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Intercooler Flow Path for Gas Turbines: CFD Design and Experiments

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Introduction

The Advanced Turbine Systems (ATS) program was created by the U.S. Department of Energy to develop ultra-high efficiency, environmentally superior, and cost competitive gas turbine systems for generating electricity. Intercooling or cooling of air between compressor stages is a feature under consideration in advanced cycles for the ATS. Intercooling entails cooling of air between the low pressure (LP) and high pressure (HP) compressor sections of the gas turbine. Lower air temperature entering the HP compressor decreases the air volume flow rate and hence, the compression work. Intercooling also lowers temperature at the HP discharge, thus allowing for more effective use of cooling air in the hot gas flow path.

The thermodynamic analyses of gas turbine cycles with modifications such as intercooling, recuperating, and reheating have shown that intercooling is important to achieving high efficiency gas turbines (Cook and Nourse, 1993; Wikes et al, 1993; Cohn, A., 1993). The gas turbine industry has considerable interest in adopting intercooling to advanced gas turbines of different capacities. This observation is reinforced by the US Navy's Intercooled-Recuperative (ICR) gas turbine development program to power the surface ships (Crisalli and Parker, 1993; Shepard et al., 1994; Valenti, M., 1995).

In an intercooler system, the air exiting the LP compressor must be decelerated to provide the necessary residence time in the heat exchanger. The cooler air must subsequently be accelerated towards the inlet of the HP compressor. The circumferential flow nonuniformities inevitably introduced by the heat exchanger, if not isolated, could lead to rotating stall in the compressors, and reduce the overall system performance and efficiency. Also, the pressure losses in the intercooler flow path adversely affect the system efficiency and hence, must be minimized. Thus, implementing intercooling requires fluid dynamically efficient flow path with minimum flow nonuniformities and consequent pressure losses.

Objectives

The objective of this research is to provide analytical tools and experimental data to help in the design of an effective intercooler flow path. Although the primary emphasis of the study is on

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the intercooler, the information obtained could be applicable to other similar flow systems such as a recuperator and/or an aftercooler.

Approach

This study involves three phases. In the first phase, computational fluid dynamics (CFD) analysis is employed to design the flow passages. The second phase involves fabrication and flow experiments in a scaled model of the intercooler flow path to characterize the flow field and frictional losses in different regions. The third phase involves validation of the analytical predictions, refinement of the analysis, and development of the design guidelines.

Project Description

Configuration of the intercooler flow path between the LP and HP compressors consisting of a diffuser, an intercooler or heat exchanger, and a contraction is shown in Figure 1. The diffuser decelerates the LP compressor discharge air while recovering the flow kinetic energy, isolates the LP compressor from the intercooler by turning air away from the turbine axis, and improves the heat transfer effectiveness by uniformly supplying low speed air to the intercooler. Downstream of the intercooler, the contraction isolates the HP compressor from the heat exchanger while minimizing the compressor inlet flow losses by gradually accelerating the cooler air.

The intercooler may be located on-axis (as shown in Figure 1) or off-axis. A water-cooled intercooler, because of its smaller size, could be on-axis while an air-cooled intercooler might be located off-axis. An off-axis intercooler would be easy to install and access; however it would require asymmetric interconnecting passages. The on-axis configuration shown in Figure 1, intended to establish the baseline information, uses a shell-and-tube heat exchanger with air flowing axially inside finned tubes. This arrangement is typical of the multi-stage centrifugal compressors and it differs from the Navy's ICR design (Shepard et al, 1994) where the space constraints led to a compact plate and fin heat exchanger with airflow normal to the turbine axis.

The current emphasis of this study is on flow nonuniformities and pressure losses in the intercooler flow path. Thus, the airflow is considered without heat transfer in the intercooler. The following is a description of the computational and experimental components of the project.

Computational

The computational part of the project is to first develop configuration(s) providing the desired flow characteristics in the intercooler flow path. Subsequently, the analysis could be used to interpret the experimental data and be refined to develop the design guidelines. Because of the complex geometry, the design of the intercooler flow path is not amenable to simple analytical procedures. Thus, the computational fluid dynamics (CFD) was used for detailed modeling and analysis.

The goal of the CFD analysis was to generate optimum diffuser and contraction wall geometries, within the specified constraints, using an iterative approach. In this procedure, the fully elliptic Navier-Stokes equations for turbulent flow are solved repeatedly on a body-fitted coordinate system. A set of parameters was developed to describe the geometry of the contoured passage. These parameters were varied systematically to yield a configuration with the desired flow characteristics. Following are some of the details of the design procedure.

Diffuser. The diffuser was required to direct airflow away from the turbine axis with minimum frictional loss, thereby, demanding a high area in a small length. An area ratio of 5.4 in a length equals 10 times the inlet annulus height was chosen. Because a straight wall diffuser with such an area ratio would be long, a contoured wall design was sought. The overall diffuser was divided into a moderately diffusing pre-diffuser and a sudden expansion. Such a design is typical of the compressor-combustor diffusers used in gas turbines (Fishenden and Stevens, 1977). The pre-diffuser recovers majority of the flow kinetic energy while the sudden expansion ensures uniformity of flow at the diffuser exit. The sudden expansion could also attenuate fluctuations caused by flow non-uniformities.

Walls of the pre-diffuser were generated from the prescribed passage axis (updated iteratively) and the passage expansion rate based on the annular Cp* diffusers (Adkins, 1983). A sinusoidal distribution was used as the initial guess for the passage axis. This guess distribution was updated iteratively using the computed flow field such that the flow decelerated uniformly at the inner and outer diffuser walls. A similar procedure was used for the sudden expansion, except that the expansion rate was also prescribed to avoid flow separation and to ensure uniform flow at the diffuser exit. Further details of the analysis are given by Agrawal et al. (1996).

Contraction. The overall area ratio of the contraction was the same as that for the diffuser. Walls of the contraction were determined from two curves; one defining the passage axis and the other prescribing distribution of the passage area. Each of these two curves was formed by two contours joining together smoothly at the inflection point. The shape of the curves was varied to yield uniform flow acceleration at the inner and outer walls of the contraction. This criterion ensured a uniform velocity profile at the contraction exit.

Experimental

The goal of the experimental work was to obtain quantitative data in a scale model of the intercooler flow path. The scale model would correctly simulate flow in the prototype if the velocity profiles at its inlet matched those at the LP compressor discharge. This matching was achieved by inducing controlled airflow through the test model by a suction type wind-tunnel. The following are the details of the test model, wind tunnel, instrumentation and data acquisition system.

Test Model. The test section consists of a diffuser, an intercooler, and a contraction. The contoured outer passages of the diffuser and contraction were designed to provide optical access, high degree of dimensional accuracy, facilities for introducing probes and pressure taps, and a reasonable cost and time for fabrication. Clear plexiglas was the material of choice. However, forming plexiglas to yield a complex contoured diffuser or contraction was not feasible. Thus, a process was devised where the plexiglas panels representing a sector of the diffuser or contraction were pressure formed. The manufacturing process consisted of the following steps: (i) machining a pair of laminated, hard mahogany wood block on a milling machine/rotary table facility to conform to the shape determined by the CFD analysis, (ii) heating and conditioning 19mm thick plexiglas sheets at 175 ^{0}C in a programmable autoclave, (iii) forming panels by sandwiching heated plexiglass between molds on a hydraulic press, (iv) assembling the segments to form the circular expansion, and (v) machining to the precise dimensions, and polishing to recover the optical quality. Figure 2 shows a photograph of a pair of molds and a formed plexiglass panel in the hydraulic press.

The contoured inner passages of the diffuser and contraction were designed to provide structural rigidity with a contrasting background. Hard Mahogany wood was the material of choice. The wood was laminated to form rectangular blocks of the desired dimensions and then machined to yield the exact profile on the exterior. Figure 3 shows a photograph of the machined inner diffuser passage.

Two co-axial rolled metal pipes were used to simulate the intercooler. This system could subsequently be replaced by a shell-and-tube heat exchanger with a square-pitch tube design to more accurately represent the test conditions.

Flow System. An open circuit wind-tunnel, shown schematically in Figure 4, provides the desired airflow through the test section. The air entering this low-turbulence wind-tunnel passes through a honeycomb, a set of screens, and a 9:1 area ratio contraction which guides the airflow to the annular flow conditioning sections.

The flow conditioning sections were designed with the following objectives: (i) ability to control the velocity profile at the inlet of the test-section, (ii) ability to characterize flow at the exit of the test-section, (iii) optical access with a clear exterior wall and contrasting interior wall, (iv) high degree of dimensional accuracy, and (v) facilities for introducing pitot-static, hot-wire anemometer and other probes. Further requirements were the interchangeability of components, resistance to wear damage due to the prolonged use, and the ability to align and traverse probes while maintaining an airtight seal.

The outer walls of the annulus were made of clear plexiglas and the interior walls were made of polished aluminum. The aluminum pipes forming the inner wall were sealed at both ends to prevent air flow inside of it. Two aircraft propeller spinners were mounted at the ends to seal as well as to streamline the flow past them. Inner tube was mounted to the wall of the wind tunnel using three airfoil shaped struts located at 120° intervals on the circumference.

Because the large plexiglas pipes of the dimensions required were not available in the market, they were formed from commercially available flat stock. The process consisted of heat rolling the plane acrylic to provide the cylindrical shape, machining to obtain the required dimensions, and polishing to yield the optical quality. Figure 5 shows a photograph of the flow conditioning sections assembled to the wind-tunnel.

The axial velocity profile at the exit of a gas turbine compressor could be nominal, OD-peaked or ID-peaked. Circular rings mounted on the inner or outer pipes of the upstream flow developing section were used to simulate the desired levels of flow nonuniformities at the inlet of the test-section.

Instrumentation. The experiments involve hot-wire anemometer, pitot-static probe, 5-hole probe, and wall pressure taps. A computer controlled scanning system is used to simultaneously scan 4 pressure input ports from a total of up to 96 ports. The hot-wire system provides voltage output which is calibrated within the velocity range using a calibration wind-tunnel. The voltage signals from the pressure sensor and the hot-wire sensor are converted to pressure and velocities using high and low-speed data acquisition boards (Mini-16 and Frash 12) and accompanying software (Quicklog and Workbench for Windows) from Straberry Tree. This icon-based data acquisition software operated in DOS and Windows environments on a Pentium microprocessor and it allowed complete control of the experiments. A system to traverse probes was also designed, fabricated and interfaced with the microcomputer. An underlying feature of the instrumentation was that the different operations such as probe traversing, scanning pressure channels, measuring pressure, temperature and velocities, and storing data on hard disk were integrated and automated within an interactive software environment. The data are stored in ASCII files on the computer hard disk. These files were accessed and processed using commercial software.

Results

Computational

The analysis was used to design annular diffuser and contraction passages. Figure 6 shows velocity vectors in the optimized diffuser with fully developed flow at its inlet. The diffuser provided uniform flow at its exit without flow separation. This diffuser had an annulus height of 0.052m (2.063") at its inlet, an area ratio of 5.4 and a length of 0.533m (21"). The exit diameters of the inner and outer walls were 0.51m (20") and 0.84m (33"), respectively.

The velocity vectors in the optimized contraction are shown in Figure 7, which indicates uniform flow at the exit. The contraction dimensions were the same as those for the diffuser except that the contraction length was 0.46m (18").

Experimental

A series of experiments involving hot-wire anemometer, pitot-static probes, and wall pressure taps were conducted to characterize the wind-tunnel and to adjust the velocity profile at the test-section inlet. These experiments indicated that the flow in the annuli was axisymmetric.

Figure 8 shows velocity profiles at 4 axial locations (72, 76, 80, and 84 inches from the entrance of the flow conditioning section) for a circumferential positions 'A' indicating fully developed flow at the exit of the upstream conditioning section. Figure 9 shows velocity profiles at an axial location 80" from the entrance of the flow conditioning section, when a circular ring was placed on either the inner or the outer pipe of the annulus. An inner ring resulted in an OD-peaked profile while an outer ring caused an ID-peaked profile. The desired velocity profile at the inlet of the test-section could be obtained by placing the ring at a proper axial location.

Future Activities

The future activities include detailed mapping of the flow field in the diffuser and contraction sections. Then, the experimental data in conjunction with the CFD design will be used to develop guidelines for the annular diffuser and contraction design.

Acknowledgements

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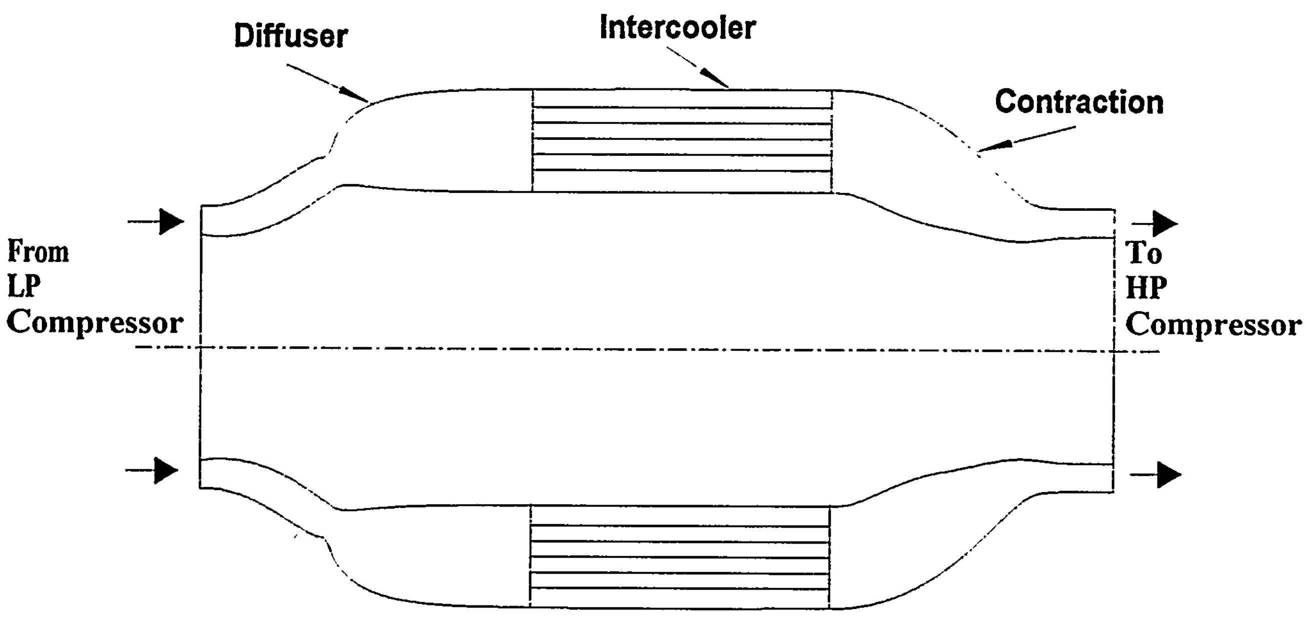

Figure 1. Schematic of the Intercooler Flow Path

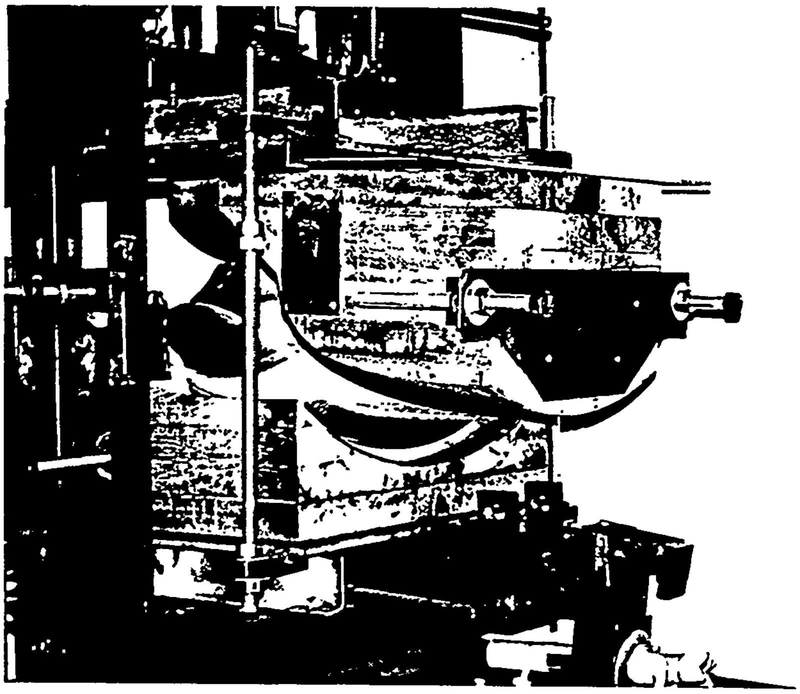

Figure 2. Molds, Plexiglass Panel and the Hydraulic Press

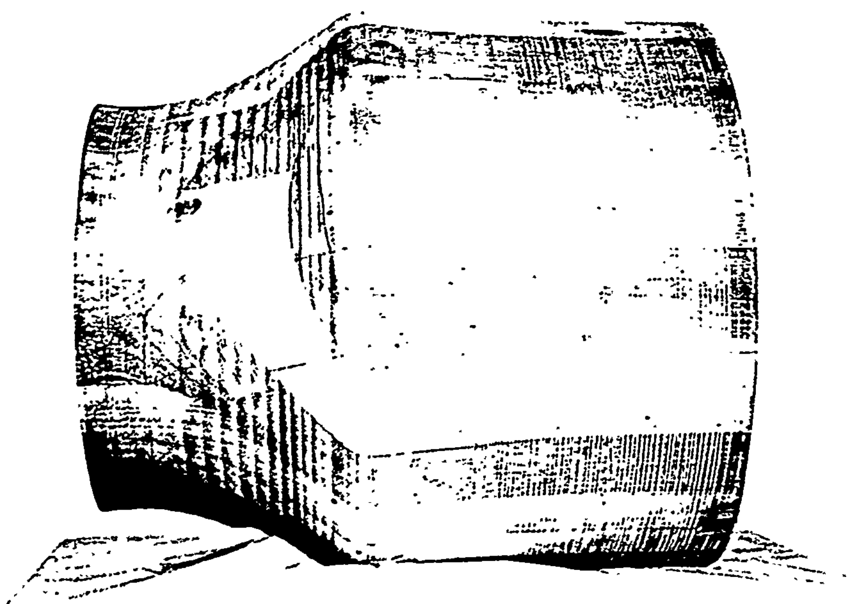

Figure 3. Machined Inner Diffuser Passage

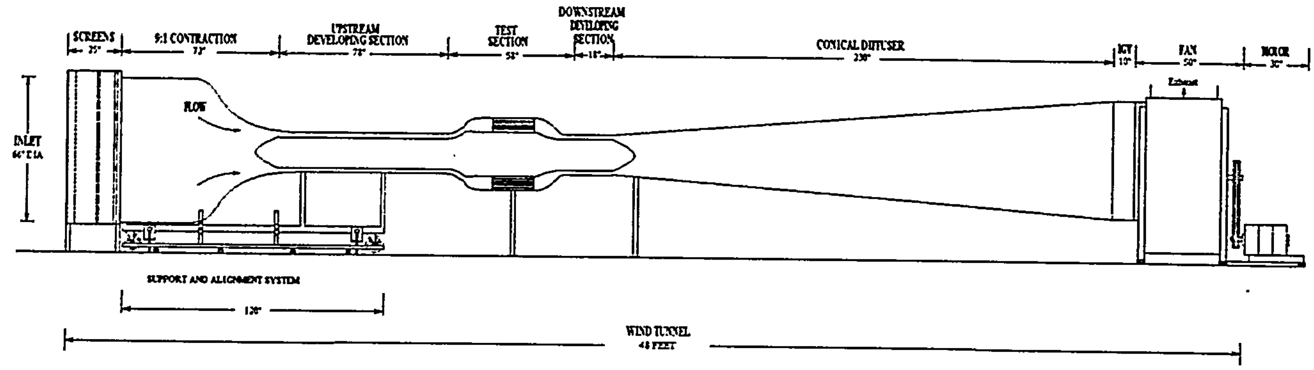

Figure 4. Schematic of the Wind-Tunnel

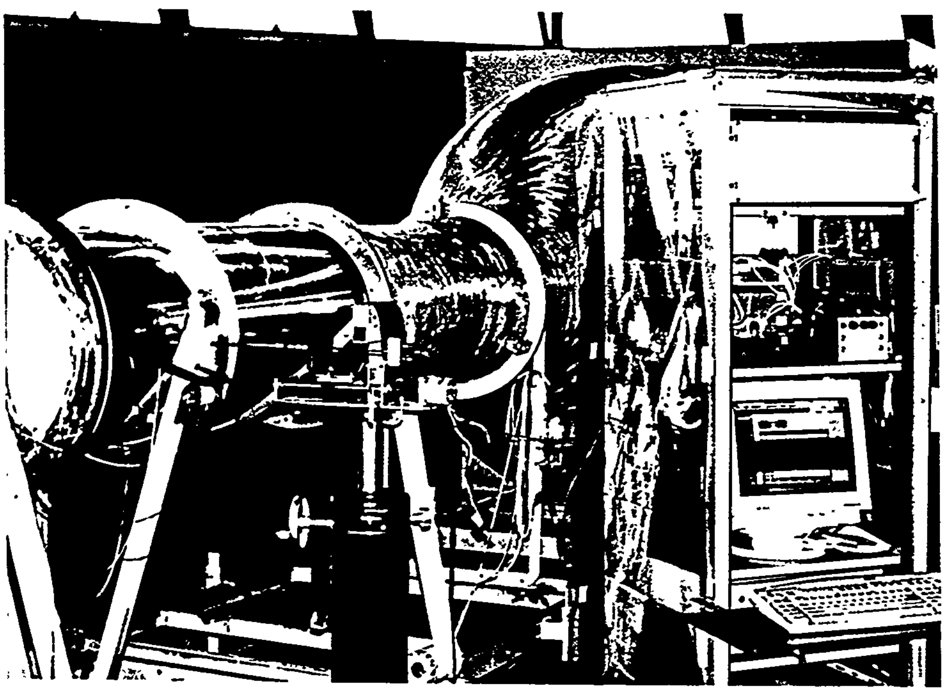

Figure 5. Wind-tunnel with the Flow Conditioning Sections

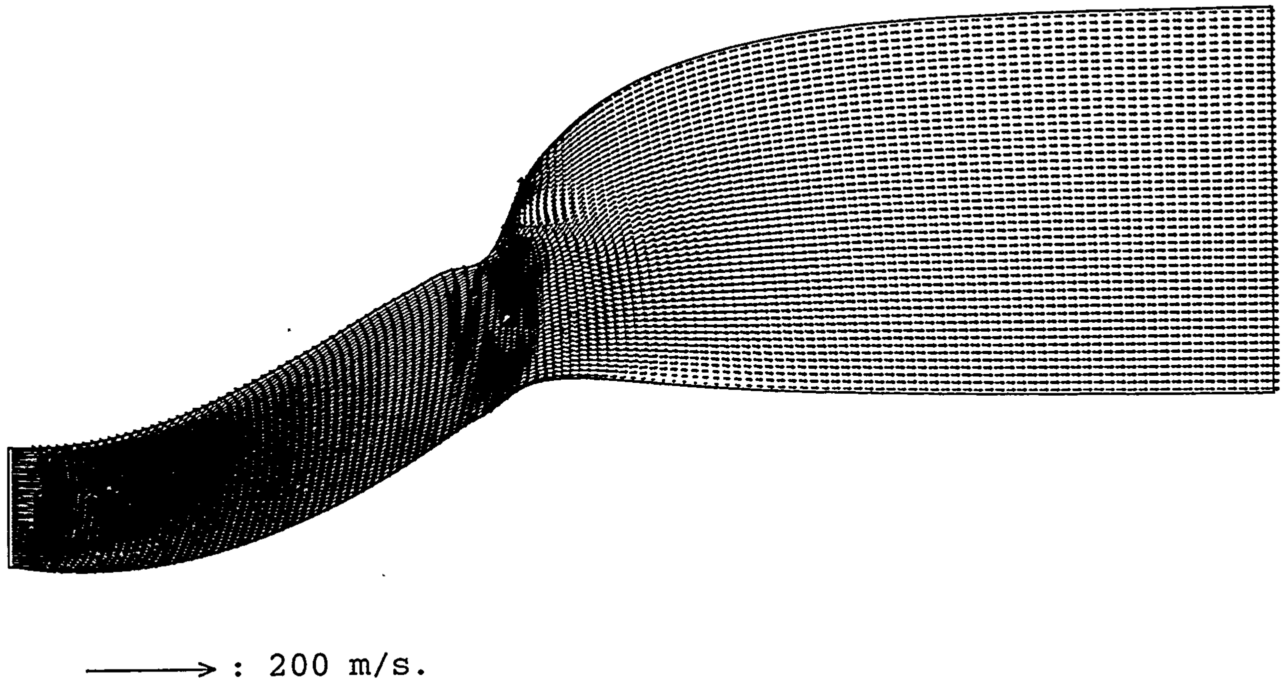

Figure 6. Velocity Vectors in the Optimized Diffuser

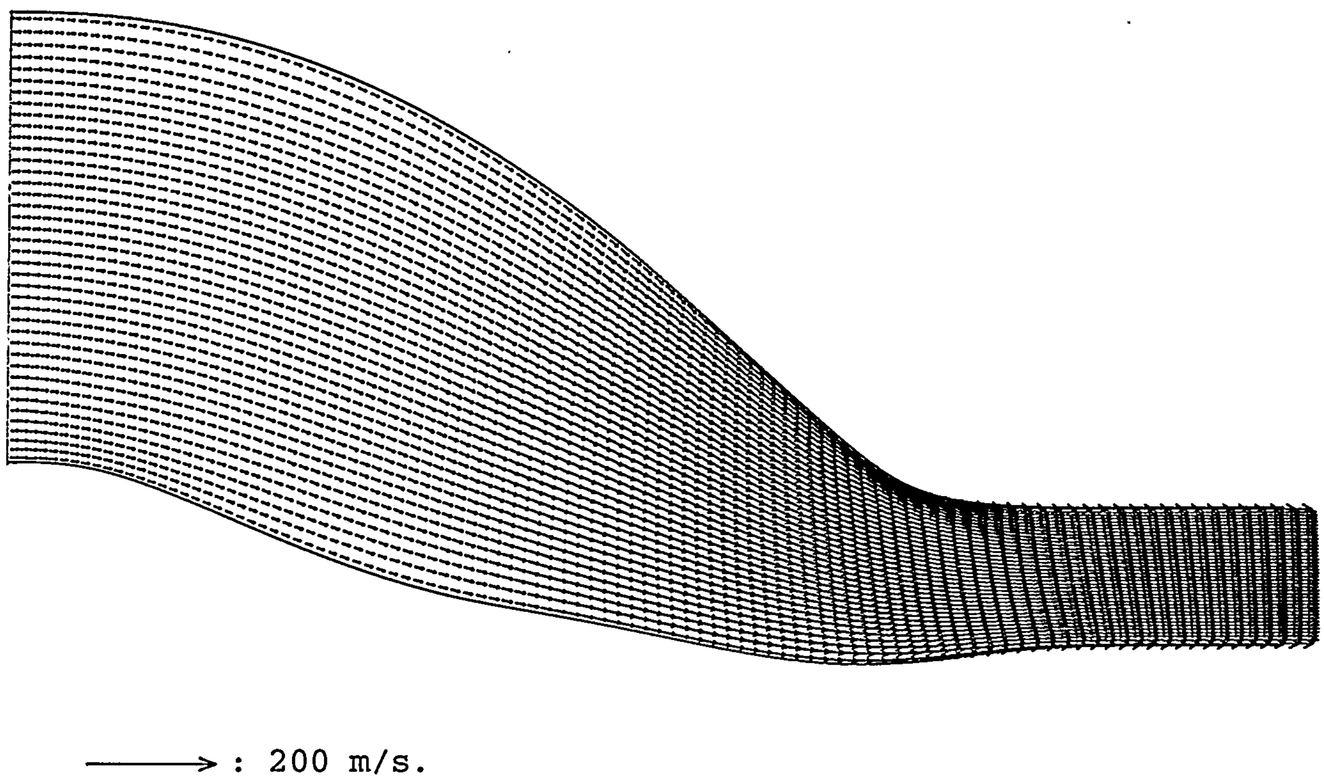

Figure 7. Velocity Vectors in the Optimized Contraction

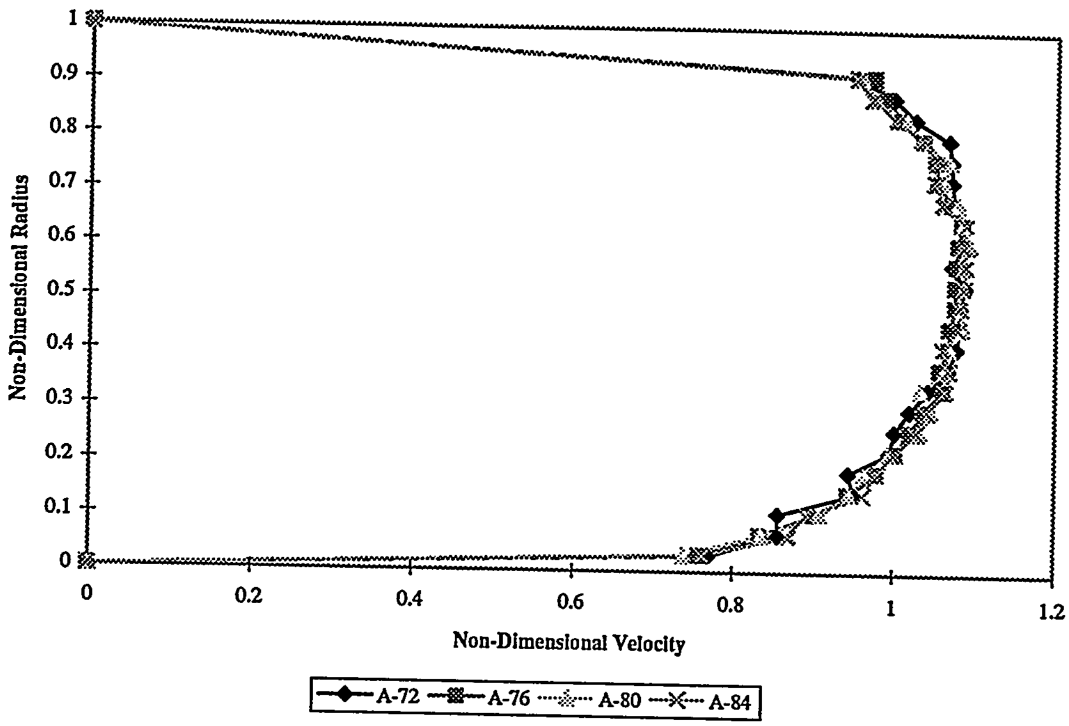

Figure 8. Axial Velocity Profiles in the Flow Developing Section (the Number after A indicates Distance, in inches, from the Entrance)

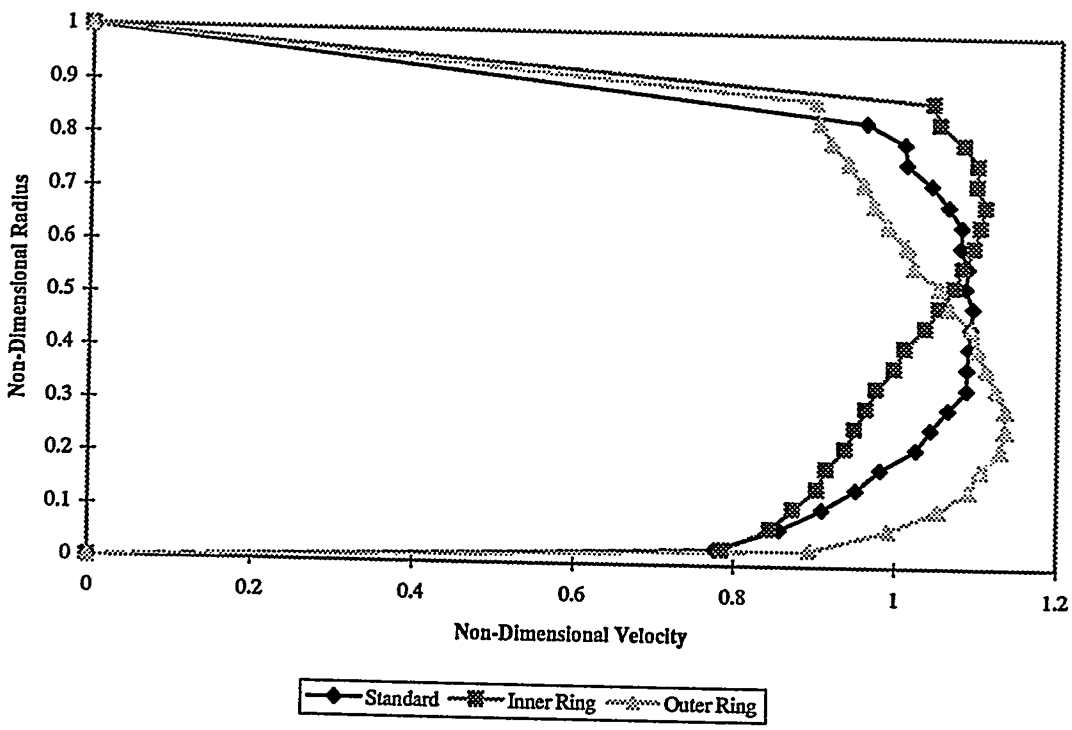

Figure 9. Axial Velocity Profiles in the Flow Developing Section at 80 inches from the Entrance (With and Without Circular Rings)