AIR EXTRACTION IN A GAS TURBINE FOR INTEGRATED GASIFICATION COMBINED CYCLE (IGCC): EXPERIMENTS AND ANALYSIS

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ABSTRACT

This paper presents an investigation of extracting air from the compressor discharge of a heavy-frame gas turbine. The study was aimed to verify results of an approximate analysis: whether extracting air from the turbine wrapper would create unacceptable non-uniformity in the flow field inside the compressor discharge casing. A combined experimental and computational approach was undertaken. Cold flow experiments were conducted in an approximately one-third scale model of a heavy-frame gas turbine; a closely approximated 3-D computational fluid dynamic analysis was also performed. This study substantiated the earlier prediction that extracting air from the turbine wrapper would be undesirable although this method of air extraction is simple to retrofit. Prediffuser inlet is suggested as an alternate location for extracting air. The results show that not only the problem of flow non-uniformity was alleviated with this alternate scheme, but the frictional power loss in the compressor discharge casing was also reduced by a factor of two.

INTRODUCTION

Air-blown, simplified Integrated Gasification Combined Cycle (IGCC) power plants with hot gas cleanup offer superior environmental performance and high thermal efficiency for generating electricity using coal. This concept utilizes a portion of the gas turbine compressor discharge air to oxidize coal in a gasifier for producing the low-Btu coal gas. Particulates and sulfur bearing gases are removed before the coal gas is burnt in the gas turbine combustor(s). IGCC systems are becoming commercially attractive partly because of system simplification by integration and recent advances in hot gas cleanup (Notestein, 1990; Todd and Allen, 1991; Becker and Schetter, 1992; Todd, 1993).

Several options are available to configure an IGCC system (Corman and Todd, 1993). In many of these options the gas turbine is integrated with the coal gasifier which requires a scheme to extract compressor discharge air. In heavy-frame gas turbines, the air exiting the compressor is decelerated in an annular prediffuser before it is discharged into a dump diffuser. Depending on the manufacturer, the dump diffuser houses 8 to 18 combustor/transition piece assemblies equally spaced in the circumferential direction. Each of these assemblies has a surrounding jacket, which receives air from the dump diffuser and then feeds it to the combustor can through primary, secondary and dilution holes and through several hundred cooling holes. The air cools the surfaces of the transition piece and the combustor can before it enters the combustor can.

In selecting an air extraction scheme, one must consider how the scheme affects the air supply to hot sections of the turbine and the frictional loss in the flow region. Proper supply of cooling air is necessary to prevent burnout and to maintain effective air/fuel mixing in the combustor, hence the combustor exit pattern factor. The overall thermal efficiency of the power plant can be increased by minimizing pressure loss associated with air extraction. Therefore, investigating airflow in the compressor discharge casing with and without air extraction would enhance our understanding of the flow behavior and thus ensure integrity and high performance of the gas turbine.

Air extraction at manholes on the turbine wrapper might be the simplest design from manufacturing and retrofitting considerations. An approximated analysis indicated adverse effects of this scheme on the impingement cooling of the transition pieces. Agrawal and Yang (1991) suggested replacing the straight wall compressor/combustor prediffuser by a curved wall diffuser with a
suction slot (Griffith diffuser), and extracting air at the inlet of the modified diffuser. The present investigation experimentally reaffirms the analytically identified problems in the impingement cooling and substantiates the benefits of using the Griffith diffuser with air extraction at the prediffuser inlet.

TECHNICAL APPROACH

A scale test model of a heavy-frame gas turbine to simulate airflow between the compressor discharge and the turbine expander was fabricated and instrumented to investigate the aerodynamic performance and flow uniformity around combustor/transition pieces. A closely approximated 3-D computational fluid dynamic analysis of the flow in the diffusers was also performed to construct a comprehensive description of the flow field. This study was conducted with two prediffuser configurations; a straight wall prediffuser and a modified prediffuser also referred to as the Griffith diffuser. The air was extracted at the turbine wrapper when the straight wall prediffuser was used. When the Griffith diffuser was used, the air was extracted at the prediffuser inlet. Brief descriptions of the experimental facility, the experimental procedure and the computational procedure are given in the following sections.

Experimental Facility

A bird's eye view of the experimental facility is shown in Figure 1. This facility includes (1) a gas turbine test model and a system to induce airflow through it, (2) the collection manifold, and (3) a system to extract airflow from the test model.

Test Model and the Through Flow System. Figure 2 shows a cross-sectional view of the test model and it also depicts the airflow path. This nearly one-third scale model was 360-degree in circumference with 14 combustor cans and transition pieces. The test model does not simulate the combustion or the heat transfer processes. However, geometric details of the hot-section components were included to simulate the cold flow within the compressor discharge casing. The test model was constructed such that the prediffuser and other components could be replaced with similar yet different designs. The configuration in Figure 2 for extracting at the prediffuser inlet was based on the analysis by Agrawal and Yang (1991).

Figure 3 shows the flow system with the test model attached to it. The entire flow system was approximately 9.5 meter long while the test model was only 0.76 meter long. The centerline of the flow tunnel was 40 inches above the floor. The ambient air entering this suction type wind tunnel went through an inlet lip, filters, a 90-degree bend, a honeycomb and transition pieces which guided the airflow to an annular flow developing section. The exit of the flow-developing section then attached to the test section. The airflow exiting the test section discharged into a plenum box which isolated the test section from vibrations or oscillations of the suction fan located downstream of the plenum box. The suction fan was an industrial fan (IAP PHB31) operating at a constant RPM by a 150 kW, 3-phase electric motor. The flow rate through the test section was regulated by a set of louvers at the fan inlet. These louvers were operated by a computer controlled stepper motor.

Collection Manifold. The collection system consisted of two manifolds, a left-sided and a right-side; each with seven arms which attached to the collection ports on the test model. An inverted Y-section then combined the two manifolds and attached them to the air extraction fan. The manifold arms enter the mainstream at an angle of 30-degrees. Prior to fabricating the manifold, an approximate analysis was conducted. According to the results, the manifold arm should enter the mainstream at an angle of 30 degrees. The main guiding rule used to design the manifold follows that the mainstream flow area increase while maintaining a constant mean flow velocity. Butterfly control valves were provided on all branches of the manifold to equalize flow rate through the branches.
Figures 4 and 5 show the extraction manifold connected to the turbine wrapper and the prediffuser inlet, respectively. The two manifold systems were similar except that the manifold to extract air at the prediffuser inlet was larger in diameter. This manifold shown in Figure 5 was connected to ports around the prediffuser outer wall using conical tubes attached to the manifold arms. These conical tubes passed between adjacent combustor/transition piece assemblies.

The manifold was calibrated independently prior to installation to equalize the flow rates through all of its branches. First, the manifold system was attached to the air extraction flow system (discussed in the next section) to provide the desired extraction airflow rate. Then, a venturimeter was used to gauge the airflow rate through each of the manifold arms. Venturimeter readings from all the branches were equalized by adjusting the corresponding butterfly valves. The flow rate measured by the venturimeter would not be the actual flow rate through the branch because the venturimeter itself adds to the flow resistance. However, the resistance added by the venturimeter will be the same if the airflow rate through the venturimeter were the same. Matching the venturimeter readings of all branches signifies equal flow through the manifold arms. The flow rate through a branch is sensitive to flow resistances in the neighboring branches. Therefore, the butterfly valves were adjusted strategically to equalize the airflow while minimizing the overall flow resistance. Once adjustments were made, the venturimeter readings were repeated at all the branches. Equalizing flow through all branches of the manifold system typically required several iterations of this process.

**Air Extraction Flow System.** The downstream ends of the manifolds connect to the flow system, which extracts air from the test model. Figure 6 shows a schematic of the air extraction flow system connected to the collection manifold. The air was extracted by a Hoffman Model 76106A, two-stage centrifugal blower driven by a 150 kW, 3-phase constant speed motor. The fan intake draws air from a 0.20 meter diameter manifold duct and a 0.30 meter diameter bypass air duct. The manifold duct extracts low pressure air from the test model while the bypass duct draws air from the ambient. Each of these two ducts has a butterfly valve to regulate their respective airflow rates. These two ducts join into a 0.3 meter diameter duct connected to the blower inlet. The total airflow rate through the blower could also be regulated by a butterfly valve at the blower inlet. The air exiting this fan was discharged outside the laboratory area by a 0.25 meter diameter duct passage. By discharging air to the outside, the temperature in the laboratory area could be regulated or kept close to the ambient temperature. In the prototype, the extracted air is used to gasify the coal.

The extraction airflow rate could vary from no flow to the maximum required flow rate by adjusting the butterfly valve on the manifold duct. The bypass airflow rate decreased when more air was extracted through the manifold. When no air was extracted and the fan was operating, only the outside air was drawn into the blower. Thus, the bypass duct ensured that the airflow rate through the blower remained within design limits.

![Figure 3. Layout of the Through Flow System](image-url)
Figure 5. Extraction Manifold at the Prediffuser Inlet

Figure 6. A Schematic of the Extraction System for Air Removal from the Prediffuser Inlet
Instrumentation and Data Acquisition

Mass Flow. The airflow rate through the test model was monitored upstream of the prediffuser by a pitot-static probe. Only a single point measurement determined this airflow rate because the wind tunnel was calibrated prior to the tests using measured profile of axial velocity at the prediffuser inlet. The maximum flow rate through the test model was 3.8 m³/s (8,000 scfm), which corresponded to a dynamic head of 0.13 meter (5.2 inches) at the monitoring location. The airflow through the test section was regulated by computer-controlled louvers located at the fan inlet.

The extraction airflow was also monitored at a single point by a pitot-static probe mounted on the manifold duct. Prior to the tests, the profile of axial velocity profile was measured at the probe location which subsequently provided the relation between the dynamic head measured with the probe and the extraction airflow rate. At full load, the maximum extraction airflow rate was 0.76 m³/s (1600 cfm) which corresponded to a dynamic head of 0.03 meter (1.2 inches) at the monitoring location. The extraction airflow rate was regulated manually by a gear-operated butterfly valve on the manifold duct.

In a simplified air-blown IGCC, the extracted air returns to the gas turbine as a constituent of the low-Btu gasified coal. Because the room air was introduced into the fuel nozzles of the model to simulate the fuel flow, the net airflow rate into the combustors was maintained same as the extraction airflow rate. The airflow through all fuel nozzles of each combustor was metered and controlled by V-notch valves mounted at the inlet of each combustor. Prior to tests, a V-notch valve was calibrated independently to correlate the mass flow rate through the valve opening and the pressure drop across the valve. During experiments, the pressure drop across the valve was monitored for one of the combustors. Then, the correct valve opening corresponding to the desired airflow rate was arrived at using the calibration curve and a trial-and-error procedure. Once the correct valve opening was found for one of the combustors, all other valves were set to that opening.

An experiment was performed to check the accuracy of the mass balance in the test model and the air extraction system. The data showed that the net mass flow rates of air into and out of the test model were within 7% of each other. The mass flow rates of air into and out of the extraction fan were within 2% of each other.

Pressure Measurements. A computer controlled pressure measurement system, Scanivalve MSS-48C, was used to measure (1) wall pressures given by the wall pressure taps, (2) total pressure given by the Kiel and impact probes, and (3) total and static pressures given by the pitot-static probes. This system had two pressure sensors with maximum pressure inputs of +/- 74 bar (5psig) and +/- 37 bar (2.5psig), respectively. Each of the two sensors could accept a maximum of 48 pressure inputs. The pressure sensors were calibrated against a 5.0 meter feet tall U-tube manometer. The calibration curves indicated significant shift in zero for both sensors, but a negligible change in their sensitivities. To avoid measurement error because of the zero shift, one of the 48 pressure input lines to each sensor was always measured the ambient pressure, which was then subtracted from the other pressure readings. For each pressure measurement 20 readings were taken in 6 seconds.

Automated Velocity Measurements. A single-wire, hot-film probe was used to measure the two components of velocity at different locations in the flow. These measurements were made at longitudinal planes with circumferential symmetry where the circumferential velocity would be zero. This was the key condition behind the applicability of the single-wire, hot-film probe in measuring the other two components: radial and axial. The voltage outputs from the hot-film probe was measured for two different orientations at each measurement location. The hot-film probe was calibrated using a blowing type wind tunnel fitted with a 9:1 area ratio nozzle which provided a uniform (within 1%) velocity profile over the central 75% portion of the 0.15 meter diameter exit.

Surface Flow Visualization. Qualitative observations were made over the outer surface of the combustor transition piece to determine effects of air extraction on impingement cooling flow pattern. One of the 14 combustor transition pieces was coated with lampblack. The air entered through the holes in the impingement sleeve and formed jets which impinged over the outer surface of the transition piece. These air jets removed the lampblack coating which, in tum, provided traces of flow at the outer surface of the transition piece. These traces were lifted off the surface by a transparent tape.

Computational Procedure

The flow description was supplemented by a closely approximated 3-D computational fluid dynamic analysis. In this analysis, the airflow path was divided into two computational domains. These two domains shared an interface region which communicated and updated the boundary condition data between the two domains. The body-fitted, curvilinear coordinate system was used to incorporate the geometric details in the computational analysis. Internal solid objects, such as the support struts and the surface of the impingement sleeve, were simulated by blocking grid flow through grids representing these areas. Figure 7 shows the computational grid in the two domains on a longitudinal plane between the support struts. The shaded grids in Figure 7 were impervious to the flow. Thus, they represent internal solid objects such as the impingement sleeve and the transition piece. The computations were done with a total of 125,856 grids; 62,208 in the lower domain and 63,648 in the upper domain. Further details of the computational procedure and boundary conditions are given by Agrawal and Yang (1993).

RESULTS AND DISCUSSIONS

At the prediffuser inlet, the Reynolds number based on the prediffuser inlet height was 1.5x10^5. The corresponding Reynolds number in the prototype would be 2.5x10^6. The flow at the prediffuser inlet was characterized as fully turbulent in both prototype and the test model. Thus, the difference in the flow behavior between those Reynolds number would be second order in nature. However, the percentage of the total pressure loss derived from the model test is expected to be slightly higher than that in the prototype. In the following section, selected results are presented for 3 test cases: (1) straight wall prediffuser without extraction (baseline configuration),
(2) straight wall prediffuser with 20% air extraction at the turbine wrapper, and (3) modified prediffuser with 20% air extraction at the prediffuser inlet.

Aerodynamic Performance

Prediffuser Flow Field. Figures 8 and 9 show measured velocity profiles at the prediffuser inlet and exit, respectively. The air extraction at the turbine wrapper has only a marginal influence on the flow field in the prediffuser. The prediffuser flow field changes when the redesigned prediffuser or the Griffith diffuser is used. A finite radial field in the prediffuser. The prediffuser flow field changes when the air is extracted at the prediffuser inlet. Figures 8 and 9 show that the axial velocity near the outer wall is higher at the prediffuser exit than that at the prediffuser inlet. The air in the prediffuser’s outer region accelerates because of the sink-effect of the combustor bypass holes. Flow acceleration is detrimental to the prediffuser’s performance because the primary function of a diffuser is to diffuse or decelerate the flow.

Prediffuser Wall Static Pressure Recovery. Figures 10 and 11 show the static pressure recovery coefficients along the inner and outer walls of the prediffuser for all test cases. The pressure increases linearly along the inner wall of the prediffuser. However, the pressure at the outer wall decreases towards the exit of the prediffuser. The outer wall pressure decreases as the flow accelerates at the prediffuser exit because of the sink-effect of the combustor bypass holes. The air extraction at the turbine wrapper has only a marginal effect on the flow field in the prediffuser. With air extraction at the prediffuser inlet, the outer wall pressure recovery coefficient reaches a high value of 0.5. The overall static pressure recovery coefficients in the prediffuser are comparable for the two extraction schemes.

Total Pressure Loss. The mass-average total pressure loss coefficients were calculated using the total pressure profiles measured by the Kiel probe, the axial velocity profiles measured by the hot-film probe and the static pressure profile measured by the pitot-static probe. Table 1 summarizes the measured and computed pressure loss coefficients in the diffusers when the air is extracted from the test section. The total pressure loss coefficients in Table 1 are based on the mass flow rate passing through the dump diffuser. All of the compressor discharge air flows through the dump diffuser when the air is extracted at the turbine wrapper. Because the air extraction at the turbine wrapper has only a marginal effect on the flow field in the prediffuser. However, a slight decrease in pressure loss in the upper dump diffuser occurs because the mass flow rate of the air reaching the impingement sleeve or combustor casing is reduced. Only 80% of the airflow reaches the combustor while the remaining 20% is extracted at the turbine wrapper. Both measured and computed pressure loss coefficients are smaller when the air is extracted at the prediffuser inlet because only 80% of the compressed air enters the dump diffuser. A lower mass flow rate through the dump diffuser decreases the loss coefficient in the upper dump diffuser by 36 percent.
The total pressure loss coefficients thus obtained are related to the total pressure loss and the power loss in the prototype. Table 2 summarizes the results and shows that when the air is extracted at the prediffuser inlet, the power loss in the diffusers is nearly half of that when the air is extracted at the turbine wrapper. This difference in the power loss translates to a difference of 0.5 percentage point in the gas turbine thermal efficiency.

Flow Uniformity

Lamp Black Traces. A significant test result includes the lampblack traces over the outer surface of the transition piece. Figure 12 shows the lampblack traces over the surface of the transition piece facing the turbine wrapper. The top panel was recorded at the base line conditions. Without extraction the air jets impinged on the transition piece providing the necessary cooling airflow. The bottom panel was recorded when the air was extracted at the turbine wrapper. This panel indicates virtually no air impingement on the transition piece surface facing the turbine wrapper.

Table 2. Summary of Results

<table>
<thead>
<tr>
<th></th>
<th>Base Case</th>
<th>Turbine Wrapper Extraction</th>
<th>Prediffuser Inlet Extraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Pressure Loss Coefficient</td>
<td>1.2</td>
<td>1.0</td>
<td>0.7</td>
</tr>
<tr>
<td>Total Pressure Loss (%)</td>
<td>2.2</td>
<td>1.8</td>
<td>1.3</td>
</tr>
<tr>
<td>Power Loss, MW</td>
<td>3.2</td>
<td>2.7</td>
<td>1.5</td>
</tr>
<tr>
<td>Thermal Efficiency (%)</td>
<td>35.0*</td>
<td>35.23</td>
<td>35.75</td>
</tr>
<tr>
<td>Change in Thermal Efficiency (% point)</td>
<td>0.23</td>
<td>0.75</td>
<td></td>
</tr>
</tbody>
</table>

* based on GE MS7001F specifications

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Table 1. Mass-Averaged Total Pressure Loss Coefficient with Air Extraction

<table>
<thead>
<tr>
<th>Measurements</th>
<th>Turbine Wrapper</th>
<th>Prediffuser Inlet</th>
<th>Turbine Wrapper</th>
<th>Prediffuser Inlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pre-diffuser</td>
<td>0.22-0.11</td>
<td>0.01-0.10</td>
<td>0.04 (4%)</td>
<td>0.04 (6%)</td>
</tr>
<tr>
<td>Dump Diffuser</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lower part</td>
<td>0.08</td>
<td>not data</td>
<td>0.22 (22%)</td>
<td>0.06 (9%)</td>
</tr>
<tr>
<td>Upper part</td>
<td>&lt;0.95</td>
<td>&lt;0.8</td>
<td>0.72 (74%)</td>
<td>0.60 (85%)</td>
</tr>
<tr>
<td>Total</td>
<td>0.98 (100%)</td>
<td>0.70 (100%)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass Flow through the diffusers (% of base line)</td>
<td>100%</td>
<td>80%</td>
<td>100%</td>
<td>80%</td>
</tr>
</tbody>
</table>

No Air Extraction (Baseline Configuration)

With Air Extraction at the Turbine Wrapper
**Pressure Measurements.** Table 3 shows measurement locations and wall pressure measurements on the impingement sleeve and the combustor casing. Table 3 shows that the wall static pressure on the inside surface of the impingement sleeve is nearly the same in the circumferential direction. However, the wall static pressures on the outer surfaces of the combustor casing and the impingement sleeve vary in the circumferential direction. The circumferential variation is small on the combustor casing (near the bypass holes) but is significant around the impingement sleeve. The wall pressure is lowest at the 3 o’clock position and highest at the 6 o’clock position. The pressure is low at the 3 o’clock position because of the venturi effect of the crossflow between adjacent combustor/transition piece assemblies. The venturi effect is more pronounced at the impingement sleeve where the crossflow area or the space between sleeves is small. The nonuniform pressure distribution would cause air to enter nonuniformly through the bypass holes resulting in nonuniform cooling of the transition piece.

Table 3 also points to an increase in the circumferential nonuniformity of the wall pressure when the air is extracted at the turbine wrapper. Extracting air at the turbine wrapper increases the crossflow and hence, the venturi effect on the side panels (or 3 o’clock position) where the wall static pressure decreases. However, extracting air at the prediffuser inlet does not adversely affect the circumferential uniformity.

Table 3. Wall Pressure Measurements, meters (inches) of water

<table>
<thead>
<tr>
<th></th>
<th>Base Case No Extraction</th>
<th>Turbine Wrapper Extraction</th>
<th>Prediffuser Inlet Extraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>12 o’clock</td>
<td>-0.577 (-22.7)</td>
<td>-0.513 (-20.2)</td>
<td>-0.483 (-19.0)</td>
</tr>
<tr>
<td>6 o’clock</td>
<td>-0.574 (-22.6)</td>
<td>-0.513 (-20.2)</td>
<td>-0.480 (-18.9)</td>
</tr>
<tr>
<td>3 o’clock</td>
<td>-0.587 (-23.1)</td>
<td>-0.523 (-20.6)</td>
<td>-0.488 (-19.2)</td>
</tr>
</tbody>
</table>

**CONCLUDING REMARKS**

From an aerodynamic point of view, air extraction at the prediffuser inlet would be beneficial. Air extraction at the turbine wrapper increases flow non-uniformity around the impingement sleeve and results in a higher total pressure loss. In the test model, the air was extracted uniformly around the turbine wrapper. Thus, the number of extraction ports was the same as the number of the combustor/transition piece assemblies. Such an idealized scheme would not be structurally suitable in the prototype. A decrease in the number of extraction ports would disturb the circumferential periodicity and distort the flow field even more, thereby, requiring deflectors or guide vanes to adequately distribute flow around the impingement sleeve. These modifications to the flow region when the air is extracted at the turbine wrapper would incur additional pressure losses and reduce the overall thermal efficiency of the power plant.

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**REFERENCES**


