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EFFECT OF VARIABLE GUIDE VANES AND NATURAL GAS HYBRIDIZATION FOR ACCOMMODATING FLUCTUATIONS IN SOLAR INPUT TO A GAS TURBINE

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ABSTRACT

In recent years, several prototype solar central receivers have been experimentally demonstrated to produce high temperature and high pressure gas capable of driving a gas turbine engine [1-4]. While these prototype receivers are generally small (< 1 MWth), advancements in this technology will allow for the development of solar powered gas turbine engines at a commercial level (sizes of at least several megawatts electric (MWe)). The current paper analyzes a recuperated solar powered gas turbine engine, and addresses engine considerations, such as material limitations, as well as the variable nature of solar input. In order to compensate for changes in solar input, two operational strategies are identified and analyzed. The first is hybridization, meaning the solar input is supplemented via the combustion of fossil fuels. Hybridization often allows for an increase in net power and efficiency by adding heat during periods of low solar thermal input. An alternative strategy is to make use of variable guide vanes on the compressor of the gas turbine engine, which schedule to change the air flow rate into the system. By altering the mass flow rate of air, and assuming a fixed level of heat addition, the operating temperature of the engine can be controlled to maximize power or efficiency. The paper examines how to *combine* hybridization with variable guide vane operation to optimize gas turbine performance over a wide range of solar thermal input, from zero to solar-only operation. A large material constraint is posed by the combustor, and to address this concern two alternative strategies – one employing a bypass valve and the other a combustor modified to allow higher temperature inlet air – are presented. Combustor modifications could include new materials and/or increased cooling air. The two strategies (bypass vs. no bypass) are compared on a thermodynamic basis. Finally, a yearly assessment of solar share and thermodynamic performance is presented for a 4.8 MWe gas turbine to identify the overall benefits of the operational strategies.

Key words: concentrating solar power, Brayton cycle, gas turbine, variable guide vanes, hybridization, bypass valve, advanced combustor.

BACKGROUND

Central receiver power plants consist of a large field of mirrors (heliostats) which track the position of the sun [5].

FIGURE 1. SOLAR ONE/TWO, DAGGET, CALIFORNIA. EXPERIMENTAL CENTRAL RECEIVER PLANT 1982-2000.

These heliostats, which can number in the thousands, focus the sun's radiation onto a central receiver located at the top of a tall tower. The central receiver converts radiant energy from the sun into thermal energy which can then be used to drive a heat engine. Figure 1 shows a photo of Solar One/Two, which operated as an experimental plant for many years near Dagget, California and served as a test platform for two liquid-cooled receiver technologies, steam (Solar One) and molten salt (Solar Two). The plant was nominally 30 MWth and 10 MWe, and included thermal storage.

Central receiver solar power plants are projected to become a very cost effective form of solar thermal energy generation, surpassing parabolic trough technology, according to [6]. This is due, among other factors, to the fact that the radiation flux reaching the central receiver is often in excess of 1000 suns (> 1 MW/m^2), which allows for much higher temperatures to be reached than with other technologies. The state of the art of central receiver power plants consists of liquid cooled receivers. These central receivers pump a liquid – either a heat transfer fluid (HTF) or simply water – through the receiver. If using an HTF, a heat exchanger is used to heat and boil water to drive a steam turbine. If water is used in the receiver, this water is converted into high temperature steam directly in the receiver, and then expanded through a turbine to generate electricity. To date, all central receiver power plants run exclusively on Rankine (i.e., steam turbine) cycles.

The development of central receivers capable of powering gas turbine engines offers several advantages over competing steam turbines. First, water usage is a major concern for new solar power plants because the ideal locations are often located in dry regions already subject to water shortages. Some current central receiver power plants are forced to use dry cooling methods, which add to the overall cost and decrease the efficiency of the system. A gas turbine engine cycle offers obvious major advantages in this area. In addition, if water usage is not a concern, a gas turbine can potentially be coupled with a bottoming Rankine cycle to achieve superior energy conversion efficiency compared to a stand-alone Rankine cycle [4]. In addition to reduced water requirements, gas turbines are generally easier to operate than steam turbines, and are expected to withstand more stops and starts. As such, they are better suited to the intermittent nature of solar energy, which can require nightly shutdown. The solar receivers for heating air to high temperature, while still under development, offer the possibility of higher cycle efficiencies compared to lower temperature liquid cooled receivers.

MECHANICAL CONSIDERATIONS FOR SOLAR POWERED GAS TURBINES

Two material constraints are dominant in dictating the performance and design of a solar powered gas turbine engine. These are the combustor liner and receiver window (certain types of air-cooled solar receivers require a window to admit the concentrated radiation, but not all do). In a normal gas turbine engine, the combustor liner is actively cooled since air entering the combustor is much cooler than the combustion products: as compressed air enters the combustion chamber, and before it is mixed with fuel or combustion gasses, it generally flows across the combustor liner, providing convective cooling. Depending on the specific material used in the combustor, the liner temperature should remain below about 850 C [7]. The cooler the liner remains, the longer its lifespan. Therefore it is advisable to keep the liner as cool as possible.

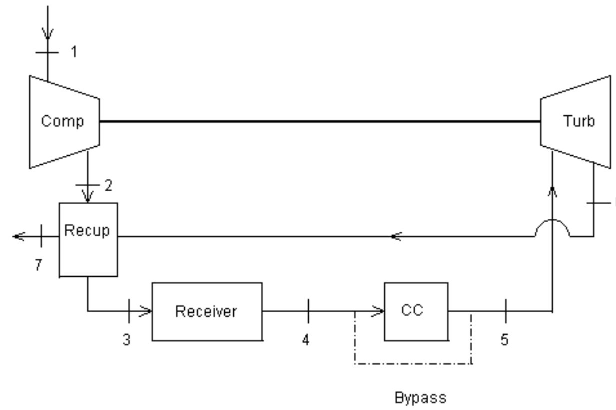

FIGURE 2. RECUPERATED GAS TURBINE CYCLE WITH SOLAR RECEIVER AND COMBUSTOR.

When facing the development of a solar receiver to power a gas turbine engine, most design strategies involve constructing a sealed, transparent (quartz) window to allow sunlight to enter into the receiver cavity [2,8]. To avoid overheating the window, then, the air entering the receiver cavity should not be significantly preheated, since the window temperature should remain below about 800 C if using quartz [1], and much cooler for alternative glasses. Alternative designs involve using black surfaces such as tubes to absorb radiation [2]. These designs also benefit from lower inlet temperatures as this reduces both thermal stresses and radiant losses.

With these constraints in mind, Figure 2 shows the thermodynamic cycle presented in this paper, although the general strategy could easily be applied to other gas turbine cycles. The cycle is a recuperated gas turbine engine, wherein the recuperator is followed immediately by the solar central receiver, and then by the combustion chamber. Note the presence of the combustor bypass, which can be activated to keep the flow of hot air from entering the combustor. The thermodynamic effects of this bypass will be shown later in this paper.

A single shaft, recuperated gas turbine engine is used for the analysis because it provides several advantages over other turbines, and it introduces a level of complexity (to be described later) not present in a simple cycle gas turbine. In addition, recuperators are commonly installed on smaller gas turbine units, and these smaller units present the best opportunity for the initial commercialization of these solar technologies. As shown in [9], the simple cycle gas turbine is intrinsically less efficient in solar applications than the recuperated cycle. Single shaft gas turbines are most commonly used for power generation, and it is for this reason that the single shaft engine was analyzed.

The recuperated gas turbine engine has a high maximum efficiency at a low pressure ratio, which is advantageous for solar receivers since this decreases stresses on the window or tube, depending on the type of receiver used. The performance of a standard recuperated gas turbine engine is shown in Figure 3, and is generated by the commercially available program GasTurb11 [10] described in the following section. Figure 3 shows thermal efficiency versus pressure ratio for a recuperated gas turbine, operating at constant mass flow. The solid lines represent different turbine inlet temperatures, and the dashed ones separating the different colors are contours of shaft power delivered. The shaft power is given in kW, labeled by the red text along each contour line.

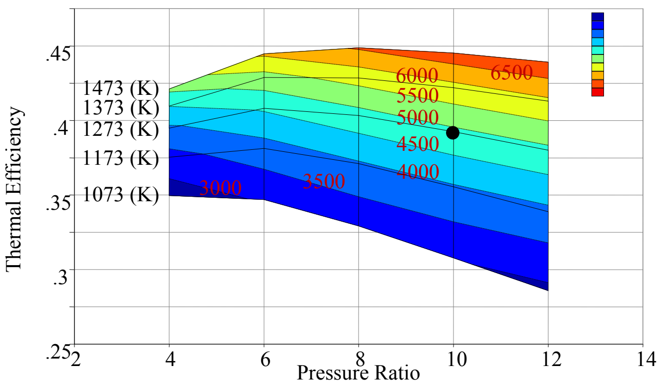

FIGURE 3. DESIGN POINT EFFICIENCY OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF PRESSURE RATIO.

Note that maximal efficiency and maximal power do not take place at the same combination of pressure ratio and turbine inlet temperature: in general, more power can be provided by increasing the pressure ratio. In this sense, a large increase in power (and thermal input) can be accomplished by raising the pressure ratio and accepting a relatively minor decrease in efficiency. However, it should be noted that this strategy is complicated in a solar powered system, because the input to the system is fixed by the available sunlight. In this case, greater efficiency implies increased power. Therefore, a high efficiency turbine is desirable if one is to use solar energy as the primary power source.

The black dot represents the design point chosen for modeling. The design point parameters are shown in Table 1. Note the pressure ratio of 9.9, which is not consistent with optimal efficiency, but generally reflects actual working conditions of the Mercury 50 recuperated gas turbine manufactured by Solar Turbines [11], which was chosen as a base point. As many of the component properties and performance characteristics remain proprietary, however, these were assigned reasonable values. As such, this modeling and the results reported here do not represent that of the actual Mercury 50, but a reasonable approximation thereof in terms of mass flow and power output.

Parameter	Value	Parameter	Value
Mass Flow	20 kg/s	**Pressure Ratio**	9.9
Isentr Comp Eff	.85	**Isentr Turb Eff**	.89
HX Effectiveness	.8	**Burner Exit Temp**	1273 K
d(M)/d(VGV°)	.01	**d(PR-1)/d(VGV)**	1
d(Efficiency)/d(VGV°)^2	.01	**Component Pressure Losses**	
Net Power	4.8 MWe	**Efficiency**	.385

TABLE 1. IMPORTANT PARAMETERS FOR MODELING THE GAS TURBINE ENGINE.

MODELING AND ENGINE PARAMETERS FOR SOLAR POWERED GAS TURBINES

Ambient conditions are assumed to be for a standard day temperature and pressure (288.15 K and 101.325 kPa). Of course, ambient temperature changes throughout the year, but these conditions were kept constant in an effort to evaluate the effect of solar input alone.

A single shaft engine is used in this calculation, so that the engine speed does not change with operating conditions. The engine speed is maintained, rather, by adjusting the electrical load placed on the generator. In this way, compressor and turbine speed remain constant, even while operating temperature, pressure ratio, and gas properties all vary. Of course, referred parameters such as referred turbine tip speed, are still adjusted according to the working conditions.

Since the availability of solar input changes markedly over the course of the day and time of year, it is essential to understand the performance of the gas turbine engine as a function of solar thermal input, rather than just ambient temperature, for instance. This is because the amount of heat entering the system cannot always be controlled, and even when it is (during hybridization) the performance of the gas turbine can change markedly simply due to variation in gas thermophysical properties.

In order to model the aerodynamics of the compressor and turbine, a commercial code called GasTurb11[10] was used to generate appropriate component maps scaled to the correct power output of 4.8 MWe with a turbine inlet temperature (TIT) of 1000 C. This temperature, while lower than that of a typical gas turbine, is a reasonable temperature to expect from a first generation commercial solar central receiver. TIT's in the range of 1100 – 1200 C are still far in advance of what has been demonstrated for any large scale prototype solar receiver.

Once the component maps are developed by GasTurb11, they can then be used within a MATLAB code developed by the authors [12] to model the thermodynamic performance of solar powered gas turbines. The MATLAB code is linked to the FORTRAN chemical equilibrium code CEA developed by Gordon and McBride of NASA to calculate gas properties [13]. This ensures accurate gas thermodynamic properties with or without combustion.

A detailed explanation of the modeling procedure is provided in the Appendix. To summarize, all components are assigned design point pressure drops and, if applicable, temperature limitations. Off-design pressure drops are assumed to vary with the square of the ratio of corrected flow to the design point flow (see Equation A2). Off-design recuperator performance is also calculated.

To match the compressor and turbine flows, the component maps, along with the pressure drops in the system exhaust, are used to ensure that conservation of mass is satisfied, and that the pressure leaving the exhaust stack matches atmospheric conditions.

Since gas properties are easily calculated from the CEA program code, the corrected mass flow at all points is calculated using the formula provided by [10]:

$$W_{corr} = \dot{m} * \frac{\sqrt{T * \frac{R}{\gamma}}}{\delta} \tag{1}$$

Since a specific gas turbine engine with variable guide vane geometry was not available, the effects of scheduling the guide vanes were estimated using the default values given in the GasTurb11 manual [10]. The guide vane angle affects efficiency, mass flow, and pressure ratio, which serves to alter the appearance of the component map. For each guide vane angle, the map is scaled according to the coefficients in Table 1.

To summarize the modeling procedure, a MATLAB code was developed which analyzes the thermodynamic performance of the gas turbine as a function of solar thermal input [9]. Gas properties are calculated from a chemical equilibrium code, and turbine component models developed by GasTurb11 are imported and used. No bleeding schedule is included to cool the nozzle guide vanes (NGV's) or first stage rotor.

PERFORMANCE OF A SOLAR POWERED RECUPERATED GAS TURBINE ENGINE

Variable Guide Vanes

Variable guide vanes (VGV's) on a gas turbine engine can be used to control the mass flow rate of air passing through the engine [14]. When this is done, the compressor (and turbine to a much smaller degree) suffers a decrease in isentropic efficiency due to the altered aerodynamics of the system. However, the behavior of these components does not dictate whether or not the total system efficiency will decrease. For example, opening the VGV's can change the pressure ratio of the cycle, which can lead to a greater expansion ratio through the turbine, and increased power output.

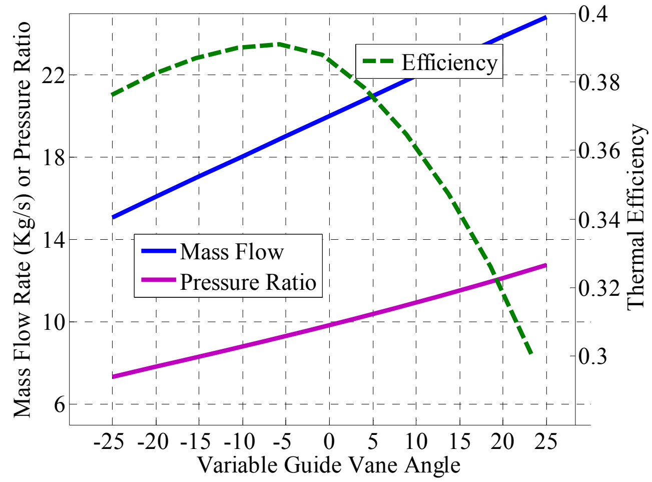

FIGURE 4. OFF-DESIGN EFFICIENCY OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF VARIABLE GUIDE VANE (VGV) ANGLE (CONSTANT TIT).

For a fixed level of solar thermal input, and assuming no hybridization, decreasing the VGV angle, and consequently the mass flow, will increase the temperature of the gas exiting the solar receiver. This, in turn, may increase system efficiency, by increasing its Carnot efficiency, meaning it simply operates at a higher maximum temperature. This is of course limited by the material constraints on the turbine itself (there exists a maximum receiver/combustor outlet temperature above which turbine lifespan decreases rapidly).

Figure 4 shows the effect of variable guide vane operation on a single shaft, recuperated gas turbine engine operating at constant turbine inlet temperature. Mass flow rate, pressure ratio, and cycle efficiency are all shown. When the TIT is held constant, the efficiency decreases as the VGV angle moves away from zero, since this corresponds to decreases in aerodynamic efficiency and changes in pressure ratio. Note that the efficiency actually shows a slight initial increase with a negative change in VGV position. This is caused by the pressure ratio decreasing towards its optimum value as shown in Figure 3. This increases system efficiency (because of increased recuperator heat transfer) even though the compressor itself becomes less efficient. This description should provide evidence of the complex nature of gas turbine performance in a recuperated system with variable guide vanes and a variable heat source.

Hybridization

Hybridization involves the combustion of fossil fuels to supplement solar thermal input [3,9,12]. The implementation of hybridization into a real system is difficult due to material limitations and huge variations in fuel flow rate and combustor inlet temperatures which can require specialized combustor designs. The issue of combustor design to accommodate a large variation in fuel flow rate and inlet temperature is not discussed here in detail.

Figure 5 shows the efficiency of two single shaft recuperated gas turbine engines as a function of solar thermal input, and Figure 6 shows the net power of these same engines. The dashed black line represents a system without a bypass valve. As can be seen, the efficiency of this configuration changes by about 1% over the course of solar thermal input. These changes are due to changes in gas composition, system pressure drops, off-design recuperator performance, and changes in mass flow due to decreased fuel input and changes in compressor performance[1]. This

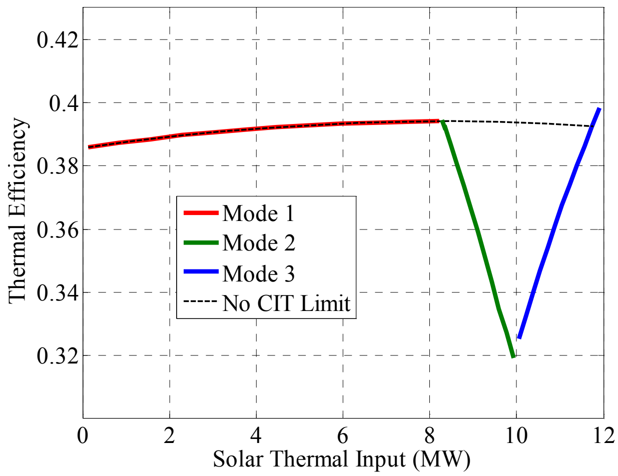

FIGURE 5. EFFICIENCY OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT.

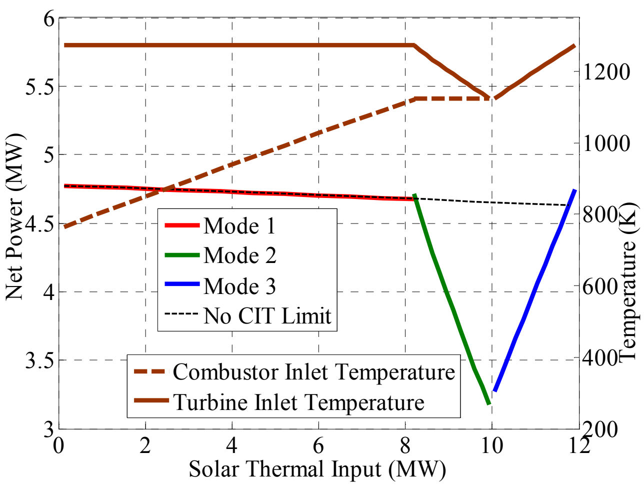

FIGURE 6. POWER OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT.

configuration requires a combustor specially designed and cooled so that it can withstand inlet temperatures approaching 1000 C while still combusting fuel to add heat to the system. In the region between 8.25 and 11.75 MW of thermal input, this system is more efficient and produces more power than the one which uses a bypass valve, indicated by the multi-colored solid line.

One basic issue investigated in this paper, with regard to hybridization, is whether or not it is necessary to redesign the combustor to accommodate these large temperature fluctuations, or if it suffices thermodynamically to install a standard combustor with a bypass valve to prevent it from overheating during periods of high solar thermal input. In order to do that, the effect of limiting the combustor inlet temperature to 850 C has been investigated heavily, as this temperature represents the upper limit of a working combustor liner temperature [7]. When a limit is imposed on the combustor, the recuperated cycle experiences three distinct modes of operation separated by the solar thermal input into the system, as first described in [9].

Mode 1 operation begins when the solar input to the system is low. During this time, the combustor inlet temperature is also low, and the system operates at its full potential: that is, sufficient fuel is added to the system to raise the temperature of the gas exiting the combustor to the design point TIT. However, as the solar input continues to increase, the temperature of the air entering the combustor approaches 850 C (Figure 6). At this point in time, Mode 2 operation begins (at 8.25 MWth in this example). During Mode 2, the combustor inlet temperature remains constant at 850 C, despite the fact that more solar energy is added to the system [9]. This is accomplished by actively controlling the amount of fuel added to the system, which in turn controls the TIT: A lower TIT will correspond to a lower turbine exhaust temperature (even though the pressure ratio is lower) and hence, due to the presence of the recuperator, a lower receiver inlet temperature. This, in turn, controls the combustor inlet temperature. Note that in Mode 2 operation, the temperature of the gas leaving the receiver is not controlled by scheduling the VGV's, but only by controlling the mass flow rate of fuel[2].

Finally, Mode 3 operation represents when only solar energy is used to drive the turbine. This occurs at solar input above about 10 MW. In Mode 3, an increase in solar input leads to an increase in temperature and hence efficiency. The maximum solar thermal input is limited by the maximum TIT for the gas turbine.

Figures 5 and 6 show a jump in thermal efficiency and power between Mode 2 and Mode 3 operation: this is caused by engaging the combustor bypass, which is assumed not to have an associated pressure drop. This leads to an increased expansion ratio through the turbine, generating more power and leading to a more efficient cycle. Therefore, for the same compressor and turbine, the two systems have different design point power outputs and thermal input. For the case of

[1] Compressor performance is dictated by corrected speed and a non-physical parameter β. β correlates pressure ratio to mass flow and efficiency at a given speed, and in off-design conditions changes to ensure component matching.

[2] In Mode 2, sufficient fuel could *theoretically* be added to raise the temperature of the gas to the design point TIT. However, this would defeat the point of the combustor bypass, since the temperature of the air entering the combustor would be above the maximum value.

a system with a bypass, the design point solar-only performance (Figure 6, end of blue line) is 4.75 MWe at 11.8 MWth solar input. For the case of system without a bypass (pressure drop across the combustor, Figure 9) the design point solar-only performance (Figure 6, end of black dashed line) is 4.63 MWe at 11.8 MWth solar input.

To further clarify Mode 2 operation, Figure 7 shows the thermal input from fossil fuel combustion as a function of solar thermal input. During Mode 1 operation, the slope of the line is -1.04, because as solar thermal input increases, the fossil fuel input must decrease by almost that same amount.

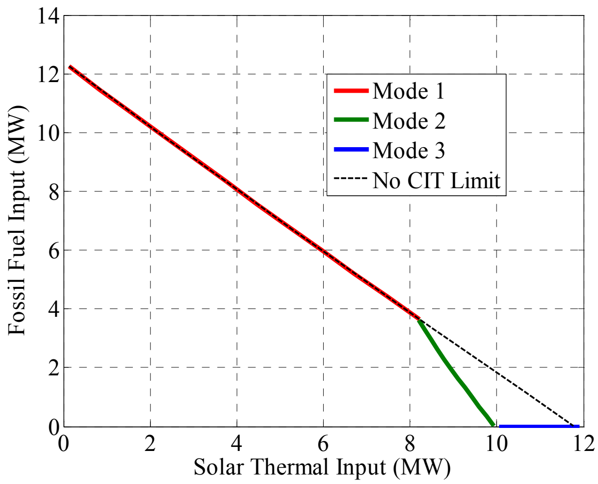

FIGURE 7. FOSSIL FUEL INPUT OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT.

(The reason for the -1.04 rather than -1.00 is due to changing gas properties which dictate that the specific heat of combustion products (mainly CO_2 and H_2O) is higher than that of air in the temperature range of interest. Consequently, as solar input continues to increase, the amount of total energy required to heat the gas is slightly less.) During Mode 2, the slope decreases so that any increase in solar input is met by a greater decrease in fossil fuel input, in order to keep the operating temperature within bounds. The slope during Mode 2 is -2.00. This is a function (among many other things) of the effectiveness of the recuperator [9], since how much heat is recycled back into the system depends on this device.

Interaction between Hybridization and Variable Guide Vanes

In a solar powered gas turbine system, there is the opportunity to combine variable guide vane operation with hybridization. One may wish to increase power at the expense of efficiency by opening the guide vanes, or, at other times, to increase both power and efficiency by scheduling the VGV's according to some optimized route. The conditions under which this should take place are not immediately obvious, and are presented below. Only power and efficiency calculations are presented, without relation to the economics or other aspects of the operational strategies.

Solar Hybrid Operation of a Variable Geometry Gas Turbine without a Bypass Valve

In this section we consider a variable geometry, single shaft recuperated gas turbine with a combustor designed to accommodate large fluctuations in inlet temperature up to 1000 C. Since the turbine inlet temperature is constant in this configuration, the efficiency of the system remains nearly constant while the solar input changes. With such a system it is possible to *guarantee* a power output from the solar plant at the rated power output of the engine. This system, then, is completely dispatchable, in that it can operate independently of the time of day or solar thermal input. As will be shown in later sections, however, operating such a system for 24 hours per day without storage leads to an annual solar share around 29%, which may be unacceptable.

Figures 8 and 9 present the thermal efficiency and net power of this cycle as a function of solar thermal input. The multiple lines represent different angles for the variable guide vanes. The red lines, which begin to fall on top of each other in Figure 8, represent Mode 1 operation, when the combustor is used to bring the temperature of the working gas to the design point turbine inlet temperature. For a given solar input, some of the red lines collapse to a single maximal efficiency about an inlet angle of 0 degrees (black line), indicating that a small change in guide vane angle (positive or negative) does not significantly affect efficiency in this regime. Since the combustor inlet temperature is not limited, there is no Mode 2 operation.

In Figures 8 and 9, Mode 3 (i.e., solar-only operation) is indicated by the blue lines. The highest point on each blue line, which represents both maximal power and maximal efficiency for that guide vane angle during solar only operation, corresponds to a TIT of 1000 C, the design point for the cycle. With a VGV angle of -25 degrees, the design point TIT is reached with a solar thermal input of about 8.65 MW. With a VGV angle of +25 degrees, the maximal solar thermal input is about 14.9 MW, which gives a variation in input of 6.25 MWth for Mode 3 operation. At 0 degrees, the design point solar thermal input for Mode 3 is about 11.8 MW. These values indicate that the VGV's allow the engine to accommodate a large variation in solar thermal input.

Of course, as the VGV angle continues to increase to allow greater mass flow to the system, aerodynamic inefficiencies increase, and the pressure ratio of the cycle changes dramatically. Because the TIT is presumably constant during Mode 3 (the VGV's are adjusted to keep the TIT at 1000 C, maximizing both power and efficiency at a given solar input) the efficiency must decrease as the solar input increases beyond the design point due to greater compressor inefficiencies caused by opening the guide vanes. Even though efficiency decreases with solar input above the design point, Figure 9 demonstrates that the net power continues to increase with solar thermal input up until the VGV angle reaches about 15 degrees. Beyond this value, the net power begins to decrease with increasing solar input due to the large aerodynamic inefficiencies in the compressor, and the fact that the pressure ratio strays farther from its design point value. Isentropic inefficiencies in the compressor act twofold on the system, first requiring more power to drive the compressor, and second lowering the heat transfer in the recuperator by increasing the compressor outlet temperature.

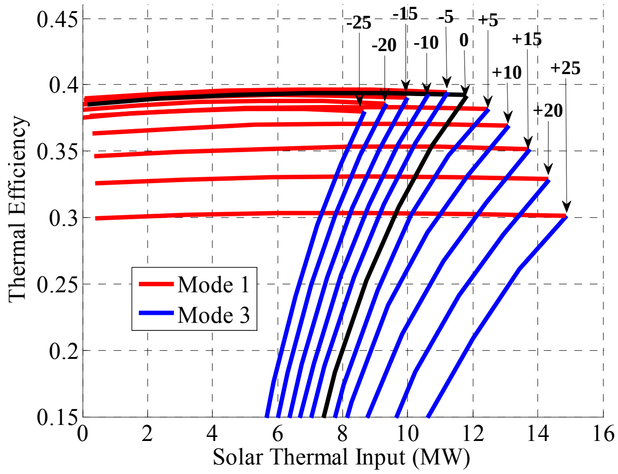

FIGURE 8. EFFICIENCY OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT. NO BYPASS (HIGH COMBUSTOR INLET TEMP.).

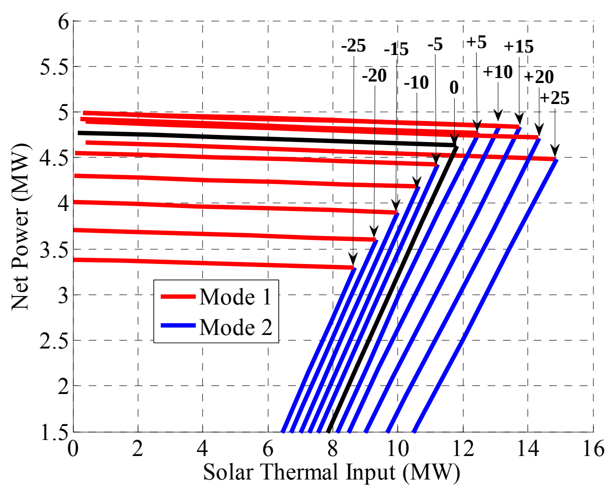

FIGURE 9. NET POWER OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT. NO BYPASS (HIGH COMBUSTOR INLET TEMP.).

Even in the case of a flexible combustor, deciding how to schedule the VGV's is not obvious, since Mode 1 and Mode 3 intersect from 8.65 MWth to 14.9 MWth. The answer to scheduling lies in the economics of the power plant: whether it is more profitable to burn extra fuel to generate extra electricity, or conserve fuel and run purely off solar energy when that opportunity is present. The best strategy probably lies somewhere in between. Efficiency can be maximized in Mode 1 by keeping the VGV angle set at 0 degrees. During medium to high design point levels of solar thermal input, the amount of fossil fuel compared to solar energy is low. The question of daily and annual solar share will be resolved in later sections.

When the VGV angle is set to zero, the power delivered is about 4.6 MWe at 11.8 MW of solar input, corresponding to essentially no fossil fuel input. At zero solar thermal input (purely fossil fuel), the power delivered is 4.77 MWe, indicating a 3.6% drop in power due largely to changes in gas composition when running in pure solar mode. This change in power can be counteracted by opening the guide vanes, but this would further decrease efficiency, as shown in Figure 8. Once 11.8 MW of solar input is reached, the guide vanes should then be opened to keep the turbine inlet temperature constant. The guide vanes should remain open as long as the solar input exceeds 11.8 MW in order to take advantage of this "extra" power delivered to the solar receiver.

Solar Hybrid Operation of a Variable Geometry Gas Turbine with a Bypass Valve

Figures 10 and 11 show the power and efficiency of a solar hybrid gas turbine with the combustor inlet temperature limited to 850 C. Figure 10 shows how the gas turbine engine operates at different VGV angles. Again, we see the presence of Modes 1, 2 and 3, indicated by red, green and blue lines. The different connecting lines represent different variable guide vane angles, and the solid black lines represent when the VGV angle is zero. For each angle, then, there are corresponding regions for Modes 1, 2 and 3.

To clarify Figure 10, consider the lowest red line, which starts at an efficiency of about .3, and indicates a VGV angle of +25 degrees. This angle corresponds to a mass flow rate of about 24.8 kg/s, compared with 20 kg/s at 0 degrees. The increase in mass flow delays the onset of Mode 2 (indicated by the green line commencing at the end of the red line, at around 10.4 MW). This delay is because more solar thermal input is required to increase the combustor inlet temperature to 850 C. At +25 degrees, then, Mode 2 spans from 10.4 MW to 12.3 MW. Once the solar thermal input increases beyond 12.3 MW, solar only operation begins, indicated by Mode 3. Since the VGV angle is fixed at +25 degrees in this example, the efficiency continues to increase with increasing solar input up until the maximum TIT is reached, at which point the system can no longer accept an increase in thermal input.

Considering further the results presented in Figure 10, it is remarkable to note that if one wishes to maximize efficiency for all levels of solar input, then the efficiency would never fall below .385, even though the combustor inlet temperature is still limited to 850 C. Considering that the design point efficiency in Mode 3 for a VGV angle of zero degrees is about .385, this operational strategy has no detrimental effect on cyclic efficiency. In order to accomplish this, the variable guide vanes and fuel flow must be precisely regulated, and in some cases change quickly between open (+5 degrees) to mostly closed (-25 degrees). This particular transition takes place at a solar input of about 8.75 MW, and is shown further in Figure 10.

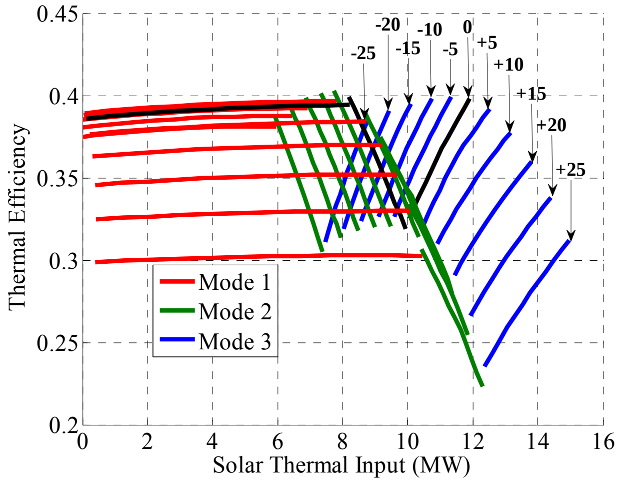

FIGURE 10. EFFICIENCY OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT. WITH BYPASS (850 C COMBUSTOR INLET TEMP.).

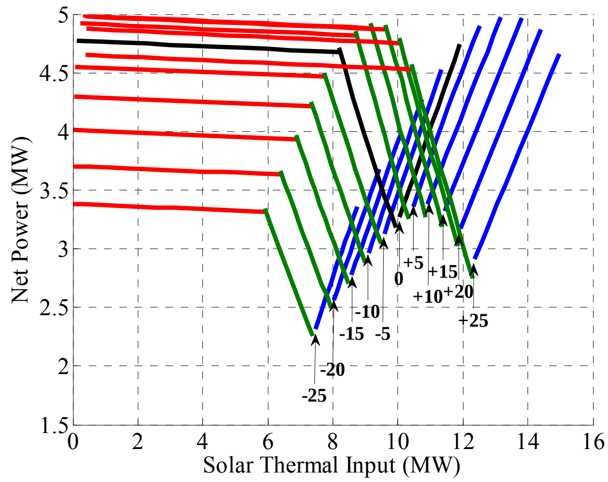

FIGURE 11. NET POWER OF THE RECUPERATED GAS TURBINE ENGINE AS A FUNCTION OF SOLAR THERMAL INPUT. WITH BYPASS (850 C COMBUSTOR INLET TEMP).

Figure 11 shows the net power as a function of solar thermal input. If we wish to maximize power for all levels of solar thermal input, then the net output would never fall below about 4.3 MWe, which is 9.5% below the Mode 3 design point output of 4.75 MWe. This minimum takes place at about 10.75 MW solar thermal input. Therefore, if we seek to maximize power *or* efficiency with a limited combustor inlet temperature, the overall effect on either value is quite minor.

The complication arises when one attempts to choose the optimal strategy, that is, one that somehow maximizes both power and efficiency. To define the difficulty fully, consider the options at a solar thermal input of just over 8.75 MWth. In order to maximize efficiency in this case, Figure 10 dictates that the VGV angle should be set to -25 degrees. This corresponds to a mass flow rate of 15.1 kg/s, and Mode 3 operation, meaning only solar input is used. The efficiency at this point is .385. However, looking at Figure 11, the corresponding power is about 3.36 MWe, or about 30% lower than the design point. This represents a much greater drop than expected from Figure 11.

The next question is whether there exists a strategy to keep output high without sacrificing too much in efficiency. Continuing with the previous example, in order to maintain 4.3 MWe output, we cannot operate in Mode 3, due to insufficient solar input. From Figure 11 we conclude that in this case, Mode 2 operation is required with a VGV angle of about +3 degrees. Returning to Figure 10, the corresponding efficiency is about .375. While more fossil fuel is added in this case to ensure sufficient power, the modest decrease in efficiency from .385 to .375 is likely compensated for by the increase in net power from 3.36 MWe to 4.3 MWe, a much more significant change.

Operation Strategies for Solar Hybrid Gas Turbines with Variable Geometry

Figures 12 and 13 show the operational strategies chosen for the cases of a bypass and no bypass. In the case of a bypass, the strategy involves keeping the power output at or above 4.3 MWe. The transition to Mode 2 takes place at just over 8.2 MW solar thermal input. From there, the VGV angle of zero degrees is kept until the output decreases to 4.3 MWe. At this point, the VGV's begin to open to keep the power constant. Efficiency decreases rapidly during this period, as seen in Figure 13. A solar thermal input of about 10.7 MW sees a singularity in the efficiency graph in Figure 13. This represents the closure of the guide vanes from Mode 2 operation to Mode 3. There is no singularity in the power graph, since it remains constant at 4.3 MWe. As the solar thermal input continues to increase beyond 10.7 MWth, the VGV's open to keep the TIT constant at its design point.

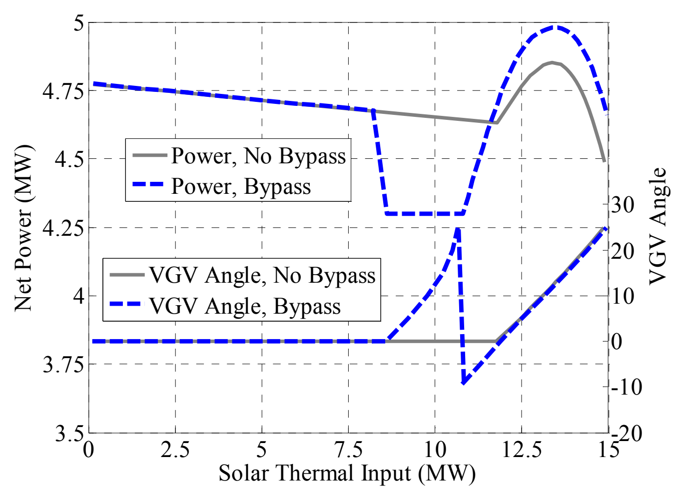

FIGURE 12. EXAMPLE OPERATIONAL STRATEGY FOR THE RECUPERATED GAS TURBINE ENGINE TO KEEP OUTPUT POWER ABOVE 4.3 MWe.

As shown in Figures 12 and 13, between 8.25 and 10.7 MW of solar thermal input, the system with no limit on the combustor inlet temperature (no bypass) out-performs the system incorporating a bypass valve. However, after the design point solar thermal input of 11.8 MW, the bypass system produces more power at higher efficiency due to the increased expansion ratio through the turbine. The significance of this on an annual basis will be discussed in the next sections. Note that for the case of no bypass, the VGV angle remains constant at zero degrees until 11.8 MWth is reached, after which the VGV's open to accommodate the increase in energy into the system.

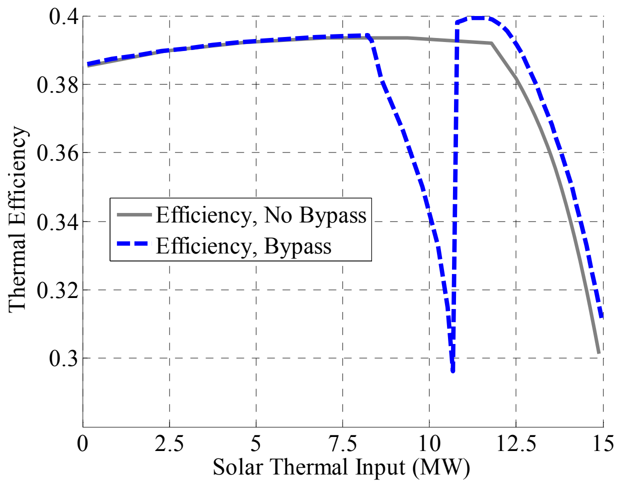

FIGURE 13. EFFICIENCY FOR THE OPERATIONAL STRATEGY SHOWN IN FIGURE 12 FOR THE RECUPERATED GAS TURBINE ENGINE.

In Figure 12, we see that the net power actually decreases beyond about 13.5 MW of solar input. This means that whenever this value is exceeded, performance suffers. Consequently, when annual performance is calculated (in the next section), it is assumed that for any available solar input in excess of this amount, mirrors are stowed so that performance is maximized. This leads to an increase in "wasted" power from the heliostat field, but also increases annual performance.

Use of the Solar Advisor Model (SAM) to Estimate Annual Performance

The Solar Advisor Model (SAM) is a solar energy evaluation tool developed by the National Renewable Energy Laboratory in order to assess different solar based technologies on a level playing field [15]. SAM can calculate optimal heliostat field layout, which can then be used to calculate the total power delivered to the solar receiver on an hourly basis for the entire year. Using this information, SAM can calculate the total power produced from a traditional central receiver power plant, and estimate costs and the levelized cost of energy (LCOE) produced at the solar power plant. However, SAM is not equipped to handle a gas cooled solar receiver and the associated power block (i.e., a Brayton Cycle). As such, any economic predictions are invalid, due to the many differences between gas and steam turbines and solar receivers. Therefore, SAM is utilized here solely as a tool to estimate the thermal energy delivered to the solar receiver. The solar receiver chosen is a cavity type receiver, meaning that its acceptance angle is 120 degrees (not a surround heliostat field). It is assumed that all of the thermal energy delivered to the receiver is used in the gas turbine cycle in order to compare the effect of a bypass to a system with no combustor bypass. This eliminates solar receiver efficiency as a variable in the analysis, something not considered here. The relevant parameters used in SAM are shown in Table 2.

Parameter	Value	Parameter	Value
Location	Dagget, CA	Receiver Type	Cavity
Tower Height	60 m	Receiver Width	4.66 m
Receiver Height	2.85 m	Design Power Output	4.5 MWe

TABLE 2. IMPORTANT PARAMETERS FROM SAM. ALL OTHER VALUES ARE UNCHANGED FROM THE 100 MWe TEMPLATE PROVIDED IN SAM.

Annual Performance of the Recuperated Gas Turbine without a Bypass

Figure 14 shows the performance of the gas turbine as a function of time during a summer day assuming 24 hour plant operation. The green bars represent solar thermal input, and the red bars fossil fuel input. The blue line at the top of the graphs shows how the power output changes with solar input. The solar share can be inferred from the relative height

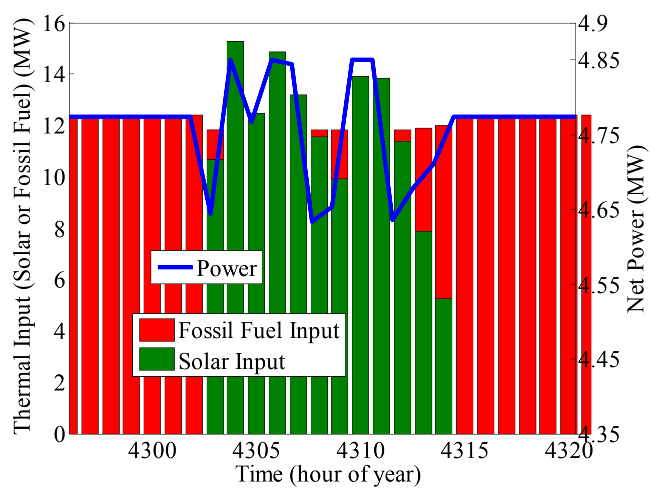

FIGURE 14. POWER, SOLAR THERMAL INPUT, AND FOSSIL FUEL INPUT OVER A SUMMER DAY FOR A SOLAR FIELD MULTIPLIER OF 1.25. NO BYPASS.

of the red and green bars. The design point solar thermal input is 11.8 MW. This value is repeatedly exceeded for several hours during this hot summer day. As the solar thermal input increases beyond 11.8 MW, the solar share remains at 1, while the net power continues to increase. As will be shown, a solar multiple of 1.25 limits the amount of solar energy wasted on an annual basis, because while the solar input is above the design point, the flexibility of a variable geometry compressor allows for compensation. During morning and evening, the system is in hybrid operation, meaning that the solar share is less than 1, and natural gas is burned to keep the TIT at its design point.

In Figure 14, the power of the gas turbine often drops when the solar thermal input increases. Recall that this is not due to any aerodynamic inefficiencies (since the VGV angle is 0 up until Mode 3) but rather due to changes in the gas properties. That is, this change is an unavoidable consequence of operating off solar thermal input. Power ranges from 4.8 MWe at zero solar input to 4.98 MWe at 100% solar input (13.5 MWth).

Table 3 summarizes the annual performance of the gas turbine in terms of average efficiency, average power and annual solar share for different solar field multipliers (SFM). The solar field multiplier dictates the thermal energy the receiver is capable of accepting, and consequently, the size of the heliostat field. For an SFM of 2, for example, the maximum thermal energy delivered is twice that of the design point input of 11.8 MW. Of course, the receiver cannot actually handle this much solar input because it would overheat, and so it is assumed that some mirrors are stowed during this time. The amount of solar energy wasted due to shutting down the mirrors is calculated. The wasted energy is given in terms of days of operation wasted at the design point of 4.63 MWe (corresponding to 11.8 MWth input.). For example, with a SFM of 1.5, enough solar radiant energy is wasted throughout the year to power the turbine for 11.8 twelve hour days. To calculate the solar share at 24 hr/day operation, divide the given value by two.

SFM	Annual Efficiency	Average Power (MWe)	Solar Share	Energy Wasted (12 hr days of operation)
1	.389	4.71	.58	0
1.25	.381	4.76	.65	11.9
1.5	.376	4.79	.69	44.5
1.75	.373	4.80	.71	87.4
2	.371	4.81	.72	135

TABLE 3. ANNUAL EFFICIENCY AND POWER, SOLAR SHARE, AND WASTED ENERGY FOR 12 HR/DAY OPERATION. NO BYPASS.

SFM	Annual Efficiency	Average Power (MWe)	Solar Share	Energy Wasted (12 hr days of operation)
1	.391	4.68	.78	0
1.25	.379	4.75	.89	11.9
1.5	.372	4.79	.94	44.5
1.75	.373	4.81	.96	96.1
2	.370	4.83	.98	135

TABLE 4. ANNUAL EFFICIENCY AND POWER, SOLAR SHARE, AND WASTED ENERGY FOR DAYLIGHT OPERATION. NO BYPASS.

If the operational hours are limited to when only sun is present, then the solar share will be significantly increased. The detrimental effect of oversizing the heliostat field is given by the total energy wasted. It is clear that oversizing the field increases the solar share at a slow rate, but the amount of energy wasted increases quickly. Therefore, a reasonable value for the SFM is around 1.25, where the waste is minimal, but the solar share is significantly better than the SFM = 1 case.

The annual efficiency generally decreases with increasing SFM. Since the efficiency is almost constant over the range from 0 to 11.8 MWth, this is caused primarily by the turbine operating with an excess of solar thermal input. With more energy in excess of the design point, the variable guide vanes open to allow more mass flow and produce more power, but lower efficiency. For this reason, *annual efficiency is not a useful parameter in that a lower efficiency may indicate increased solar share, increased power, and thus lower costs*. The net power increases with SFM because the turbine operates with open VGV for a greater period of time.

Table 4 shows how the same parameters change with SFM when the gas turbine is operated only during daylight hours. At this location, this corresponds to an average of about 8.5 hrs/day of operation (note that this varies throughout the year). When this is done, the solar share increases to .89 at a SFM of 1.25. This would be appropriate to meet with most regulatory constraints (since its less than 15% fossil fuel input). The average power and efficiency only decrease slightly from the case of 12 hr/day operation.

Annual Performance of the Recuperated Gas Turbine with a Bypass

Again, Figure 15 shows the performance of the gas turbine during the same summer day as shown in Figure 14, this time with a combustor bypass. The differences between the two systems can clearly be seen in the hours wherein the solar share is less than unity. The net power, in this case, dips down to 4.3 MWe, which is the minimum value guaranteed by this system. The efficiency (not shown) during this time is also lower. However, for the solar share of 1 (solar input at or above 11.8 MWth), or whenever the solar input is less than 8 MWth, the two systems are identical. During the off design performance, the solar share may be either higher or lower than that of the alternative system with the same solar thermal input, because the net power is less, and therefore slightly less total thermal input is required.

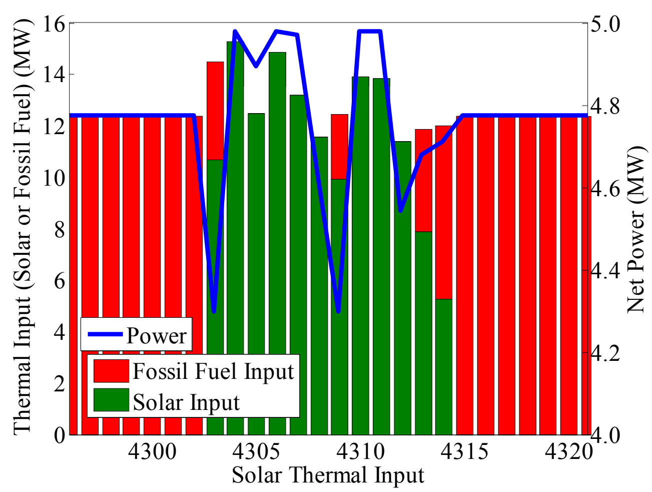

FIGURE 15. POWER, SOLAR THERMAL INPUT, AND FOSSIL FUEL INPUT OVER A SUMMER DAY FOR A SOLAR FIELD MULTIPLIER OF 1.25. WITH BYPASS.

The operation strategy first presented in Figures 12 and 13 is demonstrated in Figure 15, and highlights some of the key drawbacks which arise from the installation of a bypass system. In particular, hours 4303 and 4309 share the net power (4.3 MWe), and the solar inputs are 10.6 and 9.9 MWth, respectively. However, the total energy input (height of red plus green bars) in hour 4303 is substantially higher, and the fossil fuel input is almost the same. This is due to a severe decrease in efficiency from 8.5 MWth input to 11 MWth input, as shown in Figures 12 and 13. In between these two values of solar input, net power remains constant, but efficiency decreases sharply, requiring a large increase in thermal input to maintain that output.

Table 5 shows the annual performance of the bypass system for different values of SFM. Comparing the values from Table 3 and 5, we see that there is not a significant difference between the two systems on an annual basis. Taking SFM = 1.25, the power produced annually, and the average solar share, is equal to that for the no-bypass case. The two systems waste similar amounts of energy. It should be concluded, then, that on an annual basis operating at twelve hours per day there is not a significant difference between the two approaches (bypass vs. increasing combustion inlet temperature). Therefore, it is no longer a question of thermodynamics, but rather one of mechanical design and O&M costs associated with each approach.

SFM	Annual Efficiency	Average Power (MWe)	Solar Share	Energy Wasted (days of 12 hr operation)
1	.381	4.62	.57	0
1.25	.380	4.76	.65	10.7
1.5	.376	4.83	.69	42.4
1.75	.375	4.86	.71	84.8
2	.375	4.88	.72	132

TABLE 5. ANNUAL EFFICIENCY AND POWER, SOLAR SHARE, AND WASTED ENERGY FOR 12 HR/DAY OPERATION. WITH BYPASS.

SFM	Annual Efficiency	Average Power (MWe)	Solar Share	Energy Wasted (12 hr days of operation)
1	.379	4.56	.78	0
1.25	.377	4.75	.89	10.7
1.5	.373	4.84	.94	42.4
1.75	.372	4.89	.96	84.8
2	.37	4.92	.98	132

TABLE 6. ANNUAL EFFICIENCY AND POWER, SOLAR SHARE, AND WASTED ENERGY FOR DAYLIGHT OPERATION. WITH BYPASS.

Table 6 shows the annual performance of the cycle with a bypass when the turbine is used only during daylight hours. At a SFM of 1.25 the solar shares of the two systems are identical. The power output is also the same. Again, there is no significant thermodynamic difference between the two systems, whether they are operated 24 or 12 hours per day, or only during daylight hours. This establishes the possibility of a bypass as a thermodynamic alternative to a more heavily cooled combustor.

CONCLUSIONS

This paper examined two thermodynamic constructions of a hybridized gas turbine that can accept a combination of solar and fossil input. The gas turbine chosen was a single shaft, recuperated gas turbine due to high efficiency and low optimum pressure ratio that approximated the Solar Turbines Mercury 50.

Variable guide vanes are used in gas turbines to allow a degree of flexibility otherwise unavailable to single shaft turbines. As such, the effect of VGV's on a solar powered gas turbine was studied in depth. Scheduling the VGV's can alter the turbine inlet temperature through changes in mass flow, and can change the efficiency and pressure ratio of the system. In a non-hybridized system, variable guide vanes can be used to optimize gas turbine performance to a given level of solar thermal input by adjusting the mass flow rate as necessary.

When combustor inlet air is highly preheated, as is the case in this paper, two approaches were identified: the first is to provide extra cooling air to the combustor or design it to accept inlet air temperatures up to 1000 C, and the second is to install a combustor bypass, so that air above some limiting temperature is not passed through the combustor. Cooling air delivered to the combustor would bypass the receiver, but not result in any major thermodynamic inefficiencies since it would later be fully mixed with the burner exhaust to produce a uniform stream of air at the design point TIT.

When a maximum limit is imposed on the combustor inlet temperature and a bypass valve is installed, three distinct modes of operation appear. Mode 1 operation takes place at low solar thermal input, and natural gas is used to supplement solar input to bring the TIT up to the design point. During Mode 2 operation, the amount of fuel injected into the combustor is limited, and the turbine inlet temperature decreases with increasing solar input. Decreasing the TIT serves to decrease the amount of heat recycled back into the system via the recuperator, and consequently lowers the combustor inlet temperature. Mode 3 operation is solar only operation. If, on the other hand, no bypass is installed and the combustor can accept inlet temperatures of 1000 C, then only Modes 1 and 3 are present. In the case of a bypass, determining the optimal VGV position and fuel flow rate is complicated and at a certain value of solar thermal input a large shift in VGV position may be necessary. Whether or not the combustor has a bypass, the final decision on VGV position and fuel flow rate will rest on the economics of producing more power at the expense of reducing solar share.

Installing the bypass system has a negative effect on the efficiency of the system, since for some levels of solar thermal input the turbine inlet temperature of this system drops below the design point. This lower temperature is coupled with aerodynamic inefficiencies due to guide vane scheduling, which further decreases overall efficiency. In the case of no bypass, the turbine inlet temperature remains constant, and any change in efficiency is due solely to changes in gas properties.

When the two systems are compared on an annual basis at twelve hours per day operation, there is no significant difference between them. The average power and solar shares are within 1.5% of each other. This implies that the favored system is a function of the design and O&M costs associated with each, rather than on the thermodynamic performance. When the two systems are run only during daylight hours, the differences remain approximately the same, but the solar shares increase above 85% for a SFM above about 1.25.

ACKNOWLEDGMENTS

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NOMENCLATURE

Acronyms

CIT	Combustor Inlet Temperature
MWe	Electrical output in MW (with perfect generator)
MWth	Thermal heat input in MW
SFM	Solar Field Multiplier
TIT	Turbine Inlet Temperature
VGV	Variable Guide Vane

Variables

M	Mass Flow
PR	Pressure Ratio
W_{corr}	Corrected mass flow

Greek

θ	Dimensionless temperature, $\frac{T}{288.15\,K}$
γ	Ratio of specific heats
δ	Dimensionless Pressure, $\frac{P}{101.325\,KPa}$
Ω	Dimensionless Ideal Gas Constant, $\frac{R}{R_{std}}$
Π	Dimensionless ratio of specific heats, $\frac{\gamma}{\gamma_{std}}$
β	Non-physical parameter relating compressor (or turbine) speed to pressure ratio, flow, and efficiency.
ε_{recp}	Recuperator Effectiveness

APPENDIX: Values and Calculations

Design Point values are shown in Table A1.

Ambient Condition	**Design Point Value**
Ambient Temperature	288.15 K
Ambient Pressure	101.325 kPa
Pressure Drops	**Design Point Value**
Inlet Pressure Drop (PD)	.99
Compressor Pressure Ratio (PR)	9.9
Recuperator Cold Side PD	.98
Receiver PD	.99
Combustor PD	.97
Nozzle PD	1
Turbine Expansion Ratio	8.51
Exit Duct PD	.98
Recuperator Hot Side PD	.96
Exhaust Stack Pressure Drop	.991 (calculated value)
Temperatures	**Design Point Value**
Combustor Inlet Temperature	1123 K
Nozzle Guide Vane Inlet Temperature (Same as TIT in this example)	1273 K
Mass Flows	**Design Point Value**
Compressor	20 kg/s
Efficiencies	**Design Point Value**
Compressor Isentropic	.85
Turbine Isentropic	.89
VGV Parameters	**Design Point Value**
a_{vgv}	.01
b_{vgv}	.01
c_{vgv}	.01

TABLE A1. IMPORTANT PARAMETERS FOR EVALUATING GAS TURBINE ENGINE PERFORMANCE.

Design Point Calculation:

Assumptions: β = .5 for compressor and turbine. All other values as per Table A1. Real gas properties and combustion equilibrium calculations are used for all components.

Off-design Compressor and Turbine Performance:

The GasTurb11 program generates compressor maps whose x and y axes are defined by referred mass flow and pressure ratio, respectively. Compressor referred mass flow is given by:

$$W_{corr_{comp}} = \dot{m} * sqrt\left(\frac{\theta * \Omega}{\Pi}\right)/\delta \qquad \text{(A1)}$$

where $\theta = (\frac{T}{T_{std}})$, $\Omega = \left(\frac{R}{R_{std}}\right)$, $\Pi = \left(\frac{\gamma}{\gamma_{std}}\right)$, $\delta = \left(\frac{P}{P_{std}}\right)$, and the subscript "std" stands for standard day properties of air.

In off-design compressor conditions, some value β_c is assumed, which fully defines compressor performance at any compressor referred speed. Since ambient conditions, are known, the compressor referred speed is known, and compressor performance is known.

In off-design turbine performance, the pressure ratio across the turbine is known for a given value of β_c. Since the turbine inlet temperature is also known, and the engine operates at constant speed, the referred turbine speed is known. These values are used on the turbine map to determine efficiency and corrected flow. The corrected flow is then used to calculate actual mass flow, and this value is used to ensure continuity of mass throughout the system. If continuity is not satisfied, a new value for β_c is required.

Program Outline for Solving Off-Design Performance:

The outline of the MATLAB program created by the authors is given in Figure A1. The inputs to the system are the variable guide vane angle, the combustor inlet temperature, and the turbine inlet temperature. For example, in Mode 1, the TIT is constant, while the CIT varies from the design point recuperator outlet temperature, to the maximum CIT. In Mode 2, the CIT is fixed at 850 C, while the TIT varies from design point down to 850 C. Finally, in Mode 3, the CIT is equal to the TIT, which varies from 850 C to the design point TIT. The outline for solving the rest of the system is shown in Figure A1.

Duct Pressure Drops: All pressure drops are modified in off-design conditions using:

$$\frac{1-\frac{P_1}{P_2}}{\left(1-\frac{P_1}{P_2}\right)_{DP}} = \left(\frac{m*\frac{sqrt\left(R*\frac{T}{\gamma}\right)}{P}}{\left(m*\frac{sqrt\left(R*\frac{T}{\gamma}\right)}{P}\right)_{DP}}\right)^2 \qquad \text{(A2)}$$

VGV Effects:

The variable guide vanes alter the performance of the compressor, essentially supplying a different combination of mass flows and pressures than normal. In order to determine the effect of the VGV's the following equations are used to scale the compressor map:

$$PR_c = 1 + \left(PR_{c_{DP}} - 1\right) * \left(1 + b_{vgv} * VGV\right) \qquad \text{(A3)}$$

$$W_{corr_c} = W_{corr_{c_{DP}}} * \left(1 + a_{vgv} * VGV\right) \qquad \text{(A4)}$$

$$n_c = n_{c_{DP}} * \left(1 - VGV^2 * \left(\frac{c_{vgv}}{100}\right)\right) \qquad \text{(A5)}$$

Where PR_c, W_{corr_c} and n_c are the pressure ratio, corrected mass flow, and isentropic efficiency of the compressor, a_{vgv}, b_{vgv} and c_{vgv} are constants, and VGV is the variable guide vane angle away from zero.

Recuperator Off-Design Performance:

$$\frac{\varepsilon_{recp}-1}{(\varepsilon_{recp}-1)_{DP}} = -\frac{W_{corr}}{W_{corr_{DP}}} \qquad \text{(A6)}$$

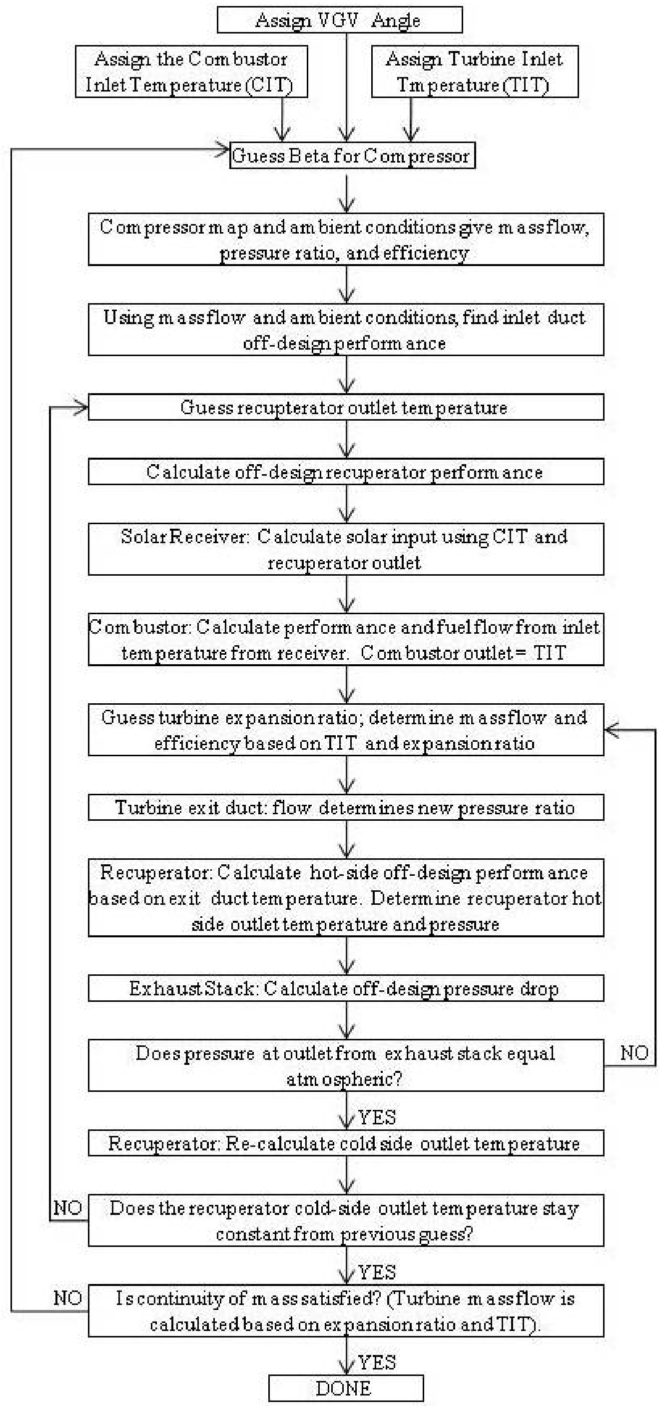

FIGURE A1. PROGRAM OUTLINE FOR SOLVING OFF-DESIGN PERFORMANCE OF SOLAR POWERED RECUPERATED GAS TURBINE.

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