ABSTRACT
Reverse-flow combustors have been used in heavy, land-based gas turbines for many decades. A sheath is typically installed over the external walls of the combustor and transition piece to provide enhanced cooling through hundreds of small jet impingement cooling, followed by a strong forced convention channel flow. However, this cooling is at the expense of large pressure loss. With the modern advancement in metallurgy and thermal-barrier coating technologies, it may become possible to remove this sheath to recover the pressure loss without causing thermal damage to the combustor chamber and the transition piece walls. Without the sheath, the flow inside the dump diffuser may exert nonuniformly reduced cooling on the combustor chamber and transition piece walls. The objective of this paper is to investigate the difference in flow pattern, pressure drop, and heat transfer distribution in the dump diffuser and over the outer surface of the combustor with and without a sheath. Both experimental and computational studies are performed and presented in Part 1 and Part 2, respectively. The experiments are conducted under low pressure and temperature laboratory conditions to provide a database to validate the computation model, which is then used to simulate the thermal-flow field surrounding the combustor and transition piece under elevated gas turbine operating conditions.

The experimental results show that the pressure loss between the dump diffuser inlet and exit is 1.15% of the total inlet pressure for the non-sheathed case and 1.9% for the sheathed case. This gives a 0.75 percentage point (or 40%) reduction in pressure losses. When the sheath is removed in the laboratory, the maximum increase of surface temperature is about 35%, with an average increase of 13%-22% based on the temperature scale of 23 K, which is the temperature difference of bulk inlet and outlet temperature.

INTRODUCTION
The heavy-frame industrial gas turbines designed by several major manufacturers typically use a two-part diffuser-combustion section to decelerate air exiting the compressor. First, air from the compressor flows through an annular pre-diffuser and recovers some of its kinetic energy before being discharged into a comparatively large chamber called the dump diffuser or the dump chamber [1]. The diffuser-combustor section has two major functions (a) to decelerate the high velocity air coming from the compressor in order to provide an adequate air velocity for the combustion process and (b) to distribute air uniformly to the combustors to reduce the non-homogeneity of the combustion process and therefore reduce the level of the exhaust emissions [1, 2].

The essential feature of the combustor is to stabilize the flame in a high-velocity stream, where sustained combustion is difficult. The combustion process must be stable over a wide range of fuel flows required for ignition, start-up, and full power. It must perform within desirable ranges of emissions, exit temperature, and fuel properties and minimize the total pressure losses between the compressor and turbine. The combustion hardware must be mechanically simple, rugged, and small enough to be properly cooled by the available air. This hardware must have acceptable life and be accessible, maintainable, and repairable. The reverse-flow, multiple-combustion system is short, compact, lightweight and mounted within the flange-to-flange machine on the same turbine base as shown in Fig. 1.
temperatures up to 1560°C (2350°F). The outer portion of the effective cooling of the transition piece for firing by air flowing reversely from the compressor. This provides which channels the high-temperature gas from the louver system.

Louver cooling, which has been highly successful and reliable over the years, has been replaced by slot cooling in the turbines with the highest firing temperatures. The slot cooling method reduces liner metal temperatures by 139°C (250°F) from an equivalent channeling to the liner walls with beneficial effect on liner life.

The combustion liner is carefully cooled to tolerate high-temperature gases located a few millimeters from the combustor liner wall. As firing temperatures increase, more air is needed to mix with the fuel for adequate combustion, and thus, less air is available for liner wall cooling. This has been offset by a more efficient cooling system and by reducing the surface area (length) of the liner. Louver cooling, which has been highly successful and reliable over the years, has been replaced by slot cooling in the turbines with the highest firing temperatures. The slot cooling method reduces liner metal temperatures by 139°C (250°F) from an equivalent louver system.

In the reverse combustor design, the transition piece, which channels the high-temperature gas from the combustor liner to the first-stage turbine nozzle is s cooled by air flowing reversely from the compressor. This provides effective cooling of the transition piece for firing temperatures up to 1560°C (2350°F). The outer portion of the transition piece near the first-stage nozzle is less effectively cooled, and at firing temperatures above 1010°C (1850°F), jet-film cooling is added.

The inner transition piece is surrounded by a perforated sleeve with the same general shape as the transition piece. This perforated sleeve forms an impingement cooling shell causing jets of compressor discharge air to be directed onto the transition piece body. The air, after impinging on the transition piece body, then flows forward (actually reversely towards the compressor) in the space between the impingement sleeve and transition piece into the annulus between the flow sleeve and the combustion liner. It then joins additional air flowing through bypass holes in the flow sleeve providing the air for the combustion/cooling/dilution processes.

In addition to external cooling of combustor liner, different cooling schemes have been applied to cool the inside of combustor liner, such as the expanding swirl flow method [4, 5] and effusion cooling method [6]. Since this study is aimed at investigating external combustor liner cooling, the following literature review will only focus on external combustor cooling in reversed-flow type combustors.

Higher firing temperatures require combustors that release more energy in a given volume. High volumetric heat release rates, which depend on higher turbulent mixing in the combustor primary zone, are achieved by raising the combustor total pressure losses. The effects of these pressure losses are unique to each design. The advantages of using a reverse-flow combustor system are:

- Reduced length will allow single shaft sitting on two bearings for some large models. This will reduce the vibration and maintenance problems.
- The reverse-flow layout effectively uses the air flow to cool down the combustor liner and the transition piece as a counterflow heat exchanging process. The absorbed heat by the air is returned back to the system. In other words, the reverse-flow combustor cooling process is actually an effective preheating process for the air entering the combustor.
- The reverse-flow process also allows warmer air to serve as the dilution air to control the NOx formation instead of using other energy to preheat the dilution air or use cold air, which could quench the flame and increase CO production.
- Similarly, the preheat air also serves as wall jet film cooling on inside of the combustor liner. Penetration of air jet into the combustion chamber can enhance air-fuel mixing to achieve more effective combustion and help reduce NOx formation near the wall.

The disadvantages include increased material and manufacturing cost and pressure losses.

Relatively few experimental studies have been performed on dump diffusers with reverse-flow combustors. Kapat, et. al. [7] conducted a study of investigating the effect of air extraction on the flow field in the dump-diffuser reverse-flow combustor assembly in a Gas Turbine for Integrated Gasification Combined Cycle (IGCC). Later, Kapat, et al. [8] conducted a detailed three-dimensional flow characteristics and aerodynamic performance in the diffuser-combustor section. They discovered that most flow tended to move toward the outer part (away from the gas turbine center axis) of the dump diffuser. As a consequence, more flow entered the combustor from the outer portion (away from the gas turbine center axis) of each combustor. With this information of skewed flow distribution around the reverse-flow combustor, Wang, et al. [9] further investigate how air extraction employed in an IGCC system could affect the flow uniformity at inlet of the combustor. They discovered that often a reverse flow would occur at the combustor inlet if the air extraction location and extracted amount had not been optimized. To supplement the discoveries from the experimental work in [6], Zhou, Wang, and Ryan [10] conducted a computational simulation and provided a more detailed flow pattern, which helped to improve the understanding of the extremely complex 3-D flow interactions between the dump diffuser and the combustor inlet flow characteristics. The potential reversed flow behavior at the combustor flow inlet was supported by the computational study.

With the modern advancement in metallurgy and thermal-barrier coating technologies, it may become possible to remove this sheath to recover the pressure losses without causing thermal damage to the combustor chamber and the transition piece walls. However, without the sheath, the flow inside the dump diffuser may exert nonuniformly, reduced cooling on the combustion chamber and transition piece walls. The objective of this paper is to investigate the difference in flow pattern, pressure drop, and heat transfer distribution in
the dump diffuser and over the outer surface of the combustor with and without a sheath. Both experimental and computational studies are performed. The experiments are conducted under low pressure and temperature laboratory conditions to provide a database to verify the computation model, which is then used to simulate the thermal-flow field surrounding the combustor and transition piece under elevated gas turbine operating conditions. Part 1 presents the experimental study.

EXPERIMENTS

Wind Tunnel System

A wind tunnel and the accompanied data acquisition systems have been constructed and established in the Aerothermal Laboratory of Energy Conversion and Conservation Center at University of New Orleans. The wind tunnel is an open-circuit design, as shown in Fig. 2. The blower, characterized by airfoil blades, is manufactured by Cincinnati Fan. It is directly driven by a 30-hp motor, operating at 1756 rpm with 12,000 CFM air flow and 10 inches water column (static pressure). The speed of the wind tunnel is controlled by a Yaskawa constant-torque, variable frequency motor controller. The temperature of the wind tunnel flow can be cooled at a rate of 4.5°C/min (8°F/min) down to 10°C (50°F) and be heated at a rate of 0.77°C/min (1.4°F/min) up to 60°C (140°F). Two data acquisition systems are used, including a 96-channel thermocouple acquisition system, and a 288-channel pressure scanning system. Other auxiliary subsystems include heating/cooling system, boundary layer control system, probe traversing system, the pressure scanning and acquisition system, the hot wire system, and the thermocouple measurement system. The test section is designed for studying the thermal/flow behaviors of the reverse-flow combustor which simulates the combustor system of a typical heavy-duty gas turbine system.

Figure 2 Plane view of the wind tunnel

To filter out particles larger than 5 μm and to protect the hot-wire sensors to be used in the test section, a filter made of polyester felt is installed at the inlet of the fan. The flow passes through a diffuser from the fan to the flow straightener section, which includes a heat exchanger for controlling flow temperature, a screen pack, and a settling chamber. The densely packed fins of the heat exchanger serve as the first-stage flow straightener to break up large eddies (swirls) from the blower. A conventional honeycomb section is therefore not needed. The screen pack serves as the second stage flow straightener to break up the small eddies and make the flow more uniform.

The main function of the heat exchanger is to make the free-stream temperature uniform and provide a means to control the temperature in the test sections. To remove the low-momentum flow near the wall, a boundary layer suction box is installed between the contraction and test section. A six-inch flexible duct connects the suction box to a 3 hp high-pressure fan. The suction blower is powered by a 3 HP, 3-phase motor, and delivers 1127 CFM at 1 inch of water static pressure.

Test Section for Reversed-Flow Combustor Study

The reversed-flow combustor test section consists of a transition section that converts the rectangular exit of the contraction nozzle to a 1/7th sector of a circle to simulate a part of the dump diffuser which accommodates reverse-flow annular gas turbine combustors. The exit of the rectangular-to-circular transition section is a section of a circular arc, which subtends an angle of 52° to the center of the circle. This accommodates one full combustor and two dummy one-half of a combustor. There are two interchangeable test sections: one for flow measurements and the other one for heat transfer study as shown in Figs. 3 and 4.
The transition piece was outfit with a sheath to provide two types of enhanced cooling (Fig. 5). First, 470 holes are drilled on the sheath to provide air passages for creating hundreds of small jets that impinge on the transition piece and provide local enhanced cooling. Second, the narrow gap between the sheath and transitional piece serves as a narrow channel that speeds up the air flow and provides additional high convective cooling of the transitional piece and combustion chamber walls. The sheath surrounding the transition piece is seamlessly connected to the sleeve outside the combustion chamber wall. The outside diameter of the combustor is 45.72 cm (1ft-6 inches) and the total length from the combustor inlet to the transition piece outlet is 147 cm (4ft-10 inches). The outer radius of the dump diffuser from the center of the turbine is 142 cm (4ft-8 inches). The inner and outer radii at the transition piece exit are 94 cm and 73.67 cm, respectively.

For the fluid mechanic study test section, pitot-static tube is inserted into the rectangular-to-circular transition section to measure flow velocity by averaging at least five different locations in the same vertical cross section. The turbulence intensity is measured with a hot-wire anemometer. To monitor the pressure changes on the combustor chamber wall, 154 pressure taps are installed. Figure 6 shows the pressure tap distribution on the mid-chamber. Similar pressure taps are installed on each of the half chamber above and below the mid-chamber. All these pressure taps are grouped into 8 serials.

In the heat transfer test section, the combustor chamber is heated by forced convection. Both ends of the mid–chamber are closed and a heat gun (1.55 kW) is installed inside the combustor chamber and blows hot air that is recirculated in the closed combustor chamber as shown in Fig. 4 to simulate forced convective heat transfer from inside the combustor.

Thermal steady state is achieved after about 2 hours, indicating the outer cooling load and the heat supplies by the heat gun has reached equilibrium. The heated section is instrumented with 80 E-type (chromel-constantan) 36 gauge (0.01 inch) thermocouples, manufactured by Omega’s Fine Duplex Insulated Thermocouple Wire (TT-E-36-1000). Each thermocouple is first taped to the designated location by applying a piece of scotch tape. Black spray-on rubber is then carefully painted on the surface to fasten the thermocouples. The power density is approximately rated 1.5W/cm². This arrangement assures that the thermocouple junctions are well protected and fixed in place.

There are eight columns (parallel to the main stream direction) of thermocouples instrumented on the transitional piece and combustion chamber walls. Each column has 10 thermocouples. These thermocouples’ locations are shown in Fig. 7. These thermocouple columns are named according to their locations on the transitional piece and combustion chamber walls.
**Fig. 7** Layout of thermocouples on the mocked combustion chamber

**Instrumentation**

A Multiple Scanivalve System (MSS-48C) has been used to collect pressure data from test section. The system accommodates a total of 6 modules and accepting a total of 288-channel pressure inputs. Each includes a 48-channel pneumatic selector switch, a 10-inch WC pressure transducer and a signal conditioner.

Two Keithley model 2700-multimeter/data acquisition systems are used to measure the thermocouple voltage readings. The Model 2700 consists of a 6-1/2-digit digital multimeter and a high-performance data logger with 80 channels (or 40 differential channels) of multiplexed measurement data acquisition function. It reads as low as 0.1 μV and provides a digitizing capability with the equivalent of 22-bit A/D resolution. The highest scanning rate is 65 channels/second and the highest sampling speed is 2000/sec, which is more than sufficient for thermocouple measurements.

**Calibrations and Operation**

The ambient conditions (atmospheric pressure, humidity, and room temperature) are monitored during each experiment including the calibration tests. For a short experiment, the initial and final temperatures are recorded. For a long experiment, several intermediate readings of ambient conditions are recorded. It is important to monitor the change of ambient conditions to assess their unsteadiness for uncertainty analysis. The offset voltage represents the bias of a pressure transducer reading at zero pressure. For this experiment, digital voltage readings from the Scanivalve system are calibrated against the water columns measured at different wind speeds. After calibrating the modules, the combustor chamber pressure measurements were able to be performed. As always before starting any pressure measurements, the room conditions are recorded and zero offset of the pressure transducers are obtained by shortening the high and low ports of the transducer.

The thermocouples are calibrated with two points: distilled water freezing point (0°C) and boiling point (100°C). The results are consistent within 2 μV with the tabulate data provided by the manufacturer, so no intermediate points between 0 and 100°C are calibrated.

**Table 1** Input data and results for zeroth, pretest and first uncertainty analysis

<table>
<thead>
<tr>
<th>Independent variable</th>
<th>Instrument</th>
<th>Nominal value</th>
<th>Uncertainty of imprecision</th>
<th>Uncertainty of unsteadiness</th>
<th>Uncertainty of calibration</th>
<th>Zeroth order (%)</th>
<th>Pretest (%)</th>
<th>First order (%)</th>
<th>Nth order (%)</th>
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<tbody>
<tr>
<td>P, V/m-H2O</td>
<td>pressure transducer</td>
<td>2</td>
<td>0</td>
<td>0.05</td>
<td>0.02</td>
<td>0</td>
<td>1</td>
<td>2.693</td>
<td>3.978</td>
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<tr>
<td>Vt, V</td>
<td>signal conditioner</td>
<td>10</td>
<td>0.0002</td>
<td>0</td>
<td>0</td>
<td>0.002</td>
<td>0.002</td>
<td>0.002</td>
<td>0.002</td>
</tr>
<tr>
<td>Tamb, °C</td>
<td>thermometer</td>
<td>25</td>
<td>0.3</td>
<td>0.1</td>
<td>0</td>
<td>0.0075</td>
<td>0.0075</td>
<td>0.008</td>
<td>0.008</td>
</tr>
<tr>
<td>ΔH water, cm</td>
<td>manometer</td>
<td>20</td>
<td>0.03</td>
<td>0.1</td>
<td>0</td>
<td>0.15</td>
<td>0.15</td>
<td>0.522</td>
<td>0.522</td>
</tr>
<tr>
<td>RH, %</td>
<td>thermometer</td>
<td>60</td>
<td>6</td>
<td>3</td>
<td>0</td>
<td>0.0001</td>
<td>0.0001</td>
<td>0.0001</td>
<td>0.0001</td>
</tr>
<tr>
<td>Pamb, pascal</td>
<td>digital manometer</td>
<td>101325</td>
<td>100</td>
<td>50</td>
<td>0</td>
<td>0.0002</td>
<td>0.0002</td>
<td>0.0002</td>
<td>0.0002</td>
</tr>
<tr>
<td>total</td>
<td></td>
<td>101325</td>
<td>100</td>
<td>50</td>
<td>0</td>
<td></td>
<td>0.151</td>
<td>1.011</td>
<td>2.743</td>
</tr>
</tbody>
</table>

The temperature measurement system uses in-house designed isothermal box (Fig. 8), which is fabricated by 1-inch thickness aluminum blocks. The isothermal box provides a massive body to isolate the thermocouple-copper junctions inside the isothermal box from the ambient temperature fluctuation and therefore reduces the unsteadiness-introduce uncertainty of the experiment. The constant ice melting temperature of the ice bath is used to be a reference temperature. The readings from the data acquisition system indicate the relative temperature difference between the thermocouple measurement location and the isothermal box temperature. To determine the absolute temperature at the thermocouple junction, the temperature difference between the isothermal box and ice bath is added on the data acquisition readings. Therefore, the actual temperature of the isothermal box is not important, rather the uniformity and steadiness of the isothermal box temperature provide a system that renders a good quality temperature measurement with negligible sensitivity to the surrounding unsteadiness.
Uncertainty Analysis
An uncertainty analysis is performed to assist in identifying large uncertainty sources and planning for the experimental procedure. The method used in this study closely follows the method of Moffat [11] and Wang and Simon [12] and will not be detailed here. Table 1 lists the uncertainty analysis of pressure measurement in terms of zeroth, pretest, first order, and N-th order. The largest contribution of uncertainty for pressure measurement comes from the pressure transducer; the total uncertainty is about 4.5%. The uncertainty for temperature measurements is about 3.5%.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Cold-Flow Pressure Measurement Results
The measurements of each case are repeated 5 times in 30-minute intervals. The average of these 5 measurements is shown in the following figures. The results of sheathed and no-sheath cases are compared in Fig. 9 for the same speed.

Removing the sheath recovers most of the total pressure losses. The local static pressure difference does not represent the pressure recovery because the flow pattern and flow direction change. Some areas are subject to flow impingement with increased pressure when the sheath is removed, while some other areas may be located in a wake flow (leeward side), showing a drop of static pressure. The pressure recovery with the removal of the sheath can be approximately obtained by measuring the total pressure losses between the inlet and outlet of the test section or indirectly derived from the increased flow rate. For the no-sheath case, the total pressure difference between inlet and exit is 1200 Pascal (0.174 psi), which is 1.15% of the absolute total inlet pressure. For the sheathed case, the total pressure difference between inlet and exit is 2000 Pascal (0.29 psi), which is 1.9% of the absolute total inlet pressure. The average pressure difference from the local pressure measurements over the combustor and transition piece surface shows a similar value of pressure difference. This gives a 0.75 percentage point (or 40%) reduction in total pressure losses by removing the sheath.

Heat Transfer Results
Figure 10 reflects the comparison of temperature measurement results between no-sheath and sheathed cases. For the no-sheath case, the highest temperature on the mid-chamber is 315K (107°F); for the sheathed case, the highest temperature on the mid-chamber is 311K (100°F). The highest temperature difference between sheathed and no-sheath cases is 10K (18°F) and the temperature difference in most area is between 3 and 5 K (5.4 and 9°F), which reflects a maximum increase of 35% in temperature and an average increase of 13-22%, based on the temperature scale of 23 K, which is the temperature difference of bulk inlet and outlet temperature.

Fig. 9 Comparison of pressure distribution on the combustor and transition walls between sheathed and no-sheath cases
Fig. 10 Temperature comparisons between no-sheath and sheathed cases

CONCLUSIONS

In this study, the experimental facility and models of a gas turbine dump-diffuser and reverse-flow combustor assembly have been constructed. The objective is to investigate whether it is possible to remove the cooling sheath to recover pressure losses without subjecting the combustor chamber and transitional piece walls to detrimental high temperature heating.

The results show that under the laboratory condition with cold-flow experiments, for the no-sheath case, the total pressure difference between inlet and exit is 1200 Pascal (0.174 psi), which is 1.15% of the absolute total inlet pressure. For the sheathed case, the total pressure difference between inlet and exit is 2000 Pascal (0.29 psi), which is 1.9% of the absolute total inlet pressure. This gives a 0.75 percentage point (or 40%) reduction in total pressure losses by removing the sheath.

For heat transfer study, removing sheath results in a maximum increase of 35% in temperature and an average increase of 13% - 22%, based on the temperature scale of 23 K, which is the temperature difference of bulk inlet and outlet temperature.

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REFERENCE