Final Report

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Project Objective

The ARES program goal is to improve the efficiency of natural gas reciprocating engines. The ARES project is structured in three phases with higher efficiency goals in each phase. The ARES objectives are as follows:

1. Achieve 44% (ARES I), 47% (ARES II), and 50% brake thermal efficiency (BTE) as a final ARES III objective
2. Achieve 0.1 g/bhp-hr NOx emissions (with after-treatment)
3. Reduce the cost of the produced electricity by 10%
4. Improve or maintain reliability, durability and maintenance costs

Background

The ARES program was initiated in 2001 to improve the overall brake thermal efficiency of stationary, natural gas, reciprocating engines. The ARES program is a joint award that is shared by Dresser, Inc., Caterpillar and Cummins. The ARES program was divided into three phases; ARES I (achieve 44% BTE), ARES II (achieve 47% BTE) and ARES III (achieve 50% BTE).

Dresser, Inc. completed ARES I in March 2005 which resulted in the commercialization of the APG1000 product line. ARES II activities were completed in September 2010 and the technology developed is currently being integrated into products. ARES III activities began in October 2010.

ARES Phase I

ARES Phase I project started using an existing Waukesha product, the 16 cylinder, 48 liter displacement, 880 kW VGF P48 engine. The VGF P48 has the following specifications shown in Table 1.

Parameter	**Value**
Bore	152 mm
Stroke	165 mm
V angle	60°
Number of cylinders	16
Displacement	48 l
Power output	880 kW @ 1800 RPM
NOx emissions	2.0 g/bhp-hr
CO emissions	1.34 g/bhp-hr
THC emissions	1.61 g/bhp-hr
Efficiency	36.3% BTE
Fuel consumption	2402 kW
Heat rejection to coolant	617 kW
Heat rejection to aux. water circuit	228 kW
Heat rejection to exhaust	665 kW
Exhaust temperature	448 °C

Table 1 Specifications for the Waukesha VGF P48 base engine

APG1000 Combustion Development

As detailed in ASME paper ICEF2006-1510, “Development of the Waukesha 16V150LTD Advanced Power Generation Engine” by Ed Reinbold and Daniel Mather [1], the APG1000 combustion system was a dramatic improvement over the VGF combustion system. To drive down NOx emissions it was necessary to increase the air-to-fuel ratio. Increasing the air-to-fuel ratio slows down combustion leading to lower engine efficiency. A unique approach to maximizing turbulent kinetic energy, and thus increasing flame speed, was used that employed state-of-art CFD and a genetic optimization methodology. The final combustion bowl shape is shown in Figure 1. Not only does this shape provide better combustion, it is also less expensive to manufacture and allows a simpler piston removal procedure compared to the VGF combustion chamber.

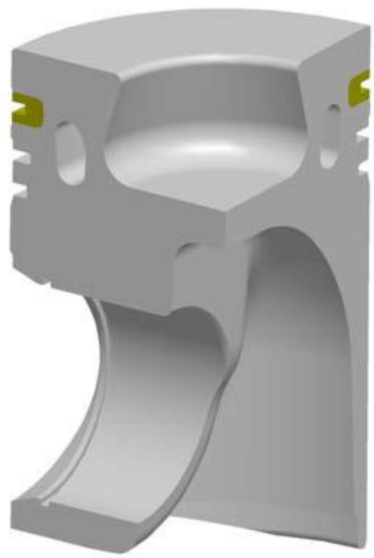

Figure 1 APG1000 combustion bowl shape

In conjunction with the new combustion bowl shape a non-fueled prechamber around the spark plug was developed. That development is detailed in ASME paper ICEF2004-821, “Optimization of a non-fueled Prechamber Ignition System for a Lean-Burn, Industrial Natural Gas Engine”, by Corey Honl [2]. Figure 2 shows a prototype of the non-fueled prechamber. Compared to previous technology fueled prechambers this design resulted in a lower cost and improved reliability, and in conjunction with the new combustion chamber shape, offers excellent combustion.

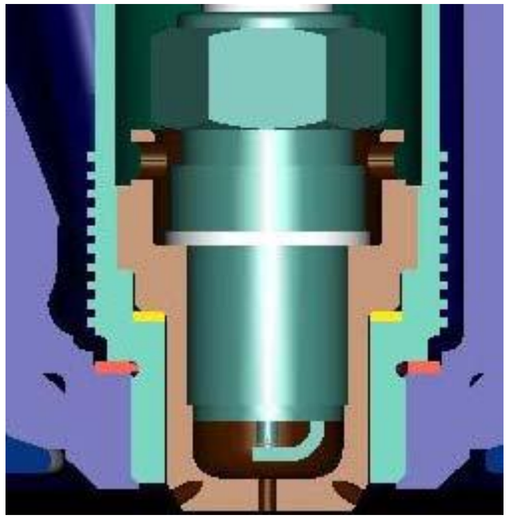

Figure 2 Prototype APG Non-Fueled Prechamber

Table 2 shows the results for the new APG1000 combustion system compared to the VGF combustion system.

Parameter	VGF P48	APG1000
0-10% Mass Fraction Burned; ignition delay	Base	13% longer
Std. Dev. 0-10% Mass Fraction Burned	Base	31% less
Ignition Timing	18°	20°
Coefficient of Variation of IMEP	Base	34% lower
Lean limit, λ	1.68	1.76

Table 2 Comparison of APG1000 combustion system to VGF combustion system operating at 1800 RPM, 1 g/bhp-hr NOx, and 16 bar BMEP

In summary the APG1000 combustion system allowed operation at lower NOx emission levels, while retaining good spark plug life, more consistent combustion, at higher power and efficiency than the previous VGF combustion system.

APG1000 Mechanical and Performance Development

There are significant mechanical differences between the VGF and APG1000 engines. ASME paper ICEF2006-1517 "Design and Analysis of the Waukesha APG1000 Engine", by Rodney Nicoson and Julian Knudsen [3] details the many changes made to deal with the higher power output of the APG1000 engine. New designs were required for the following main components: crankshaft, connecting rod, piston and cylinder heads. Modifications were made to the crankcase and camshaft. These tasks were heavily driven by FEA, CFD, and heat transfer analysis. The intake and exhaust ports of the new cylinder head feature lower flow losses than the VGF ports. The net effect of the lower loss ports and other design changes was to reduce the engine pumping loss even with 35% higher air flow through the APG1000 engine compared to the VGF. Figure 3 shows a CFD analysis of coolant flow through the lower water jacket. The APG1000 flow on the right side shows more even and higher flow velocity between the valves. The more even and higher velocity flow provides cooler valve operation leading to longer head life and minimizes the possibility of head cracking.

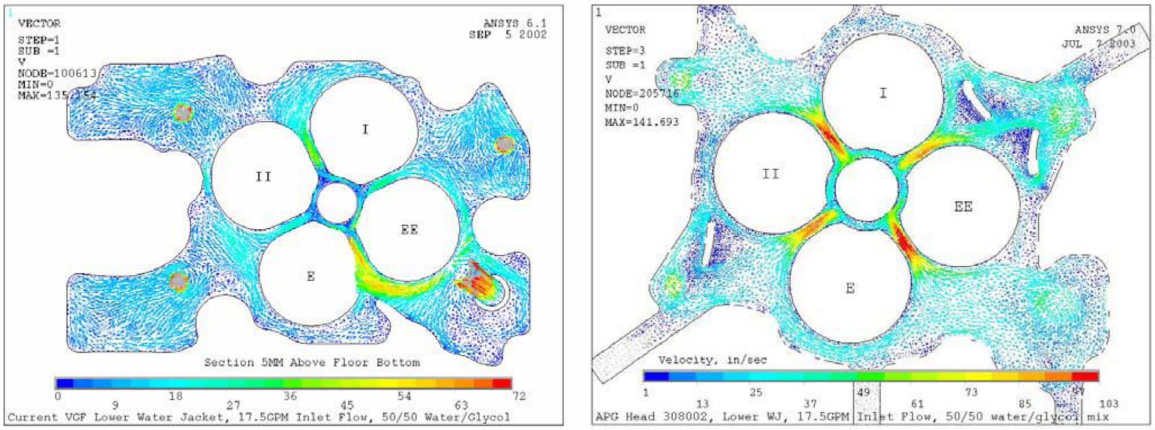

Figure 3 CFD analysis of flow through the lower water jacket. VGF shown on the left side and APG1000 shown on the right side

Fundamental Research and Development

Cooperative research with Colorado State University & the Massachusetts Institute of Technology on friction reduction technology was detailed in ASME paper ICES2006-1327, "Friction Reduction by Piston Ring Pack Modifications of a Lean-Burn 4-Stroke Natural Gas Engine: Experimental Results", by Kris Quillen, Rudolf Stanglmaier, Luke Moughon, Rosalind Takata, Victor Wong, Ed Reinbold, and Rick Donahue [4]. Friction reduction is a very cost effective way of improving engine efficiency and yields other benefits such as longer engine life all without increasing engine first cost or increasing emissions. This work had both a computational and experimental approach to modifying the ring pack to reduce friction. MIT did the computational simulation of the ring pack and CSU tested the results using a 6 cylinder version of the VGF engine. Experimental results showed that friction could be reduced, but also that lube oil consumption was compromised.

Additional cooperative research with CSU & MIT was undertaken to model and experimentally verify the opportunity for friction reduction by changing lubricating oil properties. Researchers at ExxonMobil were added to provide research guidance and test samples. This research is detailed in ASME paper JRC/ICE2007-40128, "Friction Reduction due to Lubrication Oil Changes in a Lean-Burn 4-Stroke Natural Gas Engine: Experimental Results", by Kris Quillen, Rudolf Stanglmaier, Victor Wong, Ed Reinbold, Rick Donahue, Kathleen Tellier, and Vincent Carey [5]. The results of this test showed that lower viscosity oil with better base stock could reduce engine friction by about 16.5% resulting in an efficiency improvement of about 1.6%.

While the results of this research showed very promising results for efficiency, further work was needed relative to reliability, durability, and maintenance costs. Ring pack friction reduction techniques were implemented in the production APG1000 engine, but these changes were limited in scope since long term durability testing could not be completed in-time to use the results of this research for the production APG1000 engine. However, with such promising results this research was further pursued in ARES Phase II.

Making a practical high efficiency engine involves verifying that the changes to the engine will not have any adverse consequences for the durability and the life cycle costs of the engine. One concern was whether spark plug technology would be good enough as we moved to higher BMEP and peak cylinder pressure. With that in mind the team researched the possibility of replacing the spark ignition system with a Diesel micropilot ignition system. Diesel micropilot uses a small Diesel fuel injector in place of the spark plug to ignite the main natural gas charge.

The Diesel fuel quantity is typically in the range of 0.5 – 2% of the main natural gas flow on a power basis. A 6 cylinder version of the VGF engine was modified to replace the spark ignition system with a Diesel micropilot ignition system. Since commercially available Diesel micropilot ignition systems are not available the team worked with manufacturers to procure automotive sized Diesel fuel injectors, and a prototype injector driver box. Engine testing revealed that Diesel micropilot ignition worked well at high load, but had difficulty starting the engine. In addition, the NOx emissions from the engine increased over a comparable engine with spark ignition. Data from the Diesel fuel injection manufacturer indicated that they would only have a lifetime of about 8000 hours and their replacement cost was considerably higher than spark plugs. The increase in lifecycle costs, the engine starting difficulties, and the higher NOx emissions convinced the team to terminate further development of Diesel micropilot ignition.

With the disappointing results of Diesel micropilot ignition it was clear that new spark plug technology was needed. Waukesha worked with our spark plug supplier and developed an Iridium-Iridium spark plug that offered better life with only a small cost increase. Figure 4 shows a picture of the spark plug and a key feature on the center electrode that helps decrease operating spark plug voltage which leads to longer sparkplug life. This sparkplug has been a commercial success and has been applied across all of Waukesha's lean burn engines. This sparkplug has been one of Waukesha's highest volume service parts since it was introduced in 2003.

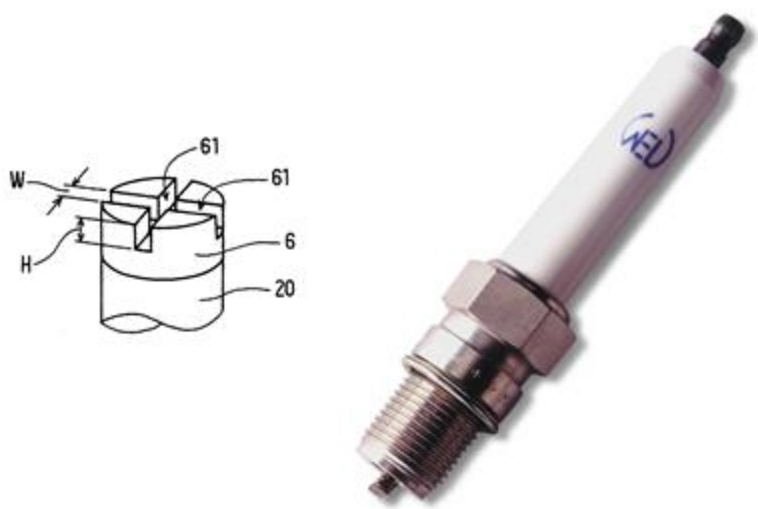

Figure 4 Iridium - Iridium spark plug with cross cut on center electrode to reduce maximum sparkplug voltage requirement

ARES Phase I Summary

ARES Phase I resulted in a large step change improvement in gas engine efficiency and competitiveness. Table 3 shows a comparison between the APG1000 engine and the VGF P48 engine that it was based on. NOx emissions were cut in half, efficiency improved by 14.6%, and power output increased by almost 30% which helped to reduce first and operating costs of the APG1000 compared to the VGF engine it replaced.

Parameter	VGF P48 Value	APG1000 Value	% Change
Bore	152 mm	152 mm	-
Stroke	165 mm	165 mm	-
V angle	60°	60°	-
Number of cylinders	16	16	-
Displacement	48 l	48 l	-
Power output @ 1800 RPM	880 kW	1,142 kW	+29.7%
NOx emissions	2.0 g/bhp-hr	1 g/bhp-hr (0.1 w/ aftertreatment)	-50%
CO emissions	1.34 g/bhp-hr	1.5 g/bhp-hr	+12%
THC emissions	1.61 g/bhp-hr	2.2 g/bhp-hr	+37.5%
Efficiency	36.3% BTE	41.6% BTE	+14.6%
Fuel consumption	2402 kW	2746 kW	+14.3%
Heat rejection to coolant	617 kW	491 kW	-20.4%
Heat rejection to aux. water circuit	228 kW	268 kW	+17%
Heat rejection to exhaust	665 kW	700 kW	+5.2%
Exhaust temperature	448 °C	407 °C	

Table 3 Specifications for the Waukesha VGF P48 base engine and the APG1000 ARES Phase I engine

Additional APG1000 Activities Outside of ARES Phase I

ARES Phase I was focused on the domestic 60 Hz. market. Most of the rest of the world operates at 50 Hz. Using our own funding we developed a 50 Hz. variant of the APG1000 for the world market. The 50 Hz. variant retained commonality with the 60 Hz. engine with only the turbocharger, carburetor insert, and the engine control calibration different between the two models. That made it simple to run both engines down the same assembly line. The larger 50 Hz. market keeps the engine volumes up reducing cost compared to a 60 Hz. only solution.

ARES Phase I was restricted to natural gas only as the fuel. Natural gas fueled engines can, with modifications, burn other gaseous fuels such as biogas that is formed in landfills, wastewater treatment plants, and agricultural digesters. Biogas is roughly a 50-50 mixture of methane and CO2. Biogas is a potent greenhouse gas (GHG) due to its methane content and thus it is advisable that it be burned. Further GHG reduction is possible by burning the biogas in an engine and producing electricity to offset power production from coal. Again, using our own funding, we developed a biogas variant of the APG1000 engine. This involved a new fuel control valve that has the range and accuracy to deal with the larger variation in fuel quality that is typical of biogas, a new control module for the control valve, a larger regulator and fuel mixer to deal with the larger volume flow rate of fuel, and a new engine control calibration. The APG1000 biogas engine has been publically announced and will be for sale in 2012.

ARES class engines are typically used in combined heat and power (CHP) applications where in addition to electricity; the heat from the engine is used in industrial processes. Sometimes the heat is used as hot water to provide heating for digesters or space heating, and sometimes the heat is used as low grade steam. A common business model is for the engine manufacturer to sell the engine to a packager, who will mount the engine to a base, and install the electric generator, switch gear, generator controls, and CHP heat exchangers. The problem with this business model is that should problems arise there is finger pointing between the customer, packager, and engine manufacturer as to who is responsible. Once again Waukesha invested our own development money to produce a complete engine + generator + switch gear + heat exchanger CHP package. Figure 5 is a rendition of the CHP package.

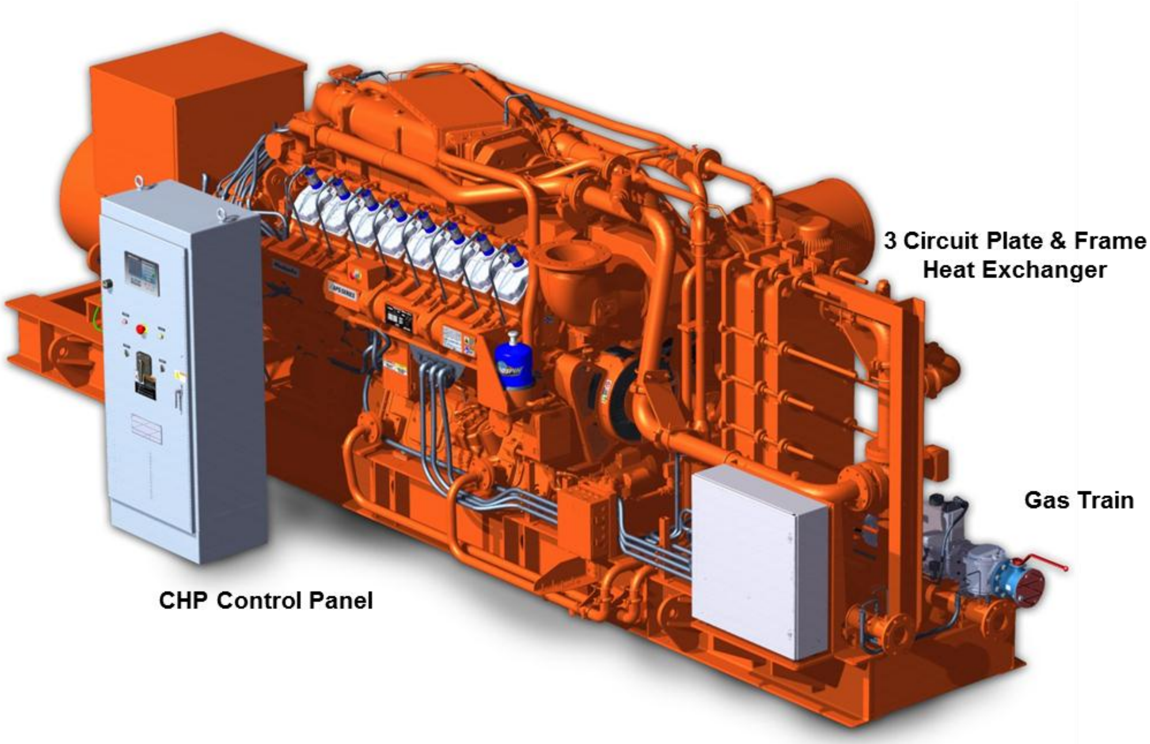

Figure 5 Rendition of complete CHP package

The CHP package features a total efficiency (electrical and thermal) of up to 89.4%. The package was also designed to support different heat usage scenarios for different CHP applications. Simple changes using jumper tubes can be used to change the distribution of heat at different temperature levels. Full production of the CHP package started in 2011 with 5 sold, one of which was biogas fueled.

ARES Phase II

Waukesha started ARES Phase II in 2005 upon successful completion of Phase I. The original plan was to use the newly acquired APG2000 product (2 MW) as the basis to develop a commercially viable 47% BTE ARES Phase II engine. The funding uncertainty caused by the moving of ARES to a different DOE division caused us to eliminate commercialization activities and focus on technology development for Phase II.

Friction Reduction

Friction reduction is a straightforward method of improving engine efficiency and it comes with other benefits. Figure 6 shows typical energy distribution in a large engine. Note that Heat Loss is the single largest portion of the energy distribution pie. It might seem that reducing heat losses should be the most important task for improving engine efficiency. Unfortunately the heat loss is at relatively low temperature and thus a second law analysis indicates that the available energy is much lower than the 49.2% indicated in Figure 6.

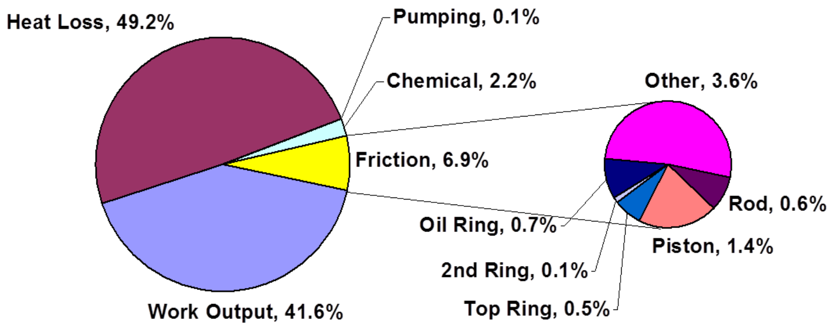

Figure 6 Typical energy distribution for a large engine.

Friction is shown as about 7% of the incoming energy. Any friction reduction goes directly into increased engine power output and thus has an availability of 100%. The secondary benefits of friction reduction are just as important to customers as the BTE improvement. Some of the secondary benefits include longer engine component life, longer oil life, minimal initial engine cost increase, and no increase in emissions. The two largest sources of friction in a large engine are the power cylinder which includes the piston, rings and rod, and the bearings. Waukesha developed technology to reduce friction from both the power cylinder and the bearings.

Low Friction Oil

As mentioned above during ARES Phase I Waukesha worked with MIT, CSU, and ExxonMobil to investigate low friction oil. Initial results were promising but more work was necessary to commercialize the technology. Working directly with ExxonMobil Waukesha began a program to test and validate low friction oil. The initial testing had been done using a 6 cylinder VGF engine at CSU. Since it was less expensive to test on a smaller engine Waukesha also used a 6 cylinder VGF engine to test different oils. Multiple 200 hour tests were performed with the baseline oil and two low friction candidates. This round of testing validated the performance improvement seen before and also allowed the team to show that the low friction oils actually had lower oil consumption than the baseline oil. Since this new low friction oil is more expensive than the baseline oil, the lowest friction oil of the two candidates was chosen for further testing. Longer 1000 hour tests were performed on the VGF engine using both the baseline and low friction oil. Wearing parts were measured before and after the test to determine wear rates. Bearing simulations were performed to ensure that the low friction oil would not adversely affect bearing life. At the end of each test the parts were removed from the engine, measured and sent to ExxonMobil for deposit formation rating using the industry standard CRC Deposit Rating Method. Figure 7 shows the piston pictures for baseline and low friction oil after the 1000 hour endurance test. In addition to the pistons, the valves, bearings, liners were also rated. During the 1000 hour test oil samples were taken and analyzed for degradation and the presence of wear metals. ExxonMobil also ran 300 hour highly accelerated life tests on the baseline and low friction oil. All test results came back favorable.

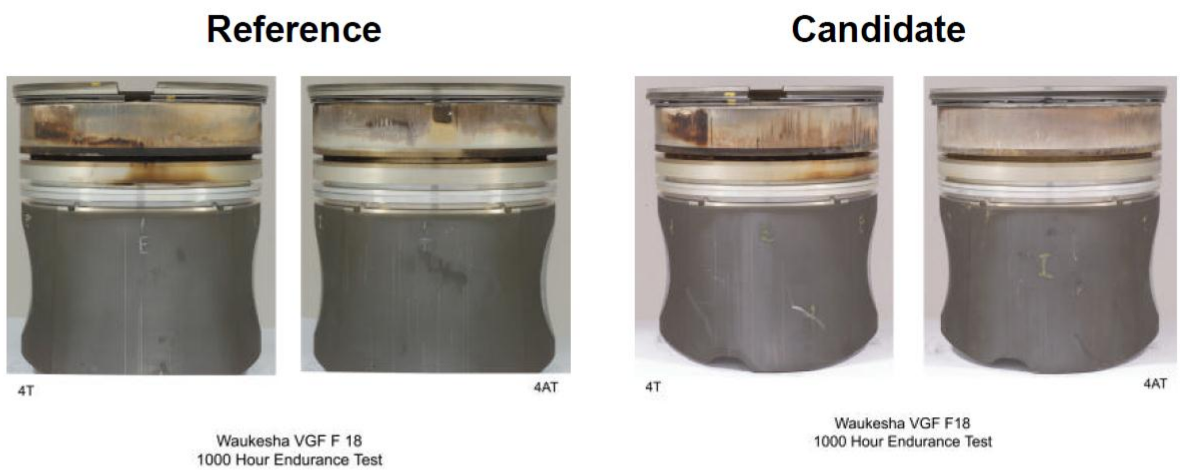

Figure 7 Piston pictures taken after 1000 hour endurance test.

Once the team was fairly certain that the low friction oil would not result in any durability, wear or deposit formation issues, testing started on the APG1000 engine. More careful tests were performed to obtain a statistically significant measurement of BTE improvement. This involved shorter 200 hour tests using humidity controlled inlet air. The measured BTE improvement was statistically significant and repeatable under controlled laboratory conditions. Finally a 1200 hour endurance test was performed with both baseline and low friction oil using the production APG1000 engine. Again components were measured for wear before and after the testing and the deposit formation was rated by ExxonMobil. One beneficial difference noted during testing is that oil consumption of the low friction oil is about 20% lower than the baseline oil. The final step towards qualifying the low friction oil for use in the APG1000 engine is a 4000 hour field test. The field test started November 2010 with the gathering of baseline performance data. In January 2011 the low friction oil was installed in the engine. There are now 3800 hours on the low friction oil with no oil change compared to the baseline oil change interval of 1500 hours. Endurance testing with the low friction oil has continued on the lab engine, we now have 4500 hours on the low friction oil with no oil or filter changes. Outside of the ARES program, ExxonMobil field testing has demonstrated over 16,000 engine hours with no oil change. In January 2010 ExxonMobil announced the commercial availability of Mobil SHC Pegasus. Mobil SHC Pegasus is the first natural gas engine oil that lowers fuel consumption.

Friction Reduction

From the collaborative work with MIT & CSU the team realized that there were significant opportunities to reduce power cylinder friction. The APG1000 power cylinder was designed using an old fashioned experimental “cut-and-try” method to reduce power cylinder friction. While the team was able to reduce friction over the VGF power cylinder, it was clear that the problem was so complicated that the only way to develop a fully optimized design would be to use modern computational tools and to use a Design of Experiments (DoE) approach.

Specialized software to simulate power cylinder performance was purchased. The software simulates lubrication and wear of the piston, rings, and liner. Figure 8 shows the components of the power cylinder.

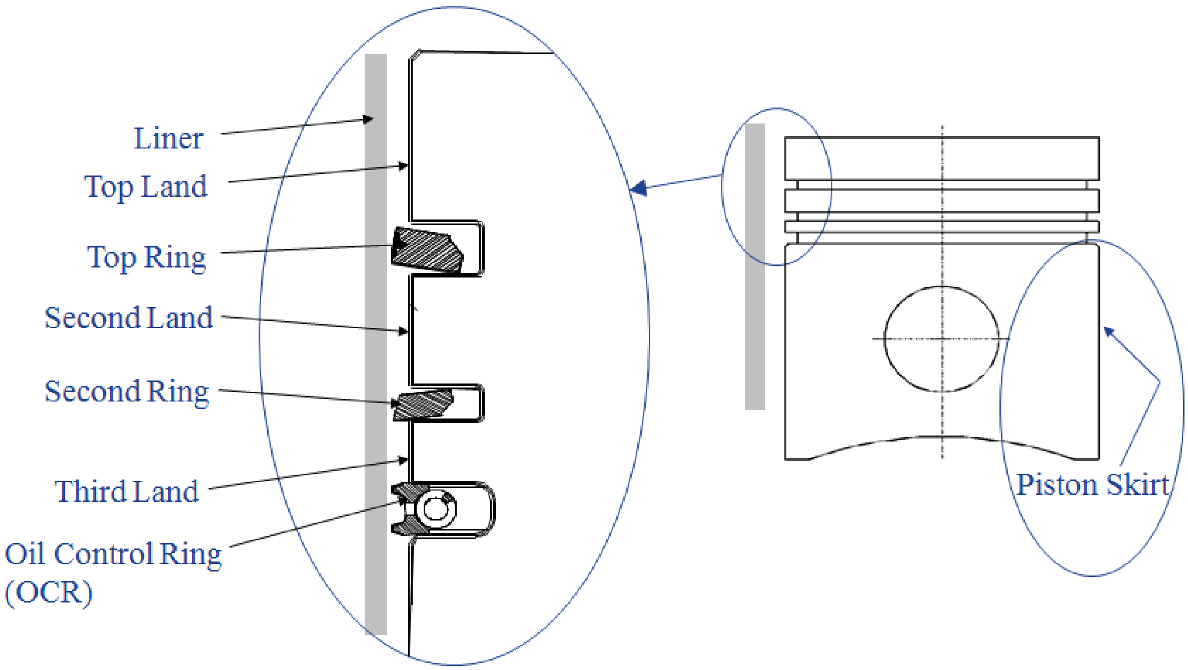

Figure 8 Components of the power cylinder.

Figure 8 does convey some of the complexity of the power cylinder, but one fact highlights this: there are over 300 different geometric parameters that can be independently changed. That level of complexity poses a problem for conventional DoE analysis. Most DoE software is limited to fewer than 50 parameters and the time necessary to perform a conventional DoE experiment of over 300 parameters makes the problem impossible to optimize with existing computer hardware. To simplify the problem several sensitivity analyses were performed. A single-factor-at-a-time analysis using blueprint tolerances was performed. The factors were ranked against the performance metrics of interest. That analysis was repeated using estimates of possible maximum ranges and once again the factors were ranked against the performance metrics of interest. It was possible to isolate fewer than 40 factors that had a strong influence on the performance metrics. Some factors affected single performance metrics, while others affected multiple performance metrics. Figure 9 shows an example of the ranking of different factors against one performance metric.

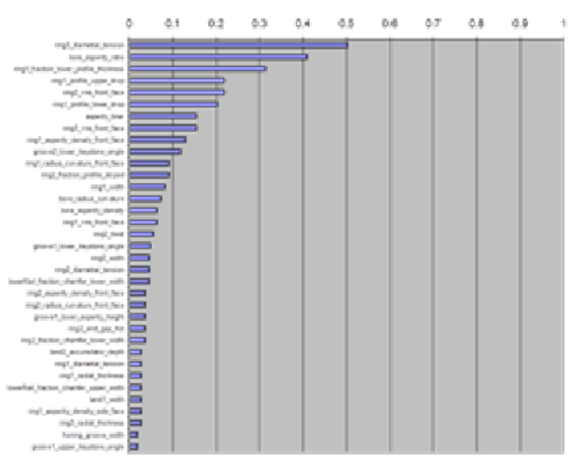

Figure 9 Example of ranking different factors against one performance metric.

This method of simplifying the problem complexity is not ideal since information on interactions between factors is lost, but it does produce a solution that is significantly better than the old fashioned experimental cut-and-try method.

A further complicating factor is that computer simulation time is significant. DoE simulations for one part of the power cylinder took 4 months of time using two multi-CPU and multi-core computers. Detailed experimental validation of the models was not possible given our resources, but it is possible to compare final results. It is quite difficult to measure actual power cylinder friction in a production engine. The team compared predicted performance metrics against actual performance metrics. Table 4 shows predicted versus actual performance improvements over the production APG1000 power cylinder.

Performance Parameter	**Predicted improvement over baseline APG1000**	**Actual improvement over baseline APG1000**	**Comment**
Friction	48% lower	Approximately 50% lower	BTE improvement used to estimate actual friction reduction.
Oil Consumption	60% lower	37% lower	Lower oil consumption leads to lower operating cost for the customer.
Blowby	26% lower	27% lower	Lower blowby increases oil change interval, lowering operating cost for the customer.
Wear Rate	40 – 90% lower	Unknown	Lower wear rate leads to longer time between engine overhaul, lower operating cost for the customer. Low hour wear rate measurements show no wear. Longer testing will be necessary.
Unburned fuel emissions	10%	0%	Lower unburned fuel emissions improve engine efficiency and reduce the level of some pollutants.

Table 4 Predicted and actual performance improvement over production APG1000 power cylinder

The results in Table 4 are outstanding especially when considering that the changes to the power cylinder added much less than 1% to the cost of the complete engine. Most industrial engine customers will be willing to spend more for the engine if the break even time from lower operating cost is less than about 1 year. This technology reduces the break even time to less than a quarter year. US patents 7,506,575, 7,493,850, 7,302,884, and 7,293,497 cover some of the technology developed in the power cylinder optimization task. This is game changing technology that is widely applicable to all natural gas engines, Diesel engines, automotive engines, and even reciprocating compressors.

Ultra Low Emissions Technology

All ARES class engines are lean burn, meaning that there is excess air in the exhaust. While lean burn technology allows high BMEP operation, high efficiency, and low engine-out NOx emissions, should even lower NOx be desired the aftertreatment solutions available are expensive. Selective Catalytic Reduction (SCR) is the most common NOx aftertreatment solution for lean burn engines. SCR requires a reductant be added to the exhaust stream to reduce NOx. The most common reductant is an aqueous solution of Urea. NOx reduction with an SCR system is limited by chemical equilibrium considerations to about 90%. Higher reduction of NOx is only possible by overdosing with Urea which leads to ammonia slip, higher operating cost, and is illegal in some locations. SCR only reduces NOx, and does nothing to reduce CO, Methane, or other hydrocarbons. Finally, the cost of Urea is approximately 1-3% of the fuel cost.

Exhaust Gas Recirculation (EGR) is a common emission control technique that was commercialized for automotive use in the early 1970's, and for heavy-duty on-highway Diesel engines in 2003. EGR involves recirculating a portion of the exhaust flow back into the engine along with the combustion air and fuel. EGR functions as a diluent, just like the excess air in a lean burn engine, but unlike lean burn there is no oxygen in the exhaust stream. The lack of oxygen in the exhaust stream allows the use of highly effective, inexpensive Non-Selective Catalytic Reduction (NSCR) aftertreatment just like the catalysts used in automotive applications. NSCR catalysts simultaneously reduce NOx, CO, Methane, and other hydrocarbons in the exhaust.

The APG1000 lean burn engine was modified for stoichiometric cooled EGR operation. A number of new components were designed and procured including an EGR mixer, EGR cooler, turbine outlet, and an EGR control system. Initial testing showed similar engine efficiency to the lean burn APG1000 engine when the EGR rate was optimized. Several beneficial differences between the EGR and lean burn engines were noted:

1) The compressor pressure ratio was lower; about 2.9 compared to 3.8 for the lean burn.
2) Combustion air flow decreased by about 40% allowing the use of a smaller air cleaner.
3) Exhaust temperature was about 100 °C higher than the lean burn.
4) Unburned Methane emissions were reduced.

Some problems were also noted which the team actively addressed.

A new EGR heat exchanger was designed that removed most of the heat at the highest possible temperature. The design was more compact than the original design and could also be mounted in front of the engine. In 4500 hours of endurance testing the EGR heat exchanger performed flawlessly.

One exhaust component failed due to an incorrect material choice for the higher exhaust temperature. That component was replaced with one made from an appropriate material.

Endurance testing of the EGR engine for 4500 hours showed comparable power cylinder wear and valve recession compared to the production lean burn engine.

A key activity was developing a reliable EGR control algorithm. After exhausting a number of ideas, the invention that became US patent 8,108,128, "Controlling Exhaust Gas Recirculation", by James Zurlo, Kevin Konkle, and Andrew May was developed. The invention uses existing sensors to control EGR flow rate. The sensors typically last the life of the engine so no additional maintenance is associated with this EGR control methodology.

The EGR engine was mated to an NSCR catalyst and the emissions performance was tested over an 1800 hour interval. Figure 10 shows engine out and catalyst out NOx emissions.

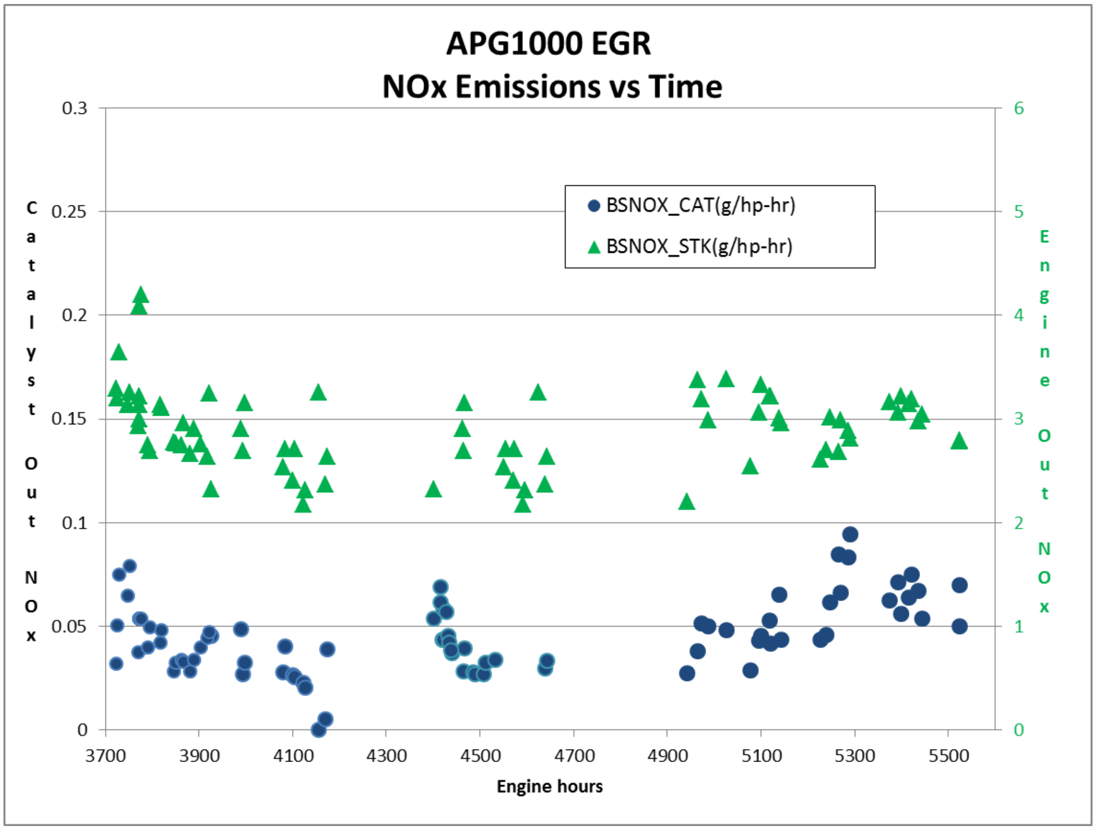

Figure 10 NOx emissions versus time.

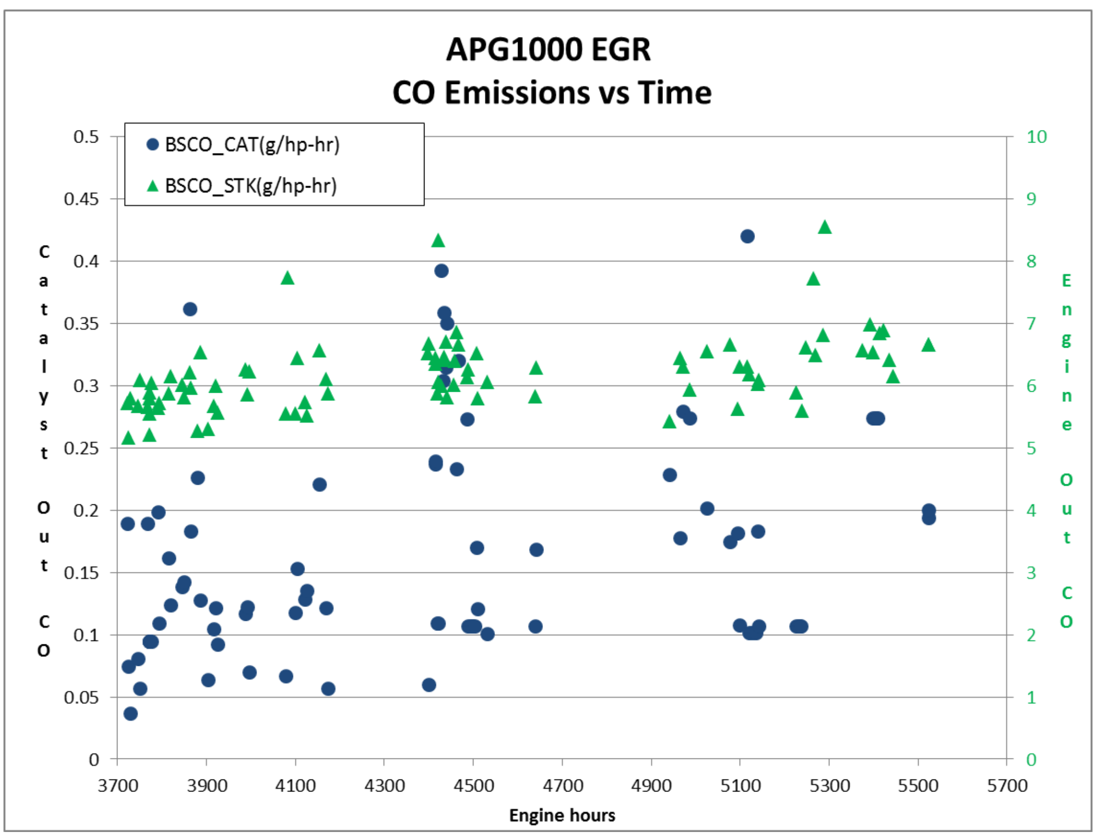

Figure 11 CO emissions versus time

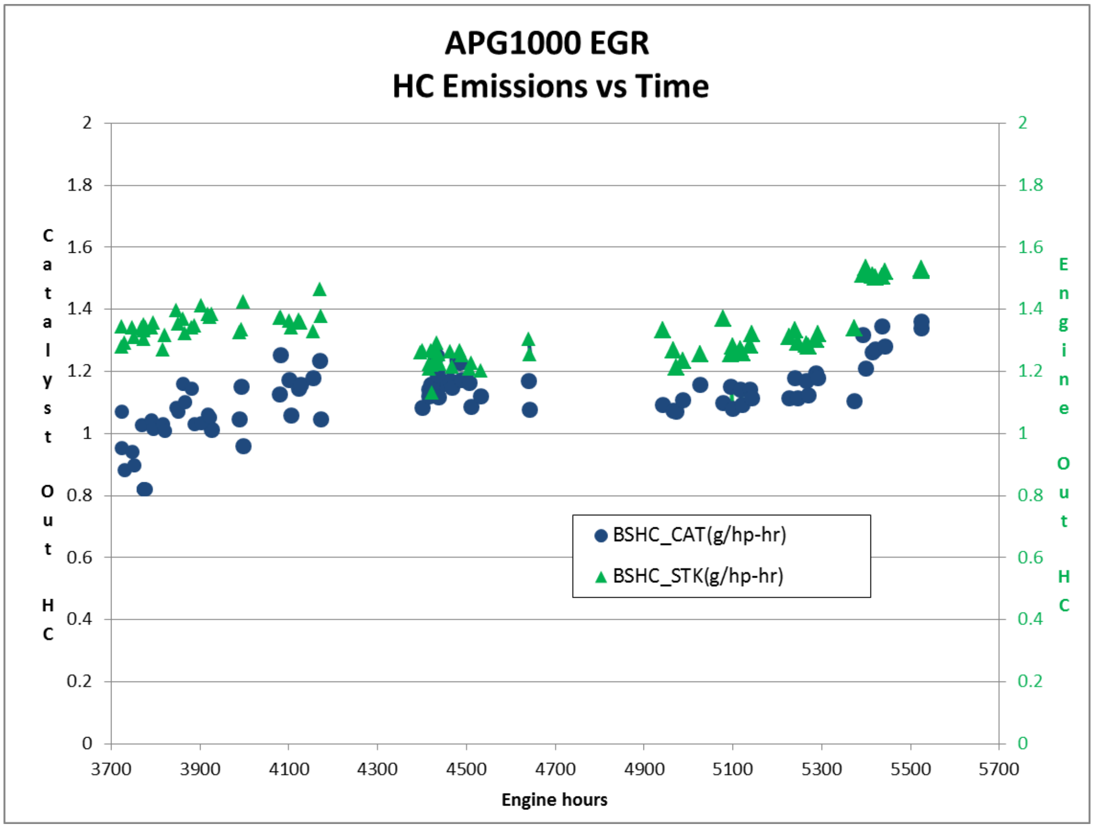

Figure 12 Unburned Hydrocarbon emissions versus time

Table 5 summarizes the emissions performance of the EGR engine. Understanding that this was still a fairly raw prototype the emissions performance was outstanding. Table 6 shows the EGR engine emissions compared to the production lean burn APG1000 engine emissions. The APG1000 EGR emissions place it among the lowest emissions of any reciprocating engine and roughly equivalent to the best gas turbine emission levels. Further emission reductions are possible with a more expensive catalyst and more engine and control optimization.

Pollutant	Avg. Engine out	Avg. Catalyst out	% reduction
NOx [g/bhp-hr]	2.98	0.056	98.1
CO [g/bhp-hr]	6.47	0.53	91.8
Hydrocarbons [g/bhp-hr]	1.36	1.15	15.3

Table 5 Summary of EGR engine emissions performance

Pollutant	APG1000 lean burn	APG1000 EGR	% reduction
NOx [g/bhp-hr]	1.0	0.056	94.4
CO [g/bhp-hr]	1.5	0.53	64.6
Hydrocarbons [g/bhp-hr]	2.2	1.15	47.8

Table 6 Comparison of production APG1000 lean burn emissions to APG1000 EGR catalyst out emissions

Pre Phase III Work

In preparation for Phase III an APG2000 engine was purchased. The APG2000 is a larger engine than the 1.1 MW APG1000. The 12 cylinder APG2000 is rated at about 2 MW and there is an 18 cylinder variant, the APG3000, which is rated at about 3 MW. Table 7 shows the differences between the APG1000, APG2000, and APG3000 engines.

	APG1000	**APG2000**	**APG3000**
Bore [mm]	152	220	220
Stroke [mm]	165	240	240
Number of cylinders	16	12	18
Power at 60 Hz. [MW]	1.1	1.86	2.8
BTE per ISO 3046/1 at 60 Hz. [%]	42.6	44.1	44.1

Table 7 Comparison between the APG1000, APG2000, and APG3000 engines

Note that the larger APG2000 and APG3000 engines have a higher efficiency than the smaller APG1000. Larger engines are more efficient than smaller engines due to reduced in-cylinder heat transfer loss, availability of higher efficiency turbochargers, and a reduction in in-cylinder crevice volume in relation to cylinder volume [6]. The inherent efficiency advantage of a larger engine is why the original plan called for using the APG2000 engine as our platform for demonstrating the ARES Phase III 50% BTE goal.

The first task was to apply ARES Phase I technologies to the engine. An optimized valve event camshaft featuring early closing Miller intake valve timing and reduced valve overlap was procured. The production APG2000 engine features intake port fuel injection that requires a fuel supply pressure of at least 60 psig. Such high pressure fuel is frequently not available, so the next task was to design and procure a new low pressure fuel system. In addition, the method of engine power control had to be changed from port open duration to a butterfly throttle valve.

Using Waukesha funding the team purchased a single cylinder version of the APG2000 engine. Simulation of the combustion was performed in an attempt to develop a non-fueled prechamber combustion system similar to the APG1000 combustion system. New pistons and prechambers were purchased that were similar in design to the APG1000 components. Waukesha suffered several mechanical setbacks that prevented proper operation of the single cylinder engine.

Next the cylinder bore was reduced from 220 mm to 200 mm. The reduction of bore reduces in-cylinder heat transfer and allows for higher peak cylinder pressure both of which improve engine efficiency. New 200 mm bore pistons and liners were designed and procured. Since new pistons were needed the team also took the opportunity to reduce the crevice volume above the top ring. The reduced crevice volume reduces unburned methane slip, improving efficiency and reducing emissions. The reduced bore size changed the ratio of prechamber to combustion chamber volume, so new smaller prechambers were designed and procured.

Efficiency is not the only criteria for a successful engine design; it also must be cost effective to purchase. In support of the objective to reduce the cost of electricity by 10% a major cost reduction effort was undertaken. Of the 56 total groups or subsystems in the engine, components in 26 different groups were cost reduced. This resulted in about a 10% reduction in engine cost.

Phase II APG1000 Demonstration Engine

Combining the technology developed during Phase II with the Phase I APG1000 engine allowed us to demonstrate the Phase II goal of 47% BTE using just the engine and without any exhaust energy recovery system, preserving the exhaust energy for Combined Heat & Power applications. The team started by reducing engine speed to 1200 RPM (60 Hz.). Reduced engine speed reduces friction and pumping losses. A newly released high efficiency turbocharger was installed on the engine. New low friction power cylinder components were designed, procured, and installed. The piston compression ratio was increased over the production APG1000 engine. Finally the low friction oil was installed in the engine. The engine was run at higher BMEP which also improves engine efficiency. Figure 13 shows the path taken to obtain 47% BTE. The combination of a higher efficiency turbocharger and 1200 RPM operation brought the efficiency to 45.2%. Adding the low friction oil and the low friction power cylinder components brought the engine efficiency up to 46.2%. Data indicated that the combination of higher efficiency turbocharger and 1200 RPM operation allowed more unburned fuel to slip into the exhaust un-combusted during the valve overlap period. The camshaft valve events had been optimized for 1800 RPM operation and needed to be re-optimized for 1200 RPM operation. There was insufficient time to design and procure a re-optimized 1200 RPM camshaft so as a test to verify that un-burned fuel was slipping straight into the exhaust during the valve overlap period the valve lash was increased. Increasing the valve lash is not an optimal solution but it did confirm the problem and reduced the unburned Methane emissions and increased the BTE to 46.6%. Finally, a new set of higher gap spark plugs was installed, which brought the efficiency to 47%.

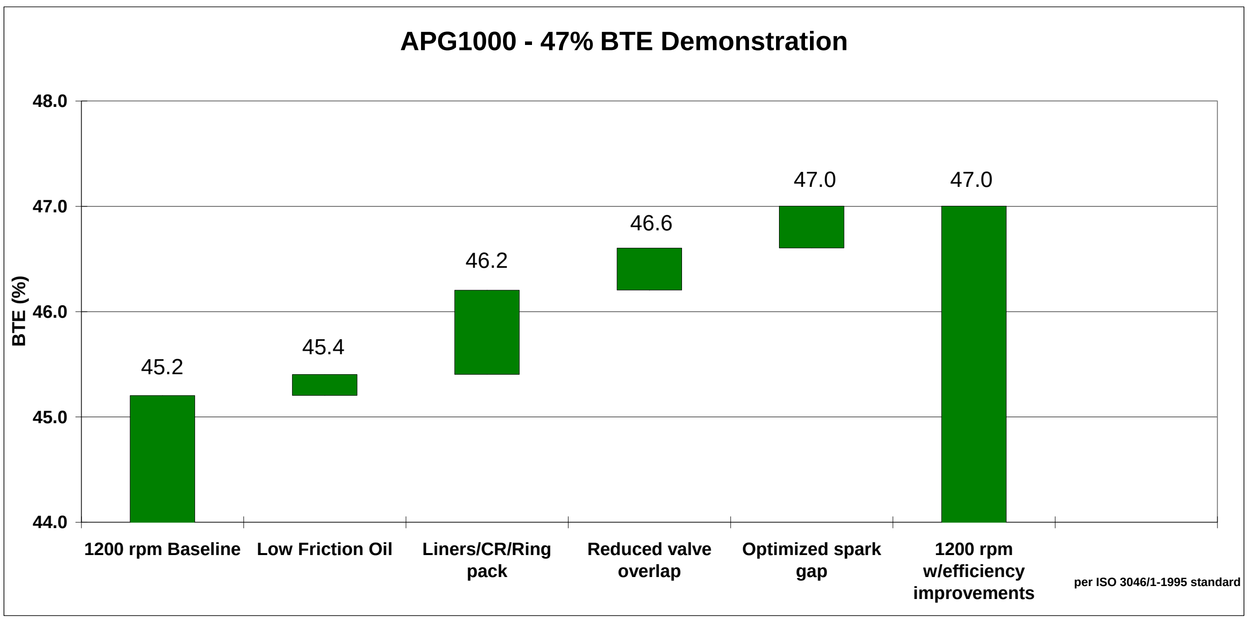

Figure 13 the path taken to demonstrate 47% BTE

Had the team taken the time to re-optimize the camshaft valve events for 1200 RPM operation even further efficiency improvements would have been demonstrated.

Figure 14 summarizes graphically the energy distribution for the production APG1000, APG1000EGR, and 47% BTE demonstration engines. As mentioned previously measuring friction reduction in a multi-cylinder engine is difficult. In the case of the 47% BTE engine the reduced friction was clearly revealed as a reduction in heat rejection to the lube oil. The oil sump temperature dropped by 10°F which could help the oil last longer. Another pleasant surprise with the performance of the 47% BTE engine was that there was only a small drop in exhaust temperature. Many technologies that improve engine efficiency do so at the expense of exhaust temperature. By retaining exhaust temperature the engine can be used in CHP applications and the performance of exhaust aftertreatment systems is maintained. Note the increased unburned Methane emissions for the 47% BTE engine even with the temporarily reduced valve overlap. That represents an efficiency loss that would have been captured with a re-optimized 1200 RPM camshaft.

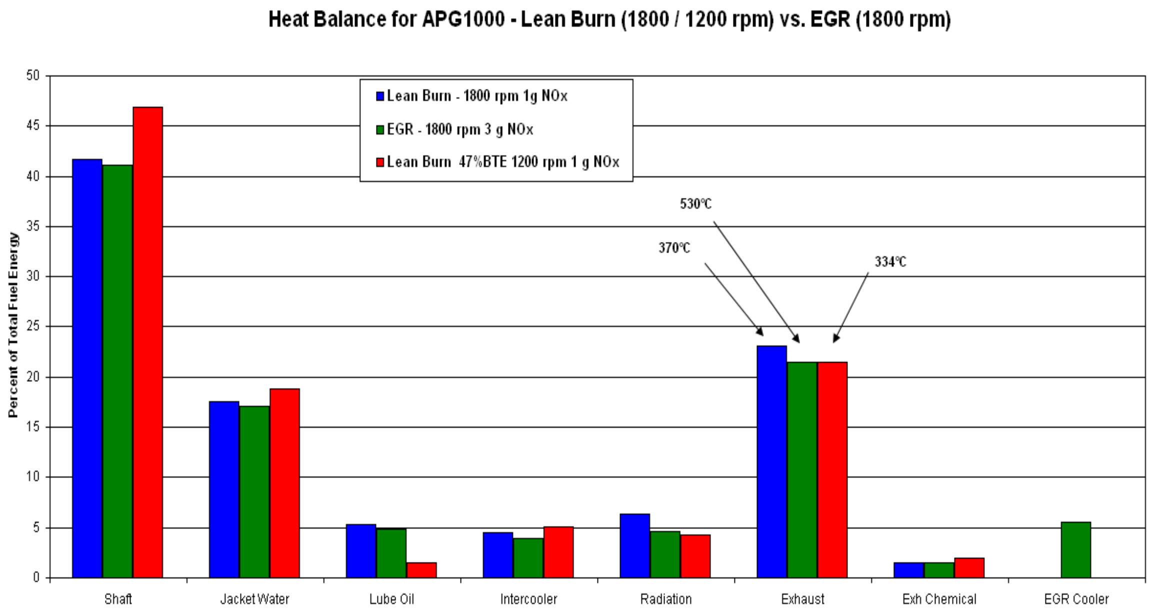

Figure 14 Energy balance for the APG1000, APG1000EGR, and 47% BTE Demonstration Engines

A single cycle efficiency w/o exhaust heat recovery of 47% puts the engine in a rarefied crowd. Of commercially available reciprocating natural gas engines, the Wärtsilä 9.7 MW 20V34SG has an efficiency of 47.7% and the Wärtsilä 18 MW 18V50SG has an efficiency of 50.1%. The GE 9.5 MW J920 has an efficiency of 50.2%. The Mitsubishi 5.75 MW KU30GSI has an efficiency of 48.7%. The Mitsubishi KU30GSI is the resulting product from a joint Mitsubishi - Japanese government NEDO program to increase natural gas engine efficiency. The GE 4.4 MW J624 engine has an efficiency of 47.9%. No other gas engines have higher than 47% efficiency and certainly no other 800 kW natural gas engine has such a high efficiency. Since the technology developed during ARES Phase II is applicable to all engines it is quite likely that we will see single cycle efficiency increase beyond 51% for the largest natural gas engines over the next decade.

ARES Phase III

Waukesha started ARES Phase III in 2010 upon successful completion of Phase II. The original plan was to use the newly acquired APG2000 product (2 MW) as the basis to develop a commercially viable 50% BTE ARES Phase III engine.

Phase III Technology Development

Recalling the friction distribution shown in Figure 6, it is clear that while the team had excellent results with our power cylinder optimization technology, there were additional friction reduction opportunities. Therefore the team decided to explore the other engine subsystems for friction reduction potential.

Lube System Modeling

A lube oil system model was developed. The lube oil system model consists of the oil pump, filters, pressure relief valve, interconnecting pipes, bearings, piston cooling jets, and valve train. The model predicts oil flow, pressure, temperature, and bearing power. The first surprising result is that unlike automotive engines, gas engine valve train friction losses are negligible. No further effort was put towards the valve train. The main bearings were responsible for slightly over half the lube system power loss with the remaining power loss comes from the connecting rod big end bearings and the power to drive the oil pump. The results for flow showed that over 50% of the total oil flow was through the main bearings, less than 10% through the connecting rod big end bearings and the bulk of the remainder through the oil cooling jets. Oil flow through the bearings provides cooling for the bearings. The oil properties were varied to simulate the low friction oil and it was demonstrated that a 10% drop in bearing power could be obtained with only a 2% increase in oil flow demand using the low friction oil compared to the standard oil.

This modeling provided guidance on opportunities to modify the lube oil system to reduce friction.

Rolling Element Bearing Modeling

There is significant effort to investigate the usage of rolling element bearings for automotive use. A quick literature review showed that rolling element bearings have the potential to significantly reduce friction when compared to standard journal bearings. Therefore the team decided to investigate whether rolling element bearings could be used in industrial gas engines. SKF was contracted to build a simplified model of the APG2000 engine main and connecting rod bearings. Modeling results showed that the bearing friction was actually higher and the predicted life significantly lower than desired. Given no space constraints to fit in place of the existing journal bearings the friction results were not much better. The continual acceleration and deceleration of the roller elements when used in the connecting rod big end bearing meant that larger diameter rollers would have a tendency to skid. Longer rollers to help support the load were also problematic since minor bearing distortion leads to higher friction and wear from roller skewing. It was clear that the highly loaded bearings of an industrial natural gas engine are sufficiently different than the automotive situation to make rolling element bearings not viable for this type of application.

APG1000 Power Cylinder Improvement Productionization

While ARES Phase II did not result in a new product it was important to continue to improve the APG1000 engine using the technology developed in Phase II. The optimized power cylinder components from the 47% BTE demonstration engine were tested further to ensure robustness and readiness for a field test. Several problems were found and the optimization was redone targeting increased robustness at the problem operating points while still maintaining the desirable performance characteristics. Further simulation found several solutions that seemed to meet all criteria. Laboratory testing was initiated and the most robust and best performing solution was chosen for further field testing. As of the time of this report, Waukesha is still searching for a suitable field test site. The proposed power cylinder components will help maintain the leadership of the APG1000 engine as the highest efficiency 1800 RPM natural gas engine.

ARES Phase III Replanning

The acquisition of Dresser, Inc. Waukesha Engine by GE Energy on February 1, 2011 resulted in a reevaluation by management of the scope and the proposed path of this program. The engine platform initially proposed for demonstration is no longer being enhanced. The plan was to utilize another engine platform within the GE Energy Gas Engines portfolio that currently is capable of higher peak cylinder pressures to help meet the cooperative agreement objectives. Numerous program plans and alternatives were created and evaluated. All these alternatives utilized an engine platform that is engineered outside of the USA, so logistics and foreign labor content were a great concern. A viable plan to continue ARES III that would limit foreign labor content to no more than 25% of total labor content could not be constructed. Therefore, GE Energy and the DOE jointly agreed to terminate the cooperative agreement.

Patents

US Patents resulting from this award:

6,883,483	7,044,103
7,293,497	7,302,884
7,493,850	7,506,575
8,108,208	

Publications

Technical publications resulting from this award:

“Development of the Waukesha 16V150LTD Advanced Power Generation Engine”. By E. Reinbold and D. Mather, ASME ICEF2006-1510, 2006

“Optimization of a non-fueled Prechamber Ignition System for a Lean-Burn, Industrial Natural Gas Engine”, by C. Honl, ASME ICEF2004-821, 2004

“Design and Analysis of the Waukesha APG1000 Engine”, by R. Nicoson and J. Knudsen, ASME ICEF2006-1517, 2006

“Friction Reduction by Piston Ring Pack Modifications of a Lean-Burn 4-Stroke Natural Gas Engine: Experimental Results”, by K. Quillen, R. Stanglmaier, L. Moughon, R. Takata, V. Wong, E. Reinbold and R. Donahue, ASME ICES2006-1327, 2006

“Friction Reduction due to Lubrication Oil Changes in a Lean-Burn 4-Stroke Natural Gas Engines: Experimental Results”, by K. Quillen, R. Stanglmaier, V. Wong, E. Reinbold, R. Donahue, K. Tellier and V. Carey, ASME JRC/ICE2007-40128, 2007

“Development of the Dresser Waukesha 16V150LTD Engine for Bio-gas Fuels”, by E. Reinbold and J. von der Ehe, ASME ICES2009-76079, 2009

Trade publications resulting from this award:

“Waukesha’s New Generation of High Efficiency Gas Engines”, by A. Dijks and L. Wilson, POWER-GEN Europe, 2006

“Gaseous Fuel Challenges for Internal Combustion Engines (The new APG Series for biogas)”, by A. Dijks, POWER-GEN Europe, 2007

"Useful Heat from Reciprocating Engines and Temperature Levels", by A. Dijks, POWER-GEN Europe, 2008

"A New Generation of High Efficiency Gas Engines", by A. Dijks, "Electricity 2008", Eilat, Israel, 2008

"Experiences with the New 16V150LTD Gas Engine/APG1000 Generator Set", by A. Dijks, POWER-GEN Europe, 2009

"A New Gas Engine Generator Set", by A. Dijks, Cogeneration & On-Site Power Production magazine, July-August 2009

Award for "One of the World's Best Power Plants", Diesel & Gas Turbine Worldwide magazine, January-February, 2010

ARES Project Summary

The ARES project has helped drive the efficiency of natural gas engines up to levels that were almost unimaginable a decade ago while dramatically cutting NOx emissions. The ARES project helped Waukesha develop the commercially successful APG1000 engine and with Waukesha using its own funding, further expanded the applications for the engine. The technology developed in ARES Phase II has wide applicability to all engine types and will be used to improve the power generation, gas compression, and Diesel engines that General Electric produces. In the next decade, the industry should see natural gas engine efficiency improve to 51%, and emissions driven down to levels that are competitive with the best gas turbines, while maintaining the cost effectiveness and application flexibility that is the traditional strong suit of reciprocating engines.

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