ADVANCED TURBOCHARGER ROTOR FOR VARIABLE GEOMETRY TURBOCHARGING SYSTEMS

R.J. Stafford, J.M. Mulloy, T.M. Yonushonis, H.G. Weber, and M.J. Patel
Cummins Engine Company, Inc,

INTRODUCTION

Turbocharging of diesel engines has enhanced fuel economy and reduced diesel engine emissions. The initial applications of turbochargers to heavy duty diesel engines during the early 1970's reduced Bosch smoke (a measure of particulate matter used at the time) from 2.4 to 0.6 units. At the same time, engine specific fuel consumption was reduced from 0.36 lb./bhp-hr to 0.35 lb./bhp-hr. Since that time, a variety of improvements including further advances in turbocharger technology have reduced particulates to the level that they are below 0.1 Bosch units (the bottom of the scale) while specific fuel consumption is at the 0.32 lb./bhp-hr level. At optimum conditions, modern diesel engines can achieve in excess of 42% thermal efficiency which is competitive with stationary power plant efficiencies. One hundred percent (100%) of Cummins heavy duty diesel and medium duty diesel engines are turbocharged. Current turbochargers are optimized at one set of engine conditions and by necessity, at the off-design conditions or transient conditions the fuel economy and emissions performance are penalized.

Cummins and Holset, a Cummins subsidiary, have been working together on development of an advanced variable geometry turbocharger system for heavy duty diesel engines to improve off-design conditions. Variable geometry turbocharging provides an additional control of the turbocharger and assists in low speed response by modifying the inlet area to the turbine rotor. Cummins has also investigated modified radial rotors to improve off-design turbocharger performance. We believe that these new rotor designs coupled with variable geometry control systems can improve turbocharger efficiency by up to 15% at off-design points, which would result in particulate reductions of up to 40% and engine fuel economy improvements of 2 to 4 %. These improvements represent major advances for state-of-the-art diesel engines.

The current variable geometry turbines utilize existing turbine rotor or impeller designs. To be successful, the variable geometry turbine must provide acceptable efficiency over a 2:1 or 3:1 range in turbine flow at a given pressure ratio. The typical turbine has great difficulty in doing this, because the current impeller designs cannot efficiently accommodate such a wide range of flow conditions. Figure 1 shows the Dry Particulate versus Response curves for standard Fixed and Variable Geometry turbine wheels.

A rotor was designed and a prototype fabricated which showed as much as a 10% efficiency improvement at off-design conditions (1, 2). The leading edges are blunt and rounded to accept the flow from the turbine nozzles at a variety of inlet conditions with a minimum of losses. Photographs of this prototype are shown in Figure 2. The rotor efficiency is better at all conditions and the advantage improves as it operates at conditions further from the design point. Unfortunately, the conventional materials from which this turbine rotor was constructed had inadequate strength to allow its use on engines, and had such high rotational inertia that transient response would have been severely compromised.

MODELING

With the introduction of high strength, light weight materials, such as silicon nitride (Si_3N_4) and titanium aluminide (TiAl), the rotor design with blunt, rounded leading edges was revisited. The original design was used as a baseline design. Stress modeling of the baseline design predicted peak stresses of 280,000 psi for TiAl and 225,000 psi for Si_3N_4. These stresses greatly exceed the yield stress for both materials.

A series of design iterations was conducted which minimized the peak stresses while providing a shape which could be manufactured. Two dimensional stress analysis of this design predicted peak stresses of 49,000 psi for TiAl. Three dimensional stress analysis and minor design optimization resulted in a final design with peak stresses of 70,000 psi for TiAl. The finite element stress comparison from the original design to final optimized design for the TiAl rotor is shown in Figure 3. Stress modeling was not conducted for Si_3N_4 because the rotor shape was considered too complex to be produced by currently available ceramic manufacturing methods. Photographs of the optimized rotor design are shown in Figure 4.

MANUFACTURING

The turbocharger rotors for testing were manufactured using production shell casting methods with a TiAl alloy. After metal casting and solidification, the rotors were chemically cleaned to remove the ceramic shell. The rotors were then hot isostatic pressed to densify the component and heat treated to develop the optimal material properties for the TiAl alloy.

The rotors were machined to develop the blade tip profiles and a shaft bore through the center. The wheels were then attached to production-like shafts using a locking nut and set of shims to set the correct operating height within the turbo-charger shroud.

TESTING

Performance testing was conducted at the Cummins Turbine Mapping Facility. This facility determines the turbine stage efficiency by measur-ing the compressor and bearing system work as well as the turbine mass flow and pressure ratio. The

Figure 1

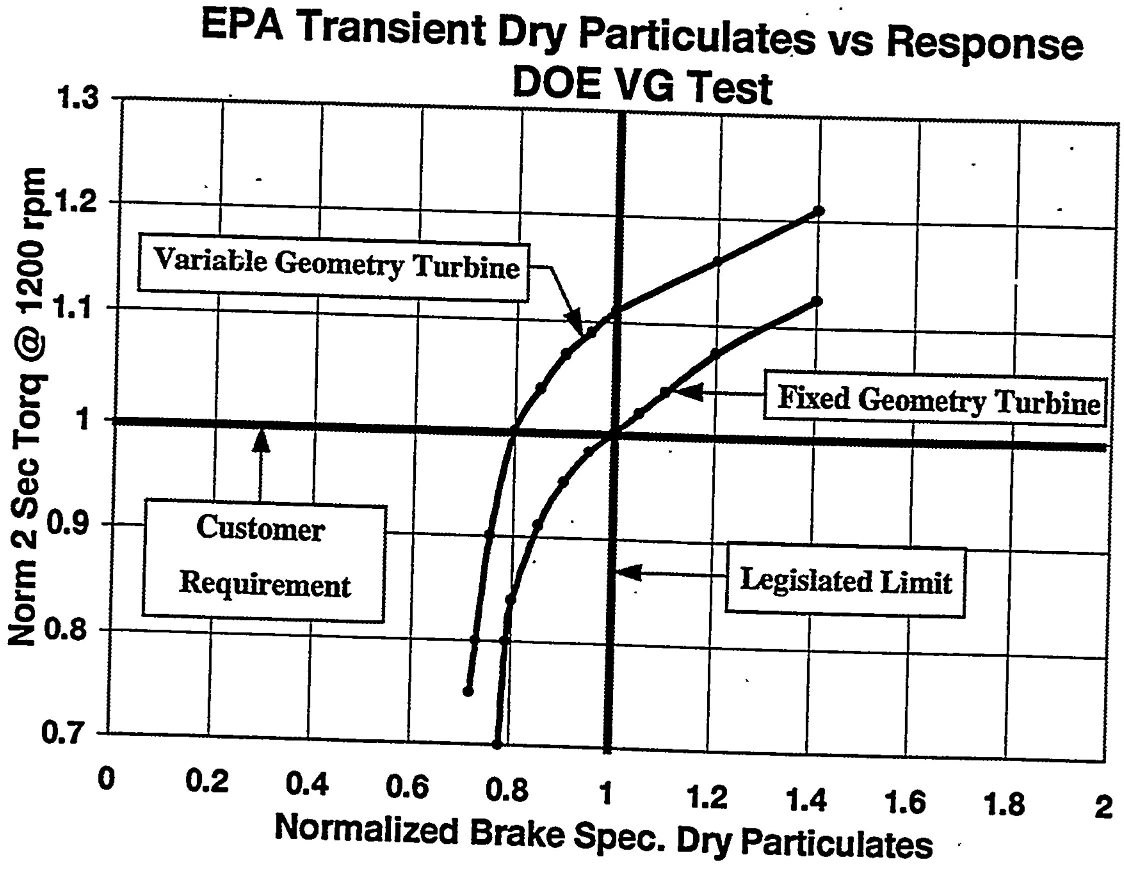
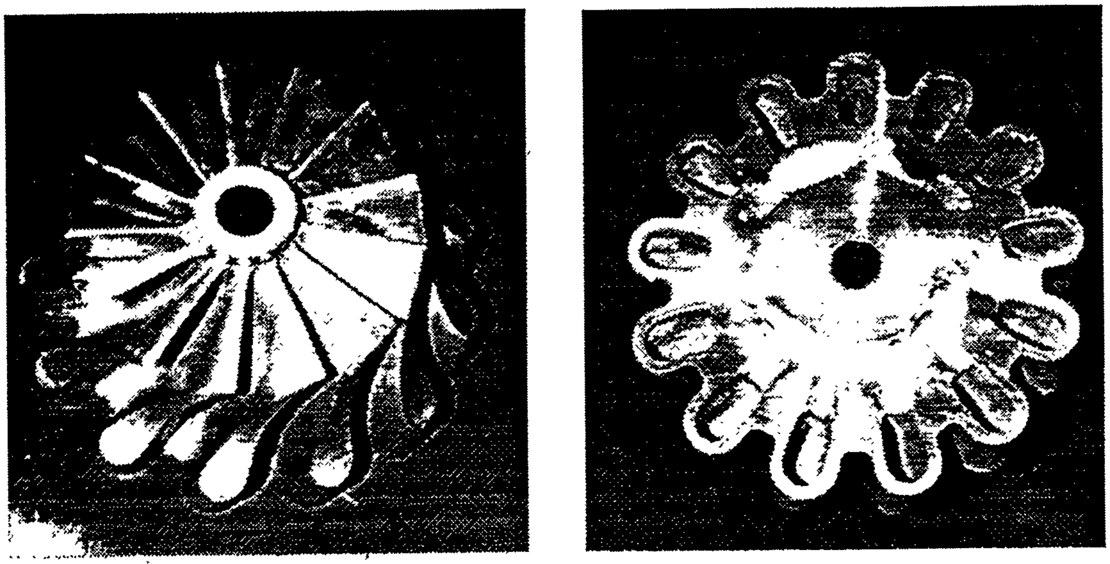


Figure 2 - Original Prototype Airfoil-style Rotor

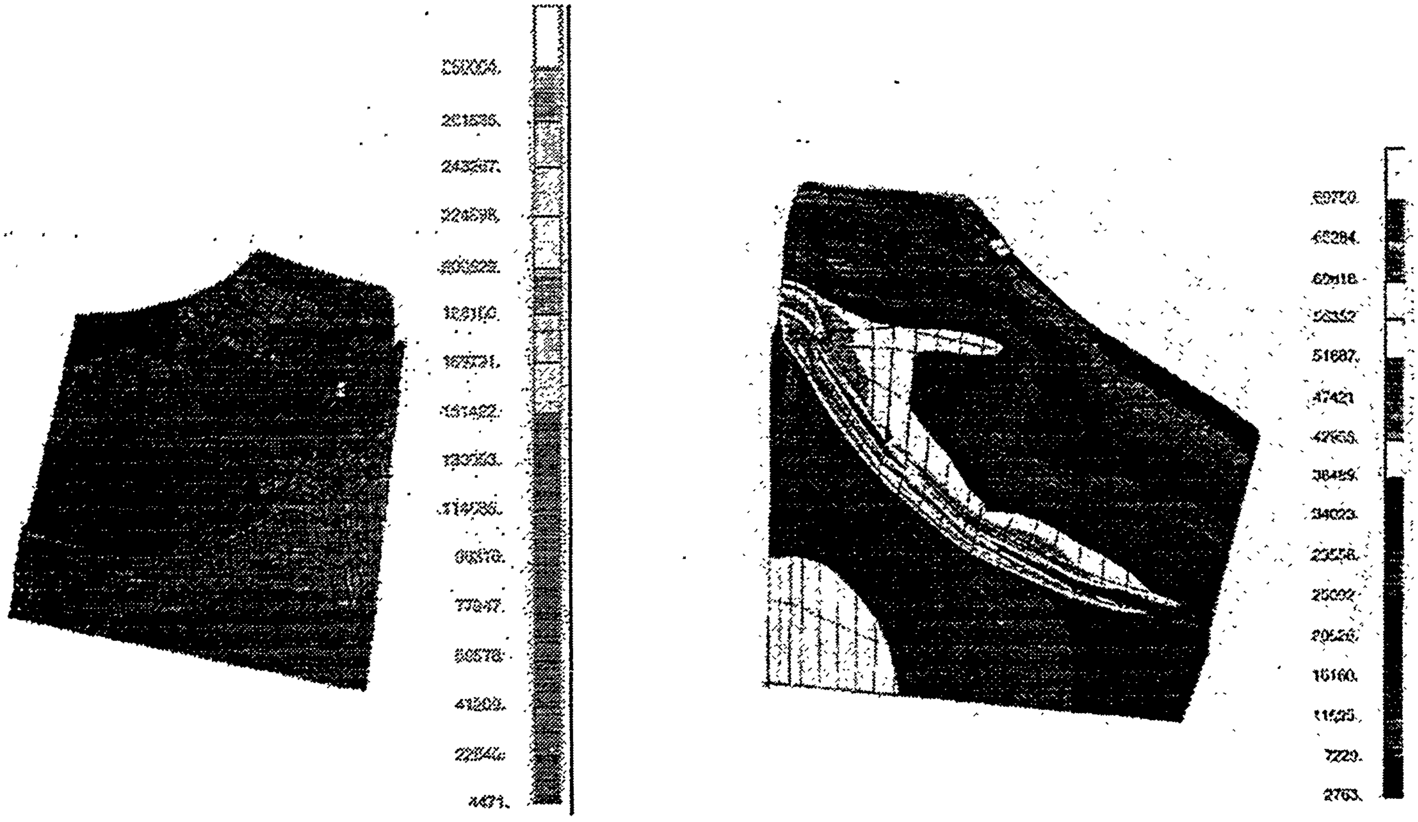


test facility measures turbine efficiency as Total to Static, meaning the turbine stage effici-ency is reduced by the kinetic energy left in the gas as it exits the turbine. This efficiency measurement represents the single staged diesel engine turbocharger efficiency.

Performance Mapping

To determine the performance characteristics of the experimental rotor in a variable geometry turbocharger (VGT) system, a Holset HX75 turbocharger unit was selected

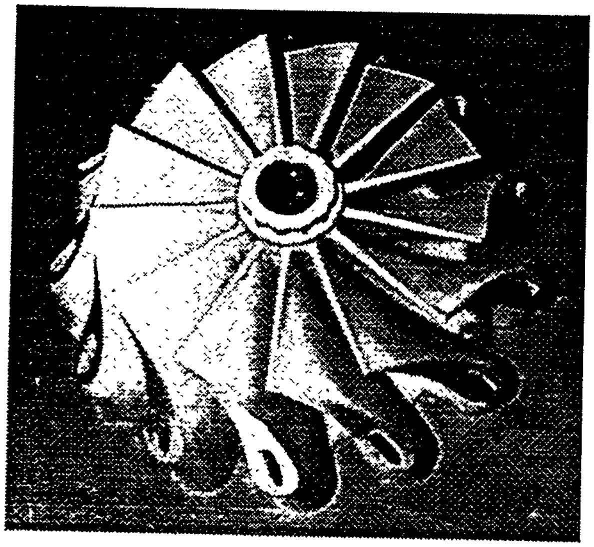
Figure 3 - Finite Element Stress Analysis Comparison

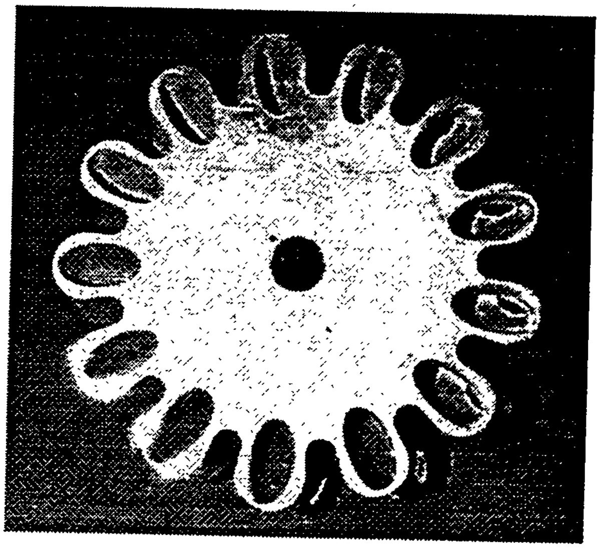


Original Design, 280 Ksi

Final Optimized Design, 70 Ksi

Figure 4 - Optimized Final Design





and nozzle rings were machined to expedite the testing. Four nozzle ring vane designs were made, one curved and three straight. The three straight vane designs had different critical areas (variations in nozzle throat areas) to increase actual air flow rates. All the nozzle rings were originally machined to 15 mm vane height to fit the HX75 unit baseline and then machined down between maps to the next desired nozzle vane height. The assembled turbo-charger rotor and nozzle ring are shown in Figure 5.

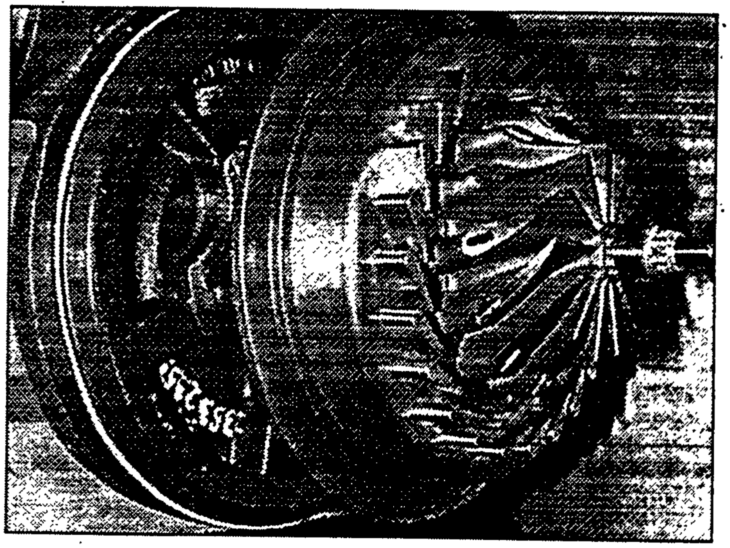
The testing was conducted using standard test procedures. During mapping, we had two major rotor bursts which destroyed the nozzle rings (see detail in Burst Testing). Once the cause of the failures was deter-mined to be related to defects in the wheel, the mapping proceeded without incident. The initial testing showed that the curved nozzle vane gave the slightly better perform-ance. Therefore, we mapped the HX75 unit using the curved nozzle vanes with vane heights of 15, 13, 11, 9, 7, 5, 3, and 2 mm. Then, in order to obtain data with higher flow rates, two of the straight vaned, increased throat area nozzle rings were mapped at a vane height of 15 mm.

Performance Comparison of the Experimental Rotor to a Conventional Rotor

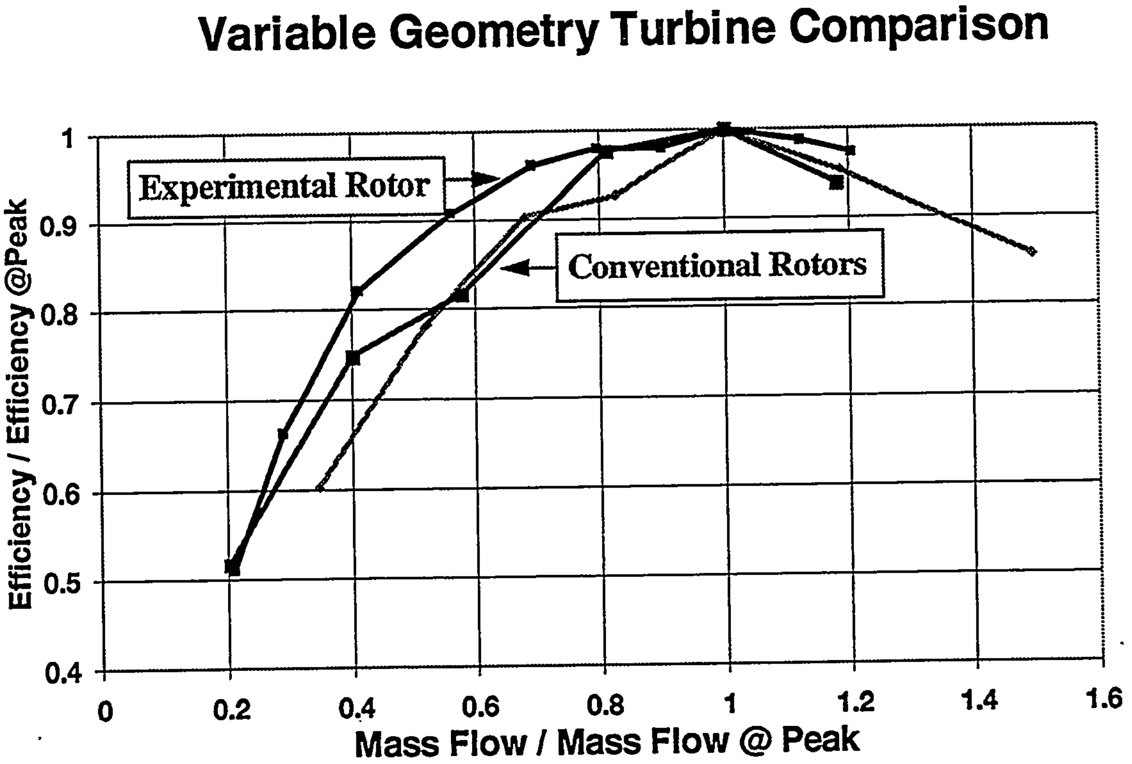
The purpose of this effort was to improve the low flow efficiency of the VGT turbine which would then increase acceleration capabilities therby reducing particulate emission and improve overall trip fuel economy. Figure 6 shows the comparison of the experimental rotor and two standard rotors in VGT systems. Since all the units had different mass flow capacities it was necessary to evaluate at the data on a non-dimensional basis. To determine the non-dimensional mass flow ba-sis, the mass flow point of peak turbine efficiency (where the nozzle vane height was equal to the rotor inlet tip height) was used as the 100% point. All other mass flows were divided by this value.

Figure 5

Assembled Turbocharger Rotor and Nozzle Ring



Firgure 6



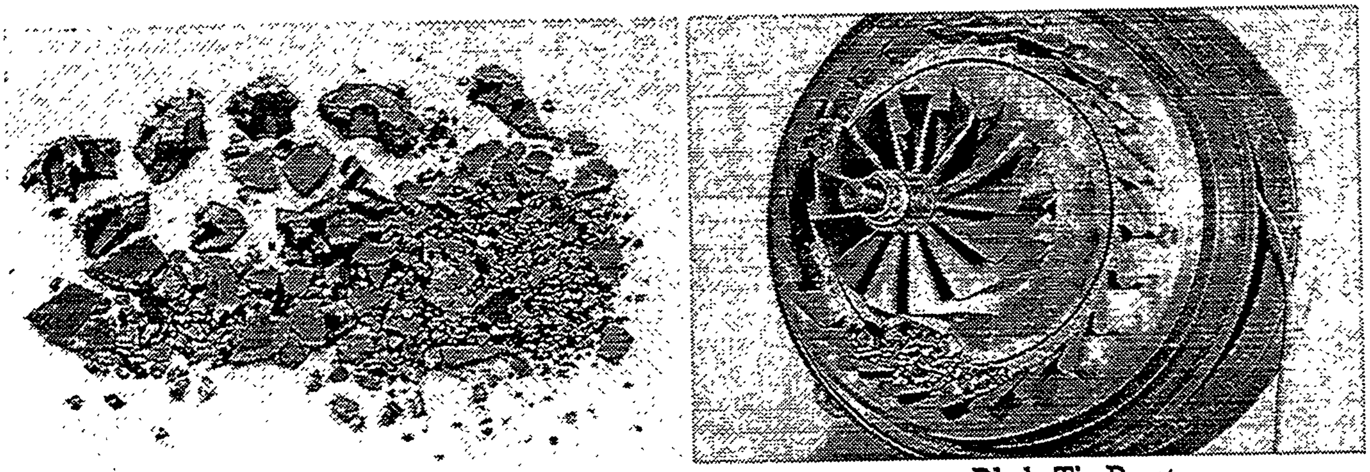
All the data points shown in Figure 6 are raw data without any curve fitting. As can be seen, the experimental rotor has significantly improved the turbine stage efficiency over the 0.30 to 0.80 range as well as showing improvement in the 1.10 to 1.20 range.

Burst Testing

The TiAl rotor was inadvertently accelerated to high speed conditions when a hose in the load absorption system which controls the rotor speed ruptured. The rotor was operating at 65,000 rpm at the time the hose ruptured, and immediately spun up to a speed which caused a hub burst of the rotor.

The TiAl rotor fragments were contained completely within the housing of the turbocharger which was a successful result. The TiAl fragments were generally small granular pieces with the largest fragments measuring less than 1 in. by 2 in. by 1 in. Analysis of the fragments did not reveal any

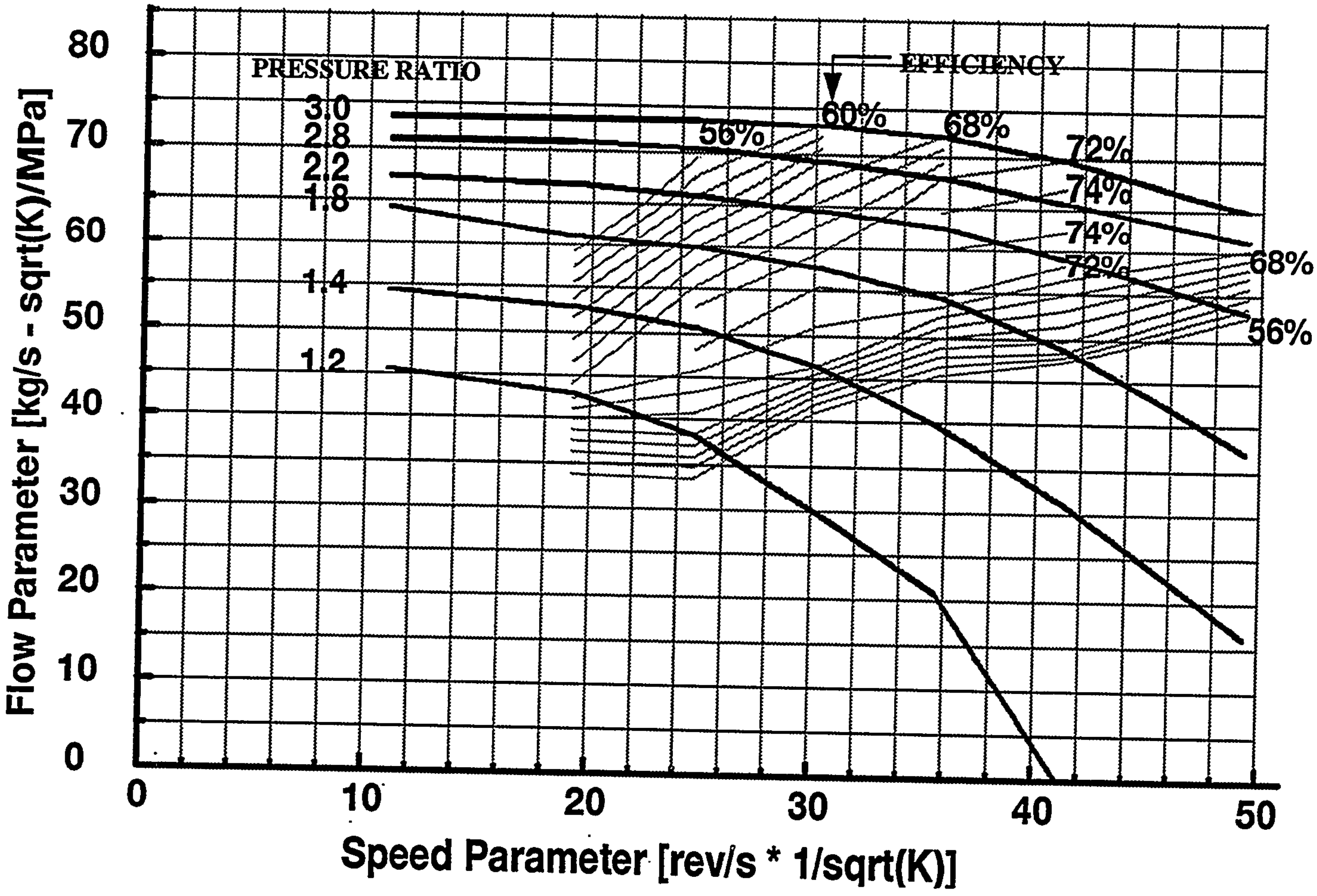
Figure 7 - Burst Test



Hub Burst

Blade Tip Burst

Figure 8 - Typical Turbine Map



mechanism for the fracture, however, dye penetrant examination of the remaining rotors showed small cracks in the machined bore on two rotors. A crack may have been present in the shaft bore causing the hub burst fracture when the rotor made a rapid change in speed.

A second TiAl rotor fractured during testing with the blade tips being removed from the hub. The blade tips again fragmented into small granular pieces and were contained completely within the housing. Examination and analysis of the hub and fragments did not reveal any mechanism for the fracture. However, based on the condition of the remaining rotors, it was suspected that a small pre-existing surface defect or crack caused one blade tip to fracture. The fracture caused rapid progressive damage of the other blade tips in the tightly toleranced space. The hub burst and blade tip burst are shown in Figure 7.

Demonstration of Advanced Turbocharger System

The objective was to evaluate the novel rotor geometry in a variable geometry

turbocharger system and compare the operation to the operation of a conventional design rotor in a variable geometry turbocharger system.

In order to compare the novel geometry turbine rotor to a standard rotor, Cummins used in-house engine simulation and vehicle simulations pro-grams. These programs are computer simulations of engine conditions which have been modeled and verified through comparison with engine performance data.

To complete this evaluation and comparison, engineers matched the engine simulation program conditions to an experimental data set which used the standard rotor in a variable geometry turbine. Then the simulation was run and used to predict the experimental system performance over the operating range. This simulation process is a standard operating procedure at Cummins and the results for various comparisons have consistently shown excellent agreement with experimental testing. For the simulation to work correctly the subsystem component maps must be supplied as input. These maps were available for the standard rotor in the variable geometry turbine but had to be generated for the experimental rotor.

For a variable geometry turbine several turbine maps must be generated to show the overall capability of the unit. Each map consists of running the unit at a constant pressure ratio while varying the rotor speed to determine the flow capacity and efficiency. This must be done at several pressure ratios since the engine will drive the turbomachinery to operate at pressure ratios from 1.0 to approximately 4.0. A typical map will include data for 1.1, 1.2, 1.4, 1.8, 2.2, 2.6 and 3.0 pressure ratios. Figure 8 is a sample turbine map showing rotor speed vs. efficiency and mass flow at different pressure ratios. For the variable geometry unit with the novel wheel, eight maps were generated at different nozzle vane heights to simulate what would be required during an actual engine test.

Once the mapping was completed, a valid comparison could be made with Cummins engine simulation programs. This comparison was accomplished by running the simulation at various operating conditions which have been determined to be major contributors to the system fuel consumption by engine measurements over a standard driving route on the highway. A different set of operating conditions, determined by engine measurements, was used for emissions reduction predictions.

The final results of the comparison after analysis with the simulation programs showed predicted improvements in cycle fuel consumption of 2 to 3 % and reduction in particulate emissions of approximately 30%.

CONCLUSIONS

The demonstrated performance of the experimental rotor in the VGT has provided the means to reduce particulate by approximately 30% and improved cycle fuel consumption by 2 to 3%. The basis for these results is a comparison to previous engine testing and cycle simulation modeling on the original prototype rotor. These results showed additional improvements in the cycle simulation model which predicts the above overall improvements compared to conventional designed rotors.

ACKNOWLEDGMENTS

This work was funded in part by the California Energy Commission, contract 500-92-042, and by the U. S. Department of Energy, contract DEN3-373. Technical review of the work was provided by Dr. Jim Patten and the senior management of the Cummins Technical Center. The oversight of Mr. John Fairbanks at the U. S. Department of Energy and Dr. Gary Yowell and his colleagues at the California Energy Commission was also greatly appreciated. Completion of this work required the efforts of many people and these people are listed below:

Holset Turbocharger
    Jacqueline Kearney
MacNeal-Schwendler Corporation, PDA Division
    Dr. J. G. Crose, Robert Holman, David Bremmer, Ernie Julian, John Norman and Robert Schindler
Consultant to MSC, PDA Division
    Douglas Marx
Cummins Engine Company, Inc.
    George Delay, Paul Park, Scott Kronhart, Stanley Bishop, Roger England, Mike Goble, Bob Gray and Scott Sims
Precision Castparts Company
    Jim Barrett, Srivats Ram and Dave Harmon
Port City Machine and Tool Company
    Doug Hartman

REFERENCES

1. Mulloy, J. M. and Weber, H. G., "A Ra-dial Inflow Turbine Impeller for Improved Off-Design Performance", ASME Paper No. 82-GT-101

2. Flynn, P. F., Weber, H. G. and Mulloy, J. M., U.S. Patent No. 4243357, Jan. 6, 1981.