1035

21.3

CONVERSION EFFICIENCIES OF LARGE AND SMALL SOLAR THERMAL POWER SYSTEMS

FINAL REPORT

PREPARED FOR THE

SOLAR ENERGY RESEARCH INSTITUTE

1536 COLE BOULEVARD

GOLDEN, COLORADO 80401

CONTRACT NO. AP-9-8035-1

BY

STEARNS-ROGER SERVICES CO.

DENVER, COLORADO

JUNE 1979

40.4

INTRODUCTION

Stearns-Roger Service Co. has performed a study of a number of different thermodynamic electric power generating cycles under funding by the Solar Energy Research Institute (SERI) of Golden, Colorado.

The purpose of the study was to provide energy conversion system efficiency data as a function of maximum system temperature for each of the thermodynamic cycles as defined in the contract Statement of Work. The energy conversion system was assumed to be that portion of a solar power system containing the prime mover.

The data generated by this study will be used by SERI in evaluating the effects of various solar thermal transport and storage systems on the power generating system.

The approach used in the analysis and presented in this report was to:

1. Calculate the performance of the reference cycle as provided by SERI in the work statement.

2. Maintain a constant cycle heat input and rejection temperature for each non-reference condition as determined from the reference cycle.

3. Analyze each non-reference system assuming that it is operating at its design point. No "off-design" conditions were analyzed.

4. Assume component efficiencies based on current design practice derived from the literature or from Stearns-Roger power plant experience.

5. Develop a schematic of the cycle components together with the cycle state points for each of the reference cycles.

6. Calculate cycle performance for variable maximum cycle temperatures for each of the reference cycles. These data are presented as curves of cycle efficiency and ratios of cycle efficiency to reference cycle efficiency vs. maximum temperature.

7. Describe each cycle together with the assumptions used and cycle limitations.

A total of eleven cycles were evaluated for this study including steam Rankine (reheat and non-reheat), open and closed Brayton, organic Rankine and a total energy system using a steam Rankine cycle.

Existing or specially developed digital computer programs were used to perform the individual cycle calculations.

The report is divided into eleven Sections. Each Section contains a brief description of the cycle analyzed with the assumptions used, and the cycle schematic and efficiency curves.

No conclusions or comparisons between cycles are drawn from this study since the sole purpose of the study is to present cycle performance data to be used by SERI in a further study of solar power generating storage systems.

TABLE OF CONTENTS

TABLE OF CONTENTS (CONT'D)

PARA. PG. NO.

LIST OF FIGURES

FIG. NO.		PG. NO.

LIST OF FIGURES (CONT'D)

FIG. NO. PG. NO.

LIST OF FIGURES (CONT'D)

FIG. NO. **PG. NO.**

SECTION 1
ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE

1.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) steam Rankine, non-reheat cycle incorporating a five heater feedwater heating system. The cycle was duplicated from that of the Barstow 100 MWe Commercial Plant using steam from a solar receiver.

1.2 CYCLE DESCRIPTION

For operation on receiver steam the throttle conditions are 1465 psia (10.10×10^6 Pa) and 950°F (510°C) for the reference cycle. At these throttle conditions and constant final feedwater conditions of 2600 psia (17.93×10^6 Pa) and 425.5°F (218.6°C), a heat input to the cycle was determined. This heat input was held constant for the varying throttle temperatures and pressures studied. The throttle temperatures were varied from the reference cycle conditions down to a temperature 825°F (441°C) which yielded approximately 84 percent minimum quality steam leaving the last stage of the turbine, and up to 1100°F (593°C) (the upper limit for existing steam turbine technology). In actual operation, it is expected that the last stage quality will not be permitted to drop significantly below 88 percent. Generator output was allowed to vary with throttle conditions.

A cycle schematic showing all component efficiencies is presented in Figure 1—1, with a plot of cycle efficiency versus throttle temperature at different pressures shown in Figure 1—2. The turbine and pump efficiencies were obtained from data given in Reference 1. The cycle efficiencies are presented as gross (total energy output divided by total energy input to the cycle) and net (assuming 8 percent of total energy output including pumping power goes to auxiliary demand). A plot of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure is shown in Figure 1—3, and generator output (gross and net) versus throttle temperature, in Figure 1—4.

The turbine used in this model is a standard-frame General Electric utility, non-reheat steam turbine exhausting at an assumed 2.5 in. HgA (0.0984 mm HgA) to a tube and shell condenser. The turbine's five extractions are connected to three closed high-pressure heaters, an open deaerating heater and one closed low-pressure heater operating at various pressures. The heater operating characteristics were duplicated from the Barstow Commercial Solar Plant as documented in Reference 1, and held constant for the variable throttle conditions. The calculation procedure used to analyze this system is as described in the appendix and Reference 2.

For the case of steam supplied entirely from thermal storage (admission steam), the steam was admitted to the turbine downstream of normal throttle steam (receiver steam). Because of the point of admission and the low thermodynamic properties of the steam (365 psia (2.52×10^6 Pa) and 565°F (292°C)), the three top heaters are taken out of service for this operating mode. A flow of 3 to 5 percent of admission steam is required for cooling the high-pressure turbine stages bypassed by the admission steam. This steam does no work in the high-pressure stages, however, it does perform work as it recombines with the admission steam.

Again, the heat input to the reference cycle was determined and held constant for the non-reference cases. Also, the admission steam temperature was varied from the reference cycle admission temperature down to a temperature of 500°F (260°C) yielding a minimum 84 percent steam quality leaving the turbine, up to 925°F (496°C). Again, the last stage quality will actually be a minimum of approximately 88 percent. At the constant heat input, several throttle pressures were studied to illustrate the effects on the cycle if the admission point were varied up to the normal (receiver operation) pressure (1465 psia).

A reference cycle diagram for the admission steam is shown in Figure 1—5. Plots of the gross and net cycle efficiencies versus throttle temperature at different pressures are shown in Figure 1—2, assuming 6 percent auxiliary power usage. A plot of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure is shown in Figure 1—3, and a plot of generator output (gross and net) versus throttle temperature and pressure, in Figure 1—4.

REFERENCES

1. McDonnell Douglas Astronautics Company, "Central Receiver Solar Thermal Power System, Phase 1," (Volume 6, EPGS, MDC-G-6776), October 1977.

2. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

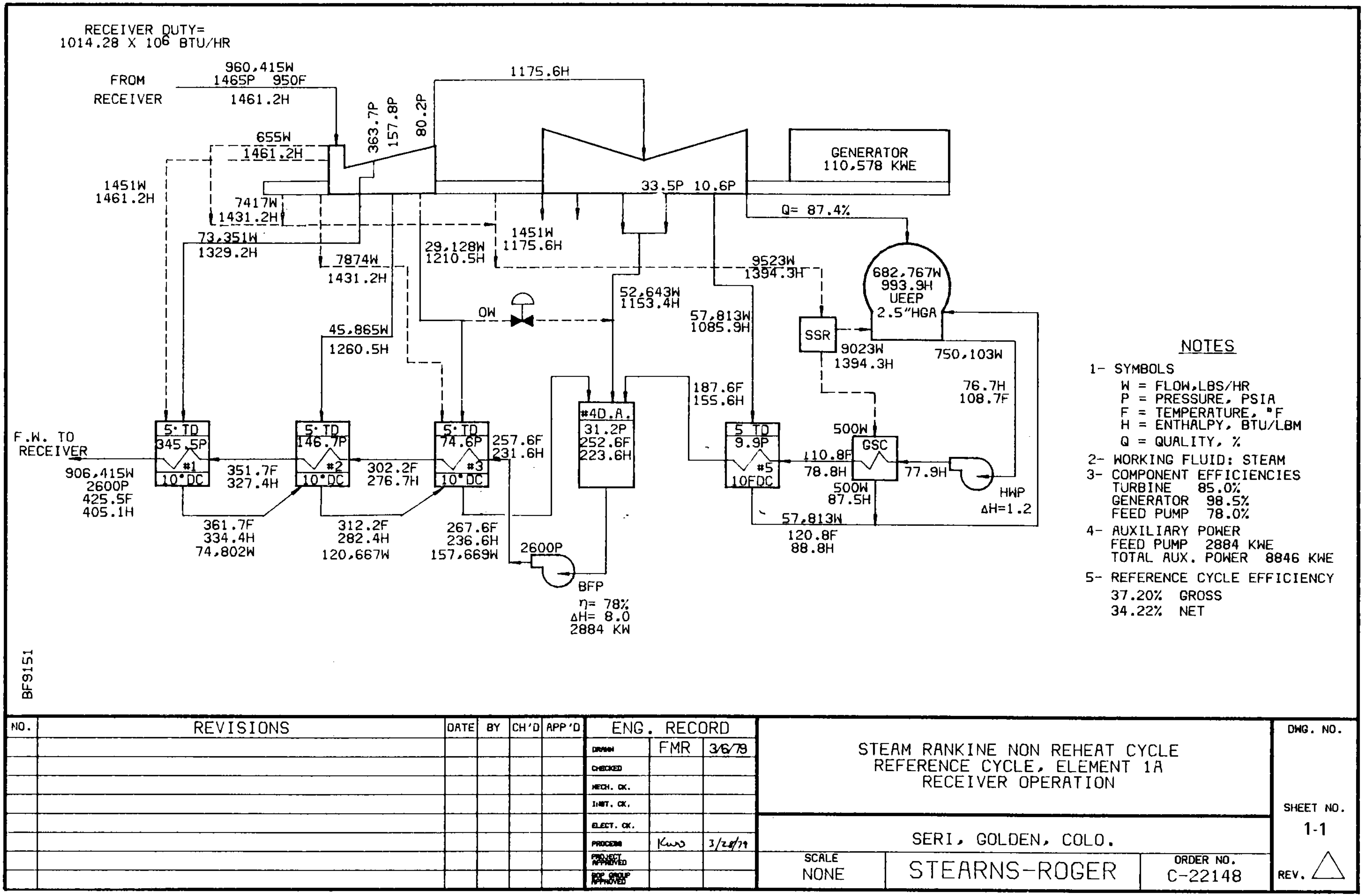


Figure 1—1. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, RECEIVER OPERATION

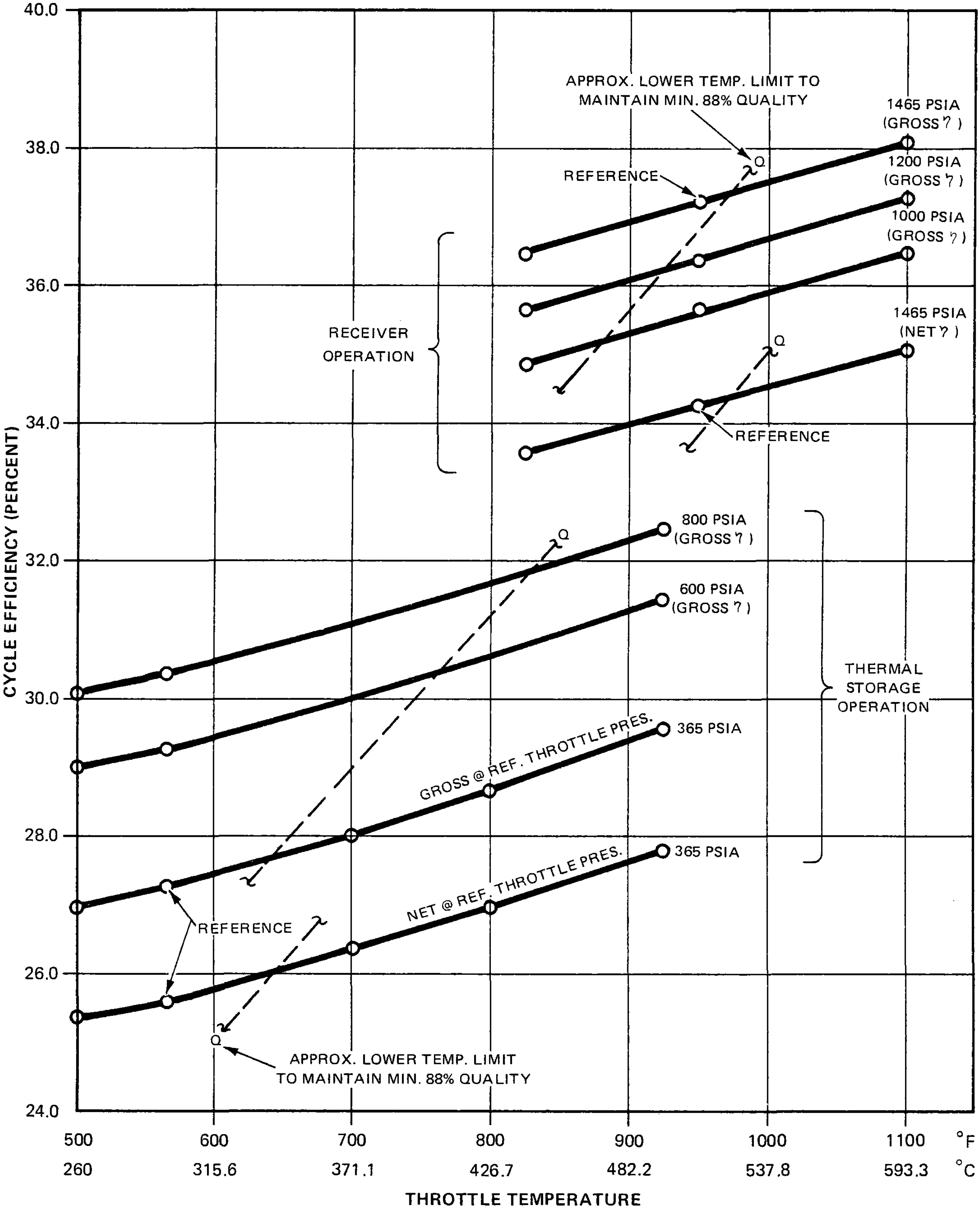


Figure 1—2. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

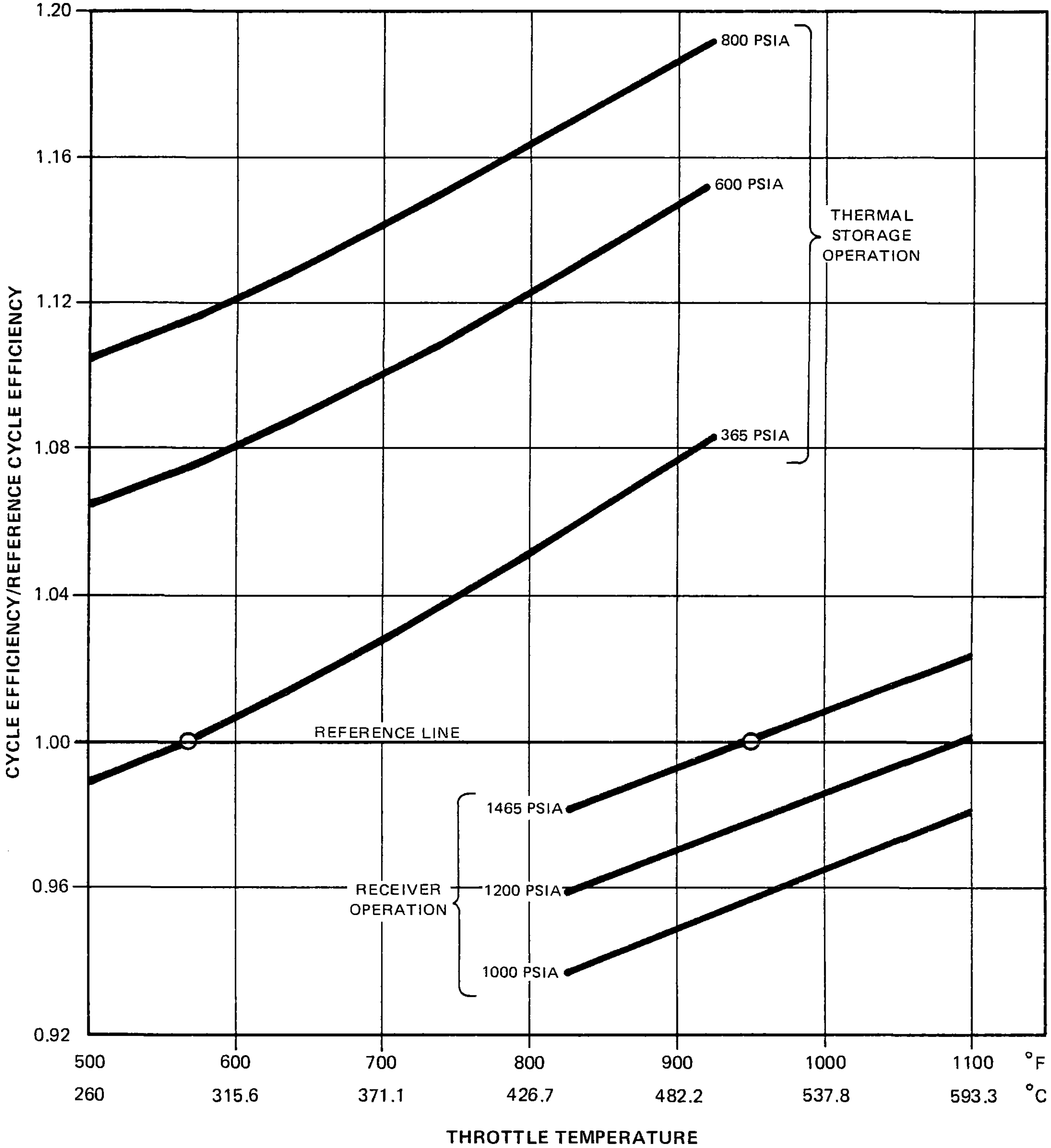


Figure 1–3. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, REFERENCE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

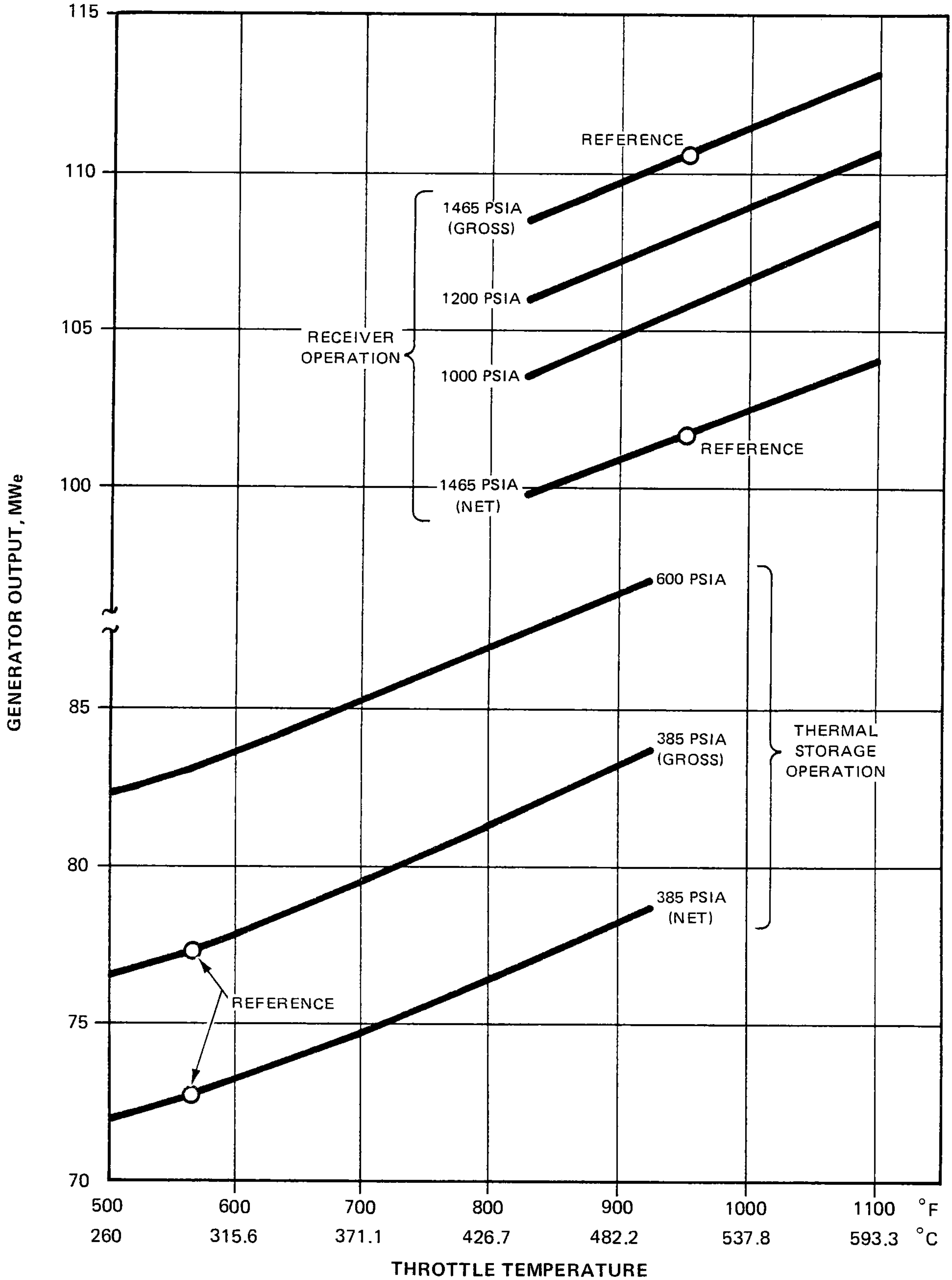


Figure 1–4. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

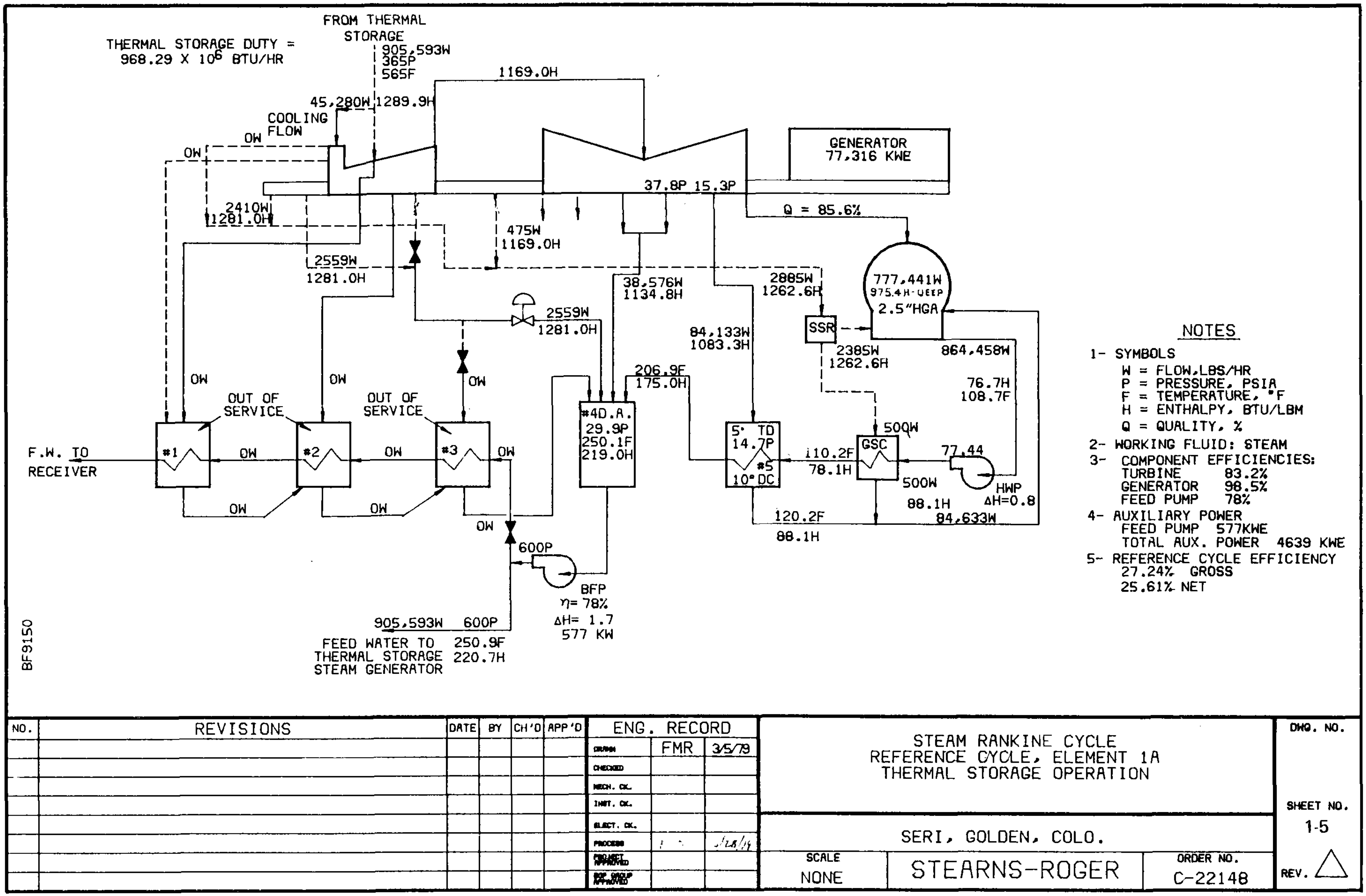


Figure 1—5. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, THERMAL STORAGE OPERATION

SECTION 2
ELEMENT 1B - REHEAT STEAM RANKINE CYCLE

2.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) reheat steam Rankine cycle employing six stages of regenerative feedwater heating. The cycle is based on the Advanced Central Receiver Power System using steam generated by a liquid metal solar receiver system. The turbine exhausts to a condenser at 2.5 inches HgA (8.46×10^3 Pa).

The cycle was analyzed using an in-house computer program (D135B), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 in Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

2.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 2400 psia (16.55×10^6 Pa) and 1000°F (537.8°C), with the reheat temperature of 1000°F (537.8°C). The final feedwater temperature selected is 480°F (248.9°C), which allows a reasonable pressure ratio across the high pressure turbine. A 10 percent pressure loss is assumed across the steam generator, and a pressure loss of 15 percent is assumed across the reheater.

For this study the total heat input to the cycle was held constant while the throttle conditions and reheat temperatures were varied. The throttle and reheat temperatures were varied over the range of 800°F to 1100°F (426.7°C to 593.3°C) and the throttle pressure was varied over the range of 1250 psia to 2400 psia (8.62×10^6 Pa to 16.55×10^6 Pa). The lower temperature limit was selected to limit the turbine exhaust steam quality to 88 percent and the upper temperature limit was selected as the limit of existing steam turbine technology. The throttle pressures selected are those normally used in the power industry. Representative high pressure and low pressure turbine efficiencies were calculated using the method specified in Reference 1. The representative turbine efficiencies were based on the throttle conditions of the reference cycle. The turbine efficiencies were held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbines will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to three high-pressure closed feedwater heaters, an open deaerating feedwater heater, and two low-pressure closed feedwater heaters. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Most turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. One leakage was included, the shaft leakage from the high pressure turbine to the low pressure turbine, as this is three percent of the throttle flow. The neglected leakages include shaft leakage from the exhaust of the HP turbine, the sealing flows to the LP turbine shaft seals, and packing leakage from the turbine stop and control valves. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 2—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 2—2 and Figure 2—3. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 2—4. A plot of generator output versus throttle temperature is shown in Figure 2—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, controls, plant lighting, plant HVAC, solar collector field usage, cooling tower fans, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

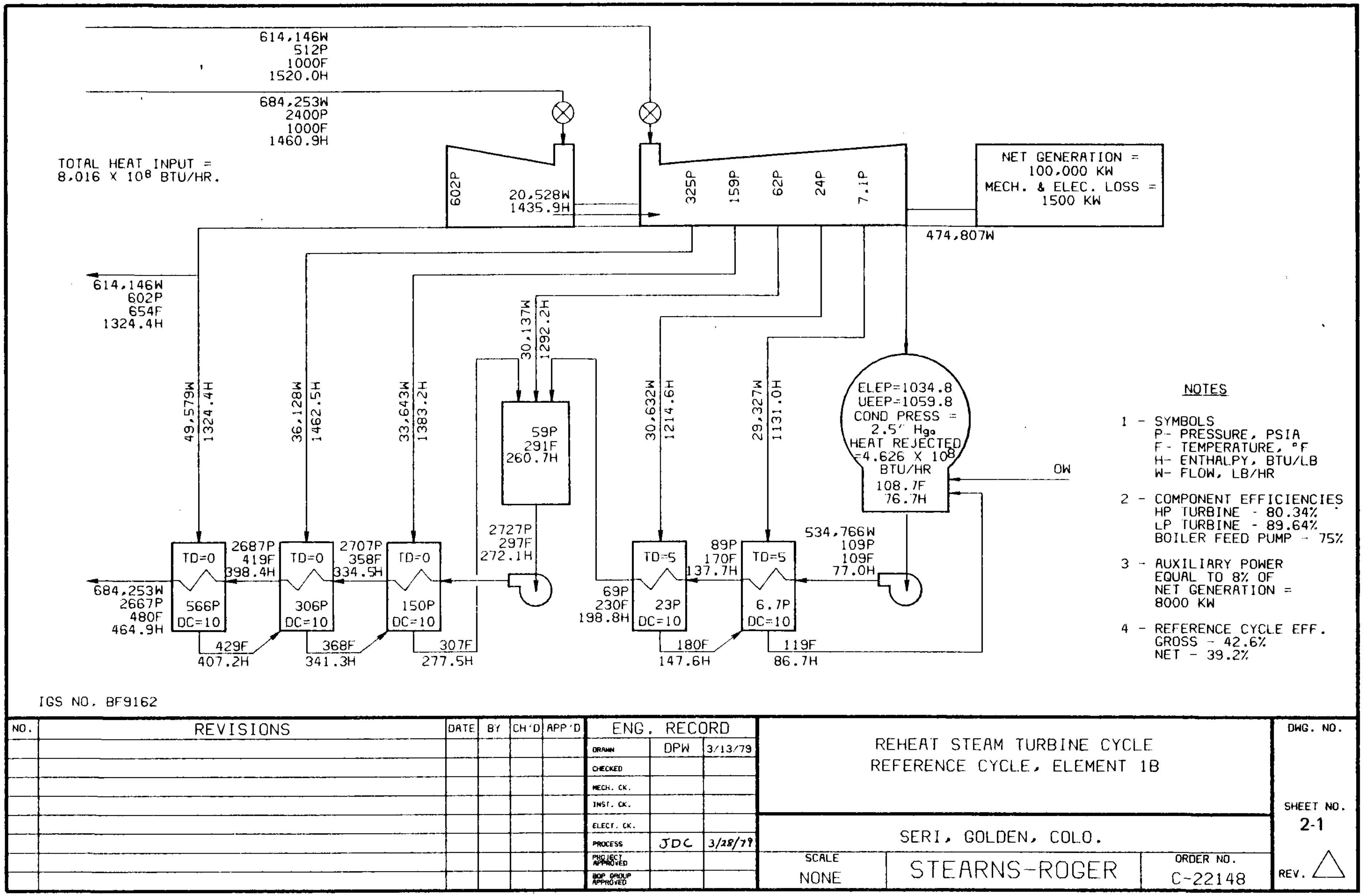


Figure 2—1. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, REFERENCE CYCLE

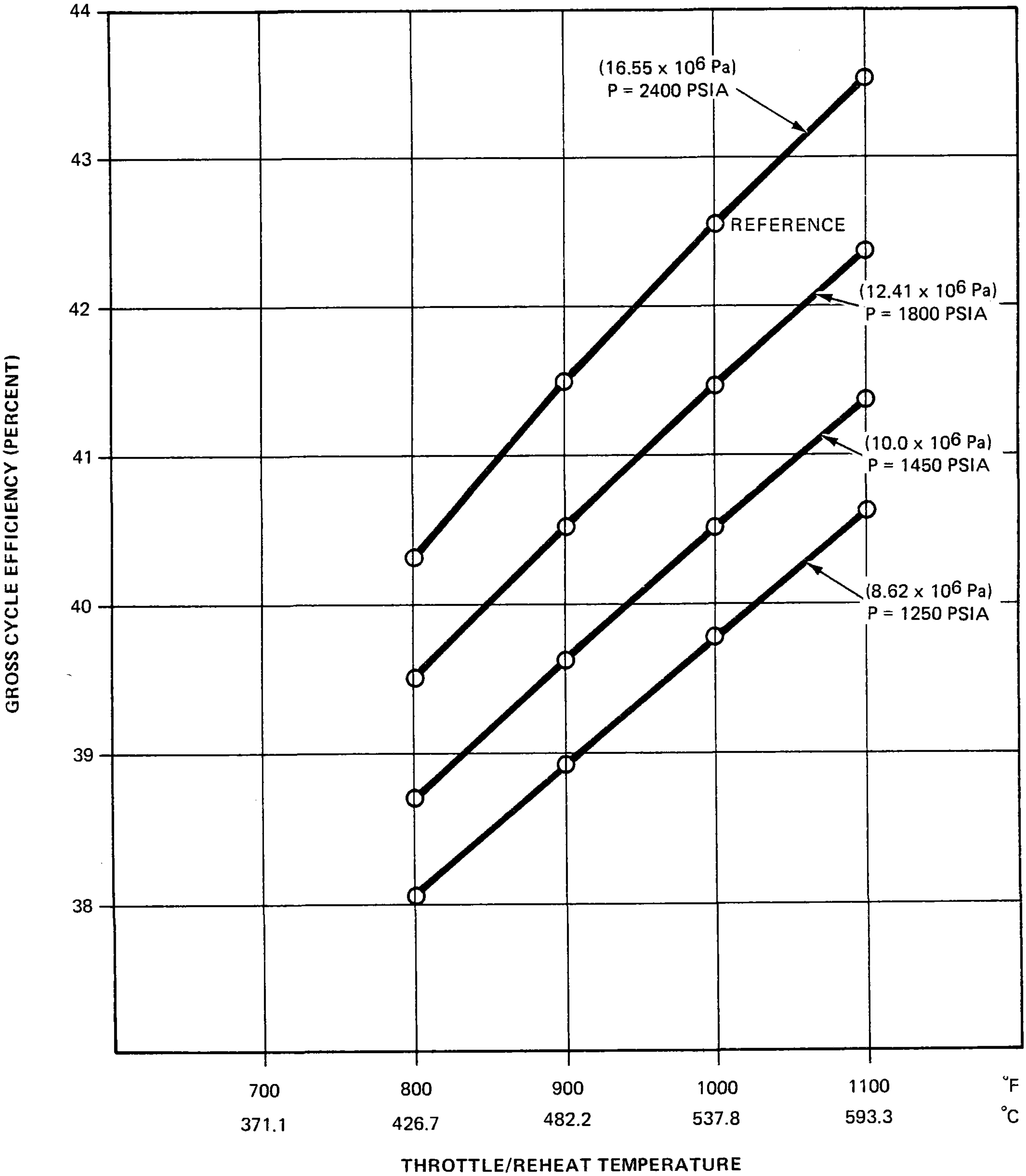


Figure 2–2. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

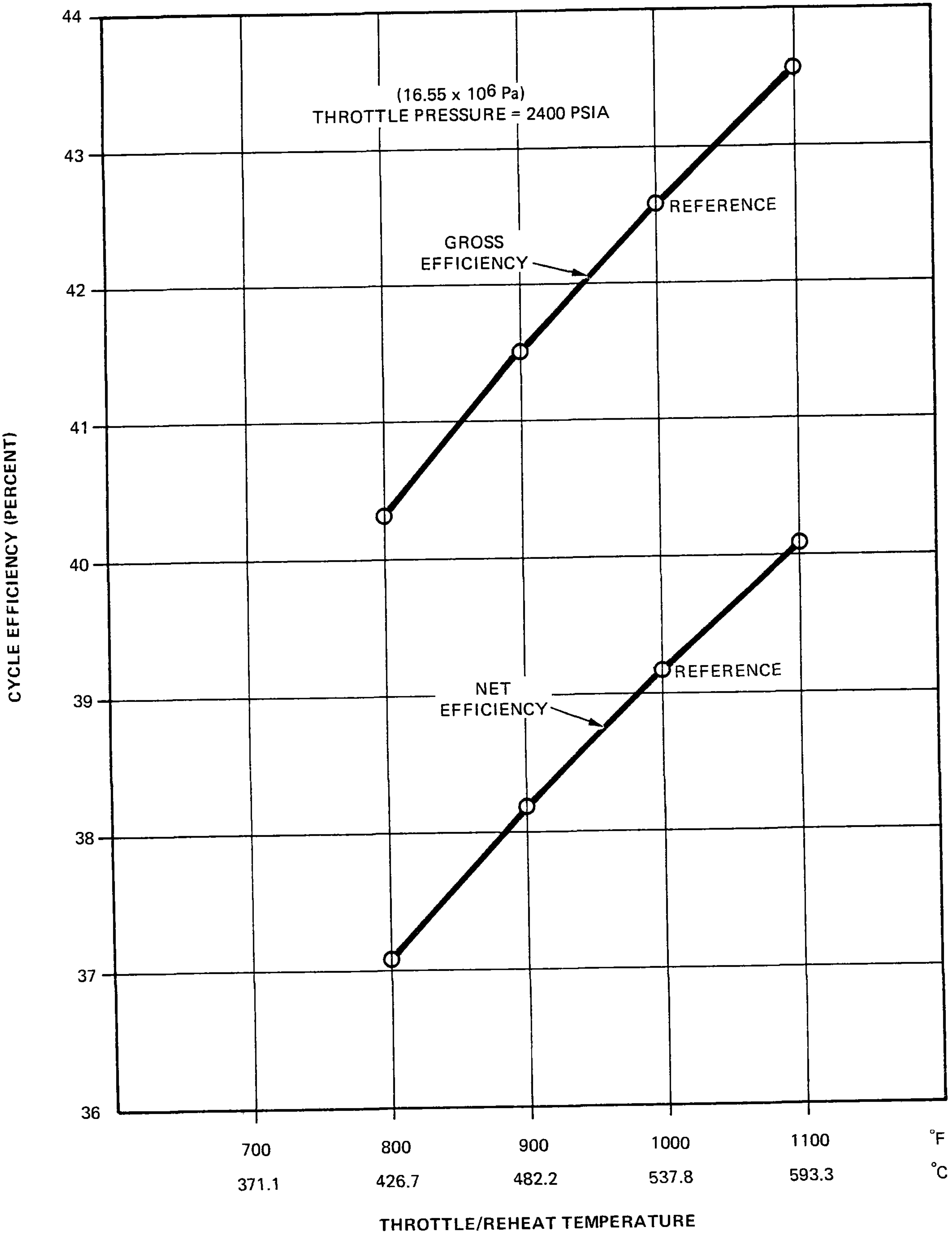


Figure 2–3. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

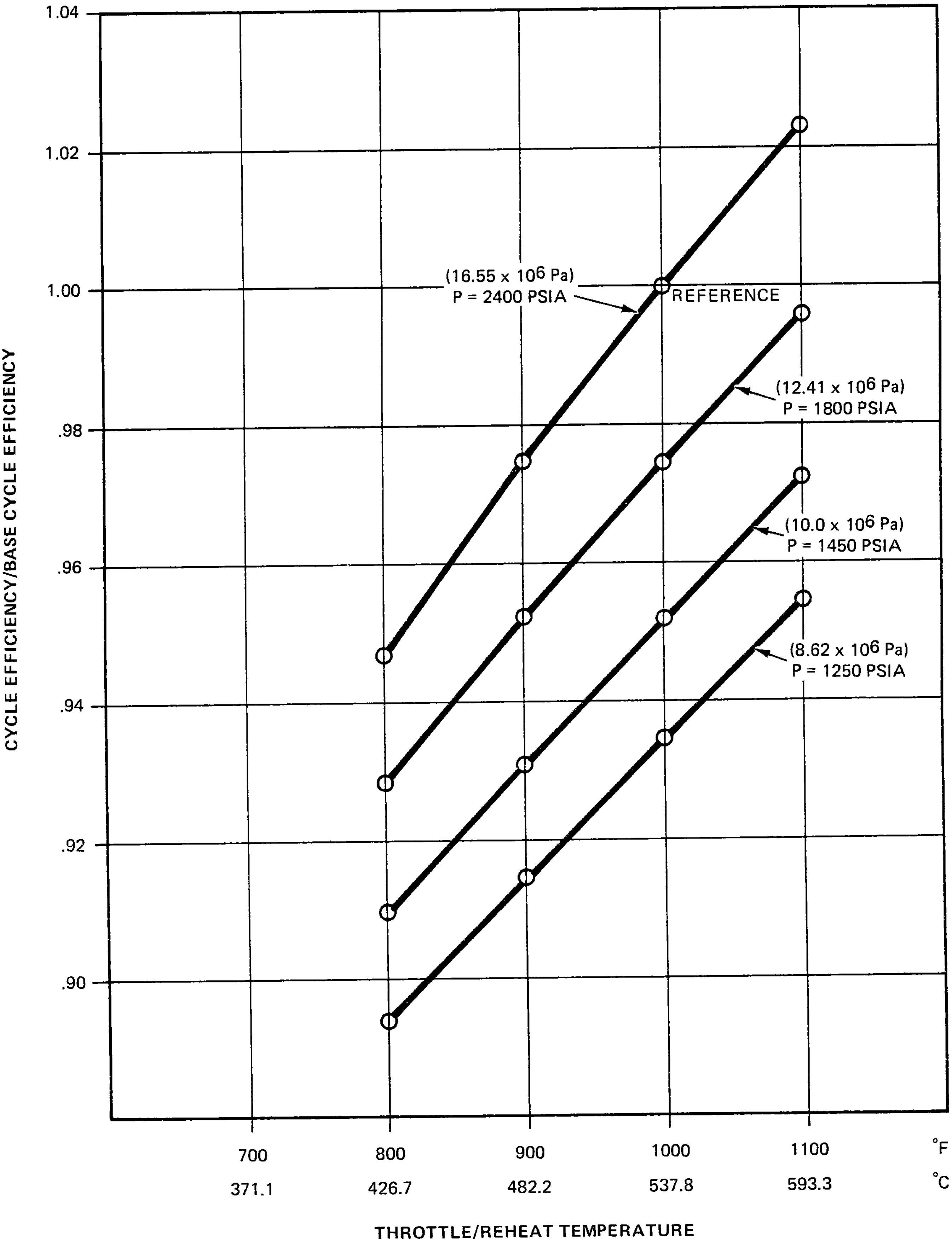


Figure 2–4. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

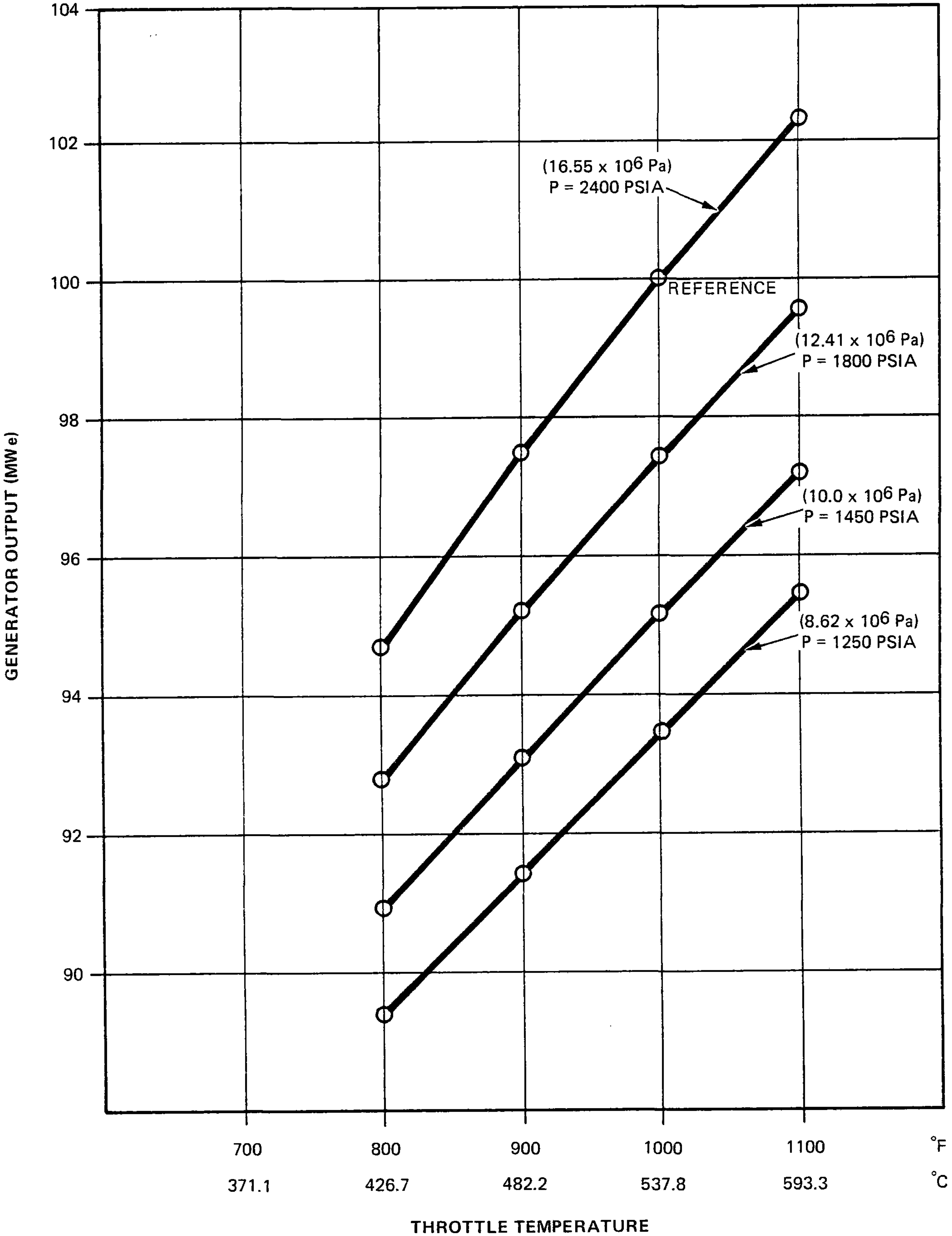


Figure 2–5. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 3
ELEMENT 2 - OPEN REGENERATIVE BRAYTON CYCLE

3.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) open regenerative, Brayton cycle. Heat input to the cycle is from a solar receiver/thermal storage system and from an oil fired air heater.

The cycle was analyzed using a computer program based on thermodynamic relationships contained in Reference 1 to perform the required mass and energy balance around the cycle. The program assumes a constant specific heat for the working fluid.

3.2 CYCLE DESCRIPTION

The reference cycle for this element is composed of a compressor with an efficiency of 80 percent and a pressure ratio of 4.75. Inlet air is compressed and discharged to the recuperator. Cooling air for the turbine is diverted from the compressor discharge to various parts of the turbine. Cooling air which is injected into the turbine in the flow path does useful work, while cooling air going to the turbine casing does no work. A general industry guide was used to determine how much of the total system flow was used as cooling air which did no work in the turbine. This guide states that cooling air is required at a turbine inlet temperature of 1700°F (926.7°C) and will be one percent of the compressor flow for every 100°F (55.6°C) of temperature increase.

The compressor flow is ducted to the recuperator where it is heated by the turbine exhaust air. For this cycle, the recuperator effectiveness is 90 percent and a pressure drop of two percent is assumed across the recuperator. The air temperature is further increased by the solar receiver/thermal storage system to a temperature of 1250°F (676.7°C). A pressure drop of three-and-one-half percent is assumed across the solar system. The air is finally heated to 2000°F (1093.3°C) by the oil-fired air heater. A pressure drop of one percent is assumed across the heater. The hot air is expanded through the turbine and produces shaft work which drives the compressor and the generator. The turbine efficiency used for this cycle is 90 percent. The air from the turbine exhaust passes through the recuperator where it is cooled by air from the compressor. A pressure drop of two percent is again assumed across the recuperator. The spent air is then exhausted to the atmosphere.

The turbine and compressor efficiencies, the recuperator effectiveness, and the amount of cooling air required are based on verbal information from several industry sources, Reference 2. These values are considered to be conservative.

For this study, the generator output was held constant for the non-reference cycles while the pressure ratio of the compressor was varied over the range of 2 to 10. The turbine inlet temperature was held at a constant 2000°F (1093.3°C). As the compressor pressure ratio is increased, the pressure ratio of the turbine also increases. The exhaust temperature of the compressor increases and the exhaust temperature of the turbine decreases with increasing pressure ratios. The overall effect is to reduce the duty of the recuperator. This relationship is shown in Figure 3—1.

For any specific compressor ratio the relationship between the temperature of the air out of the solar receiver/thermal storage system and amount of oil burned in the air heater can be calculated. In defining this relationship it is assumed that the generator output and the turbine inlet temperature are constant. This relationship is shown in Figure 3—2.

The cycle schematic showing component efficiencies and reference cycle flow data is shown in Figure 3—3. A plot of cycle efficiency versus compressor pressure ratio is shown in Figure 3—4. A normalized representation of cycle efficiency with respect to base cycle efficiency versus compressor pressure ratio is shown in Figure 3—5.

Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary requirement (which is assumed to be 8 percent of the generator output).

REFERENCES

1. "Gas Dynamics", A. D. Lewis, 1964.

2. Telephone conversations with manufacturers concerning turbine efficiencies, compressor efficiencies, recuperator effectiveness, and cooling flows.

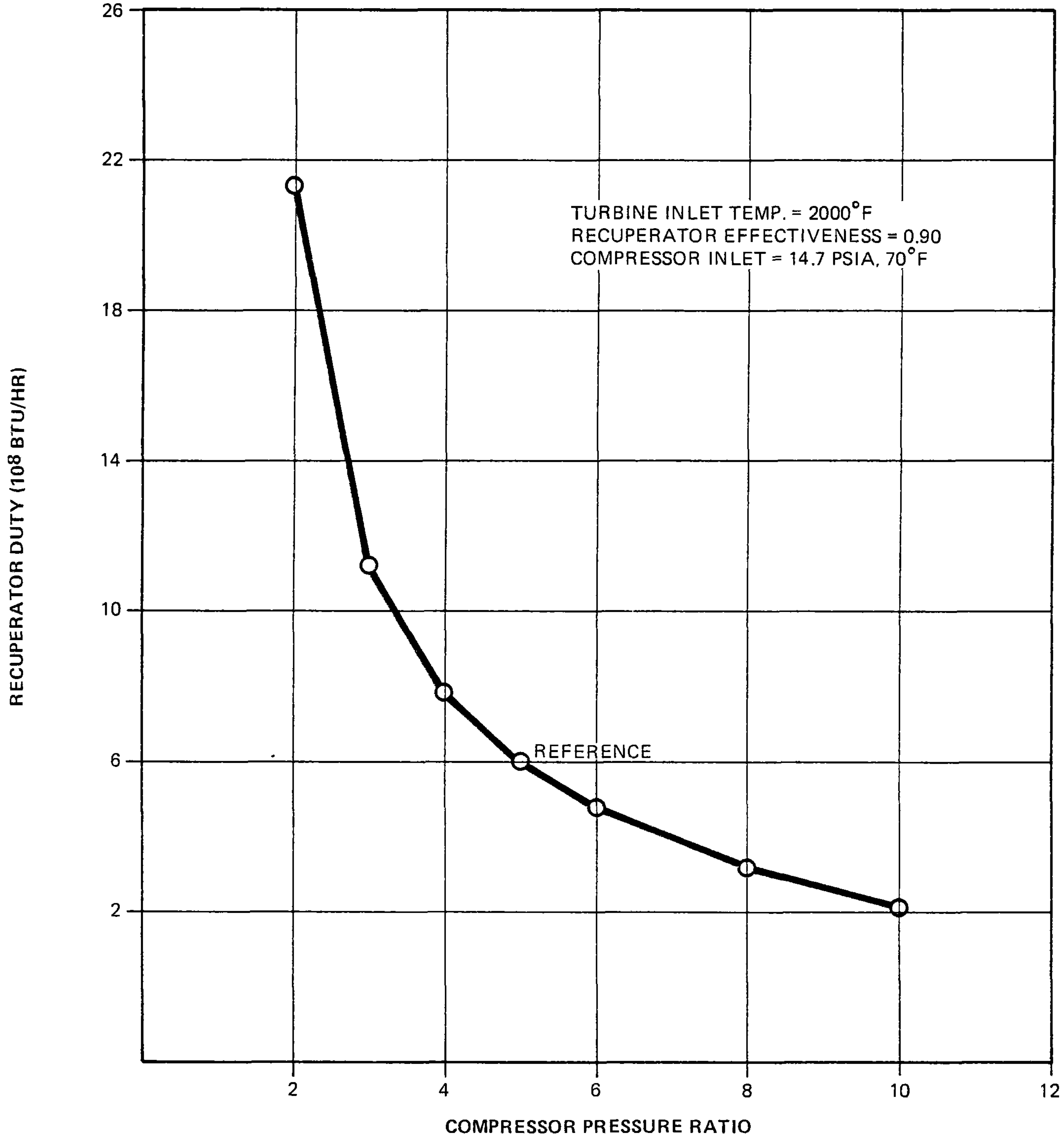


Figure 3–1. ELEMENT 2 - OPEN REGENERATIVE CYCLE, RECUPERATOR DUTY VS. COMPRESSOR PRESSURE RATIO

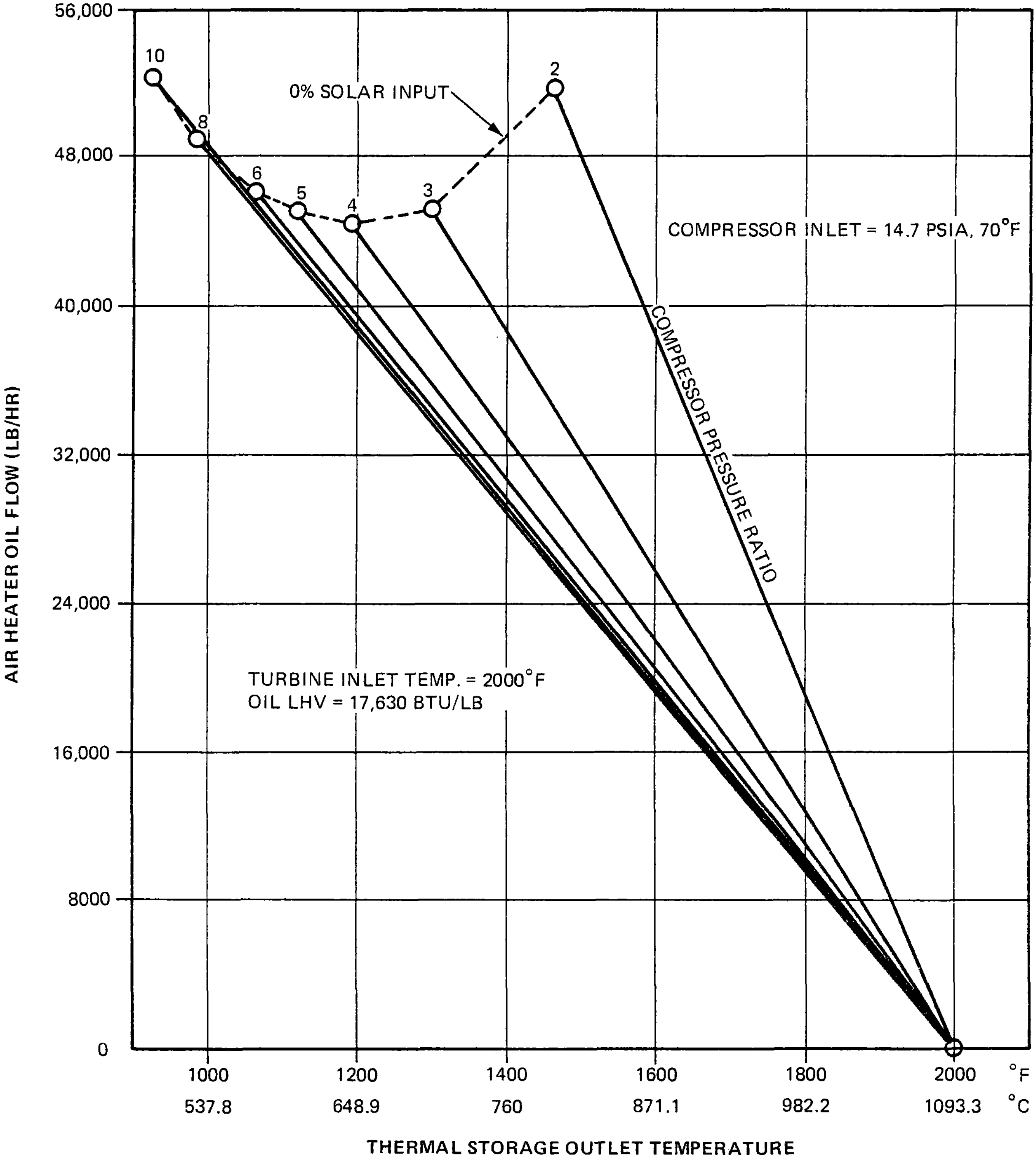


Figure 3–2. ELEMENT 2 - OPEN REGENERATIVE CYCLE,
AIR HEATER OIL FLOW VS. THERMAL STORAGE OUTLET TEMPERATURE

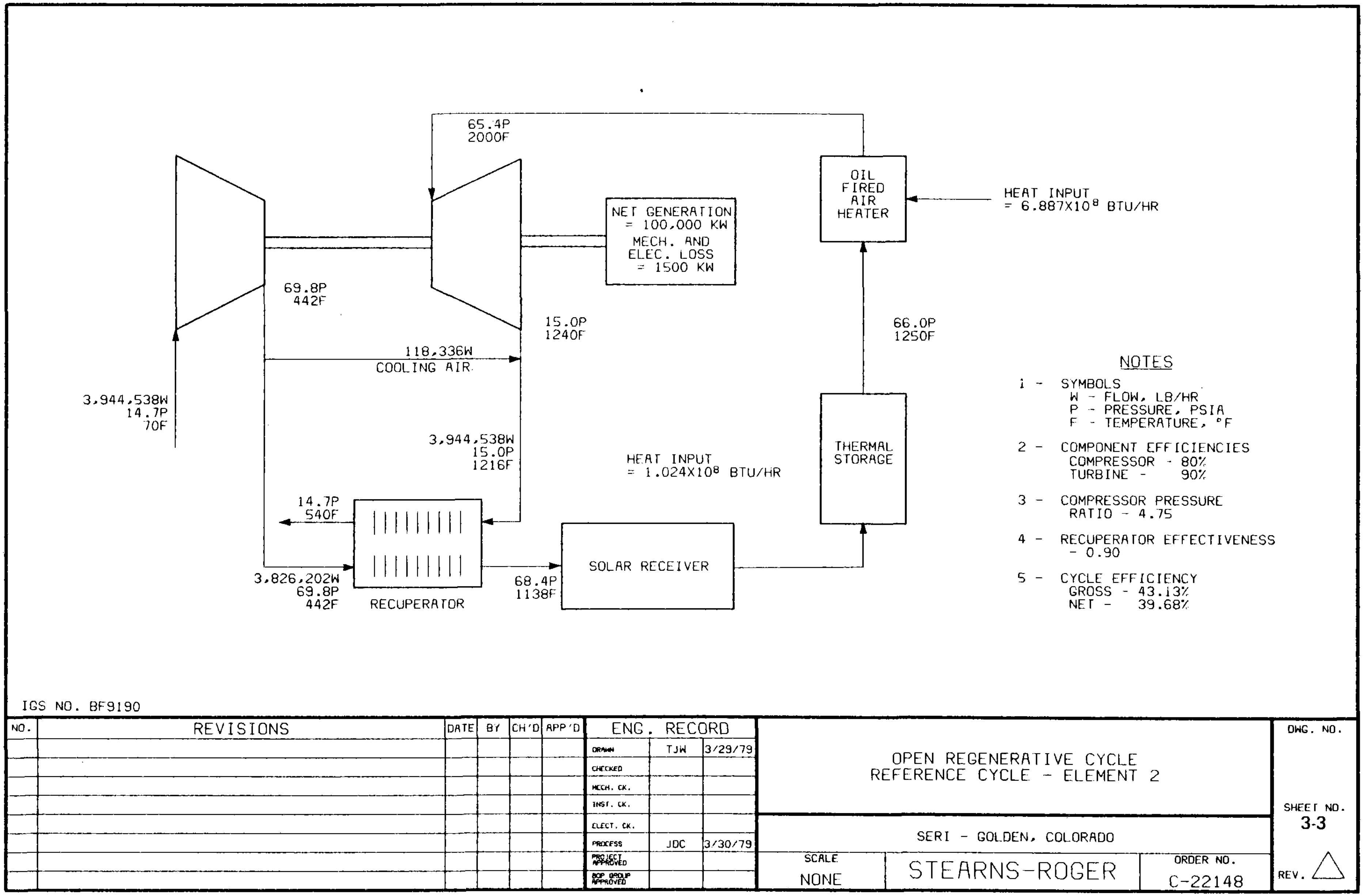


Figure 3—3. ELEMENT 2 - OPEN REGENERATIVE CYCLE

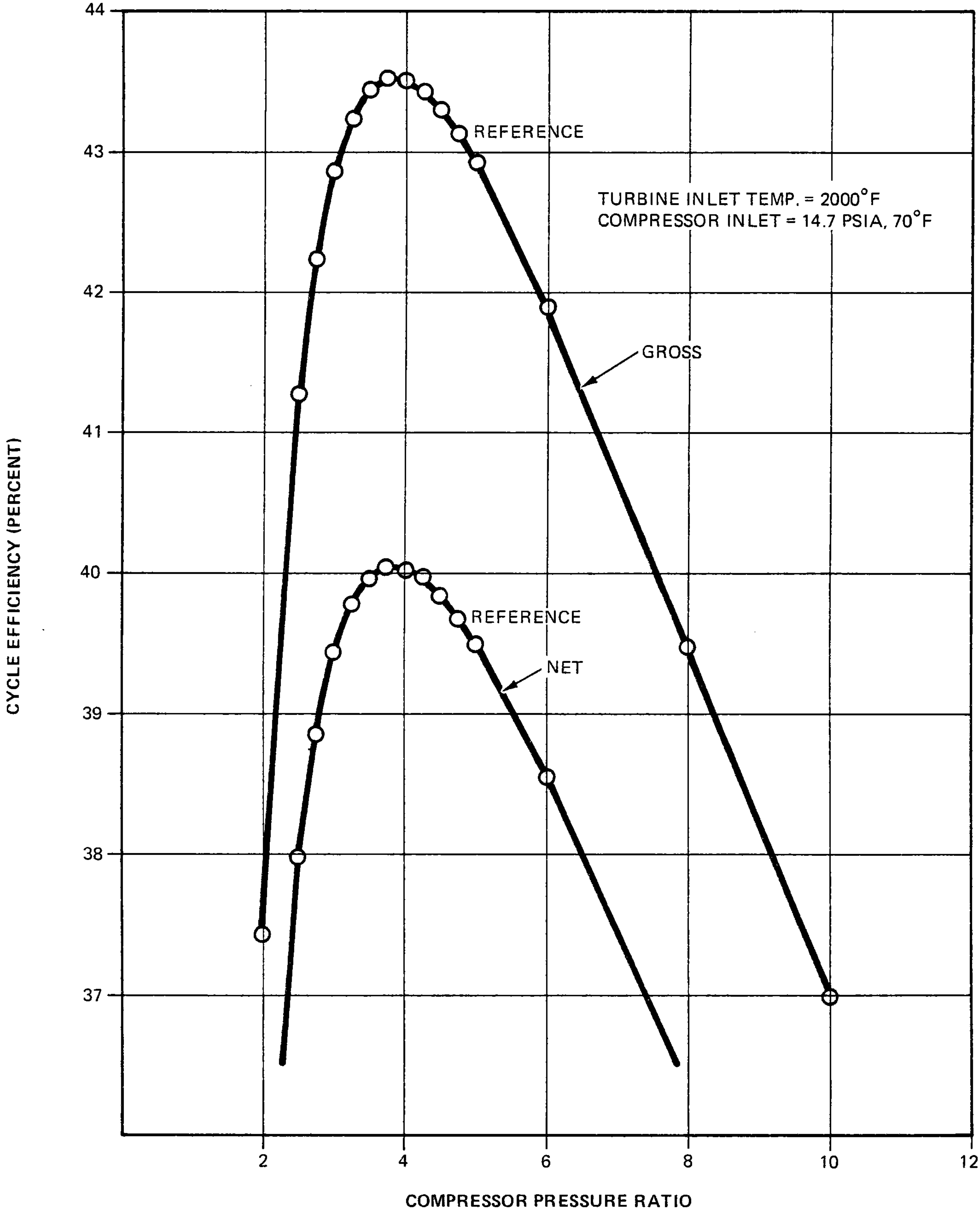


Figure 3—4. ELEMENT 2 - OPEN REGENERATIVE CYCLE, CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO

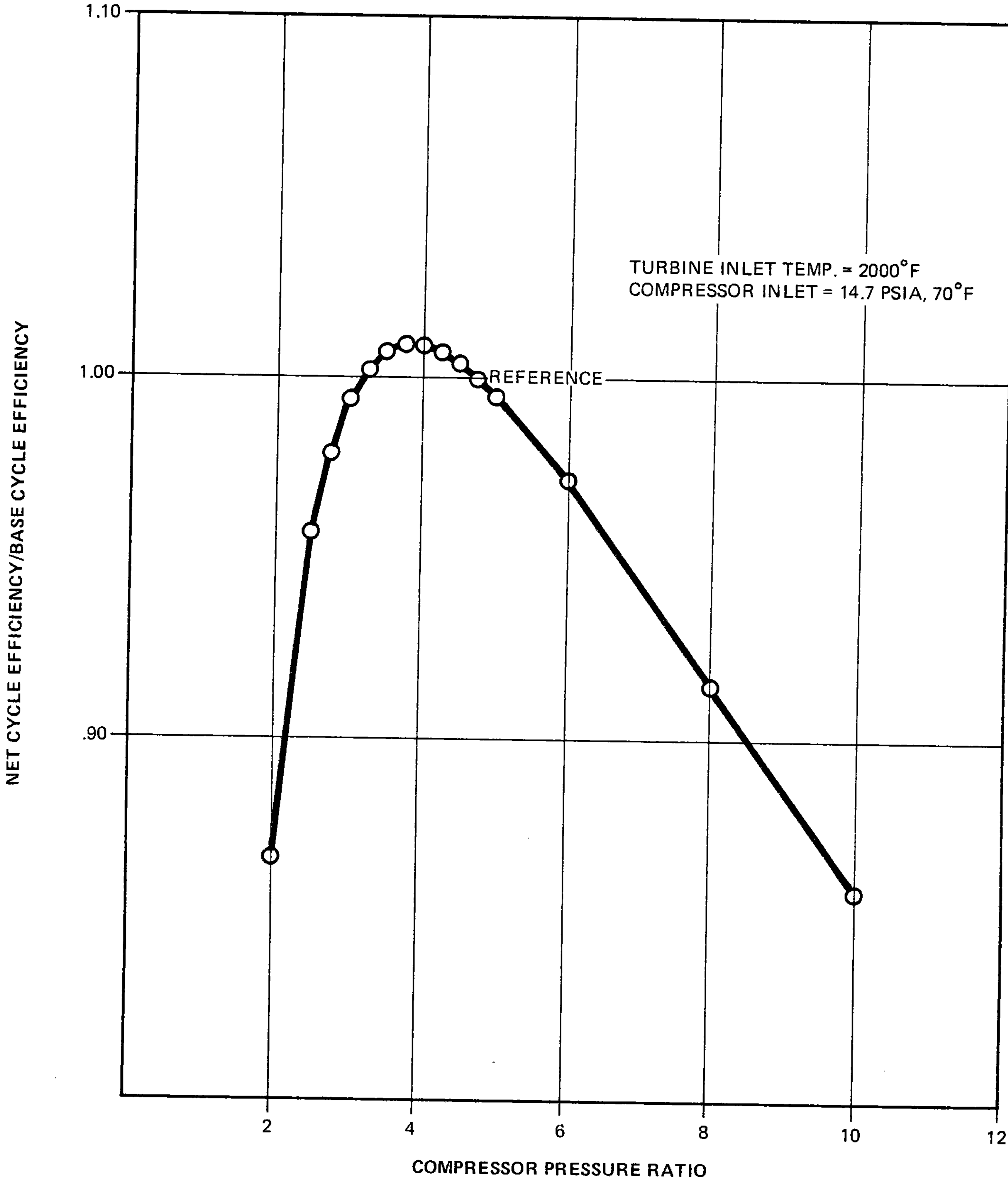


Figure 3–5. ELEMENT 2 - OPEN REGENERATIVE CYCLE, RELATIVE CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO

SECTION 4
ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE

4.1 INTRODUCTION

This cycle is similar to Element 1A, but is modeled after the Barstow 10 MWe Pilot Plant, Reference 1. A four-heater feedwater system (including two closed high-pressure heaters, one open deaerating heater and one closed low-pressure heater) is utilized. The gland steam condenser is eliminated from the feedwater system for this cycle configuration.

4.2 CYCLE DESCRIPTION

For throttle steam supplied from the receiver, the steam conditions are 1465 psia (10.10×10^6 Pa) and 950°F (510°C). The turbine exhaust conditions and basis for throttle temperature and pressure variations are identical to Element 1A for receiver operation. The throttle temperatures were varied from 800°F (427°C) up to 1100°F (593°C), while holding total heat input to the cycle constant. These calculations were based on the method described in the Appendix and Reference 2.

For the case of operation from thermal storage, the admission steam conditions are 384.7 psia (2.65×10^6 Pa) and 525°F (274°C), with 5 percent of admission steam used for cooling the high-pressure turbine. The two top heaters are out of service for operation from thermal storage. Throttle temperatures were varied from 500°F (260°C) up to 750°F (399°C), using last stage quality as an indicator of the lower temperature limit and holding heat input constant. Throttle pressures were varied to illustrate the effects on the cycle of changing the admission point up to the normal admission (receiver operation) pressure.

Reference cycle diagrams are presented as Figure 4—1 (receiver operation) and Figure 4—2 (thermal storage operation), with plots of gross and net cycle efficiencies versus throttle temperature and pressure given in Figure 4—3, for receiver and thermal storage operation (assuming 8 percent and 6 percent for total auxiliaries, respectively, including pumping power). Plots of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure, and generator output (gross and net) versus throttle temperature and pressure are shown in Figures 4—4 and 4—5.

REFERENCES

1. McDonnell Douglas Astronautics Company, "Central Receiver Solar Thermal Power System, Phase 1," (Volume 6, EPGS, MDC-G-6776), October 1977.

2. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

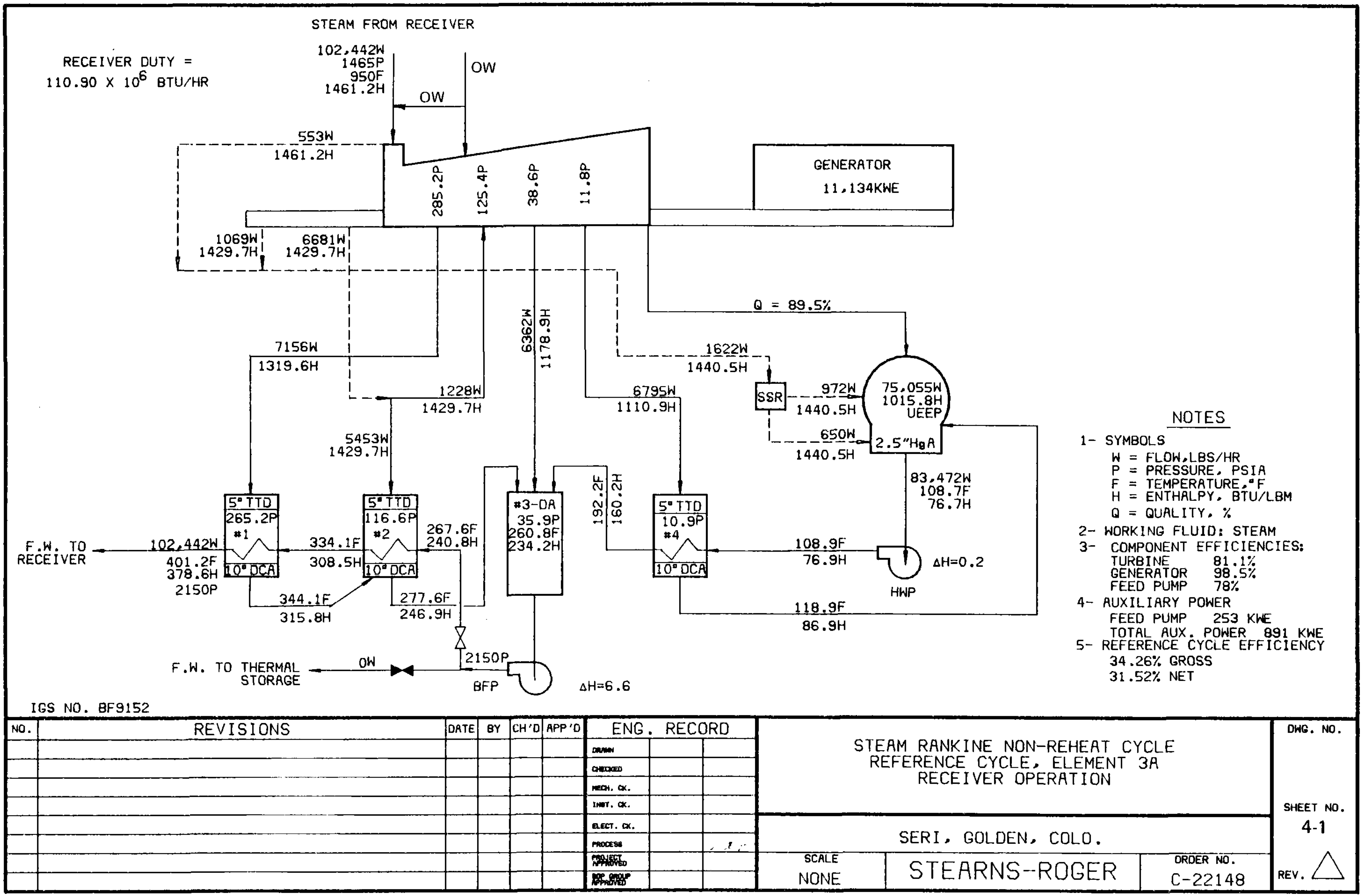


Figure 4—1. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, RECEIVER OPERATION

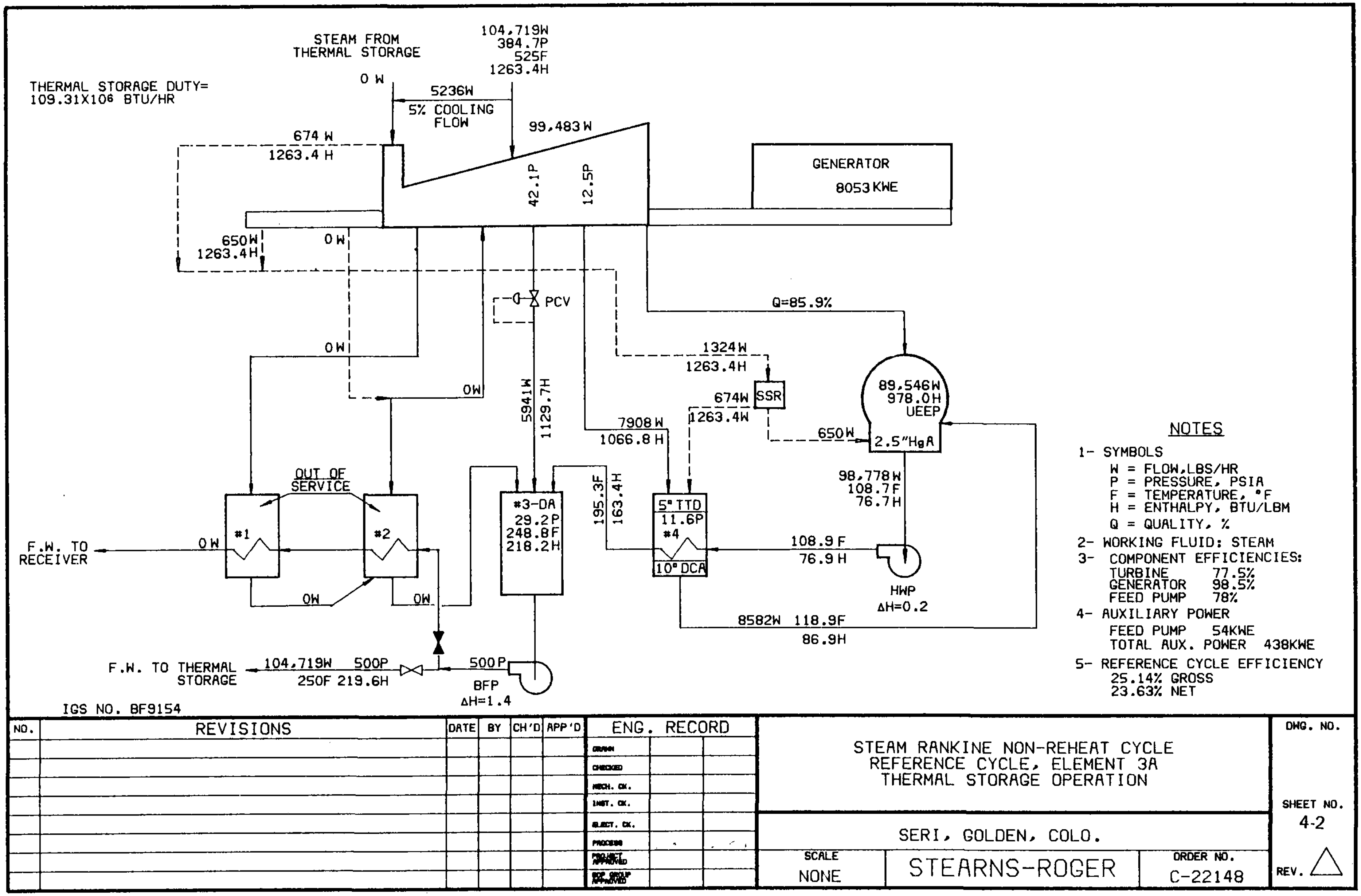


Figure 4–2. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, THERMAL STORAGE OPERATION

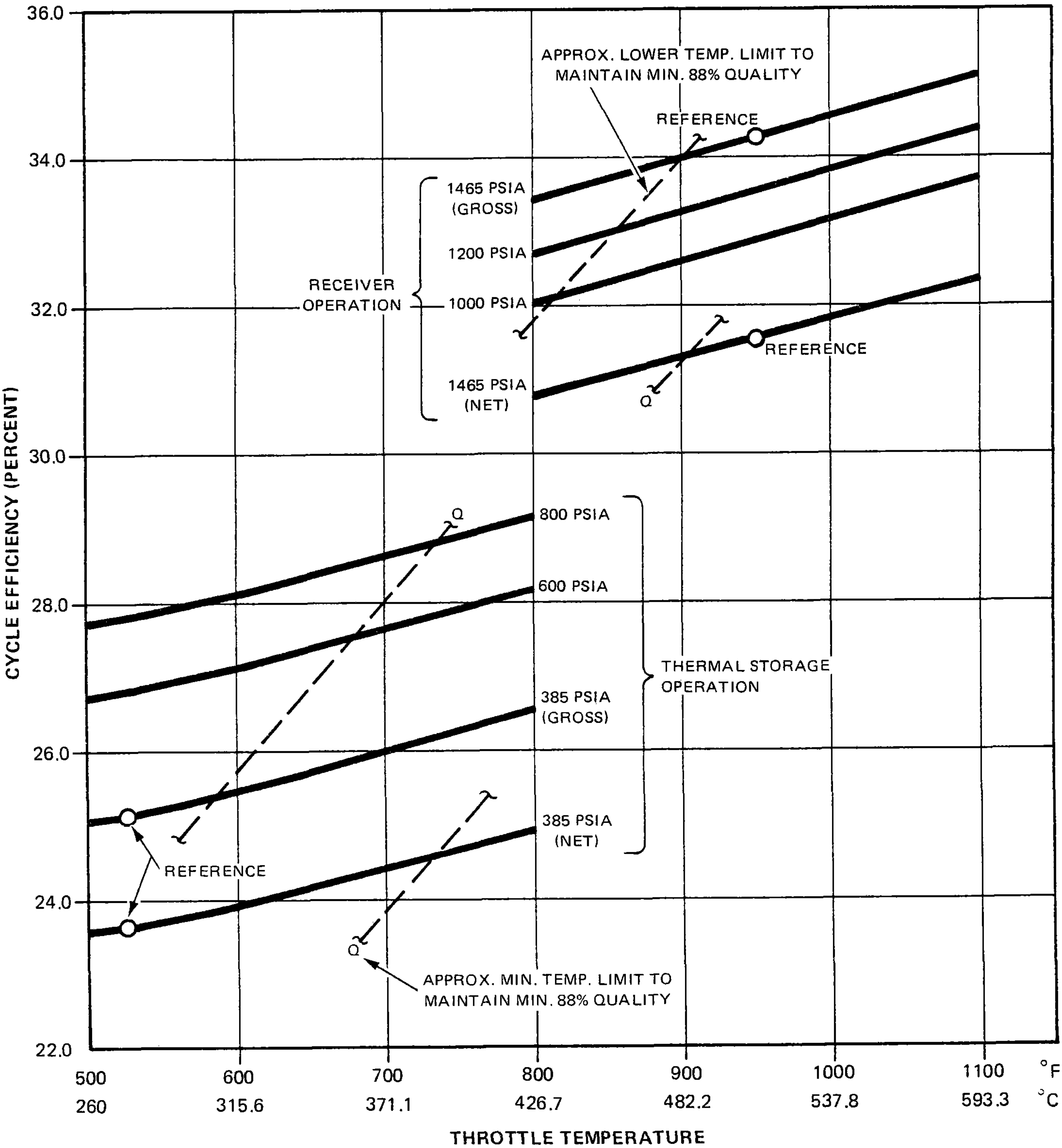


Figure 4–3. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

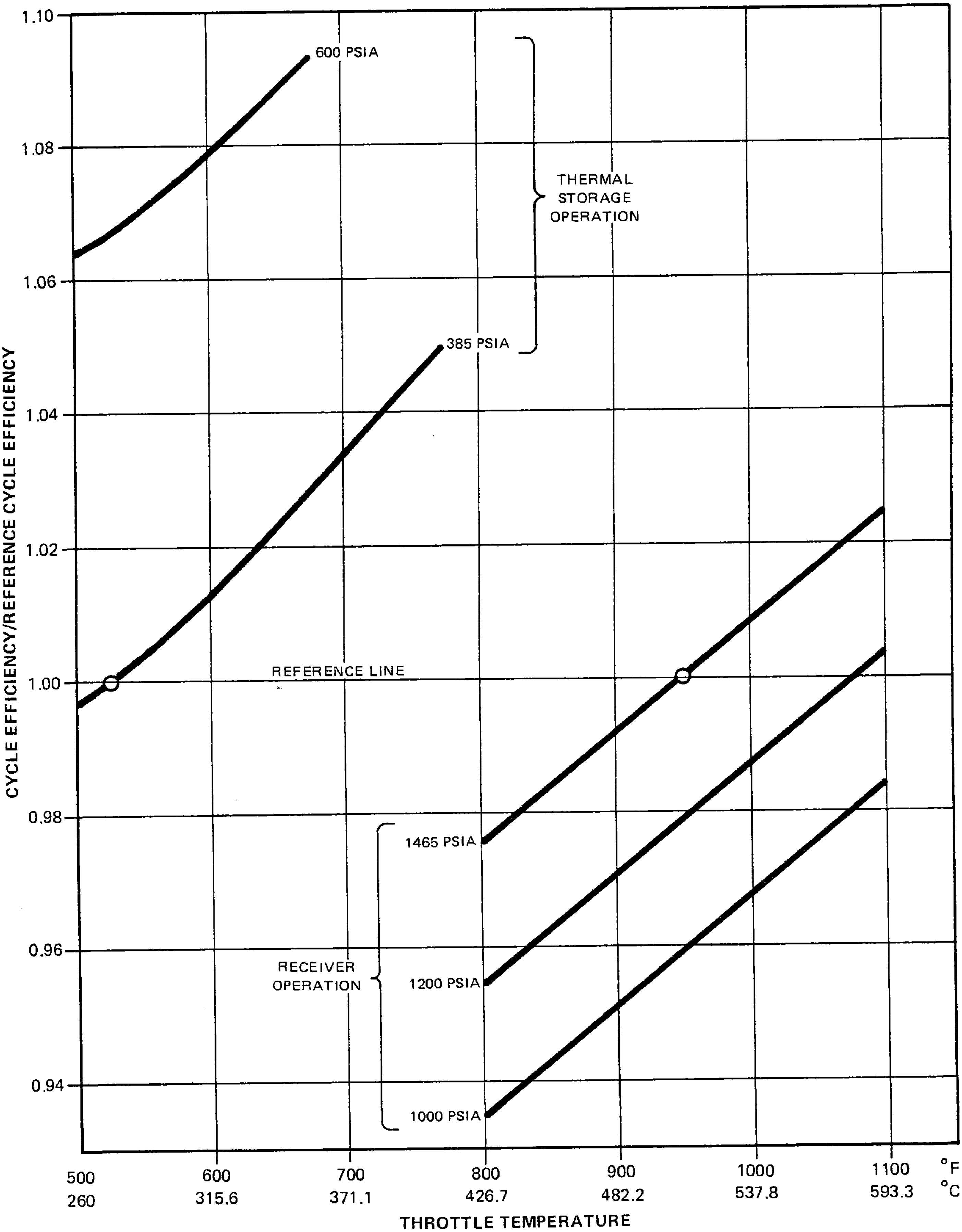


Figure 4—4. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. REFERENCE CYCLE EFFICIENCY

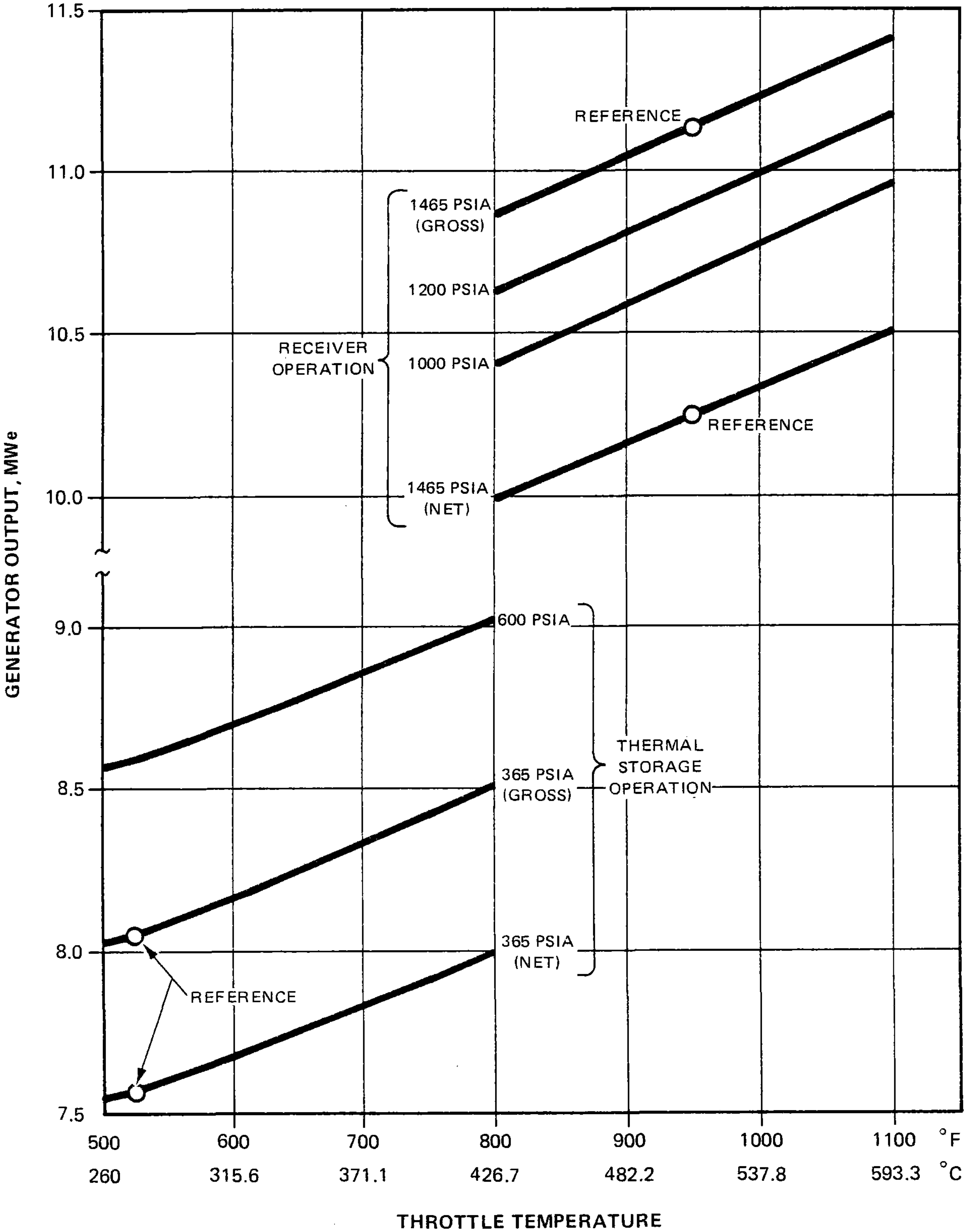


Figure 4–5. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 5
ELEMENT 3B - ORGANIC RANKINE CYCLE

5.1 INTRODUCTION

Performance data were calculated for the reference cycle as defined in the project statement of work for a 150 KWe net generation, organic Rankine Cycle Power System. Toluene (Monsanto Chemical Co. designation CP-25) was used as the working fluid as specified in the statement of work.

5.2 CYCLE DESCRIPTION

The reference cycle is based on a maximum boiler outlet temperature of 350°F (176.7°C) saturated. Saturation pressure is 71.0 psia (489.54 x 10^3 Pa).

The Organic Rankine Cycle is identical, thermodynamically to a steam Rankine cycle used in large utility generating plants, however due to the thermal properties of organic fluids there are certain component differences.

Figure 5—1 shows a schematic of the Organic Rankine Cycle analyzed together with the state points and performance for the 150 KWe (net) reference cycle.

Liquid Toluene is preheated and vaporized in the vaporizer with heat supplied by the solar receiver or thermal storage systems. In most Organic Rankine Systems, the working fluid is admitted to the turbine from the vaporizer with little or no superheat. The reason for this is that for the organic fluids and Toluene specifically, the saturated vapor line when plotted on a T-S diagram has a positive slope. This means that if the fluid leaving the vaporizer and entering the turbine is saturated, vapor expansion through the turbine will result in the turbine exit fluid being considerably superheated, with no danger of wet fluid causing blade erosion as in a conventional steam turbine.

After the working fluid is expanded in the turbine, the vapor is passed through a regenerator which is a vapor/liquid heat exchanger used to remove the superheat from the vapor and transfer this heat to the liquid Toluene at the discharge of the feed pump. The utilization of this superheat to preheat the liquid to the vaporizer improves cycle efficiency since it is not rejected in the condenser.

The condenser used in this analysis is a conventional shell-and-tube, water-cooled heat exchanger. As specified in the work statement, a condensing temperature of 100°F (37.8°C) was used. Two degrees of subcooling was assumed to take place in the condenser to provide adequate net positive suction head at the feed pump inlet.

The feed pump is assumed to be a centrifugal type, and could be powered either by an electric motor drive or be driven directly off the turbine shaft.

Typically, Organic Rankine Cycle Turbines use impluse type blade design and operate at significantly higher speeds than do conventional steam turbines due to the thermodynamic properties of the fluid (Reference 1 and 2). For this reason, Figure 5—1 shows a speed reducer between the turbine and the generator. Details of the turbine/ generator system can only be determined after an engineering design of the system has been performed.

Component efficiences which were used in this analysis are shown on Figure 5—1 and were obtained from Reference 3 and are considered typical for this cycle. These efficiencies were held constant for all the cycle temperatures investigated.

Pressure losses through the system were based on the assumption in Reference 3. The most significant pressure loss relating to cycle performance is the hot-side of the regenerator, since this loss affects the available energy in the turbine when condenser temperature is held constant. Regenerator hot-side losses will directly affect regenerator size and the losses assumed for this study are judged to be typical.

The Toluene fluid property data were obtained from Monsanto Chemical Co., Reference 4.

The analysis for the reference cycle consisted of performing a heat balance around the cycle using a 150 KWe net generator output, 350°F (176.7°C) saturated boiler output and 100°F (37.8°C) condensing temperature. A 5 percent auxiliary power requirement was assumed resulting in a gross generation of 157.5 KWe. The cycle analysis showed that the vaporizer feed pump will require 2.25 KWe with 5.25 KWe available for the remainder of the auxiliary power requirements such as circulating water pump power to the condenser and cooling tower fan power.

A vaporizer heat input requirement was calculated for the reference cycle and was held constant for the cycle calculations at the other maximum temperature conditions. Generator output was allowed to vary for each of the nonreference cycles.

A net and gross cycle efficiency was calculated for a number of fluid temperatures. Gross cycle efficiency is defined as the ratio of the gross generator output divided by the total heat into the cycle. Net efficiency includes the auxiliary power.

The data for the reference cycle are shown on Figure 5—1. The efficiency data for each of the non-reference cycles are shown on Figures 5—2 and 5—3. Cycle efficiency is plotted as a function of maximum cycle temperature on Figure 5—2. Figure 5—3 shows the ratio of cycle efficiency to reference cycle efficiency plotted as a function of maximum cycle temperature. Data were calculated for cycles both below and above the critical point of Toluene. All of the supercritical cycles were calculated at a pressure of 800 psi (5516 x 10^3 Pa). The discontinuity that exists between the subcritical and supercritical cycles is due to the pressure change and the fluid data inconsistancy at the critical point. Auxiliary power requirements of 7 to 9 percent were used for the supercritical cycles. The reason for the higher auxiliary power for these cycles is the increase in feed pump power for the supercritical pressures.

Figure 5—4 shows a plot of gross and net generator output as a function of maximum cycle temperature and based on a constant cycle heat input as determined from the reference cycle.

REFERENCES

1. Bjerklie, J. W. "Working Fluid as a Design Variable for a Family of Small Rankine Power Systems", *ASME* 67-GT-6.

2. Luchter, S. "A Quantitative Method of Screening Working Fluids for Rankine Cycle Power Plants", *ASME* 67-GT-12.

3. Barber, R. E. "Current Costs of Solar Powered Organic Rankine Cycle Engines", *Solar Energy,* Volume 20, pp 1-6, 1978.

4. Monsanto, "Toluene Thermodynamic Grid", Data Ref. 73294, St. Louis 10/12/73.

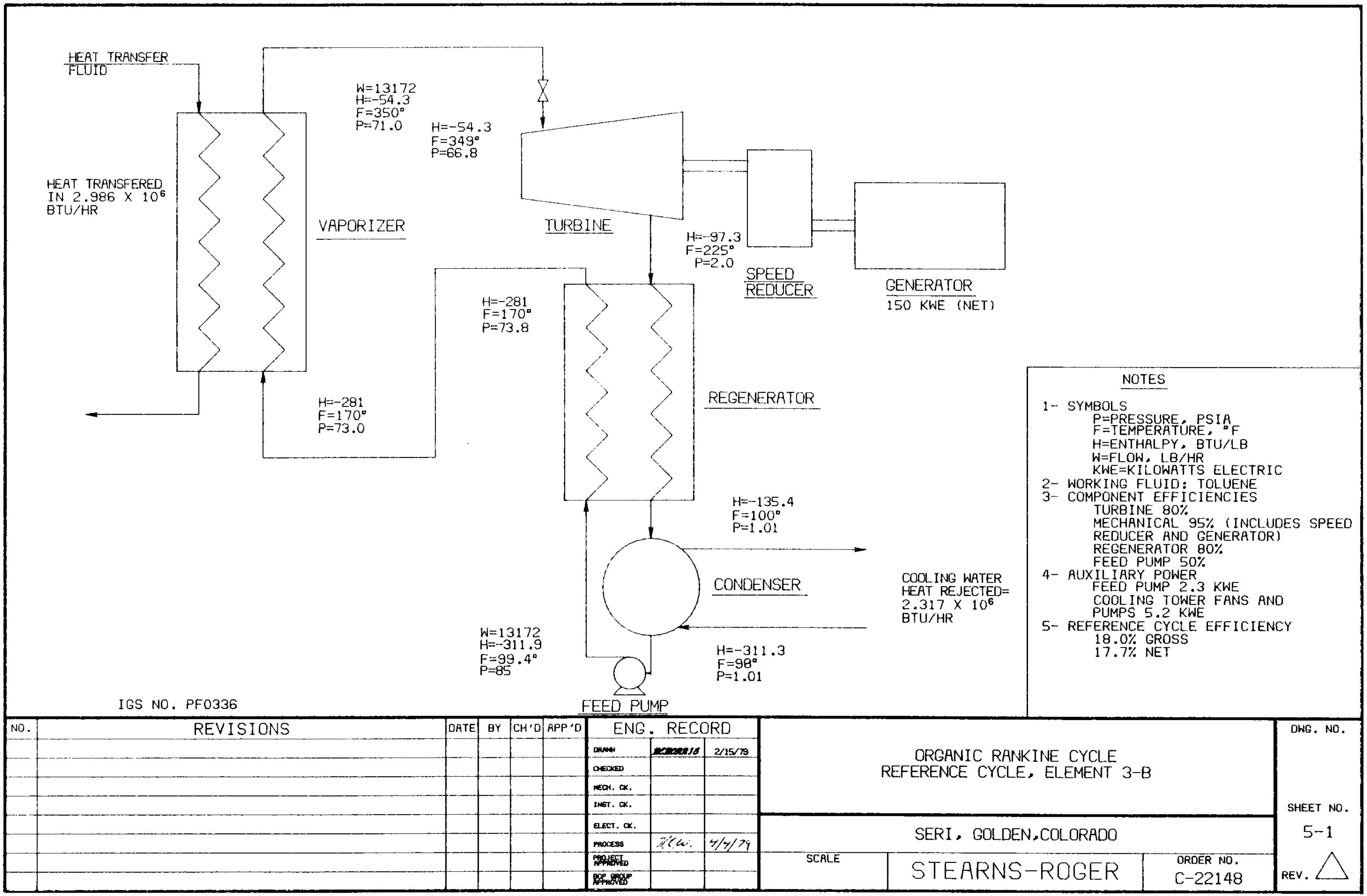


Figure 5–1. ELEMENT 3B - ORGANIC RANKINE CYCLE

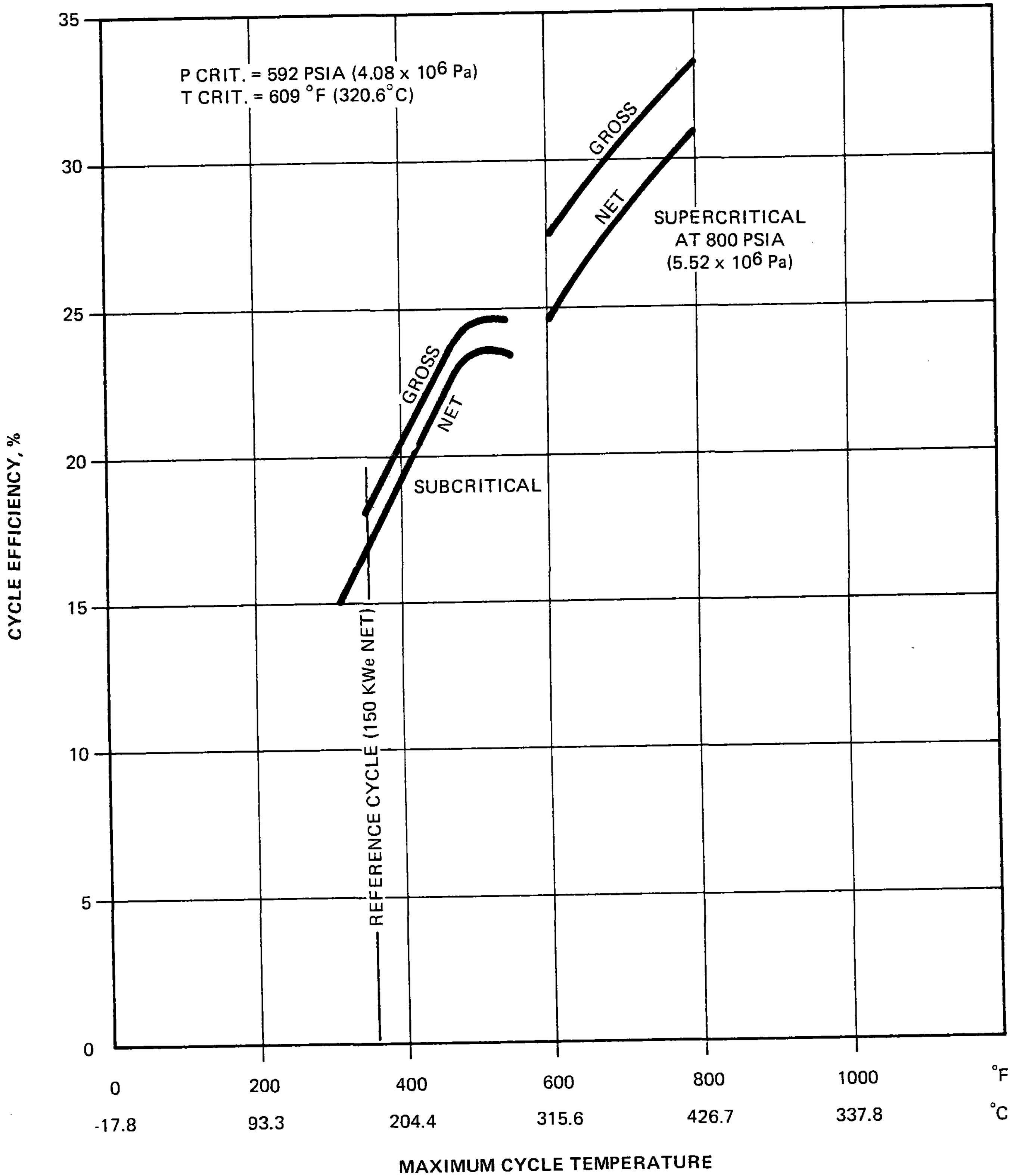


Figure 5–2. ELEMENT 3B - ORGANIC RANKINE CYCLE, CYCLE EFFICIENCY VS. MAXIMUM CYCLE TEMPERATURE

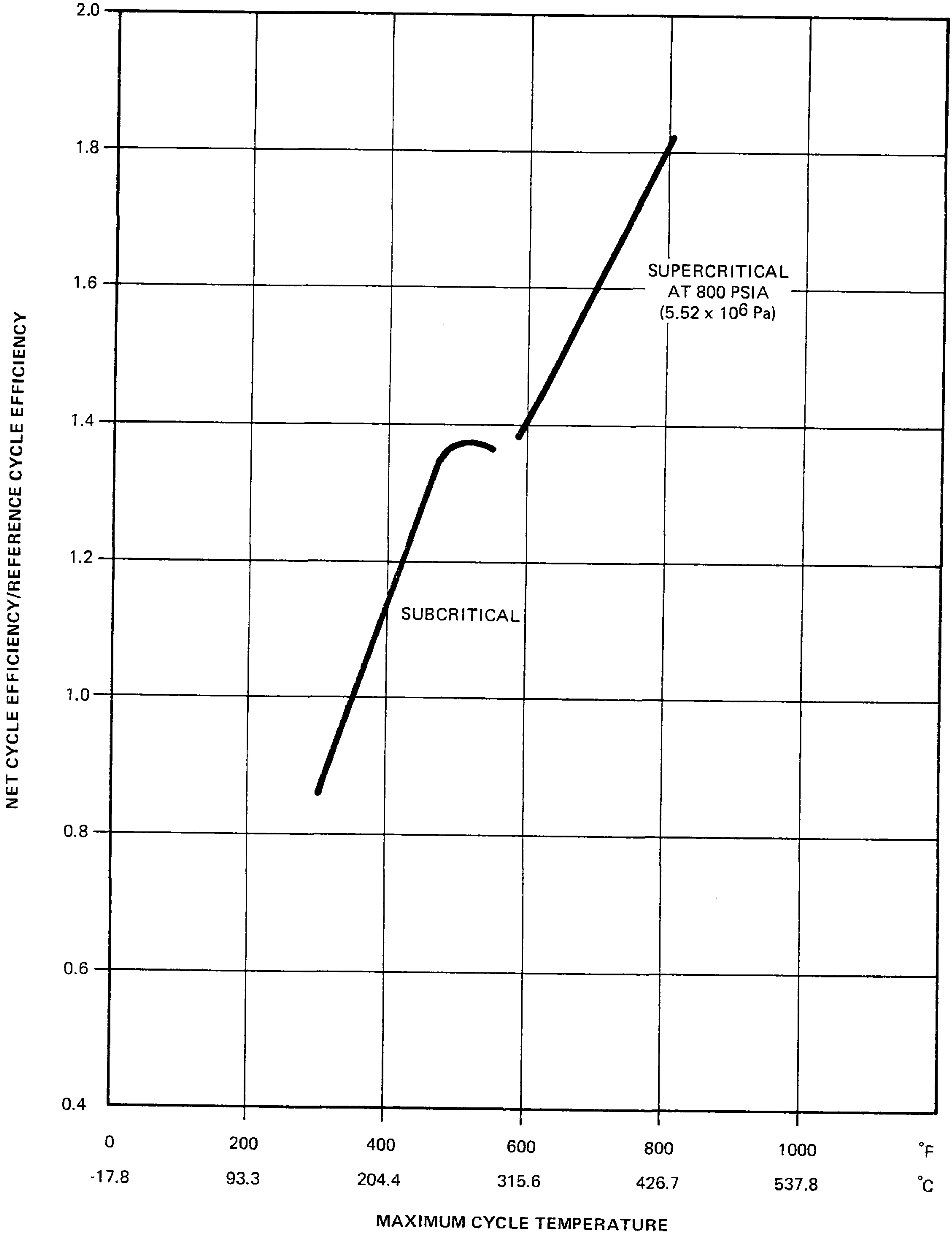


Figure 5—3. ELEMENT 3B - ORGANIC RANKINE CYCLE,
NET CYCLE EFFICIENCY VS. REFERENCE CYCLE EFFICIENCY

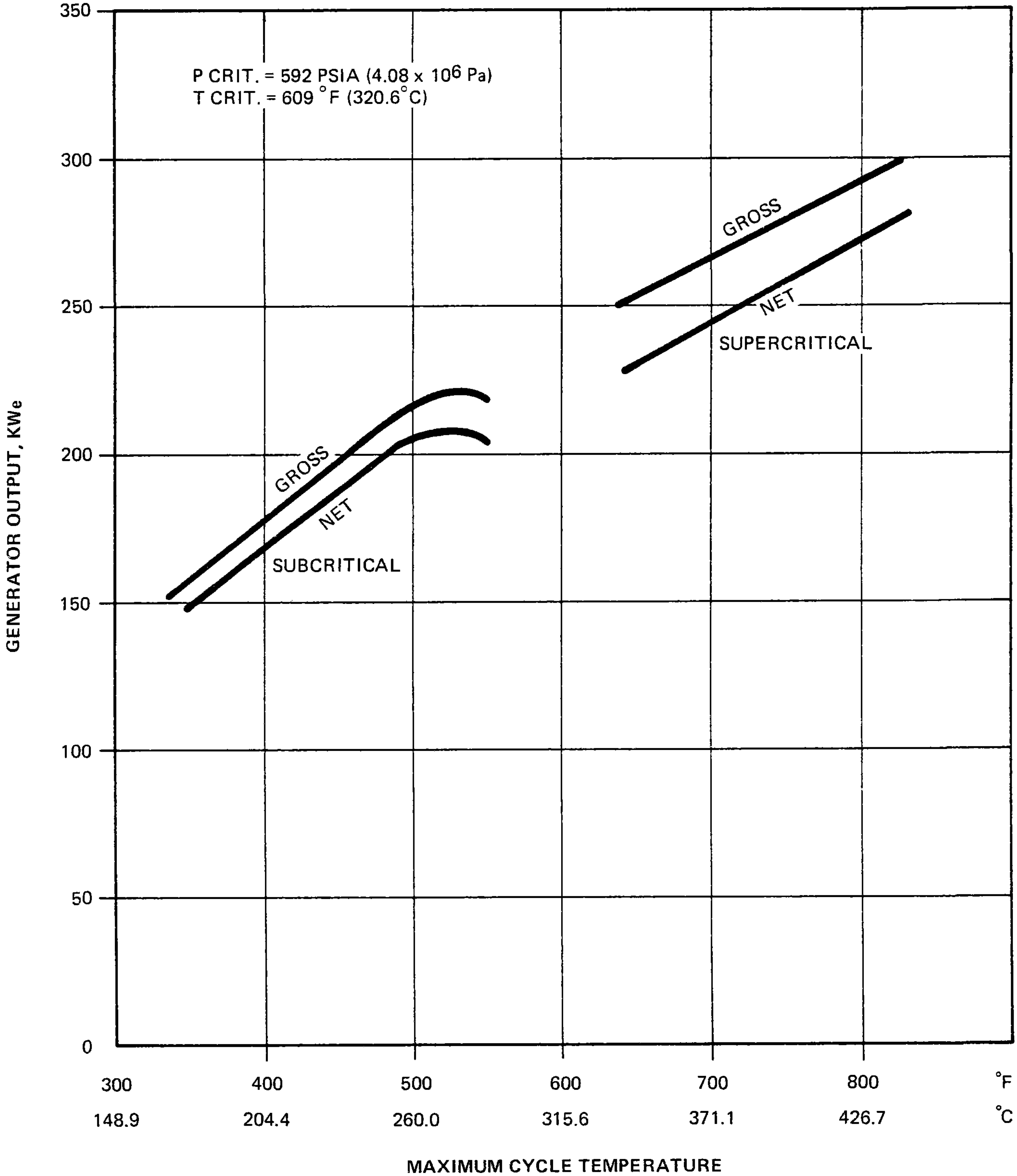


Figure 5–4. ELEMENT 3B - ORGANIC RANKINE CYCLE, GENERATOR OUTPUT VS. MAXIMUM CYCLE TEMPERATURE

SECTION 6
ELEMENT 3C - STEAM RANKINE CYCLE, TOTAL ENERGY SYSTEM

6.1 INTRODUCTION

Cycle heat balances were performed for a 400 KWe (nominal) steam Rankine cycle used in a total energy system. This cycle uses a single automatic extraction condensing turbine with one stage of regenerative feedwater heating. The extraction point on the turbine provides steam at 125 psia (8.62×10^5 Pa) which supplies the deaerator requirement and is desuperheated to 340°F (171.1°C) to supply the process requirement. The condensate is returned to the condenser at a temperature of 230°F (110°C). The turbine exhausts to a condenser at 20.78 psia (1.43×10^5 Pa). The cycle is largely based on the General Electric Solar Total Energy Cycle as referenced in the statement of work.

The cycle was analyzed using a computer program to perform the required mass and energy balances around the cycle. The program assumes that the expansion of the steam through the turbine is a straight line on a Mollier diagram. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

6.2 CYCLE DESCRIPTION

Throttle conditions used for the reference cycle are 715 psia (4.93×10^6 Pa) and 720°F (382.2°C). A pressure loss of 10 percent is assumed across the steam generator.

For this study, the total heat input to the cycle and the process steam usage was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 650°F to 1100°F (343.3°C to 593.3°C), and the throttle pressure was varied over the range of 715 psia to 1450 psia (4.93×10^6 Pa to 10.0×10^6 Pa). The throttle pressures selected are those normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower temperature limit varied with throttle pressure such that the required process steam temperature could be achieved at the extraction point. For the 715 psia (4.93×10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C); for the 850 psia (5.86×10^4 Pa) pressure, the minimum temperature is 700°F (371.1°C); and for the 1250 psia and 1450 psia (8.62×10^6 Pa and 10.0×10^6 Pa) pressures, the minimum temperature is 800°F (426.7°C). The turbine efficiency was calculated from the throttle and exhaust conditions of the reference cycle, the General Electric Solar Total Energy System. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on the overall cycle efficiency. In reality, the turbine efficiency will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

A cycle schematic showing component efficiencies and flow data is shown in Figure 6—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 6—2 and Figure 6—3. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 6—4. A plot of generator output versus throttle temperature is shown in Figure 6—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

Figure 6—6 is a plot of gross cycle efficiency versus process flow for the base case throttle conditions. The efficiency of the cycle decreases as process flow increases because the boiler duty is held constant. As more steam is extracted to the process, less steam is available to produce electric power.

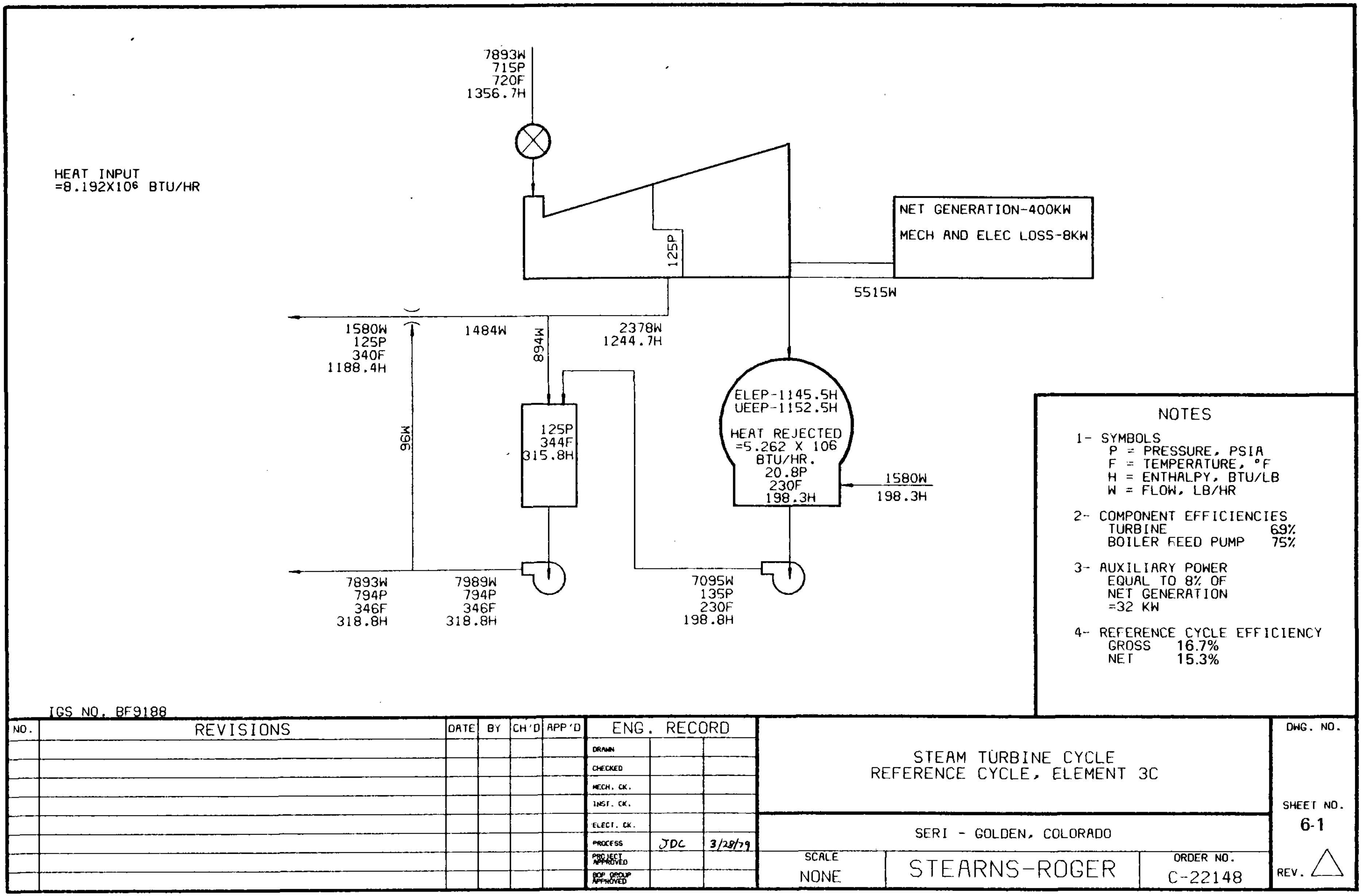


Figure 6—1. ELEMENT 3C - STEAM RANKINE CYCLE

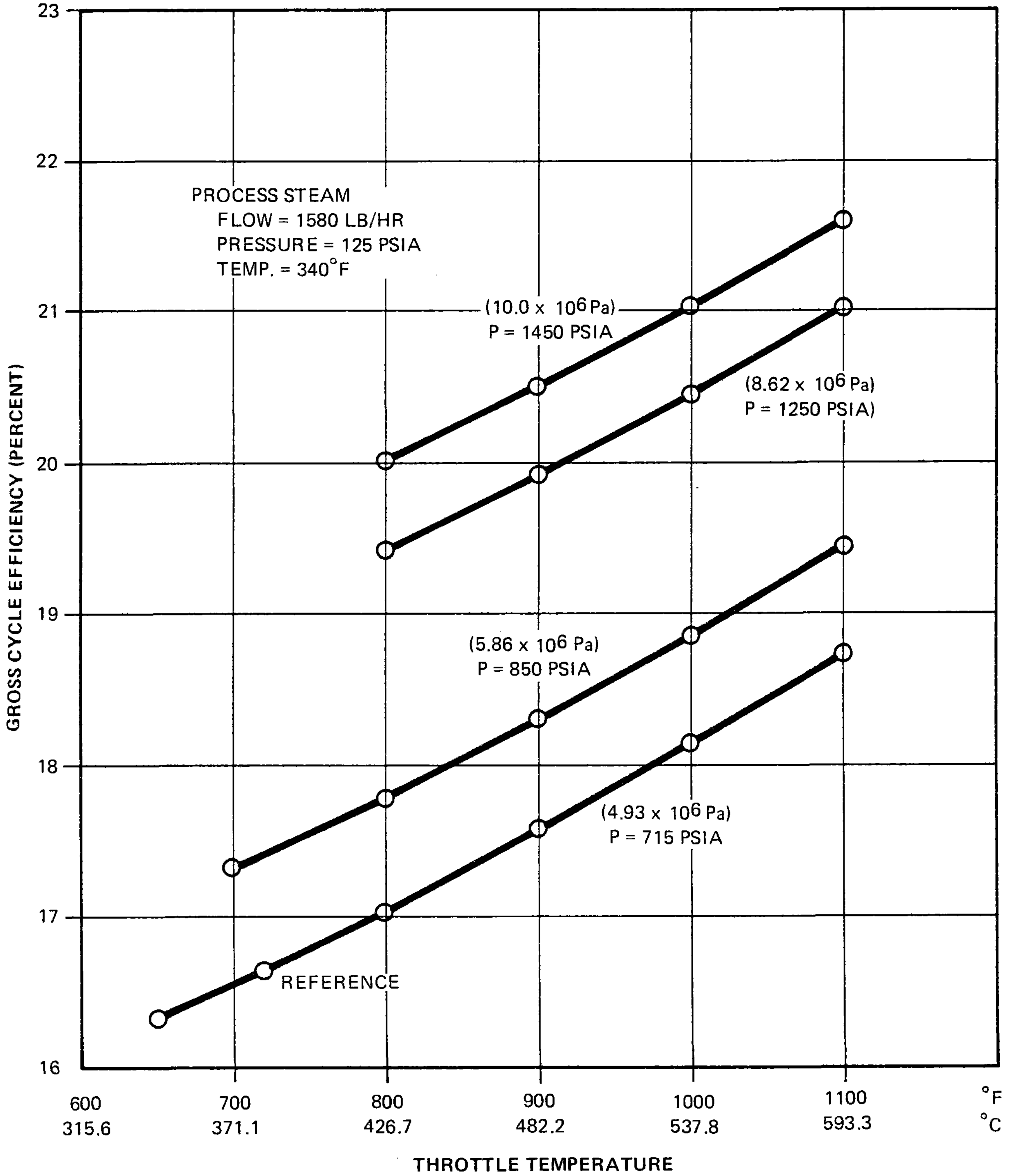


Figure 6–2. ELEMENT 3C - STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

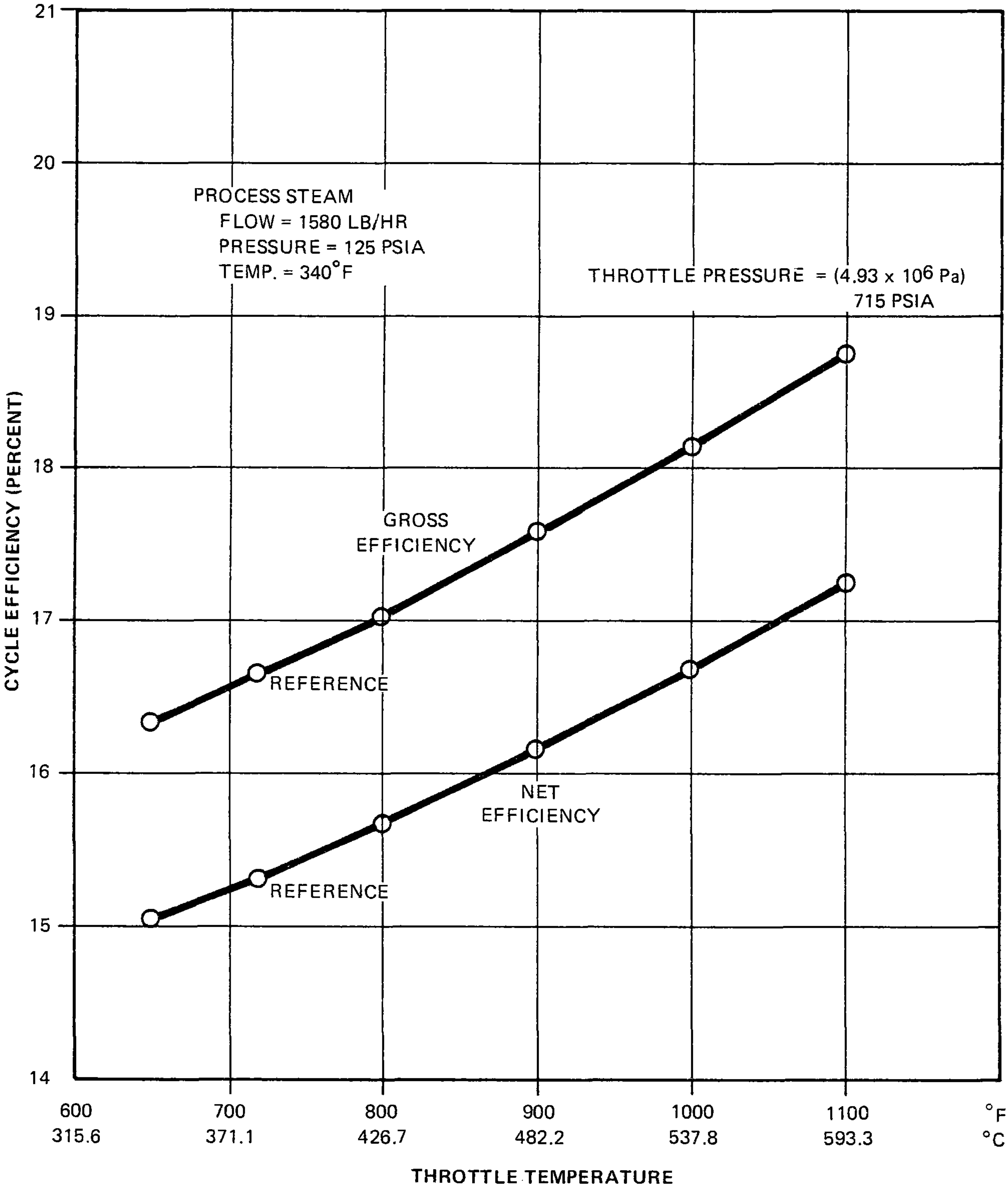


Figure 6–3. ELEMENT 3C - STEAM RANKINE CYCLE, EFFICIENCY VS. THROTTLE TEMPERATURE

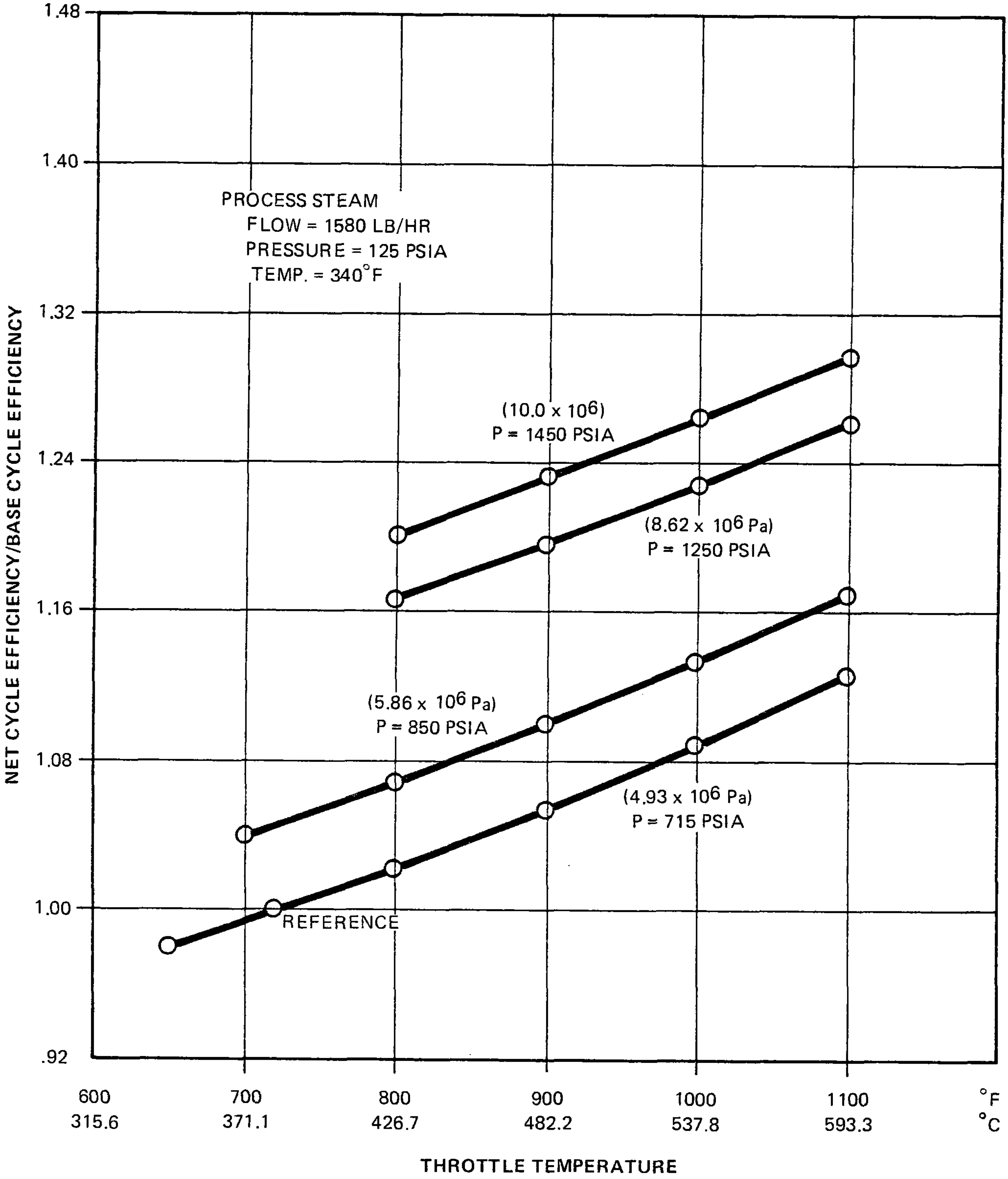


Figure 6—4. ELEMENT 3C - STEAM RANKINE CYCLE, RELEATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

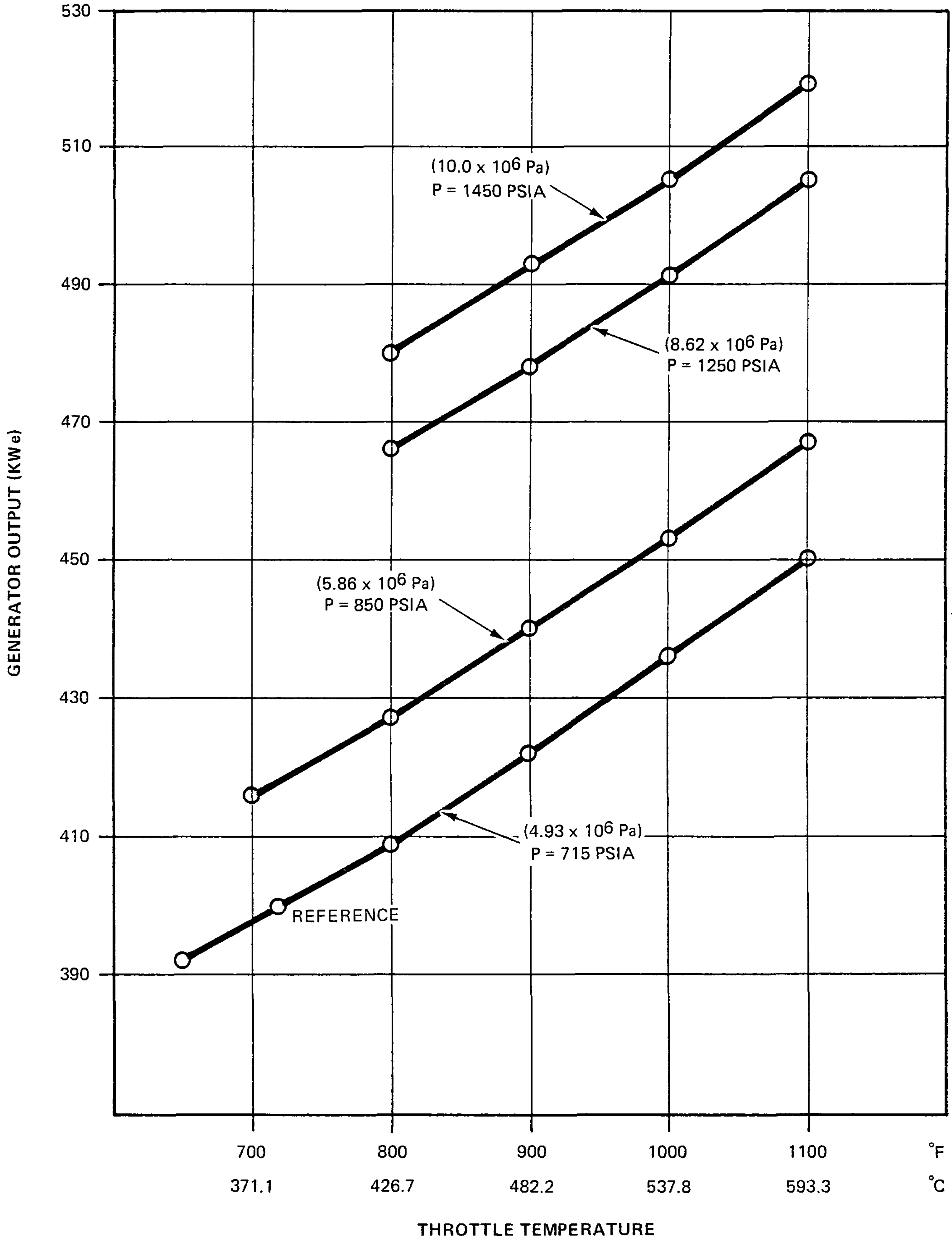


Figure 6–5. ELEMENT 3C - STEAM RANKINE CYCLE, GENERATOR OUT PUT VS. THROTTLE TEMPERATURE

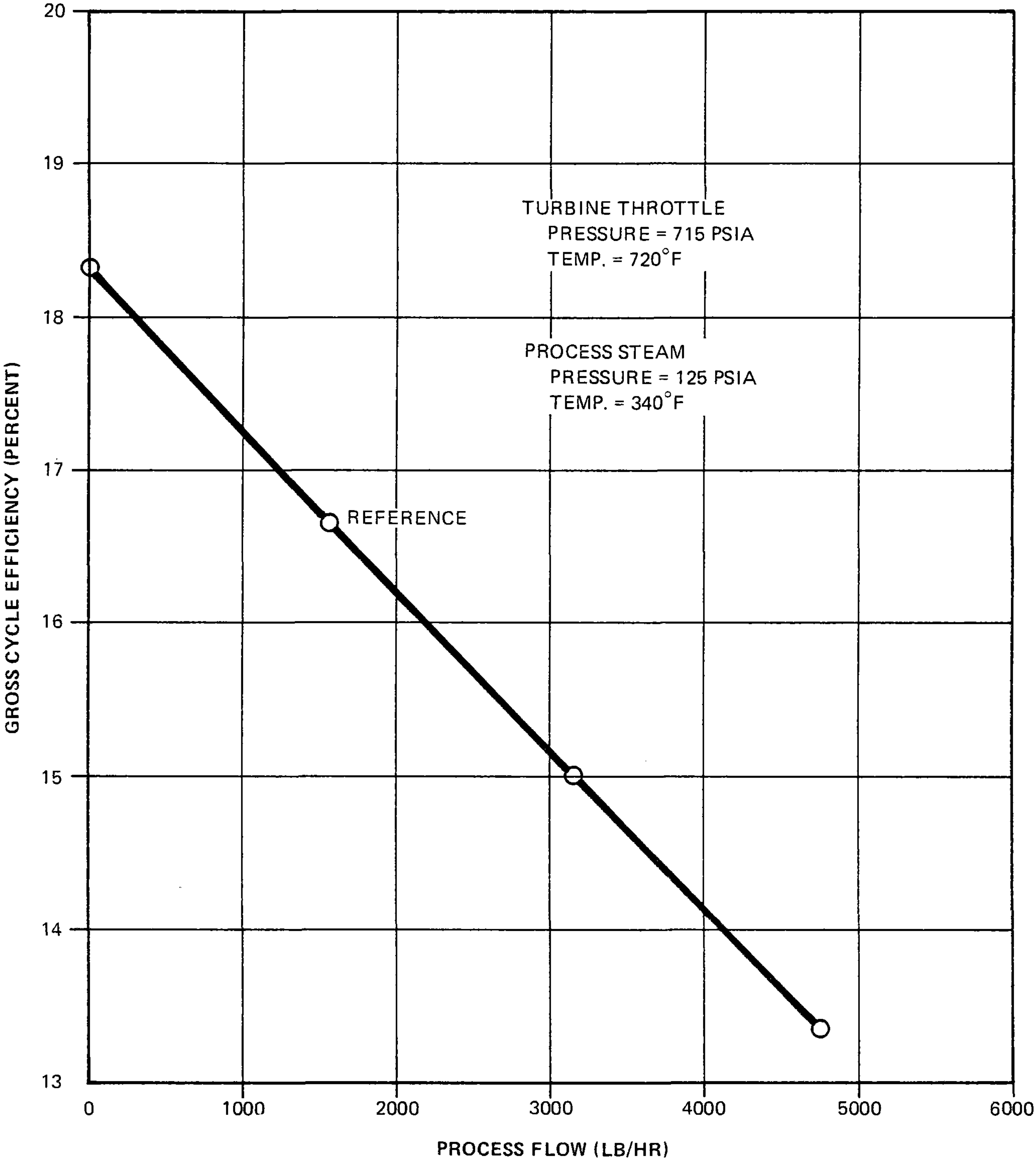


Figure 6–6. ELEMENT 3C - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. PROCESS FLOW

SECTION 7
ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE

7.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) steam Rankine, non-reheat cycle incorporating a six heater feedwater heating system. The cycle was duplicated from that of the Public Service of New Mexico Repowering Study, Reference 1, using steam from a solar receiver to repower an existing fossil-fuel-fired unit. A seventh condensing feedwater heater is added to the top of the cycle to recover heat from the discharge of the thermal storage system. It is assumed that the amount of heat absorbed in the thermal storage unit reduces the steam conditions from superheated (throttle temperature and pressure) to saturated conditions, after a pressure drop of 10 percent through the thermal storage unit. For this study the fossil-fuel-fired boiler was out of service.

7.2 CYCLE DESCRIPTION

The turbine used in this model is a standard-frame General Electric utility, non-reheat steam turbine exhausting at an assumed 2.5 inches HgA (0.0984 mm HgA) to a tube-and-shell condenser. The turbine's six extractions are connected to two closed high-pressure heaters, an open deaerating heater, and three closed low-pressure heaters operating at various pressures. The heater operating characteristics are as shown on the cycle diagram Figure 7—1.

Throttle conditions are 1250 psia (8.62×10^6 Pa) and 950°F (510°C) for the reference cycle. At these throttle conditions (no steam to thermal storage, and constant final feedwater conditions of 2400 psia (16.55×10^6 Pa) and 425°F (218.3°C)), the heat input to the reference cycle was determined. This heat input was held constant for the varying throttle temperatures and pressures, and thermal storage duties studied. The amount of steam to thermal storage was varied from zero up to the point at which the feedwater temperature leaving the thermal storage heater equals the temperature of the heater shell (i.e., saturation temperature at heater shell pressure). This yields a hot end terminal temperature difference (TTD) of 0°, for which it is assumed the thermal storage heater is designed. The throttle temperatures were varied from the reference cycle conditions down to a temperature of 800°F (427°C) which yields approximately 84 percent quality steam leaving the last stage of the turbine, and up to a temperature of 1100°F (593°C) (the upper limit for existing steam turbine technology). It is expected that the turbine will actually operate at a minimum quality of approximately 88 percent steam leaving the last stage. Generator output was allowed to vary with the throttle conditions.

A cycle schematic diagram showing all component efficiencies is presented in Figure 7—1. A parametric plot of cycle efficiency versus throttle temperature at different throttle pressures and thermal storage duties is presented in Figure 7—2. Component efficiencies were determined from Reference 2 (turbine) and from existing conventional power plant operating data (pumps). It must be kept in mind that each pressure, temperature and extraction as a percent of boiler duty shown on Figure 7—2 represents a discrete storage system and heater design. Off design conditions were not considered. The cycle efficiencies are presented as gross (total energy output divided by total energy input to the cycle) and net (assuming 8 percent of total energy output including pumping power goes to auxiliary demand). The ratio of non-reference cycle efficiency to reference cycle efficiency, and generator output (gross and net) are also presented in this Figure. The maximum percent of receiver duty to thermal storage versus throttle temperature is plotted in Figure 7—3 to illustrate the limits of heat to the thermal storage heater before the feedwater temperature reaches a maximum, based on the assumptions above.

REFERENCES

1. Maddox, J. D., Public Service Company of New Mexico, "A Technical and Economic Assessment of Solar Hybrid Repowering," SAMD 78-8511, November 1978, p. 65.

2. General Electric Company, "A Method for Predicting the Performance of Steam-Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

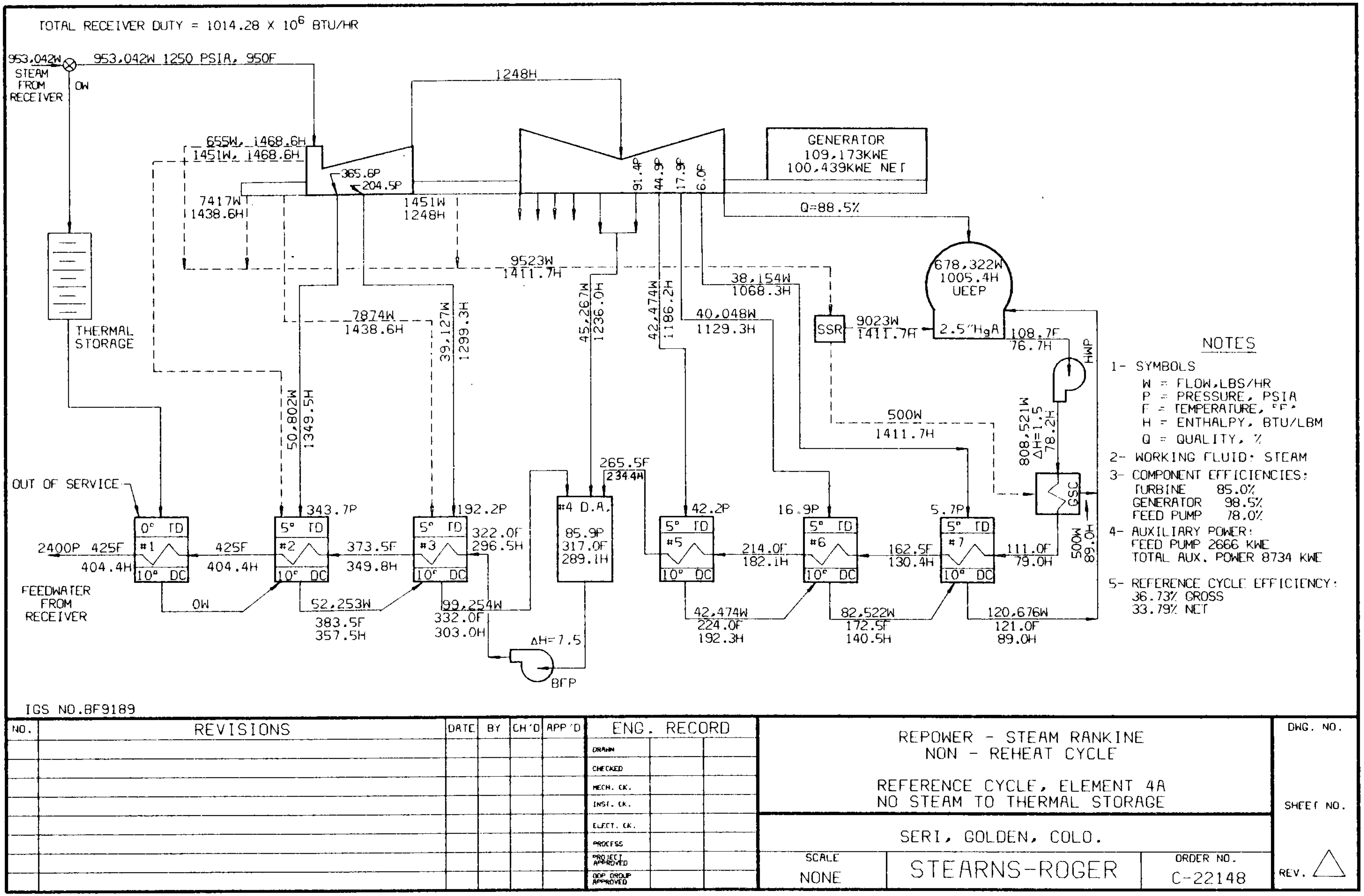
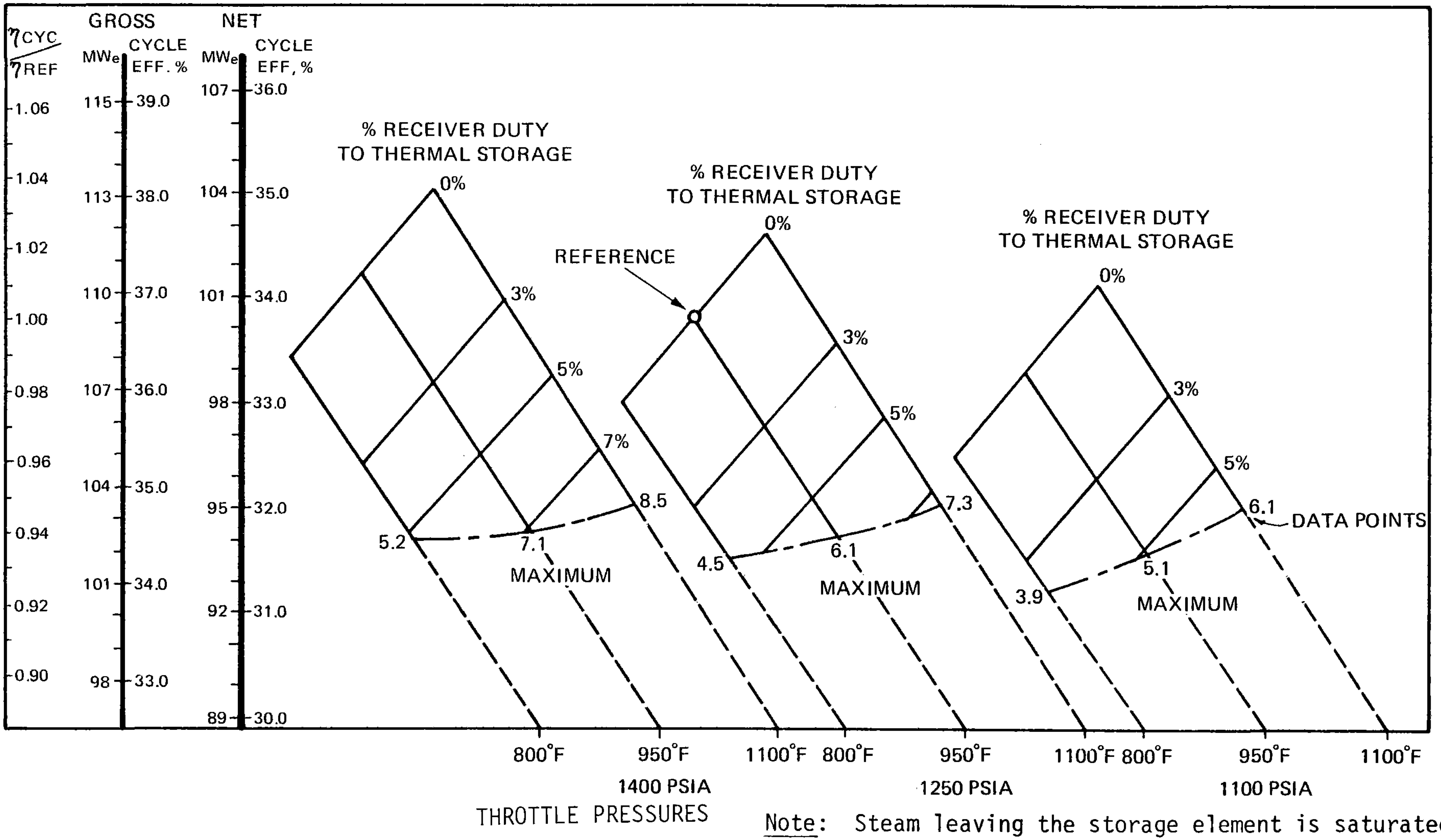


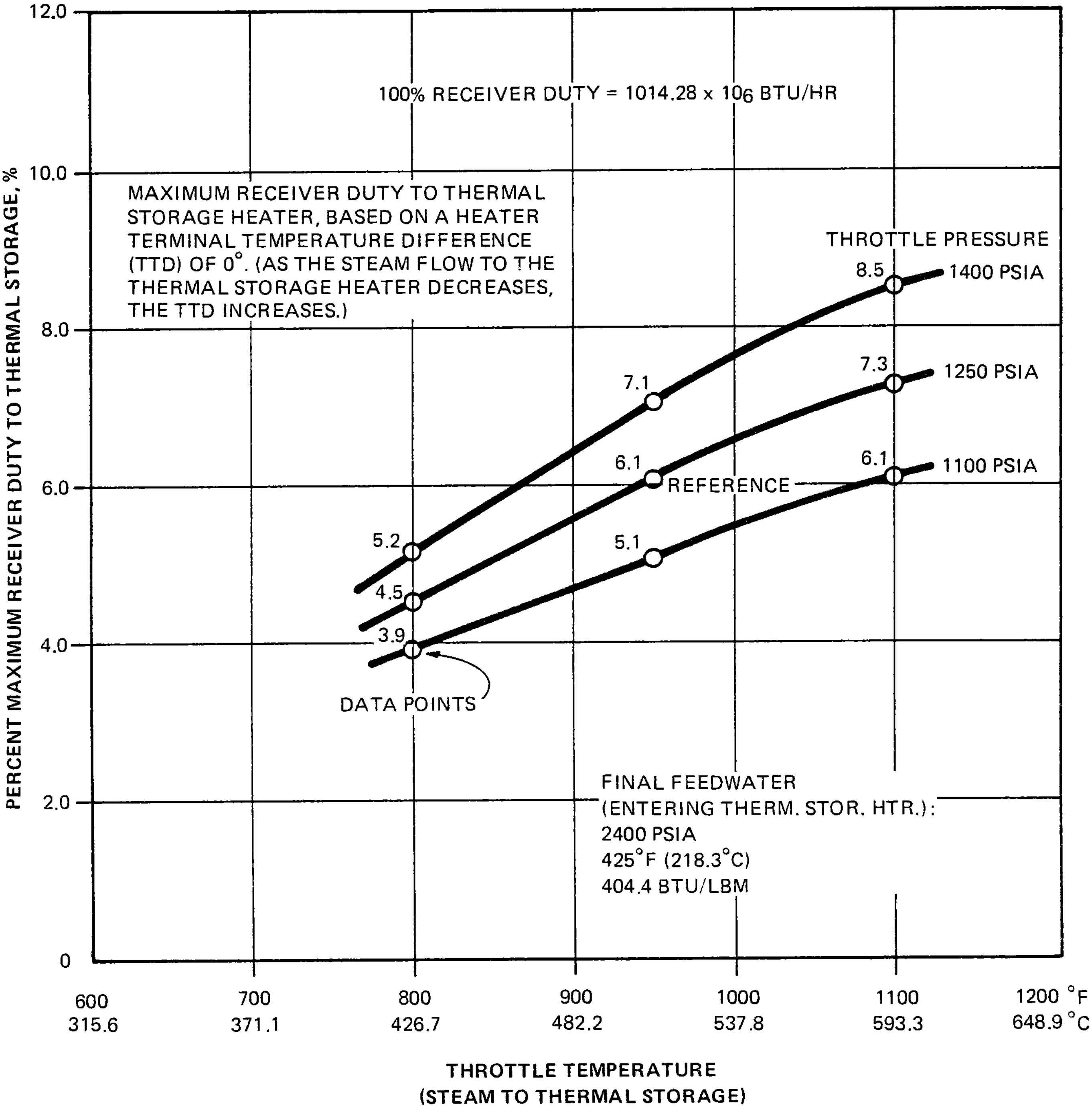
Figure 7—1. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE



REFERENCE CYCLE: 1250 PSIA; 950°F; 0% RCVR. DUTY TO THERM. STOR.
CONSTANT RECEIVER DUTY: 1014.28 MILLION BTU/HR

Note: Steam leaving the storage element is saturated vapor. The heat transferred into storage is the heat contained in the vapor superheat. The % receiver heat duty to thermal storage is, therefore, ~1/3 of the % flow. At 7% receiver heat duty to storage, ~21% of the steam flow is diverted to storage.

Figure 7–2. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE; CYCLE EFFICIENCY, GENERATOR OUTPUT, CYCLE EFFICIENCY/REFERENCE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE AT VARYING THERMAL STORAGE DUTIES AND CONSTANT RECEIVER DUTY



Note: The terminal temperature difference is the ΔT between the feedwater leaving the last feedwater heater and the steam entering the feedwater heater from the storage element.

Figure 7–3. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE, PERCENT RECEIVER DUTY TO THERMAL STORAGE VS. THROTTLE TEMPERATURE

SECTION 8
ELEMENT 4C - CLOSED ADVANCED BRAYTON AIR REGENERATIVE CYCLE

8.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) closed advanced Brayton air regenerative cycle. Heat input to the cycle is from a solar receiver/thermal storage system. References 1, 2 and 3 were used to set up the thermodynamic relationships to be used in the computer model for performance calculations and to determine component efficiencies.

8.2 CYCLE DESCRIPTION

The reference cycle for this element based on Reference 4, is composed of a two-stage compressor with efficiencies of 80 percent each and a total combined pressure ratio of 4.75. Inlet air is compressed in the first stage, cooled in a water-cooled intercooler, compressed further in the second stage, and discharged to a recuperator. Cooling air for the turbine is diverted from the compressor discharge to various parts of the turbine. Cooling air which is injected into the turbine in the flow path does useful work, while cooling air going to the turbine casing does no work. From the compressor, the air flows to the recuperator where it is heated by the turbine exhaust. The air temperature is further increased by the solar receiver/thermal storage system to a temperature of 1500°F (815.6°C) for the reference cycle. The hot air is expanded in a 90 percent efficient turbine to produce shaft work which drives the compressor and the generator. The turbine exhaust passes through the recuperator where it releases heat to the compressor discharge air. From the recuperator, the turbine exhaust is further cooled by a water-cooled precooler to a constant 100°F (37.8°C) prior to reentering the compressor.

A general industry guide was used to determine how much of the total system flow was used as cooling air which did no work in the turbine. This guide is that cooling air is required above a turbine inlet temperature of 1700°F (926.7°C), and will be one percent of the compressor flow for each 100°F (55.6°C) above 1700°F. The cooling air flow considered in this study represents that amount of flow that does no work in the turbine.

For this study, the generator output was held constant at 100 MWe gross, while the pressure ratio of the compressor was varied over the range of 2 to 9 at various turbine inlet temperatures ranging from 1500°F (815.6°C) to 2400°F (1315.6°C). A cycle schematic showing component efficiencies, pressure drops, heat exchanger effectiveness, and cycle flow data is shown in Figure 8-1 for the reference cycle. A plot of cycle efficiency (gross and net) versus compressor pressure ratio at several turbine inlet temperatures is shown in Figure 8-2. A normalized representation of cycle efficiency with respect to reference cycle efficiency versus compressor pressure ratio is also shown in Figure 8-2. The effects of the cooling air flow on cycle efficiency are demonstrated by the temperature lines of Figure 8-2 crossing each other at lower pressure ratios. Note that there is no cooling flow for turbine inlet temperatures below 1700°F (926.7°C).

Gross cycle efficiency is defined as the gross generator output divided by the total cycle input energy. Net cycle efficiency is defined as the net generator output divided by the total cycle input energy, where the total plant auxiliary requirements are assumed to be 8 percent of the gross generator output.

REFERENCES

1. Lewis, A. D., *Gas Dynamics,* 1964.
2. Faires, V. M., *Thermodynamics,* 1959.
3. Telephone conversations with various gas turbine manufacturers relative to current compressor and turbine efficiencies and cooling flows.
4. Gintz, J., Boeing Engineering and Construction, "Closed Cycle Brayton Advanced Central Receiver Solar Thermal Electric Power Plant," SAND 78-8511, November 1978, p. 85.

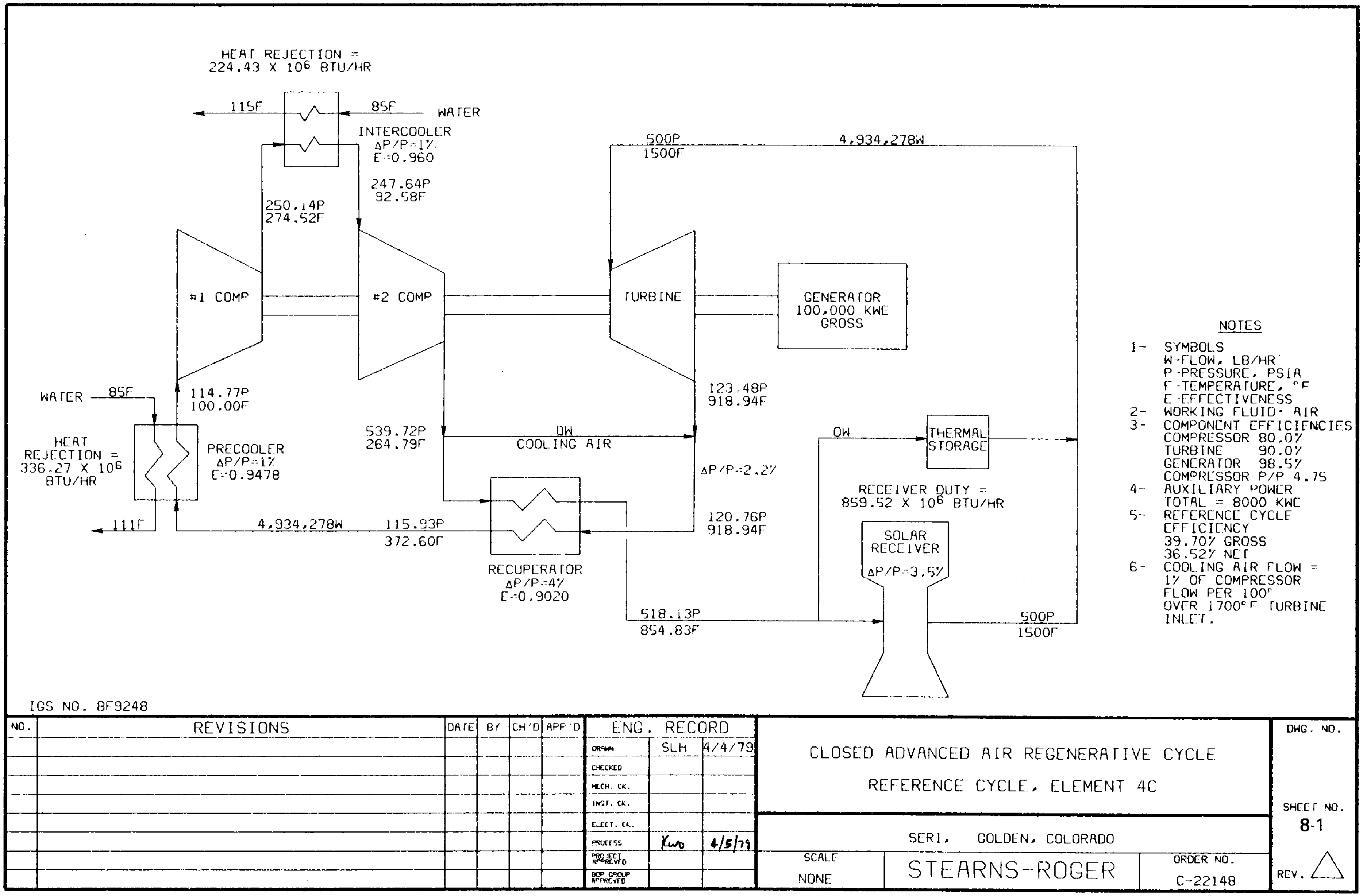


Figure 8—1. ELEMENT 4C - CLOSED ADVANCED AIR REGENERATIVE CYCLE, REFERENCE CYCLE

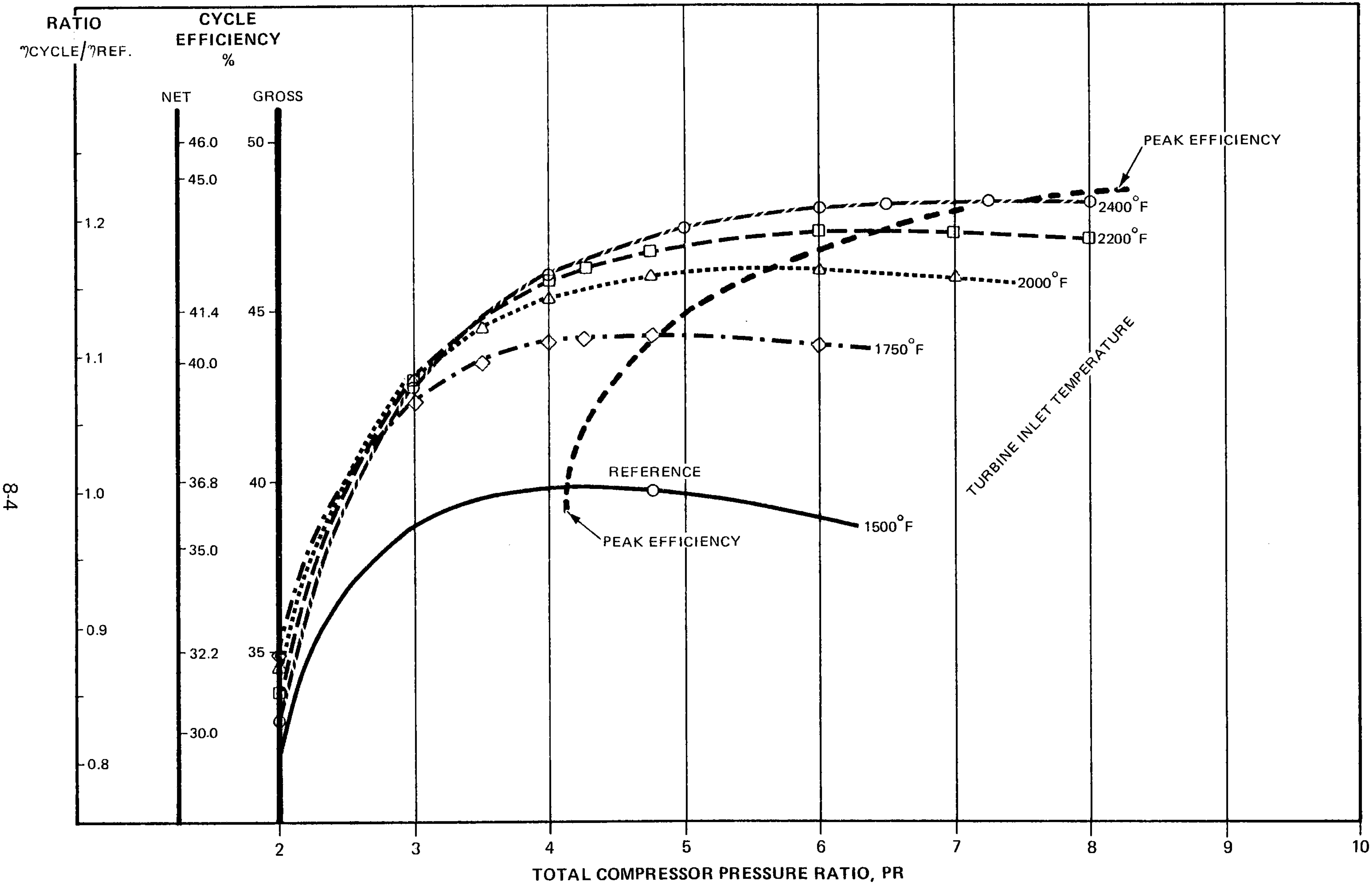


Figure 8–2. ELEMENT 4C - CLOSED ADVANCED AIR REGENERATIVE CYCLE, CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO & TURBINE INLET TEMPERATURE

SECTION 9
ELEMENT 5 - REHEAT STEAM RANKINE CYCLE

9.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) reheat steam Rankine cycle employing six stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.40×10^3 Pa). An allowance is made to extract up to 50 percent of the throttle flow from the second extraction on the low pressure turbine for use by the thermal storage system in a thermochemical reaction. The extraction pressure was varied to be 100, 150, and 200 psia (6.90×10^5, 1.03×10^6, and 1.38×10^6 Pa). Condensate is returned to the cycle in the condenser and is assumed to be at a temperature of 200°F (93.3°C).

The cycle was analyzed using an in-house computer program (D135E) which performs a mass and energy balance around the cycle. The shape of the turbine expansion curve is as specified in Figure 25 in Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

9.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 2400 psia (16.55×10^6 Pa) and 1000°F (537.8°C), with the reheat temperature of 1000°F (537.8°C). The final feedwater temperature selected is 480°F (248.9°C), which allows a reasonable pressure ratio across the high pressure turbine. A 10 percent pressure loss is assumed across the steam generator, and a pressure loss of 15 percent is assumed across the reheater.

For this study, the total heat input to the cycle was held constant while the throttle conditions and reheat temperature were varied. The throttle and reheat temperatures were varied over the range of 800°F to 1100°F (426.7°C to 593.3°C) and the throttle pressure was varied over the range of 1250 psia to 2400 psia (8.62×10^6 Pa to 16.55×10^6 Pa). The lower temperature limit was selected to limit the turbine exhaust steam quality to 88 percent and the upper temperature limit was selected as the limit of existing steam turbine technology. The throttle pressures selected are those normally used in the power industry. Representative high pressure and low pressure turbine efficiencies were calculated using the method specified in Reference 1. The representative turbine efficiencies were based on the throttle conditions of the reference cycle. The turbine efficiencies were held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbines will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

The turbine extractions provide steam to three high-pressure closed feedwater heaters, an open deaerating feedwater heater, and two low-pressure closed feedwater heaters. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Most turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. One leakage was included, the shaft leakage from the high pressure turbine to the low pressure turbine, as this is three percent of the throttle flow. The neglected leakages include shaft leakage from the exhaust of the HP turbine, the sealing flows to the LP turbine shaft seals, and packing leakage from the turbine stop and control valves. The total of these leakages are typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 9—1 for the base cycle, and in Figure 9—2 for the base cycle with an extraction flow of 50 percent of throttle flow. Plots of cycle efficiency versus throttle temperature are shown in Figure 9—3 and Figure 9—4. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 9—5. A plot of generator output versus throttle temperature is shown in Figure 9—6. The change in cycle efficiency versus the extraction flow is shown in Figure 9—7. The change in generator output versus the extraction flow is shown in Figure 9—8. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power required (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the receiver feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

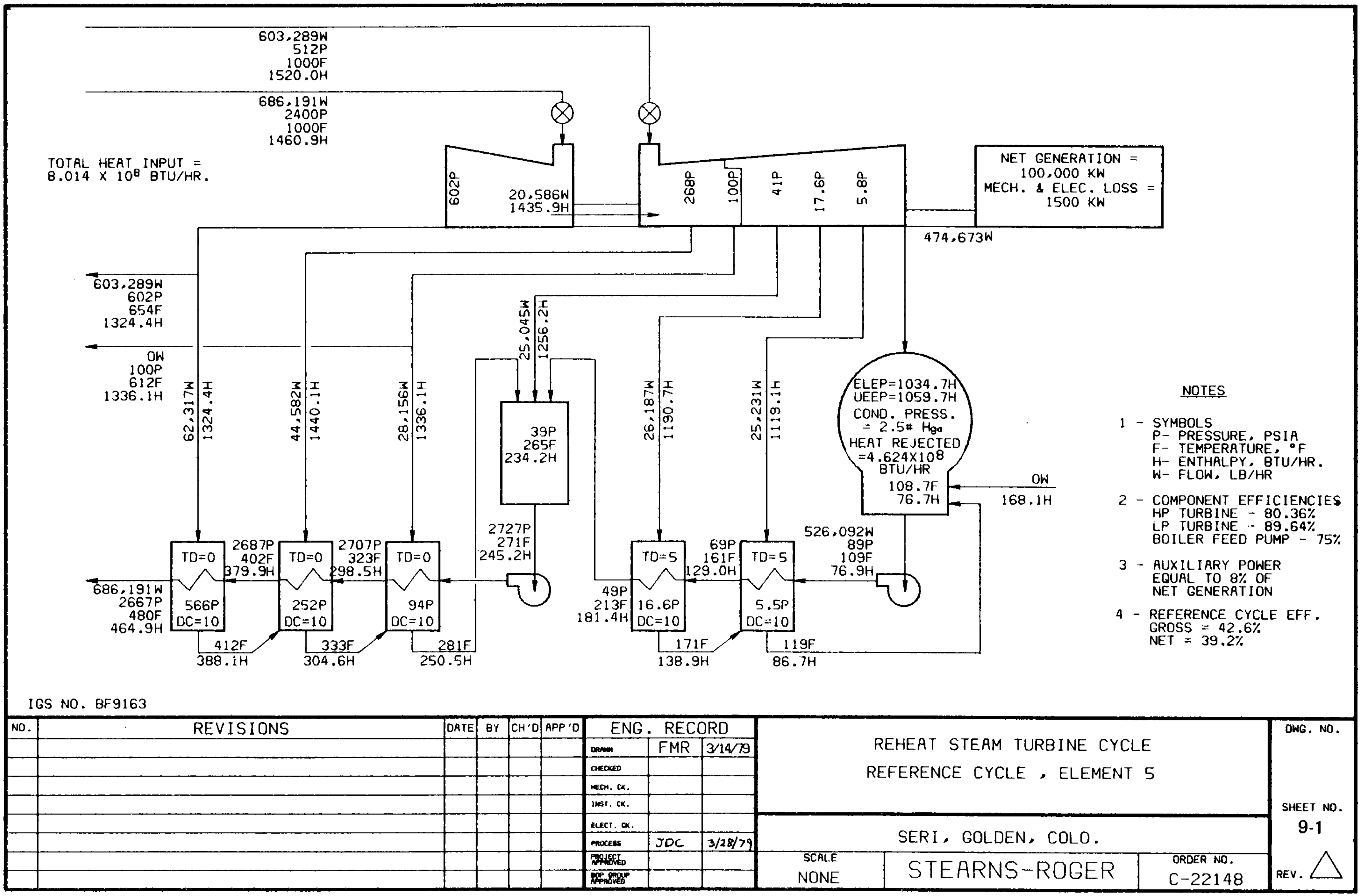


Figure 9–1. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, REFERENCE CYCLE

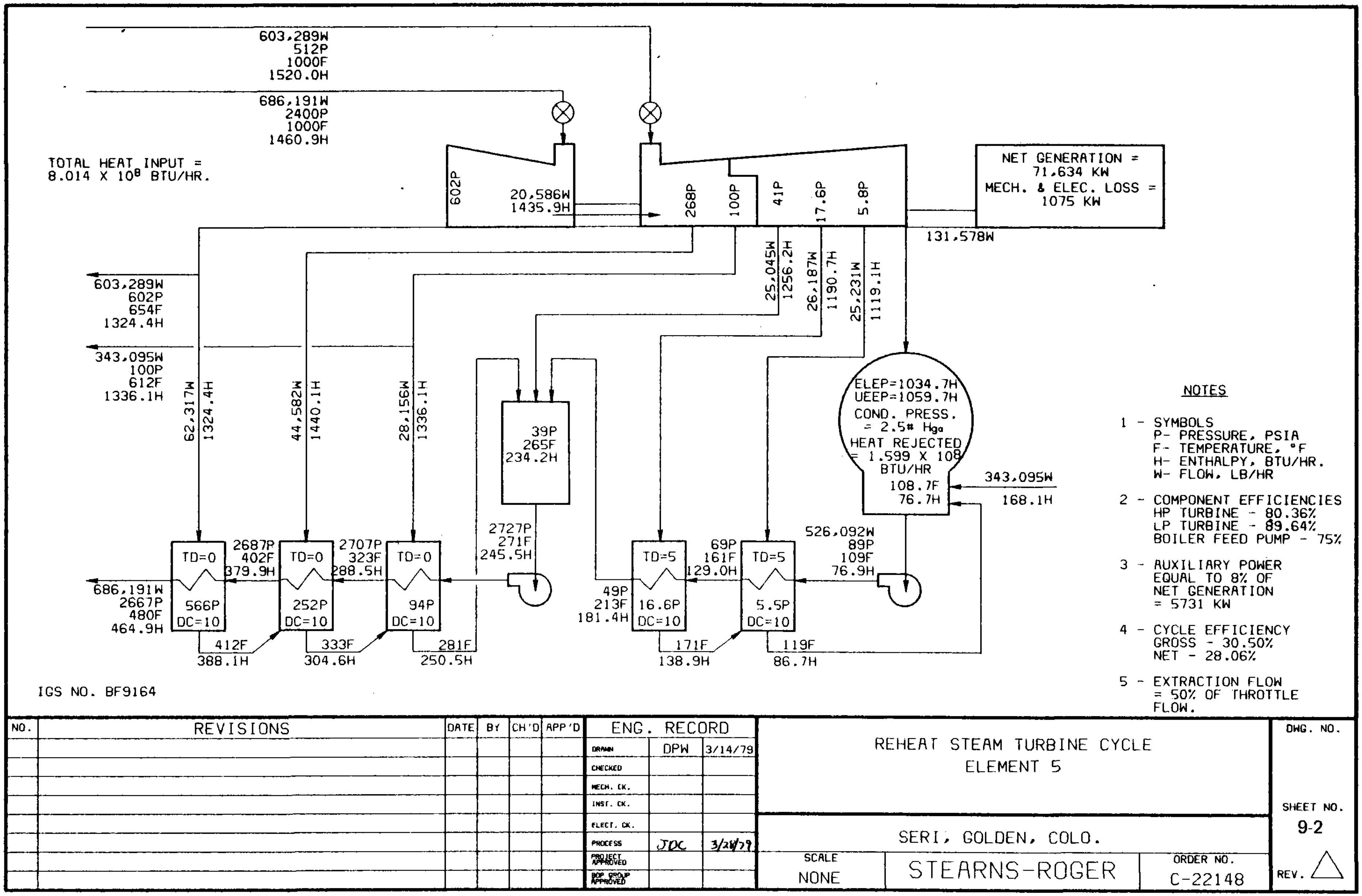


Figure 9–2. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE

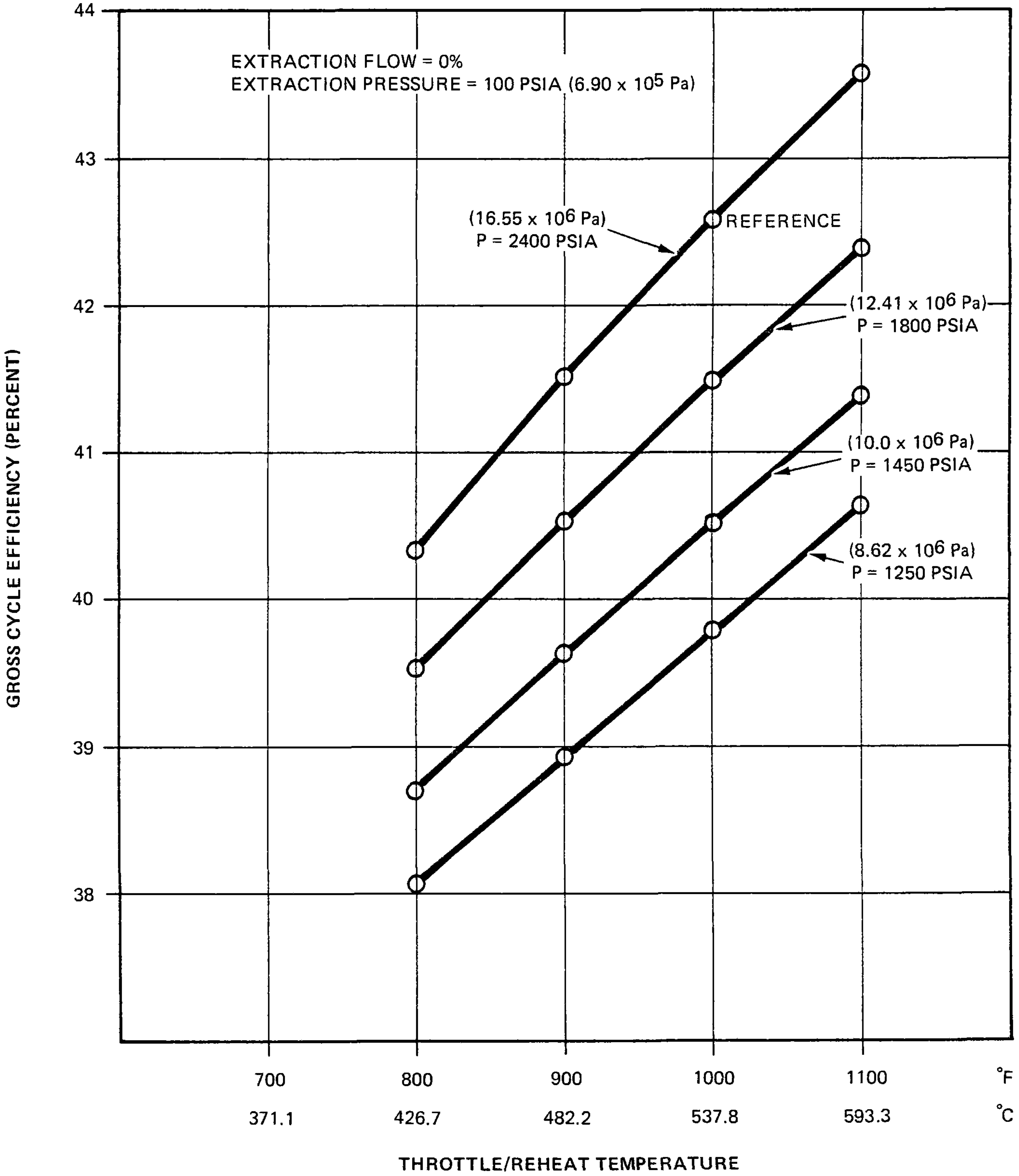


Figure 9—3. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

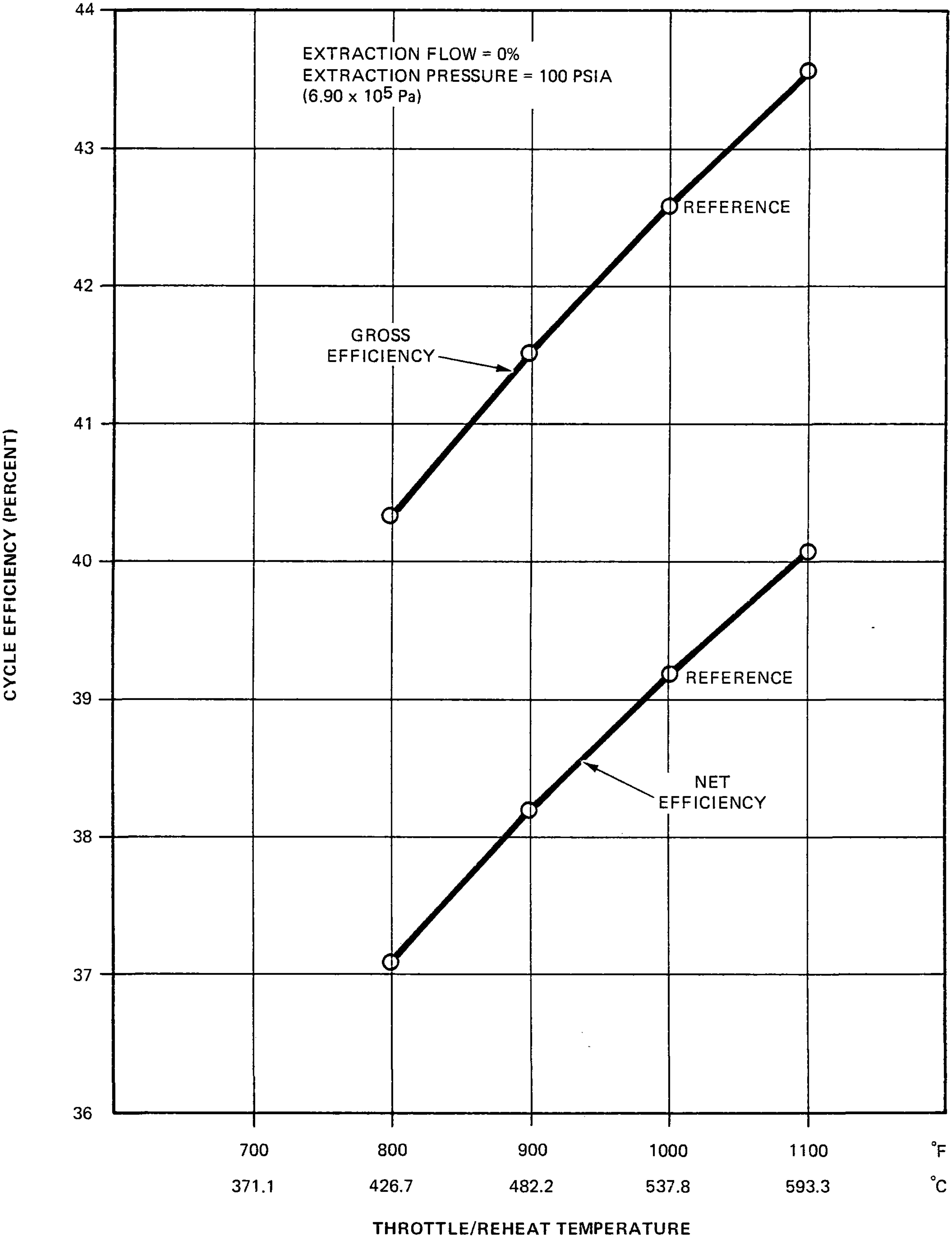


Figure 9–4. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

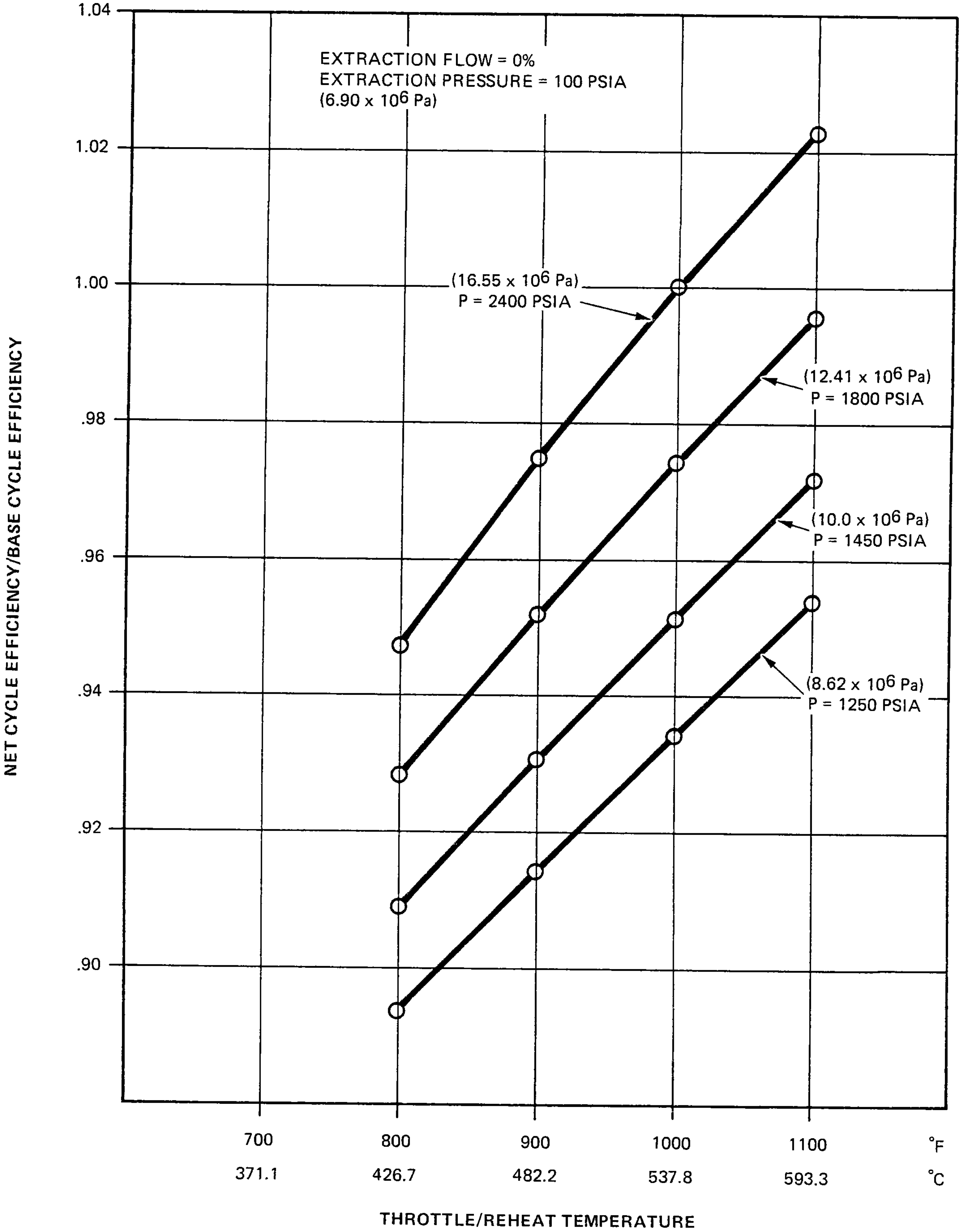


Figure 9–5. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

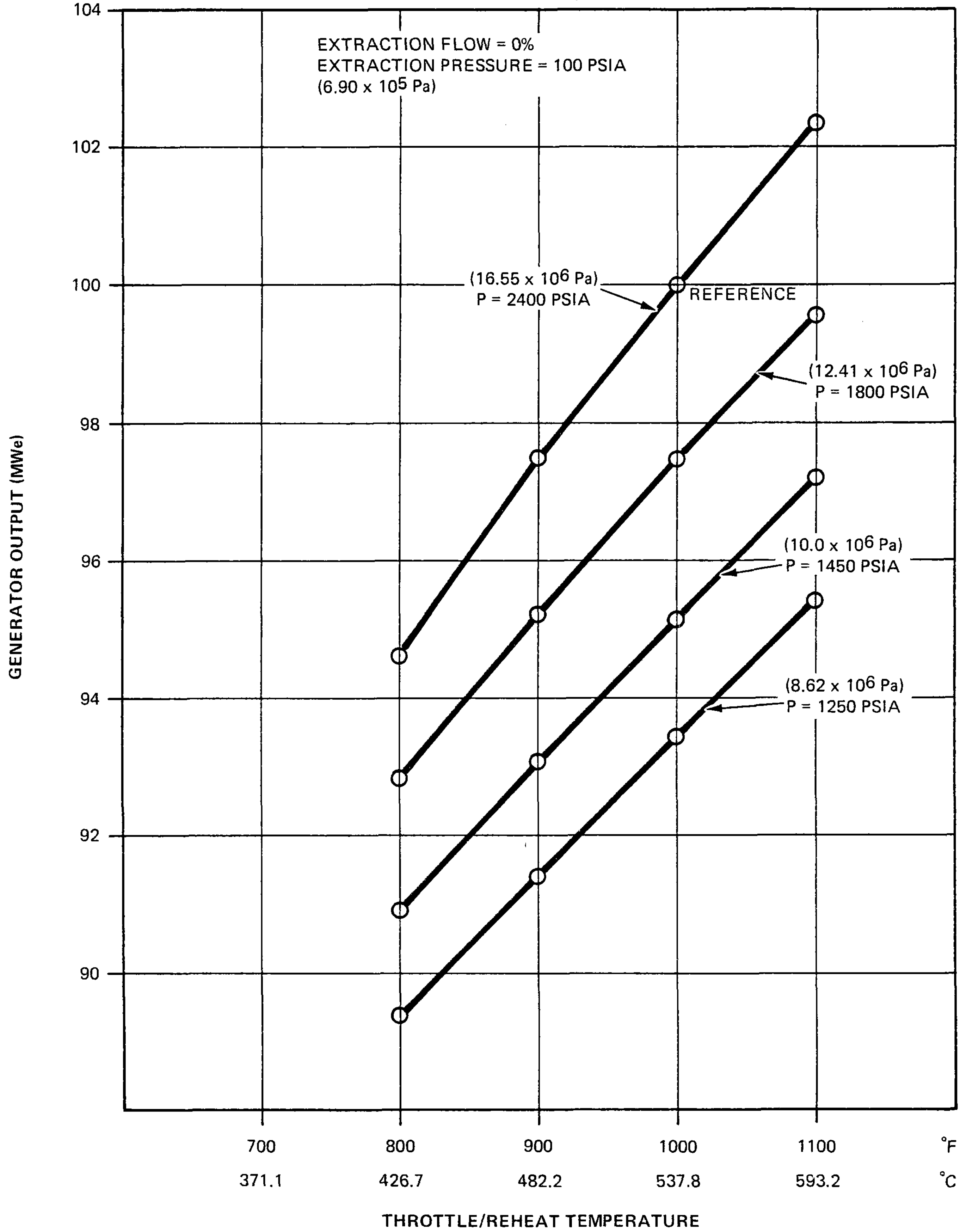


Figure 9–6. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

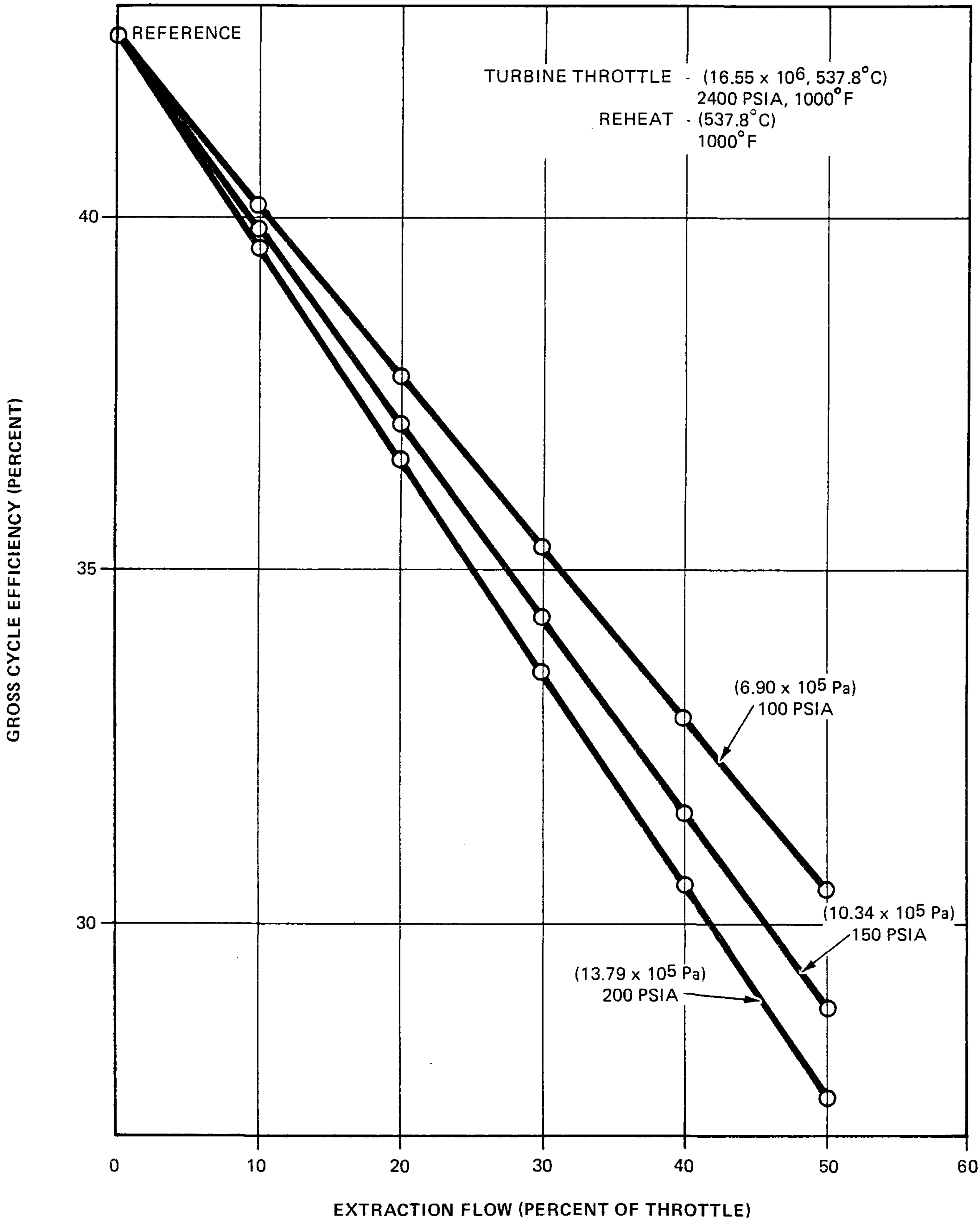


Figure 9–7. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. EXTRACTION FLOW

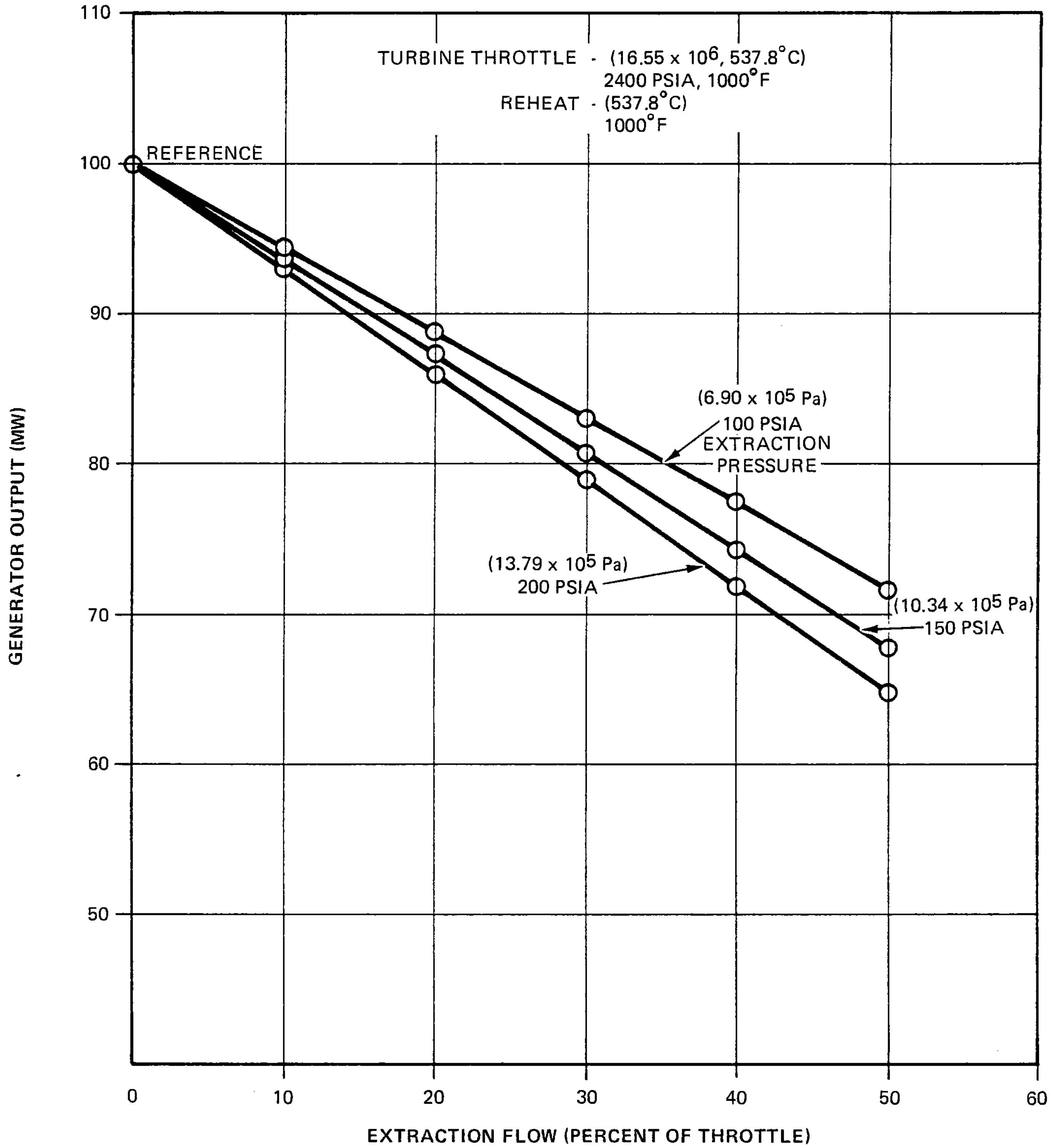


Figure 9–8. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. EXTRACTION FLOW

SECTION 10
ELEMENT 6A - STEAM RANKINE CYCLE

10.1 INTRODUCTION

Cycle heat balances were performed for a 10 MWe (nominal) non-reheat steam Rankine cycle employing four stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.46 x 10^3 Pa).

The cycle was analyzed using an in-house computer program (D135A), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 of Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

10.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 1250 psia (8.62 x 10^6 Pa) and 950°F (510°C). The final feedwater temperature is 400°F (204.4°C), which allows a reasonable pressure ratio from the throttle to the first extraction point. A 10 percent pressure loss is assumed across the steam generator.

For this study, the total heat input to the cycle was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 700°F to 1100°F (371.1°C to 593.3°C), and the throttle pressure was varied over the range of 850 psia to 1800 psia (5.86 x 10^6 Pa to 12.41 x 10^6 Pa). The pressures selected are those which are normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower limit of temperature varies with the throttle pressure, as it is desirable to maintain the turbine exhaust steam quality at a value greater than 84 percent. For the 1800 psia (12.41 x 10^6 Pa) pressure, the minimum temperature is 825°F (440.6°C); for the 1450 psia (10.0 x 10^6 Pa) pressure, the minimum temperature is 775°F (412.8°C); for the 1250 psia (8.62 x 10^6 Pa) pressure, the minimum temperature is 725°F (385°C); and for the 850 psia (5.86 x 10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C). A representative turbine efficiency was calculated using the method specified in Reference 1. The turbine efficiency was based on the throttle conditions of the reference cycle. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbine will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to two high-pressure closed feedwater heaters, an open deaerating feedwater heater, and a low-pressure closed feedwater heater. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows.

These leakages include shaft leakage from the high pressure end of the turbine, seal steam flow to the low pressure end of the turbine, and throttle stop and control valve packing leakage. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 10–1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 10–2 and Figure 10–3. A normalized representation of cycle efficiency with respect to the base cycle efficiency versus throttle temperature is shown in Figure 10–4. A plot of generator output versus throttle temperature is shown in Figure 10–5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

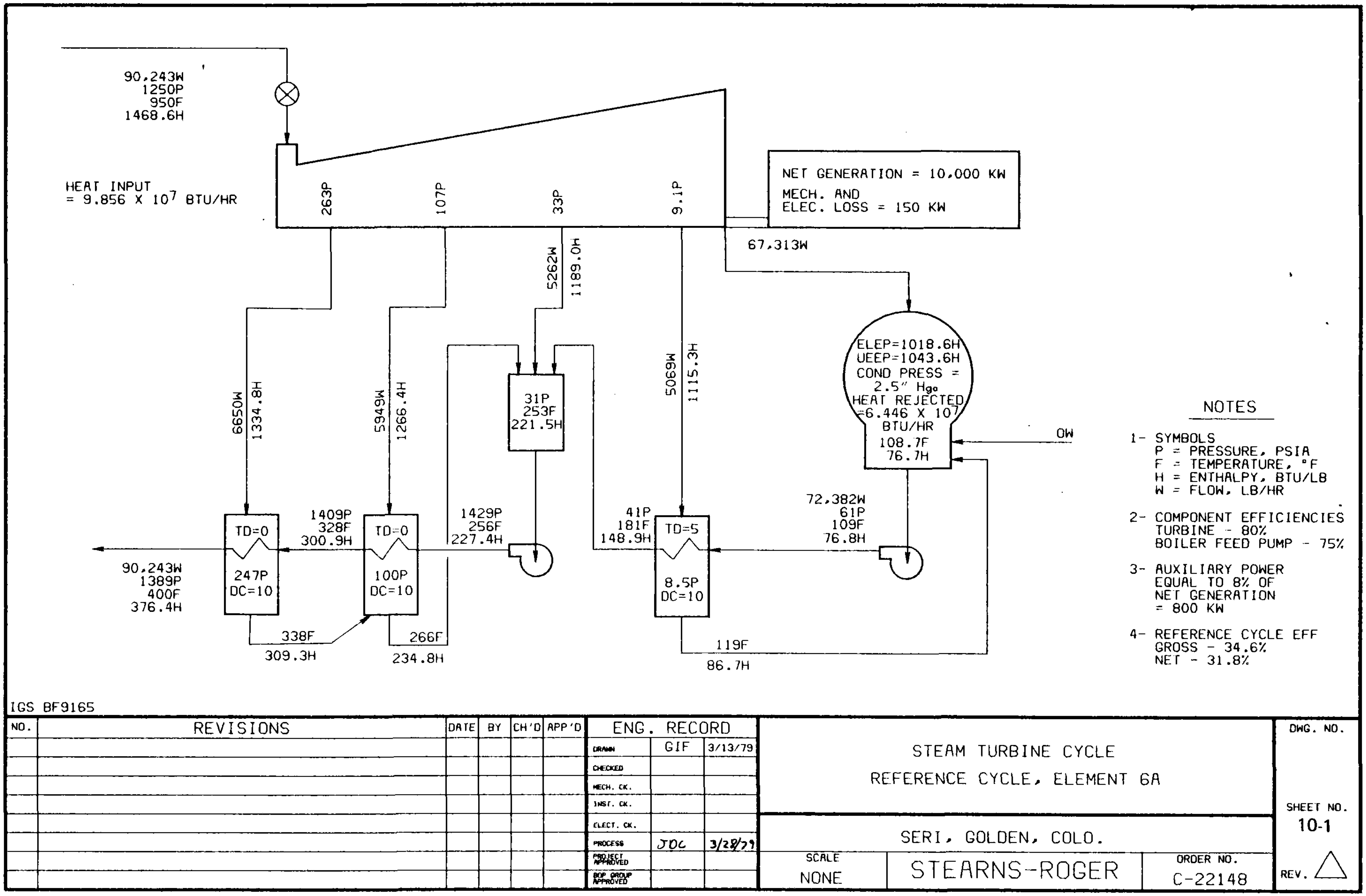


Figure 10–1. ELEMENT 6A - STEAM RANKINE CYCLE, REFERENCE CYCLE

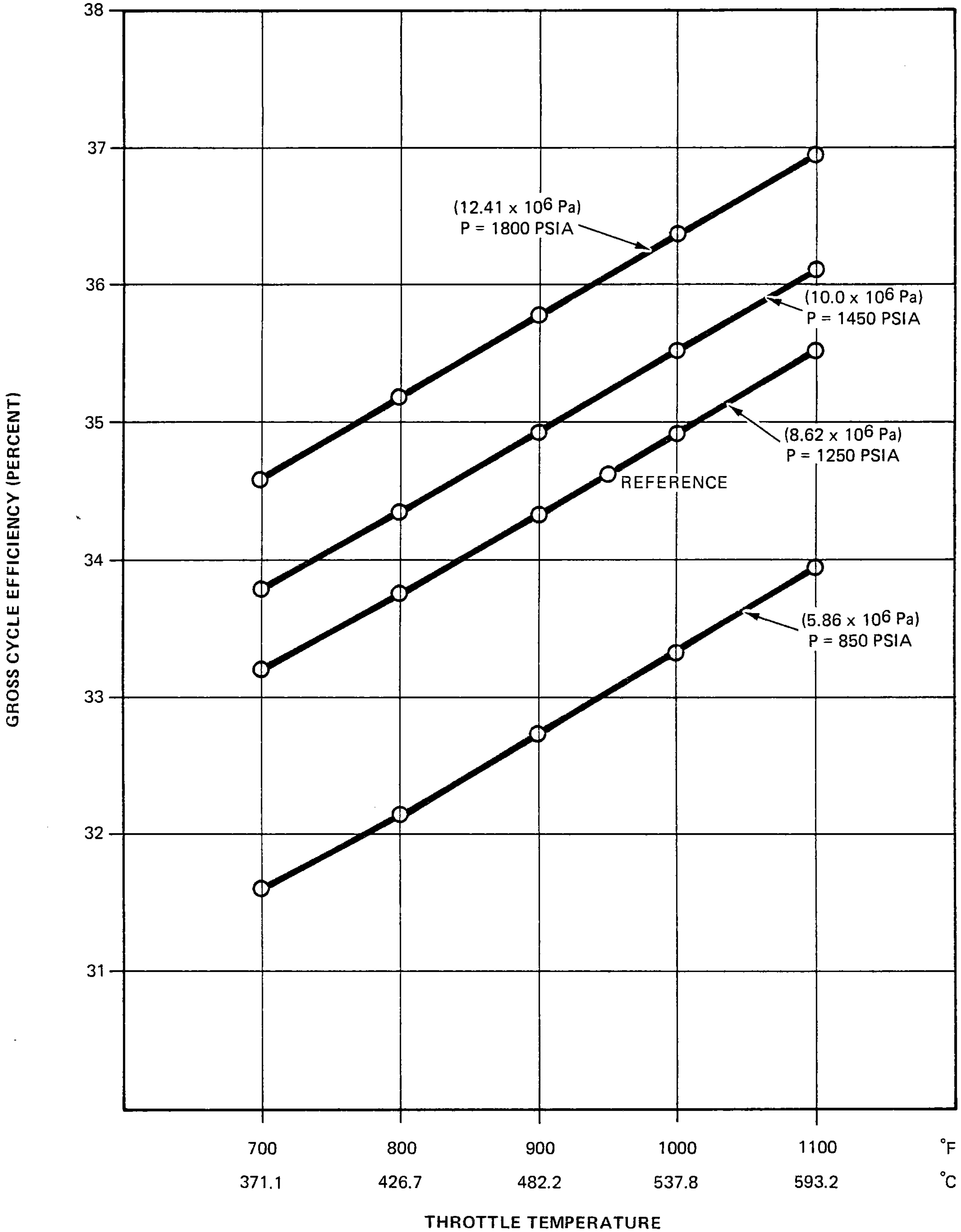


Figure 10–2. ELEMENT 6A - STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

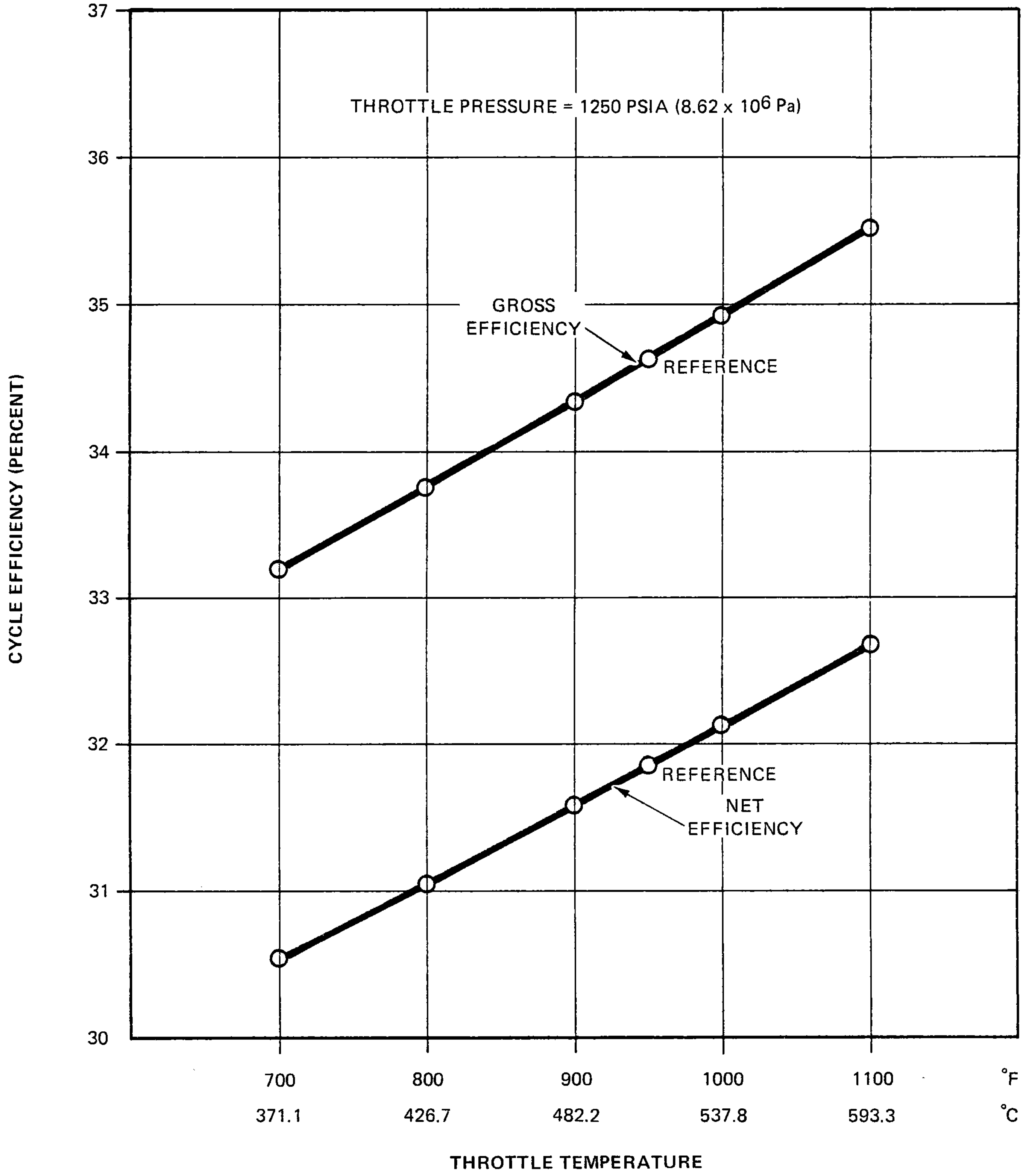


Figure 10–3. ELEMENT 6A - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

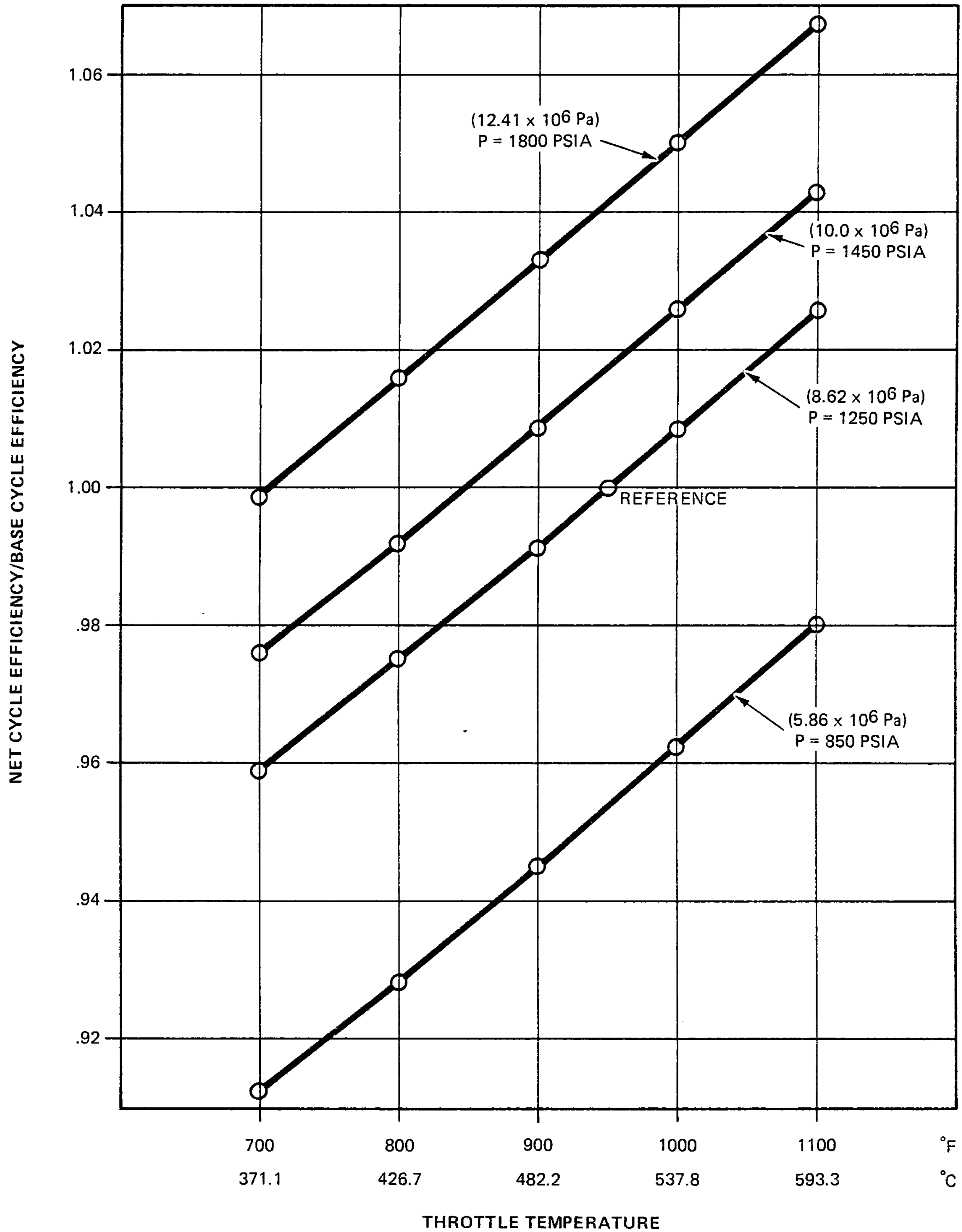


Figure 10–4. ELEMENT 6A - STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

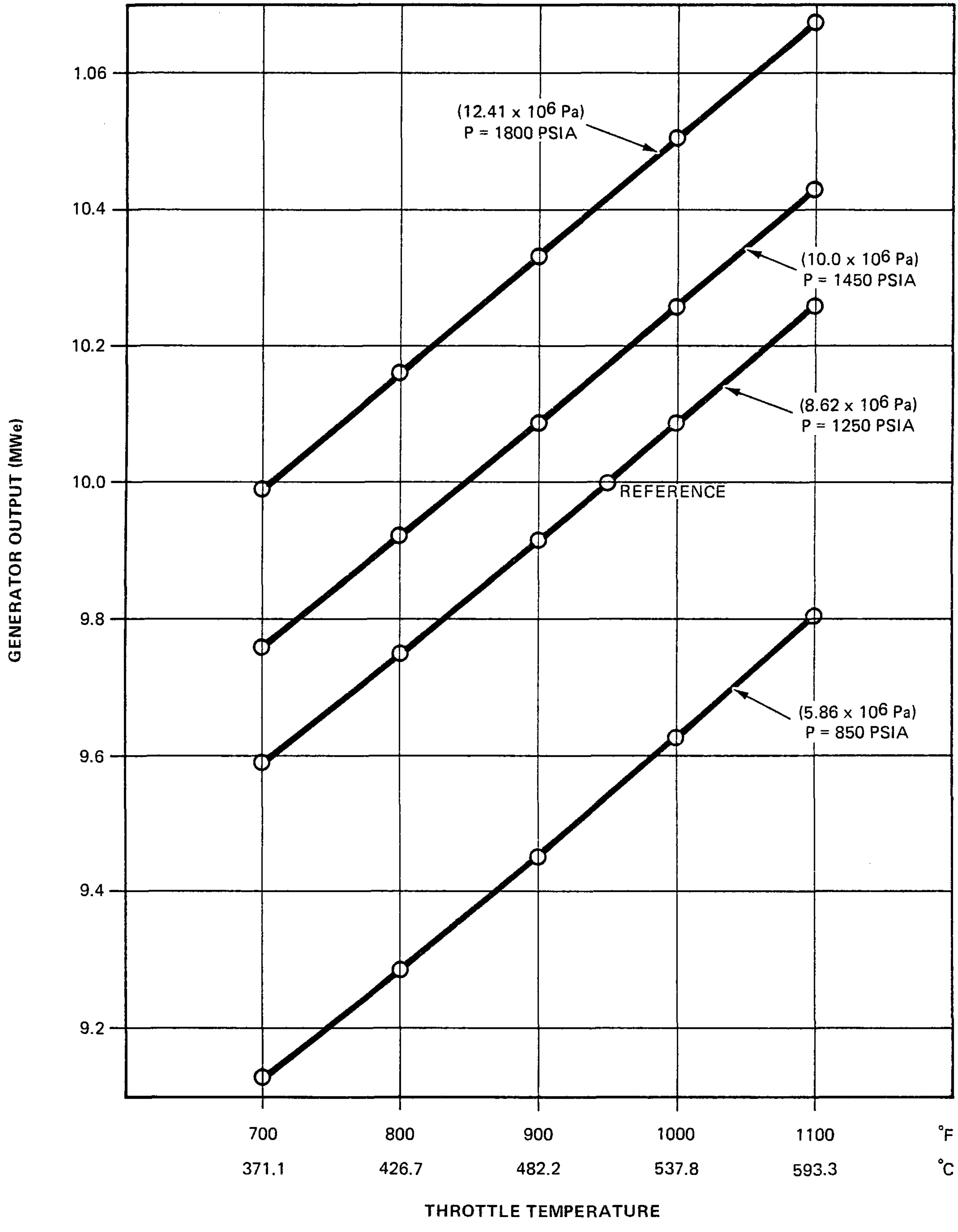


Figure 10—5. ELEMENT 6A - STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 11
ELEMENT 6B - STEAM RANKINE CYCLE

11.1 INTRODUCTION

Cycle heat balances were performed for a 300 KWe (nominal) non-reheat steam Rankine cycle employing two stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.46×10^3 Pa).

The cycle was analyzed using an in-house computer program (D135A), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 of Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

11.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 1250 psia (8.62×10^6 Pa) and 950°F (510°C). The final feedwater temperature is 400°F (204.4°C), which allows a reasonable pressure ratio from the throttle to the first extraction point. A 10 percent pressure loss is assumed across the steam generator.

For this study, the total heat input to the cycle was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 700°F to 1100°F (371.1°C to 593.3°C), and the throttle pressure was varied over the range of 850 psia to 1800 psia (5.86×10^6 Pa to 12.41×10^6 Pa). The pressures selected are those which are normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower limit of temperature varies with the throttle pressure, as it is desirable to maintain the turbine exhaust steam quality at a value greater than 84 percent. For the 1800 psia (12.41×10^6 Pa) pressure, the minimum temperature is 825°F (440.6°C); for the 1450 psia (10.0×10^6 Pa) pressure, the minimum temperature is 725°F (385°C); and for the 850 psia (5.86×10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C). A representative turbine efficiency was calculated using the method specified in Reference 1. The turbine efficiency was based on the throttle conditions of the reference cycle. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbine will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to one high-pressure closed feedwater heater and one open deaerating feedwater heater. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. These leakages include shaft leakage from the high pressure end of the turbine, seal steam flow to the low pressure end of the turbine, and throttle stop and control valve packing leakage. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 11—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 11—2 and Figure 11—3. A normalized representation of cycle efficiency with respect to the base cycle efficiency versus throttle temperature is shown in Figure 11—4. A plot of generator output versus throttle temperature is shown in Figure 11—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the receiver feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

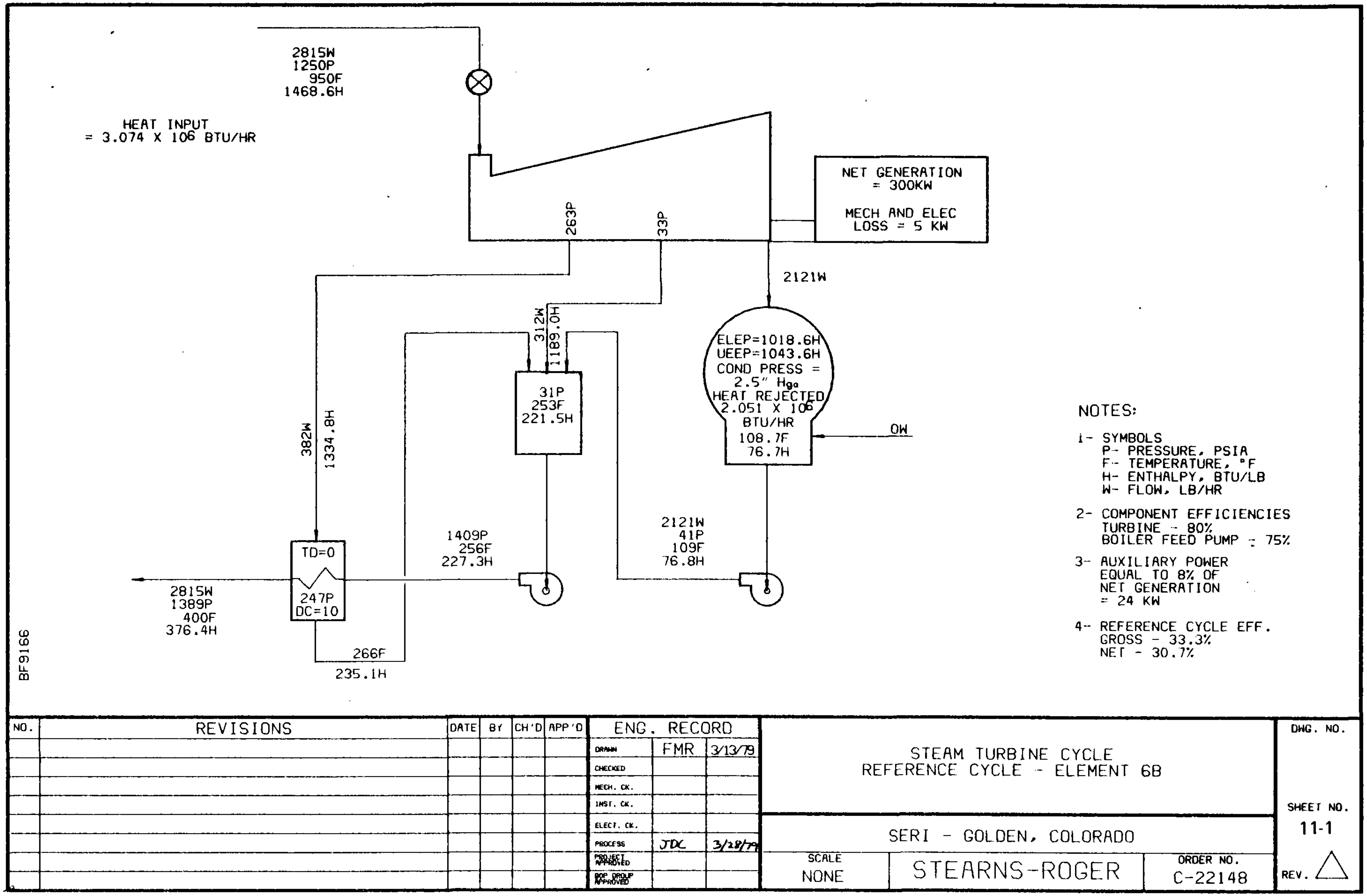


Figure 11—1. ELEMENT 6B - STEAM RANKINE CYCLE, REFERENCE CYCLE

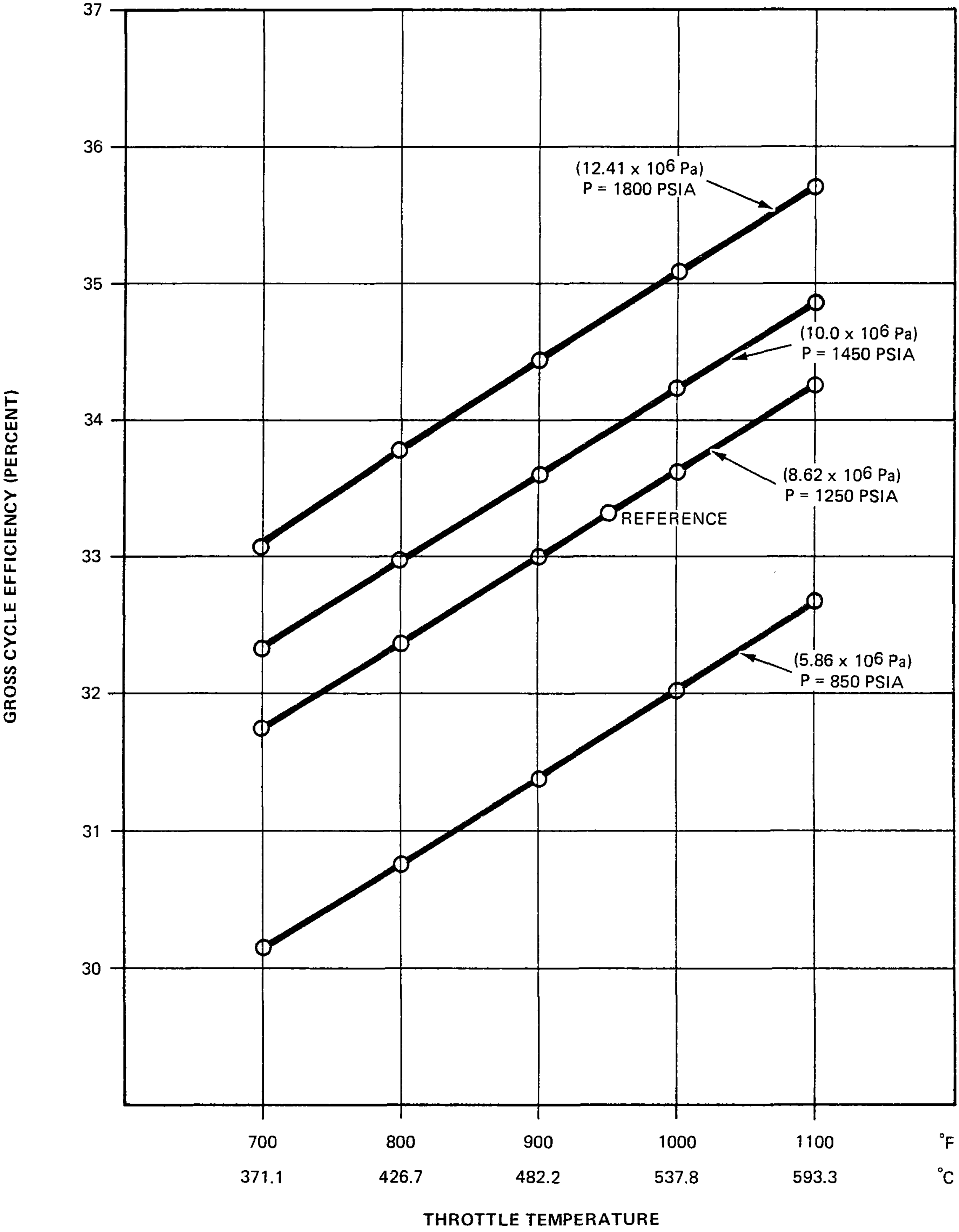


Figure 11–2. ELEMENT 6B - STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

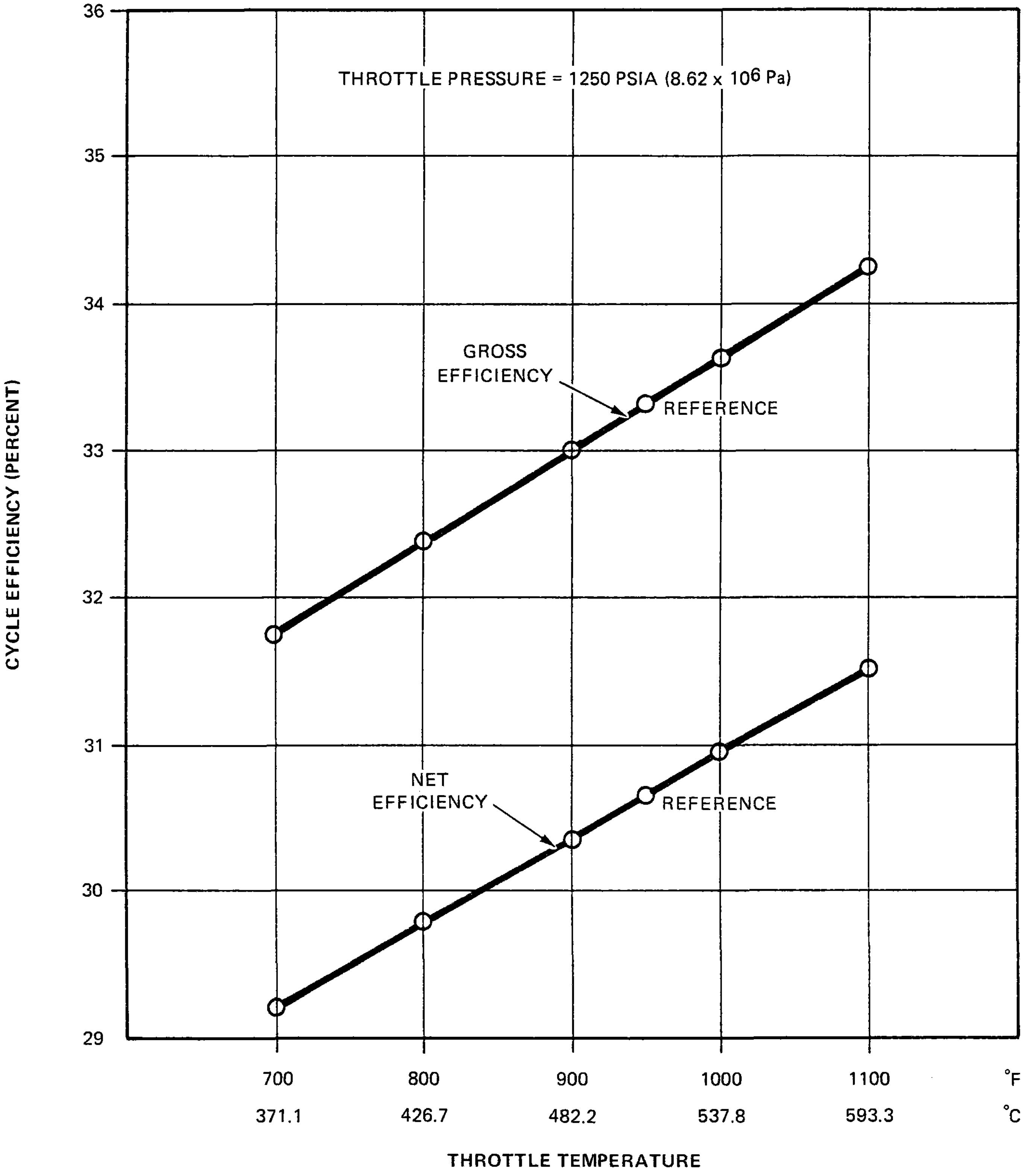


Figure 11–3. ELEMENT 6B - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

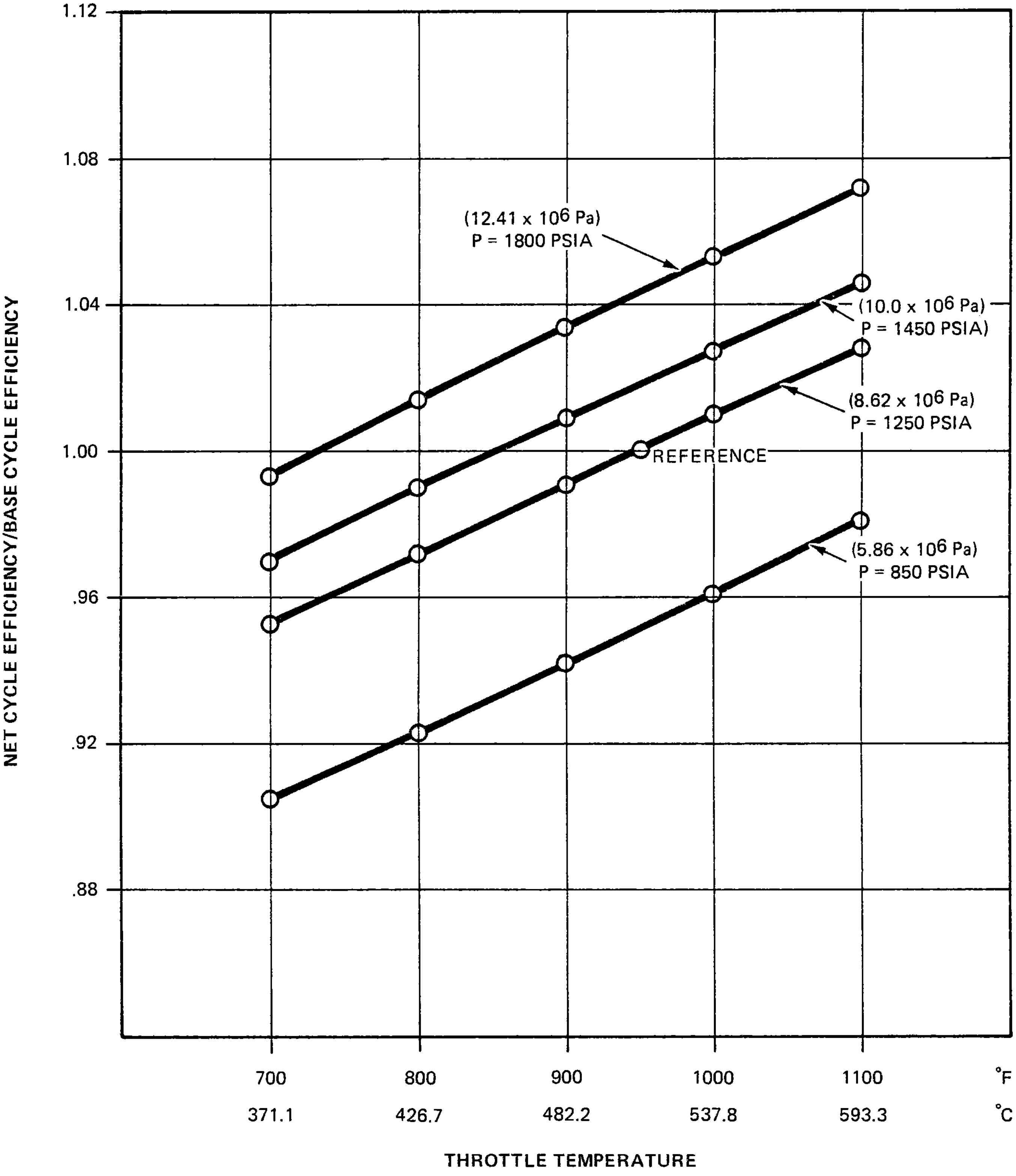


Figure 11–4. ELEMENT 6B - STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

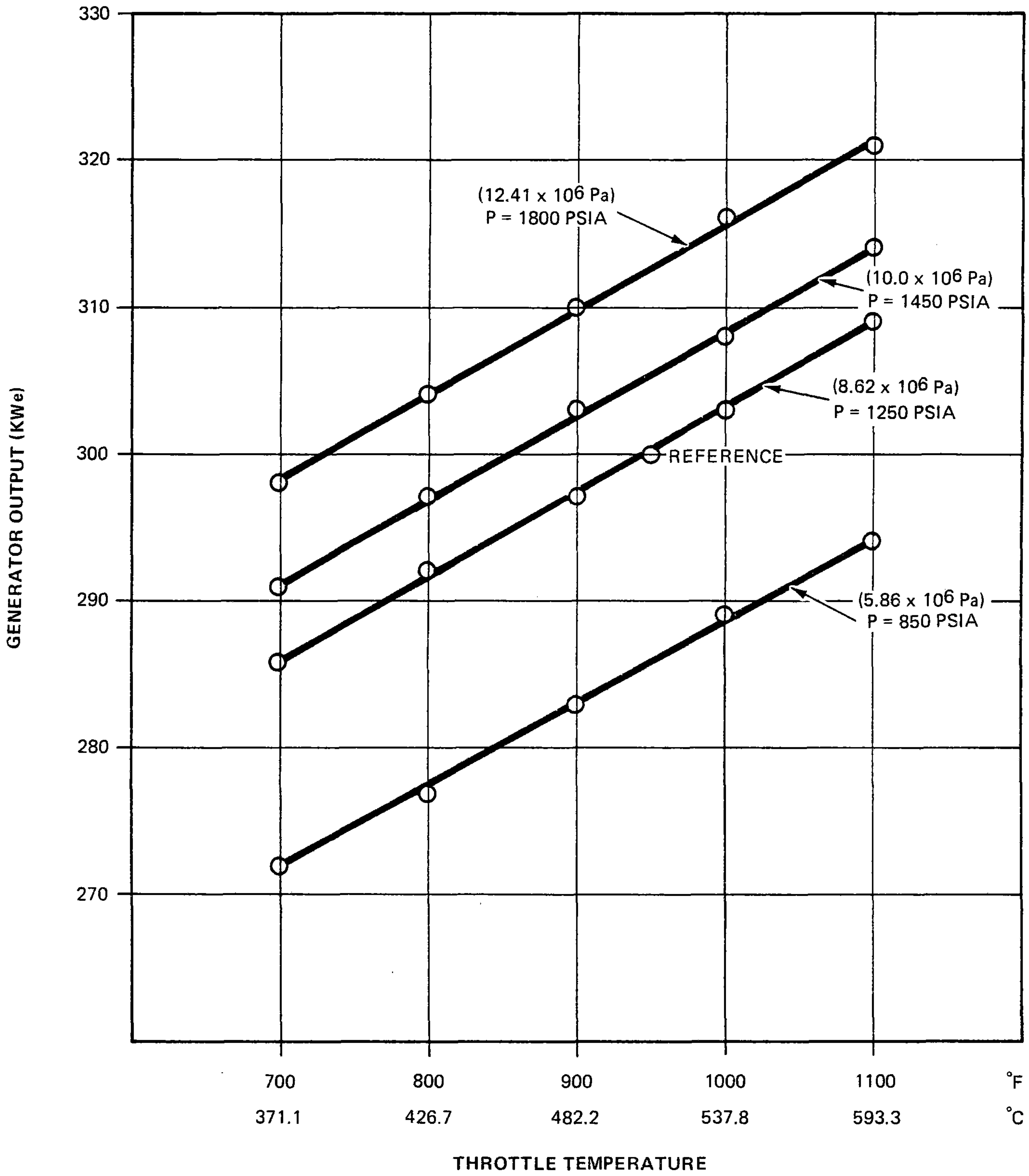


Figure 11–5. ELEMENT 6B - STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 12
APPENDIX

CALCULATION PROCEDURE FOR STEAM RANKINE CYCLES

Several computer programs were used to generate the heat balances for the various steam Rankine cycles evaluated in this study. Where possible, existing in-house programs were used to reduce the total manhour requirement. With cycles that could not be evaluated using existing programs, additional programs were generated to meet the requirements of the study. The input values, basic calculation procedure, and output results of all programs were similar with only minor changes in the physical configuration of the cycle.

The input generally consisted of desired throttle conditions, turbine efficiency, condenser pressure, performance criteria for the feedwater heaters, final feedwater temperature, pump efficiency, system mechanical and electrical losses, and required electrical output or specified throttle flow. Additional input would depend on the specific cycle.

The calculation procedure usually began with an input or assumed throttle flow value and calculated a value for the power generated. If a specific generator output was required the value of the throttle flow was modified and the required output value was obtained using a convergence technique. The procedure used to arrive at the power output for a given throttle flow was as follows:

1. A final feed flow was calculated from the throttle flow and any other boiler flows.

2. From the performance characteristics of the top heater the saturation pressure in the heater shell was determined. Using a specified pressure loss for the extraction piping, the pressure at the turbine extraction port was found. Knowing the shape and orientation of the turbine expansion line, based on the turbine efficiency, an enthalpy for the steam at that extraction point was calculated. Finally, knowing the feedwater flow, the feedwater heater performance, and the enthalpy of the extraction steam, the flow of extraction steam was calculated.

3. The above procedure was repeated for each heater.

4. The turbine exhaust conditions were calculated from the turbine efficiency and specified exhaust losses.

5. The turbine shaft power produced was found by completing an energy balance of all flows into and out of the turbine.

The output of the programs included a restatement of input data, state conditions and flows for all major components, power generated, and heat rate or thermal efficiency values.

A sample of the output of one in-house program is included on the following page.

SERI - CYCLE 6A - BASE CASE - 1250 PSIA, 950 F

GENERATCR

GENERATOR OUTPUT = 10000. KW   MECH. AND ELEC. LOSS = 150. KW

CYCLE PERFCRMANCE

HEAT RATE = (90243.(1468.6-376.4)/ 10000.) = 9856.2 BTU/KWH
BOILER DUTY = 98563376. BTU/HR   BOILER BLOWDOWN = 0.0 PERCENT
CONDENSER DUTY = 64461664. BTU/HR   CONDENSER PRESSURE = 2.5 IN HGA

	FLOW LB/HR	PRESS. PSIA	TEMP. DEG F	ENTHALPY BTU/LB	QUALITY
TURBINE THROTTLE	90243.	1250.00	950.00	1468.63	
TURBINE EXHAUST	67313.	1.23	108.70	1018.57	0.91
MAKEUP TO CONDENSER	0.				
HOTWELL	72382.	1.23	108.70	76.69	

TURBINE EFFICIENCY = 80.00 PERCENT
TURBINE FIRST EXTRACTION PRESSURE RATIO = 4.75
BOILER FEEDPUMP EFFICIENCY = 75.00 PERCENT
BOILER FEEDPUMP POWER REQUIREMENT = 155. KW / 208. HP

HEATER NO.	1	2	3	4
TTD	0.	0.	0.	5.
DC	10.	10.	0.	10.
STAGE PRESS	263.04	106.51	33.36	9.07
LINE PRESS LOSS	15.78	6.39	2.00	0.54
EXT FLOW	6650.	5949.	5262.	5069.
EXT ENTHALPY	1334.79	1266.38	1189.00	1115.26
HTR SHL PRESS	247.26	100.12	31.36	8.53
HTR SHL TEMP	400.00	327.91	252.88	185.79
HTR SHL ENTHALPY	375.10	298.63	221.52	153.81
FW IN FLOW	90243.	90243.	72382.	72382.
FW IN PRESS	1408.89	1428.89	41.36	61.36
FW IN TEMP	327.91	255.82	180.79	108.70
FW IN ENTHALPY	300.87	227.38	148.87	76.84
FW OUT FLOW	90243.	90243.	90243.	72382.
FW OUT PRESS	1388.89	1408.89	31.36	41.36
FW OUT TEMP	400.00	327.91	252.88	180.79
FW OUT ENTHALPY	376.44	300.87	221.52	148.87
DR IN FLOW	0.	6650.	12599.	0.
DR IN TEMP	0.0	337.91	265.82	0.0
DR IN ENTHALPY	0.0	309.29	234.81	0.0
DR OUT FLOW	6650.	12599.	0.	5069.
DR OUT TEMP	337.91	265.82	0.0	118.70
DR OUT ENTHALPY	309.29	234.81	0.0	86.68