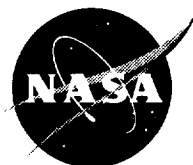


NASA/TM—1998-206971

AIAA-97-3140



A Modified Through-Flow Wave Rotor Cycle With Combustor Bypass Ducts

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Prepared for the
33rd Joint Propulsion Conference and Exhibit
sponsored by AIAA, ASME, SAE, and ASEE
Seattle, Washington, July 6-9, 1997

National Aeronautics and
Space Administration

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April 1998

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A MODIFIED THROUGH-FLOW WAVE ROTOR CYCLE WITH COMBUSTOR BYPASS DUCTS

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Abstract

A wave rotor cycle is described which avoids the inherent problem of combustor exhaust gas recirculation (EGR) found in four-port, through-flow wave rotor cycles currently under consideration for topping gas turbine engines. The recirculated hot gas is eliminated by the judicious placement of a bypass duct which transfers gas from one end of the rotor to the other. The resulting cycle, when analyzed numerically, yields an absolute mean rotor temperature 18% below the already impressive value of the conventional four-port cycle (approximately the turbine inlet temperature). The absolute temperature of the gas leading to the combustor is also reduced from the conventional four-port design by 17%. The overall design point pressure ratio of this new bypass cycle is approximately the same as the conventional four-port cycle. This paper will describe the EGR problem and the bypass cycle solution including relevant wave diagrams. Performance estimates of design and off-design operation of a specific wave rotor will be presented. The results were obtained using a one-dimensional numerical simulation and design code.

Introduction and Problem Definition

Pressure-gain wave rotors represent a promising technology for use as high pressure, high temperature topping cycles in gas turbine engines.¹⁻³ Among their potential advantages are rotor metal temperatures substantially below the combustor discharge temperature, rotational speeds which are approximately one third those of conventional turbomachinery, a wide operating range⁴, and relatively simple rotor geometry. Recent research efforts have focused largely on four-port, through-flow cycles with axially aligned passages of uniform cross section. This design, shown schematically in Fig. 1, is attractive in that, aside from the difficulty of partial to full annular transition ducts, it can be readily integrated into existing gas turbine engines as another spool⁵.

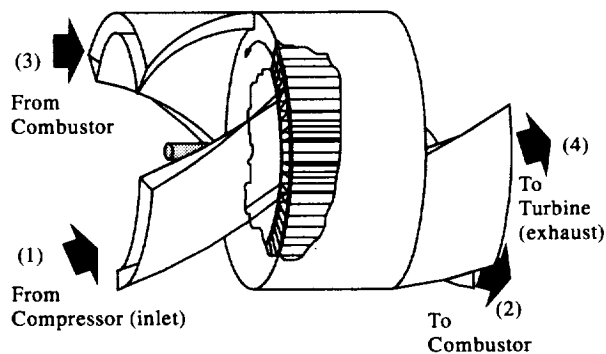


Figure 1 Generic four-port, through-flow wave rotor.

The term 'through-flow' refers to the general tendency for all of the flow entering the wave rotor to completely traverse the passage before exiting. In other words, the flow comes in one end and out the other. In contrast, reverse-flow cycles draw gas through one port and discharge it through another at the same end of the rotor. For example, with reference to Fig.1, air enters through port 1 but is sent to the combustor through port 3. Each gas stream entering the passages thus tends to remain closer to one end of the rotor.

In principle, through-flow designs take full advantage of the cooling capabilities of wave-rotors because each passage is alternately washed by hot then cold gas passing over its entire length. Estimates of the rotor wall temperature in a through-flow wave rotor designed for an overall stagnation temperature ratio (T_{04}/T_{01} in Fig. 1) of approximately 2.2 indicate that it is relatively uniform along the entire passage and that it is roughly equal to the downstream turbine inlet temperature. For most wave rotor designs this is 20-25% below the combustor exhaust temperature! In a representative small wave rotor topped gas turbine engine application, this amounts to a 660 °R difference between rotor wall and combustor exit temperature. This estimate was made using a one-dimensional CFD code developed specifically for wave rotor analysis.⁴

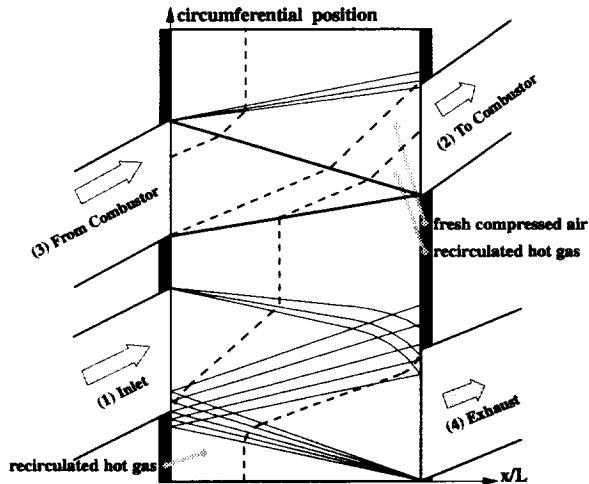


Figure 2 Wave diagram of the conventional four-port, through-flow wave rotor cycle at the design point.

While this is an impressive degree of cooling, in many engine topping applications the rotor temperatures would still be at levels requiring additional cooling if the rotor were composed of conventional materials. Furthermore, the current through-flow design contains a gas path in which the flow going to the combustor is a mixture of both fresh compressed air and recirculated hot gas from the combustor. This is illustrated in Fig. 2 which shows a wave diagram for the rotor passage as it rotates through a complete wave cycle (i.e., an $x-t$ diagram where t represents either time or circumferential position of a rotor passage). The solid lines in the figure represent the trajectories of selected wave fronts. The dashed lines represent selected particle paths. It is evident that not all of the hot gas which enters the rotor from the combustor is discharged from the port leading to the downstream turbine (port 4). As such, when a passage reaches the port leading to the combustor, the remaining hot gas is first discharged, followed by the compressed fresh air brought on board from the inlet duct (port 1). These hot and cold gases generally mix in the duct, and the resulting flow can be too hot to cool the combustor liner.

This paper will describe a new through-flow cycle which overcomes the exhaust gas recirculation problem through the use of a strategically placed bypass duct which transfers some of the working fluid from one end of the wave rotor to the other. The resulting cycle, after some repositioning of port locations and adjustment of the rotor speed to insure proper wave timing, replaces what was once hot combustor exhaust gas in the wave rotor passage with cool fresh air. This, in turn, yields a rotor with a lower mean wall temperature, equivalent performance, and with only cold fresh air going to the

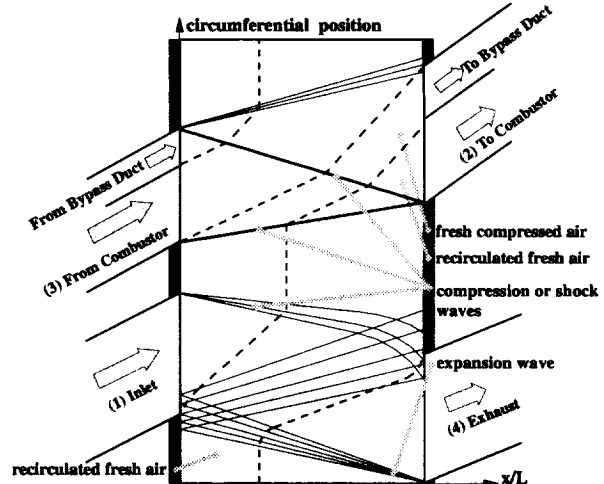


Figure 3 Wave diagram of the through-flow, bypass wave rotor cycle at the design point.

combustor. Details of the new cycle will be presented. Numerical performance estimates will be shown and compared to a conventional four-port, through-flow cycle design sized for the same mass flow rate and overall temperature ratio. Estimated wall temperatures and combustor inlet temperatures will be compared. Finally, potential technical challenges associated with this cycle will be discussed.

The results presented in this paper were obtained using a numerical wave rotor simulation which has been well documented in the literature.^{4,6} The paper relies heavily on the reader's familiarity with wave rotor operating principles, as none are provided within. Excellent descriptions may be found elsewhere in the literature,^{7,8} and a short list of description sources may be found in Ref. 9.

Bypass Cycle Description

The new cycle is illustrated in Fig. 3 which, in the manner of Fig. 2, is an $x-t$ diagram for a complete wave cycle. Fresh air enters the rotor through port 1 and is compressed by a series of shock and compression waves. Most of the compressed air then exits the rotor via the port leading to the bypass duct. A small amount of the compressed air leaves through the port leading to the combustor. Note that the amount of gas going to the bypass loop is adjustable in the design process. That is to say, cycles may be designed such that some EGR still exists, though not as much as is shown in Fig. 2. The particular cycle shown in Fig. 3 represents the minimum EGR, or alternately, the maximum useful cold bypass flow. The cycle design process is similar to that described in Ref. 3 and will not be presented here. Compressed air in the bypass duct is then routed to the other end of the rotor and re-injected via a duct located

Table 1 Port stagnation conditions at the design point of conventional four-port and bypass wave rotors sized for 4.8 lbm/s mass flow rate.

Bypass (Conventional)	P_0/P_{01}	T_0/T_{01}	\dot{m}/\dot{m}_1
Port 2	3.462 (3.257)	1.692 (2.038)	1.000 (1.540)
Port 3	3.078 (2.972)	2.900 (2.821)	1.000 (1.540)
Port 4	1.221 (1.203)	2.209 (2.208)	1.000 (1.000)
Port To Bypass	3.418	1.504	0.925
Port From Bypass	3.177	1.504	0.925

just aft of the duct coming from the combustor. This air is then expanded, along with the hot gas which came from the combustor, by the strong expansion wave generated in port 4. The expansion process purges the hot gas from the wave rotor passage, but leaves the bypass air on board. With the subsequent compression wave process, the bypass gas is sent to the combustor through port 2 along with a small amount of fresh compressed air. This air is then heated in the combustor and returned to the rotor through port 3.

Comparing Figs. 2 and 3, several salient features may be observed. First, the portion of the passage which, in the conventional four-port cycle, is filled with hot recirculated gas from the combustor is now occupied by relatively cool fresh air from the bypass loop. This is the primary mechanism leading to improved rotor cooling. Second, all of the flow going to the combustor is relatively cool fresh air. Table 1 lists, for a particular 4.8 lbm/s wave rotor described in the following section, the mixed (i.e., averaged over the width of the port) stagnation pressure and temperature, and the mass flow rate in each port, relative to the inlet state, for the computed bypass and conventional four-port cycles. The gas in port 2 of the bypass cycle is 20% cooler than that in the conventional four-port cycle.

It may appear from Fig. 3 that the gas path allows for clear distinction between the wave ‘compressor’ and wave ‘turbine’ components of the cycle. As such, the information from Table 1 could be used to calculate component efficiencies (adiabatic or polytropic). Such calculations would not be meaningful however, for several reasons. First, there is significant heat transfer from the rotor walls to the relatively cool fresh air, and from the hot gas to the relatively cool rotor wall. The heat transfer leads to apparent reductions in compression efficiency and increases in expansion efficiency. Second, although the interfaces between hot

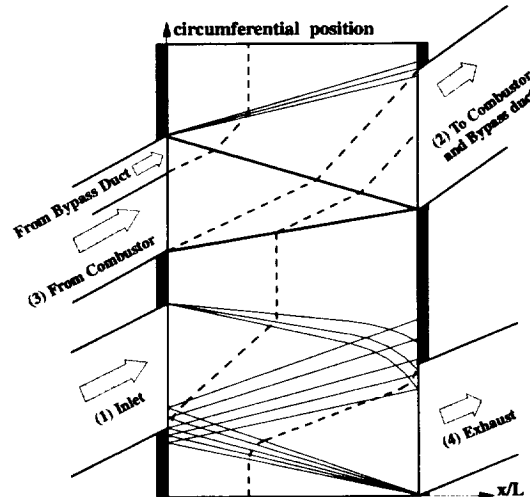


Figure 4 Wave diagram of the simplified bypass wave rotor cycle at the design point.

and cold gases appear sharp in Figs. 2 and 3, in reality they are not. The behavior of hot/cold interfaces in a wave rotor is a complex, multi-dimensional phenomenon.^{10,11} Even from a one-dimensional perspective however, it is clear that a wave rotor passage has a finite width which gives rise to a spreading or smearing of interfaces. The spreading may be compounded if the interface passes through an expansion wave. The general result of interface spreading is that some hot gas can enter ports intended for cold flow, and vice-versa. Since the interface spreading due to finite passage widths is modeled in the CFD simulation used for this paper, the effects are seen in Table 1[†]. Like the heat transfer phenomena, calculated compression efficiencies appear reduced while expansion efficiencies appear increased.

Simplified Bypass Alternative

The gas conditions in port 2 and in the port leading to the bypass duct are relatively close to one another. It may therefore be beneficial to simplify the bypass cycle by eliminating the duct wall that separates them and allow the two streams to mix. This configuration is illustrated in Fig. 4. The two streams coming from port 2 are mixed to an average state (within the simulation this is done using a constant area mixing calculation⁹) and then split somewhere downstream, prior to the combustor. As would be expected, performance of this

[†] Modifications to the boundary conditions of the one-dimensional, CFD simulation used for this investigation have been made which allow computation when the passage is simultaneously exposed to more than one port. This occurs for example as the passage moves from port 3 to the port leading from the bypass duct in Fig. 3. These modifications have not been published as of this writing.

Table 2 Port stagnation conditions at the design point of original and simplified bypass wave rotor cycles sized for 4.8 lbm/s mass flow rate.

Simplified Bypass (Original Bypass)	P_0/P_{01}	T_0/T_{01}	\dot{m}/\dot{m}_1
Port 2	3.433 (3.462)	1.669 (1.692)	1.840 (1.000)
Port 3	3.057 (3.078)	2.876 (2.900)	1.000 (1.000)
Port 4	1.231 (1.221)	2.201 (2.209)	1.000 (1.000)
Port To Bypass	(3.418)	(1.504)	(0.925)
Port From Bypass	3.144 (3.177)	1.669 (1.504)	0.840 (0.925)

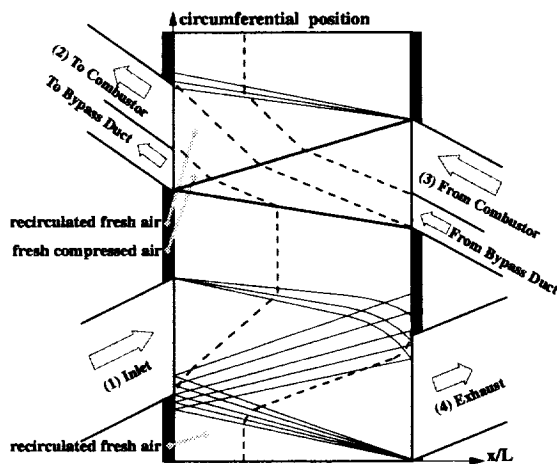


Figure 5 Wave diagram of the reverse-flow, bypass wave rotor cycle at the design point.

cycle is nearly identical to the original bypass configuration illustrated in Fig. 3. For reference, the gas states and port mass flow rates at the design point are listed in Table 2, in a manner similar to Table 1, for the original and simplified bypass cycle.

Reverse-Flow Alternative

Although the focus of this paper is primarily on through-flow cycles, it is noted in passing that the bypass modification can be implemented in a reverse-flow cycle as well. This is shown in Fig. 5 using the same technique as Figs. 2, 3 and 4. The rotor dimensions, inlet mass flow rate, and value of overall stagnation temperature ratio, T_{04}/T_{01} used to design this cycle were the same as those used for Figs. 2, 3, and 4. The overall stagnation pressure ratio, P_{04}/P_{01} is the same as the through-flow cycle shown in Fig. 3. Unlike the through-flow cycle, a conventional four-port, reverse-flow cycle, exhibits no EGR phenomenon. Instead, a mass of hot gas is perpetually trapped within each rotor

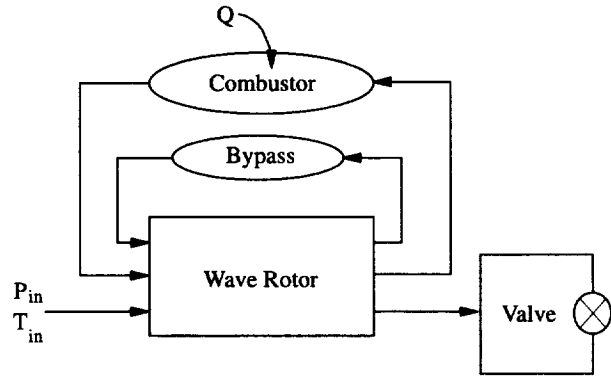


Figure 6 Simulation schematic.

passage (the same mass that in a conventional through-flow cycle gives rise to the EGR) and never leaves.¹ The bypass cycle eliminates this problem, allowing all of the gas which enters the wave rotor to eventually leave.

Overall Performance

In order to realistically assess wave rotor performance, specific flow requirements must be selected. Those selected for this paper were a mass flow rate of 4.8 lbm/s, a ratio of exhaust to inlet temperature (T_{04}/T_{01} in Fig. 1) of 2.2, and inlet conditions corresponding to an upstream compressor pressure ratio of 7.8. Both a four-port, through-flow and a bypass cycle were designed using the same rotor geometry, and requirements in order to allow direct performance comparison. In fact, the rotor geometry (e.g., length, mean radius, number of passages, etc.) was taken from a previously published four-port, through-flow design.^{4,5} As such, no optimization of the bypass cycle was performed in the manner of Ref. 3; only the port positions and rotor speed were varied to obtain correct wave timing. Both the four-port and bypass cycles were designed to be freewheeling, meaning that there is no independent drive motor. Torque is assumed to be generated by changes in angular momentum as the flow in the inlet ducts is turned from the duct angles to follow the walls of the rotor. Windage effects are neglected in the modeling; however, they are expected to be low for the rotor speeds and geometries of most pressure-gain wave rotor designs. Friction arising from bearings is also neglected. If the sum of the torque generated by the three inlet ducts is zero, the rotor speed is constant.

A schematic of the simulation components used to predict the wave rotor performance is shown in Fig. 6. The components have been documented in Refs. 4 and 6. The inlet gas state, exhaust valve area, and heat addition rate (i.e., fuel flow rate) were variables. For the performance predictions, the inlet gas state was kept constant at the design point value.

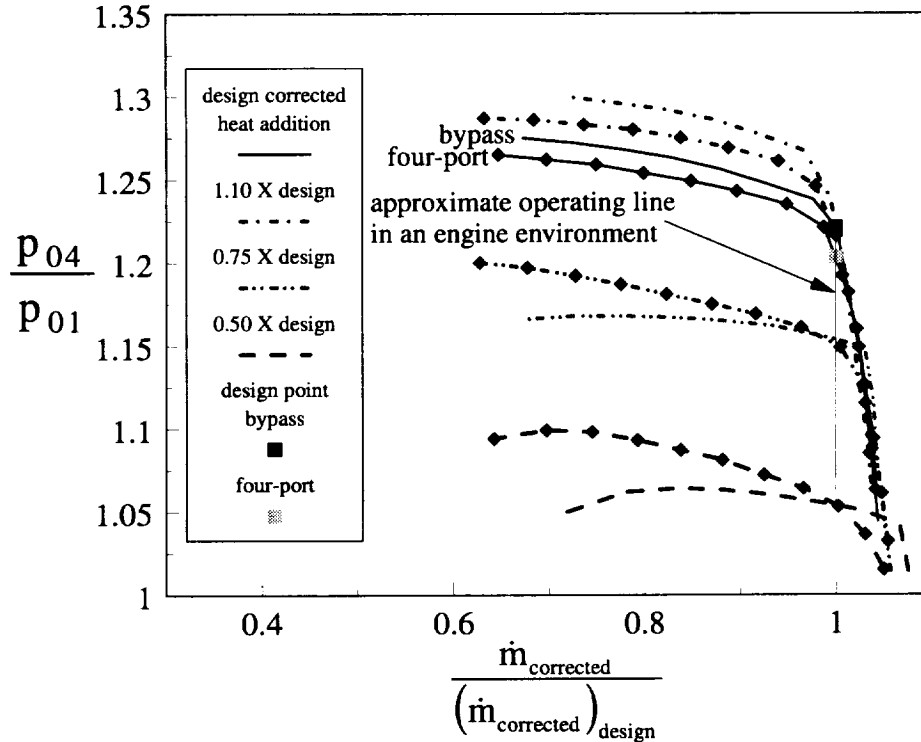


Figure 7 Performance map for bypass and four-port wave rotors sized for 4.8 lbm/s design point mass flow rate.

The original bypass and four-port performance maps are shown in Fig. 7. Each curve on the map was generated using a fixed, specified heat addition rate, while the exhaust valve area was varied. The curves without symbols represent the bypass cycles. The curves with symbols represent the four-port cycle. The design operating points for the two cycles are also shown with a dark square symbol representing the bypass cycle and a light gray square representing the four-port cycle. For each point on the map the exhaust valve area was set and the simulation was run until the sum of the mass flows from all of the wave rotor ports was zero, the time rate of change for all of the plena (i.e. combustor, bypass, rotor leakage cavity) were zero, and the net torque was zero. The horizontal axis of Fig. 7 is the non-dimensional corrected flow rate which is defined as

$$\dot{m}_c = \frac{\dot{m}}{p_{01} A_1} \sqrt{\frac{R T_{01}}{g_c}} \quad (1)$$

where \dot{m} is the mass flow rate, p_{01} and T_{01} are the inlet stagnation pressure and temperature respectively, A_1 is the inlet cross sectional area at the rotor face, R is the

gas constant for air, and g_c is the Newton constant. The corrected heat addition rate which is constant along each curve is defined as

$$\dot{Q}_c = \frac{\dot{Q}}{p_{01} A_1 \sqrt{R T_{01} g_c}} \quad (2)$$

where \dot{Q} is the heat addition rate. The vertical axis is the ratio of exhaust to inlet stagnation pressure of the wave rotor (see Fig. 1). Since wave rotor performance is sometimes stated in terms of p_{04}/p_{01} versus T_{04}/T_{01} the following relation may be used

$$\frac{T_{04}}{T_{01}} = 1.0 + \frac{(\gamma - 1) \dot{Q}_c}{\gamma \dot{m}_c} \quad (3)$$

Figure 7 also shows an approximate steady-state operating line exhibited by the rotor when it is in the topping cycle environment^{1,4}. Typically, the corrected heat addition rate, \dot{Q}_c along this line ranges from 1.02 to 0.85 of the design value representing 100% to 43% engine power ratings. In Fig. 7, it has been extended down to the value $\dot{Q}_c = 0.50$ of design in order to give some indication of performance near idle.⁵

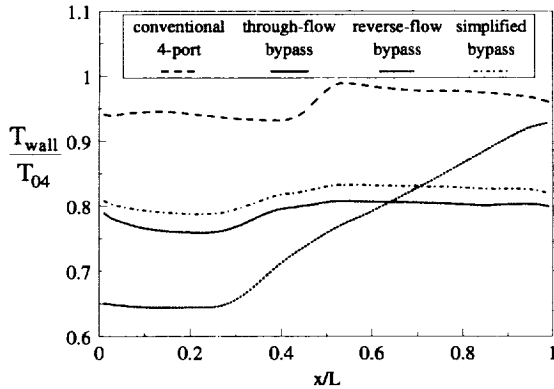


Figure 8 Design point wall temperature distributions for a conventional four-port and several bypass wave rotor cycles sized for 4.8 lbm/s mass flow rate.

It can be seen from Fig. 7 that the computed performance of the bypass cycle is somewhat better than the four-port design along the entire operating line. The improvement is most likely because the bypass cycle is, fortuitously, a more optimal design. Reference 3 suggests an optimal rotor speed which is below what was used in the present investigation, for a four-port, through-flow wave rotor with nearly the same flow requirements. The optimal speed was not used because it would have resulted in substantial period of time during which both ends of the rotor passages were adjacent to endwalls. This could lead to unnecessarily high leakage losses. Thus, the design four-port, through-flow rotor speed was increased in order to eliminate the extra endwall space. In order to ensure correct wave timing however, the bypass cycle required a speed reduction back toward the original optimal configuration. The two cycles are expected to perform equivalently when each is optimized, as long as leakage losses can be minimized.

The performance of the bypass cycle appears to drop off more rapidly than the four-port cycle at the lower heat addition rates when the corrected mass flow rate is reduced. Since numerically simulations have indicated that operation of the wave rotor in regions where the slopes of the curves in Fig. 7 are positive may lead to instabilities⁴, this observation may mean that the bypass cycle is more susceptible to unstable operation near idle than the four-port cycle; however, this is by no means a definitive result.

The computed design point distribution of wall temperature along a passage for the through-flow, four-port, the through-flow, bypass, the simplified bypass, and the reverse-flow, bypass cycles are shown in Fig. 8.

Table 3 Mean design point rotor wall temperature estimates, for a conventional four-port and several bypass wave rotor cycles sized for 4.8 lbm/s mass flow rate.

	Through-flow Four-port	Reverse-flow Bypass	Simplified Bypass
$\frac{T_{wall}}{T_{04}}$	0.96	0.79	0.82

The wall temperatures have been scaled by the exhaust port stagnation temperature, T_{04} . It is noted that the wall temperature estimates are based on the assumption of radial heat transfer and conduction only, on the top and bottom walls of the passage. No account is made for heat conduction along the passage (in the axial direction). It is seen in the figure that the all of the bypass cycles are significantly cooler than the four-port cycle and far below the peak cycle temperature listed for port 3 in Table 1. The average values of the distributions in Fig. 8, are listed in Table 3. Note that although the reverse-flow, bypass cycle yields the lowest mean wall temperature, it is clear from Fig. 8 that there is significant variation from end to end. In contrast, both the original through-flow, bypass and simplified, bypass cycles have relatively uniform temperature distributions.

Discussion

The results presented above indicate that the bypass cycle, when compared to the four-port, through-flow cycle, provides equal or improved aerodynamic performance throughout the normal operating range, significantly reduced rotor temperatures, and a combustor inlet temperature. These benefits are substantial and in many applications may make the difference between a wave rotor that is truly self-cooling, with metal temperatures well within the material limits of conventional construction materials, and one which is merely cooler than the combustor exhaust gas. Nevertheless, it is worthwhile to consider some possible challenges which arise from the bypass design, keeping in mind that such considerations are preliminary. These are outlined below.

Ducting

A bypass rotor may require more complex ducting compared to the four-port design. It would appear, since the pressures and velocities in the combustor and bypass loop ports are nearly identical, that the two duct sets could be directly adjacent to one another; separated only by a single wall. By definition however, the bypass flow does not pass through the combustor and must have a separate path around it. This, may somewhat

complicate the integration of the wave rotor within the gas turbine engine. It should be noted however that, regardless of the potential increased complexity, the combined volumetric flow rate of bypass cycle combustor and bypass loop flows is nearly identical to the four-port cycle combustor loop flow, meaning that the size of the ducting in the two cycles is comparable.

Combustor Pressure Drop

A difficulty of the bypass cycle, which has also been noted on the reverse flow four-port cycles, is the large combustor pressure drop required to properly balance the wave cycle. Examination of Table 1 shows that for the through-flow, four-port cycle, the combustor pressure drop, $\Delta p_0/p_{02}$ is 8.8%, while on the bypass cycle it is 11.1%. It may be argued that the complex ducting of the wave rotor will incur a larger loss than the usual 3-5% seen on conventional turbomachinery combustor sections, but it is difficult to envision values as large as those required by the bypass cycle.

Interface Distortion

A third potential issue with the bypass cycle involves the structure of the hot/cold interface. That is the regions of the gas path when hot flow is roughly adjacent to cold. These occur in both the four-port and bypass cycles and can be seen in Fig. 2 between the compressed flow from the inlet port and that coming from the combustor, and between the combustor EGR gas and that coming on board the rotor from the inlet. In Fig.3 they are seen between the compressed flow from the inlet and that coming from the combustor, and between the flow from the combustor and the flow from the bypass duct. It has been observed computationally¹⁰ that this interface becomes highly distorted as it traverses the passage; so much so that mass averaged port pressures and temperatures are different from values computed using one dimensional models. Whether the interface distortions are worse in the bypass than in the four-port cycle is an issue which requires investigation both computationally and experimentally.

Bleed Flow for Downstream Turbine Cooling

The question of what is the best site on the wave rotor from which to extract cooling flow for downstream turbomachinery is still unanswered. In some applications it may be practical to take this air from the high pressure regions of the cycle. In the four-port, through-flow cycle it can only be accessed via strategically positioned taps in the duct leading to the combustor. Furthermore, given the above description of interface distortion, it is questionable if the fresh air could be completely isolated from the recirculated hot gas. In the bypass cycle however, high pressure fresh

air is readily available after it is brought on board through the duct leading from the bypass loop. A tap located on the endplate could be used to extract the required amount using a relatively weak expansion wave. A second possible coolant extraction scheme would utilize the region between ports 1 and 3 in Figs. 2 and 3. If this latter scheme is practical however, the bypass cycle offers no extraction advantage over the four-port cycle.

Conclusions

A new bypass wave rotor cycle has been described which appears to solve the problems of exhaust gas recirculation and insufficient rotor cooling found in the conventional four-port cycle. Such a cycle has been successfully designed using a one-dimensional numerical simulation. Results from the simulation show that the bypass cycle, when compared with the four-port, through-flow cycle yields an 18% reduction in average rotor wall temperature, a 17% reduction in combustor inlet temperature, and equivalent or improved overall pressure ratio, p_{04}/p_{01} . The required combustor pressure drop is increased from 8.7% to 11.1% and the new bypass cycle may require more ducting; however, further experimental and numerical investigation is warranted given the substantial rotor and duct cooling benefits predicted.

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REPORT DOCUMENTATION PAGE

Form Approved
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1. AGENCY USE ONLY (Leave blank)		2. REPORT DATE April 1998	3. REPORT TYPE AND DATES COVERED Technical Memorandum	
4. TITLE AND SUBTITLE A Modified Through-Flow Wave Rotor Cycle With Combustor Bypass Ducts			5. FUNDING NUMBERS WU-523-26-33-00	
6. AUTHOR(S) Daniel E. Paxson and M. Razi Nalim				
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135-3191			8. PERFORMING ORGANIZATION REPORT NUMBER E-10987-1	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) National Aeronautics and Space Administration Washington, DC 20546-0001			10. SPONSORING/MONITORING AGENCY REPORT NUMBER NASA TM-1998-206971 AIAA-97-3140	
11. SUPPLEMENTARY NOTES Prepared for the 33rd Joint Propulsion Conference and Exhibit sponsored by AIAA, ASME, SAE, and ASEE, Seattle, Washington, July 6-9, 1997. Responsible person, Daniel E. Paxson, organization code 5530, (216) 433-8334.				
12a. DISTRIBUTION/AVAILABILITY STATEMENT Unclassified - Unlimited Subject Category: 07 This publication is available from the NASA Center for AeroSpace Information, (301) 621-0390.			12b. DISTRIBUTION CODE Distribution: Nonstandard	
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14. SUBJECT TERMS Wave rotor; Simulation; Computational fluid mechanics			15. NUMBER OF PAGES 14	
			16. PRICE CODE A03	
17. SECURITY CLASSIFICATION OF REPORT Unclassified	18. SECURITY CLASSIFICATION OF THIS PAGE Unclassified	19. SECURITY CLASSIFICATION OF ABSTRACT Unclassified	20. LIMITATION OF ABSTRACT	