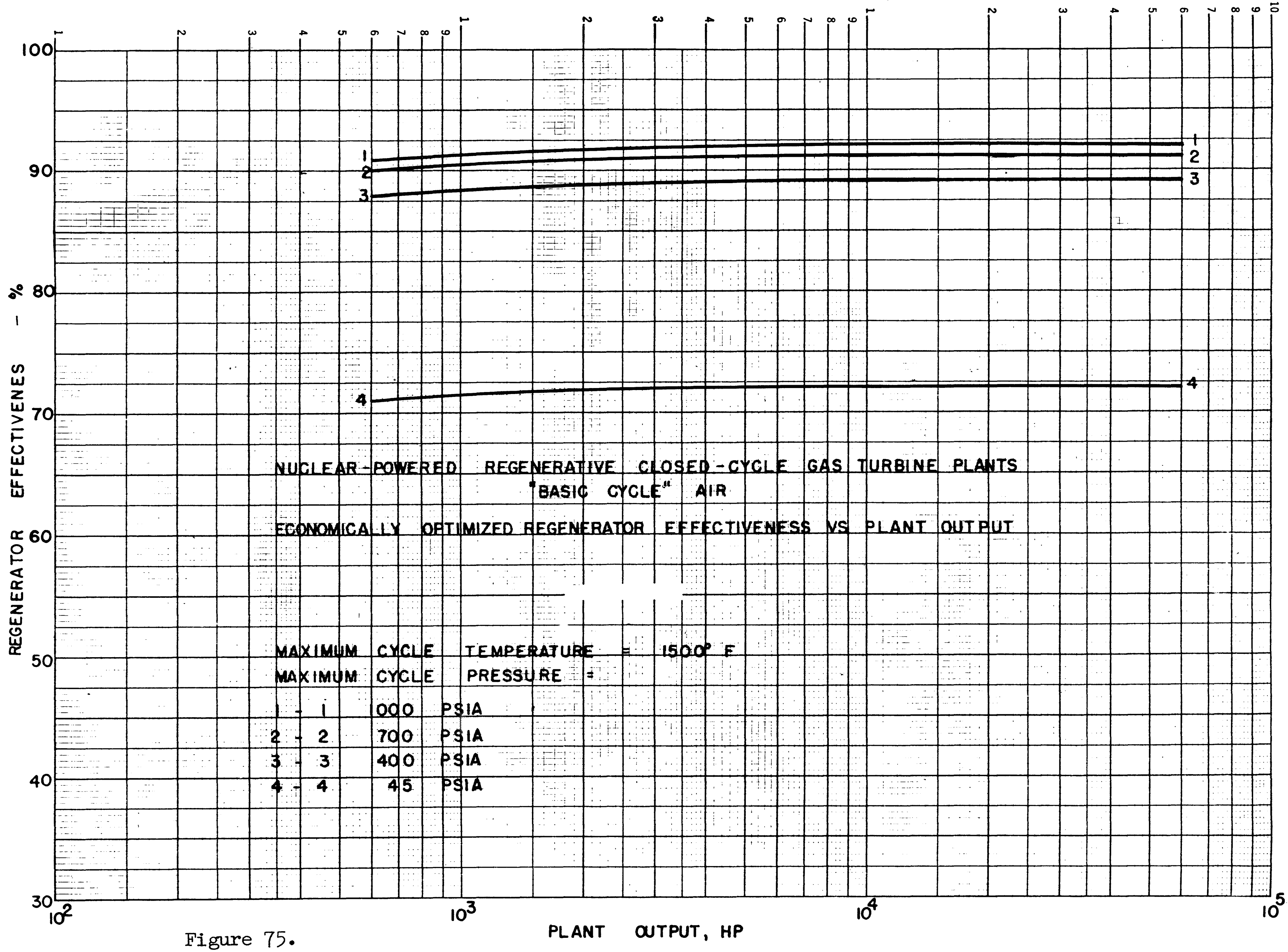

NUCLEAR-POWERED REGENERATIVE CLOSED-CYCLE GAS TURBINE PLANTS
"BASIC CYCLE" AIR
ECONOMICALLY OPTIMIZED REGENERATOR EFFECTIVENESS VS PLANT OUTPUT
MAXIMUM CYCLE TEMPERATURE = 1500° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
1
2
3
4
REGENERATOR EFFECTIVENES - %
100
90
80
70
60
50
40
30
PLANT OUTPUT, HP
10^2
10^3
10^4
10^5

Figure 75.

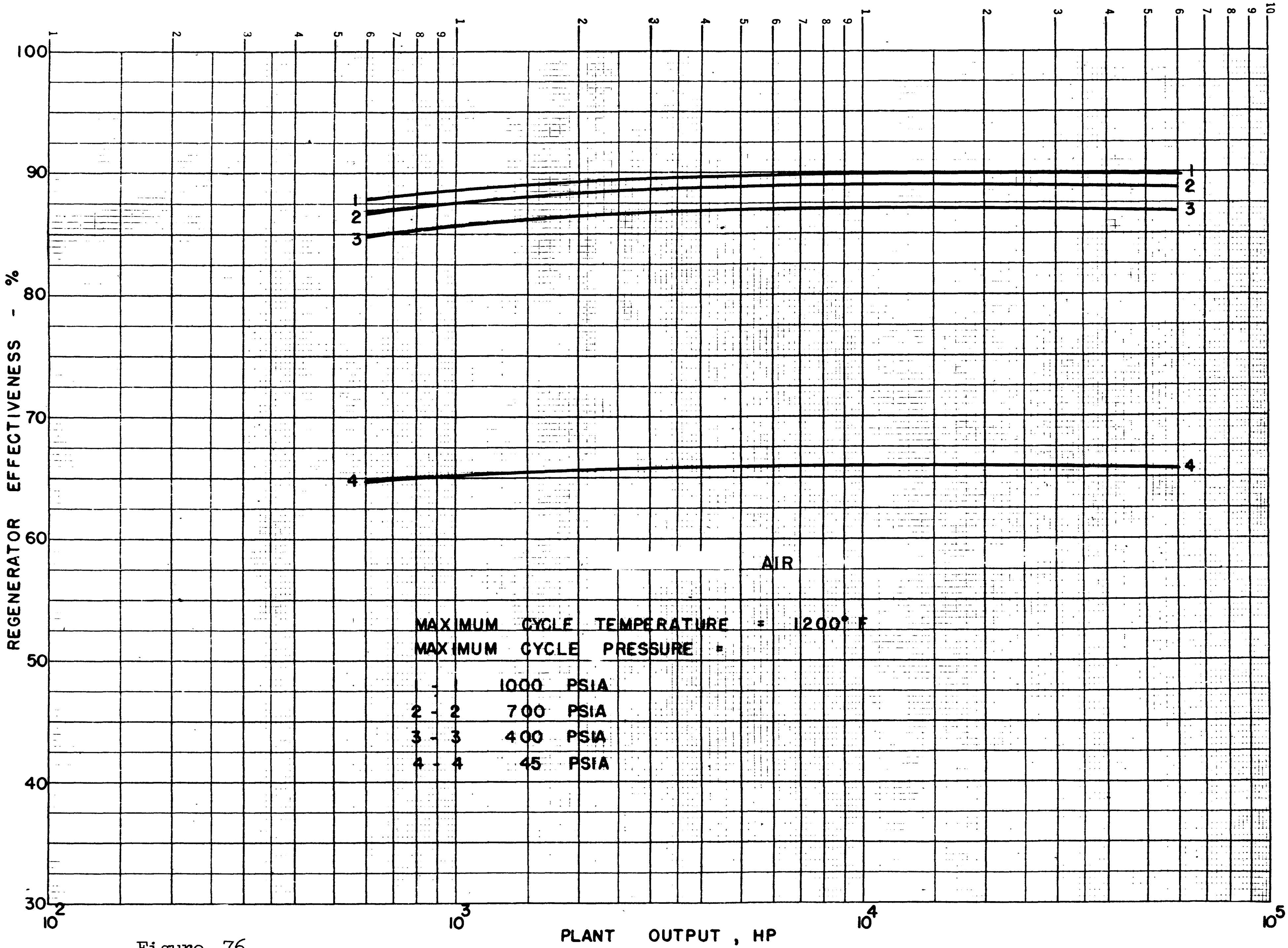

100
90
80
70
60
50
40
30
REGENERATOR EFFECTIVENESS - %
10^2
10^3
10^4
10^5
PLANT OUTPUT , HP
AIR
MAXIMUM CYCLE TEMPERATURE = 1200° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA

Figure 76.

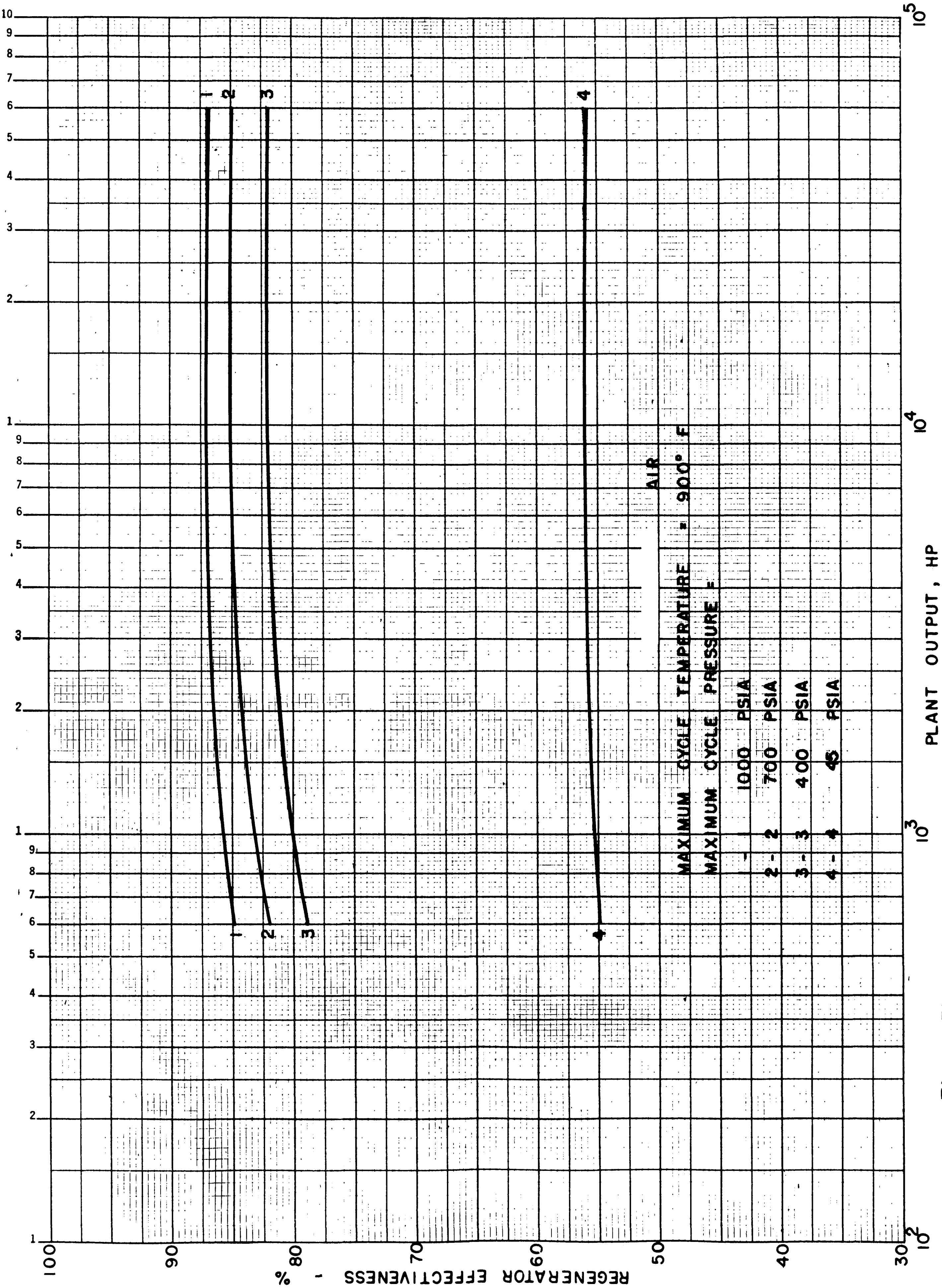

AIR
MAXIMUM CYCLE TEMPERATURE = 900° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
PLANT OUTPUT, HP
REGENERATOR EFFECTIVENESS - %
100
90
80
70
60
50
40
30
10^2
10^3
10^4
10^5

Figure 77.

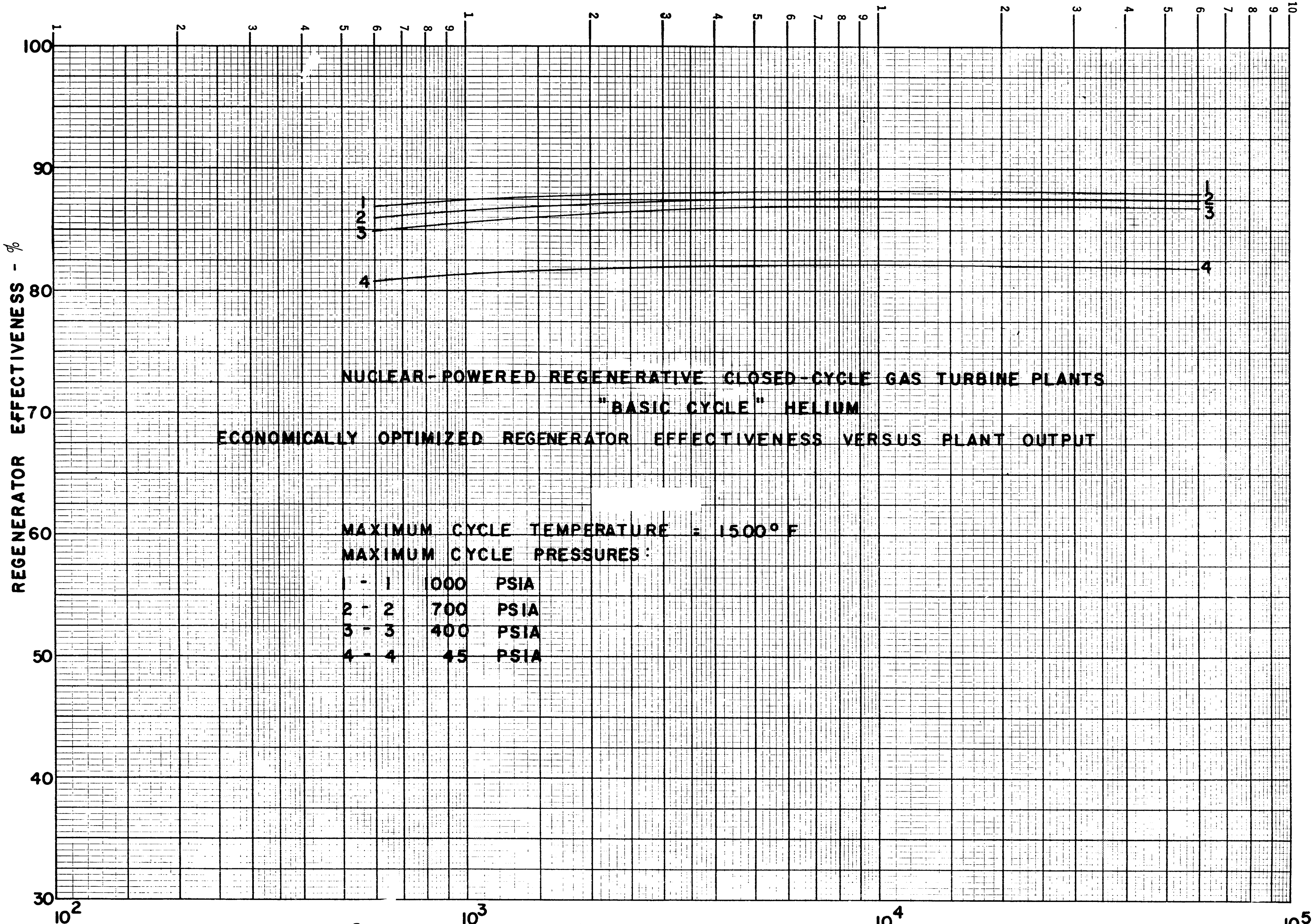

NUCLEAR-POWERED REGENERATIVE CLOSED-CYCLE GAS TURBINE PLANTS
"BASIC CYCLE" HELIUM
ECONOMICALLY OPTIMIZED REGENERATOR EFFECTIVENESS VERSUS PLANT OUTPUT
MAXIMUM CYCLE TEMPERATURE = 1500° F
MAXIMUM CYCLE PRESSURES:
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR EFFECTIVENESS - %
PLANT OUTPUT, HP

Figure 78.

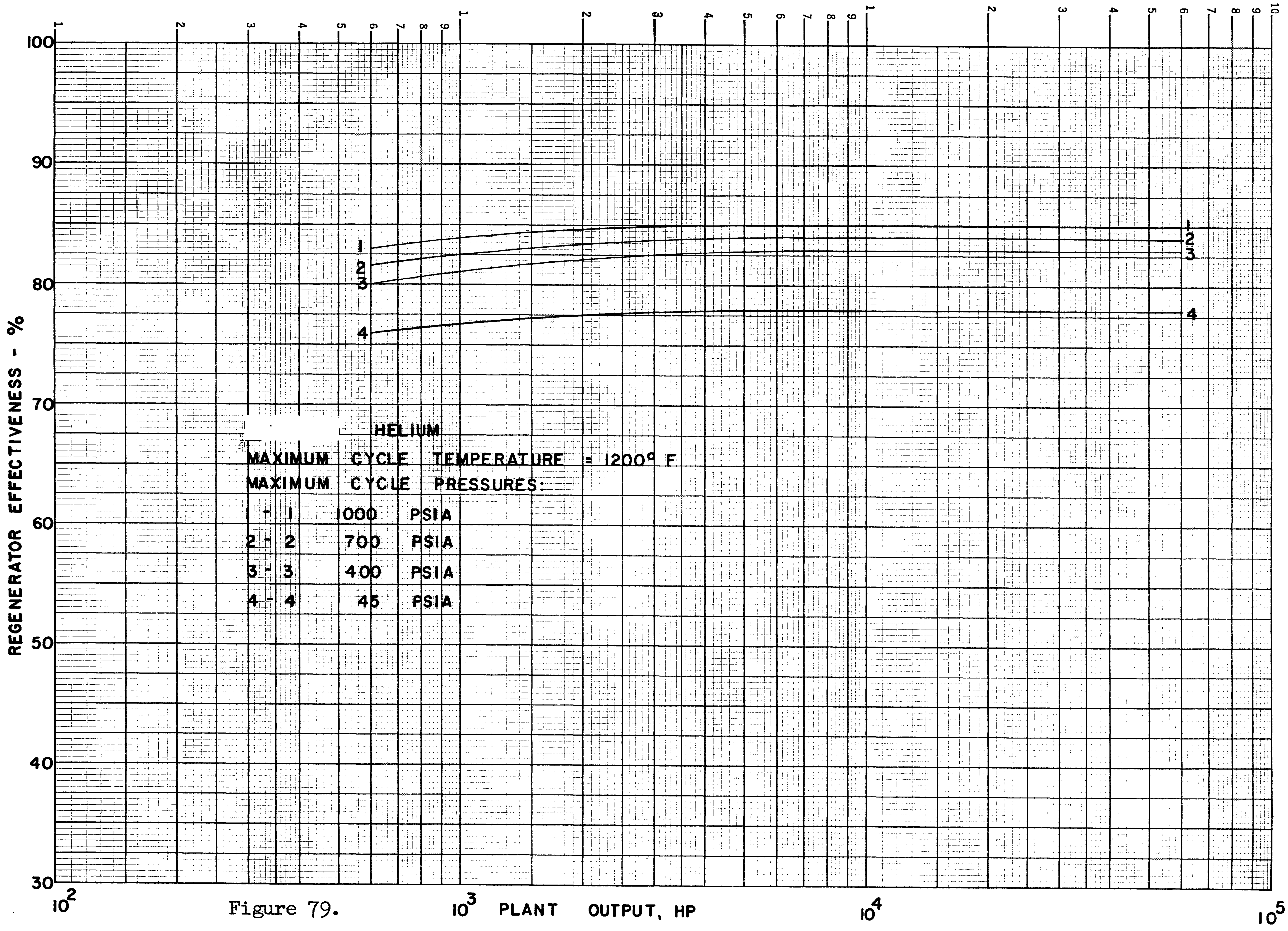

HELIUM
MAXIMUM CYCLE TEMPERATURE = 1200° F
MAXIMUM CYCLE PRESSURES:
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
REGENERATOR EFFECTIVENESS - %
PLANT OUTPUT, HP

Figure 79.

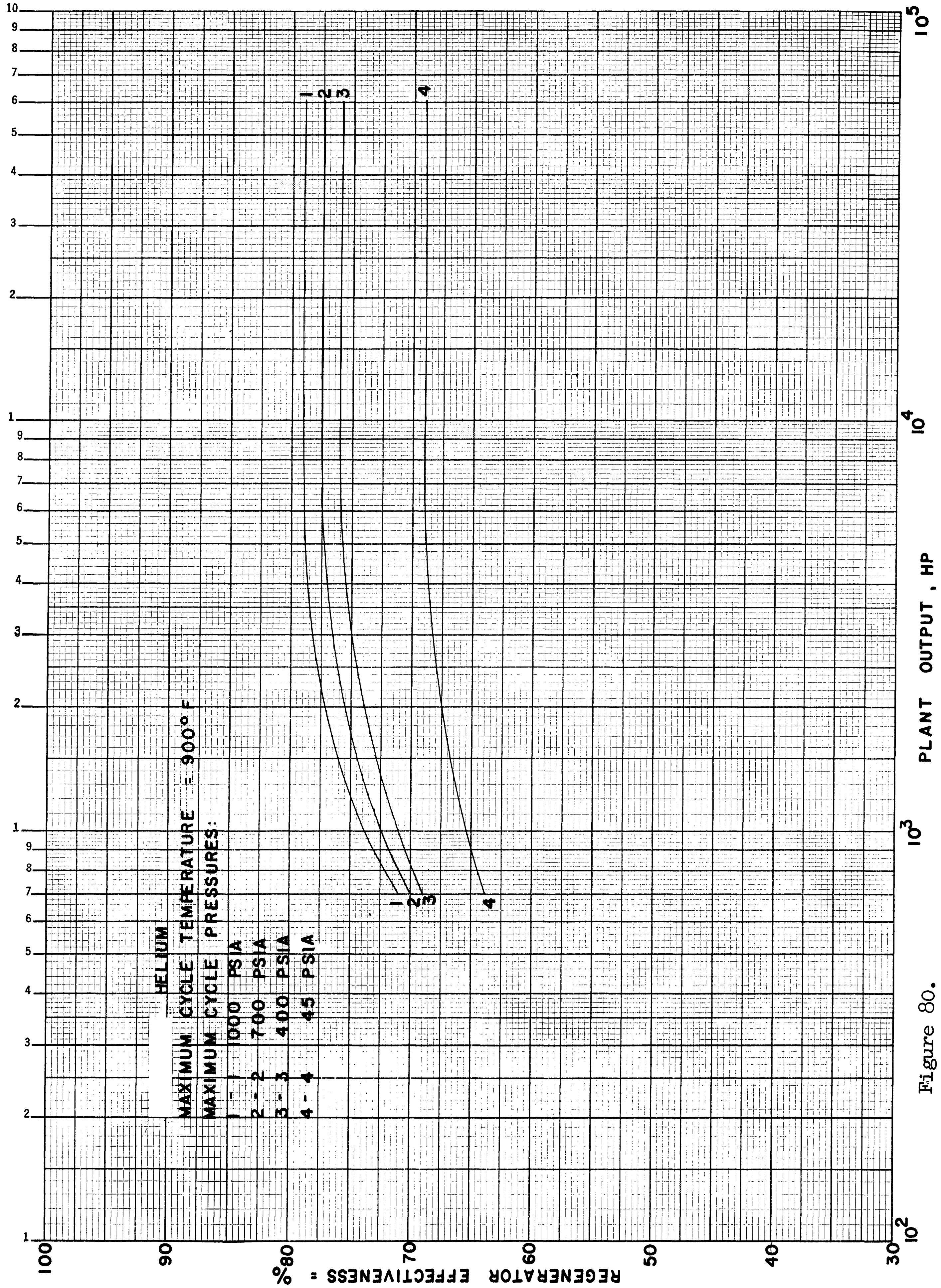

HELIUM
MAXIMUM CYCLE TEMPERATURE = 900°F
MAXIMUM CYCLE PRESSURES:
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
PLANT OUTPUT, HP
REGENERATOR EFFECTIVENESS = %
100
90
80
70
60
50
40
30
10^2
10^3
10^4
10^5

Figure 80.

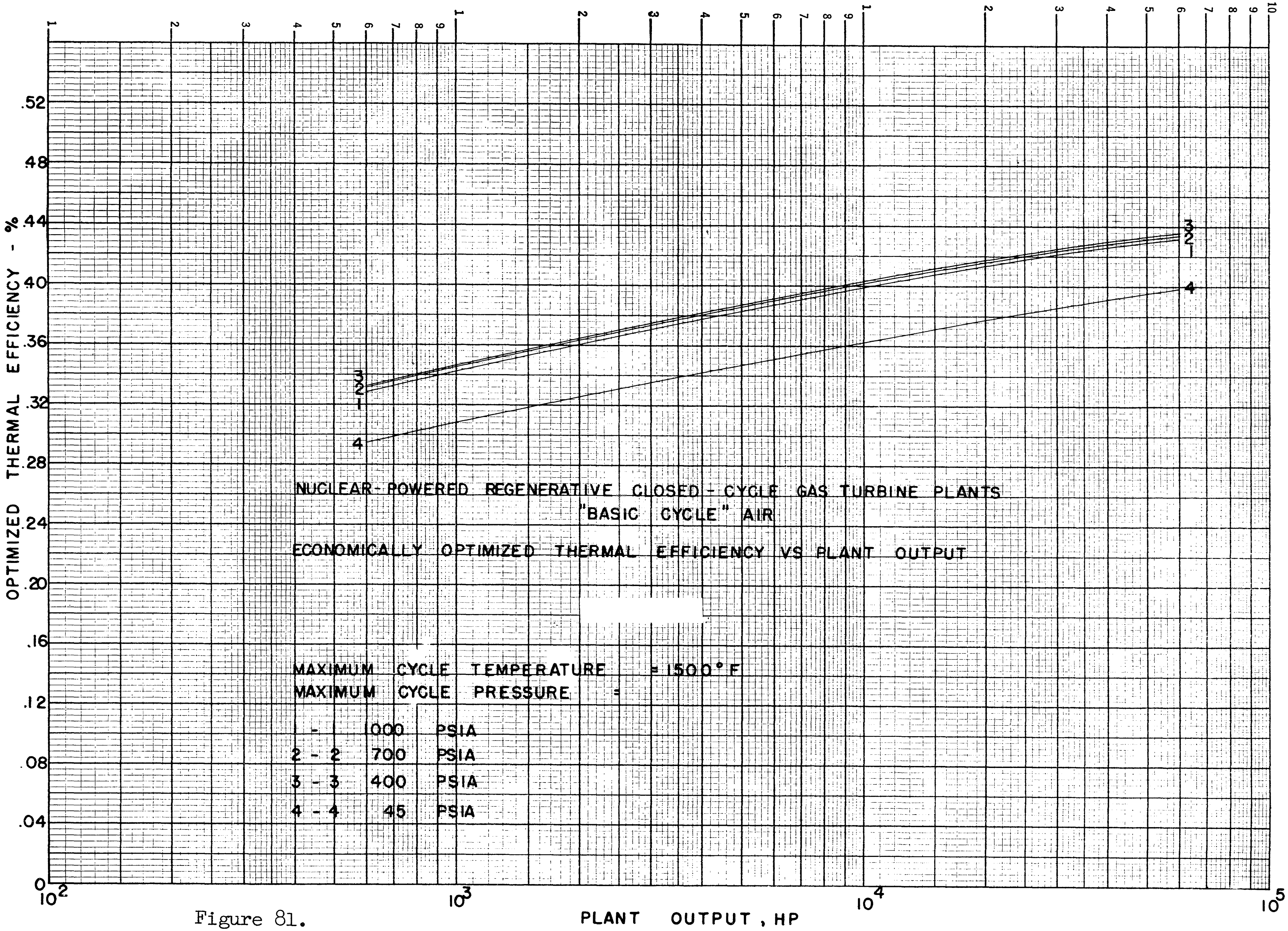

NUCLEAR - POWERED REGENERATIVE CLOSED - CYCLE GAS TURBINE PLANTS
"BASIC CYCLE" AIR
ECONOMICALLY OPTIMIZED THERMAL EFFICIENCY VS PLANT OUTPUT
MAXIMUM CYCLE TEMPERATURE = 1500° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
1
2
3
4
OPTIMIZED THERMAL EFFICIENCY - %
0
.04
.08
.12
.16
.20
.24
.28
.32
.36
.40
.44
.48
.52
PLANT OUTPUT, HP
10^2
10^3
10^4
10^5

Figure 81.

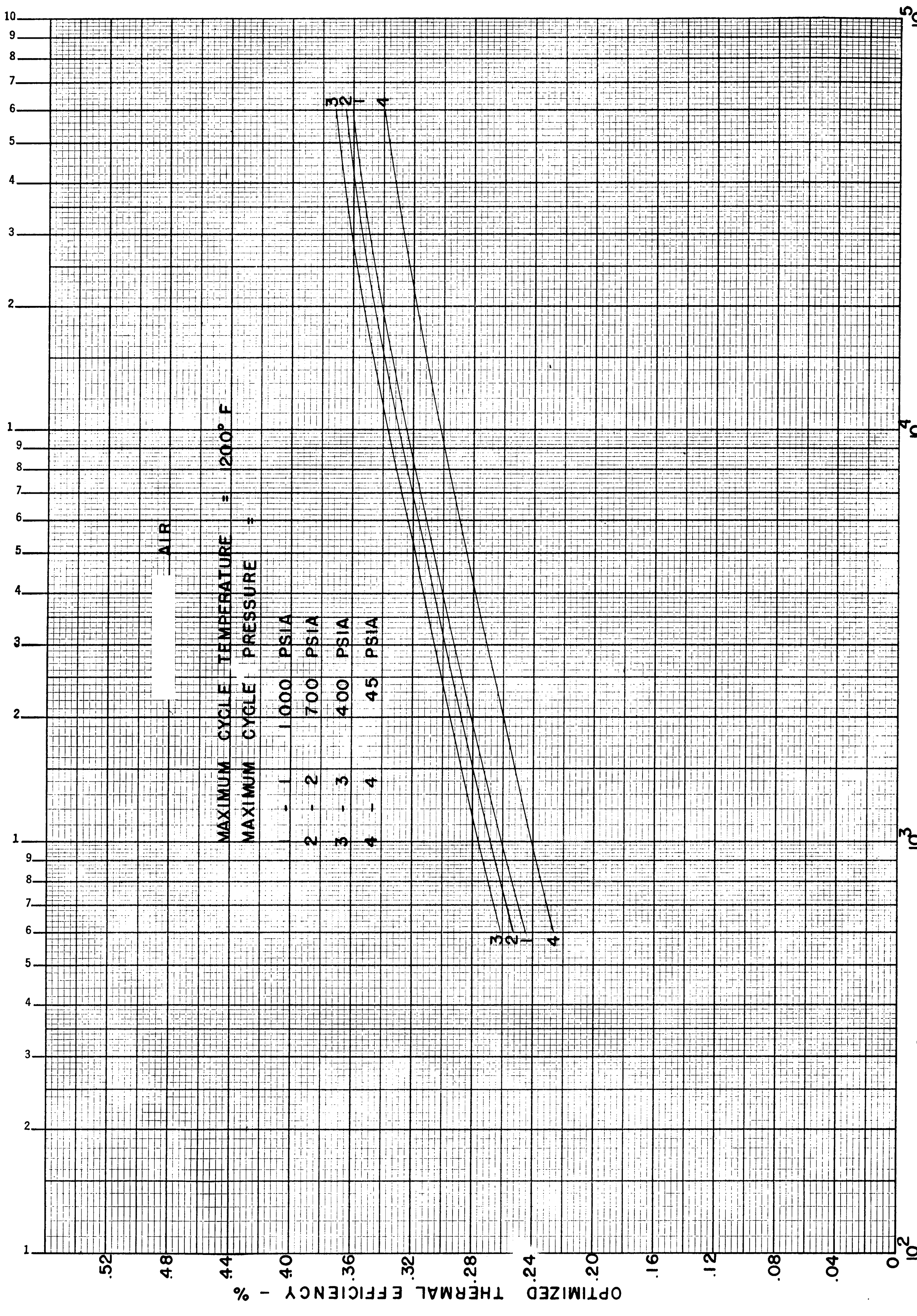

AIR
MAXIMUM CYCLE TEMPERATURE = 1200° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1,000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
PLANT OUTPUT, HP
OPTIMIZED THERMAL EFFICIENCY - %
.52
.48
.44
.40
.36
.32
.28
.24
.20
.16
.12
.08
.04
0
10^2
10^3
10^4
10^5

Figure 82.

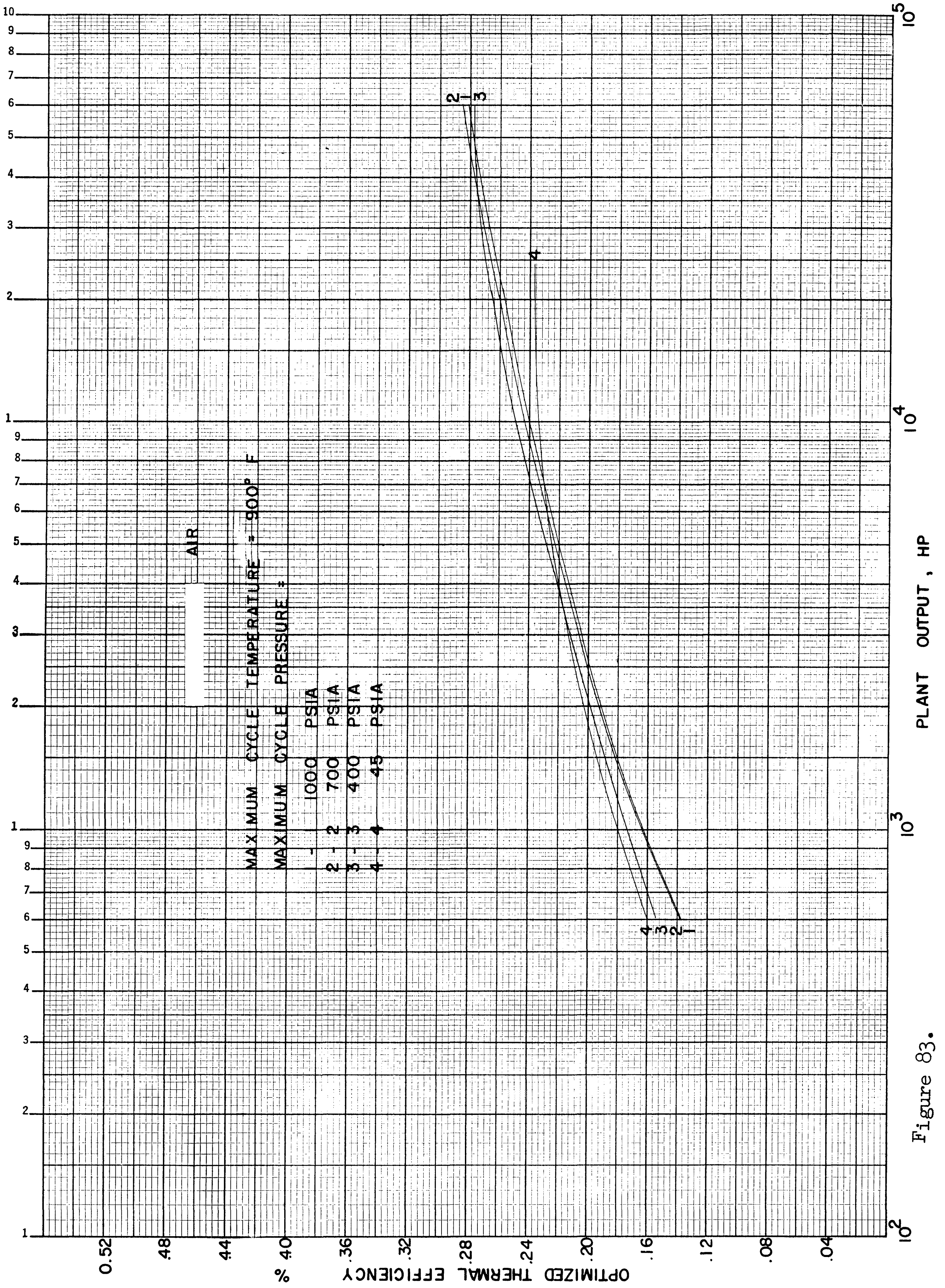

AIR
MAXIMUM CYCLE TEMPERATURE = 900° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
OPTIMIZED THERMAL EFFICIENCY %
PLANT OUTPUT, HP
0.52
.48
.44
.40
.36
.32
.28
.24
.20
.16
.12
.08
.04
10^2
10^3
10^4
10^5

Figure 83.

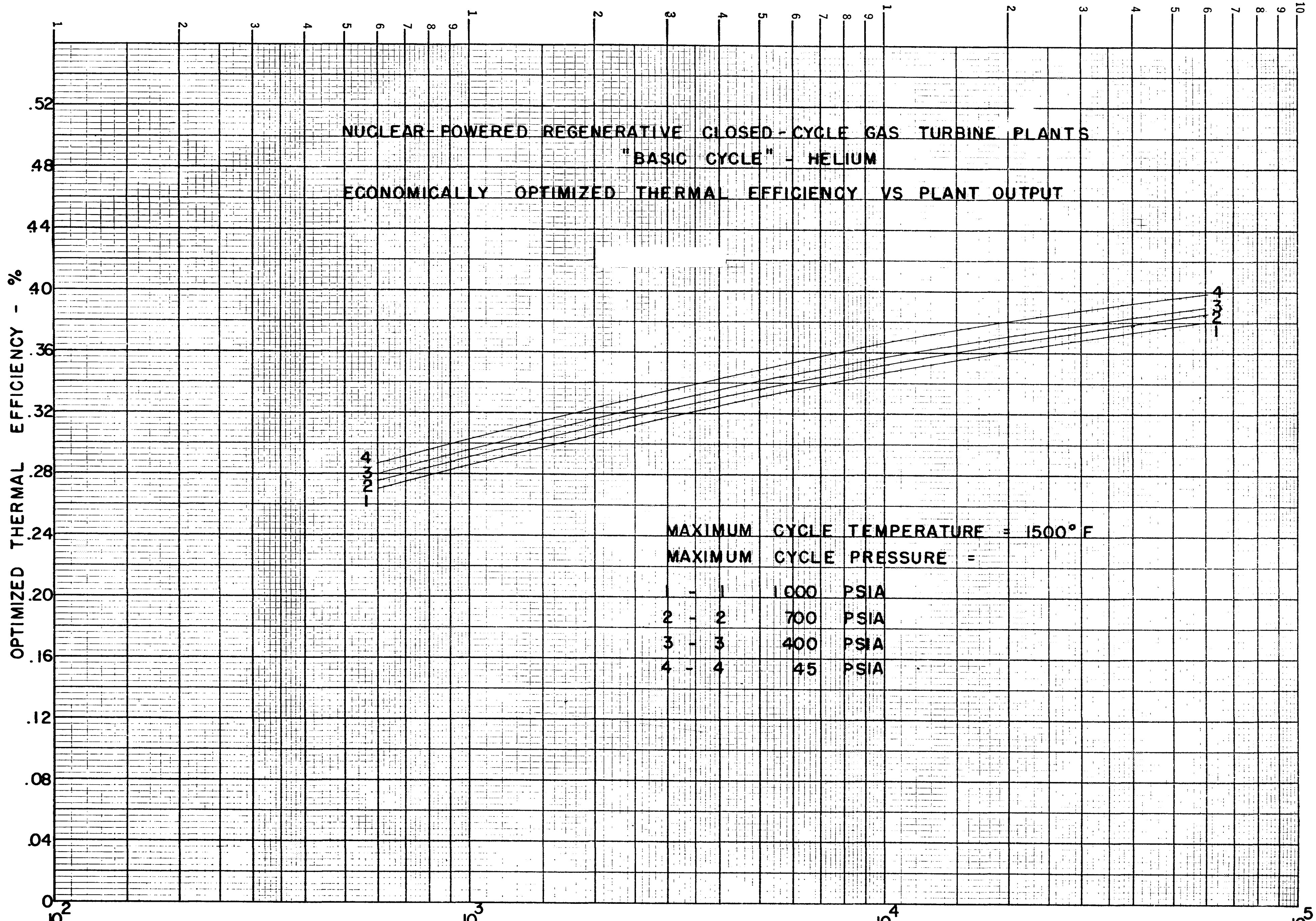

NUCLEAR-POWERED REGENERATIVE CLOSED-CYCLE GAS TURBINE PLANTS
"BASIC CYCLE" - HELIUM
ECONOMICALLY OPTIMIZED THERMAL EFFICIENCY VS PLANT OUTPUT
MAXIMUM CYCLE TEMPERATURE = 1500° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
1
2
3
4
OPTIMIZED THERMAL EFFICIENCY - %
0
.04
.08
.12
.16
.20
.24
.28
.32
.36
.40
.44
.48
.52
PLANT OUTPUT , HP
10^2
10^3
10^4
10^5

Figure 84.

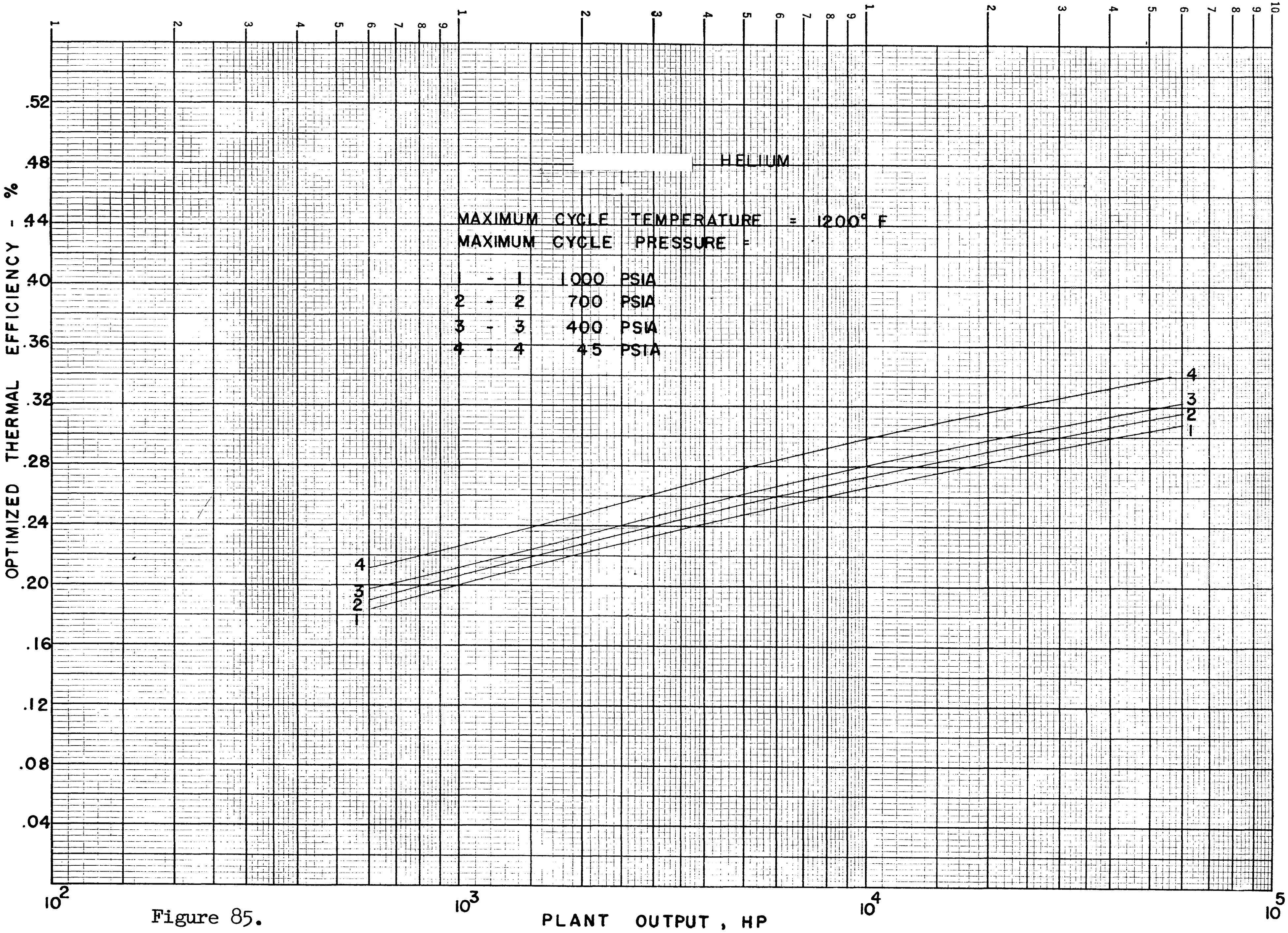

HELIUM
MAXIMUM CYCLE TEMPERATURE = 1200° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
OPTIMIZED THERMAL EFFICIENCY - %
.52
.48
.44
.40
.36
.32
.28
.24
.20
.16
.12
.08
.04
10^2
10^3
10^4
10^5
PLANT OUTPUT, HP
1
2
3
4

Figure 85.

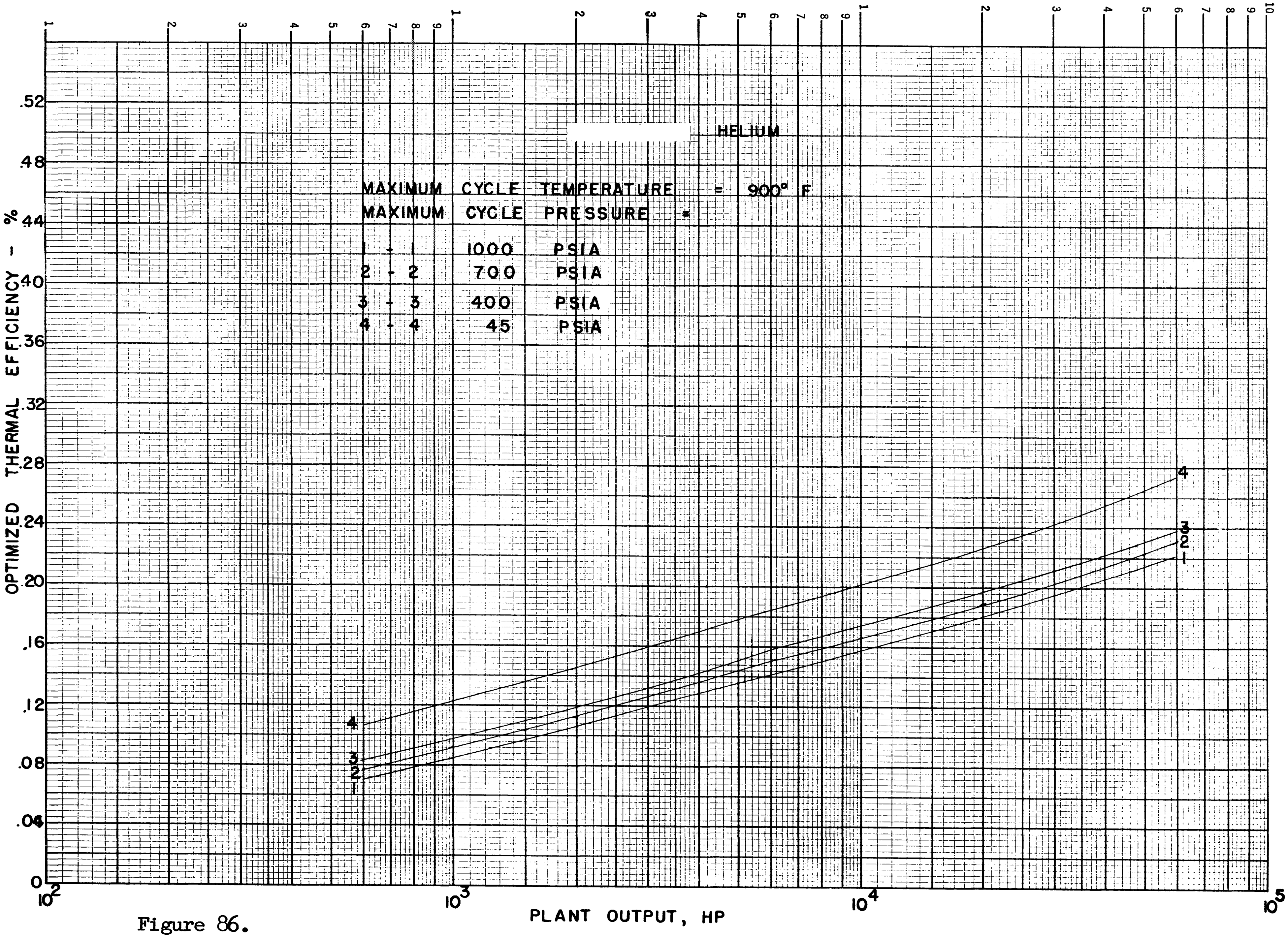
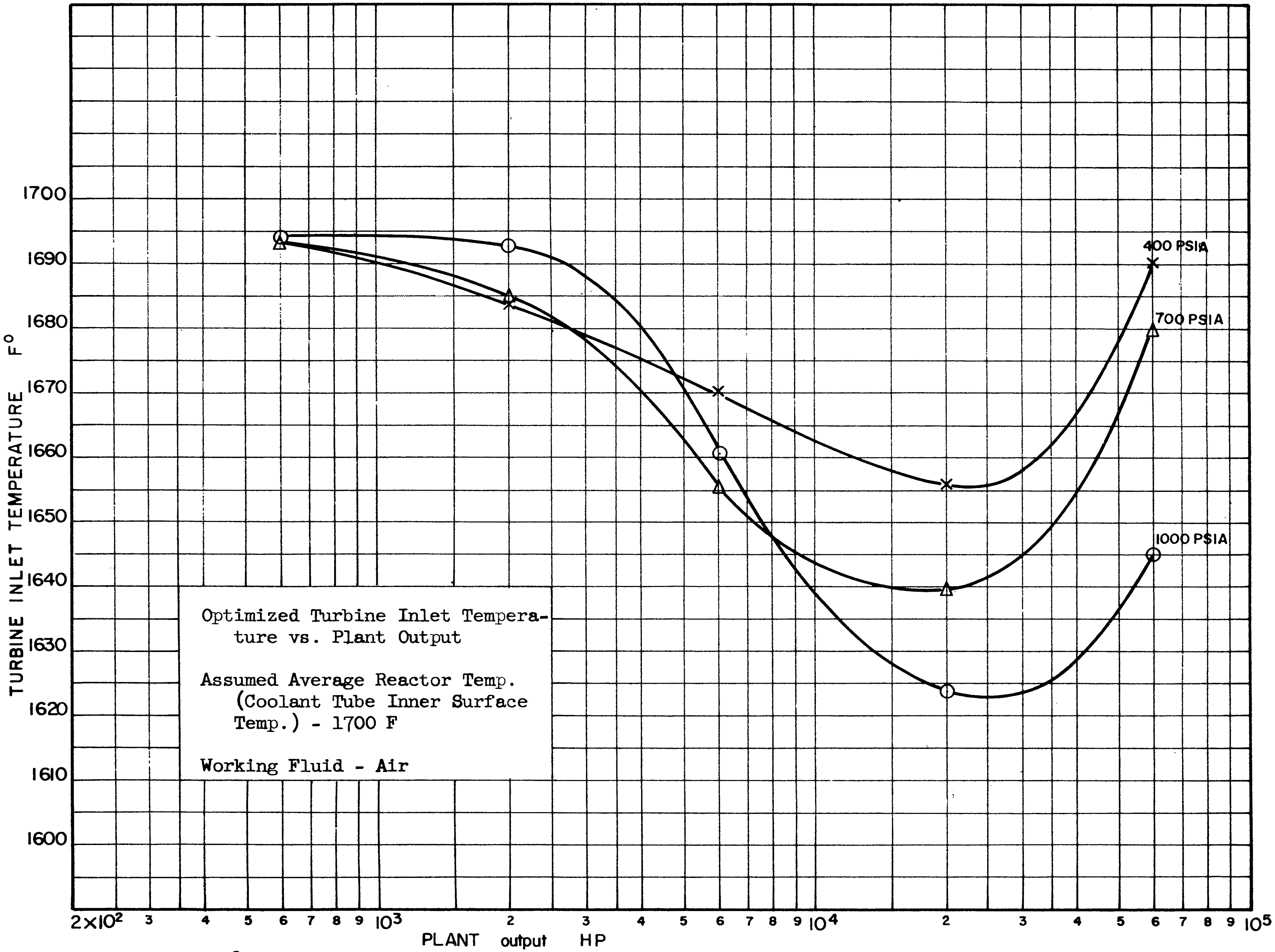
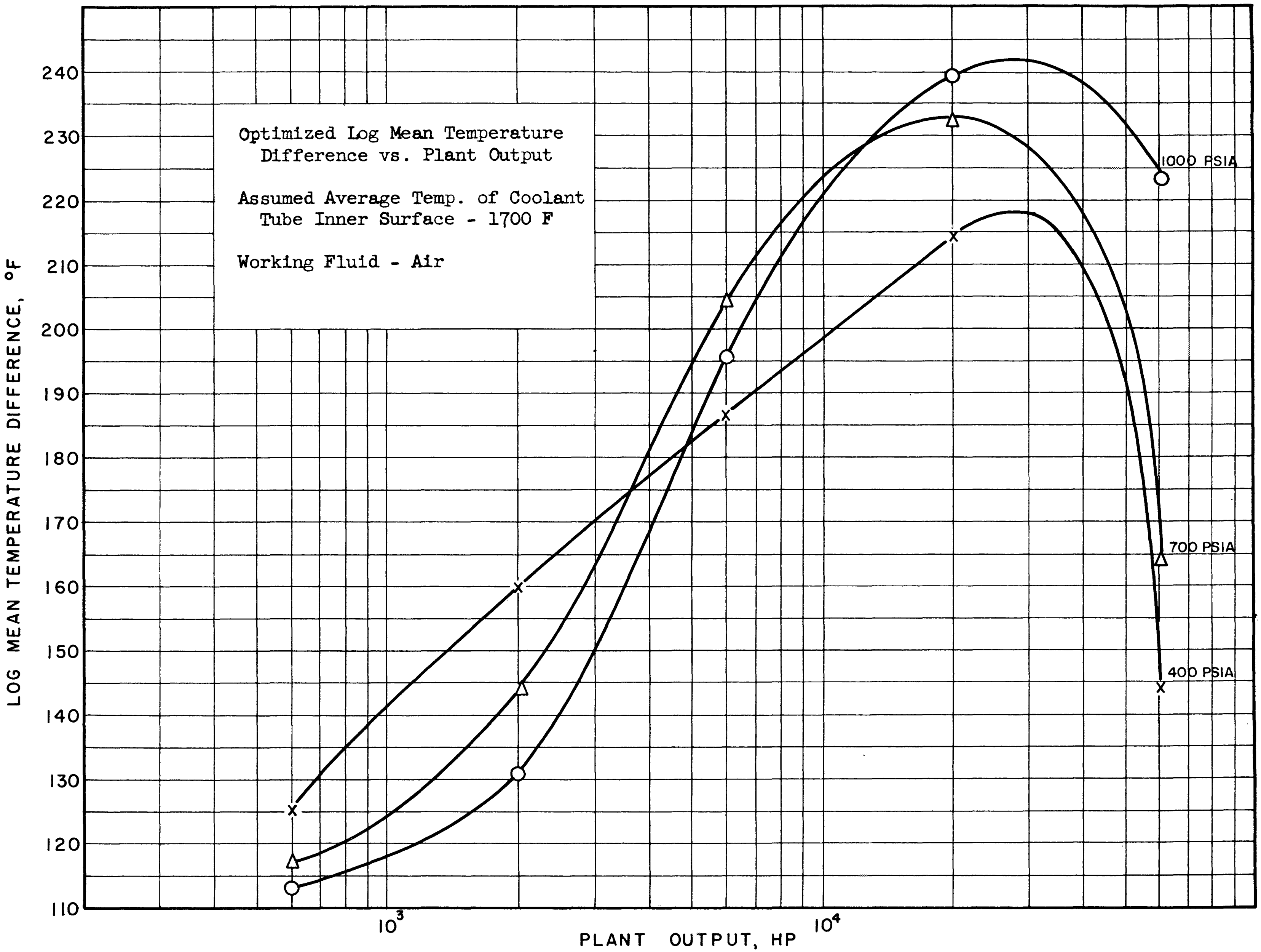

HELIUM
MAXIMUM CYCLE TEMPERATURE = 900° F
MAXIMUM CYCLE PRESSURE =
1 - 1 1000 PSIA
2 - 2 700 PSIA
3 - 3 400 PSIA
4 - 4 45 PSIA
OPTIMIZED THERMAL EFFICIENCY - %
PLANT OUTPUT, HP

Figure 86.



1700
1690
1680
1670
1660
1650
1640
1630
1620
1610
1600
TURBINE INLET TEMPERATURE F°
2×10²
3
4
5
6
7
8
9
10³
2
3
4
5
6
7
8
9
10⁴
2
3
4
5
6
7
8
9
10⁵
PLANT output HP
400 PSIA
700 PSIA
1000 PSIA
Optimized Turbine Inlet Temperature vs. Plant Output
Assumed Average Reactor Temp. (Coolant Tube Inner Surface Temp.) - 1700 F
Working Fluid - Air

Figure 87.


Optimized Log Mean Temperature Difference vs. Plant Output
Assumed Average Temp. of Coolant Tube Inner Surface - 1700 F
Working Fluid - Air
1000 PSIA
700 PSIA
400 PSIA
LOG MEAN TEMPERATURE DIFFERENCE, °F
240
230
220
210
200
190
180
170
160
150
140
130
120
110
10^3
10^4
PLANT OUTPUT, HP
Figure 88.

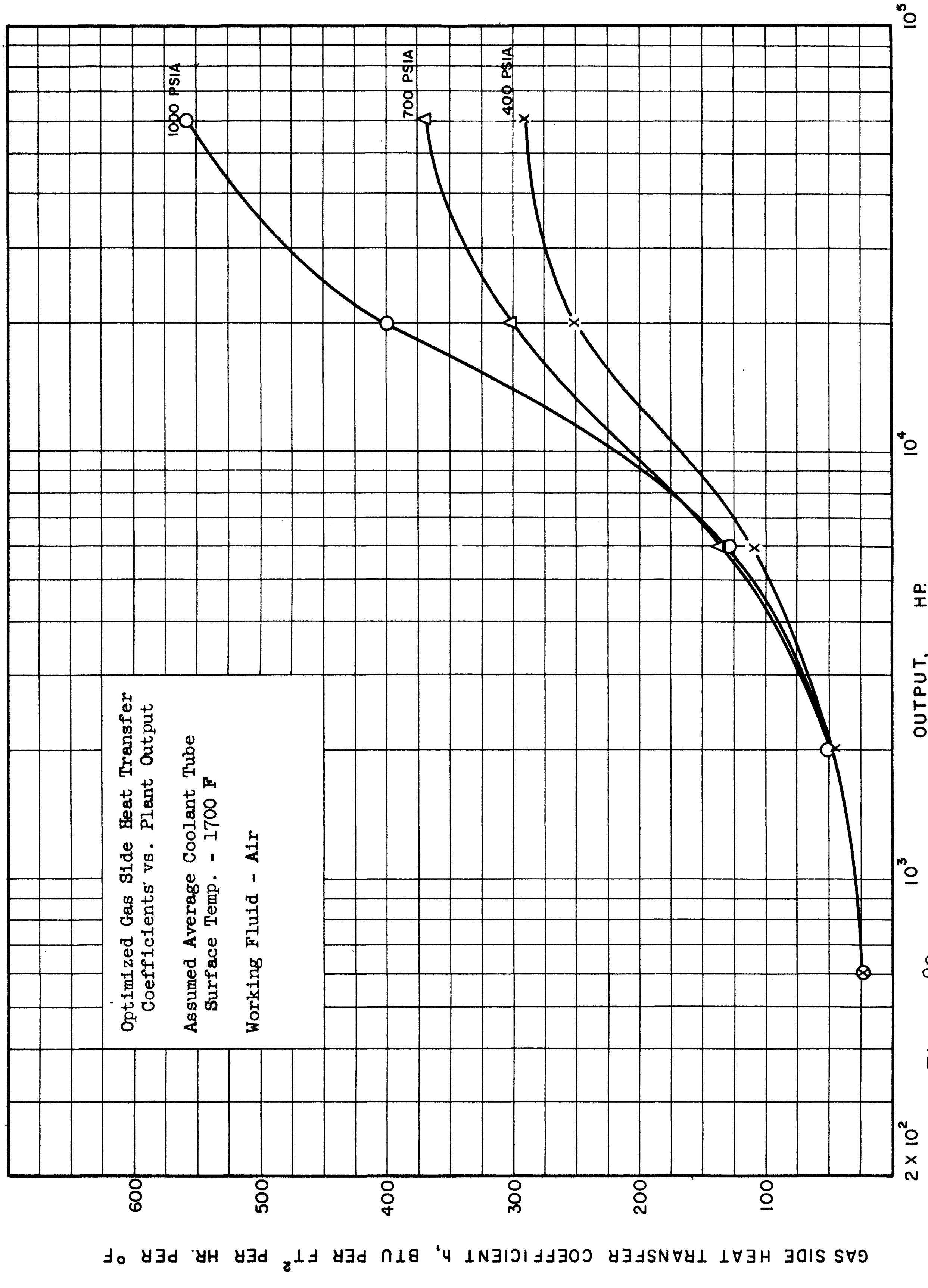

Optimized Gas Side Heat Transfer
Coefficients vs. Plant Output
Assumed Average Coolant Tube
Surface Temp. - 1700 F
Working Fluid - Air
1000 PSIA
700 PSIA
400 PSIA
600
500
400
300
200
100
2 X 10²
10³
10⁴
10⁵
OUTPUT, HP.
GAS SIDE HEAT TRANSFER COEFFICIENT h, BTU PER FT² PER HR. PER °F

Figure 89.

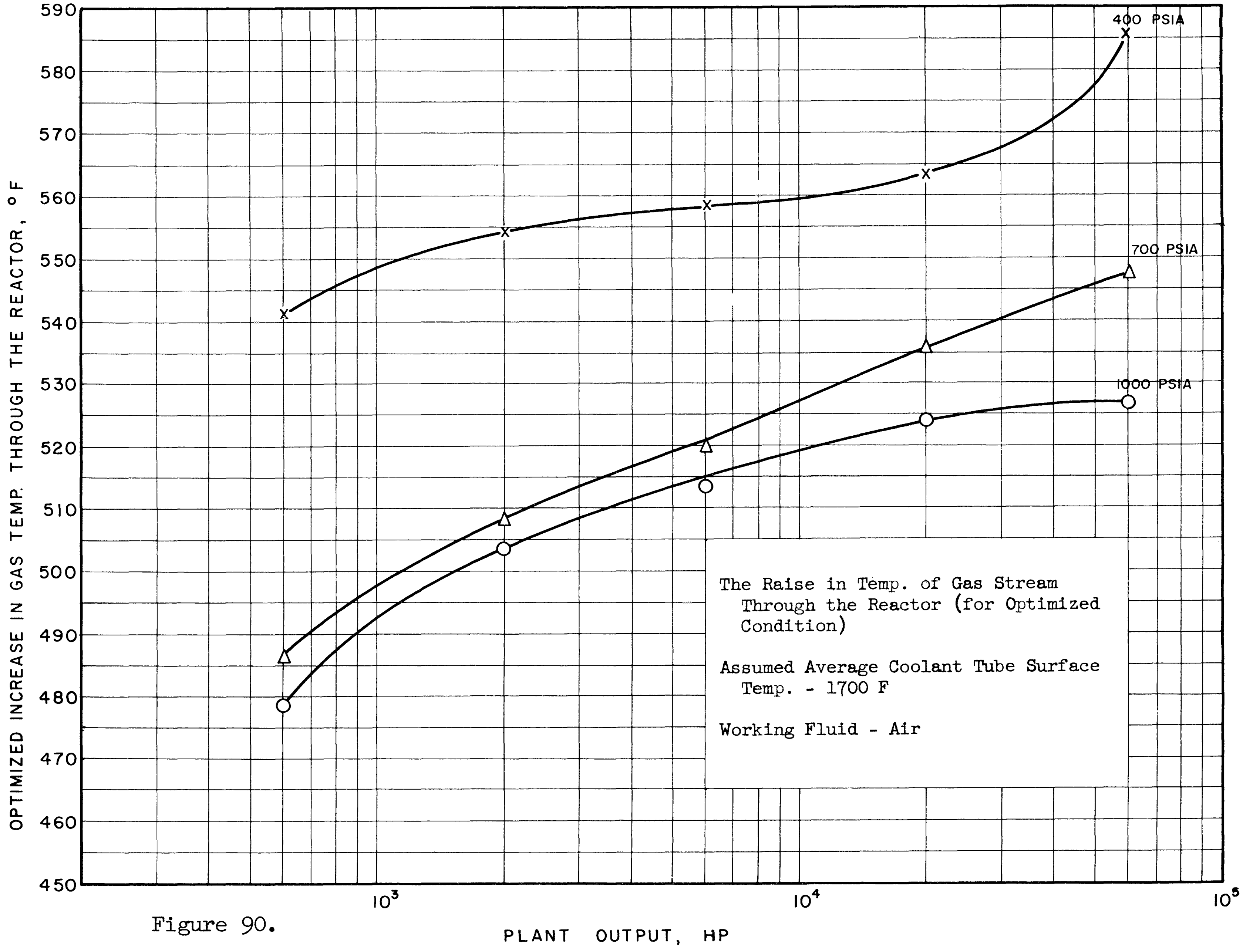
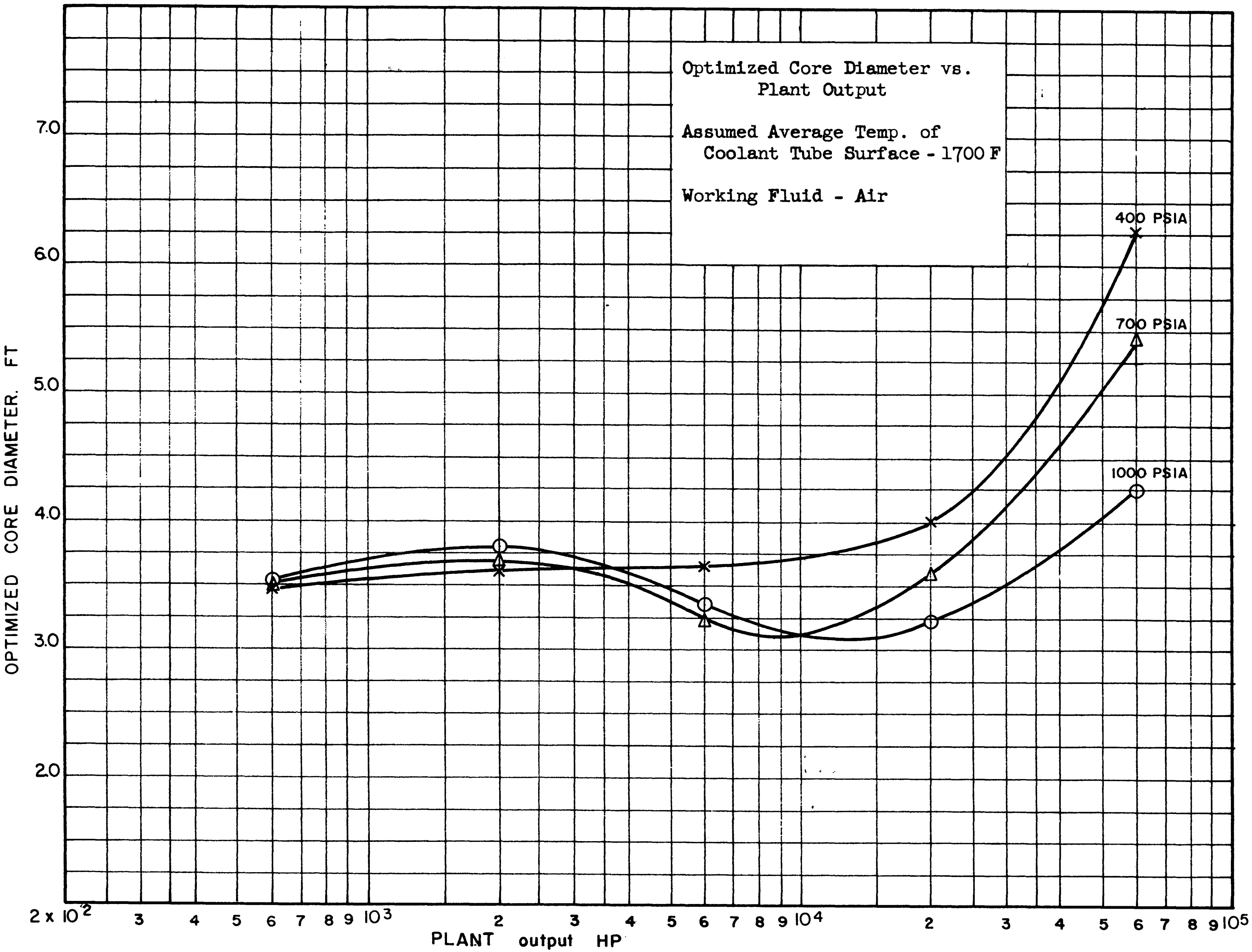

590
580
570
560
550
540
530
520
510
500
490
480
470
460
450
OPTIMIZED INCREASE IN GAS TEMP. THROUGH THE REACTOR, °F
10^3
10^4
10^5
PLANT OUTPUT, HP
400 PSIA
700 PSIA
1000 PSIA
The Raise in Temp. of Gas Stream Through the Reactor (for Optimized Condition)
Assumed Average Coolant Tube Surface Temp. - 1700 F
Working Fluid - Air

Figure 90.



Optimized Core Diameter vs. Plant Output
Assumed Average Temp. of Coolant Tube Surface - 1700 F
Working Fluid - Air
400 PSIA
700 PSIA
1000 PSIA
OPTIMIZED CORE DIAMETER. FT
7.0
6.0
5.0
4.0
3.0
2.0
2 x 10^2
3
4
5
6
7
8
9
10^3
2
3
4
5
6
7
8
9
10^4
2
3
4
5
6
7
8
9
10^5
PLANT output HP

Figure 91.

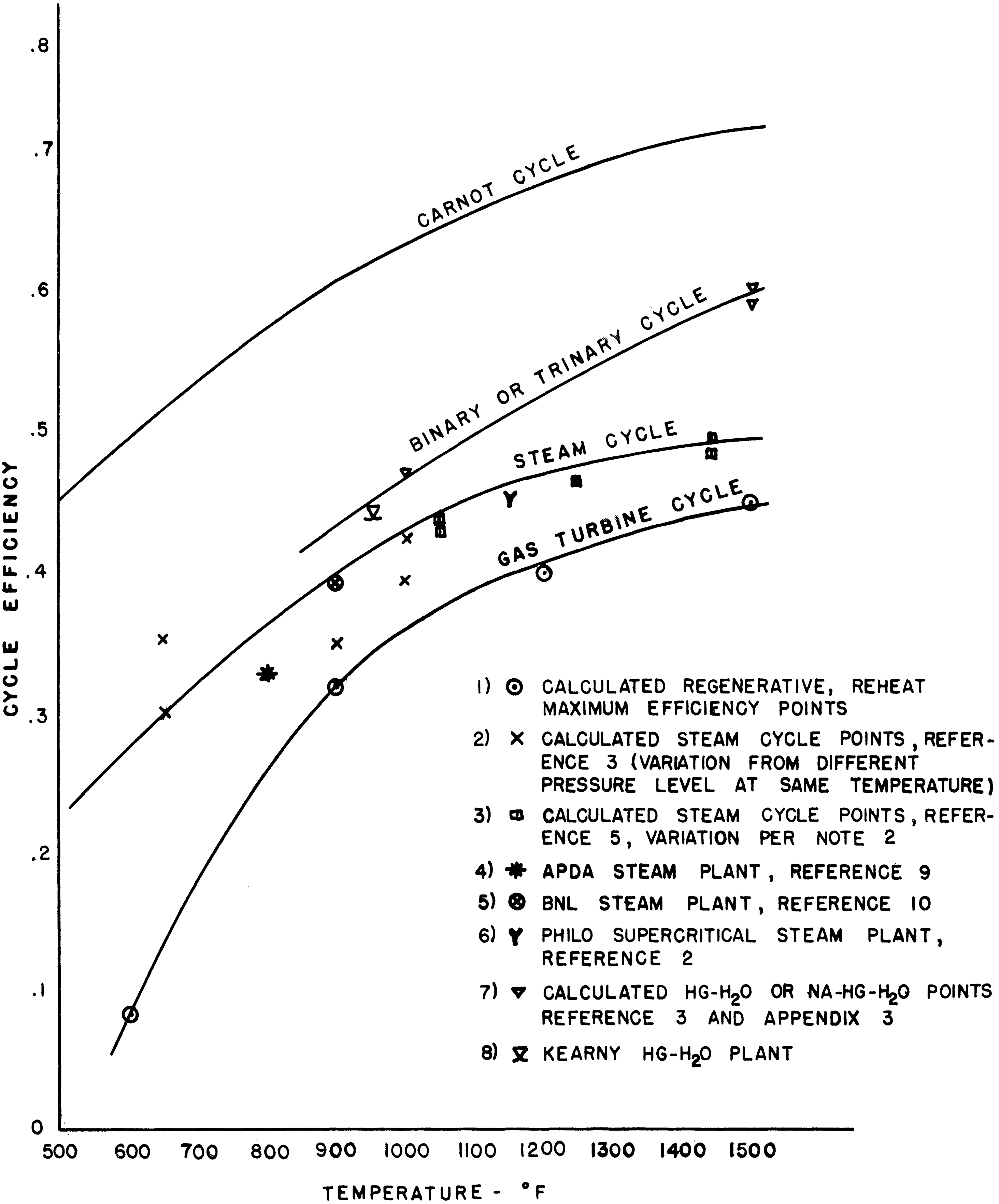


Figure 92. **MAXIMUM FEASIBLE EFFICIENCY VS TEMPERATURE VARIOUS HEAT ENGINE CYCLES COOLING WATER AT 70° F**

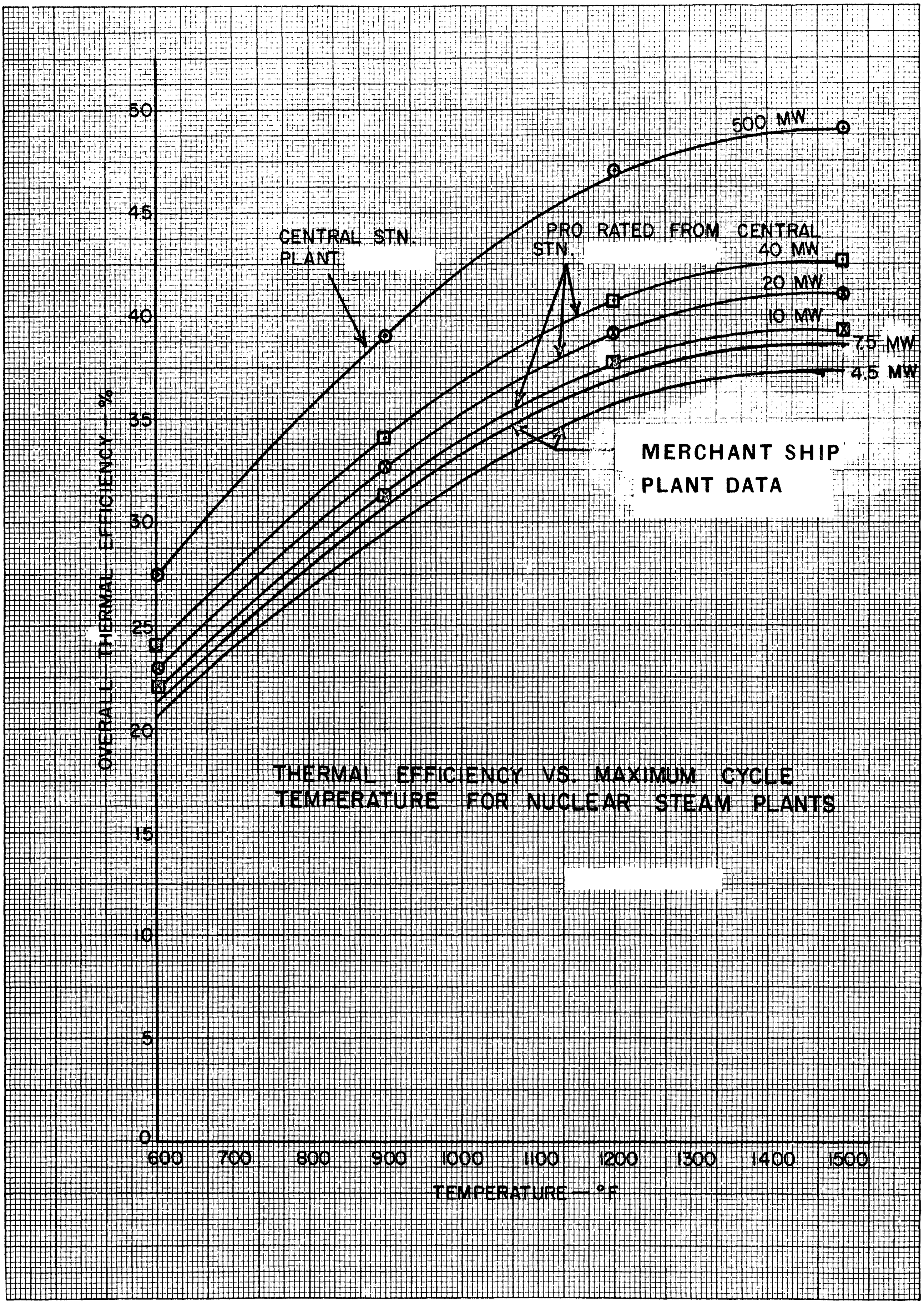

500 MW
CENTRAL STN.
PLANT
PRO RATED FROM CENTRAL
STN.
40 MW
20 MW
10 MW
7.5 MW
4.5 MW
MERCHANT SHIP
PLANT DATA
OVERALL THERMAL EFFICIENCY — %
50
45
40
35
30
25
20
15
10
5
0
600
700
800
900
1000
1100
1200
1300
1400
1500
TEMPERATURE — °F
THERMAL EFFICIENCY VS. MAXIMUM CYCLE
TEMPERATURE FOR NUCLEAR STEAM PLANTS

Figure 93.

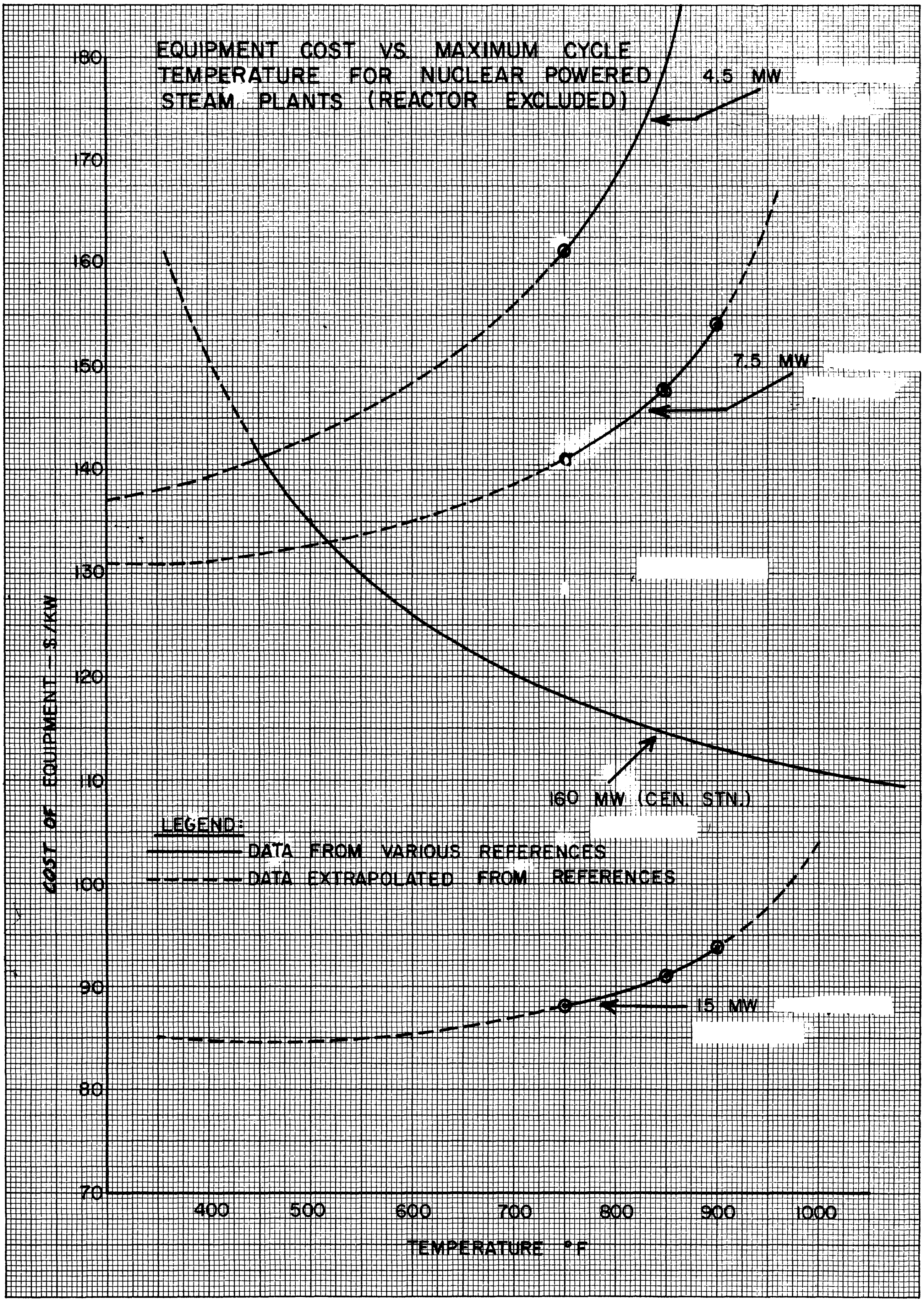

EQUIPMENT COST VS. MAXIMUM CYCLE TEMPERATURE FOR NUCLEAR POWERED STEAM PLANTS (REACTOR EXCLUDED)
4.5 MW
7.5 MW
160 MW (CEN. STN.)
15 MW
LEGEND:
DATA FROM VARIOUS REFERENCES
DATA EXTRAPOLATED FROM REFERENCES
COST OF EQUIPMENT — $/KW
TEMPERATURE °F
180
170
160
150
140
130
120
110
100
90
80
70
400
500
600
700
800
900
1000

Figure 94.

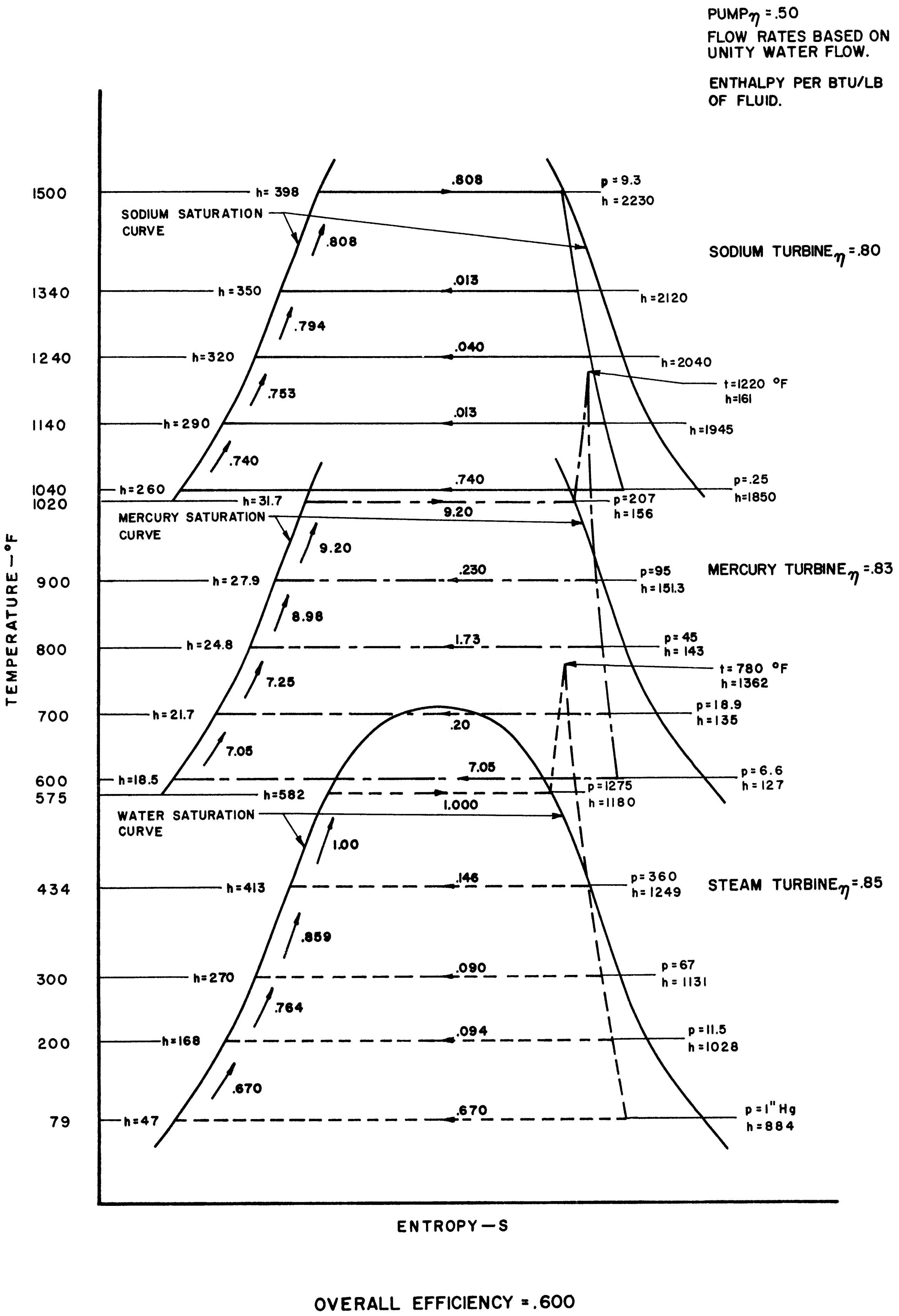
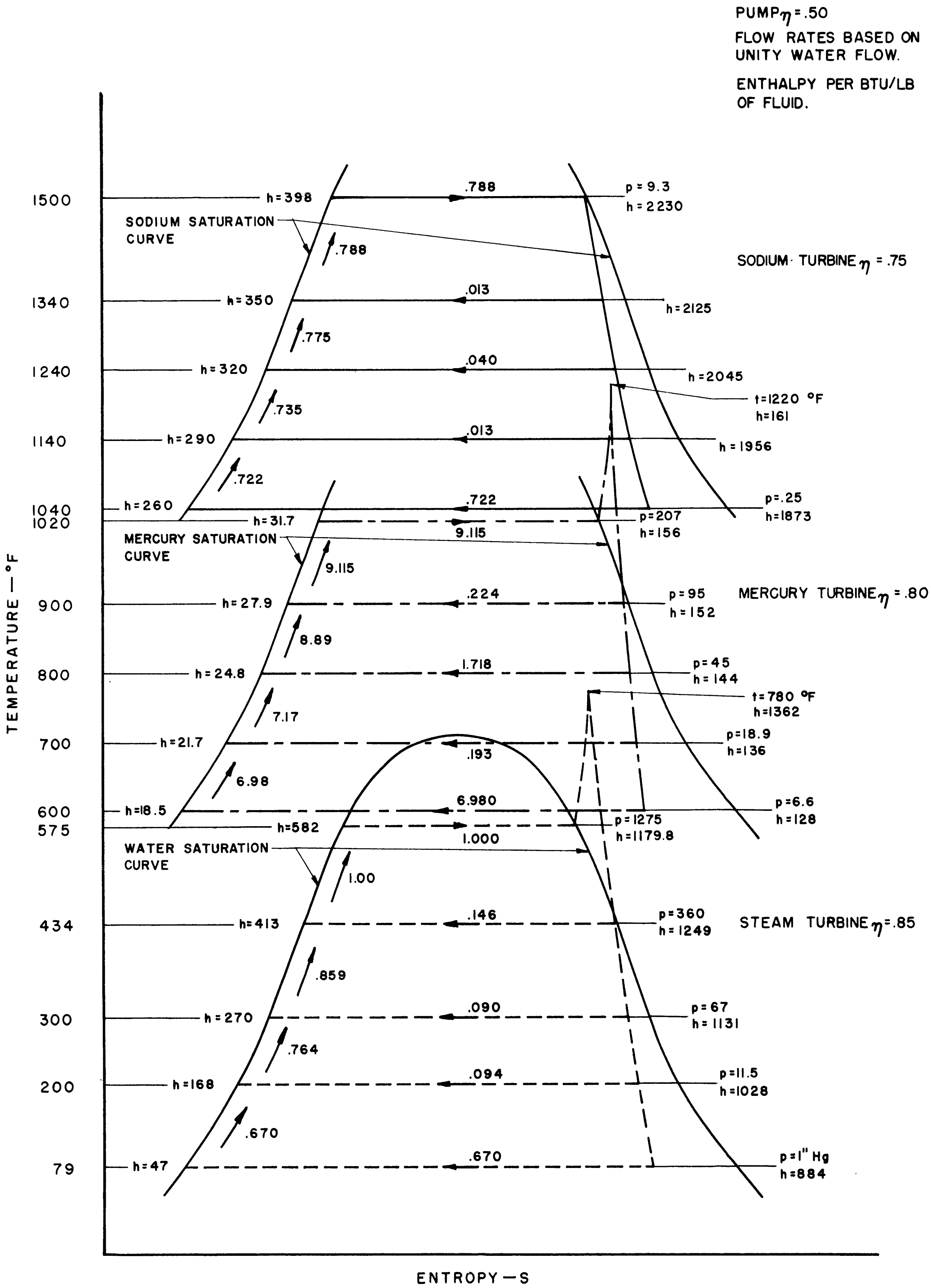
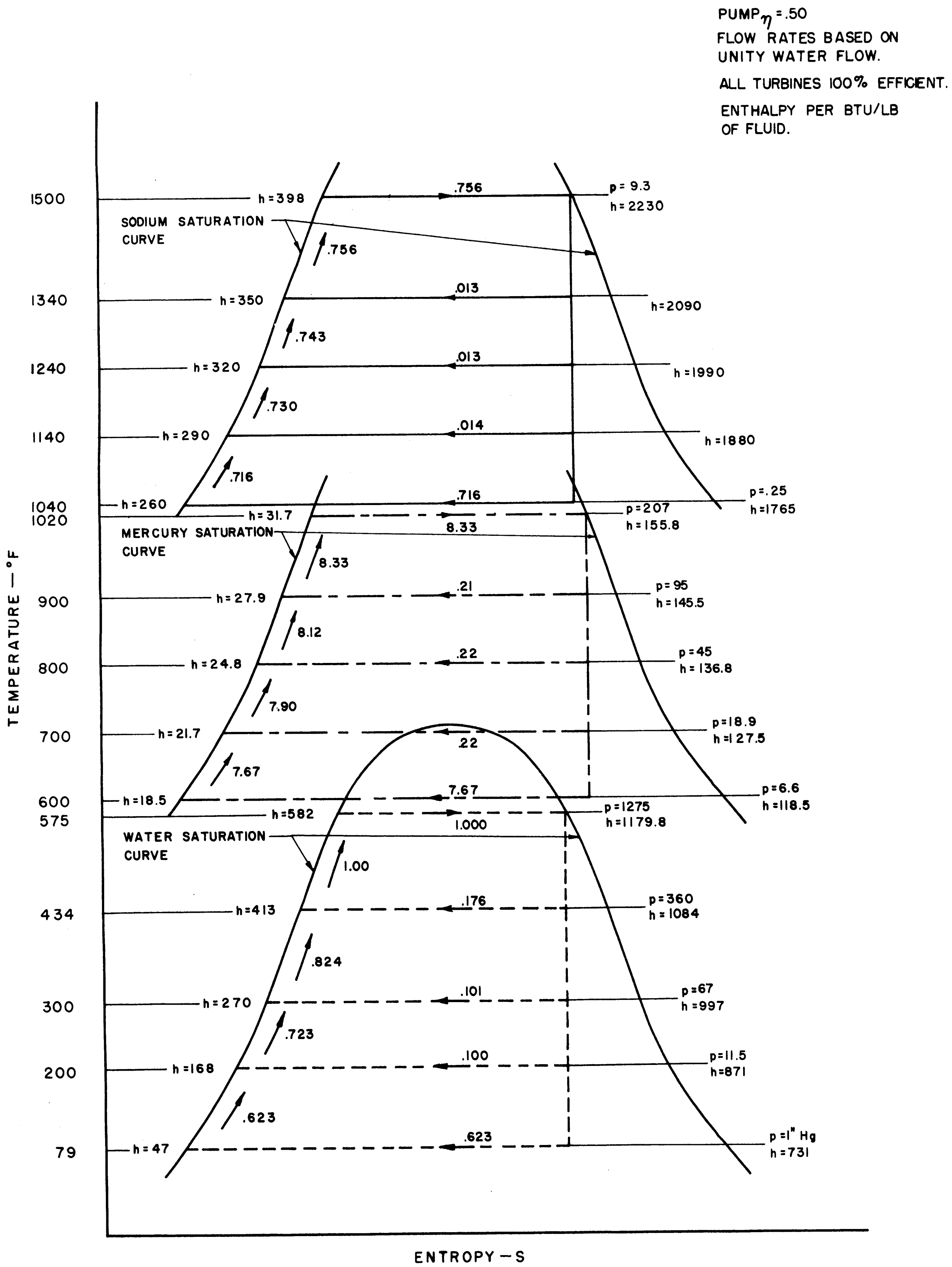


Figure 95. SODIUM-MERCURY-STEAM EXTRACTION TRINARY CYCLE I TEMPERATURE-ENTROPY DIAGRAM



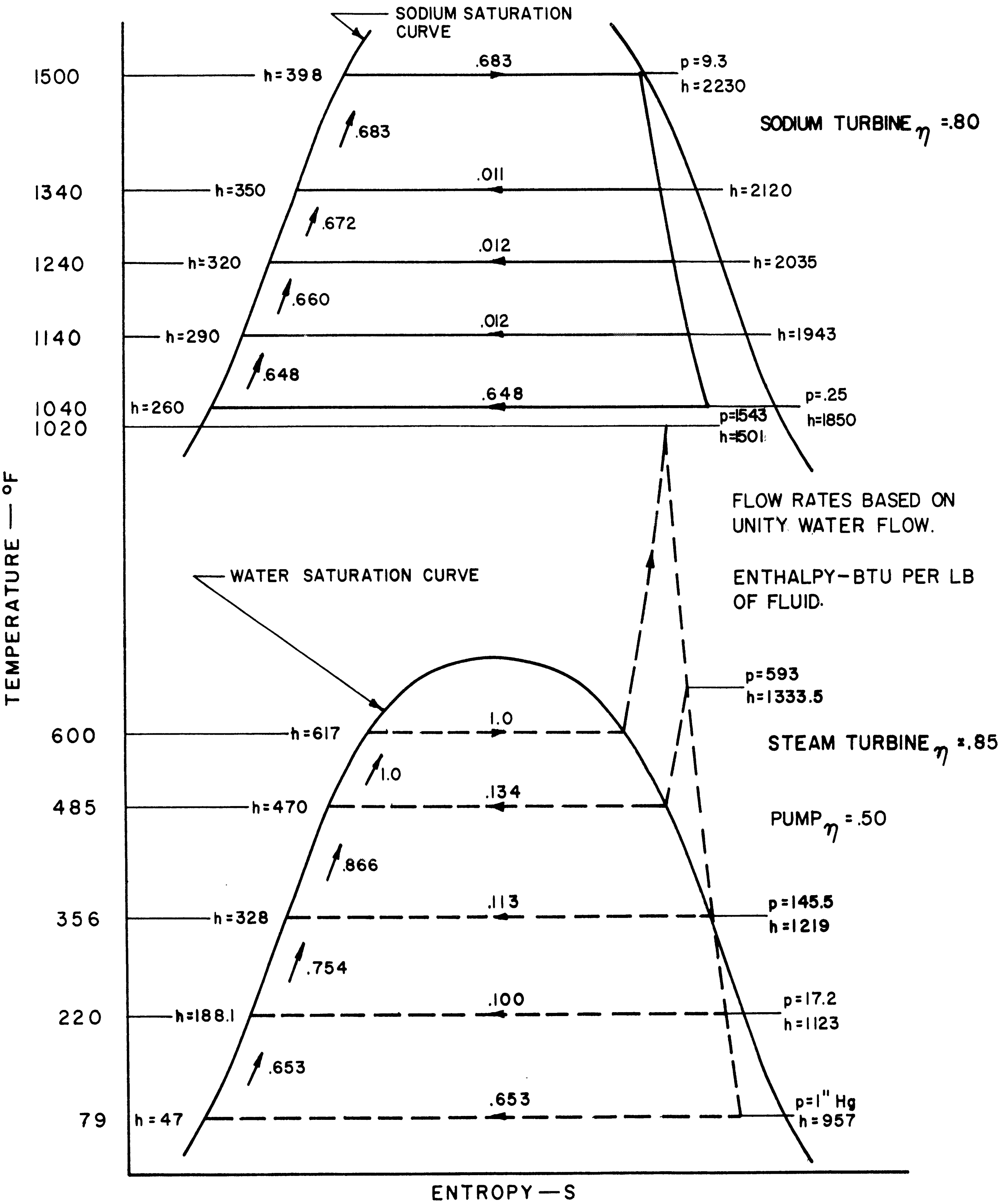
OVERALL EFFICIENCY = .588

Figure 96. SODIUM-MERCURY-STEAM EXTRACTION TRINARY CYCLE II
TEMPERATURE-ENTROPY DIAGRAM



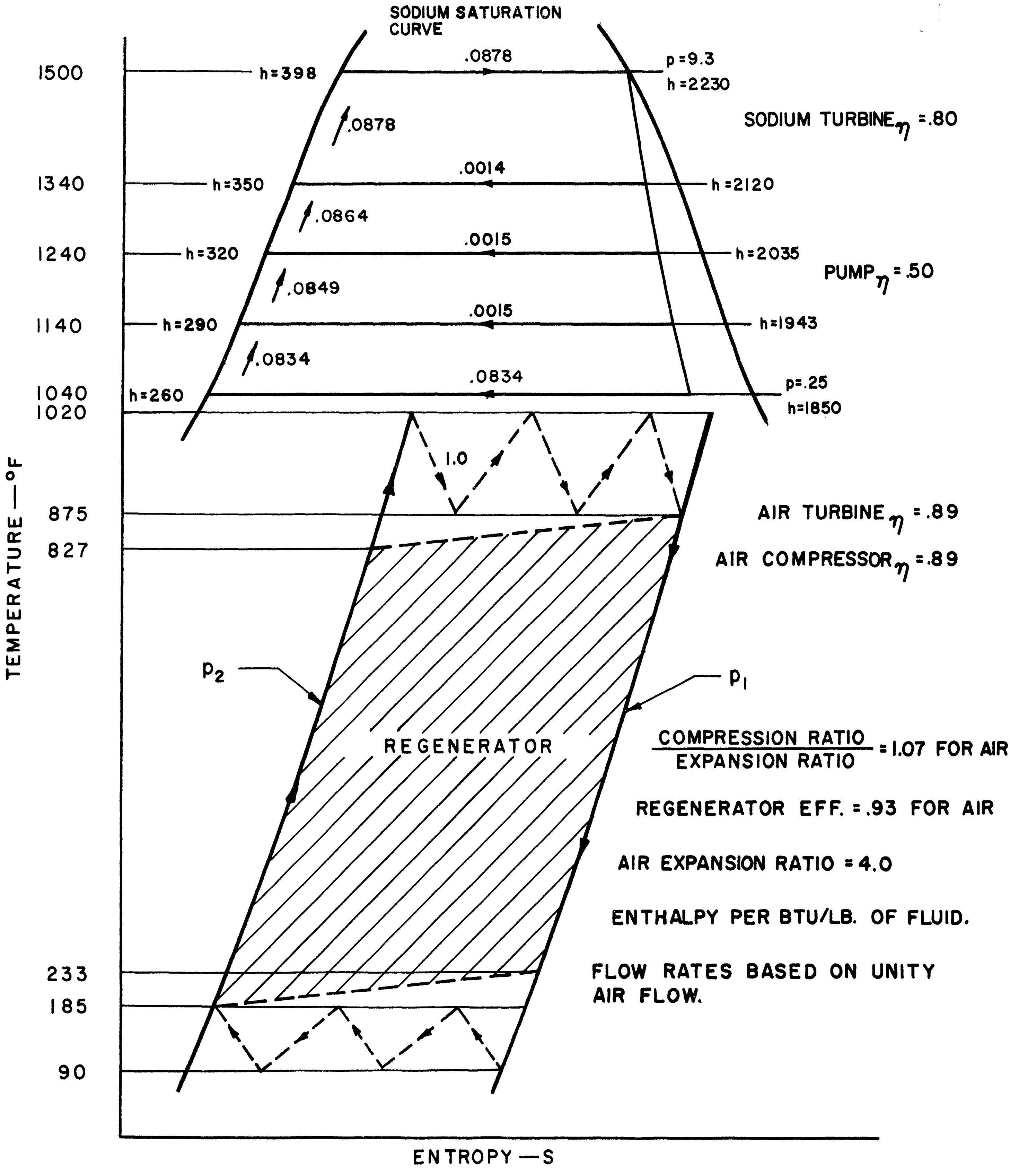
OVERALL EFFICIENCY = .671

Figure 97. SODIUM-MERCURY-STEAM IDEAL EXTRACTION TRINARY CYCLE TEMPERATURE—ENTROPY DIAGRAM



OVERALL EFFICIENCY = .516

Figure 98. SODIUM—STEAM BINARY EXTRACTION CYCLE TEMPERATURE-ENTROPY DIAGRAM



OVERALL EFFICIENCY = .486

Figure 99. SODIUM–AIR BINARY EXTRACTION–REHEAT CYCLE TEMPERATURE–ENTROPY DIAGRAM

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1. Hammitt, F. G. and Ohlgren, H. A., "Nuclear Powered Gas Turbines for Light Weight Power Plants." Paper No. 57-NESC-79, Second Nuclear Engineering and Science Conference, March 11-14, 1957, Philadelphia, Pa.

2. Ohlgren, H. A. and Hammitt, F. G., "Component Optimizations for Nuclear-Powered Closed-Cycle Gas Turbine Power Plants." ASME Paper No. 57-A-259. December, 1957.

3. Chernick, J., "Small Liquid Metal Fueled Reactor Systems," Nuclear Science and Engineering, Vol. I, (1956), pp. 135-155.

4. Sheehan, T. V., "Proposed Design for 500 MW LMFR Power Plant," Outline of talk given at BNL Industry Conference, Nov., 1955.

5. Cost estimates received from industrial sources.

6. Mabuce, E. M. and Hub, K. A., "A Study of Favorable Design Values for a Reactor Operated for Power Generation Only," AEP 99, Atomic Electric Project, St. Louis, Mo., Sept. 30, 1953.

7. Butcher, R. O., "Engine Room Study - Secondary Steam System for Nuclear Propelled Merchant Ships," MS-75 Aviation and Defense Industries, Sales Dept., General Electric Co., Schenectady, N.Y. 1955.