**SECOND STAGE INTERCOOLING USING LNG FOR TURBOCHARGED
HEAVY DUTY ROAD VEHICLES
PHASE I**

FINAL REPORT

to

**Brookhaven National Laboratories
Upton, New York 11973-5000**

Submitted by

**Advanced Technologies Management, Inc.
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September 21, 1999

SECOND STAGE INTERCOOLING USING LNG FOR TURBOCHARGED HEAVY DUTY ROAD VEHICLES

Phase I Report

Introduction

It is well documented in engine performance literature that reduced engine inlet air temperature increases power output and reduces NO_x emissions for both diesel and spark ignited (SI) engines. In addition, reduced inlet temperature increases the knock resistance of SI engines. In that most HD natural gas engines are SI derivatives of diesel engines it is appropriate to evaluate the benefits of reduced engine air temperature through LNG fuel.

This project investigated the “real world” possibilities of a patented process for utilizing the “cold” in LNG to chill engine inlet air. The results support the conclusion that doing so is a practical means to increase engine power and reduce engine-out NO_x.

ATM employed two principal subcontractors for the Phase I project:

- Advanced Mechanical Technology, Inc. (AMTI), Watertown, MA performed well instrumented dynamometer testing of the Cummins “B” 5.9G natural gas test engine removed from a Roadway Express truck tractor after 17,000 miles of service on LNG. The engine inlet air was cooled additionally by the LNG feed in a second stage intercooler. The results were compared to normal operating, baseline conditions (LNG warmed externally). AMTI retained the services and advice of Robert Raymond, recently retired from Thermo Electron and formerly with Cummins Engine. While at Thermo Electron Mr. Raymond managed testing of a Navistar 466 engine under simulated LNG intercooled conditions under the sponsorship of Consolidated Natural Gas Co. A 22% power increase was achieved in that project by increased turbo boost and spark advance schedule, taking advantage of increased knock resistance.
- Senior Consultant Paul Swenson, recently retired Director of Research and Development for Consolidated Natural Gas, designed and oversaw the construction of the LNG/engine air heat transfer apparatus, using a propane intermediate loop. The engine air heat exchanger was fabricated by Dynamic Specialties, Inc., of Mentor, Ohio. The design report is attached.

ATM acknowledges the material assistance provided by Roadway Express, Akron, Ohio in providing the test engine at depreciated cost and also valuable advice. ATM thanks Delphi (GM) for advice on selecting the proper Delphi component for use in the engine air heat exchanger.

A 20% boost on the currently available Cummins B5.9G natural gas engines would raise rated power from 230 hp to 275 hp, which is the rating of their next larger model C8.3G. A 20% boost in the C8.3G engine would raise its rating to 330 hp, greater than the next model, the L10G, rated at 300 hp. With proposed ATM technology engine selection would move one size down, reducing weight, size and cost. There would be a 385 lb. reduction in going from a series C to a series B engine.

Phase I Results

- The test results tabulated in Figure 1 show that the LNG-cooled Second Stage Intercooler reduced engine inlet air temperature by 38°F for a simulated outdoor ambient air temperature of 60°F, and reduced the air temperature by 56°F at 95°F ambient (first stage intercooler air exit temperatures were set at 35°F above simulated ambient). These results are described in the attached AMTI Test Report. In the 60°F day case, the inlet air captured only 70% of the LNG cooling capacity. Without environmental heat leak the air would have been cooled an additional 15°F. The inlet air exit temperature from the second stage intercooler was 20°F cooler than the actual ambient temperature. In the 95°F day case the inlet air captured essentially 100% of the LNG cooling capacity. In this case the heat leak to the LNG just balanced the environmental heat loss from the much warmer inlet air. It will be important to provide increased insulation in future designs to better preserve the available cooling from LNG.
- Torque and power increases ranged from 6% to 10%, all a result of increased engine inlet air density. The engine was run at maximum torque, nominally 1600 rpm. The Cummins engine was tested in its on-the-road condition with its provided turbocharger and electronic control module (ECM). The ECM turned out to be faulty and was replaced with a new unit from Cummins. The engine could not be tested at maximum power because misfiring brought on by the ECM setting a air/fuel ratio in excess of the lean burn limit, which could not be overridden. Increasing turbo boost and more aggressive spark advance schedules to test for increased knock resistance were not possible on the currently programmed Cummins ECM.
- NO_x emissions were reduced to 1.8 gm/HP hr or less at maximum torque by second stage intercooling, even at the 95°F ambient condition. Figure 2 shows the NO_x emissions dependence on inlet temperature. The engine inlet temperature must be below 90°F for NO_x emissions to be below 2 gm/l-IP hr. CO emissions were essentially unaffected by reducing inlet air temperature. Unburned hydrocarbons and CO were adversely affected by operating so close to the lean burn limit.
- The intermediate propane heat exchange loop performed well. The results suggest that a more vehicle-practical unitary heat exchange module can be made, eliminating the pump and utilizing R-22 instead of propane as a heat transfer medium. The space next to the first stage intercooler under the truck tractor hood is adequate to install the unitary module.
- The use of propane as an intermediate heat transfer fluid was discussed at length in person with Roadway Express. Roadway advised that a propane containing vessel and piping would be a major negative in the trucking industry. In addition the distributed heat transfer loop was too complex and vulnerable for use on a HD vehicle. The temperature difference between LNG at one atmosphere pressure and the freeze point of R-22 is negligible. R-22 is nonflammable and is the most widely used refrigerant presently utilized in North America and is widely accepted as environmentally safe.

Constraints Limiting Phase I Results

- Subcontractor AMTI was unable to circumvent the Cummins Engine Control Module (ECM). Therefore, all testing was conducted within the envelope of the engine operating conditions (A/F ratio, spark timing and turbo boost pressure) prescribed by Cummins. This inability was in spite of assistance from the Boston area Cummins service organization, who is familiar with the natural gas 5.9G engine. As a result, the torque and power increases were severely limited to the 6 to 10% range, predicted by engine intake air density increase. These limitations can be bypassed through manual control of all variables affecting engine performance. Southwest Research's manual control module should provide the necessary solution to eliminate these issues for the purposes of future work.
- Allen Wells, formerly HD natural gas engine director at GRI, now with Navistar, advised that the Cummins 5.9G can easily be boosted to 22 psig without over-stressing the "bottom end," which includes main and rod bearings and pistons. The current Cummins 5.9G natural gas engine ECM limits boost to 17 psig. As a result the knock limit was not reached, either on baseline or second stage intercooled conditions. ATM is discussing with Borg Warner Turbocharger Division their providing a replacement unit capable of easily reaching pressure boost and air flow for a 20% power increase believed achievable with second stage intercooling with LNG.
- AMTI reported suspected engine misfiring. "High" total unburned hydrocarbons were measured in the exhaust. The data shown in the AMTI Test Report ranged from 0.25% to 0.30% of exhaust volume. This data supports the existence of misfire. In the Cummins lean burn operation strategy HD natural gas engines are calibrated to operate on the thin margin near to the lean limit to achieve NO_x emissions compliance. When the AT ratio is controlled too near to the lean limit, the engine will misfire. Near the lean limit both UHC and CO emissions begin to rise.
- The LNG storage tanks used by AMTI, were transit bus LNG tanks, obtained from the City of El Paso, TX. The Essex tanks obtained from Roadway Express were too small to be practical. The tanks were self pressurized by heat leak and could only deliver saturated liquid at a vapor pressure sufficient to feed the engine. This fueling strategy sacrifices much of the low enthalpy possible in subcooled LNG. It is anticipated that future tests will be run with well insulated tanks, permitting subcooled storage.
- The composition of the LNG fuel supplied to AMTI as well as the humidity of the local air supply were uncontrolled by active means. The fuel composition will affect knock limit, but this was not a factor in the tests with the current Cummins ECM. Fuel composition and humidity slightly affect emissions. It would be desirable to control these variables in future laboratory testing.

Discussion and Conclusions

- The testing suffered from constraints imposed by the Cummins ECM. The existing ECM protocol is not appropriate for an LNG Second Stage Intercooled engine. Potential for further turbo boost and more aggressive spark advance schedules will be realized in future testing through the use of the Southwest Research Institute manually operated ECM.
- Greater turbo boosting will be needed to reach a 20% increase in engine power output. Future testing will be done with a Borg Warner Turbocharger Division upgraded replacement unit .
- The technology has been proven to offer two large benefits; increased engine power and torque in spite of the Cummins ECM imposed constraints, and reduced NO_x emissions with no increase in CO emissions. The full 20% power increase expected has not yet been demonstrated.

MEASURED DATA		Baseline	2nd Stage Air Cooler	Baseline	2nd Stage Air Cooler
1 st Stage Cooler					
Exit Temperature	°F	95	95	130	130
Actual Ambient	°F		77		76
Engine Torque	ft-lb	411	451	416	441
Engine Seed	rpm	1614	1606	1610	1595
Engine Power	HP	126	138	127	134
Inlet Pressure	"H abs	56	56	57	57
Inlet Temperature	°F	95	60	125	79
Fuel Flow Rate	lb/hr	42.7	45.0	41.4	43.4
Air Flow Rate	lb/hr	1091	1145	1056	1105
Air-Fuel Ratio		25.6	25.4	25.5	25.5
BSFC	lb/HP hr	0.338	0.326	0.325	0.324
Exhaust O ₂	%	8.0	7.8	7.9	7.9
CO ₂	%	7.26	7.32	7.32	7.32
CO (air free)	ppm	1972	2008	1838	1833
Total U H C	%	0.30	0.47	0.25	0.27
NO _x	gm/HP hr	2.05	1.74	2.96	1.82

Figure 1
Second Stage Intercooling Using LNG Test Data

Figure 2
NO_x Emissions – all data

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DUTY ROAD VEHICLES**

TEST REPORT

July 21, 1999

Submitted to

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SECOND STAGE INTERCOOLING USING LNG FOR TURBOCHARGED HEAVY DUTY ROAD VEHICLES

Introduction And Summary

The objective of this project was to demonstrate the feasibility of increasing the power and torque of a natural gas engine fueled by LNG by using the low temperature of LNG to reduce charge-air temperature. Theory predicts that reducing the charge air temperature will increase the air flow and power of an IC engine. Furthermore, on the assumption that the boost pressure of a turbocharged engine is limited by the necessity of avoiding engine knock, cooling the charge air should reduce the peak temperature reached during compression, thereby permitting the boost pressure, and hence the power and torque, to be further increased. Turbocharged gas engines are equipped with an air-cooled charge-air cooler that typically cools the air to about 20 - 40°F above ambient. The addition of a second-stage LNG cooled charge-air cooler can cool the air by an additional 40 to 60°F.

The test engine was a Cummins Model B5.9G six-cylinder 5.9 liter natural gas engine that had been removed from a Roadway tractor after about 17,000 miles of operation on LNG. The engine was fitted with a second-stage charge-air cooler cooled by a closed propane loop, in which propane was circulated between the charge-air cooler, where it was evaporated, and the LNG evaporator, where it was condensed. The intermediate propane loop was utilized to prevent frosting that would occur if moist air was cooled directly by cryogenic LNG.

The test plan was to compare the power and efficiency for the second-stage charge air-cooled case versus a baseline condition in which only a conventional first-stage air-cooled charge-air cooler was used. The comparison was to be made at several engine speeds at wide open throttle at a turbocharger boost pressure determined by the knock limit, and at first-stage air-cooler leaving temperatures of 95°F and 130°F. The boost pressure would be varied by closing the turbine waste gate, which is normally partially open at wide-open throttle. This would allow the engine to operate at a higher boost pressure. The waste gate would be closed in varying degrees until engine knock was observed.

It was not possible to achieve all of the test plan objectives due to several factors beyond our control. These effects were all engine related, and in no way connected with the modified charge air cooling. First, a consistent comparison at the planned test conditions was not possible because of erratic combustion efficiency due to unknown causes. Unpredictable and non-reproducible misfiring occurred at most operating conditions, and especially at high engine speeds. Second, the engine's electronic control module (ECM) limited the boost pressure and controlled the air-fuel ratio and spark timing. This prevented the test from being run at the knock limit.

The tests demonstrated the feasibility of increasing charge-air cooling using LNG with conventional automotive-type heat transfer components, and that the high degree of cooling produced no adverse effects. The performance of the second-stage charge-air cooler and LNG evaporator exceeded their design requirements. The tests indicated that secondary cooling using LNG is capable of reducing charge-air temperatures by 40 to 60°F using commercially available

heat exchangers of moderate size. Because of the erratic engine performance and the inability to reach a knock limit, the performance comparison was only conducted at the peak torque point (1600 RPM), and at constant intake manifold (boost) pressure. Consequently, only the modest increase in torque enabled by cooling the intake air with LNG at constant boost pressure was demonstrated. Demonstration of increased torque and power enabled by increasing the knocklimited boost pressure will require removal or modification of the present Cummins ECM, and possibly upgrading of the engine's turbocharger.

Test Apparatus

Figure 1 shows the schematic of the engine test apparatus. This is located in a 13' x 13' cement-block walled room, which is sound-insulated and equipped with a 5000 CFM exhaust fan. The room is supplied with cooling water from a 1700 gallon water reservoir by a 100 GPM pump. The water provides engine cooling, first-stage charge-air cooling, water-brake supply, and propane vaporization while running in baseline mode. The water from all these devices except the engine coolant heat exchanger empties into a sump. A 3/4 hp pump returns the water to the reservoir via heat exchanger cooled by a closed coolant-loop dissipating its heat to a roofmounted fan-coil.

The cell is equipped with a remotely operated, 50 lb dry-chemical fire extinguisher system and manual emergency shut-down system. The emergency stop system disables the engine's ignition, closes the LNG supply, stops the exhaust fan, and vents the propane out of the room.

The Cummins Model B5.9G 5.9 liter, 6 cylinder, turbocharged, natural-gas engine is mounted on a steel test stand attached to the test cell floor through four air springs to isolate engine vibrations. The engine is loaded by a Go-Power DT-1000 500 hp water-brake dynamometer, as shown in Figure 2. The rotor of the water brake is bolted to the engine flywheel via an adapter plate. The stator transfers the load to hydraulic load cells mounted in the dynamometer housing, which is bolted to the engine bell housing. A 3 hp pump supplies the water to the dynamometer. Dynamometer torque is controlled by varying the water level in the dynamometer using a manual valve.

Fuel Supply

Fuel for the tests is delivered from one of a pair of vacuum insulated LNG tanks (Cryenco LNG-0077). For safety reasons, the tanks are placed outside of the test cell. An insulated liquid line is piped into the test cell and is connected to the propane-heated LNG evaporator. In normal practice, tank pressure is governed by the heat leak from the ambient into the tank, and is limited by venting gas through a relief valve. To control tank pressure during the tests, the tanks were either cooled by circulating liquid nitrogen through heat exchangers located within the tanks, or heated by allowing some of the stored LNG to circulate through an external heat exchanger.

Pressurized LNG is supplied to an LNG vaporizer which is heated by the closed propane loop. The high-pressure vapor from the vaporizer is supplied to the engine through a first-stage pressure regulator at a pressure of approximately 50 psig. The gas pressure is throttled to approximately 35 psig by a secondary regulator within the engine's air/fuel mixer. Gas flow is regulated by a gas control valve within the engine's and air/fuel mixer, which is controlled by the engine's electronic control module (ECM).

Propane Loop

The closed propane loop was designed to transfer heat from the warm charge-air to vaporize the LNG. This intermediate heat transfer loop is employed instead of cooling the air directly with LNG to avoid frosting that would occur if warm moist air is exposed directly to cryogenic temperatures. The propane is supplied to the second-stage air cooler as a liquid at a temperature very close to the air temperature leaving the second-stage air cooler. The liquid propane is condensed in a heat exchanger by the evaporating LNG.

A schematic of the propane/LNG loop is shown in Figure 3. Propane is pumped from a 1-gallon sump by a sealed-rotor, magnetic-drive pump (Tuthill). Net propane flow to the cooling loop is manually controlled by bypassing excess flow back into the sump. The liquid propane first flows to the second-stage air cooler. There the propane is evaporated by the warm charge-air in a cross-flow plate-fin type heat exchanger. In the event that there is insufficient heat in the charge-air to vaporize the LNG, or if the second-stage charge-air cooler is not operating, the propane is heated in a secondary water-heated evaporator in series with the second-stage air cooler. The vaporized propane flows into a coaxial (tube-in-tube) heat exchanger, where it is condensed by evaporating LNG. The liquid propane then drains out of the heat exchanger back into the sump. Propane and LNG flows are co-current in the LNG evaporator in order to avoid excessive subcooling of the condensed propane, which could frost the second-stage air cooler. The propane loop components are shown in Figure 4.

Air System

The stock engine configuration in a truck installation is to place an air-cooled charge-air cooler in-line with the air-cooled radiator and cooling fan. This arrangement was impractical to duplicate in the test-cell for several reasons:

1. The amount of cooling air flowing over the first-stage charge-air cooler and radiator in a vehicle is strongly affected by "ram-air" (the air flow induced by the vehicle's forward motion). The relationship between vehicle speed (and hence ram-air) and engine speed is variable, as it depends upon the specific gear ratio. Since the amount of cooling air flow is variable, there is no single relationship between air-cooler/radiator heat rejection and engine speed, even if actual ambient air temperature is properly simulated.
2. The engine dynamometer measures brake horsepower after the shaft power delivered to the fan and accessories. The fan power at a given fan speed will vary with air flow and

temperature. The fan speed at a given engine speed in turn will vary with radiator leaving air temperature (LAT) due to the effect of the thermostatically controlled viscous clutch. Without a fan dynamometer, the fan power will be unknown.

To avoid this uncertainty introduced by the radiator fan, it was removed and the normal air-cooled radiator was replaced by a water-cooled shell-and-tube heat exchanger. The air-cooled charge-air cooler was placed in a water bath, and the first-stage ECM was controlled by manually controlling the temperature and flow of the bath cooling water. This arrangement provided simulation of any desired ambient temperature/road-speed condition by instead permitting control of the corresponding first-stage ECM.

The second-stage air cooler was inserted in between the first-stage and the air/gas mixer. For baseline tests, this heat exchanger was bypassed, and the first-stage was connected directly to the mixer. Thus, the second-stage test condition includes the additional air-side pressure-drop of the second-stage air cooler.

Instrumentation

LNG, propane, air and water loops were instrumented with thermocouples, pressure gauges and flow meters as indicated in Figures 1 and 3. Temperatures and propane pressure were recorded on a data acquisition computer. A Cummins diagnostic tool, 'INSITE', was used to monitor and record engine parameters directly from the ECM in real time. Engine speed, coolant temperature, fuel feedback control parameters, boost pressure, and intake manifold temperature and pressure were recorded with this tool. The remaining pressures, some temperatures, and exhaust emissions were recorded manually. Engine torque was measured by manually recording the displayed value of the analog console of the Go-Power dynamometer.

Gas flow was measured by a turbine-type flowmeter, and subsequently when this measurement proved to be unreliable, by weighing the fuel tanks over a period of several minutes of steady-state engine operation. Air flow was both measured directly with an orifice meter upstream of the turbocharger, and was calculated indirectly using the gas flow measurement and the air/fuel ratio as determined from the exhaust oxygen measurement.

Exhaust gases were sampled and analyzed using an Enerac 2000 electrochemical type analyzer. O₂, NO_x, and CO and unburnt hydrocarbons (as methane) were directly measured using separate sensors in the device. All sensors in the unit were carefully calibrated before each test.

Test Plan

The test plan was to measure the effect of reduced charge-air temperature on engine performance at three points along the wide-open throttle power curve: at the peak torque point (nominally 1600 RPM), maximum power (nominally 2800 RPM), and at an intermediate point to be determined. Each of these points was to be tested at two charge-air temperatures exiting the first

stage air cooler: 95° F and 130° F. All of these test points were to be with a turbocharger boost pressure at the knock limit.

It became apparent while performing instrumentation shakedown tests that the engine was not operating properly. The engine could not reach rated power. A considerable amount of time and effort was spent in re-calibrating and upgrading instrumentation and in servicing or replacing engine components in an effort to bring the engine up to its published performance, which met with only partial success. One symptom of the problem was that the exhaust oxygen would increase with engine speed¹. It appeared that the ECM was controlling the fuel flow to a desired air-fuel ratio, but the ratio was beyond the value specified by the manual. After treating all the troubleshooting symptoms possible in our shop, a Cummins technician and engineer were consulted. The engineer suspected the ECM was malfunctioning and suggested a replacement. He indicated that this was a 'first generation' ECM and several bugs have been corrected since its release. He suggested that it be replaced with the most recent 'third generation' module and calibration.

After the ECM was replaced and calibrated, the engine operated noticeably better in terms of air-fuel ratio control, but excessive levels of unburnt hydrocarbons were observed, especially at high engine speed. The engine still could not reach rated power or torque. While at times close to rated performance was achieved with relatively low unburned hydrocarbons, this was erratic and un-reproducible².

The other obstacle to achievement of the test plan was that the boost pressure could not be increased sufficiently to exceed the knock limit. The boost pressure is controlled by the ECM through actuation of the turbine waste-gate. It was possible to override the waste-gate control signal in order to increase boost pressure. However, the ECM also operated a boost overpressure limit, which shut down the engine if the boost pressure exceeded a pre-set limit that could not be overridden.

Since at this point all of the remedies that were feasible to implement on-site had been exhausted, the decision was made to move ahead with the test plan, making use of the best engine performance that could be attained. The most consistent operation that was of interest had been achieved at the baseline 1600 RPM peak torque condition. Accordingly, the test plan was modified to focus on this point. Data was then acquired at the two first-stage air-cooler exit temperatures of 95° F and 130° F, with and without second-stage air cooling. The boost pressure was set at the normal baseline value.

¹ At this stage of the testing the exhaust hydrocarbon sensor was not operational, so it could not be determined to what extent unburned fuel contributed to the high exhaust oxygen.

² For one transitory period, rated torque was achieved with 0.05% hydrocarbons measured in the exhaust. However, this point could not be duplicated.

Results and Discussion

The engine and dynamometer were controlled manually during the tests. Start-up and operation under manual control was straightforward. The LNG evaporated without active cooling at startup of the engine and during idle. After a warm-up period, torque and speed were increased by opening the engine throttle and increasing the dynamometer load. The propane pump was turned on once liquid LNG appeared at the inlet to the LNG evaporator. The LNG was heated by the supplementary water-heated propane evaporator for the baseline tests, and in the second-stage charge-air cooler tests until the temperature of the air leaving the first-stage charge-air cooler was warm enough to evaporate the propane. Operation of the charge-air/propane/LNG heat transfer system was smooth and stable, and was not affected by the fluctuations noted below.

After the propane loop was operational, the engine was gradually brought to the desired test condition and was allowed to come to steady state. Fuel consumption was measured between the start and end of the steady-state test by weighing the LNG tank. Typical test data recorded by the Cummins INSITE diagnostic tool is shown in Figure 5. The fluctuations of the fuel control valve are controlled by the ECM, which is monitoring exhaust oxygen and is attempting to maintain a desired air/fuel ratio. The cause of these fluctuations was not determined, but might have been related to the apparent misfiring/incomplete combustion. The fluctuations in engine speed/torque and intake manifold pressure appear to be coupled to the fuel control valve fluctuations. Similar fluctuations appeared in the dynamometer torque, which was not recorded automatically.

Engine performance

Table 1 shows the data for the two baseline runs at 95°F and 130°F first stage LAT, and for the corresponding tests with the second-stage air cooling. Here engine torque is recorded manually from the Go-Power dynamometer readout, while engine speed is averaged over the test run. It should be noted that torque and speed fluctuated on the order of about 2% - 3% during the run, which is about the level of uncertainty of the power measurement. Fuel flow is an average determined from the weight loss of the fuel tank during the run³. Air flow was determined from air/fuel ratio, as determined by measured exhaust oxygen, and from measured gas flow. This determination proved to be more consistent and reliable than the direct measurement of air flow by the air orifice flow meter located at the charge-air inlet.

Table 2 compares the predicted versus measured performance. Theory predicts that the torque varies linearly with the air flow. At constant speed, it can be shown⁴ that the air flow is related to the inlet temperature and pressure by the relation:

³ Although the test apparatus was equipped with a calibrated turbine gas flow meter, this gave unreliable readings when the engine was operating.

⁴ R. Raymond, "Gas Engine Power Enhancement Through LNG Cooling of the Inlet Air Charge," Final Report for the Consolidated Natural Gas Service, Project No. 5827.

$$[AirFlow] \sim \frac{P_i}{\sqrt{T_i}}$$

where P_i and T_i represent the intake manifold pressure and temperature. Using this relation, and accounting for the slight differences in engine speed, the predicted air flow and torque increase by 4.2% and 2.8%, versus a measured air-flow increase of 5.0% and 4.7%, for 95°F and 130°F first stage LAT, respectively. The measured torque increases by a larger amount, at 9.8% and 6.1%, for 95°F and 130°F first stage LAT, respectively. In each case, the measured air flow and torque exceed the predicted values. The differences between predicted and measured performance are judged to be due to the non-reproducible engine performance and to measurement inaccuracy.

Second-Stage Charge-Air Cooler Performance

Table 3 shows the performance of the second-stage charge-air cooler for the two cases. The cooler reduced the air temperature by 38°F for the 95°F case, and by 56°F for the 130°F case. The pressure drop on the air-side was 0.2 psi, and 4 psi on the propane side.

The measured thermal effectiveness of the second-stage air cooler was between 94% to 96%, resulting in a leaving air temperature within about 3°F of the propane temperature. Based upon this result, the overall thermal conductance (product of overall heat transfer coefficient times surface area) is estimated to be approximately 800 Btu/h-ft²-°F. This compares with a design specification of 500 Btu/h-ft²-°F.

Similarly, the propane-LNG evaporator heated the LNG vapor to within about 2°F of the propane temperature. Thus, the air and gas are within about 5 degrees of each other leaving their respective heat exchangers. Considering the fact that the mixed air/gas temperature should not depend on the terminal temperatures leaving the respective heat exchangers, but only on their initial enthalpies, it is likely that the effectiveness of these heat exchangers could be reduced without compromising performance of the engine.

Heat Balance

A heat balance of the air/propane/LNG system is shown in Table 4. This compares the heat input to the LNG and the heat out of the second-stage charge-air cooler. The calculations assume that the LNG inlet condition is saturated liquid, i.e., the conditions at the storage tanks. These calculations indicate that for the case of 130°F air entering the second-stage air cooler, the LNG and air heat flows are comparable; while at 95°F entering air temperature (EAT), the heat into the LNG is about 4,000 Btu/h greater than the heat out of the second-stage air cooler.

The probable explanation is that there is a heat gain from the ambient to the cryogenic LNG components, apparently on the order of 4,000 Btu/h. When the second-stage charge-air cooler inlet air temperature was 130°F and its outlet temperature was 74°F, there was a correspondingly

large heat loss from the air-cooler. At air temperatures of 95°F/57°F, the heat loss from the air cooler was negligible. For the case of 130°F second-stage EAT, the propane loop temperature at 71°F was close to ambient temperature (76°F), resulting in a negligible heat gain to the propane. At 95°F second-stage EAT, the propane loop was at 54°F, resulting in a small but significant heat gain from the environment.

In the present test, the LNG piping and evaporator and the propane loop were insulated, but the charge-air cooler was not. In a practical system, it might be feasible to achieve better insulation of the LNG/propane piping and components, but it would be counterproductive to insulate the air system, except possibly downstream of the second-stage air-cooler. Accordingly, it might be feasible, but more costly, to increase the second-stage air cooling by perhaps another 20% at the lower first-stage LAT's (e.g., at 95°F).

Initial estimates of the degree of charge-air cooling assumed that the LNG would be available at a temperature of -260°F, based on the atmospheric boiling point of methane. This proved to be overly optimistic for two reasons. On the one hand, the engine's fuel system requires a delivery pressure of at least 50 psig. This would raise the boiling point of methane to -220°F. However, the LNG boiling point differs significantly from that of pure methane. The 96% methane LNG used for the test had a boiling point of -194°F at 54 psig, rather than -218°F if it were pure methane. The difference between an LNG temperature of -260°F and -194°F is calculated to result in a change of 8°F in the charge-air temperature.

Emissions

An additional benefit of charge-air cooling is expected to be a small but significant reduction in NO_x emissions. Figure 6 shows a plot of the NO_x concentration in the exhaust versus intake manifold temperature under steady-state operation. The exhaust oxygen was between 7.8% to 8.0% (air/fuel ratio between 26.2 to 26.8) for all these points. The trend of the data indicates a greater reduction of NO_x due to cooling as the temperature level increases. The NO_x reduction of roughly 10% by cooling from about 100°F to 60°F is similar to that observed by Raymond⁵.

CO emissions are shown in Figure 7. Air-free CO is on the order of 2000 ppm, and appears to be relatively insensitive to manifold temperature.

Conclusions and Recommendations

The test program demonstrated the feasibility of utilizing the enthalpy of the LNG to reduce intake manifold temperature and thereby to increase engine power. Because the engine control system did not permit the boost pressure to be increased up to the knock limit, the effect of charge-air cooling on increasing the knock limit, and thereby permitting higher boost pressures to be utilized, could not be demonstrated. Only the partial effect of increasing the air density by

⁵

Ibid. Raymond found roughly a 10% reduction for 50°F of cooling.

cooling was demonstrated, and resulted in an measured increase of torque (power) at constant boost pressure and speed of 6% to 10%. Had the boost pressure been increased as a result of raising the knock limit, a further increase in torque and power could have been realized.

The tests demonstrated that utilization of LNG cooling in a second-stage charge-air cooler is straightforward and uncomplicated, with no adverse effects on engine operation. The LNG evaporator and second-stage charge-air cooler were highly effective, and resulted in only modest pressure drops. The thermal effectiveness of the charge-air and LNG heat exchangers required for a practical system can probably be reduced significantly from the tested values without adverse effect on performance.

While operation of the indirect propane heat transfer loop was straightforward and uncomplicated, elimination of this loop by direct heat transfer between the charge-air and the LNG should be pursued as a means of reducing cost and simplifying the system.

Raising the boost pressure to the knock limit was prevented by the engine's control module (ECM). In future tests, the engine should be operated in a manual mode, or at least provisions should be made to override selected functions of the ECM.

Even if the ECM had allowed the boost pressure to be raised, the present turbocharger may not have had sufficient capacity to reach the knock limit over the speed range of interest. Therefore, the option of a higher-pressure turbocharger should be available in future tests.

Testing was severely hampered by erratic and incomplete combustion. It was not feasible to remedy these deficiencies within the confines of test cell nor within the resource constraints of the project. In future tests, the test engine should be thoroughly checked out beforehand and its proper performance verified, preferably by a factory-authorized service facility.

Table 1. Engine performance data.

		Baseline		Second Stage Air Cooling	
Exit temperature of 1st stage CAC	°F	95	130	95	130
Engine torque	ft-lb	411	416	451	441
Engine speed	RPM	1,614	1,610	1,606	1,595
Engine power	HP	126	128	138	134
Intake manifold pressure	inHg (abs)	56	57	56	57
Intake manifold temperature	°F	95	125	60	79
Fuel flow rate	lb/hr	43	41	45	43
Exhaust O ₂	%	8.0%	7.9%	7.8%	7.9%
Air-Fuel ratio	lb/lb	25.6	25.5	25.4	25.5
Air flow rate	lb/hr	1,091	1,056	1,145	1,105
Unburnt hydrocarbons	%	0.30%	0.25%	0.23%	0.27%
CO	PPM	1122	1055	1047	1052
NO _x @3% O ₂	PPM	489	736	588	453
BSFC	lb/HP-hr	0.338	0.325	0.326	0.324

Table 2. Measured versus predicted engine performance.

MEASURED DATA		Baseline	2nd Stage Air Cooler	Baseline	2nd Stage Air Cooler
1st-Stage LAT	°F	95	95	130	130
Engine Torque	ft-lb	411	451	416	441
		100.0%	109.8%	100.0%	106.1 %
Engine Speed	RPM	1,614	1,606	1,610	1,595
Engine Power	HP	126	138	127	134
Manifold Pressure	in H abs	56	56	57	57
Manifold Temperature	°F	95	60	125	79
N*P _i /sqrt (T _i)		3,803	3,964	3,795	3,901
Air Flow	lb/hr	1,091	1,145	1,056	1,105
		100.0%	105.0%	100.0%	104.7%
Fuel Flow	lb/hr	42.7	45.0	41.4	43.4
BSFC	lb/HP-hr	0.338	0.326	0.325	0.324
Air/Fuel Ratio	lb/lb	25.6	25.4	25.5	25.5
PREDICTED RESULTS					
Air Flow	lb/hr		1,137		1,086
		100.0%	104.2%	100.0%	102.8%
Torque	ft-lb		428		427
		100.0%	104.2%	100.0%	102.8%

Table 3. Charge-air cooler performance.

		Second Stage Air Cooling	
Air flow rate	lb/hr	1,145	1,105
Air inlet pressure	psig	13.4	13.4
Air exit pressure	psig	13.2	13.2
Air inlet temperature	°F	95	130
Air exit temperature	°F	57	74
Propane pressure	psig	85	114
Propane temperature	°F	54	72
LMTD	°F	14	17
Heat transfer rate	Btu/hr	10,444	14,856
UA	Btu/hr-°F	760	860
LNG temperature into HX	°F	-193	-194
NG temperature out of HX	°F	53	70

Table 4. Heat balance of propane system.

Air inlet temperature	°F	95	130
Heat into LNG	Btu/hr	14,643	14,374
Enthalpy of vaporization	Btu/lb	199	199
LNG flow rate	lb/hr	45.0	43.4
LNG specific heat	Btu/lb-°F	0.5	0.5
LNG inlet temperature	°F	-193	-194
LNG outlet temperature	°F	60	71
Heat out of air	Btu/hr	10,442	14,851
Air flow rate	lb/hr	1,145	1,105
Air specific heat	Btu/lb-°F	0.24	0.24
Air outlet temperature	°F	57	74
Air ΔT	°F	38	56
Ambient Temperature	°F	77	76
Air Intake Manifold Temp	°F	60	79
Heat Imbalance	Btu/hr	4,200	(477)

Figure 1. Air and Water Loop.

Figure 2. Cummins B5.9G Engine in Test Cell.

Figure 3. Propane/LNG Loop.

Figure 4. Propane Loop in Test Cell.

Figure 5. Sample Steady-State Data Set.

Figure 6. NO_x versus Manifold Temperature.

Figure 7. CO versus Manifold Temperature.

Design of LNG Second Stage Intercooler for Test

November 5, 1998

System Description

In LNG-fueled heavy duty vehicles, liquid natural gas from an on-board fuel tank is vaporized and superheated to near local ambient temperature for use as the engine's fuel. The negative enthalpy available in this process is used by ATM to cool turbocharged engine supply air after it exits a first stage intercooler. First stage intercoolers are usually ambient air-cooled cross-flow heat exchangers and are customary on turbocharged heavy duty (HD) vehicles that operate on diesel fuel or natural gas.

The worst case design operating condition chosen for a HD truck tractor is a 95°F day, climbing a steep hill in low gear with a fully loaded trailer in tow. Engine power becomes the important issue. This is especially so for a natural gas-fueled truck tractor. The available engine power is inversely related to engine intake air temperature. First, increased air density more than compensates for the volume displacement of the natural gas fuel. Second and more importantly, the margin on engine knock limit is substantially enlarged, allowing greater compression. An additional factor of growing importance, reduced engine charge air temperature reduces engine-out NO_x. The customary first stage intercooler reduces engine air temperature to within 30°F of ambient at the worst case design condition, or to about 125°F on a 95°F day. The available cooling with the ATM technology (Second Stage Intercooler) reduces the engine air delivery by an additional 65°F, or to about 70°F on a 95°F day.

ATM uses an intermediate fluid, propane, to first vaporize and warm the LNG and then cool the intake air. Direct heat exchange of LNG and engine air would freeze moisture in the air. The Second Stage Intercooler consists of a LNG evaporator/superheater (condenses propane vapor), an engine supply air cooler (vaporizes propane), a propane circulation pump, and an engine coolant heat exchanger to warm the propane at lower ambient temperatures to prevent freezing of engine air moisture.

A schematic is attached of the current LNG Second Stage Intercooler heat transfer circuit design to be tested at AMTI. A detailed tabulation of a range of heat transfer circuit parameters is attached in the AMTI memorandum: "Analysis Of Propane Loop."

Design And Performance Specifications

Propane Condenser/LNG Evaporator-Superheater

The selected LNG evaporator-superheater is designed to vaporize and warm the LNG to near ambient temperature for a 195 HP engine operating at an increase of 20% in power (235 HP). It will handle a heat flow of 31,500 BTU/Hr with saturated LNG supplied at 50 psig and -220°F, and 33,000 BTU/Hr with saturated LNG or methane supplied at 25 psig and -235°F. Propane enters the condenser side as a saturated or partially wet vapor at 35°F and 55 psig. Methane vapor will exit the evaporator side superheated at 20 - 32°F over the range of full load engine conditions anticipated for the test program. The natural gas vapor line from the evaporator to the engine's carburetor will not be insulated, so that the vapor will arrive at approximately engine

supply air temperature. This allows the carburetor to function in an optimal manner.

Water-To-Propane Heat Exchanger

This heat exchanger vaporizes propane during baseline testing to establish normal operation without second stage intercooling. For testing convenience, hot water indirectly supplies the heat required to vaporize and bring the LNG to near ambient temperature. Current practice uses engine coolant to directly vaporize the LNG.

Second Stage Intercooler

This heat exchanger provides second stage intercooling for engine supply air exiting the customary first stage intercooler. For test program purposes, entering engine air will be a controlled parameter, set at 125°F and 98°F (30°F over the two test conditions: 95°F and 68°F ambient air, respectively). The heat flow will be 31,500 - 33,000 BTU/Hr. Exiting engine supply air will be at 65 - 68°F and 33 - 35°F, respectively.

Propane will be supplied at 35°F and 55 psig as a saturated or slightly subcooled liquid. It will leave as a saturated or partially wet vapor.

Liquid Propane Circulator

This pump will circulate liquid propane at up to 1.4 GPM to overcome pressure losses up to 20 psi. A bypass valve will be provided to accommodate test conditions where less than 1.4 GPM is required. The pump is magnetically coupled to the motor to eliminate risk of propane leak, and as an added precaution, the pump motor is rated explosion proof.

Design Parameter Selection

For each of the above cited components, pressure, temperature and flow parameters have been selected. Safety and heat balance requirements dominated considerations.

Flow. As cited for the Circulator, 1.4 GPM has been identified as the necessary maximum propane flow rate. This is for the purpose of providing saturated or somewhat wet propane vapor to the Second Stage Intercooler, consistent with achieving desired heat balance conditions. The 20 psi output requirement was conservatively determined primarily by the pressure drops through interconnecting piping and Second Stage Intercooler tubes.

Engine air requirements, based on the Cummins B model engine, were calculated on the basis of an output of 235 BHP, representing a 20% increase over the present natural gas rating for that engine of 195 hp. A 20% increase is anticipated.

Temperature. For this application temperature affects materials selection. The cold end of the system can potentially, under unusual conditions, encounter -260°F. Accordingly, all components and piping at the cold end will be of either 304 or 316 series stainless steel.

Pressure. The maximum static pressures are imposed on components comprising the propane heat transfer loop. The critical component pressure-wise containing propane is the item selected for the Second Stage Intercooler evaporator coil. This is an automotive air conditioning evaporator manufactured by Harrison, a division of GM.

Harrison shared with ATM their policy on static pressure rating. Minimum burst must exceed maximum operating pressure by 2X. Although this does not satisfy Section VIII the ASME Unfired Pressure Vessel standard, ATM has reached a determination to respect GM's internal standard Minimum burst criteria. This standard provides a 2X safety factor between a worst case set of manufacturing problems and extremes in operating experience. The selected evaporator coil has a minimum burst pressure rating of 400 psig.

AMTI will limit test cell maximum temperature to 109°F, which corresponds to a propane vapor pressure of 200 psig. Also, a 200 psig pressure relief valve is being installed in the propane loop, vented to outdoors. Test cell temperature is an issue due to heat soak following test shut down.

ATM has specified that the packager of the GM evaporator for project purposes (Dynamic Specialties, Inc., Mentor, OH) pressure test the evaporator as follows:

- Propane-side tubes, at 150% maximum operating pressure. (Test @ 300 psig)
- Air side package housing pressure, at 150% maximum operating pressure. (Test @ 22 psig)

The purpose of these tests is to assure that ATM has not acquired a defective evaporator component or designed an inadequate package housing.

Under either of the above tests either a burst or visible evidence of material yield will require testing an additional evaporator coil sample and/or modifying the housing. The pressure testing will be conducted at room temperature. Both the evaporator coil and the housing are built from 6000 series aluminum, which becomes stronger at reduced temperature without embrittlement. Under test conditions these components are expected to operate at or below room temperature.

The other propane loop components, including tubing, fittings and other heat exchangers, have minimum burst pressures considerably in excess of 400 psig.

Component Selection

Component selections have been made based on satisfying design and performance specifications, and design parameters, at least cost to the program.

Propane Condenser/LNG Evaporator- Superheater.

A coaxial tube within a tube, spiral wound heat exchanger has been ordered from Doucette Co., Model No. cc 200 TT. The material is 304 stainless steel. Uncoiled, the heat exchanger is 23 feet long. This configuration maximizes counter flow heat exchange effect, which is necessary to both vaporize and superheat the LNG.

Water-To-Propane Heat Exchanger.

A conventional shell and tube design has been ordered: Doucette Co. Model No. HED 8058-31.

Second Stage Intercooler.

An automotive air conditioner evaporator, recommended by GM engineers to meet the specific heat transfer requirements specified by ATM, has been purchased, plus a spare. ATM has confirmed the heat transfer capacity of this evaporator, following purchase and inspection, to within 14% of GM's numbers. We expect that this evaporator incorporates heat transfer sophistication not documented in textbooks or other generally available information, and were not disclosed to ATM following our inquiry. This evaporator is used in the largest passenger cars GM has built since 1976: the 1995-97 Chevrolet Caprice and the 1995-97 Buick Roadmaster. It is an all aluminum, furnace brazed fin and rectangular tube assembly, manufactured by the Harrison Heat Transfer Division of GM's Delphi Unit.

ATM has subcontracted Dynamic Specialties Inc. (DSI), Mentor OH, to construct the necessary housing and to assemble the housing and evaporator. In addition, DSI will execute exterior modifications to the evaporator to accommodate the housing, and provide inlet and outlet interface for Swagelok-type tube fittings. DSI has manufactured aluminum air conditioner components for GM during the past 20 years. The assembled intercooler will be delivered to AMTI fitted with the type T thermocouples compatible with AMTI's data acquisition system.

Liquid Propane Circulator.

A vane-rotor pump manufactured by Tuthill Pump of California has been ordered. Model NO.DDPPET2NM1.6. This pump has a body of 316 stainless steel, vanes and bearings of carbon/PTFE impregnated PPS. It is magnetically coupled to an explosion proof motor.

Tuthill Pump of California builds an identical but lower cost pump with a plastic body compatible with propane. However ATM's minimum burst pressure (400 psig) specification led Tuthill engineers to recommend the 316 stainless body version.

Test Engine.

A Cummins model B engine has been acquired from Roadway Express. This engine was dressed for natural gas, as new, by Cummins Engine Co. It has been road tested by Roadway since 1994. Roadway has agreed to provide records of the engine's performance. Cummins' Boston, MA shop has provided AMTI very useful support services in setting the engine up for test purposes.

Second State Intercooler System Assembly And Test Plans

- All of the Second Stage Intercooler system-unique components are presently specified and on order. The Cummins B model engine has been installed on the test frame built by AMTI, and coupled to a water brake dynamometer.

- All system test components are expected to be delivered to AMTI before December 15, 1998. Some will have been previously pressure tested. This delivery schedule should allow system testing to begin by January 15, 1999.
- As resolved immediately since the September 16 project review meeting in Watertown, the air flow to the Second Stage Intercooler will be supplied from a real first stage intercooler, fooled to provide engine air at two simulated air delivery temperatures, corresponding to 95 and 68°F ambient air. Turbocharger ambient air engine delivery first stage exit intercooler temperatures will correspond to 68 and 95°F.
- The one remaining unresolved barrier to test start up scheduling is availability of liquid 99% methane to AMTI's Watertown, MA location. ATM and AMTI would much prefer to conduct the testing on 99% methane "LNG" rather than commercial LNG. For now our concern is in regard to consistent test results. "Dirty" LNG varies considerably in ethane and propane content, up to 10% in ethane and 4% in propane, which will shift as the stored LNG is consumed.

Implications For A Truck Tractor On-Board Prototype System

The three heat exchangers and pump that make up the Second Stage Intercooler were purchased for the laboratory test for about \$1500. Their combined weight is about 70 pounds. The necessary plumbing will add perhaps another 10 pounds. There is room for considerable improvement in cost and complexity of this first generation design. This will be pursued under the BNL contract as part of the cost/benefit study.

A conceptual drawing of a second generation design is attached. It was in the ATM proposal to BNL. This design incorporates the three heat exchangers in one shell half filled with liquid propane. The design eliminates the propane pump. LNG vaporization takes place in the propane vapor space. Heat exchange with engine air and heat exchange with engine coolant take place submerged in the liquid propane. The propane transport is by the "heat pipe" mechanism. The propane vapor condensed against the LNG falls back into the pool of liquid propane. The propane liquid boiled against engine air and engine coolant rises into the propane vapor space. The engine coolant flow is controlled to maintain proper pressure. This modular design will achieve all of the heat exchange, heat balance and safety goals ATM has established. It has the potential for very low cost, low weight and ease of even retrofit installations on LNG engines.

An earlier study by a potential manufacturer revealed a design problem relative to heat expansion/contraction related stresses. Resolution of this problem and the construction and test of the module on the Roadway test engine would be a natural follow-on to the present contract.

ATM believes the multi-pass heat exchange module represents the basis for a viable on-truck device, rugged and practical in the context of a truck tractor. The device shown will require engineering regarding fabrication details and materials requirements. A monolithic on-truck device can be properly designed in preparation for an on-road field test.