Proceedings of ASME:
TURBO EXPO 2002
June 3-6, 2002, Amsterdam, The Netherlands

GT-2002-30081

INTEGRATING THE STAGED PREVAPORIZER-PREMIXER INTO GAS TURBINE CYCLES

J. S. Campbell, Jr.
Department of Mechanical Engineering
University of Washington
Box 352600
Seattle, Washington 98195-2600

P. C. Malte*
Department of Mechanical Engineering
University of Washington
Box 352600
Seattle, Washington 98195-2600
(*corresponding author)

S. de Bruyn Kops
Department of Mechanical Engineering
University of Washington
Box 352600
Seattle, Washington 98195-2600

I. Novosselov
Department of Mechanical Engineering
University of Washington
Box 352600
Seattle, Washington 98195-2600

John C.Y. Lee
Solar Turbines, Inc.
San Diego, CA 92186

M. A. Benjamin
Gas Turbine Fuel System Division
Parker Hannifin Corporation
Mentor, OH 44060

ABSTRACT

This paper describes a cycle analysis study on the use of the staged prevaporizer-premixer injector (SPP) in high-pressure gas turbine systems fired with liquid fuel. A review of the SPP is given, including discussions of its operational concepts and previous research. The main portions of the paper consist of analyzing the use of the SPP in three different gas turbine systems: a steam-injected gas turbine (STIG) engine, a Frame H gas turbine in combined cycle, and a reheat gas turbine in combined cycle. Focus is placed on determining the effect of the SPP on cycle efficiency. In addition, SPP use in an engine conventionally recuperated by heat exchange from the exhaust gas stream to the compressor discharge air is examined. The SPP offers the potential of low NOx emissions for liquid-fired gas turbines. Because water injection is a method currently practiced for the reduction of NOx, simulations of engines without the SPP but with water injection into the combustor are also performed and comparisons are made. The simulation process is described, as are methods of how the SPP is implemented into the various engines.

Results of the study are given, showing the effect of SPP use on cycle efficiency. In general, except for application to the conventionally recuperated engine, use of the SPP causes a decrease in cycle efficiency of around 1-3 percent (relative). The impact of water injection is somewhat greater, causing a 2.5-4 percent (relative) decrease in cycle efficiency. Further, the water injection does not provide as much NOx control as the lean prevaporized-premixed combustion.

INTRODUCTION

With environmental regulations applied to the power generation industry becoming more stringent in recent years, the importance of low-NOx combustion has increased. In combustors fired by natural gas, lean-premixed combustion is used to achieve NO_x emissions typically below 25 ppmv (at 15% O_2 dry), and sometimes under 10 ppmv. However, when liquid fuels such as No. 2 diesel oil are used, such low pollutant levels are difficult to reach. Water injection at the combustor can be used, of course, to lower the flame temperature and with it NOx emissions. This practice, unfortunately, has the drawbacks of decreased system efficiency and increased capital cost, and typically results in minimum NOx emissions substantially greater than 25 ppmv. Therefore, for engines fired with liquid fuels, effective prevaporizing and premixing of the fuel prior to combustion is a potentially more economically and environmentally attractive practice.

An injector that performs the prevaporizing and premixing tasks, the staged prevaporizer-premixer (SPP), has been studied at the University of Washington (Lee, 2000; and Lee et al., 2001). In order to succeed in industrial applications, the staged prevaporizing and premixing technology must have the ability to operate at the high pressures and temperatures of gas turbine engines. One-atmosphere testing of the SPP has been performed, and high-pressure testing is underway. As a complementary effort, the present cycle analysis study of SPP use in high-pressure gas turbine systems has been conducted. The goal is to consider the integration of the SPP into various gas turbine cycles and to assess the impact of the SPP on cycle efficiency

REVIEW OF SPP METHOD

The SPP achieves lean prevaporizing and premixing of liquid fuels by using two levels of air temperature. In stage 1, the cooler air enters the SPP. In practice, the stage 1 air would be air from the compressor discharge that would be cooled. Liquid fuel is atomized and injected into stage 1, where it mixes with the stage 1 air. Second stage air is mixed into the SPP flow through several staggered jets. The stage 2 air is hotter than the stage 1 air, bringing the fuel-air mixture temperature up to the combustor inlet temperature. In practice, the stage 2 air would be the balance of the compressor discharge air. The light ends and middle boiling compounds of the blended fuel are vaporized in stage 1, with the vaporization and mixing processes being completed in stage 2.

In Figure 1, the SPP is illustrated using results from a computational fluid dynamics (CFD) simulation of the laboratory-scale SPP of Lee (2000). From top to bottom in this figure, the SPP is illustrated by gas temperature, fuel vapor mass fraction, and gas velocity magnitude. The flow is from left to right. At the left (upstream) end of the SPP, the stage 1 air enters as streams on both sides of an annular air flow splitter. The No. 2 diesel fuel spray enters at the upstream end, dispersed as a cone around the centerline. Initial fuel droplet size is 15 μm. At stage 2 (right 40% of the figure), air enters through 45° radial jets. The outlet pressure is 30 atm, and the pressure drop between the air inlet manifolds and the SPP outlet is 5%. The flow conditions in stages 1 and 2 of the SPP are given in Table 1. The pressure is 30 atm (at the SPP outlet) and the 2nd stage air inlet temperature is 823 K. The residence time in the 2nd stage is reduced to 1.63 ms by increasing the mass flow rate of the SPP. In order to maintain the overall air pressure loss of the SPP at 5%, the 2nd stage air inlet jets are increased in size (each of the 16 jets is given a diameter of about 1 mm). For the exit plane of the SPP, the CFD solution of the Reynolds averaged Navier Stokes equations gives a fairly small spatial variation in the fuel vapor mass fraction, indicated by a standard deviation over mean value of about 0.08. Because of the injection of the 2nd stage air towards the centerline of the SPP, slightly reduced levels of fuel vapor are seen in the center of the exit plane. The exit diameter of the laboratory-scale SPP is 12.6 mm.

Table 1: SPP conditions for pressure of 30 atm.

Value	Stage 1	Stage 2
Temperature of inlet air, °C (K)	473 (746)	550 (823)
Mass flow rate within SPP, kg/s	0.0613	0.184
Residence time within SPP, ms	4.16	1.63

NOx REDUCTION AND RELATION TO THIS STUDY

The SPP reduces NOx formation by prevaporizing and premixing the fuel. Two laboratory studies previously conducted at the University of Washington are particularly relevant in this regard.

Lee (2000) conducted an atmospheric pressure study of the laboratory-scale SPP firing a jet-stirred reactor (JSR). The main findings of this study were reported by Lee et al. (2001). The study involved the variation of the air temperatures and flow rates of the two stages of the SPP, the measurement of NOx formed within the JSR, and the indication of complete vaporization of the liquid fuels and a high degree of mixedness of the fuel vapor and air through laser-probing of the outlet stream of the SPP. Several liquid and gaseous fuel were studied, the two of greatest importance being No. 2 diesel fuel and natural gas. For all experiments, the combustion temperature of the JSR was held at 1790K by adjusting the fuel-air ratio, and the residence time of the JSR was 2.3±0.1 ms. The nominal fuel-air equivalence ratio was 0.6. The NOx level (adjusted to 15% O_2 dry) was found to be 8.8-11.5 ppmv for No. 2 diesel fuel, and 4.8-5.0 ppmv for natural gas, which served as the "benchmark" NOx level for the study. The fuel bound nitrogen (FBN) content of the diesel fuel ranged from about 45 to 90 ppmw, and primarily accounted for the variation of 8.8-11.5 ppmv stated for the NOx. The importance of these NOx measurements is their demonstration that low levels of NOx can be obtained from No. 2 diesel oil combustion if the fuel is fully prevaporized and significantly premixed with the combustion air.

The second University of Washington laboratory study of relevance to the present paper is that of Steele (1995). The main findings of this study were reported by Steele et al. (1998). The NOx levels obtained from the lean premixed burning of methane in JSRs were examined as a function of combustion temperature (about 1550 to 1950K), inlet temperature (about 300 to 600K), pressure (1 to 7.1 atm), and residence time (about 2 to 7 ms). Of most importance, the JSR measurements of NOx agreed with the data of other investigators, using other bench-scale flames as well as gas turbine (GT) combustors. The JSR NOx exhibited sensitivity to combustion temperature similar to that seen in other combustors, and mainly depended on combustion temperature, a finding also observed elsewhere, eg, see Leonard and Stegmaier (1993). Additionally, the JSR measurements exhibited NOx levels as low as any of the bench-scale scale flames and GT combustors. The importance of these NOx measurements is their demonstration that JSR measurements are indicative of NOx behavior in lean premixed GT combustors and of the low levels of NOx that can be obtained in lean premixed combustion.

The lowest levels of NOx obtained from natural gas-fired GT engines in the field are just into the single-digit regime, about 9 ppmv (15% O_2 dry). If the factor two increase in NOx from natural gas to No. 2 diesel fuel measured for Lee's (2000) SPP-JSR laboratory system is assumed to hold for full-scale systems, the NOx emission from a GT engine equipped with an advanced prevaporizer-premixer and fired on No. 2 diesel fuel (with a maximum of about 100 ppmw FBN) should be less than 25 ppmv (15% O_2 dry). This level of NOx is significantly lower than the levels of NOx obtained by water-injecting conventional (diffusion flame) oil-fired gas turbine combustors. For such engines, NOx emission is seldom below 40 ppmv (15% O_2 dry), and can be about 75 ppmv.

The challenge for obtaining low NOx emission for oil-fired gas turbines engines is the prevaporizer-premixer technology, which must not only provide good vaporization and mixing, but must resist autoignition and carbon fouling. The SPP is an approach for accomplishing this. However, since its performance relies on cooling the portion of the gas turbine engine compressor discharge air routed to the first stage of the SPP, the engine cycle is modified by the inclusion of the SPP and there is the potential for a decrease in the thermodynamic efficiency of the cycle. Exploration of this is the focus of this paper: determining how the cooling of the first stage air and the flow split between the first and second stage air of the SPP affect cycle efficiency. A key trade-off the gas turbine engine designer may face is that of cycle efficiency versus resistance to autoignition. As the temperature of the first stage air of the SPP drops, and as the percentage of the air flow of the SPP assigned to the first stage increases, the autoignition tendency decreases though the cycle efficiency may suffer. Another trade-off of potential concern is that of cycle efficiency versus NOx reduction. However, the measurements of Lee (2000) showed only a weak effect of the SPP temperatures and stage air flow split on NOx. That is, so long as the SPP outlet stream is well vaporized and mixed, the particular conditions used within the SPP do not much matter to the NOx formation process. Thus, a significant trade-off between cycle efficiency and NOx reduction is not anticipated (so long as the SPP is operated for good vaporization and mixing). In Lee's experiments, the temperature of the first stage air was reduced by 100-200 degrees C relative the temperature of the second stage air, and the percentage of the total air passed through the first stage of the SPP was varied from 33 to 67%. These changes had little effect on the NOx. Decreasing the outlet temperature of SPP did cause a mild increase in NOx, but only for firing on natural gas. This effect was attributed to the increase in fuel-air ratio required to maintain the constant combustion temperature of 1790K, an effect that would occur with or without the SPP.

CYCLE SIMULATION AND ANALYSIS

Simulation Development

For this study, the engine cycle analyses are performed using GT Pro and Thermoflex software. These software products were obtained from Thermoflow, Inc. (2002). The products are fairly widely used in the gas turbine industry, and have been previously used in IGTI papers involving cycle assessment, eg, see Vermes et al. (2001).

For the present study, GT Pro is used to provide baseline results for a GT system operating without the SPP, and Thermoflex is used to quantitatively determine the extent of the system changes required for the integration of the SPP into the GT and the subsequent impact of the SPP on the system performance, especially the overall efficiency of the cycle. Specific features of the cycle simulation process vary depending on the type of engine under analysis.

The general model development process is shown in flowchart form in Figure 2. A model is built in GT Pro in order to calculate baseline operating states of the system. The model includes the gas turbine, the properties of which are contained in the GT Pro database, a multi-pressure heat recovery steam generator (HRSG), and a deaerator. Steam turbines are included if a combined cycle system is being analyzed. If augmentation or confirmation of the GT Pro database is required, the relevant engine literature is sought and used. [This has been done for the reheat engine, based on Farmer (1993).] After the GT Pro model is constructed, it is converted to a Thermoflex model, which permits system modification and the addition of the SPP. The Thermoflex model is then tested to ensure that its results for a baseline case match those of the GT Pro model for the same case. After testing, the baseline Thermoflex model is ready for modification to include the SPP.

Once a working Thermoflex model of the gas turbine system is in place, the model is modified to include the SPP and its supporting hardware. For example, Figure 3 shows a schematic of the steam injected gas turbine (STIG) system. Since the SPP receives two streams of compressor discharge air, the first of which is cooler than the second, for mixing with the GT fuel, an air flow divider and heat exchanger (HX) are placed between the compressor discharge and the SPP. Part of the compressed air flows to the HX, where it is cooled by steam in the HRSG, and then flows to stage 1 of the SPP, where it mixes with and vaporizes the fuel. The remaining, uncooled air flows to stage 2 of the SPP, where it mixes with the air/fuel stream from stage 1. The set up of the SPP in the combined cycles is similar. However, in the conventionally recuperated engine, lacking the HRSG, rather than cooling the stage 1 air, the stage 2 air is heated by routing the GT exhaust through the recuperator HX.

Thermoflex does not include fuel injector modeling capability, so the SPP is represented in the simulation by an air stream mixer that combines the stage 1 and 2 air streams. Further, Thermoflex permits the GT fuel to be added only at the combustor and separate from the entering air. Because the fuel is injected at the combustor, neither of the two air streams actually combines with fuel in the SPP simulation. Hence the SPP temperatures are not debited by the cooling effect of the fuel undergoing vaporization. If included, this effect would decrease the SPP outlet temperature by about 12°C. For the reheat engine, the SPP is employed only at the first combustor. The second combustor operates on high-temperature, vitiated air, for which the SPP is not currently rated.

The baseline compressor-discharge-to-turbine-inlet pressure drop for the engines considered in this analysis is 4 percent. This is reapportioned as follows: a 3-percent drop in all of the components between the compressor discharge and the combustor liner interior (including the HX and SPP) and a 1-percent drop within the combustor liner interior. Additionally, in order to simulate increased pressure loss, the compressor-discharge-to-turbine-inlet pressure drop is increased to 5.5 and 7 percent. In all cases simulated, the pressure drop within the combustor liner interior is maintained at 1 percent. The SPP stage 2 air is throttled upon entry into the SPP so that its pressure matches the SPP stage 1 air pressure within the SPP.

In this analysis, the fluid used to cool the SPP stage 1 air is extracted from the HRSG. For the Frame H and reheat engines, water from the HRSG HP line (extracted just upstream of the HP feed pump) is used. After being converted to steam in the HX, this fluid is routed back to the HRSG. For the STIG engine, HP steam is used. The STIG HP steam line is divided into two streams, one of which is routed to the HX, where it receives heat from the SPP stage 1 air, then recombines with the remainder of the HP steam before being sent to the combustor. Streams are chosen for cooling because of suitable mass flows and enthalpies.

Cases Run

Cases run in this study are based on equipment sizes and specifications calculated for the baseline case. As such, the results of the study indicate the performance of the given system altered by the replacement of the GT fuel nozzles by the SPP injector and the addition of the HX hardware and plumbing. The effectiveness of the HX is assumed to be 0.85 (for the conventionally recuperated engine, the effectiveness is 0.95), the ambient temperature is 15°C (59°F), and the GT fuel is No. 2 diesel, using properties obtained from a fuel manufacturer.

SPP cases are run for the following variations:

- Air-side pressure drop between compressor discharge and combustor interior: 3 percent (baseline), 4.5 percent, and 6 percent. The corresponding overall compressor-to-turbine pressure losses are 4, 5.5, and 7%, respectively.
- Water-side pressure drop: 0, 2, 4, and 6 percent.
- Percentage of SPP air flow through stage 1: 10, 20, 30, 40, and 50 percent.

The overall pressure loss of 4 to 7 percent covers the range of pressure losses estimated to occur in lean-premixed combustors. However, in some applications, the inclusion of the heat exchanger might increase the pressure loss of the SPP system to somewhat greater levels, say, to 8 to 10 percent. Although these levels are not treated in the present analysis, their impact on system performance may be easily determined by extrapolating the results described below.

In addition to the parameters listed above, the amount of temperature decrease in the SPP stage 1 air is varied. The SPP stage 1 air temperatures are determined from different SPP stage 1/stage 2 temperature ratios. The temperatures are defined as those of the air entering the SPP. In this analysis, ratios of 1.1, 1.2, and 1.3, based on absolute temperatures, are used. The SPP stage 2 air temperature (except for the conventionally recuperated engine) is equal to the compressor discharge temperature. The temperature variations are listed below:

- STIG engine: Compressor discharge pressure = 33 atm. Compressor discharge temperature = 569°C (1056°F). SPP stage 1 temperatures = 498°C (929°F), 434°C (813°F), 379°C (715°F).
- Frame H engine: Compressor discharge pressure = 23 atm. Compressor discharge temperature = 496°C (925°F). SPP stage 1 temperatures = 405°C (762°F), 349°C (661°F), 301°C (574°F).
- Reheat engine: Compressor discharge pressure = 30 atm. Compressor discharge temperature = 547°C (1017°F). SPP stage 1 temperatures = 473°C (883°F), 411°C (771°F), 358°C (676°F).

For the conventionally recuperated engine, cases are run for increasing SPP stage 1 air flow rates and constant values of HX effectiveness and turbine inlet temperature (TIT). The compressor discharge pressure is 10.5 atm, and the air-pressure drop is held constant at 3 percent with the combustor liner interior pressure drop constant at 1 percent. The stage 1 air flow rate is increased from 0 percent to 33 percent of the total compressor discharge air flow. Constant values of HX effectiveness and TIT are 0.95 and 1260°C (2300°F), respectively.

Water injection analyses are conducted on only the Frame H and reheat engines. In these analyses, the systems are modified by the addition of water injection at the combustor – the SPP and its supporting hardware are not used. For each engine, the first case run consists of injecting an amount of water equal to the baseline GT fuel flow rate for No. 2 diesel fuel. In the Frame H engine, this amount is 18.0 kg/s (39.6 lbm/s); in the reheat engine, it is 10.7 kg/s (23.5 lbm/s). For a given fuel flow rate and combustor inlet temperature, the water injection has the effect of lowering the turbine inlet temperature (TIT). In subsequent cases, the water flow rate into the combustor is held constant and the fuel flow rate is increased until the baseline TIT value [1445°C (2630°F) and 1230°C (2250°F) for the Frame H and reheat engines, respectively] is reached. For all cases, makeup water at 15°C (59°F) is used for combustor injection.

RESULTS

STIG Engine

The steam injected gas turbine (STIG) cycle, based on the LM5000 GT, is reviewed by Weston (1992). This engine has been selected for this study because of its high pressure (33 atm). The present analysis shows the SPP affects the thermodynamic efficiency and power output of the STIG engine mainly through the air pressure drop of the SPP; that is, through the pressure drop of the air between the compressor discharge and the inside of the combustion liner, rather than through thermal energy transfers. In the simulation, thermal energy is conserved as energy given up by the air in the HX is picked up by the HX coolant steam and then re-enters the cycle at the combustor with the injected steam.

The HX steam-side pressure loss has no effect on cycle performance. This is because Thermoflex throttles the steam that is injected into the combustor to a constant pressure based on the combustor inlet gas pressure.

Table 2 shows the variation in cycle efficiency and power output with changes in the air-pressure drop. For all cases, the pressure loss within the combustion liner is 1 percent. The 3 percent loss in air-pressure corresponds to the baseline 4 percent loss in pressure between the compressor discharge and the turbine inlet. The larger pressure drops result in small decreases in cycle performance. Thus, the impact of the SPP on the efficiency of the STIG cycle is quite small.

Table 2: Variation of cycle efficiency and power output with air ΔP, STIG engine.

Air pressure loss (%)	Cycle efficiency (%)	Power output (MW)
3 (baseline)	42.60	48.2
4.5	42.34	47.9
6	42.08	47.6

Frame H Engine

The variation of combined cycle combustion turbine (CCCT) efficiency with changes in SPP stage 1 air flow rate and temperature for the Frame H engine is shown in Figure 4. As the figure demonstrates, the inclusion of the SPP has a small impact on the overall system performance. The lowering of the combined cycle efficiency is quite small at low SPP stage 1 air flow rates and high SPP stage 1 temperature values, though it is larger as the stage 1 air flow rate increases or as the stage 1 temperature decreases. The drop in efficiency is a result of the increased percentage of the total power produced by the less-efficient steam cycle. This is caused as heat is extracted from the air stream and deposited in the water/steam. The SPP appears best suited for this application if low stage 1 flow rates and relatively high stage 1 temperatures can be used.

The effect of the air-pressure drop on cycle efficiency is shown in Figure 5 for a SPP stage 1 temperature of 405°C (762°F) and a water/steam-side pressure drop of 0 percent. As the figure shows, increasing the air-pressure drop results in a decrease in cycle efficiency because of the reduced turbine expansion ratio. As the decrease in efficiency is small for small HX air-pressure drops, the SPP is best suited for use when the pressure loss can be held close to the baseline pressure loss of the engine.

Changes in the water/steam-side pressure drop are determined to affect only the temperature increase in the HX water/steam.

Reheat Engine

The variation of combined cycle (CCCT) efficiency with changes in SPP stage 1 air flow and temperature for the reheat engine is shown in Figure 6. Figure 7 shows the effect of the air-pressure drop on cycle efficiency for an SPP stage 1 temperature of 473°C (883°F). As the figures demonstrate, the effect of the SPP on overall system performance here is similar to that of the Frame H engine. Again, the drop in efficiency is a result of the increased percentage of the total power produced by the less-efficient steam cycle.

Autoignition

The reheat and STIG engines, because of their high pressures of 30 and 33, respectively, are especially susceptible to auto-ignition within the SPP. For the STIG engine, the temperature of stage 1 of the SPP can be significantly reduced without causing much impact on the cycle efficiency, because the heat of the compressor discharge air is transferred to the steam entering the combustor. However, for the reheat engine, an excessive decrease in the stage 1 temperature of the SPP would negatively impact the cycle efficiency. In Table 3 below, the autoignition delay times, based on Spadaccini and TeVelde's (1982) "inlet temperature" equation for No. 2 diesel fuel, are shown for the two stages of the SPP for application in the reheat engine. The stage 2 temperature used in Table 3 is the mean of the inlet air temperatures of stages 1 and 2. Two situations are shown: 30% and 50% stage 1 air flow.

These results indicate a weak autoignition tendency in stage 1 for the 358 and 411°C inlet air temperatures. However, the calculations also imply a risk for autoignition in stage 2, unless the stage 1 inlet air temperature is decreased to 358°C and the percentage of air in stage 1 is increased to 50% (thereby decreasing the temperature in stage 2). By the results plotted in Figure 6, this configuration would decrease the CCCT efficiency of the reheat engine to about 55.3%, which corresponds to a 3% (relative) decrease from the baseline efficiency of 57%.

Table 3: Autoignition delay times in SPP stages 1 and 2 for the 30 atm reheat engine.

Stage 1 Inlet Air Temp °C (°F)	Auto-ignition Delay Time ms	**Stage 2** Temp (30% Air Stage 1) °C	Auto-ignition Delay Time ms	**Stage 2** Temp (50% Air Stage 1) °C	Auto-ignition Delay Time ms
547 (1017)	0.3	547	0.3	547	0.3
473 (883)	4.1	525	0.7	510	1.1
411 (771)	51	506	1.2	479	3.2
358 (676)	671	490	2.2	453	8.9

Water Injection

Figures 8 and 9 show water injection analysis results for the Frame H and reheat engines, respectively. As the figures indicate, the addition of combustor water injection has a detrimental impact on combined cycle (CCCT) efficiency. For Figure 8, the point for a water-to-fuel ratio of 0.88 corresponds to running the engine at the baseline TIT of 1445°C (2630°F). At this point, the cycle efficiency is 56.7 percent, a drop of 2.4 percent (absolute) from the 59.1 percent efficiency of the oil-fired, dry Frame H engine. For Figure 9, pertaining to the reheat engine, the point for a water-to-fuel ratio of 0.89 corresponds to running the engine at the baseline TIT of 1230°C (2250°F). At this point, the cycle efficiency is 55.6 percent, a drop of 1.4 percent (absolute) from the 57.0 percent efficiency of the oil-fired, dry reheat engine.

Conventionally Recuperated Cycle

Figure 10 shows the variation of cycle efficiency with increasing SPP stage 1 air flow rate as a percentage of compressor discharge flow. Clearly, increasing this flow rate causes a decrease in efficiency. This decrease can be explained by the fact that increasing the flow rate through stage 1 of the SPP decreases the flow through stage 2, and therefore decreases the amount of heat removed from the GT exhaust gases. The peak efficiency of 39.2 percent shown in the figure is for the system with the recuperator only; i.e. no SPP. The decrease in efficiency in this system is greater than that in the combined cycle combustion turbine systems analyzed previously. This is due to the fact that, in those systems, the lower GT efficiency is lumped into the overall system efficiency, and is therefore buffered by the steam cycle efficiency.

CONCLUSIONS

Analysis of SPP use in a variety of gas turbine systems has been completed. The results of the study show the SPP can be implemented into a STIG cycle or a combined cycle, powered by a Frame H or reheat gas turbine, with small decreases in combined cycle efficiency. For the combined cycle engines in particular, the cycle efficiency will decrease by about 1 percent (relative) for an SPP (and HX) air-pressure drop of 3 percent and low stage 1 cooling requirements. On the other hand, if relatively large amounts of stage 1 cooling are required to control autoignition, the combined cycle efficiency may decrease by about 3 percent (relative). Further, viewing the SPP as a NOx control technology, important points are that water injection into the combustor to control NOx would probably result in as great or greater efficiency losses and would not reduce the NOx to the low levels possible by prevaporizing and premixing the liquid fuel.

REFERENCES

Farmer, R. (1993). "Reheat GT's Boost 250 and 365-MW Combined Cycle Efficiency to 58%," *Gas Turbine World*, Vol. 23, No. 4, pp. 18-23.

Lee, John C. Y (2000). "Reduction of NO_x Emission for Lean Prevaporized-Premixed Combustors," Ph.D. Dissertation, University of Washington, Seattle, WA.

Lee, John C. Y., Malte, P. C., and Benjamin, M. A.(2001). "Low NO_x Combustion of Liquid Fuels: Atmospheric Pressure Experiments Using a Staged Prevaporizer-Premixer," ASME Paper No. 2001-GT-0081 presented at the ASME International Gas Turbine and Aeroengine Congress, New Orleans, LA.

Leonard, G. and Stegmaier, J. (1993). "Development of an Aeroderivative Gas Turbine Dry Low NOx Combustion System," ASME Paper No. 93-GT-288 presented at the ASME International Gas Turbine and Aeroengine Congress, Cincinnati, OH.

Spadaccini, L. J., and TeVelde, J. A. (1982). "Autoignition Characteristics of Aircraft-Type Fuels," *Combustion and Flame*, Vol. 46, pp. 283-300.

Steele, R. C. (1995). "NOx and N_2O Formation in Lean-Premixed Jet-Stirred Reactors Operated from 1 to 7 atm," Ph.D. Dissertation, University of Washington, Seattle, WA.

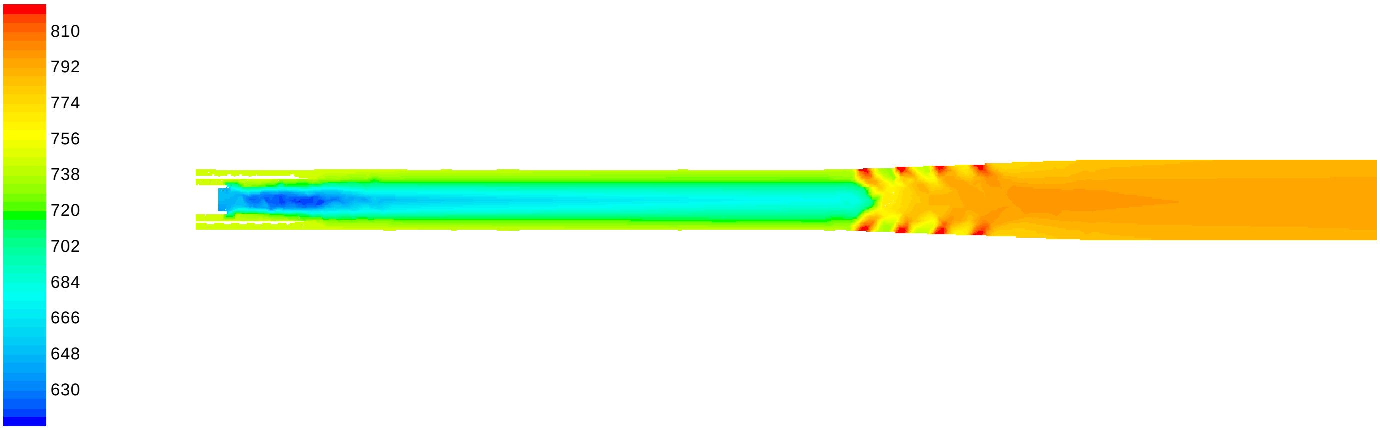
Steele, R. C., Tonouchi, J. H., Nicol, D. G., Horning, D. C., Malte, P. C., and Pratt, D. T. (1998). "Characteristics of NOx, N_2O, and CO for Lean-Premixed Combustion in a High-Pressure Jet-Stirred Reactor," *ASME Transactions Journal of Engineering for Gas Turbines and Power*, Vol. 120, pp. 303-310.

Thermoflow (2002). Thermoflow, Inc. web site: (www.thermoflow.com).

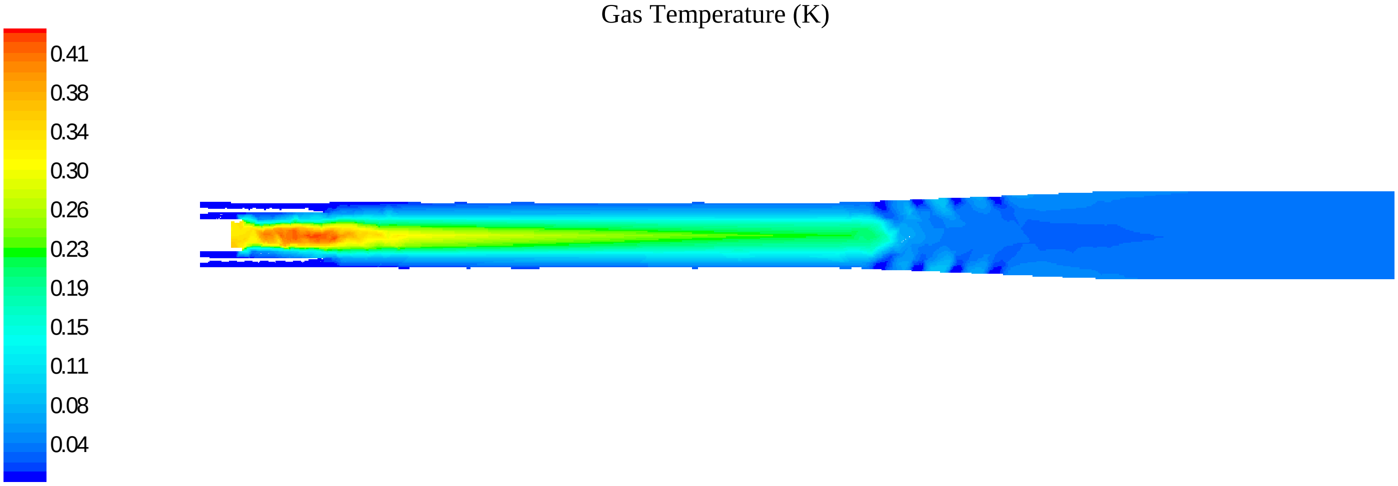
Vermes, G., Barta, L. E., and Beér, J. M. (2001). "Low NOx Emission from an Ambient Pressure Diffusion Flame Fired Gas

Turbine Cycle (APGC)," pamphlet paper presented at the ASME International Joint Power Generation Conference, New Orleans, LA.

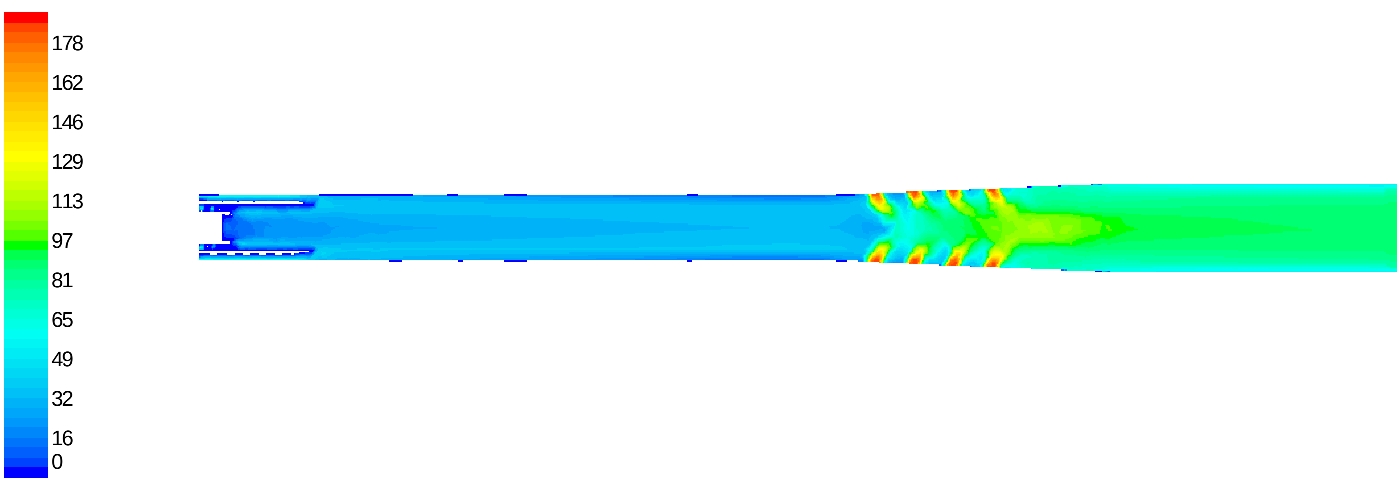
Weston, K. C. (1992). *Energy Conversion*, West Publishing Co., St Paul, MN, pp. 345-352.



Gas Temperature (K)



Fuel Vapor Mass Fraction



Gas Velocity Magnitude (m/s)

Figure 1: SPP illustrated by CFD simulation of 30 atm operation of the laboratory-scale SPP

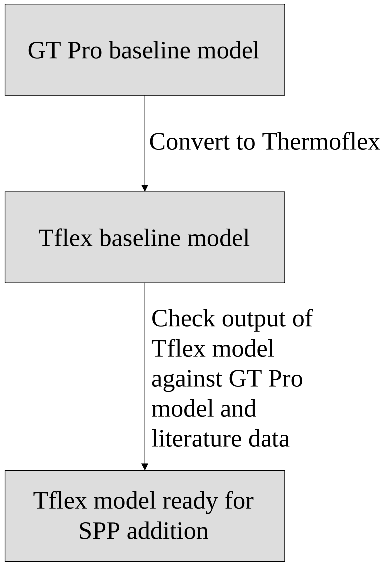


Figure 2: Model development flowchart.

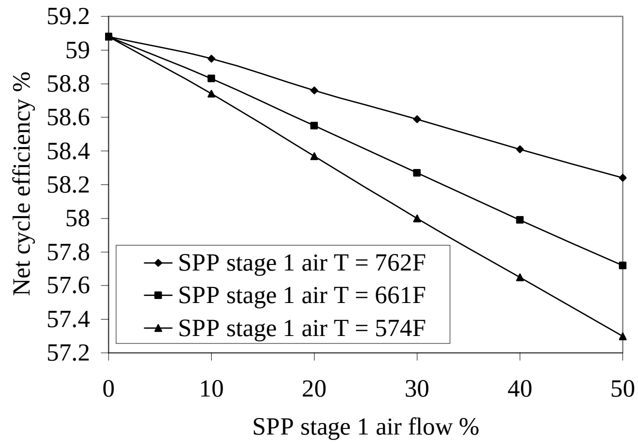


Figure 4: Variation of combined cycle efficiency with stage 1 air flow %, Frame H engine. Air ΔP = 3%.

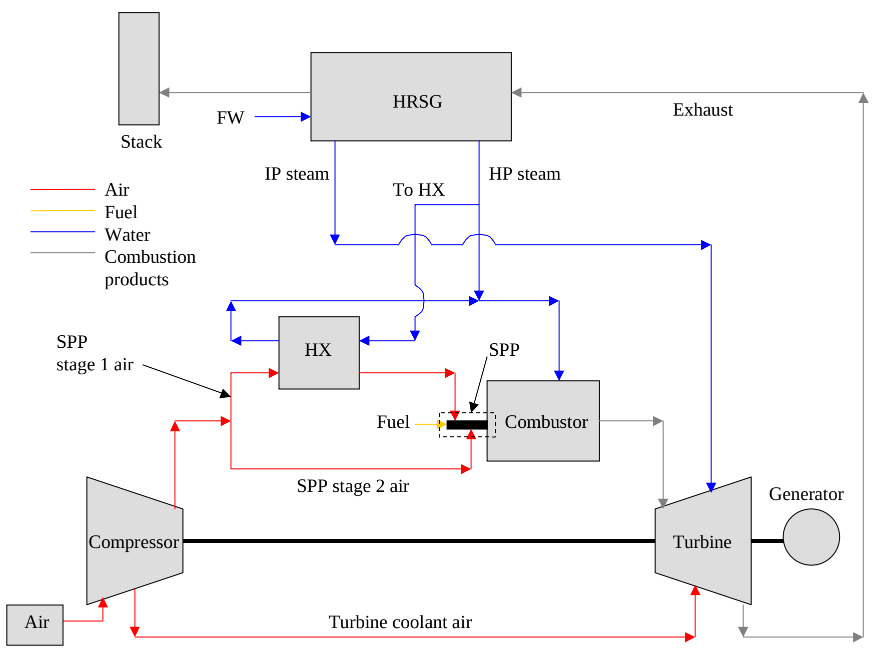


Figure 3: STIG system schematic.

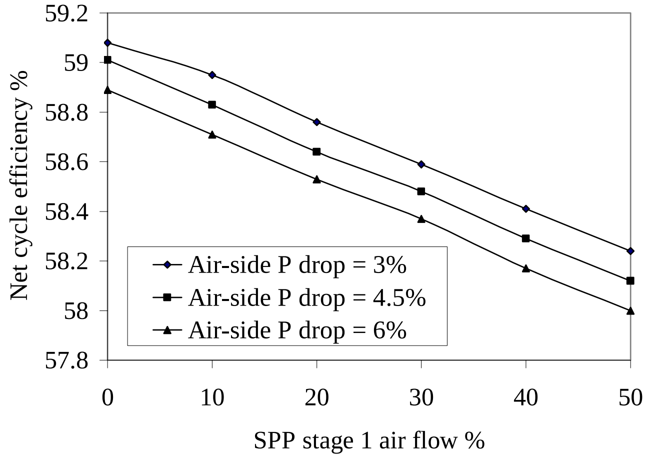


Figure 5: Variation of combined cycle efficiency with changes in air ΔP, Frame H engine. Stage 1 T = 405°C (762°F).

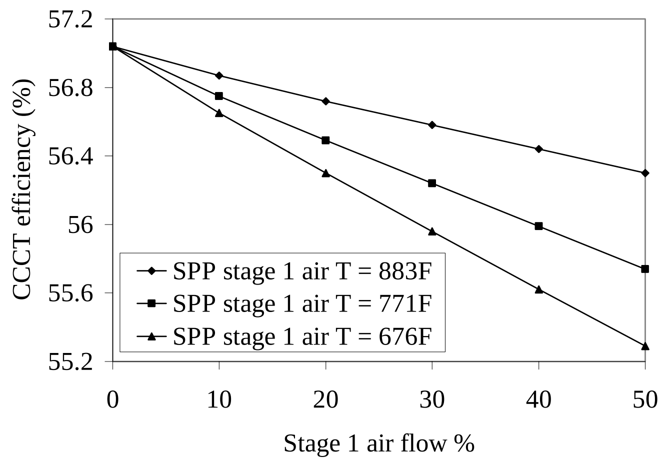


Figure 6: Variation of combined cycle efficiency with stage 1 air flow %, reheat engine. Air ΔP = 3%.

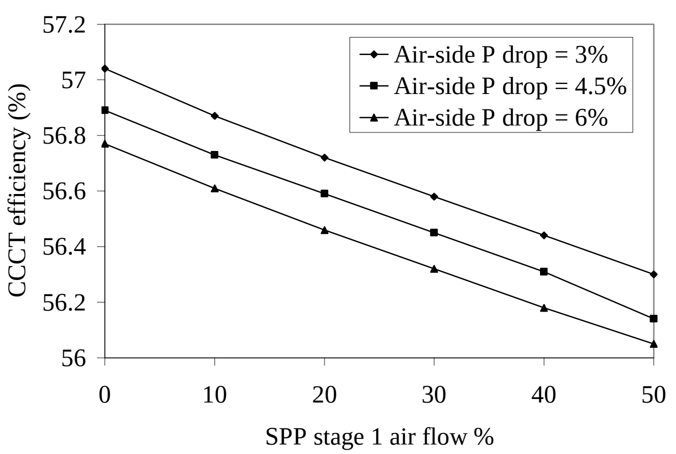


Figure 7: Variation of combined cycle efficiency with changes in air ΔP, reheat engine. Stage 1 T = 473°C (883°F).

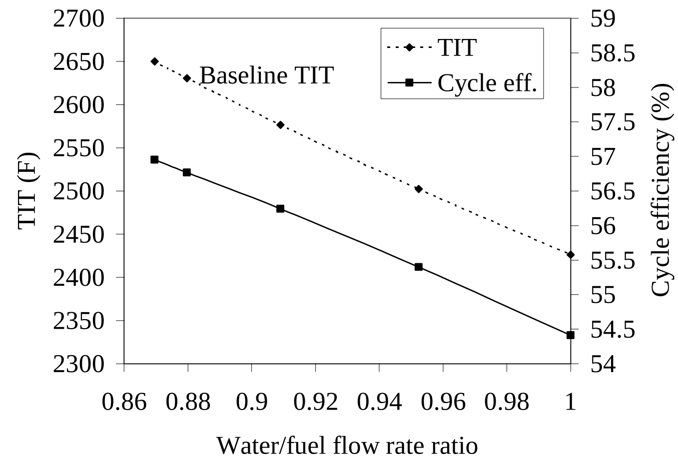


Figure 8: Variation of combined cycle efficiency and TIT with water/fuel flow rate ratio, Frame H engine.

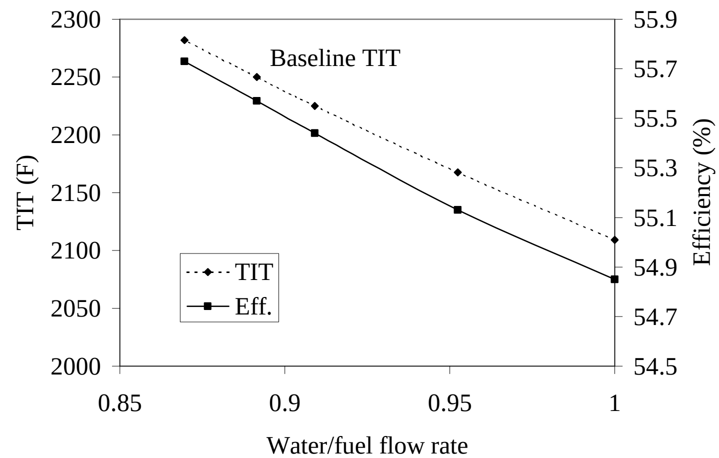


Figure 9: Variation of combined cycle efficiency and TIT with water/fuel flow rate ratio, reheat engine.

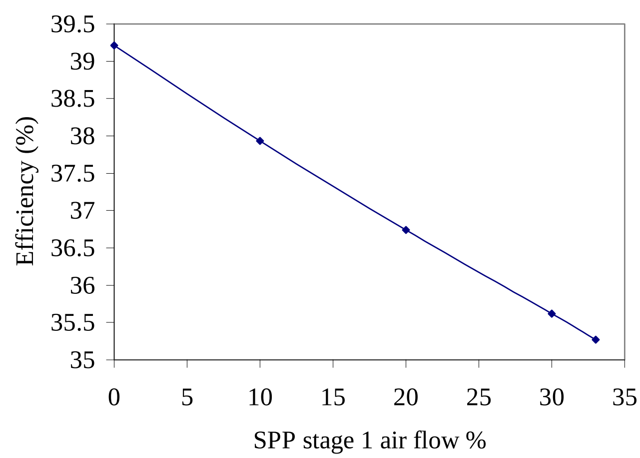


Figure 10: Variation of recuperated cycle efficiency with increasing SPP stage 1 air flow rate.