Proceedings of ASME 2005
ASME TURBO EXPO 2005
June 6-9, 2005, Reno, Nevada

GT2005-68203

INSERTION OF SHOCK WAVE COMPRESSION TECHNOLOGY INTO MICRO TURBINES FOR INCREASED EFFICIENCY AND REDUCED COSTS

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ABSTRACT

The following analysis is presented to serve as a preliminary design guide for micro turbine engine designers to consider the potential advantages of incorporating the Rampressor into their recuperated engine designs. It is shown that the increase in compressor efficiency and the shift in optimum pressure will increase the efficiency of the engine and lower the recuperator inlet temperature and specific cost. This also provides the opportunity to increase the turbine inlet temperature and specific power without incorporating more costly air-cooled metal or ceramic components into the turbine design.

Ramgen Power Systems, Inc. (RPS) is developing a family of high performance supersonic compressor designs that combine many of the aspects of shock compression systems, commonly used in supersonic flight inlet design, with turbo-machinery design practices employed in conventional axial and centrifugal compressor design. The result is a high efficiency compressor that is capable of single stage pressure ratios in excess of those available in existing axial or centrifugal compressor designs.

This technology provides a tremendous opportunity for replacement and/or de-staging of multi-stage centrifugal or axial compressors in gas turbines for greater efficiency, less cost, fewer parts, lower weight, and reduced footprint. A conceptual single-stage supersonic compressor has been defined for integration with a micro turbine in the 200 to 500 kWe class. This configuration offers the potential to achieve the DOE Advanced Micro Turbine Systems goals of greater than 40% LHV electric efficiency and $500 per kWe package selling price.

INTRODUCTION

Global electric power capacity additions over the next 20 years are projected to reach over 1500 GW, or approximately twice the present operating capacity. Aging and congested power grids, rising fuel costs and lower emissions requirements have all caused stationary power generation manufacturers to respond with aggressive and costly “distributed generation” (DG) technology development programs. In addition, DOE funded activities, such as the Advanced Turbine Systems and the Advanced Micro Turbine Systems programs, are further evidence of a newly developing age in electricity production, transmission, and distribution.

Micro and mini gas turbines have been identified as part of the evolving DG resource technology portfolio. There is a great deal of interest in these products, although in recent years some of that interest has subsided due to shifting dynamics of DG concepts. This is largely a result of regulatory uncertainty, volatile and rising natural gas prices and the continued electric utility resistance to on-site generation, seen as a competitive threat.

These are not new phenomena, but their impact has been magnified by the general inability of the equipment developers to meet their claimed efficiency and selling price targets. Today’s micro turbine at best is 28% LHV net electric efficiency and it appears that, as currently designed, the highest these units will achieve is 34% LHV net electric efficiency. This limits their practical application to CHP projects, and as a result, also limits the number of units sold.

The basic problem is that these recuperated simple cycle designs cannot use the full turbine rotor inlet temperature

(TRIT) potential because of material cost vs. life limitations on the recuperator materials. A recuperator as shown in Figure 1 is a high temperature heat exchanger that serves to re-cycle the waste-heat from the turbine exhaust back to the cycle and before fuel addition. In current recuperated turbines the recuperator inlet temperature is at or very near the metallurgical limit of type 347 stainless steel, the generally preferred and affordable material. Alloys such as Inconel 625 have higher temperature capabilities but cost approximately three times more per pound than 347 SS. A switch to such alloys will increase the overall cost of the turbine and could negate the value of any potential efficiency gains.

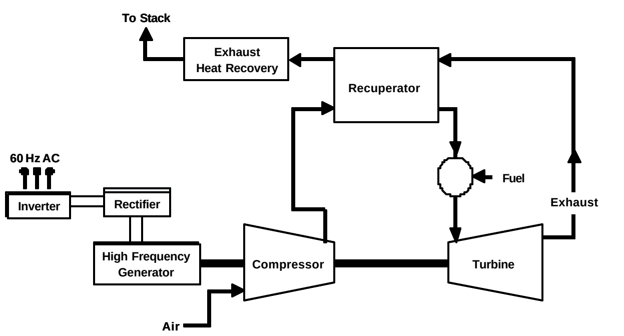

Figure 1. Single-Shaft Recuperated Micro Turbine

An approach to avoid high recuperator material costs is to decrease the turbine exhaust temperatures (EGT) by increasing the turbine pressure ratio. Presented in the paper are four recuperated engine configurations that show the potential advantages of inserting the Ramgen compressor technology (Rampressor™) into a micro turbine for greater compression efficiency, higher specific power, and lower overall manufacturing costs.

THEORY OF OPERATION

The Rampressor™ will deliver exceptionally high compression ratios per stage, with breakthrough efficiencies. This shock wave compression technology combines many of the aspects of shock compression systems, commonly used in supersonic

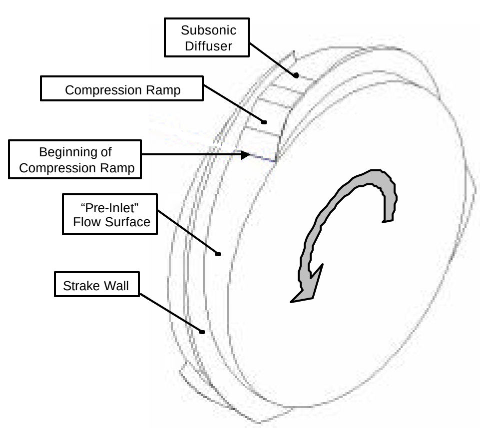

Figure 2. Supersonic Compression Stage Rotor

flight inlet design, with turbo-machinery practices employed in conventional axial and centrifugal compressor designs. The unique features of this shock wave compressor provide an additional degree of freedom, and can open up new and creative options for the gas turbine designer and package integrator that have not been considered before.

While supersonic axial compressor stages have been suggested by a number of other investigators [1-5], those rotor stages have invariably been configured so as to resemble conventional axial flow rotors. They have relatively high blade counts and inefficient shock systems generated by those blade surfaces.

The present RPS rotor is configured to use an oblique shock system, followed by a terminal normal shock and using the design guidelines typically employed in supersonic flight inlet design. By using this approach, it is anticipated that proven performance characteristics for the supersonic flight inlet systems would serve as a basis for the design.

The supersonic rotor flow-path is formed by three elongated blades or strakes mounted on the rim of the rotor. These strakes are mounted on the rotor at a shallow angle, typically 5° – 10°, and form the axial boundaries of each of the three flow-paths as shown in Figure 2. The strakes themselves are meant to do a minimum of compressive work. The principal shock system is generated by a compressive ramp, a largely planar surface integrated into the rim of the rotor. The ramp is developed so as to create a series of oblique shock waves that are reflected off the non-rotating compressor case.

The pattern of shock waves is comparable to the pattern of shock waves that might be found in a supersonic inlet, as would be applied to a missile or supersonic aircraft. The shock forming compression ramp of this flight inlet is analogous to the shock forming ramp on the rim of the supersonic compression rotor.

This configuration forms a rectangular flow-path with an inflow Mach number that is determined by the rotation rate of the rotor, in combination with any pre-swirl in the flow upstream of the rotor. Figure 3 shows an unwrapped view of the rotor together with the pre-swirl cascade, used to establish the rotor inflow field. Figure 3 defines the basic flow-path stations surrounding the rotor (shown in the inset). The direction of rotation of the rim shown in Figure 3 is left to right. The combination of the rotor speed and the pre-swirl from the upstream cascade of pre-swirl nozzles created a rotor inflow that is supersonic relative to the moving rotor.

PROOF-OF-CONCEPT TESTS

A proof-of-concept Rampressor system was designed and tested to demonstrate the basic operational characteristics of rotating shock compression when operating on air. The test unit processed 1.43 kg/s and produced a pressure ratio across the supersonic rotor flowpath of 2.25:1. The test unit operated at an inlet relative Mach number of 1.6. The goal of the tests completed in 2003 was to demonstrate the basic operability of the design including starting, surge characteristics, sensitivity to bleed flow and tip gap, as well as the ability of the analysis/design methodology to predict the behavior of the

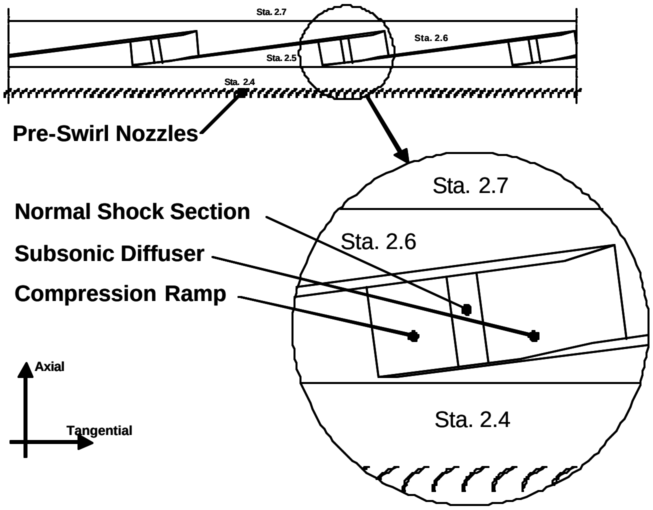

Figure 3. Rotor Station Numbering Convention

system. It was not the goal of this initial rig to achieve optimized system performance.

The experiments demonstrated that the Rampressor operates like a true compressor as described by Lawlor et al. [6]. The results indicated that the compression performance was highly dependant on tip gap and tip leakage. Another issue that was of concern was the performance of the system during maximum back-pressure conditions. The potential phenomenon of surge or inlet "un-start" was of significant concern. The results of these tests indicated that the supersonic flow-path in the rotor had very benign un-start characteristics. The off-design process occurred in a smooth and continuous manner resulting in a constant decrease in mass flow from 30% to 110% speed.

Performance prediction tools were used to model the rotor-only compression data. RPS-developed 1-D codes in combination with 2-D and 3-D CFD models were used to analyze the results. It was not the intention of the tests to optimize the pre-swirl system or recover the energy in the highly swirling rotor discharge flow.

In order to realize more of the potential of the system, follow-on tests are planned at higher rotor speeds. Based on performance modeling, these higher pressure ratio and higher speed systems have the potential to achieve increased efficiency levels, provided that upstream pre-swirl and downstream de-swirl losses can be minimized.

The test results were used to calibrate the models and make design predictions for higher pressure ratio designs. A higher pressure-ratio rig has been proposed and the design process for that system is underway. The goal for that design is to demonstrate a supersonic stage that would be traceable to a compressor capable of supplying air at ~125 psig. In such a system, the combination of a pre-swirl cascade, the supersonic rotor and a de-swirl/diffuser would produce an overall pressure ratio of 9.9:1. These tests are scheduled to commence in 4th Q, 2005. This program will finalize the proof-of-concept tests for pressure ratios from 2 to 10 and confirm the viability of the Rampressor technology. Positive results will lead to potential opportunities for air and gas compression, turbine compressor replacement/de-staging, and innovative engine designs for industrial and military applications.

SHIFT IN OPTIMUM PRESSURE

The improved performance of the Rampressor relative to a current industrial centrifugal compressor yields an overall cycle efficiency increase as shown in Figure 4. An industrial recuperated turbine configuration is presented with TRIT limited to maintain the recuperated inlet temperature for 347 SS. With a 90% effective recuperator and standard assumptions presented in Figure 4, the Baseline microturbine

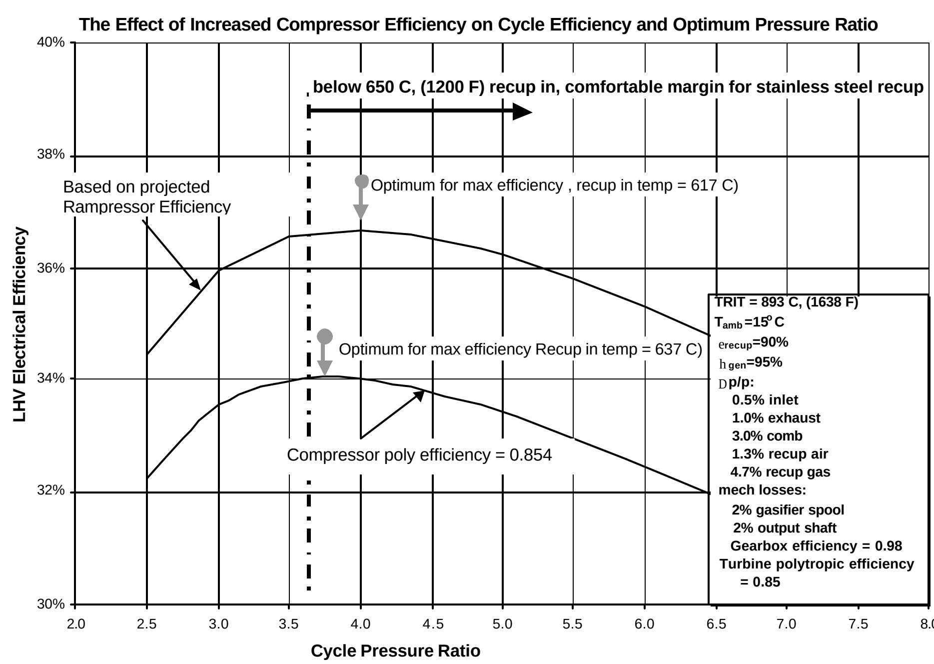

Figure 4. Effect of Increased Compressor Efficiency on Cycle Performance

cycle efficiency optimizes at a pressure ratio of 3.6:1 and a cycle efficiency of 34%. The recuperated engine, utilizing a Rampressor compressor stage, optimizes at a pressure ratio of 4.0:1, a cycle efficiency of greater than 36.8%, and a decrease in the recuperator inlet temperature of 20°C.

For economic reasons, recuperated gas turbines are commonly designed at higher pressure ratios for reduced component size and cost. The assumptions for centrifugal compressor polytropic efficiency are based on the equivalent adiabatic efficiency of 83% at the pressure ratio of 3.0. The Rampressor is predicted to achieve an adiabatic efficiency of 88% at the pressure ratio of 5.0. The assumptions for the analysis are included in Figure 4. The electrical efficiency is defined as the thermal input over the AC electric output on an LHV fuel basis.

The increase in compressor efficiency and the increase in optimum pressure ratio will increase the efficiency of the engine, and lower the recuperator inlet temperature as well as the recuperator specific cost. The increase in optimum pressure ratio will also allow full utilization of the temperature capability of metallic turbine rotors, without exceeding recuperator material life expectancy.

EFFICIENCY AND COST ANALYSIS

A comparison of performance and recuperator cost parameters for five cases is presented in Table 1 and Figure 5. The cases represent an industrial recuperated turbine in the 200 to 500 kWe class. The progression from one case to the next demonstrates an improvement in engine efficiency and specific power, as well as a significant decrease in recuperator specific cost by incorporating the Rampressor technology. The performance predictions and the normalized recuperator costs are based on models from the work of Kesseli et al. [7].

	Case 0	1	2	3	4
TRIT, C	893	893	983	1149	1205
PRc	3.6	5	5	10	8
Compressor type	Centrif	Ram	Ram	Ram	Ram
Compressor efficiency-adiabatic	0.826	0.879	0.879	0.847	0.858
Recup in Temp, C	637	576	645	642	725
Electrical LHV efficiency,	0.340	0.362	0.385	0.385	0.410
Specific power, kJ/kg	141.9	176.2	206.3	301.4	316.6
Recommended recup alloy	AISI 347	AISI 347	AISI 347	AISI 347	IN625
Recup core Specific Cost, $/kWe	92	64	56	36	61
Assumptions	See figure 5				
Stainless steel cost	3.18 $/lbm (Kesseli et al., IGTI 2003)				
Alloy-625 cost	11.08 $/lbm (Kesseli et al., IGTI 2003)				

Table 1. Comparison of Performance and Cost Parameters for Five Cases

The computational models are validated by instrumented industrial core rig tests. The recuperator core design parameters are integrated into Brayton-cycle based engine prediction codes. The costs have been estimated using material and labor rates from an industrial manufacturer. The specific costs presented are solely for the purpose of providing estimates for cycle optimization analysis and should not be construed as indicative of an industrial quotation.

	Case 0	1	2	3	4
Power (kW)	193	246	287	433	446
Core engine specific cost ($/kW)	420	338	289	198	188
Generator/power electronics, transmission $/kW	80	80	80	80	80
Recuperator package ($/kW)	138	93	83	51	89
Packaged engine specific cost ($/kW)	638	511	452	329	357
% cost decrease	-	20%	29%	48%	44%

Table 2. Comparison of Manufacturing Cost Parameters for Five Cases

A comparison of costs per kWe of the conceptual micro turbines and recuperators for the five cases is presented in Table 2. The power for each turbine is based on the specific power from Table 1 and an assumed air flow of 1.36 kg/s (3 pps) to match the current Rampressor design. A target cost of $500 per kWe for the Baseline engine yields a manufacturing cost of $96,500. The shaft to electric power conversion elements of the engine are assumed to represent $80/kWe of the Baseline cost. For scaling purposes, the normalized cost per kWe of the power conversion elements are constant, while the absolute cost of the core engine subassemblies are assumed constant.

The basis of this assumption is that the core engine turbomachinery cost does not change with the addition of the Rampressor and operation at increased pressure and temperature ratios. Alternatively stated, the core engine turbomachinery cost is most strongly influenced by the engine mass flow and component size, and less dependent on component speed and temperature. This engine cost is assumed for the cost predictions for each configuration. The recuperator package costs in Table 2 are estimated by increasing the core recuperator costs in Table 1 by a factor of 1.5.

Case 0 - Metal Turbine

Case 0 is established as the Baseline, with component efficiencies and turbine temperatures typical of today's recuperated microturbines. The pressure ratio was selected near the cycle's optimum for maximum efficiency, while respecting the 650°C operating limit of today's SS (AISI 347) recuperators.

Un-cooled metallic turbine rotors are commonly applied between 950°C and 983°C. The recuperator cost goal, however, compels the use of a stainless steel alloy, but the 40,000 hour life in service objective limits the EGT between 625°C and 650°C. Unfortunately, this also necessitates limiting TRIT, and therefore the turbine efficiency potential. In the baseline scenario, TRIT is limited to 893°C which limits the net electric LHV efficiency to 34% and a high recuperator core specific cost of $92 per kWe. The overall cost of the Baseline engine is an estimated $638 per kWe. The following case studies present various scenarios to improve both cycle efficiency and specific cost.

Case 1 - Metal Turbine

Case 1 demonstrates the efficiency improvement achieved by replacing the conventional centrifugal compressor with the Rampressor at an adiabatic compression efficiency of 88%. Maintaining the same turbine inlet temperature as the Baseline, but increasing the pressure ratio from 3.6:1 to 5.0:1, increases the cycle efficiency from 34.0% to 36.2%, and increases the specific power by 28%. There is a 30% reduction in the size and cost of the recuperator, and the manufacturing cost of the engine has decreased by 20%, to $511 per kWe.

The recuperator inlet temperature is 576°C, nominally 76°C below the working limit of common stainless steel alloys. The work by Kesseli et al. (2003) shows a dramatic reduction in recuperator cost and size by operating at higher pressures, with some small sacrifice in engine efficiency. Figure 5 indicates that the increased Rampressor efficiency results in a less severe drop off in engine efficiency vs. centrifugal compressors at these higher pressures.

Case 2 - Metal Turbine

Case 2 exploits the advantages of the Rampressor's higher efficiency by increasing the TRIT to 983°C, while respecting the working threshold limit of the stainless steel recuperator. The selected turbine inlet temperature is achievable with state of the art metal alloys and un-cooled blades. This configuration can achieve an increase in cycle efficiency to 38.5%, and a 45% increase in specific power over Case 0. There is a 39% reduction in the size and cost of the recuperator, and a reduction in overall engine manufacturing cost of 29% over Case 0, to $452 per kWe.

Case 3 - Ceramic or Air-Cooled Metallic Turbine

Case 3 explores the benefit of selecting either air-cooled metal or ceramic hot section components by increasing the TRIT to 1149°C and the pressure ratio to 10:1. The shift in pressure to the right in Figure 7 allows for the continued use of the less expensive stainless steel recuperator material at the threshold temperature of 642°C. The Rampressor's high efficiency combined with the high temperature ratio on the turbine results in a minimal roll-off in efficiency as shown in Figure 7. This configuration will achieve the same cycle efficiency of 38.5% as Case 2 but also show a factor of two increase in specific power and a 61% reduction in the size and cost of the recuperator vs. Case 0. This is a 48% reduction in overall manufacturing cost vs. Case 0.

The high Rampressor efficiency works to improve the feasibility of ceramics by lowering the required turbine tip speed. As compared to the Baseline, the gasifier turbine speed is reduced by about 5.3%, to a moderate level of 568 m/s. This results in a reduction in rotor stress of nominally 11%. Though these conditions represent a significant technical challenge, the

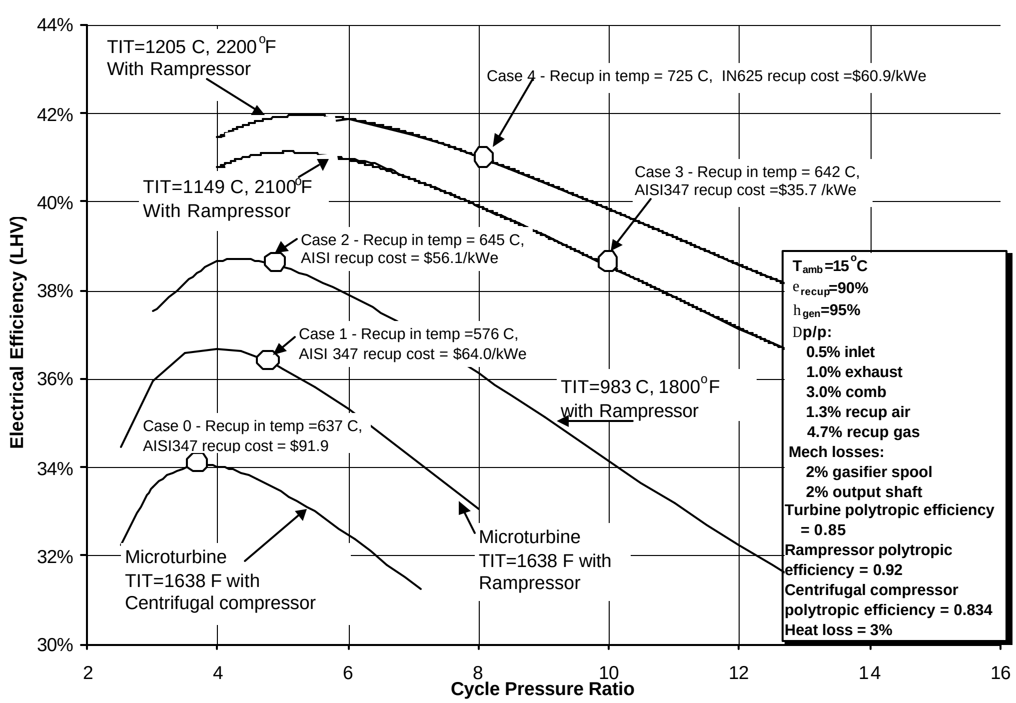

Figure 5. Correlations Representing Five Case Studies

stress levels are below those of ongoing DoE Advanced Micro Turbine programs.

Case 4 - Ceramic Turbine

Case 4 tests the range of turbine inlet temperatures under investigation for the US DOE advanced ceramic development program. Operation at 1205°C gas inlet temperature constrains the ceramic rotor to operate at relatively low tip speeds and stress levels. Selecting a pressure ratio of 8:0, combined with the high Rampressor efficiency, results in a gasifier tip speed of 531 m/s, and an acceptable rotor stress in the 200 MPa range. Progress by the US ORNL, Kyocera, and as reported by Kesseli et al. [8] indicates that a radial turbine formed from silicon nitride operating within this low stress range will achieve very long life.

Due to the lower pressure ratio, the recuperator gas inlet temperature will move into the range where common super-alloys such as Alloy 625 (IN625) can be successfully applied. The premium cost of this alloy is moderated by the engine's relatively high specific power. This configuration will achieve an increase in cycle efficiency to 41%, surpassing the DOE target of 40%. Although the more expensive super-alloy is required for the recuperator, Case 4 still demonstrates a path to achieving a 34% reduction in the core recuperator cost and a 44% reduction in engine manufacturing cost at $357 per kWe

This level of improved efficiency and specific power, and reduction in manufacturing costs per kWe will greatly enhance the potential to meet the DOE advanced micro turbine program goals of 40% cycle efficiency and $500 per kWe for the complete turbine-generator package. This will also allow broader participation in the "power only" markets where sales opportunities are several times larger than for combined heat and power alone.

SUMMARY AND CONCLUSIONS

A conceptual single-stage supersonic compressor has been defined for integration with a microturbine in the 200 to 500 kWe class. The principal claims for the technology are that it can achieve single stage pressure ratios up to 10:1 with adiabatic stage efficiencies of up to 88%. The superior compression performance of this technology will greatly enhance the potential to achieve the DOE Advanced Micro Turbine program goal of 40% LHV net electric efficiency and the overall package cost of $500 per kWe, without resorting to the use of ceramics.

The preceding analysis has been presented to serve as a preliminary design guide for micro turbine engine designers to consider the potential advantages of incorporating the Rampressor into their recuperated engine designs. It is shown that the increase in compressor efficiency and the shift in optimum pressure will increase the efficiency of the engine and lower the recuperator inlet temperature and specific cost. This also provides the opportunity to increase the turbine inlet temperature and specific power without incorporating more costly air-cooled metal or ceramic components into the turbine design.

ACKNOWLEDGMENTS

This effort was funded in part by the Department of Energy under the cooperative agreement #DE-FC26 00NT40, awarded between the period of October 1, 2003 and September 30, 2004.

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