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THE WISC GAS TURBINE ENGINE

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ABSTRACT

Combined gas turbine-steam turbine cycles have gained widespread acceptance as the most efficient utilization of the gas turbine for power generation, particularly for large power plants. In order to maximize the achievable thermal efficiency, more than one exhaust heat recovery boiler is used. The current trend is to use three boilers at three different operating pressures, which improves thermal efficiency but significantly increases the initial cost of the plant.

There are advantages in replacing an exhaust heat recovery system using multiple boilers by a single heat exchanger in which the water side pressure is above the critical pressure of water; we shall refer to such a heat exchanger as a supercritical heat exchanger. The supercritical steam leaving the heat exchanger is expanded in a two phase turbine and then fed into the engine combustor. A condenser and a water treatment system are used to recover most of the water in the exhaust stream. A turbine system identical to the basic engine turbine system is added in parallel in order to allow for the operation with increased mass flow due to the steam injection. To achieve maximum efficiency such a turbine should be provided with variable area nozzles. With this arrangement, it becomes possible to inject sufficient steam to produce stoichiometric combustion at the desired turbine inlet temperature. We shall refer to this cycle as the Water Injected Stoichiometric Combustion (WISC) gas turbine cycle. The various components described above can be added to any existing gas turbine engine to change it to the WISC configuration.

The WISC engine offers significant economical advantages. The specific power output per pound of air for the WISC engine is more than five times that of the basic engine, the thermal efficiency is 75% higher than that of the basic engine. This produces a significant reduction in the initial investment in the plant as well as its operating expenses.

NOMENCLATURE

T_H	Heat source temperature
T_L	Heat rejection temperature
WISC	Water Injected Stoichiometric Combustion
η_{th}	Thermal efficiency

INTRODUCTION

Research and development efforts to improve the performance of gas turbines for the last fifty years have been mainly directed toward increasing the turbine inlet temperature. This effort has produced innovative blade cooling technology and more demand for research to produce ceramic materials that can withstand higher temperatures. Today the use of the combined cycle for power generation is coupled with a high turbine inlet temperature. However, the effect of the combined cycle alone has a greater influence than raising the turbine inlet temperature. In order to make this point clear, we shall consider a Carnot engine with the thermal efficiency

$$\eta_{th} = \frac{T_H - T_L}{T_L} \tag{1}$$

The marginal effect of raising T_H is

$$\frac{\partial \eta_{th}}{\partial T_H} = \frac{1}{T_L} \tag{2}$$

On the other hand, the marginal effect of lowering T_L is

$$-\frac{\partial \eta_{th}}{\partial T_L} = \frac{T_H}{T_L^2} \tag{3}$$

Lowering the heat rejection temperature is more effective than raising the maximum temperature by a factor of T_H/T_L, which suggests that lowering the heat

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rejection temperature is likely to be a more cost effective way of raising efficiency than is raising turbine inlet temperature. Currently, the most popular way to do this is to incorporate an additional bottomming Rankine cycle using water as the working fluid, but this is neither the only nor necessarily the best way.

Advanced gas turbine cycles incorporating such features as steam injection, intercooling, and regeneration, while promising high efficiency, have not gained the widespread acceptance of the combined cycle (Boland and Stadaas 1995). The combined cycle remains popular because it is economical; an alternative cycle must prove itself more economical even when the costs of development are taken into account.

The objective of this paper is to present a cycle, the Water Injected Stoichiometric Combustion cycle (WISC) which the authors believe is more attractive economically than the combined cycle.

The WISC idea was conceived as a minimal development cost route to an advanced gas turbine cycle after a meeting with the US Navy regarding a patent of Patton and Shouman (1989) for a supercritical steam-injected gas turbine engine. Shouman and Shouman (1995) undertook the feasibility study and characterization of the WISC engine, and differentiate it from the steam-augmented gas turbine (SAGT) technology investigated by Urbach et al. (1993).

Almost every gas turbine manufacturer is using steam injection for power augmentation and NO_x control. Rice (1995) discusses the effect of steam injection on gas turbine performance, presenting a schematic diagram of the high pressure section of the steam-injected General Electric STIG LM 5000 engine. The steam injected to the turbine is 6.5% of the air mass flow, increasing power output from 34 MW to 52 MW. The turbine nozzle areas had to be increased and the power turbine redesigned in order to accept the increased mass flow and expansion ratio necessary for this power output.

The approach described by Rice is typical of that taken by other manufacturers. The amount of steam injected into the combustion zone of the engine is limited by the need to keep the compressor operating point at a sufficient distance from the surge line to allow stable operation. The turbine nozzle areas are changed to accommodate the change in mass flow. Steam addition to the combustion system has a positive effect on efficiency, power output, and NO_x emissions.

Cheng (1976) originally suggested that a gas turbine power plant be operated with enough steam injection to reach the stoichiometric limit. The original demonstration of the Cheng technology in California was the only means for meeting the EPA emissions standards. Although the technology showed improvements in power output and thermal efficiency relative to non-steam-injected plants, it fell short of stoichiometric combustion because the steam injection rate was limited by the surge limit of the compressor.

Urbach et al. (1993) et al studied the feasibility of steam-augmented gas turbine engine technology for Navy ships. They recommended the use of steam until the stoichiometric combustion is reached and showed the advantages of such an engine. However, in their study they showed that such an engine would need a desalination plant for its operation.

THE WISC POWER PLANT

Almost any existing gas turbine power plant may be converted to the WISC configuration. The problem of the compressor operating limitation is solved by adding a second turbine similar to the engine turbine and in parallel to it. The second turbine may be identical to the original engine turbine, however it should preferably be of variable geometry in order to achieve the maximum possible efficiency. The second turbine may be mounted on the same shaft as the main turbine for a single shaft machine or on a separate shaft connected separately to its load. For the two shaft gas turbines, there will be two turbines in parallel, one equivalent to the compressor turbine and the second equivalent to the power turbine. The two turbines may be run independently or coupled together and with the load turbine to run on a single shaft.

In the WISC plant, the waste heat boiler is replaced by a supercritical heat exchanger where the water is supplied at a pressure above its critical presure in order to maximize the work recoverable in the steam cycle. The steam leaving the supercritical heat exchanger is expanded through a two-phase expansion turbine which exhausts into the combustor of the plant. This two phase expander may be run either coupled with the other turbines, or on a separate shaft. The authors are currently investigating ways of staging heat recovery exchangers so that a more conventional steam turbine may be used to expand some of the steam produced. Any such steam turbine will exhaust at the combustor pressure, so it will be small in comparison to the low pressure turbine necessary for a steam bottomming cycle.

The exhaust gases leaving the supercritical heat exchanger are fed into a water cooled condenser which cools the non-condensible gases and condenses most of the steam. The non-condensibles are vented to the atmosphere and the water discharged to the condenser

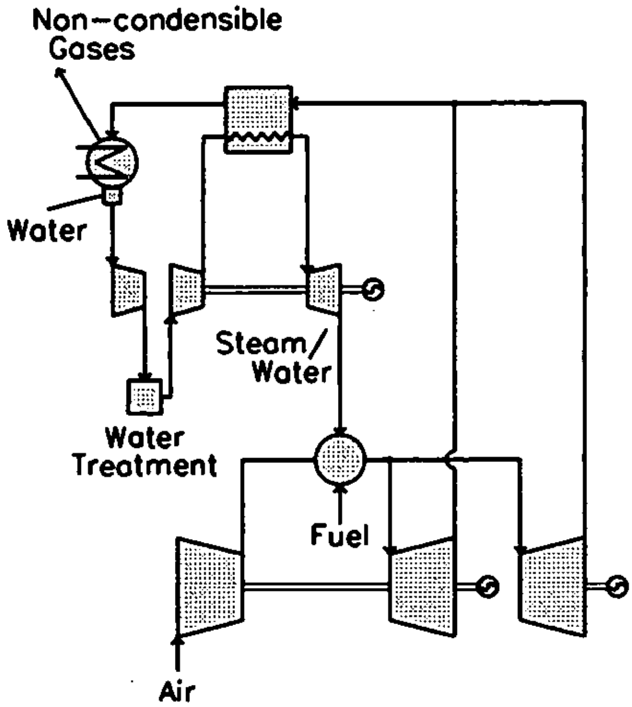

Figure 1: Typical WISC configuration

sump. A feed water treatment system is provided to treat the feed water before it is pumped back to the supercritical heat exchanger. Any excess water produced can be stored for use whenever make up water might be needed. The amount of steam entering the condenser will always be greater than that supplied to the combustor because of the water present in the combustion products. Make up water may still need to be supplied; the condenser should be sized to balance the costs of construction and the cost of gas pressure drop through the condenser against the cost of the make up water.

Figure 1 is a schematic diagram of a WISC plant constructed from a single shaft gas turbine engine – all the turbine shafts are shown as independent of each other. The gas turbine is intended to operate in its original mode until its original rated power is reached. At that point, water is pumped to the heat exchanger thus feeding steam to the combustor. The mass flow rate through the main turbine will slightly decrease because of the presence of the steam. However, the power output from the main turbine will increase. The excess mass flowing through the compressor is diverted to the secondary turbine using the variable geometry of the turbine to maintain the desired speed. If the turbine geometry is not variable, a throttling valve at the inlet or exhaust of the secondary turbine must be used to provide control. Naturally, using such a throttling valve entails a penalty in thermal efficiency.

Increasing the rate of water flow while adjusting the fuel flow to maintain constant turbine inlet temperature will increase the power output until the point of stoichiometric combustion is reached. Of course, other control strategies might be designed to tailor the system to the requirements imposed.

The two-phase expander is the component of the WISC system requiring significant development. Conventional steam turbines are not usable with a large fraction of condensate in the exhaust. An alternate expander design is illustrated in 2. Since the development of these expanders is not yet accomplished, the performance of a steam injected cycle without the two-phase expander and with a waste heat boiler operating at the combustor pressure was also evaluated. In the most of the following figures, this case is included for comparison with the full WISC configuration. Of course, there are other possibilities, for example, some fraction of the injection water could be superheated in a high temperature section of the heat exchanger and expanded through a conventional steam turbine, while the remainder would be heated to whatever temperature was available in a low temperature section of the heat exchanger. The authors are currently exploring this and other configuration options.

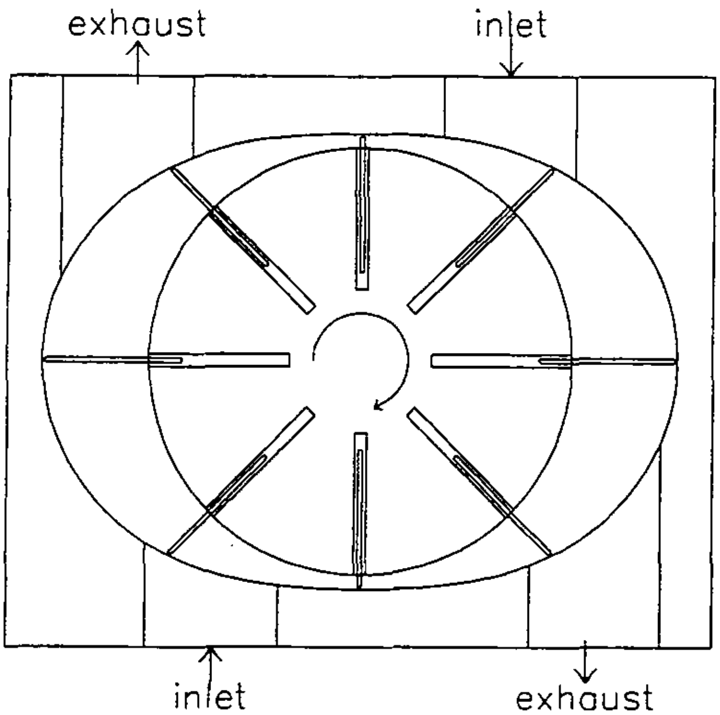

Figure 2: Cross section of a sliding-vane two-phase expander

APPLYING WISC TO THE STATE OF THE ART ENGINE

Boyle (1976) studied the effect of steam addition on cycle performance of simple and regenerated gas turbines. His calculations showed the same trend shown by Cheng. However, Boyle considered the effect of the pinch point on the boiler design, an effect which Cheng did not consider. Boyle assumed that a waste heat boiler would be used to recover as much energy as possible, and that a minimum $\Delta T$of 50°F (28°C) would be present between the hot and cold streams of the countercurrent heat exchanger. This minimum might occur either at the hot gas entrance or at a pinch point partway through the exchanger. He concluded that "the efficiency increases with increased steam addition until there is a shift in location of the minimum $\Delta T$for the boiler. this shift is from the steam boiler exit to the pinch point within the boiler. After the minimum ΔTshifts, there is a decrease in cycle efficiency with increased steam addition".

The authors have likewise found that the pinch point always limits heat transfer in the exhaust heat exchanger for sufficiently large steam injection rates. Even when the water in the heat exchanger is kept above the critical pressure, the shape of the enthalpy-temperature curve causes a pinch point.

The authors have developed software for the thermodynamic analysis of the WISC engine. In these calculations, non-condensible gases are assumed to be ideal, with heat capacity equations as given by Sonntag and Wylen (1971). The equation of state for steam was obtained from the paper of Young (1988). Agreement to within 1% for values of specific enthalpy with the tabulated values of the ASME steam tables was obtained with this equation. A correction using data from the steam tables was necessary for the properties of steam at supercritical pressure.

Turbines, compressors, and pumps were characterized by an assumed adiabatic efficiency, and countercurrent heat exchangers by a minimum ΔTbetween the two fluids. The system of equations resulting from material balances and the component efficiencies were solved to within small tolerances by Newton-Raphson iteration.

The case of a heat recovery boiler (as opposed to that of a supercritical heat exchanger and two phase expander) was analyzed first in order to compare results with those available in the literature, for example Boyle (1976), as well as to document the effect of the changes associated with the WISC configuration.

Table 1 summarizes environmental and engine parameters assumed for the purposes of this study, they are the same as those assumed by Boyle.

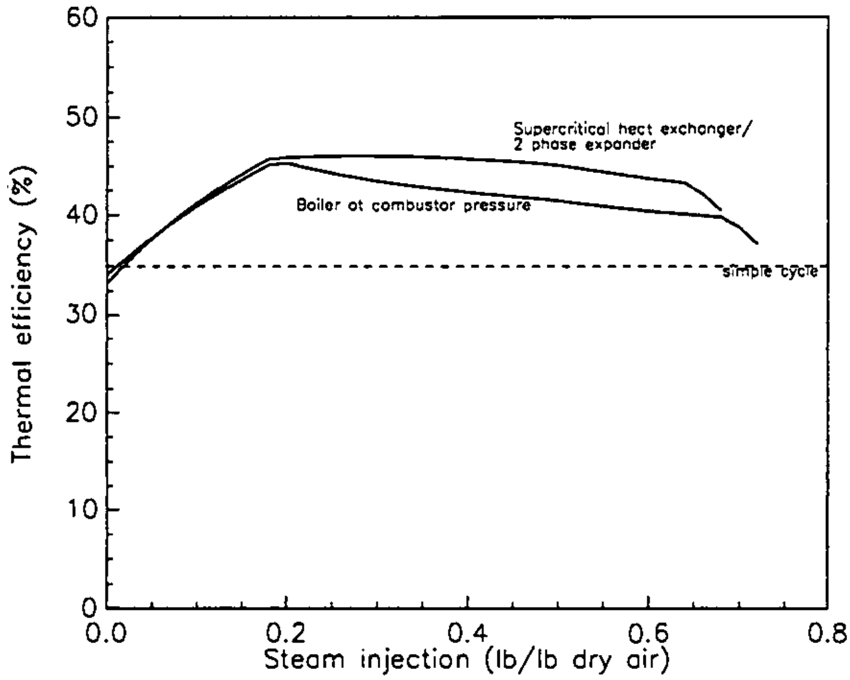

Figure 3: Efficiency as a function of water injection

Table 1: Basic Engine Parameters

Inlet pressure	14.7 psia (1.013 MPa)
Inlet temperature	60 °F (16 °C)
Inlet relative humidity	60%
Pressure ratio	16:1
Turbine inlet temperature	2000 °F (1094 °C)
Turbine adiabatic efficiency	87%
Compressor adiabatic efficiency	87%
Two-phase expander adiabatic efficiency	87%
Pressure of supercritical heat exchanger	5000 psia (344.6 MPa)
Pump adiabatic efficiency	80%
Heat exchanger pressure drop	4%
Condenser Pressure Drop	4%
Minimum heat exchanger ΔT	50°F (28 °C)
Fuel lower heating value	18600 Btu/lb_m (43200 Kj/Kg)

RESULTS AND DISCUSSION

Figure 3 shows the effect of steam injection on the thermal efficiency for an engine with a heat recovery boiler and for the WISC engine. It can be seen that the maximum efficiency occurs at a steam injection rate of about 20% by mass. The WISC engine shows a better thermal efficiency than does the engine with the boiler. However, the most significant difference is from the maximum efficiency point to the full power point.

The dashed line in the figure indicates the efficiency of a simple cycle gas turbine engine with the same component efficiencies as the steam-injected engines. It is slightly higher than the zero water injection point for those engines because of the pressure drop in the heat exchangers. In this figure and the following ones, the WISC (supercritical heat exchanger / 2-phase expander) case includes the pressure drop across the condenser, while the waste heat boiler case does not, this relatively small difference may be seen at the zero water injection point.

The breaks in the curves at the right end of the figure represent the stoichiometric point. The drooping part of the curve to the right of the break was calculated using the highest possible turbine inlet temperature, which decreased from the specified 2000°F (1094 °C).

Figure 4 shows the effect of steam injection on the specific work output for the engine with the boiler and

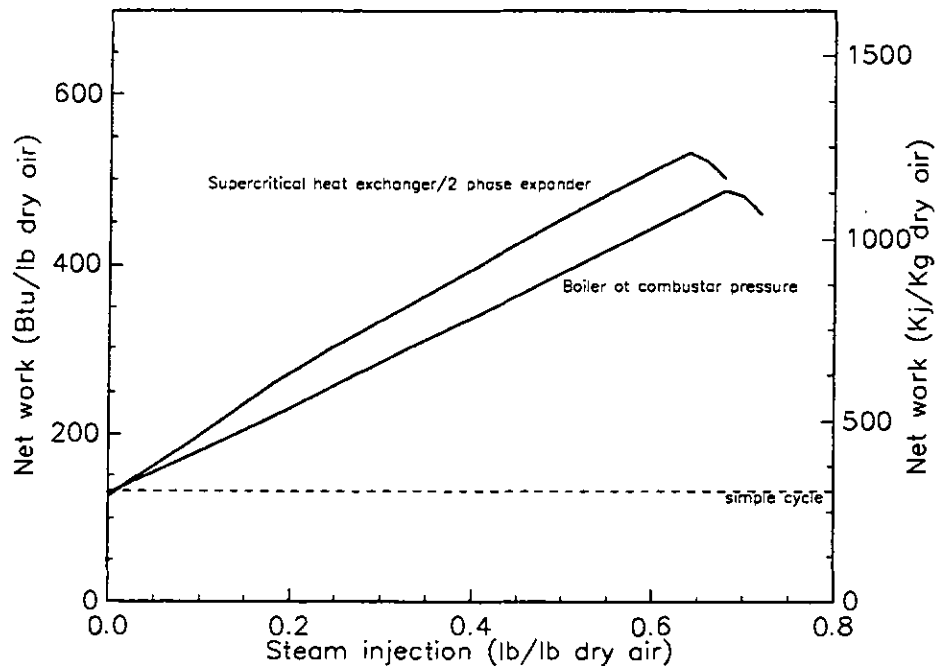

Figure 4: Net work as a function of water injection

for the WISC configuration. The engine with the boiler increases the specific work output to 3.75 times that of the basic engine, while the WISC configuration increases the specific work output to to four times that of the basic engine. This is quite significant, especially when compared with the specific work output of a combined cycle which is approximately 1.5 times that of a simple gas turbine cycle.

Figure 5 shows the effect of steam injection on the turbine exit temperature and the heat exchanger exit temperature for the engine with a boiler and for the WISC configuration. The difference in turbine exit temperatures in this figure is a result only of the different assumptions as to condenser pressure drop. It may be seen that a waste heat boiler recovers more heat than a supercritical heat exchanger, the latter is more efficient because some of the recovered heat is converted to work when the water is expanded to the combustor pressure.

Figures 6 and 7 show the effect of changing the compressor pressure ratio while keeping all other parameters constant on thermal efficiency and the specific work. It is clear that increasing the pressure ratio without intercooling is detrimental to performance, assuming that large rates of steam injection will be used.

Figures 8 and 9 show the effect of raising the turbine inlet temperature from 2000°F (1094°C) to 2500°F (1371°C) while the compressor pressure ratio is held at 16:1. Significant gains in both efficiency and net work may be seen from increasing the turbine inlet temperature, these gains increase with water injection rate.

CONCLUSIONS

The present analysis of the performance of the steam injected gas turbine engine until stoichiometric combustion is reached (WISC) with the engine using either a simple boiler or a supercritical heat exchanger supports our claim that this technology should have an important impact on future gas turbine power plants. The WISC power plant can be much more economical than the combined cycle power plant because it requires less additional equipment relative to the simple cycle.

We also offer the recommendation that this technology can be used to upgrade existing gas turbine power plants and make them competitive with the best available technology for a fraction of the cost of building new plants.

ACKNOWLEDGEMENTS

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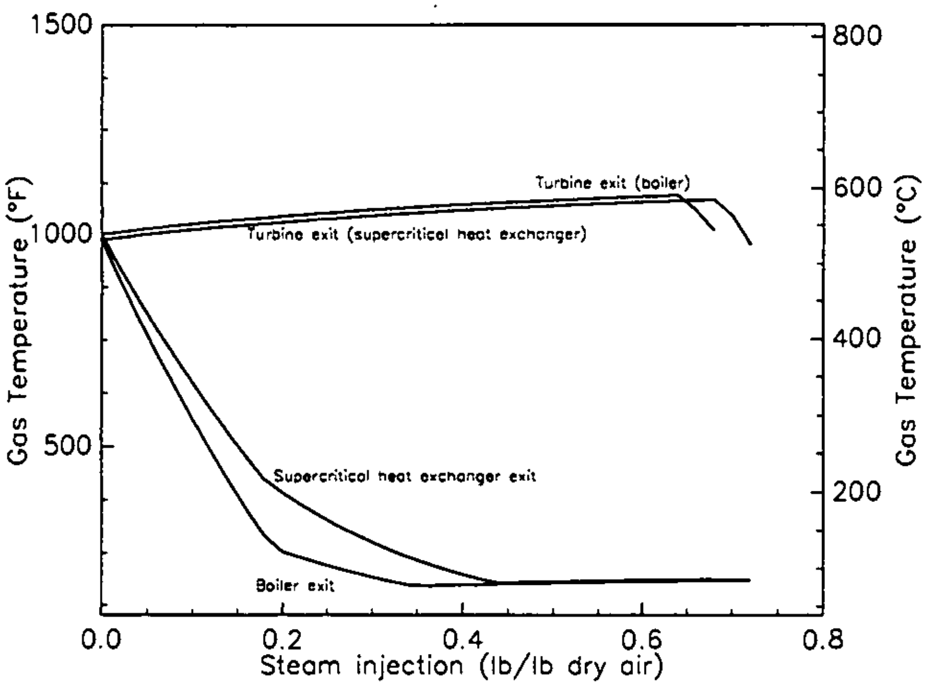

Figure 5: Exhaust temperatures as functions of water injection

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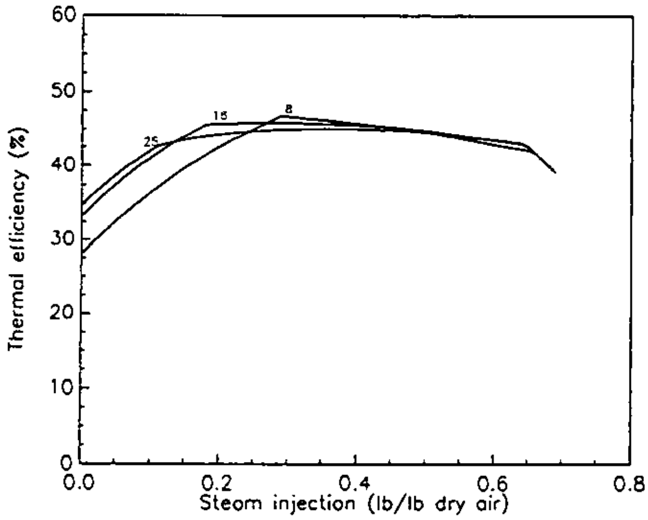

Figure 6: Effect of pressure ratio on efficiency (WISC case)

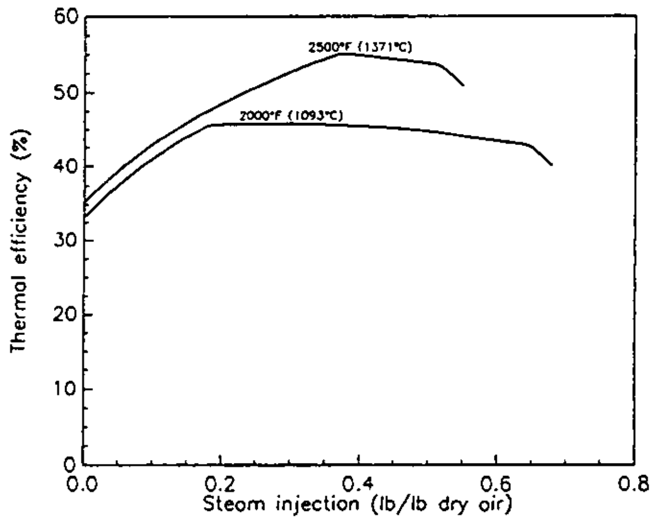

Figure 8: Effect of turbine inlet temperature on efficiency (WISC case)

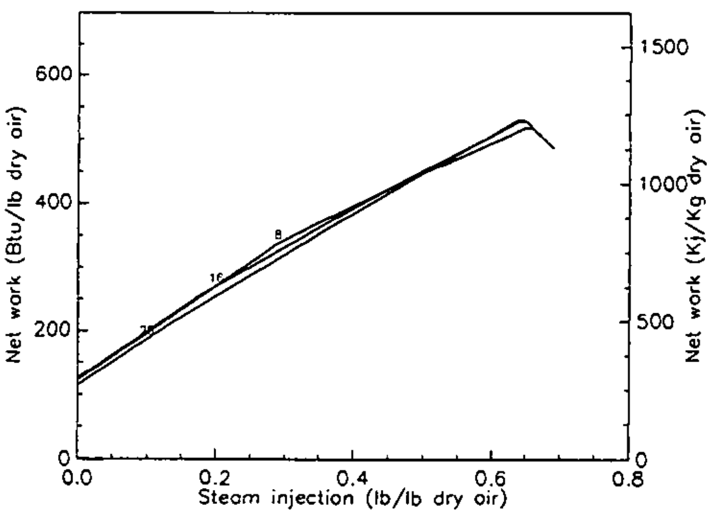

Figure 7: Effect of pressure ratio on net work (WISC case)

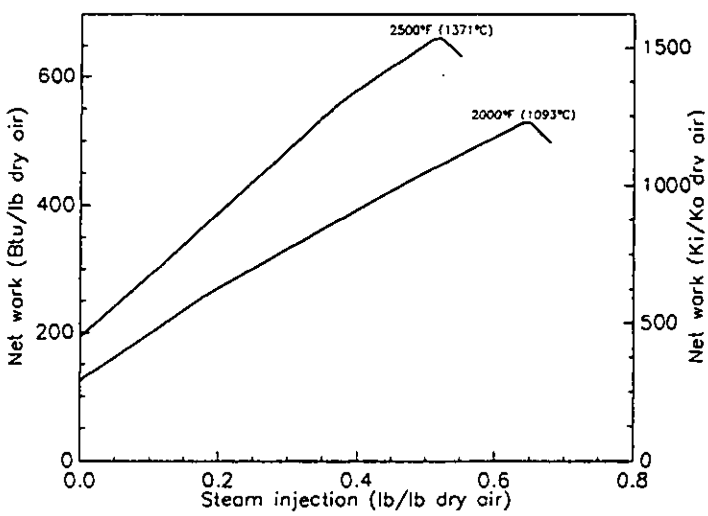

Figure 9: Effect of turbine inlet temperature on net work (WISC case)