Comparison of Power Enhancement Options for Retrofit to Combined Cycle Power Plants

Phase 2 Report

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Purpose

Turbine Air Systems (TAS) began this analysis effort in early 2002 with the concept of studying the relative impact of several options of power augmentation, first for new "Greenfield" combined cycle plants, and then for “Retrofit” options to existing plants. We originally undertook this effort in response to several of our customers, who expressed confusion regarding the competing claims of many power augmentation vendors who sometimes made assertions that were difficult to back up.

Our customers were also largely unaware that during the past five years, the total installed cost of inlet chilling has dropped dramatically, and that the efficiency and maturity of these systems has increased significantly. Thus, many of the previously held assumptions of otherwise well-intentioned people needed to be challenged with new data that reflected the state-of-the-art.

Our goal was to develop a methodology that would allow a power plant Owner to be able to clearly understand the inputs and assumptions that we would model, and more importantly, to be able to use independent and commercially available software to re-create the same results.

The first phase of this project was presented at Power-Gen International 2003. The first phase was the study of “Greenfield” installation options.

The second phase of the project is presented herein. This second phase is concerned with the retrofit of turbine inlet chilling to a plant that was not originally configured for chilling. As such, the retrofit of most power augmentation technologies is severely challenged in a retrofit environment for technical and commercial reasons.

As in the first phase, the effort was conducted using the industry-standard GT Pro / GT Master suite of software. Our goal has been to maintain maximum credibility throughout the development of this material by explaining our input assumptions, and using third-party software, so as to allow repeatability of results by an independent examiner.

This narrative makes a case for the relative technical and economic feasibility of the retrofit of chilling technology to a plant that already has fogging or evaporative cooling technology (and of course, for plants with no prior cooling at all).

We believe that this study is important to an industry that has a fleet of combined cycle plants designed for base-load and intermediate-load operations, yet in reality operate more like summer peakers. The economic viability of these plants relies on maximum economic return during a very limited portion of the operating year, when maximum spark-spread opportunity exists.

Introduction

The purpose of this paper is to explore the economic and performance implications of several common forms of power enhancement for combined cycle power plants, including the interaction of several forms of power augmentation when employed together. Options compared are:

- Turbine inlet air cooling using fogging (evaporative cooling)
- Turbine inlet chilling (TIC) using mechanical chillers (refrigeration)
- HRSG supplemental duct firing

In this paper, we are exploring the installation and resulting performance of augmentation technologies on a retrofit basis. That is, installing technologies on an existing plant that never included that possibility in the original design.

The impacts on power output and heat rate, as well as the capital cost of the plant, in both absolute and in incremental terms, are calculated for various scenarios. The results will show several important design aspects for turbine inlet cooling by means of mechanical chillers:

- The incremental cost per kW for output attributed to chiller systems is superior to the existing base plant unit cost, where the original plant employed no inlet cooling such as fogging or evaporative coolers.

- The incremental cost per kW for incremental output attributed to chiller systems for a plant that was originally built for foggers is competitive when Thermal Energy Storage is added to the design scheme.

- Both chillers and duct firing provide "active" controls that maximize the output and flexibility of the front-end and rear-end systems. Active systems allow the operators to react to changing economic conditions.

- Power augmentation by chilling can be likened to a built-in peaker, except with superior operating economic characteristics as compared to a new stand-alone aero-derivative GT.

- Performance and economic results will challenge fleet Owners to consider the retrofit of chiller technology to existing plants for their next round of peaking capacity construction.

A typical 2x1 F-Class gas turbine combined cycle power plant is considered for this study. We will consider four common versions of the existing plant "Base Case", representative of the existing fleet of F Class combined cycles in operation.

Modeling of the Original Power Plant

A 2x1 GE 7FA (PG7241FA) gas turbine combined cycle power plant, typically referred to as a STAG207FA, was considered for this study. The HRSG and other plant hardware were generic and designed for the gas turbine exhaust conditions. The steam turbine was generic but an attempt had been made to conform to standard GE D-11 sizes available in the market. All models were designed in Thermoflow's[1] GT Pro software and then simulated for off-design using the GT Master software. The Thermoflow PEACE option provided an estimate of the cost of the equipment as well as construction and installation cost of the base facility.

The following cases are simulated in this Phase 2 effort:

Original Configuration	Base Plant Technologies	Modifications
BASE CASE 1	No fogging; no chilling; no firing	ADD Chillers
BASE CASE 2	w/ Fogging; no chilling; no firing	ADD Chillers

It continues to be our assumption (carried over from the Phase 1 Report) that the use of the GE F-class GT in the models would be representative of similar results for all advanced gas turbines such as the Westinghouse W501FD, the Alstom GT24, the MHI 501F, and the Siemens V84.3A, and all of these turbines' 50-cycle counterparts. Moreover, the general concepts of this study can be reasonably extrapolated down to older E-class non-reheat cycles, as well as up to G/H class advanced steam-cooled cycles.

However, this Phase 2 Retrofit study is still best considered for the GE F-class family of gas turbines, due to concerns developed in the F fleet due to potential compressor damage[2] associated with fogging. The OEM has identified interim repair methods[3], blade coatings, and new blade materials[4]. Although some operators have resumed fogging with a more frequent inspection schedule[5], there is a lingering concern for long-term expense, either through a scheduled blade repair/re-coating/replacement program or, worse, an "unscheduled blade replacement".

Moreover, this paper does not consider the intentional use of fogger "over-spray" or "wet compression" as an augmentation technology because this method is still not

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1 www.Thermoflow.com. This software was chosen because of its wide acceptance among power project developers, not only for initial screening studies, but also for detailed plant design as well.

2 GE TIL-1389-1R1, 2003. "Compressor Rotor Blade Erosion From Water Ingestion Used In Power Augmentation"

3 "Boosting Gas Turbine Power…", Phillips and Levine, Turbomachinery International, volume 45, no. 4, July/August 2004

4 GER 3569f, Advanced Gas Turbine Materials And Coatings

5 GER 3620j, Heavy-Duty Gas Turbine Operating and Maintenance Considerations

recognized as a commercial option by this GT OEM. For other GT OEMs, the use of wet compression technology can be considered as its own stand-alone economic exercise, due to the belief that the effects of chilling and wet compression are most likely complimentary (additive) and not mutually exclusive[6].

The simulation of new plant designs begins with the plants that were described in Phase 1 of this report. In review, all Base Case models were designed using the following assumptions:

- Average ambient conditions for plant design:
  Dry Bulb Temperature: 77°F[7] (25°C)
  Relative Humidity: 50%[8]
  Wet Bulb Temperature: 64°F (17.8°C)
  Elevation: 0 ft. MSL (0 meters MSL)

- Gas Turbines:
  - Two GE PG7241FA gas turbines with DLN combustors
  - Performance is based on the default library model in GT Pro
  - Inlet filter house losses are 3 in. water gauge (wg).
  - Total exhaust losses are set at 16 in. wg, which includes the losses due to catalytic reactors in the HRSG for NOx and CO reduction.
  - Gas turbine natural gas fuel is heated from 59°F to 365°F (15° to 185°C) in a heat exchanger using hot water from HRSG IP economizer.
  - A typical natural gas fuel composition is one used with heating value of 20695 Btu/lb (LHV). The HHV/LHV ratio is 1.1076. No provision has been made for liquid fuel / dual fuel firing.

- Steam Turbine:
  - This was a three-pressure reheat steam turbine. An attempt was made to conform to GE's D-11 structured steam turbine.
  - The steam turbine was designed with two LPT exhaust ends with down exhaust. Two sizes of steam turbine were used in the models depending on the power enhancement option considered. All supplementary duct-fired models had a steam turbine with 33.5 inch Last Stage Blade (LSB). The rest of the models (base and fog) had a steam turbine with a standard 30-inch

6 While the use of chilling concurrent with traditional fogging is not technical viable or even prudent, it is presumed that the wet compression in a chilled environment could yield additive results. Without data on an actual installation, this remains an assumption. However, the load profile of a plant that employs wet compression is not "flat-lined" as would be expected in a chilled plant. Therefore, despite reasonable potential "value pricing" and performance of wet compression, it still seems to not fundamentally address the critical and qualitative issue of *predictability* of plant output.

7 77°F corresponds to 25°C, a common design point for balance-of-plant equipment.

8 50% RH corresponds to a linear interpolation between the design summer conditions of 95°F (35 C) / 40% RH, and ISO conditions of 59°F (15 C) / 60% RH.

LSB. Therefore, the costs associated with a larger steam turbine was be borne by the duct-fired cases.
  - The maximum output of the steam turbine was limited by the HP inlet steam flow of 1300 kpph.
  - The steam turbine operates on sliding pressure in off-design cases.

- HRSG:
  - The HRSG was designed as a three pressure reheat type.
  - The HRSG arrangement was optimized by the software for the gas turbine exhaust and steam conditions, based on ambient conditions of 77°F (25°C) and 40% RH.
  - The low-pressure economizer at the back end of the HRSG contains a re-circulation loop to maintain inlet feedwater temperature higher than the acid dew point of the sulfur in the exhaust gases.

- Condenser and Cooling System:
  - The plant has a deaerating condenser.
  - The condenser was designed at 2.0 in. Hg. backpressure, based on ambient conditions of 77°F (25°C) and 50% RH.
  - The range of the cooling tower is 18°F (10°C)
  - The approach is 11°F (6.1°C)
  - A mechanical air-draft wet cooling tower was included for heat dissipation.

- Fogger:
  - For a "Greenfield" application, the Fogging system was designed for 95%[9] effectiveness with ultra-fine droplet size. No additional inlet air pressure drop was associated with the fogging system. In order to achieve 95% effectiveness in an application it is usually necessary to allow for a certain amount of "over-spray". This would result in undesirable droplets entering the compressor. It may be possible however to use a mist eliminator to over-spray the fog, without risking droplet impingement. This design would require a special filter house design.

9 95% is considered to be an excellent performance specification for a typical fogging system. Real-world applications are largely limited to 75% to 85% effectiveness, as measured on an "approach to wet bulb" basis. In order to achieve 95% effectiveness, it is usually necessary to over-spray the fog considerably, resulting in water droplet carryover into the GT compressor. For the purposes of this paper, which is to demonstrate the relative impact of chilling technology as compared to fogging technology, using the higher fogging effectiveness causes the relative chiller results to be highly conservative.

- An over-spray[10] or "wet compression" style of fogging system was not considered for this study. This system is not offered or authorized for use by the OEM for the 7FA machine.
- Evaporative cooling by method of "evaporative media"[11] was not considered.

- Selected PEACE input variables used for estimating the cost of the power plant:
  - The gas turbine, steam turbine and HRSG are not located indoors. The water treatment center is located indoors.
  - Cost multipliers of 1.0 are used for all estimates. These factors can be changed in the "Cost Modifiers" worksheet in the Cost Report spreadsheet.
  - The gas turbine has a single-fuel package option, Hydrogen-cooled generator and electric motor starter.
  - One 36 kpph auxiliary boiler, running on natural gas for the plant.
  - Steam turbine has a downdraft exhaust duct.
  - SCR or Catalytic reactor in the HRSG for NOx and CO reduction.
  - Inclusion of continuous emission monitoring system (CEMS).
  - The plant has a DCS.

It is important to note that in setting up the original GT Pro models, certain aspects of plant performance were "fixed", such as steam turbine generator (STG) last stage blade length and cooling tower approach and range. However, other design parameters were allowed to "float" to find the best economic and technical results. The most important of these features would be:

1. The heat transfer surface of the HRSG, particularly the superheater section
2. The steam turbine condenser
3. The cooling tower.

The critical design factor is that as GT inlet temperatures change, so would the GT exhaust temperatures. The critical design step is to allow the GT Pro program to find the best heat transfer surface area derived from the changing exhaust conditions. Such changes in exhaust temperatures would change the approach for the superheater section of the HRSG. Moreover, as the GT inlet temperature changes, there would be

10 "Over-spray" fogging systems are proprietary, with limited applications, and are not used for comparison purposes for this paper. The methodology for determining incremental costs and incremental performance throughout this study can also be used for evaluating over-spray systems.

11 Evaporative cooling through wetted media continues to be a viable alternative to fogging. Typical design effectiveness is 85% approach to WB. The use of evaporative cooling requires a larger filter house than a non-augmented filter house. The air pressure drop associated with the evaporative media (and drift eliminator) can be approximated at 1" w.g., same as for the chiller coil. Because of the similar performance and cost impacts of evaporative media and fogging, we only study fogging due to fogging's higher relative effectiveness, which makes the chiller comparison conservative. Some operators continue to choose evaporative cooling over fogging due to safety concerns resulting from water droplet carryover into the compressor inlet.

significant changes in exhaust mass flow. For the chilled Base Cases, the HRSG needed to be optimized for higher mass flow and lower exhaust temperatures, in order to attain the highest possible steam cycle output. This was a critical design factor in optimizing the combined cycle for chiller operations. An HRSG not designed for chiller operations will clearly under perform as compared to a properly designed HRSG.

However, in this Retrofit study, there was no “float” or redesign of HRSG or other steam path components. This is intended to reflect the reality that the plant must be expected to operate with original hardware, even if it does not reflect the most optimal design.

This figure shows a S207FA in Texas, USA, with a nominal power rating of 633 MW. This plant was originally designed for both chilling and high rates of supplemental firing. In summer operations, the chiller capacity is dispatched before the firing capacity, which is held in reserve for the hottest days only when economic dispatch is made possible. Two TAS “F-50” chiller systems are shown on the right side of the picture. Combined net coil demand is approximately 10,500 tons (refrigeration).

Designing the plant for maximum output while still “on paper” has yielded one of the most powerful and efficient plants in the fleet.

This figure shows one of three W501D5's located near Houston, Texas, USA. This older cogeneration plant was originally designed with no inlet cooling at all. It was later retrofit with inlet fogging. In order to recover even more power losses associated with summer weather, the Owner went the next step to install chillers. The picture shows a new modular filter house fitted in 1999. This system included new barrier filters, cooling coils, drip pans, and mist eliminators. Outside the filter house, the chilled water pipe manifolds are visible. Each gas turbine has two such filter houses, aligned perpendicular to each other.

This is a unique installation in that the older fogger system was not removed during the chiller coil installation. It is possible to stand in the transition duct of this unit during operation. The air in the duct runs very wet during fogger operation, with the floor of the duct visibly "running" with water, while the system stays dry with chiller operation due to the conservative design and the use of mist eliminators.

Technical and Economic Assumptions for Adding Chillers

The various Base Case combined cycle models were originally designed at the average ambient conditions of 77°F (25°C) and 50% RH (which corresponds to 64°F WB). For simulating performance in the summer, we used off-design conditions of 95°F (35°C) and 40% relative humidity (which corresponds to 75°F WB). This temperature was selected as a typical high ambient temperature when the need for summer power enhancement is most beneficial.

The reason for these two ambient conditions could be explained as follows. The design conditions were optimized for a combined cycle plant in "intermediate" service, that is, neither a base-loaded plant nor a peaker. This design was representative of the "merchant" plants pressed into service in the past seven years. Such a plant could be expected to operate approximately 4,000 hours per year (this unfortunately turned out to be optimistic for many new US merchant plants). The fogger and chiller systems, on the other hand, are designed for summer peak design conditions, as is typical design practice for the industry, and not coincidently, when these combined cycle plants have been getting so much of their dispatch time.

The graphical output of the simulation of power plant cycle for each of the case runs described earlier is given in Appendix A. The description of the "Base Cases" is as shown in the following table:

Case Description		With Duct Firing
No Inlet Conditioning		Case 1
Fogging		Case 2

Chiller technical design:

- Chiller:
  - A water-cooled electrical chiller with its own auxiliary cooling tower for condenser cooling.
  - Chilled Air nominal approach to chilled water: 7.25°F (4°C)
  - Chilled water nominal range ("Delta T"): 18°F (10°C)
  - Condenser water nominal approach to wet bulb: 7°F (3.9°C)
  - The capacity of the chillers and its associated equipment is designed such that it will chill the turbine inlet air down to 50°F (10°C) at the summer ambient reference conditions. The cold air temperature downstream of the chiller coils is often referred to as "T2" (where T1 would be the ambient temperature). This summer design ambient condition is always used for chiller system design, even for cases where the "balance of plant" (BOP) (HRSG, STG, condenser, cooling tower, etc.) equipment is designed at the default 77°F (25°C) / 50% RH ambient condition. Accordingly, when the ambient conditions are less severe, the chiller system will be operating at part-load.

- The GT Pro / PEACE model does not yet support an integrated chiller *system* design such as the TAS F-50 which TAS would recommend for this application. Accordingly in the GT Pro models, each gas turbine has three chillers, each chiller with a nominal capacity of 1,850 tons. Nonetheless, we have forced the model to provide performance characteristics commensurate with the function of (2) TAS F-50 chiller systems.
- The GT Pro model was set up with the following inputs to mimic the performance characteristics of the TAS F-50 chiller system, with a nominal capacity of 5,700 tons (11,400 tons total) at the following nominal conditions:
    - 82°F (27.8°C) condenser water supply temperature from the dedicated condenser cooling tower.
    - 42.5°F (5.83°C) chilled water temperature
    - System electrical parasitic loads are "grossed-up" for ARI allowable tolerances ("zero negative tolerance")
    - The overall chilling system is based on the most common "two on two" configuration, with two chiller plants on a common piping manifold that supply the two GT's.
    - The TAS F-50 system is operating within the guaranteed performance envelope (output and efficiency) typically quoted for this system.

Construction Economics:

The typical contracting method for the retrofit of a chiller system to an existing plant would be a "lump sum turnkey" (LSTK) with the chiller packager. The Scope of retrofit supply would require:

1. Packaged Chiller systems
2. Dedicated cooling towers
3. Filter house modification kit and support structures
4. Cooling coils, and pipe manifolds, and condensate collection system
5. Step-down electrical transformer (13.8kV primary) with power distribution switchgear.
6. Field installation and construction services (includes civil work, pipe racks, distribution piping, insulation, etc.)
7. Field engineering technical support
8. Shipping (domestic)

While every site is different, we used a typical installation in our own cost estimating models, and developed a LSTK project cost of $12,000,000 for a S207FA. This cost will be used in the analysis exercises in this report.

ANALYSIS - Power Output Results

The subject of this paper is to describe power augmentation technologies. Hence, the *incremental output* of each option is a critical factor in the evaluation of the technology.

All values are based on the simulation at the off-design ambient summer conditions of 95°F and 40% relative humidity. All values are in kilowatts (kW). The absolute and incremental results are shown for fogging, chilling, and supplementary duct firing. These results are analyzed in further detail in the following pages.

Phase 1 Original "Base Plant" designs Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning		Output	Incremental Output, kW	
1	None		452,183		
2	Fog		481,853	29,670	6.6%
3	Chiller		505,922	53,739	11.9%

Phase 2, Add Chiller Technology to the Base Plant Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning		Output	Incremental Output, kW	
1	None		452,183		
	Add Chillers		503,472	51,289	11.3%

As can be seen from the above tables, the addition of Chillers on a retrofit basis will not provide quite as much power augmentation on a retrofit basis as it would when designed in from the beginning. If chilling is planned from the beginning, the incremental power gain is 53,739 kW, versus 51,289 kW for a retrofit application. Planning from the beginning provides an improvement of 2,450 kW, or 4.8% over the retrofit of the same technology.

The reason for this can largely be found in the HRSG / STG train. The drop in GT exhaust temperature, despite the higher mass flow, impacts HP steam production. It is best to design for additional HP superheater surface area in the original plant design. When that surface is not available, the relative increase of steam production does not increase proportionally to the GT power increase.

Another consideration is that the increase in steam flow will challenge the existing condenser and cooling tower. In a retrofit application, there could be expected a slight increase in condenser backpressure. Therefore, STG output would not increase proportionally to the GT power increase.

One important lesson to be learned here is that if the rollout of chiller technology is planned in a phased construction approach, then the steam plant BOP must account for the planned exhaust flow that would be experienced when the chillers are finally added.

In this second scenario, we consider the addition of Chillers to a plant that already has foggers. Again, as can be seen from the below tables, the addition of Chillers on a retrofit basis will not provide quite as much power augmentation on a retrofit basis as it would when designed in from the beginning. If chilling were planned from the beginning, the incremental power gain over foggers would be 24,069 kW, versus 23,108 kW for a retrofit application.

Phase 1 Original "Base Plant" designs Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning		Output	Incremental Output, kW	
1	None		452,183		
2	Fog		481,853	29,670	6.6%
3	Chiller		505,922	53,739	11.9%
	(Chiller over fog):			24,069	+81.1%

Phase 2, Add Chiller Technology to the Base Plant Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning		Output	Incremental Output. kW	
2	Fog		481,853		
	Add Chillers		504,961	23,108	4.8%

Phase 2, Add Chiller Technology to the Base Plant Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning		Output	Incremental Output, kW	
1	Add Chillers		503,472		
2	Add Chillers		504,961	1,489	+6.4%

Nonetheless, the fogged plant (Base Case 2) appears to allow for a greater net output when chillers are added. This is evidenced in the final net output of 504,961, which is 1,489 kW more than the net output of the original Base 1 plant when chillers were added. This represents a 6.4% improvement in output. Again, the reason for this is that if the plant was properly designed in the beginning, the engineer would have accounted for the slightly higher steam mass flow and slightly lower GT exhaust temperature in the design of the plant BOP. The addition of the Chiller does not represent as much of a radical change (with respect to the BOP) when added to a plant already designed for foggers.

There is a substantial economic challenge to adding chillers to a plant that is already equipped for foggers. Consider that the incremental power change for the chiller addition for the Base Case1 plant is about double that of the chiller addition for the Base Case 2 plant.

Phase 2, Add Chiller Technology to the Base Plant					
Base Case	Inlet Conditioning		Output	Incremental Output, kW	
1	None		452,183		
	Add Chillers		503,472	51,289	11.3%
2	Fog		481,853		
	Add Chillers		504,961	23,108	4.8%

There are very few plants in the fleet that do not have any form of inlet cooling at all. Most plants have either foggers or evaporative cooling (which we will consider roughly equivalent in output herein). Therefore the real change in output for adding chillers seems comparatively small. Some people might consider adding chillers to be on the wrong side of “the law of diminishing returns”.

In the critical factor of “unit cost”, or “$ / kW”, the chiller technology addition to a fogged plant (Base Case 2) includes all of the capital costs as compared to adding chiller technology to an unaugmented plant (Base Case 1). However, there are fewer new kilowatts produced to amortize the cost. So the simple evaluation of “$ / kW” doesn’t look as promising for adding chillers to a fogged plant.

Phase 2, Add Chiller Technology to the Base Plant					
Base Case	Inlet Conditioning	Output	Incremental, kW	Cost	$ / kW
1	None	452,183			
	Add Chillers	503,472	51,289	$12,000,000	$234
2	Fog	481,853			
	Add Chillers	504,961	23,108	$12,000,000	$519

The addition of peaking technology at a unit price of $234 per kW for Base Case 1 is very reasonable and attractive. However, the same chilling technology when added to the Base Case 2 plant yields an incremental cost of $519 per kW. By most measures, this is would not be considered as attractive an investment as the former case. However, there are mitigating factors that will be considered.

1. The plant equipped with foggers might not run the foggers, due to the technical safety concerns discussed previously. Therefore, the reference case for economic evaluation would be the non-fogged plant, and the incremental unit cost would be the more attractive value of $234.

2. Fogging and other evaporative technologies work best when the spread between dry bulb and wet bulb temperatures is greatest, most typically in the middle of the afternoon. However, many utilities also have an early morning demand peak, as well as an evening

peak. There could be as many as three peaks per day. There could be expected less spread during those hours, and less assistance from the foggers.

3. Increased overall plant turndown capability from the chiller capacity, including using chillers to track small load swings, or even to follow load automatically.

4. There are means for improving the chilled plant output with Thermal Energy Storage (TES) systems, that we will show later in this paper.

ANALYSIS - Environmental Impact

One of the heretofore-unquantified benefits of using power augmentation on the F Class CC fleet is the potential incremental net decrease in pollutants. Even if there are simple cycle peakers available, many of these could be older units with poor fuel efficiency and high unit NOx output.

For example, the table below provides an estimate of using the incremental output of chiller capacity in lieu of older simple cycle peakers (unabated FT-4).

Displacement of Simple Cycle Megawatts		
	SC Peakers	Chillers
Summer Hours	750	750
Approx. Peak MW Available	50	50
MWH/Year (summer only)	37,500	37,500
Seasonal NOx (pounds)	300,000	1,125
Emissions Reductions		149 tons
Fuel Savings (CC vs. SC)*		$1,400,000*

* Spot fuel cost of $7 per mmBTU; 4,500 HHV heat rate delta

There has been much talk within the industry of the potential to reduce unit NOx output (on a "ppm" basis) with the use of foggers. This is because humid air has a greater heat capacity than dry air, which helps to reduce flame temperature. We suspect that like most other system changes that lower NOx output through reduced flame temperature, the possibility exists that CO emissions will increase concurrently. We are uncomfortable with claims within the fogger industry that purport to change engine operation in such a manner. On the contrary, the chiller technology replicates as much as possible a naturally occurring cold day, for which the engine controls are already calibrated. We do not know of any chiller systems that have made claims for reducing the NOx emissions for a GT on a "ppm" basis.

Nonetheless, serious company-wide NOx reductions are available by simply doing two things:

1. Displace the company's dirty unit operations with clean unit augmentation to the maximum extent possible;
2. Improve peaker capacity fuel efficiency, because less fuel means fewer emissions.

ANALYSIS – Heat rate Results

As shown in the Phase 1 report presented in 2003, the heat rate for chilling is not as good as for fogging, but is better than for duct firing.

Phase 1 Original "Base Plant" designs Summer Peak Conditions of 95°F (35°C) and 40% RH				
Case	**Inlet Conditioning**	**Duct Firing**	**Heat Rate**	**Incremental Heat Rate**
1	None	No	6,371	
2	Fogging	No	6,356	6,127
3	Chiller	No	6,453	7,143
4	None	Yes	6,668	8,389
5	Fogging	Yes	6,638	7,789
6	Chiller	Yes	6,705	7,893

In order to properly justify the incremental costs of a technology, we need to consider the "value proposition" of its operational economics. Technologies that have high fuel operating costs should require less up-front capital cost than high-efficiency technologies. Hence, simple-cycle peaker plants use their low capital costs (and limited operating hours) to justify their poor heat rate. Conversely, for a plant that is designed for a base load or an intermediate load operating plan, then higher-priced technologies should be added that would improve the plant's heat rate.

In order to make this analysis, we are going to need the costs and performance of simple-cycle peakers as a reference point for cost and efficiency references. We add two more "Base Cases" to the above table taken from the original Phase 1 report. Case "A" will be two simple cycle peakers, both GE PG7241FA engines like those applied to the original study. No inlet cooling is applied. Case "B" will be similar to Case A, but will include fogging.

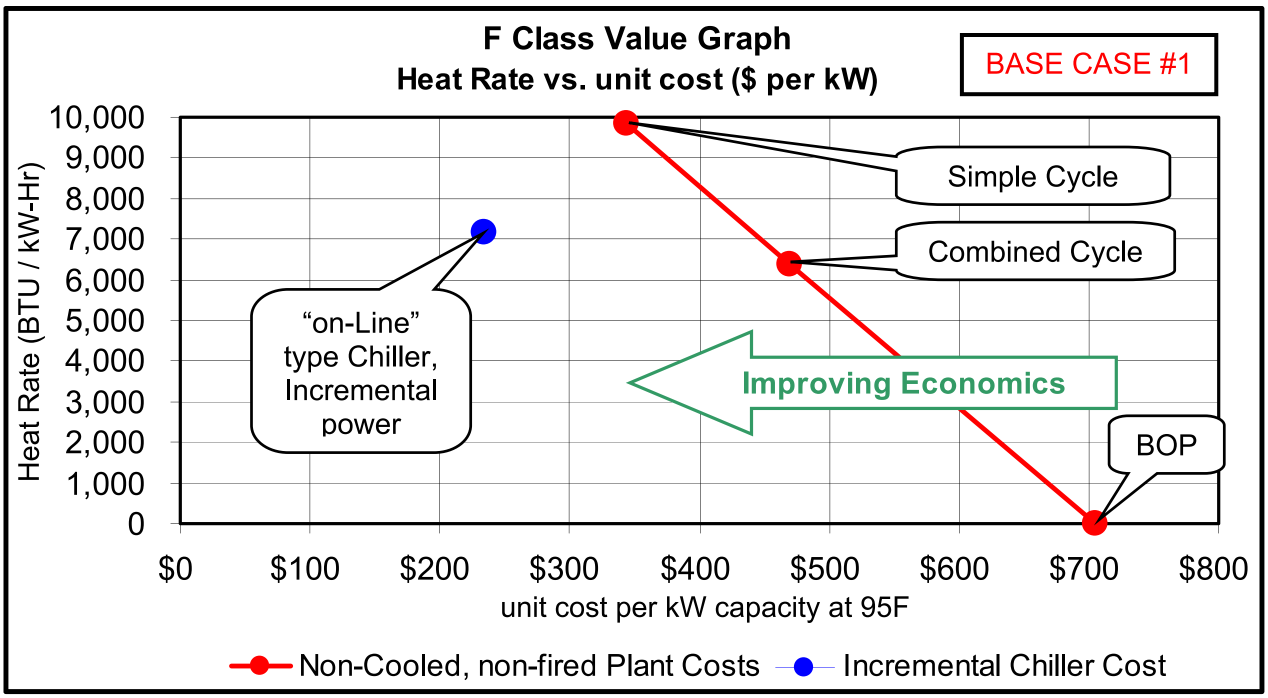
Phase 1 Original "Base Plant" designs Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning	Heat Rate	Output	Ref Cost (millions)	Unit Cost
A	None	9,835	294,832	$101.4	$344
B	Fog	9,532	319,855	$102.9	$322

The PG7241FA is not the typical first choice for simple cycle peaking plants. However, many utilities have actually installed these engines as simple cycle peakers, with the intention of one day converting them to combined cycle units as their need for intermediate and base-load power increases.

The next step is to look at the BOP equipment associated with the difference between a simple cycle peaker and a combined cycle plant. This would include the typical BOP of HRSGs, STG, cooling tower, water treatment plant, etc. This equipment adds $111 million to the project, but also adds 157,351 kW, with a unit cost of $705 per kW. The incremental heat rate of this additional power is essential zero, because it involves the recovery of waste heat. This exercise illustrates quite well that the industry is willing to pay a premium for additional power output, if the heat rate is sufficiently low. In this case, when the engineer decides to convert a simple cycle plant to a combined cycle plant, there is a willingness to pay $705 per kW at an associated heat rate of 0 BTU / kW-Hr.

Phase 1 Original "Base Plant" designs Summer Peak Conditions of 95°F (35°C) and 40% RH					
Base Case	Inlet Conditioning	Heat Rate	Output	Ref Cost (millions)	Unit Cost
A	None	9,835	294,832	$101.4	$344
1	None	6,371	452,183	$212.4	$470
Delta:		0	157,351	$111.0	$705

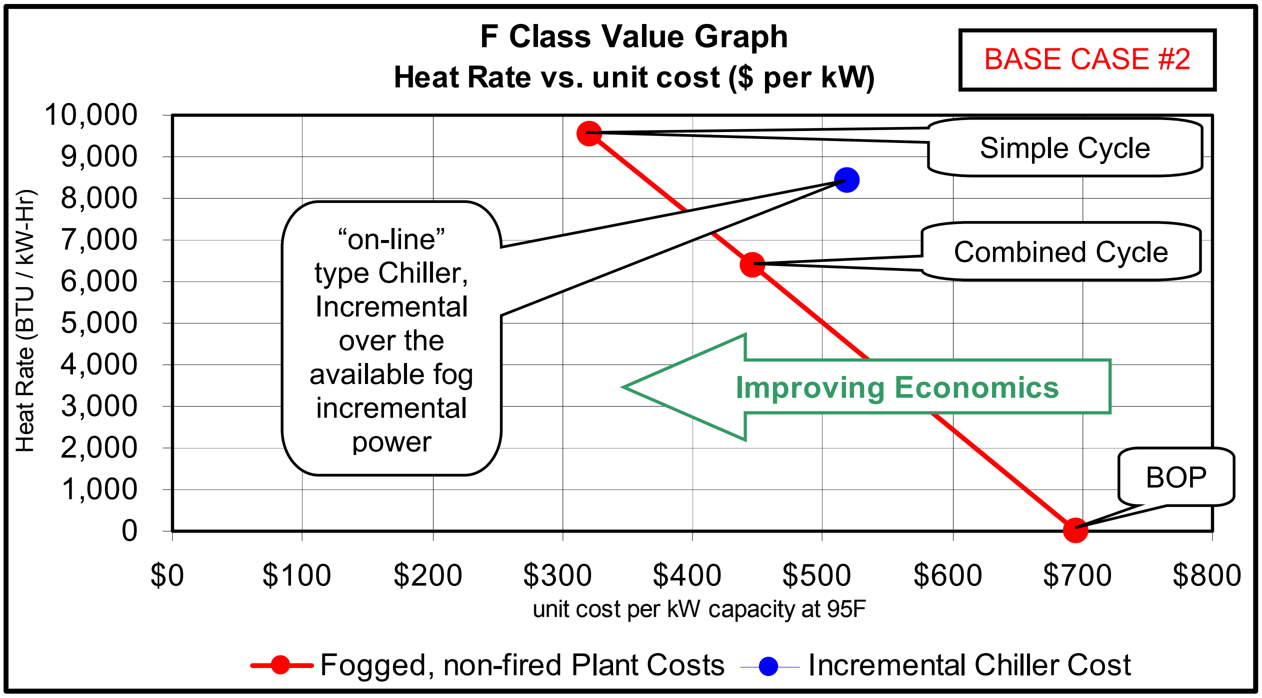
With this data in mind, it is possible to develop an interesting graphic that shows the relationship of heat rate and unit price (for **BASE CASE #1**):



This graphic shows the three reference points for a simple cycle plant, a combined cycle plant, and the BOP required to change simple cycle plant to a combined cycle unit. Everything to the left of this line will be less expensive and/or more efficient than the state of accepted system economics / efficiency trade-off. This is considered a very simplistic "go / no-go" gauge.

When we over-lay the data for a chiller plant retrofit, we see that the chiller cost is less than all of the accepted plant options. The chiller incremental heat rate is not quite as good as a new combined cycle plant, but it is far superior to that of a simple cycle peaker. This relationship best illustrates the "first cost" attractiveness of a chiller system retrofit to a plant that has no cooling at all.

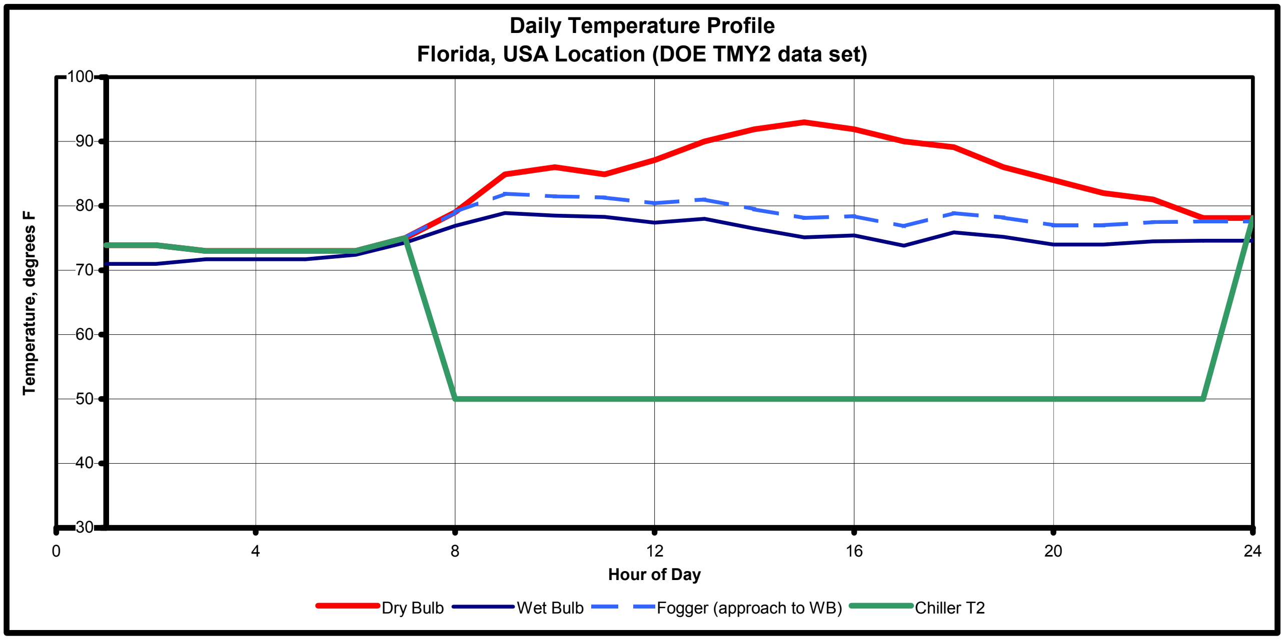
The preceding example and graphic uses Base Case 1 as the reference point. This plant had no original inlet cooling. However, as expected, we get less attractive results graphically for an original plant configuration that included foggers (**BASE CASE 2**).



Despite the seemingly unattractive economics of the chiller addition for Base Case 2, this has not kept power plant owners from considering this option. The reasons are many, but include the following:

1. Desire to retire their fogging systems due to technical risk.
2. Need for additional power quickly without adding a new plant.
3. Desire for the predictability of the chiller output.

Reasons number 1 and 2 are self-explanatory, so lets explore the third reason. As described in the Phase 1 report, gas turbines react passively to ambient temperature changes. Further, while the addition of foggers only slightly mitigates the variability of gas turbine output, the plant will still react passively to ambient weather conditions, with little that an operator can do to impact plant output, and as importantly, output stability and predictability. Note the graphic below that shows a typical weather profile for a single summer day location in Florida. We show 24 hourly data points for dry bulb temperature, wet bulb temperature, the "operating line" for a fogging system (where approach to WB can't be more than 3 degrees F for safety reasons), and finally the green line which shows the typical chiller plant "T2" temperature. In this case, the chillers are started during the "Peak" period of the day, from 07:00 until 23:00, as is typical for many plants with 5 X 16 commitments.



Of course, because the output of the plant is inversely proportional to the gas turbine compressor inlet temperature, the plant operator will get the benefit of high output and a very predictable "flat-line" power profile. This is critical for system operators who need to be able to predict plant output on a day-ahead (or longer term) planning schedules.

The above graph shows a peak dry bulb temperature of 93 degrees F and a corresponding wet bulb temperature of 75 degrees F. This matches up with our design criteria well (95 /75). Using the 3-degree approach to WB rule, the foggers can achieve a dry bulb temperature drop of approximately 15 F. On the other hand, the chillers will achieve 43 F temperature drop. Accordingly, the incremental power increase for the chiller technology should be approximately 3 times that of the fogger technology. However, our *net plant output* tally does not show that. The reason is quite simple: approximately 15% of the total incremental gross output attributed to the chillers is lost to parasitic electrical losses of the chiller system. In order to make the chiller technology competitive to foggers on a retrofit basis, it may be necessary to perform the installation in such a way that mitigates the parasitic electrical draw of the chillers.

The Impact of Thermal Energy Storage

For a plant that has foggers already installed and that intends to continue to operate the foggers, as a minimum on a limited or emergency basis, then the chiller system needs to be configured in such a way as to make the economic comparison more attractive. The answer to this concern is Thermal Energy Storage (TES).

TES is an option for new plants too, and is not limited to retrofit applications. However, during the recent US building boom from 1998 until 2001, plant developers were more interested in getting their plants to market quickly, rather than finding the most optimal design configuration. In the Phase 1 Report, the topic of TES was only briefly mentioned, with the plan to provide more details in this Phase 2 report.

In TES applications, fewer chillers are added to a plant site. To make up for lost capacity, a chilled water storage tank is added to the project. The loss of chiller capacity is made up with chilled water stored in the tank. There are two typical design / operating modes for TES:

1. Partial Storage. In a Partial Storage design, approximately one half of the chiller capacity that would normally be installed is deleted. The remaining half capacity chiller system runs 24 hours per day creating chilled water. During the day, this created chilled water is pumped to the GT cooling coils. Because there is not enough chilled water creation capacity to keep up with the coil demand, an equal amount of chilled water is withdrawn from the storage tank to satisfy the coil demand. In such a manner, the coils can be fed approximately 12 hours per day. During the 12-hour overnight period, the secondary chilled water pumps that feed the cooling coils are turned off. All chiller capacity is directed towards recharging the storage tank. Warm water in the tank is chilled and returned to the tank for the next operations day.

   Because only approximately 50% of the chiller capacity is installed in a Partial Storage design, the parasitic load associated with the chiller system is halved, leaving more net electrical power to be delivered to the grid.

2. Full Storage: In a full storage design, typically one half of the chiller capacity that would normally be installed is deleted. The remaining half capacity chiller system runs only on the off-peak periods, which could be as few as 8 hours and as long as 16 hours. During the peak period, the stored chilled water is pumped to the GT cooling coils (this pumping pulls very little parasitic electrical load). The chillers do not operate at all during the peak period. The chillers are re-started at the end of the peak period, and the secondary pumps sending water to the coils are shut down. All chiller capacity goes towards recharging the tank.

If approximately 8 hours of coil demand are needed for an installation, then 50% of the nominal chiller capacity (as a percentage of peak coil demand levels) will be installed. This allows for 8 hours of discharge and 16 hours for recharge. If 12 hours of discharge is required, then there will be no decrease in chiller capacity, but there will still be a tank. If a discharge period of 16 hours is required, then upwards of double the nominal chiller capacity is required, because the discharge period of 16 hours is twice as much as the allowable recharge period of 8 hours.

Because the chillers do not operate at all during the peak period, then nearly all of the incremental power output is available on a *net* basis.

For typical water-cooled chiller plants, the addition of TES will typical result in modest capital cost savings. The reason is that the unit cost of "tank capacity" is fairly close to the cost of "chiller capacity". (However, for projects that use air-cooled chillers, or absorption chillers, where the unit cost of the chiller plant is higher (typically 40% to 50%), then displacing expensive chiller capacity with tank capacity will decrease overall capital costs of the project.)

Pictured below is a 6+ million gallon tank installed in 1999 for a partial storage design involving 3 W501D5s GTs.

In the table below, we show the relative cost estimates for several options. Most chiller projects (since 1987) have been "on-line chilling", which means that there is no thermal storage. In the 1990's, there were several projects that explored ice storage, but as the chilled water technology has matured, we expect no more ice storage projects due to the much higher efficiency and reliability as well as cost benefits offered by chilled water storage vs ice storage systems.

Most chilled water storage projects have been of the partial storage design; but we expect more full storage designs in the future. This is because thermal storage represents one of the most cost-effective and efficient means of shifting on-peak MW to off-peak hrs and full-storage TES maximizes the amount of MW that is shifted.

	On-Line	Partial TES	Full TES	Full TES	Hybrid
Discharge Hours	24	12	8	12	16
Charge Hours	N/A	12	16	12	12
Relative capacity of chillers	100%	50%	50%	100%	133%
Tank Size (Million gallons)	0	3.5	4.6	6.9	3.1
Capital Cost (millions USD)	$12.0	$11.25	$11.75	$16.75	$11.75

In order to attain an economical TES installation, it is necessary to employ a "low flow" chilled water design plan. The heat capacity of water is only 1 BTU per pound per degree change in temperature. By comparison, one pound of ice creates 144 BTUs of latent energy during melting. So, the low flow plan demands a high change in water temperature (high delta T) across the cooling coils. Where many chiller systems have delta T's as low as 12 to 15 degrees F, it is important to "open up the delta T" in a chilled water TES design, using a delta T of 20 to 30 degrees or more. This is best accomplished by operating several smaller chillers in series instead of operating fewer larger chillers in parallel. The series chilled water configuration breaks down the chilling work into smaller progressively colder temperature segments (sequential cooling), which improves chiller output and efficiency.

Low flow technology creates two advantages:

- The first is that the pumping horsepower of moving chilled water is decreased due to the lower flow rates thus reducing the parasitic power. There can also be a modest decrease in secondary pump capital costs.
- The second reason is that the size (and capital cost) of the storage tank is greatly diminished when there is a higher water delta T.

	On-Line	Partial TES	Full TES	Full TES	Hybrid
Discharge Hours	24	12	8	12	16
Peak Parasitic Load, kW	8,900	4,600	300	300	300
Case 1 incremental power	51,289	55,889	59,889	59,889	59,889
Unit Cost $ / kW	$234	$202	$196	$280	$196
Incremental Heat Rate	7,157	6,603	6,129	6,129	6,129
Case 2 Incremental Power	23,108	27,708	32,308	32,308	32,308
Unit Cost $ / kW	$519	$406	$364	$518	$364
Incremental Heat Rate	8,420	7,022	6,022	6,022	6,022

The above table shows the extent of potential unit prices for retrofit options. The options for less than $400 per kW should appear to be attractive for the quality of the output (summer peaking) and the concurrent low heat rate. The options near $200 should be attractive at pretty much any heat rate, but are outstanding at the heat rate shown.

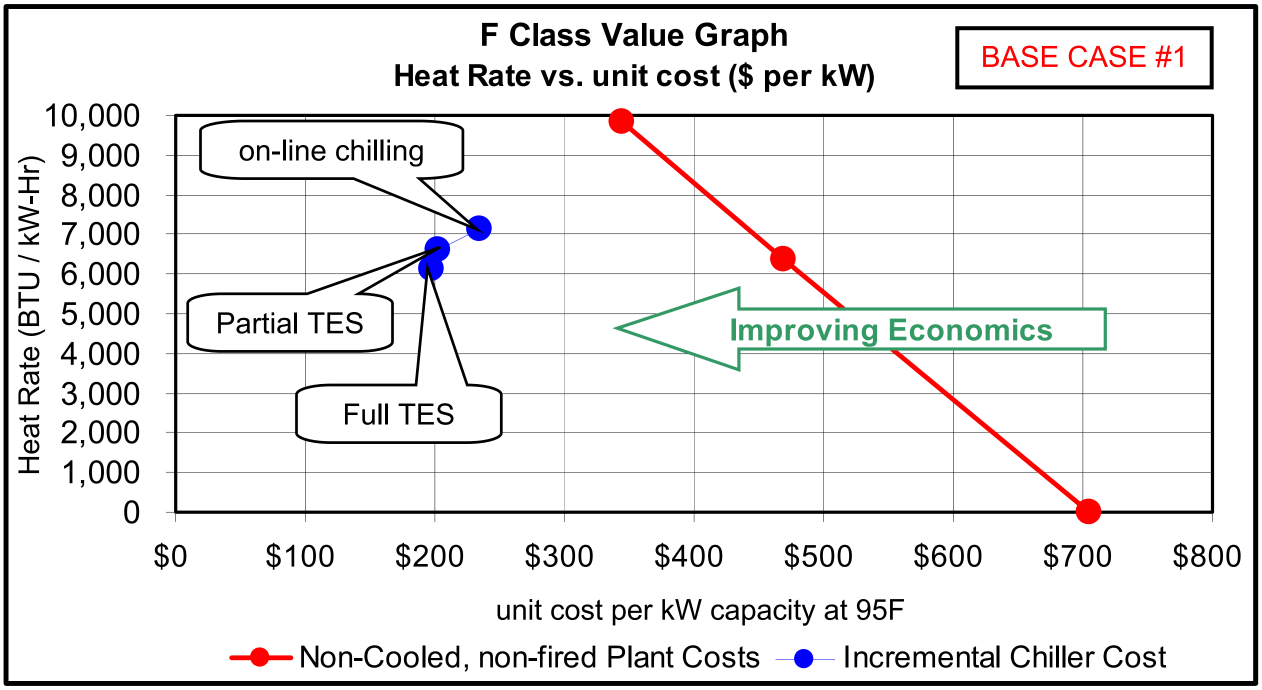
While the 8-hour Full TES option is the lowest cost plan for retrofit, the higher discharge hour Full TES plan (such as 12 full hours of discharge) increase in cost quickly. The planner should know that a system that is based on a single design point for 8 hours will likely provide upwards of 12 hours on a “real” day, when the actual ambient conditions are less than the design point for virtually all hours of the day. A 24-hour weather profile can help in fine-tuning the size of the chiller plant and storage tank for more realistic ambient weather conditions.

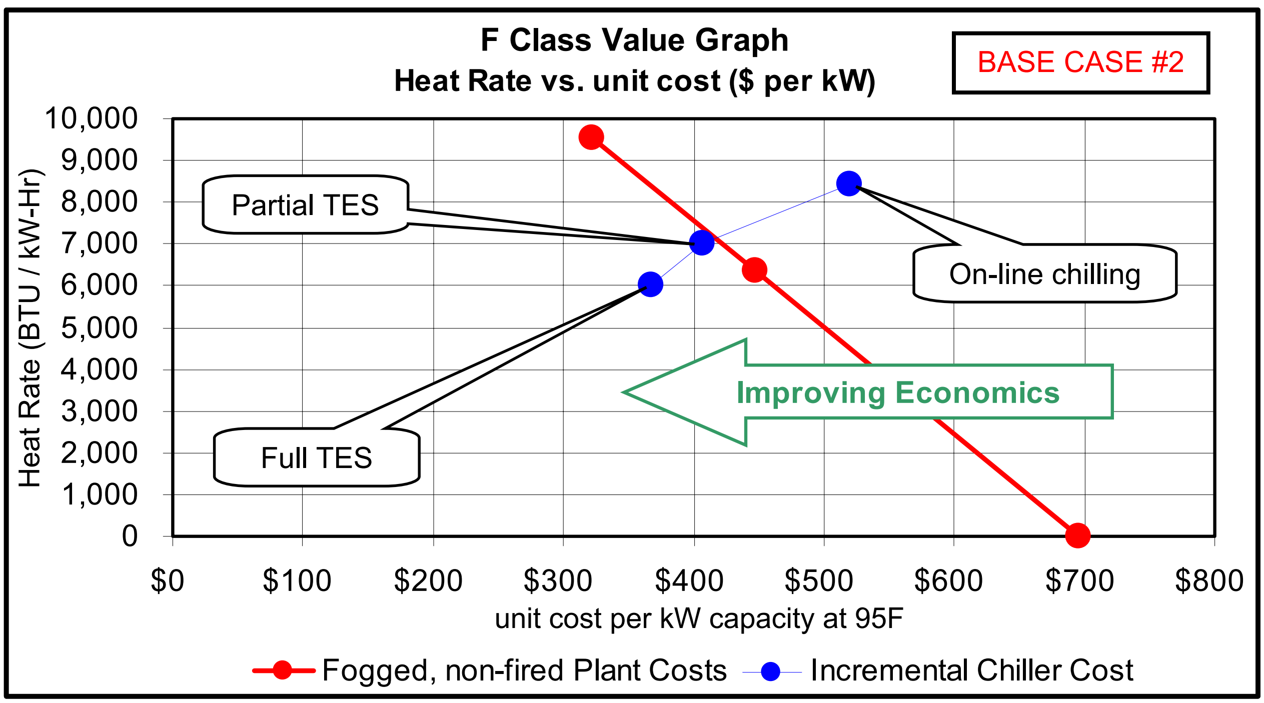
Likewise, a 12-hour Full TES system could provide closer to 16 hours of discharge, particularly if a “hybrid” operating profile is undertaken where the early morning and late evening peak hours are used for chiller operation.

Perhaps the most compelling design is the Hybrid system operates like a Full Storage system during the hottest 8 hours of the day (“Super-Peak”), and like a Partial Storage system during the remaining 8 peak hours of the day (what we call the “shoulder hours”). The unit cost ($/kW) is shown for the super-peak operating hours.

The advantage of a TES system is that once the system is installed, the operator can choose to run the system as a Partial, Hybrid, or Full storage system, depending on changing economic conditions.

Having another look at the relative economics of the chilled water options, this time including TES, provides the following results, shown graphically below.





As shown in the preceding graphics, for the Base Case 1 plant that had no previous cooling, the TES options improve the relative economics of chilling. However, because the costs of all the chilling options are already low, the relative impact does not seem as great. Nonetheless, the economic improvement for the TES options are measurable and attractive, and there are additional qualitative reasons for including TES in all projects.

For the Base Case 2 plant that already had foggers, the two chilling options put the relative economics on the correct side of the evaluation line. The incremental economics of the partial storage design are very similar to the base plant economics. The Full Storage economics (8 hours discharge) are far superior to the other chilling options.

It could be said by detractors that the incremental heat rate figures shown for the Partial and Full Storage cases are artificially low, because the parasitic chiller electrical load is taken at night (or off-peak) and is not counted in the instantaneous heat rate equation. This may be true. However, because of the great disparity of electrical prices between peak and off-peak periods, the use of the chiller electrical draw in the heat rate equation would not be an apples-to-apples comparison. After all, heat rate is a proxy for an economic ratio analysis. We recommend that the cost of operating the chillers at night be categorized as a non-fuel operating expense.

In fact, in some locations, clearing prices for electricity drop below the marginal cost of fuel during the off-peak period. Many combined cycle plants attempt to cycle down to minimum load during the off-peak period. This could mean taking both GT's to minimum load, or even taking one GT off line completely. During this time, the plant heat rate is very adversely affected, causing the plant to lose money overnight. Therefore, the electrical draw of the chillers during the off-peak period can in some cases actually *increase* daily net revenue by displacing money-losing kilowatts.

Commentary

The US power industry went on what some might consider to be a drunken rampage from 1998 to 2001. The hangover kicked in with the combined events of the Enron collapse and the 9-11 attack. Since 2002, the industry has been recovering, but not without the exit of several large players from the Merchant Industry. A partial rollback is forming to an industry that looks a little more like the 1992-1998 model: assets backed with PPA's.

Plants that were designed for Base Load have been lucky to achieve Intermediate Dispatch. Intermediate Plants have been relegated to daily Peakers in the summer only. The industry is learning the hard way what the costs are for cycling F-Class combined cycle plants: in GT LTSA costs, HRSG thermal cycling, increased operator labor, environmental violations for excess NOx during start-ups, etc. These issues, along with low capacity factors and soaring fuel prices, have made a mockery of the original financial pro-forma plans for these plants.

Is there any silver lining in this bleak outlook? One thing that is known for sure is that the only period of high clearing prices for electricity is during the hottest hours of the summer. A recent winter cold snap in New York and New England in early 2004 saw low electrical clearing prices, tied closely to the marginal cost of oil at approximately $70 per MW-Hr, when plant operators sold their gas supplies. Not even a threatened electrical shortage could draw electrical prices up into the triple digits.

We have heard much about the overcapacity glut in Texas, the Southeast and New England. The state of Mississippi, as an example, is littered with half-finished projects. "Over-capacity" as reported in the media and even the trade press must be a relative term; because there have been several "close calls" in the summer of 2004 with respect to power supply during heat waves, particularly in California. Florida and the Front Range of the Rocky Mountains have on average finally gone from being winter peaking regions to summer peaking.

On the international side, new gas turbine power projects are pretty much within the tropical and subtropical regions of the globe: Viet-Nam, China, the Gulf States region, Mexico, etc. So far (as of the time of the writing of this paper), Europe has thankfully not seen a repeat of the devastating heat wave and power shortage that took so many lives in 2003; but a more powerful warning sign for failure to adequately plan could not possibly be imagined.

The common theme is that summer weather, and the increasing use of air-conditioning, is going to drive the power construction demand for the foreseeable future. The use of "ISO Rating" for power plants continues to become an anachronism. Power plant operators and developers should have by now learned the economic lessons of not designing their assets for the economic realities of the market: hot-weather power sells consistently and profitably. Fortunately there is a fairly easy way to retrofit these assets to "take weather out of the equation" to produce above-ISO power regardless of the ambient temperature.

Conclusion

In this second Phase of this report, we have analyzed the retrofit capabilities of Chilling to existing F Class combined cycle plants. The reason for exploring the retrofit economics of chilling are clear: there are comparatively few Greenfield power projects planned for the next few years, as the power demand grows into the existing excess reserve margin. However, as reserve margins grow small, as they always will, fleet operators will be looking for the next supply of incremental peaking capacity. It is expected that the recovered financial conservatism of the industry will force planners to look for power augmentation sources within their existing fleet before expanding into new Greenfield projects.

For the few F-Class combined cycle plants that don't yet have any form of inlet cooling, a chiller retrofit represents an economical and safe project that combines a capital cost ($/kW) less than a dedicated peaker while maintaining incremental $/kW-Hr O&M costs approx 1/10 of that of the base plant, and with a heat rate near that of a combined cycle plant.

Even for plants that already have foggers, there is a growing voice of dissatisfaction within the fleet over the real-world performance of this technology. (Surprisingly, the evaporative cooler is regaining the ground it lost to foggers a few years ago. Evaporative coolers are more reliable, predictable; and if properly constructed, do not contribute to moisture carryover.) We expect several more plants that were built with foggers to make the upgrade to chillers.

Chilling technology can be more expensive and less efficient on a retrofit basis. This is pretty much a truism for almost any technology: after-thoughts are just not a good substitution for good up-front planning. Designs for filter-house retrofit kits make the retrofit of chillers less intrusive to the front end of the GT; but the fixed-geometry of the back-end of the plant BOP (HRSG, etc.) makes the recovery of all waste heat slightly more difficult.

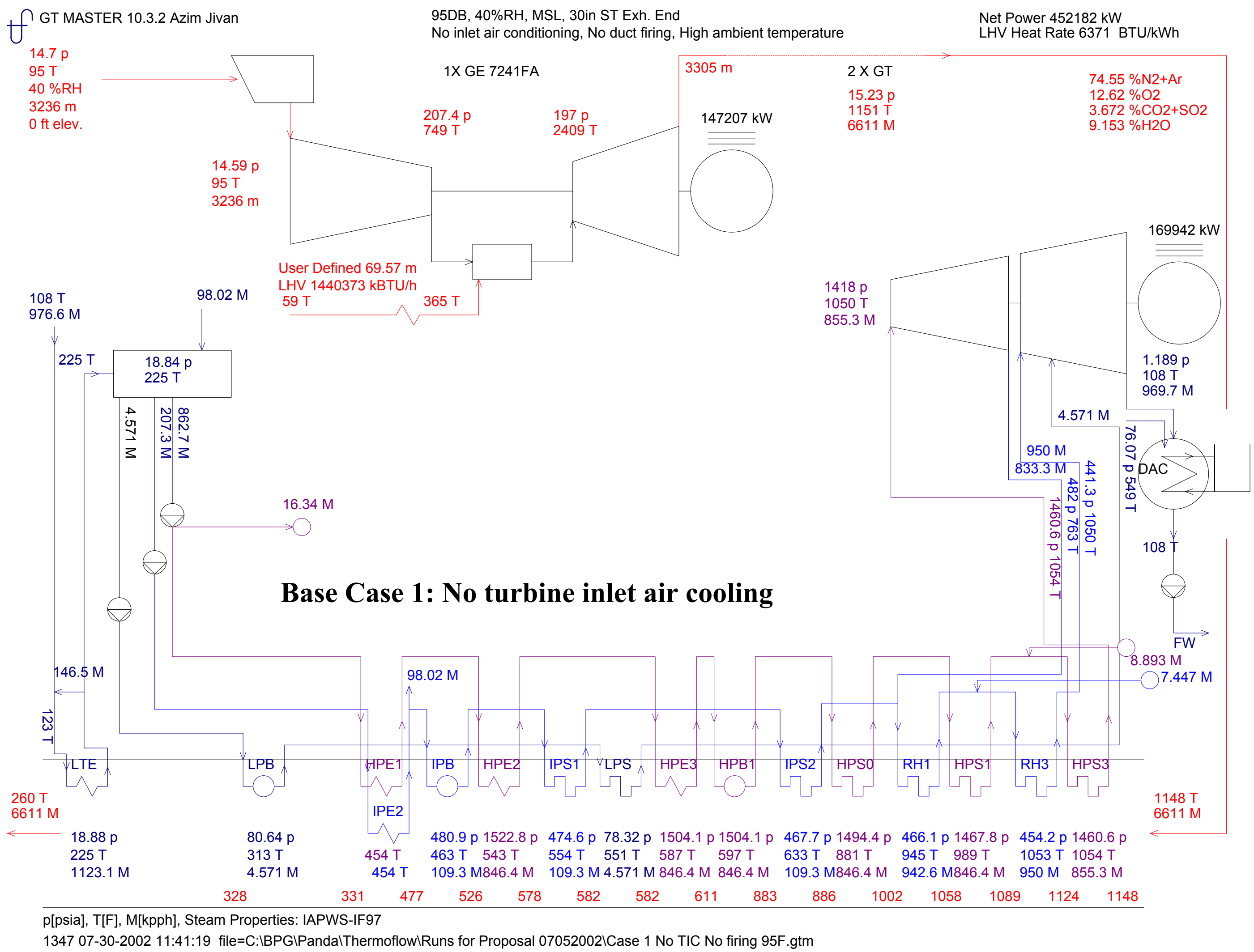
Nonetheless, the economic analysis for a chiller retrofit shows that there is a compelling financial argument, particularly when TES is employed in the plan. (For that matter, TES makes just as much sense for Greenfield projects and for simple-cycle plants.)

For a typical combined cycle plant on a typical summer day, the addition of chillers can yield an incremental 51,289 kW for a previously unchilled plant and 23,108 kW for a previously fogged plant. This comes at an approximate all-in capital cost of $12M dollars. While unit price alone is usually too simplistic a measure of the value of a project, the unit price of a chiller retrofit could be as little as $196/kW for a plant with no previous cooling (or for decommissioned foggers) and $364/kW for a plant that still operates their foggers. Even at the high end of this range, it would be difficult to site and construct a dedicated peaking plant for this cost, and certainly not at the chiller incremental heat rate.

APPENDIX A

Graphical Output of the power plant cycle for the following Cases:

- Base Case 1: No turbine inlet air cooling.
- Base Case 2: Turbine inlet air conditioning using foggers.
- Case 1: plus chiller retrofit
- Case 2: plus chiller retrofit



GT MASTER 10.3.2 Azim Jivan
95DB, 40%RH, MSL, 30in ST Exh. End
No inlet air conditioning, No duct firing, High ambient temperature
Net Power 452182 kW
LHV Heat Rate 6371 BTU/kWh
14.7 p
95 T
40 %RH
3236 m
0 ft elev.
1X GE 7241FA
3305 m
2 X GT
15.23 p
1151 T
6611 M
74.55 %N2+Ar
12.62 %O2
3.672 %CO2+SO2
9.153 %H2O
207.4 p
749 T
197 p
2409 T
147207 kW
14.59 p
95 T
3236 m
169942 kW
User Defined 69.57 m
LHV 1440373 kBTU/h
59 T
365 T
1418 p
1050 T
855.3 M
108 T
976.6 M
98.02 M
225 T
18.84 p
225 T
1.189 p
108 T
969.7 M
4.571 M
207.3 M
862.7 M
4.571 M
950 M
833.3 M
76.07 p 549 T
441.3 p 1050 T
482 p 763 T
1460.6 p 1054 T
DAC
16.34 M
108 T
Base Case 1: No turbine inlet air cooling
FW
146.5 M
8.893 M
98.02 M
7.447 M
123 T
LTE
LPB
HPE1
IPB
HPE2
IPS1
LPS
HPE3
HPB1
IPS2
HPS0
RH1
HPS1
RH3
HPS3
260 T
6611 M
IPE2
1148 T
6611 M
18.88 p
225 T
1123.1 M
80.64 p
313 T
4.571 M
454 T
454 T
480.9 p
463 T
109.3 M
1522.8 p
543 T
846.4 M
474.6 p
554 T
109.3 M
78.32 p
551 T
4.571 M
1504.1 p
587 T
846.4 M
1504.1 p
597 T
846.4 M
467.7 p
633 T
109.3 M
1494.4 p
881 T
846.4 M
466.1 p
945 T
942.6 M
1467.8 p
989 T
846.4 M
454.2 p
1053 T
950 M
1460.6 p
1054 T
855.3 M
328
331
477
526
578
582
582
611
883
886
1002
1058
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1148
p[psia], T[F], M[kpph], Steam Properties: IAPWS-IF97
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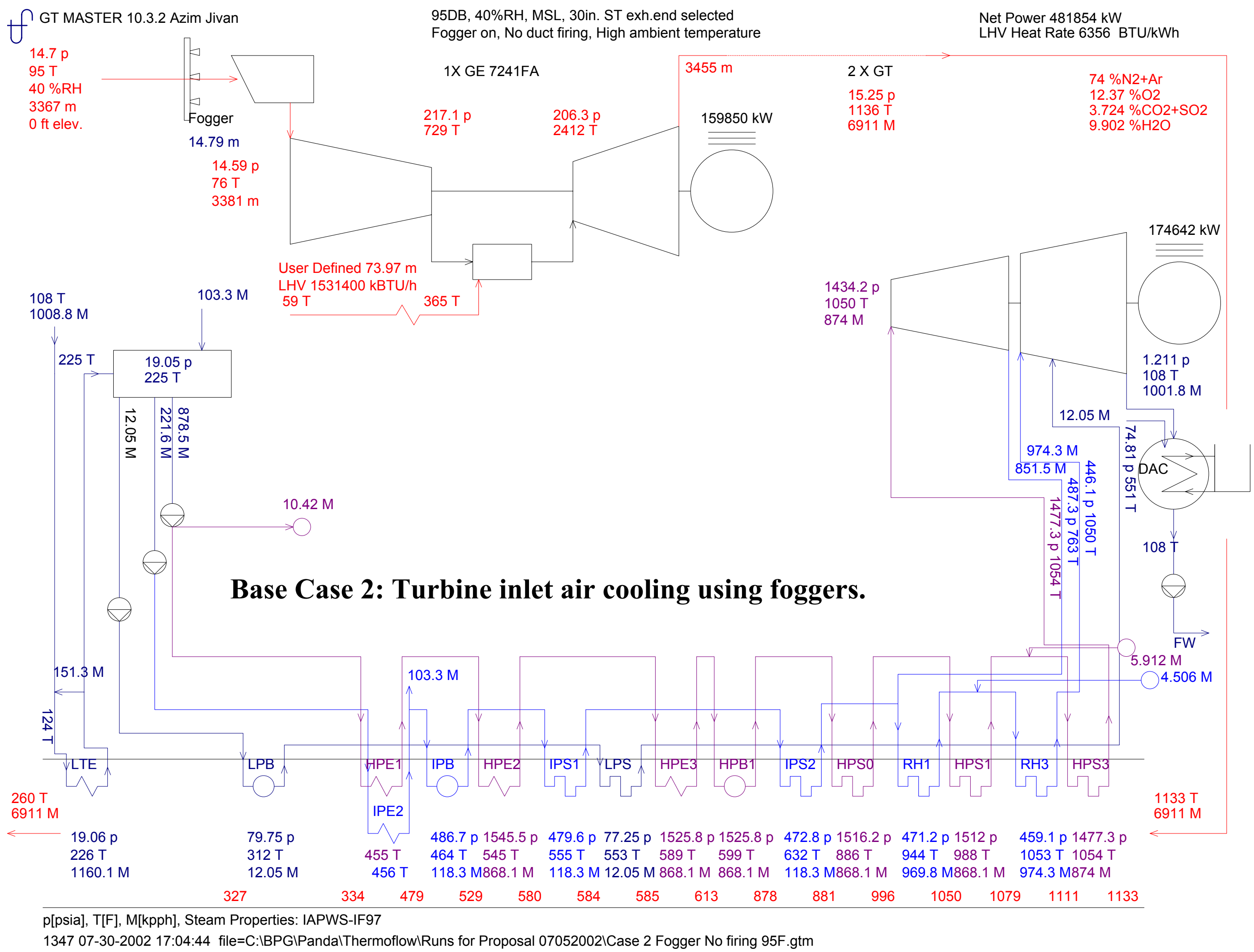
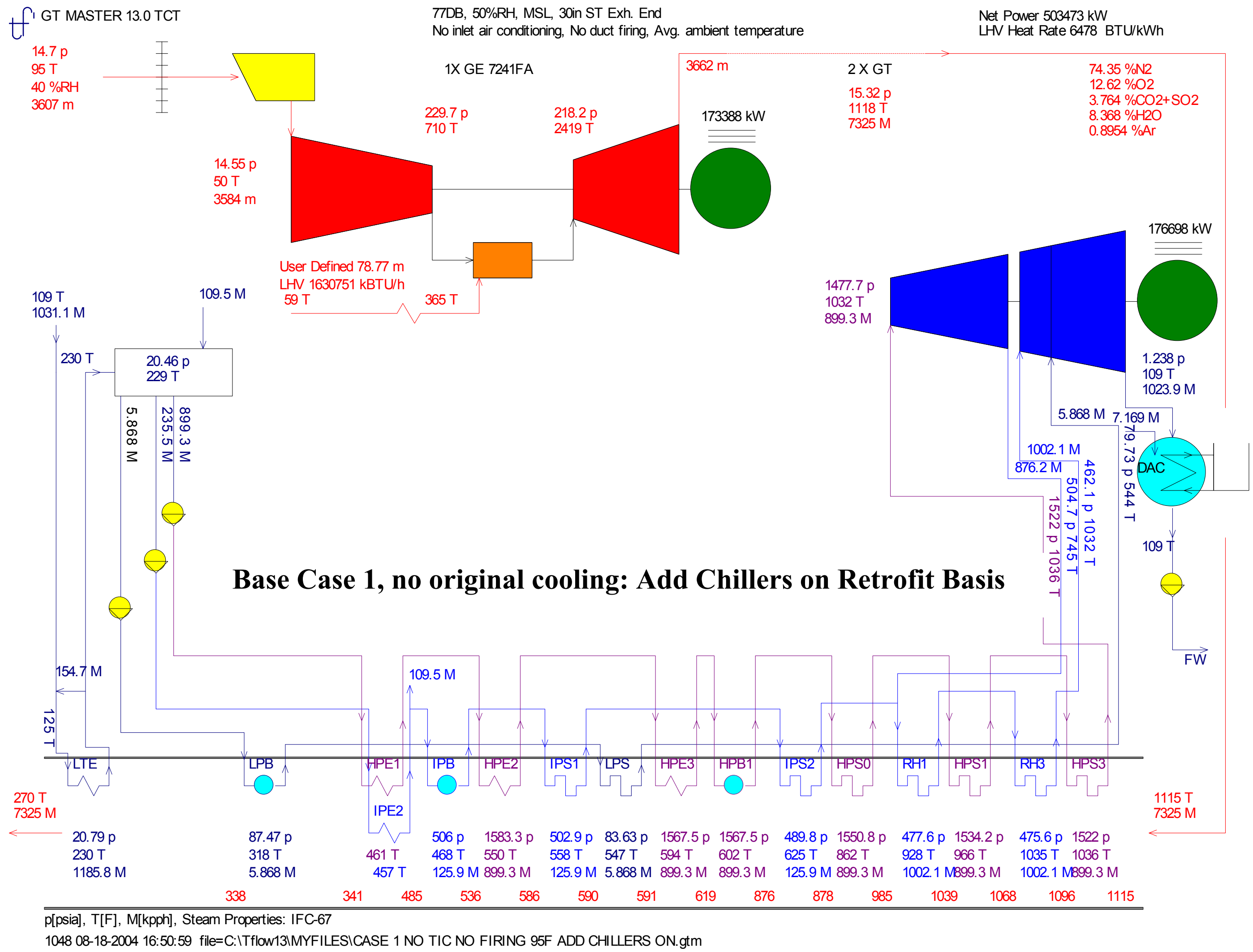
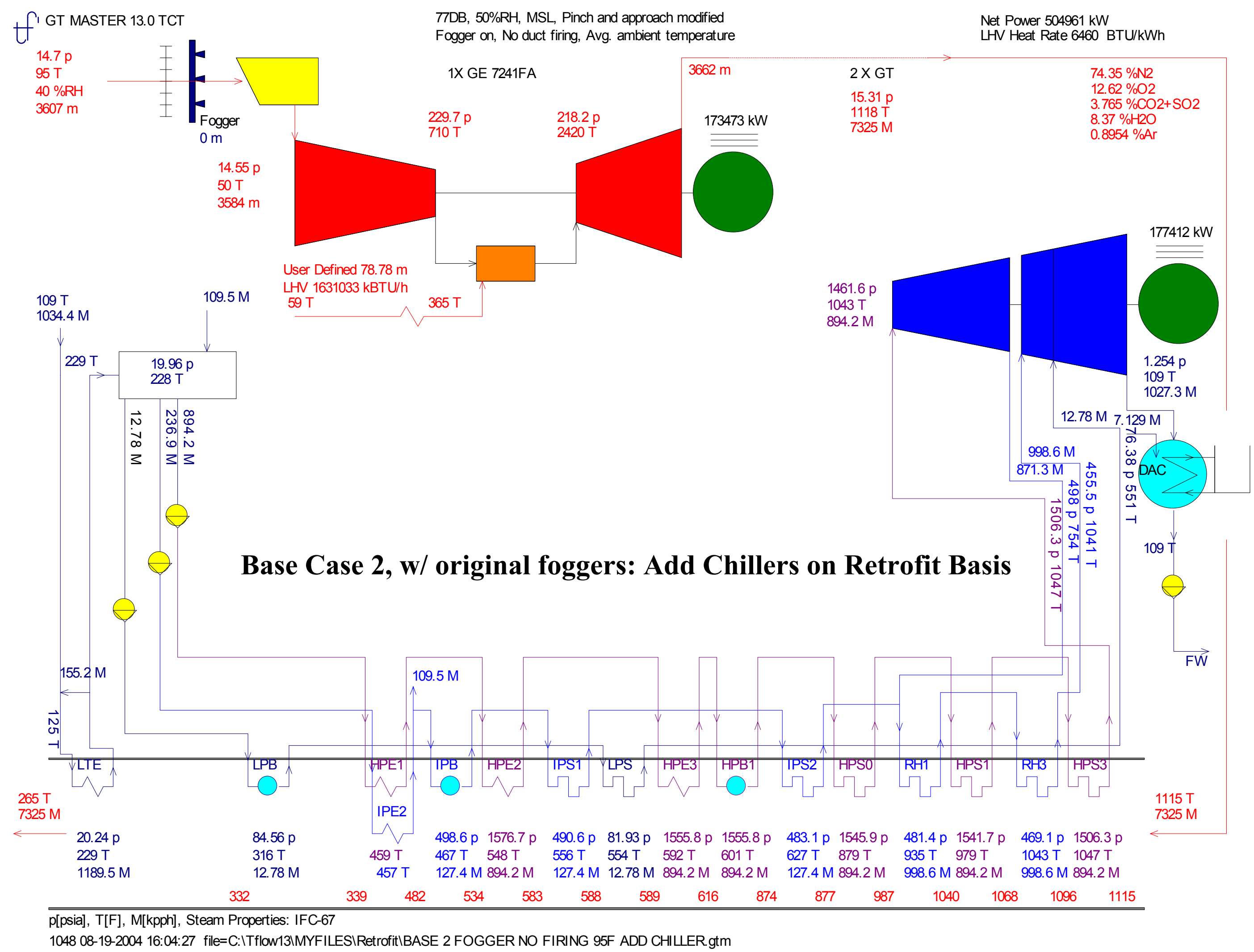


Base Case 2: Turbine inlet air cooling using foggers.


GT MASTER 13.0 TCT
77DB, 50%RH, MSL, 30in ST Exh. End
No inlet air conditioning, No duct firing, Avg. ambient temperature
Net Power 503473 kW
LHV Heat Rate 6478 BTU/kWh
14.7 p
95 T
40 %RH
3607 m
1X GE 7241FA
3662 m
2 X GT
15.32 p
1118 T
7325 M
74.35 %N2
12.62 %O2
3.764 %CO2+SO2
8.368 %H2O
0.8954 %Ar
229.7 p
710 T
218.2 p
2419 T
173388 kW
14.55 p
50 T
3584 m
User Defined 78.77 m
LHV 1630751 kBTU/h
59 T
365 T
176698 kW
1477.7 p
1032 T
899.3 M
1.238 p
109 T
1023.9 M
109 T
1031.1 M
109.5 M
230 T
20.46 p
229 T
5.868 M
235.5 M
899.3 M
5.868 M
7.169 M
1002.1 M
876.2 M
79.73 p 544 T
462.1 p 1032 T
504.7 p 745 T
1522 p 1036 T
DAC
109 T
FW
Base Case 1, no original cooling: Add Chillers on Retrofit Basis
154.7 M
109.5 M
125 T
LTE
LPB
HPE1
IPE2
IPB
HPE2
IPS1
LPS
HPE3
HPB1
IPS2
HPS0
RH1
HPS1
RH3
HPS3
270 T
7325 M
1115 T
7325 M
20.79 p
230 T
1185.8 M
87.47 p
318 T
5.868 M
461 T
457 T
506 p
468 T
125.9 M
1583.3 p
550 T
899.3 M
502.9 p
558 T
125.9 M
83.63 p
547 T
5.868 M
1567.5 p
594 T
899.3 M
1567.5 p
602 T
899.3 M
489.8 p
625 T
125.9 M
1550.8 p
862 T
899.3 M
477.6 p
928 T
1002.1 M
1534.2 p
966 T
899.3 M
475.6 p
1035 T
1002.1 M
1522 p
1036 T
899.3 M
338
341
485
536
586
590
591
619
876
878
985
1039
1068
1096
1115
p[psia], T[F], M[kpph], Steam Properties: IFC-67
1048 08-18-2004 16:50:59 file=C:\Tflow13\MYFILES\CASE 1 NO TIC NO FIRING 95F ADD CHILLERS ON.gtm



Base Case 2, w/ original foggers: Add Chillers on Retrofit Basis