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# Thermal and Economic Analyses of Energy Saving by Enclosing Gas Turbine Combustor Section

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**ABSTRACT**

An infrared (IR) thermography inspection indicated a high-temperature area (500~560°F) at the combustor section of the GE Frame 5 gas turbine of Dynegy Gas Processing Plant at Venice, Louisiana. To improve the thermal efficiency and reduce energy cost, thermal and economic analyses are conducted by enclosing the hot combustor section. This paper presents the methodology and procedure developed to calculate the thermal losses from the gas turbine combustor surface with and without the enclosure. Both natural convection and thermal radiation are considered. Iteration is employed to obtain the enclosure surface temperature. Sensitivities of the thermal losses to surface temperature, emissivity and radiation view factor are examined. It is found that the thermal losses from the combustor surface of the studied gas turbine are about 0.26 MMBtu/h (77.35 kW). The proposed enclosure can reduce the thermal loss to 0.029 MMBtu/h (8.5 kW). The saving is equivalent to about 0.17% in fuel saving. The return of investment is estimated to be one year. However, implementation of this proposed insulation enclosure requires approval from the turbine manufacturer.

## 1. INTRODUCTION

Venice Gas Process Plant is located approximately 75 miles southeast of New Orleans, Louisiana and is operated by Dynegy Midstream Services, LLP. The Venice facilities include an offshore natural gas gathering system, a cryogenic gas processing plant, a lean-oil processing plant, a liquids fractionation facility, an underground gas liquids storage facility, and a multi-barge gas liquids terminal. The facility processes about 800 MCF of natural gas per day and is expanding to a gas processing capacity of 1.3 billion cubic feet per day (BCFD).

Due to competition within the natural gas industry, the Venice plant is seeking various means to reduce cost. As part of the project to improve the energy efficiency of the plant and thus reduce energy costs, Dynegy contracted the Energy Conversion & Conservation Center (ECCC) of University of New Orleans to inspect selected energy-intensive devices using infrared thermograph technology. The inspected results were reported in [1]. The IR results indicated a high-temperature area (500~560°F) at the combustor section of the GE Frame 5 gas turbine, as shown in Fig. 1. The objective of this project is to assess the energy loss and the potential energy (or natural gas) savings by enclosing the combustor section.

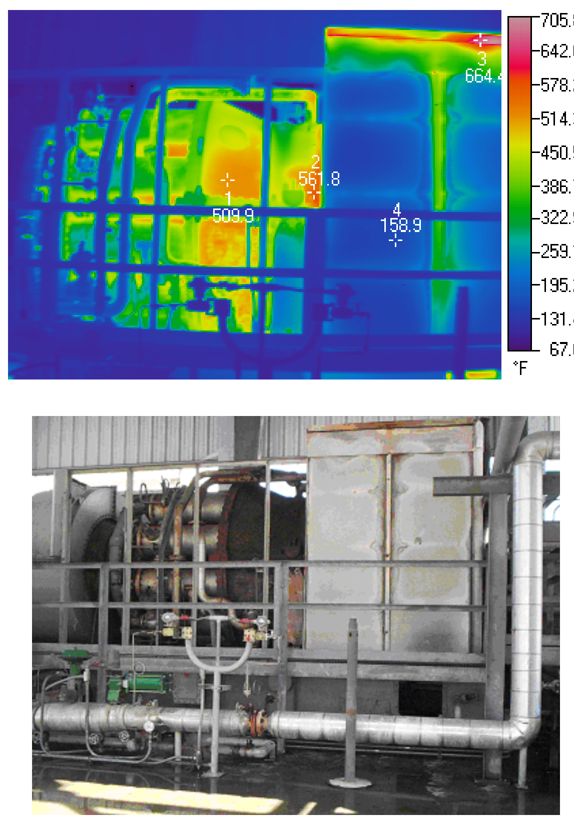


**Figure 1: IR image of the combustor section of GE Frame 5 gas turbine and the corresponding photo.**

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## 2. ANALYSIS AND RESULTS

2.1 Geometry and Dimensions

The dimensions of the combustion section are obtained from the drawings in the gas turbine manual provided by Dynegy [2]. Figure 2 shows part of the drawings used in this study. Since some of the dimensions can't be read directly from the drawings, estimations are made by actually measuring the drawing and then multiplying the scale. As a result, Figure 3 shows the major dimensions applied in the calculation of this study.

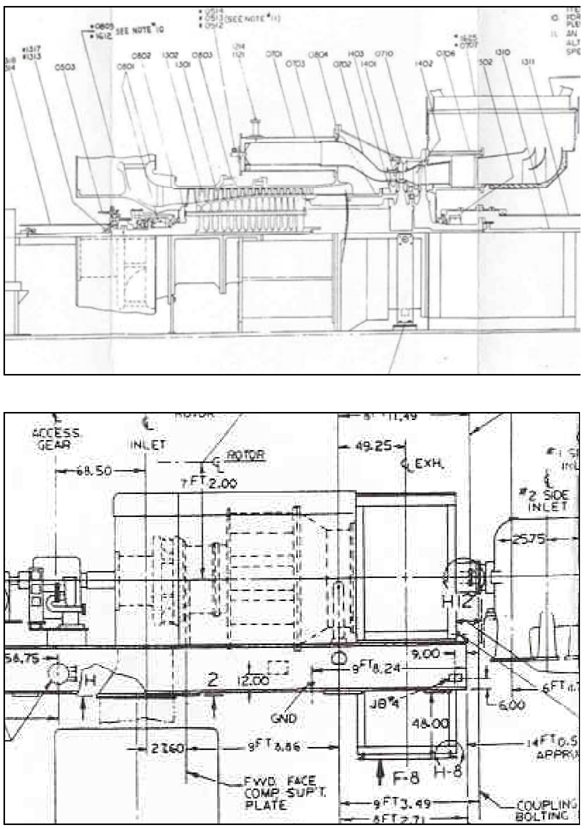


**Figure 2: Part of the drawings used to obtain the geometry and dimensions of the combustor section**

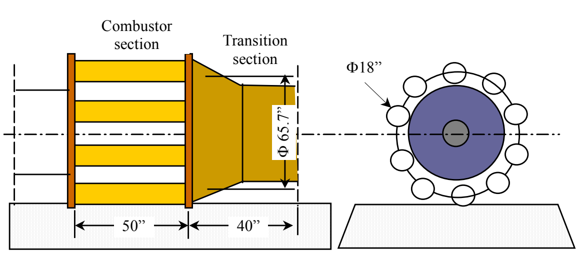


**Figure 3: Simplified dimensions used in the analysis.**

The 10 combustion chambers are assumed to be cylinders with an outer diameter of 18 inches and a length of 50 inches. The dump-diffuser, which encloses the transition piece, shapes like a frustum and a cylinder with the diameter changing from 54.9 inches to 76.5 inches. For simplicity, it is considered a big cylinder with a length of 40 inches and a constant (averaged) diameter of 65.7 inches.

2.2 Models and Assumptions

The basic procedure in this study is to estimate the thermal losses from the combustor section under the existing situation without insulation, and then evaluate the potential energy savings by enclosing the combustor section.

The thermal losses are calculated by considering natural convection as well as thermal radiation. Empirical equations of a heated cylinder are used for the natural convection heat transfer. The surface temperature is first assumed to be constant with an averaged value of 450 $^{o}$F (232 $^{o}$C) based on the IR images. Two other average temperatures (400 and 500 $^{o}$F) are later used to examine the sensitivity of the thermal losses to the surface temperature variations.

2.2.1 Natural Convection

In general, the natural convection heat transfer can be correlated with the Grashof number (Gr), which is defined as

$$Gr = \frac{g\beta\Delta T d^3}{\nu^2} \quad (1)$$

where $\beta$ is the volumetric expansion coefficient and is equal to $1/T$ for ideal gases as used in this study. $\Delta T$ is the temperature difference between the heated surface and the surrounding fluid, d is the length scale or the diameter of the cylinder in this study, g is the gravity acceleration, and $\nu$ is the fluid kinematic viscosity. The characteristic temperature is evaluated as the average film temperature by averaging the wall temperature ($T_w$) and the ambient temperature ($T_0$).

$$T_m = (T_w + T_0)/2$$

The heat transfer coefficient (h) is scaled to a non-dimensional number, Nusselt number (Nu), by

$$Nu_d = hd/k$$

where k is the fluid heat conductivity. While more complicated formula are presented in different references, a general form of the relationship between Nusselt number and Grashof number can be given as

$$Nu_d = c(GrPr)^n \quad (2)$$

where the constants c and n are experimentally obtained, and Pr is the Prandtl number. When Gr = $10^4 \sim 10^7$, which means a laminar flow, the correlation becomes [3]

$$Nu_d = 0.48(GrPr)^{0.25} \quad (3)$$

When the Grashof number is higher ($10^7 \sim 10^{12}$), the following equation should be used,

$$Nu_d = 0.125(GrPr)^{0.333} \quad (4)$$

Once the Nusselt number is known, the heat transfer coefficient can be obtained.

$$h = \frac{Nu_d k}{d} \quad (5)$$

The heat flux due to the natural convection will be obtained by

$$q''_{conv} = Ah\Delta T \quad (6)$$

where A is the area of heat transfer surface, and $\Delta T$ is the same temperature difference as in Eq. (1).

2.2.2 Thermal Radiation

Radiation heat transfer is given by

$$E = \varepsilon\sigma(T_w^4 - T_0^4) \quad (7)$$

where $\varepsilon$ is the surface emissivity and $\sigma$ is the Stefan-Boltzmann constant ($5.67 \times 10^{-8}$ W/m$^2$K$^4$). The temperatures in this equation need to be in Kelvin. Considering that not all the surfaces can radiate thermal energy directly to the ambient, a view factor is incorporated to take into account only the radiation heat flux effectively transferred. The equation for net radiation heat transfer exchange between two gray, diffuse surfaces at $T_1$ and $T_2$ is given as

$$q''_{rad} = \sigma(T_1^4 - T_2^4)/(\frac{1-\varepsilon_1}{\varepsilon_1 A_1} + \frac{1}{F_{12}A_1} + \frac{1-\varepsilon_2}{\varepsilon_2 A_2})$$

where $\varepsilon_1$ and $\varepsilon_2$ are the emissivity, and $A_1$ and $A_2$ are the areas of Surfaces 1 and 2, respectively. $F_{12}$ is the view factor from Surface 1 to 2. If $A_1 << A_2$ and $\varepsilon_2$ is not very small, the net radiation heat transfer exchange between two surfaces can be simplified to:

$$q''_{rad} = \varepsilon_1 A_1 F_{12}\sigma(T_1^4 - T_2^4)/[(F_{12}(1-\varepsilon_1) + \varepsilon_1)] \quad (8)$$

which is used later to calculate the radiation between the combustor and the enclosure layer in this study. The surfaces are assumed gray (independent of radiation wave length) and diffuse (independent of radiation direction) in all the calculations of this study.

2.3 Assessment of Thermal Loses without Insulation

The thermal losses due to natural convection are calculated first. The ambient temperature is assumed to be 59 °F (15 °C). The heat transfer area of the transition section is calculated as a cylinder with a constant averaged diameter of 65.7 inches (1.67 m). The calculation is shown in Table 1. The thermal loss due to natural convection is 26,532 and 90,876 Btu/hr (7.78 and 26.63 kW) for the transition section and the combustion chambers, respectively.

Calculation of the radiation heat transfer without the enclosure is shown in Table 2. Note that the emissivity is assumed to be 0.9 for all the surfaces. The overall view factor for the combustion chamber is approximated as 0.5 since the combustors can "see themselves." The view factor for the transition section is assigned to 1.0. The radiation thermal loss is calculated as 54,048 Btu/hr (15.84 kW) from the transition section and 92,469 Btu/hr (27.10 kW) from all the 10 combustors. Therefore, the total thermal loss from the combustor sector without insulation is **263,929 Btu/hr or 77.35 kW (7.78 + 15.84 + 26.63 + 27.10).**

Calculation is also conducted to examine the sensitivity of the thermal losses to the surface temperature variations. As shown in Table 3, a variation of ±50 °F (±27.8 °C) of the surface temperature results in a –15 ~ +17 kW (-19% ~ +22%) difference of thermal losses. Similarly, the effect of the surface emissivity and view factor on the thermal loss is also examined. The sensitivity studies allow us to obtain a possible range of the results due to variations of the parameter values as well as the assumptions used in this study. The effect of these parameters on thermal losses is listed in Table 3.

**Table 1: Natural convection heat transfer without the enclosure**

| Parameters | Notations | Transition Section | Combustors | Unit |
|---|---|---|---|---|
| Nominal diameter | d | 1.67<br>(65.7) | 0.457<br>(18) | m<br>(in) |
| Ambient temperature | $T_0$ | 15<br>(59) | 15<br>(59) | $^oC$<br>($^oF$) |
| Surface temperature | $T_w$ | 232.22<br>(450)<br>(505.22) | 232.22<br>(450)<br>(505.22) | $^oC$<br>($^oF$)<br>(K) |
| Expansion coefficient | $\beta=1/T$ | 0.001979 | 0.001979 | 1/K |
| Temperature difference ($\Delta T$) | $(T_w-T_0)$ | 217.22 | 217.22 | $^oC$ |
| Average film temperature ($T_m$) | $0.5*(T_0+T_w)$ | 123.61 | 123.61 | $^oC$ |
| Viscosity | $\nu$ | 2.70E-05 | 2.70E-05 | $m^2/s$ |
| Prandtl number | Pr | 0.7 | 0.7 | |
| Heat conductivity of the air | k | 0.034 | 0.034 | J/m-s-K |
| Grashof number (Eq. 1) | Gr | 2.69E+10 | 5.52E+08 | |
| Nusselt number (Eq. 3 or 4) | Nu | 330.02 | 90.43 | |
| Heat transfer coefficient (Eq. 5) | h | 6.719 | 6.728 | $W/m^2$-K |
| Section length | L | 1.016<br>(40) | 1.27<br>(50) | m<br>(in) |
| Heat transfer surface | A | 5.33 | 1.82 | $m^2$ |
| Number of sections | | 1 | 10 | |
| Total heat transfer surface | A' | 5.33 | 18.22 | $m^2$ |
| Total natural convection heat transfer loss | $q_{conv}$ | 7,775.84<br>(26,532) | 26,632.99<br>(90,876) | W<br>(Btu/h) |

**Table 2: Thermal radiation loss without the enclosure**

| Parameters | Notations | Transition Section | Combustors | Units |
|---|---|---|---|---|
| Ambient temperature | $T_0$ | 15<br>(59)<br>(288) | 15<br>(59)<br>(288) | $^oC$<br>($^oF$)<br>(K) |
| Surface temperature | $T_w$ | 232.22<br>(450)<br>(505.22) | 232.22<br>(450)<br>(505.22) | $^oC$<br>($^oF$)<br>(K) |
| Constant | $\sigma$ | 5.67E-08 | 5.67E-08 | $W/m^2K^4$ |
| Emissivity | $\varepsilon$ | 0.9 | 0.9 | |
| Heat flux | E | 2973.65<br>(109,236) | 2973.65<br>(109,236) | $W/m^2$<br>(Btu/hr-$ft^2$) |
| Section length | L | 1.016<br>(40) | 1.27<br>(50) | m<br>(in) |
| Surface | A | 5.33 | 18.22 | $m^2$ |
| View factor | F | 1 | 0.5 | |
| Radiation energy loss, Eq. (8) | $q_{rad}$ | 15,842.72<br>(54,048) | 27,096.27<br>(92,469) | W<br>(Btu/hr) |

**Table 3a: Effect of combustor surface temperature on thermal loss**

| Surface Temperature, $^{o}C$ ($^{o}F$) | Thermal Loss, kW (Btu/hr) | Thermal Loss Variations |
|---|---|---|
| 200.4 (400) | 62.44 (213054) | -19% |
| 232.2 (450) | 77.35 (263929) | Reference |
| 260.0 (500) | 94.08 (321014) | +22% |

**Table 3b: Effect of surface emissivity on thermal loss**

| Surface Emissivity | Thermal Loss, kW (Btu/hr) | Thermal Loss Variations |
|---|---|---|
| 0.8 | 72.58 (247653) | -6% |
| 0.9 | 77.35 (263929) | Reference |
| 0.95 | 79.73 (272045) | +3% |

**Table 3c: Effect of view factor on thermal loss**

| Combustor View Factor | Thermal Loss, kW (Btu/hr) | Thermal Loss Variations |
|---|---|---|
| 0.5 | 77.35 (263929) | Reference |
| 0.6 | 82.77 (282422) | +7% |
| 0.8 | 93.61 (319410) | +21% |

2.4 Assessment of Thermal Loss with the Enclosure

An insulation housing (enclosure) is designed to reduce thermal losses. The thermal losses with the enclosure are calculated and compared with those without the enclosure. The enclosure consists of 3 layers. The inner and outer layers are galvanized metal sheet with a thickness thin enough to ignore the thermal resistance of heat conduction. In addition, it is assumed that the surface has a low emissivity of 0.3. The middle layer is fiberglass of 1-inch in thickness. The enclosure is structured with a semi-cylinder of 120 inches in diameter and two straight walls of 60 inches in height. The length of the enclosure is 120 inches, which covers the entire combustor section. Figure 4 shows the designed insulation housing. Related data is summarized in Table 4, and the overall area is 257.1 $ft^2$ (23.9 $m^2$).

To facilitate the analysis, the thermal circuit with the insulation housing is shown in Fig. 5, which includes different thermal resistance of conduction, convention and radiation. The methodology to calculate the thermal losses with the enclosure is documented as below, where iteration is needed to obtain the final result.

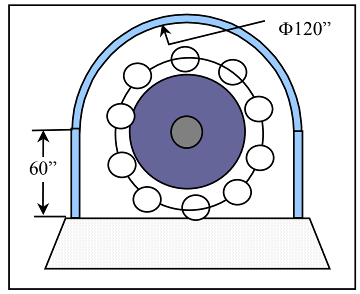


**Figure 4: Schematic of the enclosure structure**

**Table 4: Parameters for the enclosure layer**

| Parameters | Values |
|---|---|
| Thickness of galvanized metal sheet | 0.0015m (1/16 in) |
| Conductivity of stainless steel | 12 W/m-K |
| Thermal emissivity | 0.3 |
| Thickness of fiberglass layer | 0.0254 m (1 in) |
| Thermal conductivity of fiberglass | 0.038 W/m-K |
| Diameter of the semi-cylinder | 3.05 m (120 in) |
| Height of the straight section | 1.52 m (60 in) |
| Length | 3.05 m (120 in) |

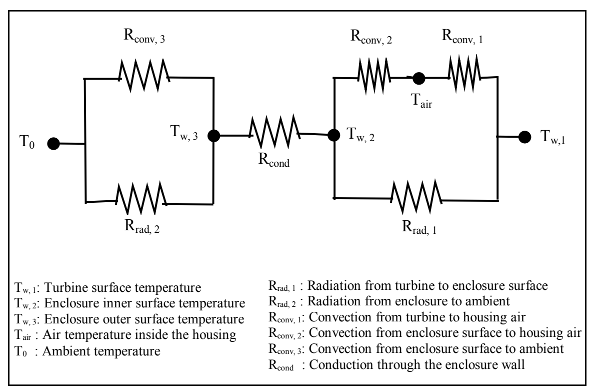


**Figure 5: Thermal circuit of heat transfer from the gas turbine with insulation housing**

Calculation and iteration procedure:

1. Guess the temperature of the outer surface of the enclosure layer. Then calculate the thermal loss from the outer surface, including both natural convection and thermal radiation.

2. Based on the steady-state heat flux of the thermal losses, apply a one-dimensional analysis to the enclosure wall. The temperature of the inner surface of the enclosure can be calculated.

3. With the temperature of the inner surface, the heat transfer between the combustor and enclosure can be calculated including both natural convection and radiation. The bulk temperature of the air inside the insulation housing is considered as the average of the temperature of the enclosure inner surface and the surface temperature of the combustor section.

4. Thermal balance requires the heat flux from the combustor to the inner surface of the enclosure equal to the heat flux from the outer surface of the enclosure. If these two heat fluxes are not equal, the guessed temperature of the outer surface is then decreased or increased. A new calculation begins until the correct temperature of outer surface of the enclosure is found.

Notice that this methodology assumes the temperature of the combustor surface is known. Due to the insulated enclosure, the surface temperature of the combustor section will increase. The actual value depends on the thermal equilibrium between the combustor internal heat transfer to the wall and the heat transfer through the insulation walls, which would complicate this analysis. A simplified method is adopted by examining the effect of combustor wall temperature variation to the thermal loss. To this end, four (4) different combustor surface temperatures are introduced to cover the possible range of combustor wall temperature variations.

Without changing the combustor surface temperature 450$^o$F (232.2$^o$C), the calculation with the insulated enclosure is shown in Tables 5~9. The results show that the outer surface temperature of the enclosure is 126.3 $^o$F (52.4 $^o$C), and the thermal loss reduces to 20,130 Btu/hr (5.9 kW) from previous 263,918 Btu/hr (77.35 kW) without the enclosure. Notice that the external natural convection coefficient of the enclosure with the ambient is estimated with the correlation for a cylinder, and the heat flux is calculated with the actual surface area of the enclosure.

**Table 5: Natural convection from the outer wall of the enclosure.**

| Parameters | | | Units |
|---|---|---|---|
| Nominal diameter | d | 3.05 | m |
| Ambient temperature | $T_0$ | 15 | $^o$C |
| Enclosure outer surface temperature | $T_w$ | 52.39<br>(126.3)<br>(325.39) | $^o$C<br>($^o$F)<br>(K) |
| Expansion coefficient | $\beta$ | 0.003073 | 1/K |
| Temperature difference ($\Delta$T) | ($T_w$-$T_0$) | 37.39 | $^o$C |
| Average film temperature ($T_m$) | 0.5*($T_0$+$T_w$) | 33.69 | $^o$C |
| Viscosity | $\nu$ | 1.70E-05 | m$^2$/s |
| Prandtl number | Pr | 0.7 | |
| Heat conductivity of the air | k | 0.027 | J/m-s-K |
| Grashof number | Gr | 1.11E+11 | |
| Nusselt number | Nu | 528.25 | |
| Heat transfer coefficient | h | 4.676 | W/m$^2$-K |
| Section length | L | 3.05 | m |
| Heat transfer surface | A | 23.90 | m$^2$ |
| Number of sections | | 1 | |
| Total heat transfer surface | A' | 23.90 | m$^2$ |
| Total heat transfer loss | $q_{conv}$ | 4178.69<br>(14,295) | W<br>(Btu/hr) |

**Table 6: Thermal radiation loss from the outer wall of the enclosure.**

| Parameters | | Values | Units |
|---|---|---|---|
| Ambient temperature | $T_0$ | 15<br>(288) | $^oC$<br>(K) |
| Surface temperature | $T_w$ | 52.39<br>(126.3)<br>(325.39) | $^oC$<br>($^oF$)<br>(K) |
| Stefan-Boltzmann constant | $\sigma$ | 5.67E-08 | $W/m^2K^4$ |
| Emissivity | $\varepsilon$ | 0.3 | |
| Heat flux | E | 73.66<br>(2705.5) | $W/m^2$<br>($Btu/hr\text{-}ft^2$) |
| Section length | L | 3.05 | m |
| Surface | A | 23.90 | $m^2$ |
| View factor | F | 1 | |
| Radiation energy loss | $q_{rad}$ | 1760.49<br>(6007.1) | W<br>(Btu/hr) |

**Table 7: Heat conduction within the enclosure wall**

| Parameters | Values | Units |
|---|---|---|
| Thermal flow<br>($q_{conv}$ +$q_{rad}$ ) | 5.9<br>(20131) | KW<br>(Btu/hr) |
| Area (A) | 23.9 | $m^2$ |
| Thermal flux<br>($q_{conv}$ +$q_{rad}$ )/A | 246.9<br>(9068.4) | $W/ m^2$<br>($Btu/hr\text{-}ft^2$) |
| Thickness | 0.0254<br>(1) | m<br>(in) |
| Conductivity | 0.038 | W/m-K |
| Temperature difference<br>($\Delta T$) | 166.1<br>299.0 | $^oC$<br>( $^oF$) |
| Enclosure outer wall temperature | 52.4<br>(126.3) | $^oC$<br>( $^oF$) |
| Enclosure inner wall temperature | 218.49<br>(425.3) | $^oC$<br>( $^oF$) |

**Table 8: Natural convection from gas turbine inside the insulated enclosure**

| Parameters | | Transition Section | Combustors | Units |
|---|---|---|---|---|
| Nominal diameter | d | 1.67<br>(65.7) | 0.457<br>(18) | m<br>(in) |
| Temperature of the inner surface of the enclosure | $T_{w,inner}$ | **218.49**<br>(425.3) | **218.49**<br>(425.3) | $^oC$<br>( $^oF$) |
| Combustor outer surface temperature (Measured) | $T_w$ | 232.22<br>**(450)**<br>(505.22) | 232.22<br>**(450)**<br>(505.22) | $^oC$<br>($^oF$)<br>(K) |
| Expansion coefficient | $\beta$ | 0.001979 | 0.001979 | 1/K |
| Temperature difference ($\Delta T$) | ($T_w$-$T_{w,\ inner}$) | 13.73 | 13.73 | $^oC$ |
| Average/film temperature ($T_m$) | 0.5*($T_{w,\ inner}$ +$T_w$) | 225.36 | 225.36 | $^oC$ |
| Viscosity | $\nu$ | 2.70E-05 | 2.70E-05 | $m^2/s$ |
| Prandtl number | Pr | 0.7 | 0.7 | |
| Heat conductivity of the air | k | 0.034 | 0.034 | J/m-s-K |
| Grashof number | Gr | 1.70E+09 | 3.49E+07 | |
| Nusselt number | Nu | 131.58 | 36.05 | |
| Heat transfer coefficient | h | 2.679 | 2.682 | $W/m^2\text{-}K$ |
| Section length | L | 1.016<br>(40) | 1.27<br>(50) | m<br>(in) |
| Heat transfer surface | A | 5.33 | 1.82 | $m^2$ |
| Number of sections | | 1 | 10 | |
| Total heat transfer surface | A' | 5.33 | 18.22 | $m^2$ |
| Total heat transfer loss | $q_{conv}$ | 195.96<br>(668.9) | 671.18<br>(2290) | W<br>(Btu/hr) |

**Table 9: Thermal radiation from gas turbine with the insulated enclosure**

| Parameters | | Transition Section | Combustors | Units |
|---|---|---|---|---|
| Surface temperature of enclosure (inner wall) | $T_{w, inner}$ | **218.49** (491.49) | **218.49** (491.49) | $^{o}C$ (K) |
| Combustor outer surface temperature | $T_w$ | 232.22 (450) (505.22) | 232.22 (450) (505.22) | $^{o}C$ ($^{o}F$) (K) |
| Stefan-Boltzmann constant | $\sigma$ | 5.67E-08 | 5.67E-08 | $W/m^2K^4$ |
| Emissivity | $\varepsilon$ | 0.9 | 0.9 | |
| Heat flux | E | 346.94 | 346.94 | $W/m^2$ |
| Section length | L | 1.016 | 1.27 | m |
| GT Surface | A | 5.33 | 18.22 | $m^2$ |
| View factor | F | 1 | 0.5 | |
| Radiation energy loss | $q_{rad}$ | 1848.41 (6307) | 3161.39 (1078) | W (Btu/hr) |

The total thermal loss from the enclosure outer wall to the ambient (Tables 5 and 6) is 20,268 Btu/hr (5.94 kW), and the total thermal loss from the gas turbine (Tables 8 and 9) to the enclosure inner wall is 20,063 Btu/hr (5.88 kW) (within 1% error). It can be concluded that the converged surface temperature of the enclosure is 126.3 $^{o}F$ (52.39 $^{o}C$) in this case.

As discussed above, the combustor surface temperature will increase due to the installed enclosure. The sensitivity of the overall thermal loss from the increased combustor surface temperature is estimated by assuming different combustor surface temperature levels as shown in Table 10. Compared with the result of non-insulated case in Table 3, the overall thermal loss is less sensitive to the surface temperature of the combustor due to the insulated enclosure. A 150$^{o}$F increase of combustor surface temperature will increase 44% of the thermal losses, but the absolute value is still small. The highest thermal loss (8.5 kW or 29,000 Btu/hr) in Table 10 is selected for cost saving analysis, which is given with a surface temperature of 600 $^{o}F$, an increase of 150 $^{o}F$ from the baseline case.

2.5 Energy Saving Assessment

The above analysis shows that without insulation, the thermal loss is about 263,918 Btu/hr (77.35 kW). After installing the insulated enclosure, the thermal loss reduces to 29,000 Btu/hr (8.5 kW). Therefore, an energy saving of 234,918 Btu/hr or 68.85 kW (77.35 - 8.5) on the fuel (natural gas) can be gained from installing the enclosure. This reduction of energy loss corresponds to a saving of 3.8 SCFM (108 kg/day) natural gas with a heating value of 55, 000 kJ/kg (68.85 × 24 ×3 600 / 55,000 = 108.2 kg). The total saving is about 2,050 MMBtu per year. Based on the specifications of the Dynegy gas turbine, its shaft output power is 19,530 HP, or an equivalent of 14,364 kW (19,530×0.7355) or 49.0 MMBtu/h. Assuming a thermal efficiency of 36%, the input of fuel will be 39,900 kW (136 MMBtu/h). The percentage of fuel saving can be found as 0.17% (68.85/39,000).

The current market price of the natural gas is at about \$6/MMBtu, which is roughly equivalent to \$0.02 per kilowatt-hour ($6\times3600/10^6/1.055$). Based on this price, the annual capital savings due to the insulation will be \$12,062 (68.85×0.02×24×365). An alternative way to evaluate the savings is to consider the savings of electricity. Assuming the thermal saving can be converted to electricity with an efficiency of 36%, the total saving of electricity will be 68.85×24×365×0.36 = 226,172 kW-hr. With a price of \$0.07/kW-hr, the annual savings will be \$15,832. The cost of installing the insulated enclosure is estimated to be \$10,000. Therefore, the payback period should be less than 1 year. The increased surface temperature in the combustor may cause thermal strain at the component joints. For safety, it is necessary to consult with the gas turbine manufacturer before the insulation enclosure is implemented.

**Table 10: Effect of combustor surface temperature on thermal loss with the insulated enclosure**

| | | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|
| Combustor surface temperature | $^oF$ | 450 | 500 | 550 | 600 |
| Enclosure outer surface temperature | $^oC$ ($^oF$) | 52.4 (126.3) | 56.5 (133.7) | 60.6 (141.1) | 64.6 (148.2) |
| Enclosure inner surface temperature | $^oC$ ($^oF$) | 218.5 (425.3) | 246.1 (474.9) | 274.3 (525.7) | 302.0 (575.6) |
| Thermal loss | KW (Btu/hr) | 5.9 (20,132) | 6.8 (23203) | 7.6 (25932) | 8.5 (29003) |
| Increase percentage | % | 0 | 15 | 29 | 44 |

**Table 11: Analysis of savings with installed enclosure**

| | GE/ Dynegy |
|---|---|
| Power, MMBtu/hr (kW) | 49.0 (14,364) |
| Efficiency (%) | 36 |
| Fuel input, MMBtu/hr (kW) | 136 (39900) |
| Loss , MMBtu/hr (kW) | 0.236 (68.85) |
| Fuel saving (%) | 0.17 |

## 3. CONCLUSIONS

- The thermal loss from the combustor surface of the gas turbine under study is calculated to be about 77.35 kW (0.26 MMBtu/h), which is sensitive to the surface temperature.

- A simple design of an insulation housing is proposed. The proposed insulation enclosure can reduce the thermal loss to 8.5 kW (0.029 MMBtu/h).

- The energy saving from installing the enclosure is estimated to be $12,062 per year. This annual saving will cover the capital cost of enclosure within one year.

- It is necessary to ask the manufacturer if the enclosure can cause any negative impact on reliability and performance before it is installed.

## 4. REFERENCES

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