Engineered Approach to Reach Emissions Targets on a Worthington LTC Engine

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Objective

Perform analysis, field testing, and further analysis as required to establish the turbocharger specification for a field retrofit "pure-turbocharged" Worthington LTC-8H installed at El Paso Corporation's Compressor Station 32 located in Jasper, Texas. The turbocharger specification and other components of the overall engine system were constrained by oxides of nitrogen (NO_X) limits of 8 g/bhp-hr and carbon monoxide (CO) limits of 3.0 g/bhp-hr.

Background

A summary of relevant background information follows. This summary includes the regulatory drivers, a description of the affected engines, prior attempts at retrofitting the engines for regulatory compliance, and the engineering process used by the team to ultimately design, specify, and test the successful engine retrofit technologies.

Regulatory Driver

The Texas Commission on Environmental Quality (TCEQ) promulgated the Grandfathered Facilities Permit Program in 2002. Under this rule, many previously grandfathered engines stood to lose this status and would subsequently fall under the federal Title V air permit program. Under the assumption that a handful of engines would be subject to the mandatory rule, El Paso Corporation (EPC) took advantage of the Voluntary Emission Reduction Permit (VERP) program, enacted by the TCEQ, which provided for capital reimbursements on voluntary modifications[1].

Facility

El Paso Corporation operates eleven Worthington LTC-8H integral engine-compressor units at its Compressor Station 32 located in Jasper, Texas. Seven of these units are targeted for retrofits under the VERP program. As such, EPC must reduce NO_X emissions from these engines, from an estimated 15 g/bhp-hr to 10 g/bhp-hr with a corresponding CO emissions of 1.96 g/bhp-hr. Table I summarizes the engine particulars.

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[1] Up to 80% of associated expenditures.

Table I: Engine specifications.

Bore (inches)	15
Stroke (inches)	15
Connecting Rod Length (inches)	39.38
Number of Cylinders	8
Exhaust Port Closure (°ATDC)	220
Estimated Geometric Compression Ratio	8
Rated Speed (rpm)	300
Rated Power Output (bhp)	1,100
Rated BMEP (psi)	68.5

In view of the relatively modest reductions required and low unit output, low cost solutions have continued to be of particular interest.

Prior History

El Paso Corporation utilized a phased approach to achieve the emissions targets. This approach was based on prior successful history with improved mixing technologies and an increased air flow rate through the engine. The Digicon Super-Sonic Fuel Injector (SSFI) was chosen to enhance in-cylinder mixing, and an upgraded Globe turbocharger was chosen to increase the air flow rate to the engine.

Pre-Conversion Testing

El Paso Corporation conducted baseline testing of Unit 6A in September 2002 over a wide range of operation. The type of emissions analyzers used is not known. The NO_X emissions varied between about 6 and 20 g/ bhp-hr and CO varied between 2 and 3.5 g/bhp-hr. In or around October 2002, Digicon installed their SSFI valves and EPC re-tested the engine using a portable analyzer. The NO_X ranged between 4 and 15 g/bhp-hr and CO ranged from less than 0.5 g/bhp-hr to over 5 g/bhp-hr. The engine heat rate improved by approximately 10%.

The data indicated that, while enhanced mixing significantly improved engine performance, the engine lacked sufficient air to achieve the desired emissions at rated conditions. The engine did appear capable of achieving those emissions levels at reduced load. This suggested a relatively minor increase in air flow and/or charge density would result in the desired emissions level at rated conditions.

El Paso Corporation contracted AETC to assist in developing an air specification. To determine the appropriate air specification for this engine, the team installed a motor-driven blower and tested the engine over the desired range of operation. Based on these results, AETC developed an air specification for either a motor driven blower or a pure turbocharger configuration. Following a cost analysis, EPC elected to utilize a pure turbocharger solution rather than a motor driven blower[2] and ultimately selected Digicon to perform all of the necessary engine modifications.

2005 Digicon Conversion

To achieve the project objectives, Digicon elected to use a "pure" turbocharger solution. Based on a limited set of field data and concerns that the current liner port configuration was too restrictive for this approach, Digicon made several changes to the liner porting and piston crown. After several iterations by Digicon, AETC mapped the engine in June 2005. The engine

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2 Economics of installing utility lines to a facility that maintains production of 100% of needed power were not attractive.

exhibited extremely unfavorable trade-offs between NO_X and CO, and between NO_X and BSFC with significant degradation in engine operability, stability, and sustainability.

El Paso Corporation, Digicon, and AETC staff met onsite in July 2005 to review the project status. Based on those and subsequent discussions, the team determined blow-back of burned charge into the intake manifold was the primary problem. This blow-back was solely due to the relatively short delay between exhaust port opening (EPO) and intake port opening (IPO) created by liner port and piston crown modifications. Subsequent CFD analysis by Digicon confirmed this hypothesis. Digicon therefore further modified the piston and port geometry to increase the delay between EPO and IPO on one engine. In conjunction with this change, Digicon also prepared a lower-pressure ratio turbocharger based on AETC's recommendation to reduce the boost level. The lower boost level would result in air manifold pressures more in line with the original air specification that was developed following development testing with the motor-driven blower.

AETC tested the modified Unit 9A in October 2005 with the modified ports and liners and with the high- and low-boost turbocharger configurations referenced above. At the time, the engine still exhibited extremely unfavorable trade-offs between NO_X and CO, and between NO_X and BSFC with no improvement in engine operability. Digicon had run additional CFD scenarios by this time, likewise indicating further modifications to liner ports and piston crowns may be required.

November 2005 Engineering Design Review

Based on the difficulties encountered to this point, EPC retained AETC and ScavengeTech in November 2005 to participate with EPC staff in an engineering evaluation of the effort to date and to develop an engineered path forward. This evaluation included a meeting with Bo Mikkelsen of EPSI, who had converted an LTC in the late 1990's to run under emissions constraints of 1.5 g/bhp-hr NO_x and 2.5 g/bhp-hr CO. EPSI had used a combination of series turbocharger-scavenger pump operation and micro pre-combustion chambers (MPCC) to achieve the emissions constrained operating level. As part of the conversion, no modifications to existing liner ports or pistons were made. The design review team also consulted with Dresser on a similar conversion to an HBA.

Digicon reviewed their conversion experience with the team. Based on input from Enginuity, Digicon originally modified the liner port heights to increase the blowdown energy to the turbocharger. This resulted in the unsatisfactory operation because (1) scavenging air short circuited through the cylinder liner and (2) the port configuration created the exhaust blowback noted above. Digicon reported they had attempted to operate the engine in a pure turbocharger mode with OEM liners and pistons and Digicon's specified turbocharger configuration. The results were that the turbocharger would not self sustain and that the engine back-fired and eventually stalled soon after start up. Digicon interpreted this as validating the need to continue pursuing modifications of the pistons and liners.

After reviewing options, risks, and costs associated with various courses, the design review team elected to first evaluate the engineered approach of a pure turbocharged solution with OEM liners and pistons. This approach best leveraged the current infrastructure installed by Digicon, minimized cost and, if technically feasible, minimized long term maintenance. The team recommended concurrently pursuing an engineering evaluation of a series turbo-scavenger option as a back up if the pure turbocharging approach was found to not be technically feasible[3].

The team recognized that the key constraint to pure-turbocharging the LTC engine is that the brake mean effective pressure (BMEP) is significantly lower than other turbocharged engines. This constraint translates into the following physical facts:

1. Engine:
    a. The low BMEP indicates a relatively low trapped fuel mass in the cylinders;
    b. The ignition system is designed to ignite an air/fuel mixture that is greater than a specific lean limit;
    c. The relatively low OEM boost level of about 6 inHg created by the scavenger pump most undoubtedly created a specific trapped equivalence ratio within the engine cylinders;
    d. This trapped equivalence ratio was in the range where combustion would occur with the OEM ignition system;
    e. Increasing the boost level traps additional air inside the cylinders, but the trapped fuel mass does not increase. This leads to a leaner trapped equivalence ratio. At some increased boost level, the trapped equivalence ratio would decrease (become more lean) to a point where the ignition system would not reliably ignite the mixture;
2. Turbocharger
    a. The sustainability of any turbocharger is a function of turbocharger efficiency, boost level, turbine inlet temperature, and the pressure decrease across the engine cylinders;
    b. Typical emissions-constrained turbocharged engines have a boost level that represents at least a factor of 1.7 times barometric pressure. This is a boost level of <u>at least</u> 21 inHg, which is over three times the OEM boost level of the LTC engines;
    c. For a given turbocharger efficiency and pressure decrease across the engine, decreasing the boost level from, say, 21 inHg to 10 inHg physically requires an increased turbocharger turbine inlet temperature (blowdown/exhaust gas energy). This requirement is based on the thermo- and fluid-dynamic relationship between the engine and turbocharger components;
    d. The turbine inlet temperature is itself constrained by the NO_X constraint. Increased turbine inlet temperature (engine exhaust temperature) will lead to increased NO_X emissions.

These facts clearly point to a trade-off between

1. Increasing the engine boost and the ability to ignite the air/fuel mixture trapped in the cylinders; and

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[3] Due to an abbreviated project timeline, it was necessary to have a fully designed turbo/scavenge cylinder solution on hand in the event that pure-turbocharging proved to not be technically feasible.

2. Finding the minimum boost level where, with constrained blowdown/exhaust energy content, the turbocharger will sustain itself.

The engineering challenge facing the team is to determine the engine system design that will successfully operate within these trade-offs.

ScavengeTech investigated these trade-offs with the Turbocharger-Reciprocating Engine Computer Simulation software (T-RECS). The software package was used to estimate the blow-down energy, the largest unknown in the pure turbocharging option, for a range of engine operating conditions. Prior to conducting this investigation, ScavengeTech benchmarked T-RECS with LTC-8H field test data and subsequently used the results to develop a comprehensive turbocharger specification, including net efficiency.

The missing link for the complete turbocharger specification was the pressure differential between the air manifold and the exhaust manifold. This differential is paramount in specifying the turbocharger operating point. The solution was to measure the pressure decrease across an LTC cylinder at the Kansas State University National Gas Machinery Laboratory (NGML). The pressure differential across the cylinder liner provided additional information that was used to develop the turbocharger air specification.

Globe Turbocharger Specialties, the turbocharger supplier, used this specification to develop and assist in mapping a range of turbocharger builds at the NGML to ensure that the turbocharger satisfied and bracketed this specification.

The analyses completed by ScavengeTech indicated the engine would operate with a scavenging ratio of less than unity. While this would theoretically satisfy the emissions limits, the operable range of the engine between emission limit achievement point and lean misfire avoidance might have proven quite small with conventional open chamber combustion (OCC). Therefore, and concurrent with the above efforts, AETC continued to review the scavenging effects of low brake mean effective pressure (BMEP) engines.

AETC subsequently recommended procurement of screw-in pre-combustion chambers (SIPCC) as a fallback plan to increase the in-cylinder ignition energy. The SIPCCs would allow the engine to operate at the higher boost levels, which in turn create leaner air/fuel ratios, and expand the engine operational range. Operation at higher air/fuel ratios (higher boost) would also increase the range over which the turbocharger could self sustain. It was estimated that NO_X emissions would reduce to 25% of the permitted limit and combustion stability would improve, allowing for reduced CO production.

January 2006 Engineering Review

El Paso Corporation convened a review of AETC and ScavengeTech's analysis in January 2006. This meeting included EPC Engineering, Reliability, and Field-Operations personnel. Hoerbiger also attended as the contractor employed for mechanical implementation.

El Paso Corporation Plant Services personnel presented the highlights of the November 2005 meeting. ScavengeTech and AETC then presented the results of the analysis. The results suggested that a pure turbocharger option, using a modified Globe-produced Alco 730 turbocharger, would work based on reasonable estimates of engine pressure drop, blow-down energy, and turbocharger efficiency. At this point, the cylinder liner tests had not been completed at the NGML. Based on this presentation the assembled team decided to proceed with the pure turbocharger option. The team also decided to terminate further development of liner and piston modification or other engine redesign and to proceed with OEM liners and pistons.

AETC reported that recent experience with engines operating at relatively low scavenge ratios resulted in a very narrow operating range between detonation and lean misfire. AETC therefore proposed maintaining a SIPCC or MPCC back up. Again, the addition of high-energy ignition would allow the engine to (1) operate at leaner air/fuel ratios; (2) expand the range of stable engine operation; and (3) permit the higher boost levels to prevent unsustainable turbocharger operation.

At the conclusion of the meeting EPC staff decided to conduct a pilot test of the pure turbo option that, if required, included MPCCs or SIPCCs. Based on the status of the various engines, the team decided to restore Unit 9A (which had already been converted by Digicon) to OEM pistons and liners and install the modified Globe turbocharger. The team also directed Hoerbiger to develop a design for the series turbocharger-scavenger pump option as a fallback.

The team conducted the pilot test in March 2006 to assess the feasibility of operating the engine in a pure turbocharger mode with OEM pistons and liners as described herein.

This report summarizes the results of that testing after first reviewing the unique aspects of performance modeling with T-RECS and testing of the turbocharger and engine liners at the NGML.

T-REC Simulation

This section of the report explains and documents the T-RECS calculations that were conducted to develop the air specification for the LTC engines.

Goals of Simulation

The specific goal of the simulation effort was to fully understand the fundamental physical relationships that govern the operation of the LTC engine. This understanding would then be used to fully specify the turbocharger operating point, the blowdown and exhaust gas energy, the boost level, the in-cylinder trapped equivalence, and the air flow rate through the engine. Once these parameters were determined, the only remaining effort was to assemble the necessary engine system components as characterized by the modeling effort.

Basic Model Description

The T-RECS program separates the engine system into the six components that are illustrated in Figure 1. The components are: 1) air filter; 2) compressor; 3) aftercooler; 4) engine; 5) turbine; and 6) silencer. The methodology used to develop T-RECS takes into account the interaction of all the components. The T-RECS program solves for the temperature, pressure, and flow rate at various points in the turbocharged engine system. In all cases, the mathematical algorithms used to determine the pressure, temperature, and flow rate are based on fundamental engineering principles to provide the highest reliability in the calculated values. The three fundamental principles that are used throughout the calculations are:

1) Energy conservation;
2) Mass conservation; and
3) Compressible flow.

The following discussion provides a high-level overview of the algorithms that are used to calculate the specific values and of the overall solution scheme.

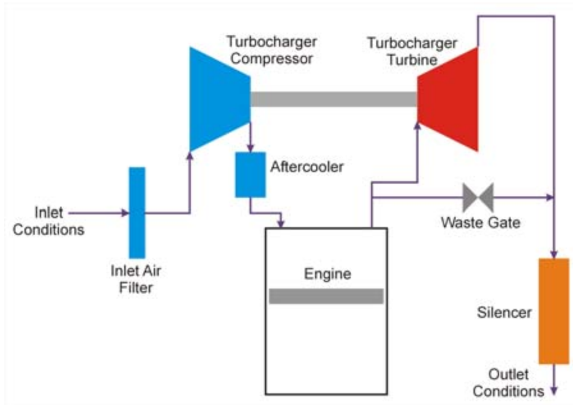

Figure 1: Engine system.

Solution Scheme

The premise of the T-RECS algorithm is that each component can be treated as a box that accepts an input stream and transforms it into an output stream. As an example, Figure 2 illustrates an aftercooler that is positioned between the turbocharger compressor and the air intake manifold. As shown in the figure, the aftercooler accepts an input stream of air at a certain pressure and temperature from the turbocharger compressor. The aftercooler then processes this input stream into the output stream by establishing the outlet temperature and pressure for the given air flow rate.

Mathematically, the process can be generalized as:

$$\begin{aligned} T_{out} &= f_{AC,T}\left(\dot{m}_a, T_{in}\right) \\ p_{out} &= f_{AC,p}\left(\dot{m}_a, p_{in}, \text{geometry}\right) \end{aligned} \tag{1}$$

In this general model, the functions $f_{AC,T}$ and $f_{AC,p}$ are the processes that transform the inlet information into the outlet information. In this particular case, the temperature processor needs information on the air flow rate, the inlet temperature, and the heat removal rate of the aftercooler. The pressure processor requires the inlet pressure, the air flow rate, and information about the geometry of the aftercooler.

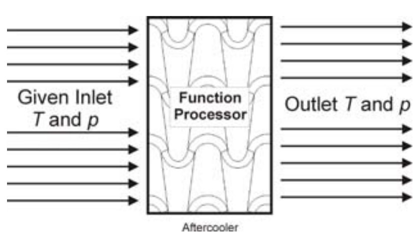

Figure 2: Aftercooler modeled as a processor.

By developing a processor for each component in the turbocharged-engine system, a set of *N* equations with *N* unknowns can be written. This equation set is iteratively solved for the pressure, temperature, and flow rate throughout the system. The iterative solution scheme is illustrated in Figure 3. Each rectangle in the diagram represents a sub-model processor that takes the input information and transforms it into the output information.

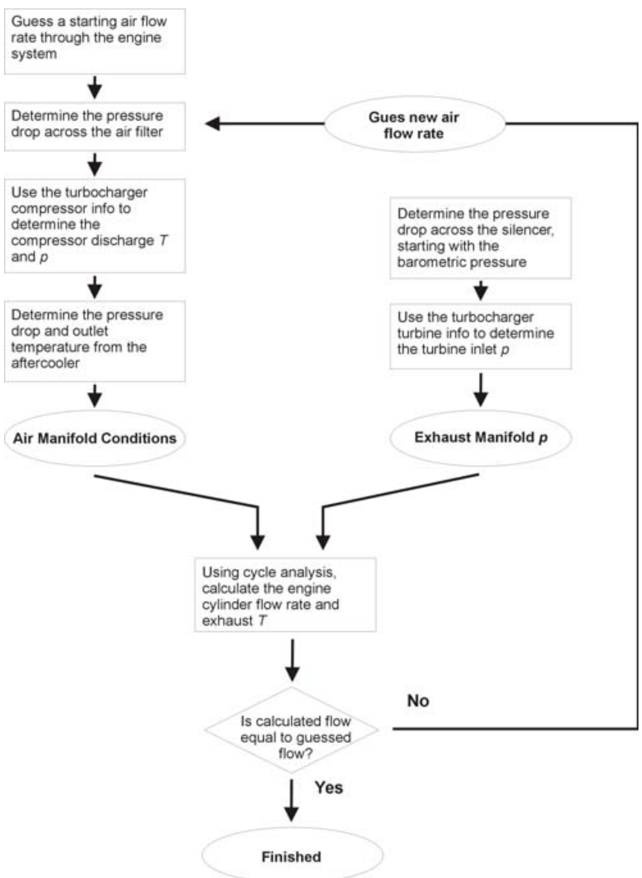

Figure 3: Iterative solution scheme.

The iterative solution scheme begins with a guessed flow through the engine. The guess is an intelligent guess based on the trapped cylinder volume and a nominal delivery ratio. The guessed flow, coupled with the ambient conditions, then determines the exit conditions of the air filter. The air filter exit information is used as the input to the turbocharger compressor processor, which produces the compressor output temperature and pressure. Continuing the iterative solution, this information serves as the input to the aftercooler processor, and then the output is used as the air manifold conditions. At this point, the engine inlet temperature and pressure are now determined for the guessed air flow rate.

Starting at the exhaust pipe, this procedure is repeated for the exhaust system. The total gas flow rate is now the sum of the guessed air flow rate and the added fuel. Given the barometric pressure, the silencer inlet pressure can be determined. Using the engine exhaust temperature from the previous iteration and the information about the turbocharger turbine, the turbine inlet pressure is determined. At this point, the exhaust manifold pressure is determined, and the engine cylinder is now bounded with the pressure and temperature of the air manifold, and the pressure of the exhaust manifold.

T-RECS Pressure Drop Models: The Processors

Several components in the engine air and exhaust flow path behave fluid dynamically as a pressure reducing component. For example, the flow rate through the air filter causes a very predictable pressure drop across the filter. The pressure drop is a function of the air flow rate through the filter and the flow geometry of the filter.

In general terms, the air and exhaust flow rate through any flow restriction is governed by the fundamental compressible flow equation. This equation is commonly written as:

$$\dot{m} = \frac{C_d A_R p_1}{\sqrt{RT_1}} \left[\frac{2\gamma}{\gamma - 1} \left(\frac{p_2}{p_1} \right)^{\gamma/2} \left[1 - \left(\frac{p_2}{p_1} \right) \right]^{(\gamma-1)/\gamma} \right]^{1/2} \quad (2)$$

In this equation, the subscripts 1 and 2 represent the temperature and pressure upstream and downstream of the flow restriction, respectively. The discharge coefficient, C_d, and the reference area A_R represent the characteristic curve that defines the relationship between the flow rate through the restriction and the pressure drop across the restriction. In short, the product C_dA_R is unique to each flow restriction, whether that restriction is a filter, heat exchanger, exhaust silencer, etc.

In the T-RECS program, the compressible flow equation is used to determine the flow and pressure drop across the air filter, the aftercooler, and the exhaust silencer. The procedure is the same for each, and is described in the following steps:

1) Determine a data point that describes the flow rate through the restriction and the pressure drop across the restriction. This data point comes from the manufacturer of the component, and must include:
    a) the flow rate through the component
    b) the pressure drop across the component
    c) the inlet temperature and pressure
2) Determine the geometric flow factor $C_d A_R$ for this component by using the data point determined in the previous step and then solve for the $C_d A_R$ term:

$$C_d A_R = \frac{\dot{m}\sqrt{RT_1}}{p_1} \left\{ \frac{2\gamma}{\gamma - 1} \left(\frac{p_2}{p_1} \right)^{\gamma/2} \left[1 - \left(\frac{p_2}{p_1} \right)^{(\gamma-1)/\gamma} \right] \right\}^{-1/2} \tag{3}$$

Once the data point and the geometric constant are determined, the component is fully characterized and can be used to solve for the flow rate through the turbocharged engine system.

The Engine Processor

With the manifold information now known, a complete cycle and flow analysis is completed at each crank angle. The cycle analysis covers two phases. The first phase is the gas exchange process when the ports are uncovered, and the second phase occurs when the ports are covered (compression, combustion, and expansion). The necessity of the cycle analysis calculation is to determine the blowdown and exhaust gas energy that is available to the turbocharger turbine. Without this information, the turbocharger specification is a complete guess.

The purpose of the gas exchange process is to: 1) eliminate the burned gas from the cylinder; and 2) admit the fresh charge into the cylinder. For the T-RECS simulation, the flow through the ports is determined using a quasi-steady model. The quasi-steady modeling process determines the air flow rate through the ports at some incremental crank angle. The result is the air and exhaust flow rates into and out of the engine cylinders as a function of crank angle. The process makes use of the compressible flow equation. The ports are treated in much the same way as the air filter. The information that is required is the reference area, the discharge coefficient, the inlet information, and the cylinder pressure. The discharge coefficient and the reference area are determined from experimental data.

The flow rates through the intake and exhaust ports, as well as the flow rate through the cylinder, define the scavenging process. During the scavenging process, the intake and exhaust processes occur almost simultaneously. With an auxiliary pumping device, the fresh charge is pressurized so that it enters the cylinder through the intake ports and displaces the burned gas into the cylinder to the exhaust port.

The scavenging process initiates when the exhaust ports are uncovered by the piston during the outward piston stroke toward bottom dead center (BDC). Because of the cylinder pressure is higher than the exhaust manifold pressure, burned gas flows into the exhaust port. This initial exhaust phase, commonly referred to as the blowdown phase, depressurizes the cylinder. As the piston moves further toward BDC, inlet ports are uncovered and, if the air manifold pressure is greater than the cylinder pressure, fresh air flows into the cylinder.

The scavenging process includes complex in-cylinder flow events. At the early stage of a scavenging process, a pure displacement of burned gas from the cylinder occurs. The mixing between fresh charge and burned gas at their interface also takes place. A scavenging process is an imperfect gas exchange process. First, a portion of fresh charge can be driven out through the exhaust port without being used. This phenomenon is called short-circuiting. Second, some other portion is mixed with the burned gas that would either remain in the cylinder or would flow out the exhaust port. Finally, a part of the burned gas could stagnate totally inside the cylinder forming so-called pockets or a dead zone. From a modeling perspective, it is difficult to accurately model the scavenging process.

Instead of directly modeling the fluid dynamics during the scavenging process, measurable parameters are typically correlated to the scavenging efficiency. More than one parameter is often utilized to indicate the overall effectiveness of a scavenging process. The scavenging ratio, often denoted by Λ, is one of these parameters and describes the ratio of total fresh charge mass delivered to the cylinder per cycle to the mass of fresh charge that would occupy the cylinder trapped volume at air manifold conditions:

$$\Lambda = \frac{\text{Mass of delivered fresh air}}{\text{Trapped Volume} \times \text{Air Manifold Density}} = \frac{m_{a,in}}{V_d \rho_a} \tag{4}$$

The scavenging efficiency, η_{sc}, indicates the portion of the burned gas replaced by the fresh charge:

$$\eta_{sc} = \frac{\text{Mass of delivered fresh charge retained}}{\text{Mass of trapped cylinder charge}} = \frac{m_{re}}{m_{tr}} \tag{5}$$

The mass of trapped cylinder charge shown in the denominator includes both fresh charge and residual gas. These two parameters are most frequently used with regard to the performance of the scavenging process.

Several scavenging efficiency models are listed in the literature. The model used in the T-RECS simulation is the perfect mixing model. This model correlates the scavenging efficiency to the scavenging ratio, and is mathematically written as:

$$\eta_{sc} = 1 - \mathrm{e}^{-\Lambda} \tag{6}$$

The trapped cycle analysis calculations focus on thermodynamic-based in-cylinder models. These models compute the cylinder pressure and energy transfer between the piston and working fluid as well as gas states that could be utilized for emission calculations. Simultaneously solving the conservation equations of mass and energy is required. The information on the mass transfer into and out of the cylinder during the gas exchange process, heat transfer between the cylinder gases and the surrounding engine components, and the rate of energy release from the fuel is necessary to complete the simulation. These models successively calculate the changes in thermodynamic and chemical states of the trapped gases throughout the compression, combustion, and expansion phases of the trapped processes.

The combustion process is the most complicated process of the all, and it is also the least understood (Watson and Janota, 1982). Therefore, the combustion process simulation involves the most complex computations. Many attempts to develop combustion models have been conducted. The complexity for such models varies significantly and depends on the purpose and result in different accuracies. For spark ignition engines, the spark discharge is the commencement of the combustion process. The simplest approach has been a one-zone model that assumes a single thermodynamic system for the entire combustion contents, and the rate of energy release from the fuel is obtained by experimentally derived functions that become part of the simulation input (Heywood, 1988). For more sophisticated simulations, quasi-geometric modeling of the combustion chamber gas as two or three zones of unburned and burned gases is employed (Heywood, 1988). The burned mass fraction with respect to advancing crank angle is derived from a first law analysis of the pressure history actually measured on the engines. Such data is only valid for a particular engine and the operating condition from which the pressure history was obtained. The technique is utilized to establish a collection of heat release data for different engine operating conditions as well as different engine types (Watson and Janota, 1982). For the engine simulation purpose, one way often used to represent this burned mass fraction is the Wiebe function (Heywood, 1988):

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{n+1}\right] \tag{7}$$

where, x_b is the burned mass fraction, θ is any given crank angle during combustion, θ_0 represents the beginning of combustion in crank angle, and $\Delta\theta$ is the total combustion duration. The variables a and n are the parameters that fix the shape of the combustion profile and they are empirically acquired. The T-RECS simulation uses the Weibe function approach coupled with a three-zone model. The conservation equations of mass and energy are applied individually to the burned, unburned, and boundary layer gas zones where mass and energy transfer from one zone to another.

The result of the cycle simulation represents the crank angle-resolved pressure, temperature, burned and unburned mass fractions, and gas exchange rates for the cylinder. To see how these principles work together to determine the flow rate and exhaust temperature from the engine, sample cycle analysis output data is illustrated in Figure 4. The data in this figure are the results from a Cooper GMV engine. The top panel shows the unburned and burned gas temperature, and the in-cylinder pressure. Of note is the increase in the temperature of the unburned gases shortly after top dead center (0°). This increase is due to the increase in pressure of the cylinder that actually compresses the unburned gases, resulting in the temperature increase. The blowdown phase begins with the uncovering of the exhaust ports at approximately 100° after TDC. The gas temperature decreases rapidly due to the rapid decompression of the gases in the cylinder.

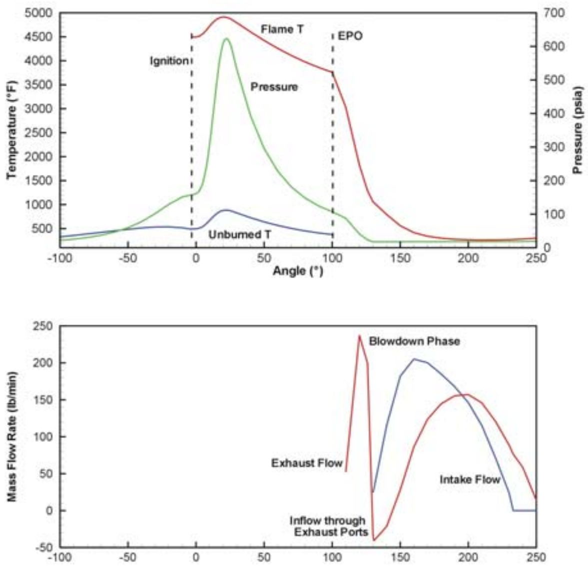

Figure 4: Cooper GMV cycle analysis results.

As importantly, the cylinder pressure is not significantly affected until the intake ports are uncovered. This is predominantly due to the fact that the blowdown flow through the exhaust port is at, or very near, sonic conditions, and is, therefore, choked. Consequently, the cylinder pressure during the blowdown phase is unaffected by the pressure in the exhaust manifold.

The lower panel shows the inflow and outflow of gases through the intake and exhaust ports. The exhaust flow exhibits a rapid spike in the flow rate as the port opens that is characteristic of blowdown. The blowdown phase ends with the opening of the intake ports, at which time exhaust gases actually flow back into the cylinder from the exhaust manifold (identified as negative flow).

Overall, the relatively simple cycle analysis simulation provides a substantial amount of information about the processes and events that occur within the engine cylinder. Of prime importance for this project is that the cycle analysis readily provides the exhaust gas temperature, blowdown conditions, and the air flow rate through the engine cylinders.

Engine Turbocharger Performance Matching

Typically an engine operating point is defined based on the air flow through the engine and air manifold pressure (AMP). However, one must assess the thermodynamic feasibility of actually attaining the desired performance point. This requires a turbocharger performance model as described in the following paragraphs.

The turbocharger utilizes a turbine wheel and housing to transfer energy from a hot gas stream into rotational energy to drive a compressor wheel. The compressor wheel raises the pressure of a separate inlet gas stream from the supply pressure to an elevated pressure. On the turbine end, the fluid imparts angular displacement on the rotor as it flows through the turbine blades. This displacement in turn is transmitted through a shaft to the centrifugal compressor wheel where the rotor imparts both velocity and a radial displacement to the fluid. The main components of the centrifugal compressor are the impellers where work is done on the fluid, and the diffuser where the velocity decreases and static pressure increases (Bathie, 1996). The isentropic efficiency of turbocharger components compare the actual work transfer to that which would occur in an ideal process. For this particular application, the ideal process is considered one that is isentropic and adiabatic.

The isentropic efficiency of the compressor is the ratio of the isentropic work to the actual work. The isentropic work is the amount of work required to ideally raise the pressure from inlet conditions to the exit pressure. Conversely the amount of work required to perform the same pressure change in a real machine is the actual work. The isentropic power is always lower than the actual power. Mathematically, the isentropic compressor efficiency is defined as:

$$\eta_{comp} = \frac{\dot{W}_{ideal}}{\dot{W}_{actual}} \tag{8}$$

The isentropic efficiency of the turbine is the inverse equation (8).

In order for a turbocharger to operate, the power produced by the turbine must precisely match the power consumed by the compressor and any losses what occur within the turbocharger. Consequently, by using three separate control volumes, a complete analysis of the turbocharger can be performed:

Control Volume 1 – Compressor:

$$\dot{W}_{comp} = \dot{m}_{air}\left(h_{in} - h_{out}\right)_{comp} \approx \dot{m}_{air} c_p \left(T_{in} - T_{out}\right)_{comp} \tag{9}$$

Control Volume 2 – Turbine:

$$\dot{W}_{turb} = \dot{m}_{exh}\left(h_{in} - h_{out}\right)_{gas} + \dot{m}_{oil}\left(h_{in} - h_{out}\right)_{oil} + \dot{m}_{water}\left(h_{in} - h_{out}\right)_{water} + \dot{Q} \tag{10}$$

Control Volume 3 – Entire Turbocharger:

$$\dot{W}_{turb} = \dot{W}_{comp} + \dot{W}_{mechanical} \tag{11}$$

The mechanical losses are commonly expressed in terms of the mechanical efficiency:

$$\eta_{mech} = 1 - \frac{\dot{W}_{losses}}{\dot{W}_{turb}} \rightarrow \dot{W}_{losses} = \dot{W}_{turb}\left(1 - \eta_{mech}\right) \tag{12}$$

Substituting equation (12) into equation (11) provides the final equation necessary to determine turbocharger performance:

$$\dot{W}_{turb} \times \eta_{mech} = \dot{W}_{comp} \tag{13}$$

At this point, equations (9), (10) and (13) provide the information that is necessary to analyze the turbocharger.

To determine if a given turbocharger will provide the desired performance, one proceeds through the following steps:

1. Set the compressor and turbine isentropic efficiencies, and the turbocharger mechanical efficiency from performance maps and other available data;
2. Calculate the compressor power required from the air inlet temperature and pressure, and the required AMP used in the T-RECS simulation;
3. Calculate the average engine exhaust gas temperature using T-RECS;
4. Calculate the power production required from the turbine;
5. Calculate the EMP that is necessary for the turbine to provide the power
    a. Determine the actual turbine discharge temperature
    b. Determine the necessary turbine inlet pressure (EMP)

   c. Compare this EMP with the minimum available based on the estimated pressure drop across the engine and the T-RECS calculations;

6. If the necessary EMP to drive the turbine is greater than that determined from T-RECS, then repeat the design process by selecting different turbocharger components.

T-RECS Upgrades

The T-RECS and turbocharger matching procedure explains how T-RECS computed results at the beginning of this project. Two major changes had to be incorporated into the T-RECS simulator for this project. The first was an improved scavenging efficiency model and the second was flexibility in using the actual fuel flow rate as an input into the simulator.

Scavenging Efficiency Model

Over the past three years, AETC and ScavengeTech developed an improved scavenging efficiency model that accurately represents the scavenging process in two-stroke cycle (2SC) engines used in the pipeline industry. Emissions of NO_X are controlled by the peak combustion temperature which is, in turn, governed by the heat capacity of the in-cylinder gases. For typical pipeline engines in which the fresh air and residual have similar thermodynamic characteristics the trapped charge density at the time of port closing governs the cylinder heat capacity. The challenge is determining the actual scavenging efficiency.

As noted earlier, under ideal mixing the scavenging efficiency is directly related to the scavenging ratio:

$$\eta_{se} = 1 - e^{-\Lambda} \tag{14}$$

Under non-ideal mixing, data further shows that:

$$\eta_{se} = 1 - \frac{T_{fresh}}{T_{blow}} e^{-\Lambda} \tag{15}$$

where:

T_{fresh} = Fresh Charge Temperature

T_{blow} = Blowdown Temperature

While T_{blow} is equally unknown, it is greater than T_{fresh}. This suggests scavenging efficiency can be modeled by:

$$\eta_{se} = 1 - k_1 e^{-\Lambda} \tag{16}$$

where k_1 is a curve fit parameter < 1.

To obtain k_1 the team reversed the problem using NO_X as a surrogate to determine the relative peak temperature and the trapped in-cylinder charge. Specifically, the team assumed that for a given "true" trapped equivalence ratio the engine would generate the same NO_X emissions regardless of speed. The value of k_1 can then be readily determined from existing data.

To support comparisons of scavenging performance between various engine models, the team further assumed that for the same "true" trapped equivalence ratio any 2SC engine with similar mixing characteristics would generate the same NO_X emissions. This requires the addition of a second curve fit parameter as follows:

$$\eta_{se} = k_2(1 - k_1 e^{-\Lambda}) \tag{17}$$

where k_2 is a second curve fit parameter that is approximately 1.

In this formulation k_1 defines the relative scavenging efficiency for a given scavenging ratio and k_2 adjusts the scavenging efficiency between engines relative to each other and relative to ideal mixing. With proper k_1 and k_2 values, the scavenging efficiency can be accurately determined.

Fuel Flow Rate Input

The original version of T-RECS determined the mass of trapped fuel using an input trapped equivalence ratio and the computed trapped air flow rate. The program was modified so that it now accepts either the trapped and corrected equivalence ratio or the fuel flow rate. The details of this effort were entirely based on computer programming changes and are, therefore, not included.

Simulation Results-Turbo Specification

The simulation results were generated in two sequential phases. Specific parameters were tuned in the first phase and the turbocharger air specification was developed in the second phase.

Tuning Phase

The T-RECS simulator was first tuned with existing data prior to using it as an engineering tool to develop the turbocharger specification. Blower data was collected from a test conducted on Unit 6A at Station 32. During this test, the scavenging pump was bypassed by connecting a Roots blower directly to the LTC air manifold. A series of tests were conducted to determine if emissions compliance could be achieved using a motor-driven blower instead of turbocharging the engine. While the motor-driven blower concept was deemed infeasible because of the electrical power supply cost, the test provided the needed engine data to tune. Several data points were averaged to create a tuning “target.” Averaged data, shown in Table II, then were selected to tune T-RECS.

The results of the T-RECS tuning are provided in Table III. The Weibe coefficients, combustion

Table II: Engine test parameters used to tune T-RECS.

Parameter	Value	Parameter	Value
AMP	5.99 inHgG	Fuel Flow Rate	158.68 ft^3/min
AMT	113.42°F	Delivery Ratio	1.15
Air Flow Rate	5.68 lbm/s	Peak Pressure	~350 psig
Engine Speed	301 rpm	Location of Peak Pressure	~22°ATDC
IT	10°BTDC	Engine Δp	5.3 inHgG
LVH	900 Btu/ft^3		

Table III: Key engine parameters resulting from the T-RECS tuning procedure.

Fixed Parameters	Value	Parameter	Value
AMP	5.99 inHgG	IMEP	83.11 psi
AMT	113.42°F	Delivery Ratio	1.16
Fuel Flow Rate	158.68 ft^3/min	Peak Pressure	352.5 psig
Engine Speed	301 rpm	Location of Peak Pressure	22°ATDC
IT	10°BTDC	Trapped Eq. Ratio	0.802
LVH	900 Btu/ft^3	Air Flow Rate	5.68 lbm/s

duration, and port flow coefficients were fine-tuned to achieve these results. The "Fixed Parameters" on the left side of the table are boundary conditions that were not varied during the tuning process. The parameters on the right side of the table are calculated values that can, in most cases, be compared to the Roots blower field test data listed. All the calculated values compare within 0.5% of the measured averaged parameters in Table III.

For demonstration purposes, Figure 5 shows the in-cylinder pressure, mass in-flow rate, and exhaust flow rate as functions of crank angle. These data were generated from the tuning process. The intake and exhaust port open/close events are designated by the vertical dashed lines. Of particular interest is that the in-flow of air is negative, which implies that there is

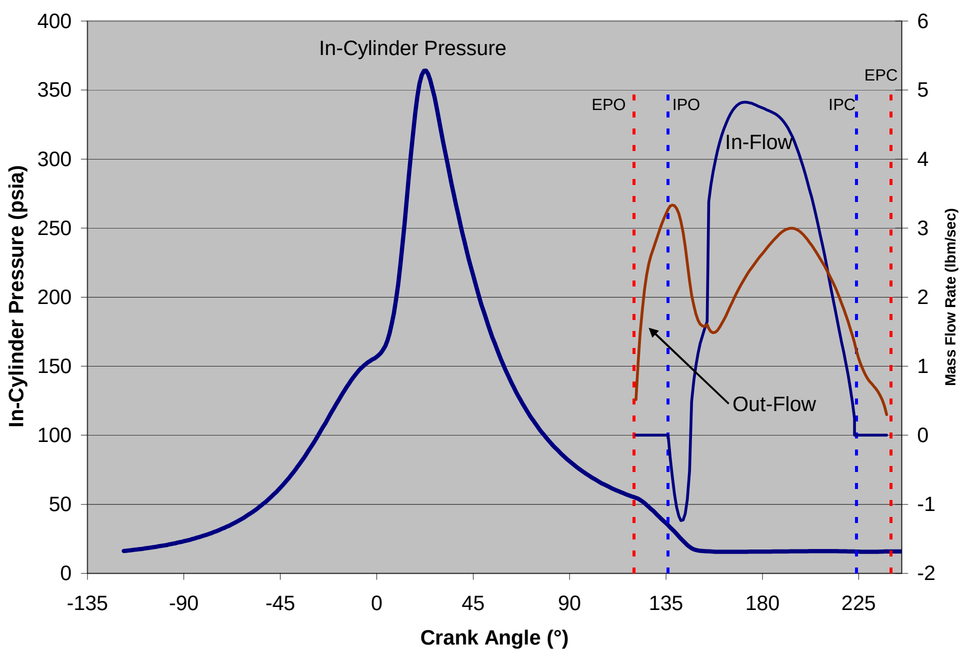

Figure 5: Pressure versus crank angle for the Roots blower-driven LTC configuration.

actually an outflow of exhaust products for the first few crank angles following the IPO event. One can also observe the high rate of exhaust port out-flow during the blowdown phase, which occurs between the EPO and IPO events. Once the IPO event occurs, the in-cylinder pressure decreases very quickly, which in turn decreases the exhaust port out-flow rate.

Phase 2: Developing the Air Specification

The T-RECS simulation, tuned to accurately model the Worthington LTC engine, was used to develop the turbocharger specification. The boundary conditions were changed to simulate the case where a turbocharger was installed on the engine.

The calculations were completed to determine the following:

1. The lowest turbocharger component efficiencies that result in sustainable turbocharger operation;
2. The boost level (AMP) that reduces the trapped equivalence ratio to 0.60. At this point, expectations are that the engine would begin to misfire;
3. Determine the compressor pressure ratio, turbine inlet temperature, and air mass flow rate that represents the turbocharger design point; and
4. Determine if the turbocharger can sustain itself at conditions other than full-torque, full-speed. Self sustaining is defined as an operating point where the turbocharger turbine power is greater than the turbocharger compressor power. This balance depends on the AMP, EMP, and EMT, among other parameters.

The one missing piece of information was the pressure reduction across the cylinder liner as a function of flow rate through the liner. For these calculations, a pressure reduction of 2 inHg was used. This value was determined by DigiCon personnel during the previous effort to upgrade these engines.

During the series of calculations, the AMP was varied from a low of 8 inHg to a high of 20 inHg.

Field Test Overview

An overview of the field test follows.

Data Collected

AETC provided its transportable Continuous Emissions Monitoring System (CEMS) and staff to monitor:

- NO_x
- O_2
- CO

CEMS data was automatically logged into AETC's mapping program.

AETC automatically recorded primary engine parameter data from the PLC into the mapping program including:

- AMP
- AMT

- Speed
- Fuel Flow
- Load
- IT

ScavengeTech installed its Turbocharger Monitoring System (TuMS) on the test turbocharger. This system automatically recorded, logged and transmitted via TCP/IP the following parameters to AETC's mapping program:

- Compressor inlet temperature
- Compressor inlet vacuum
- Compressor discharge temperature
- Compressor discharge pressure
- Turbocharger speed
- Turbine discharge temperature
- Turbine discharge pressure
- Turbo bypass position (manually recorded)
- Air flow

AETC installed Portable CPM™ system to measure the following:

- Power cylinder combustion pressure
    - Pressure versus crank angle
    - Peak pressure
    - Location of peak pressure
    - Standard deviation of peak pressure
- Intake and exhaust manifold pressure
    - Pressure versus crank angle

EPC provided a local technician to monitor the indicated compressor power.

Test Phases

A review of the roles and responsibilities of the pilot test follows.

Commissioning and Feasibility Test

The team first commissioned the engine and the turbo in the OCC configuration. This included:

- Motoring the engine on starting air
- Motoring the turbocharger on jet assist
- Starting and idling the engine and turbocharger
- Loading the engine by closing the bypass
- Progressively operating the engine over its speed and toque range

The test procedure included a number of detailed fall back tests if the turbocharger could not self sustain, all of which proved unnecessary. Since the engine had new pistons and liners commissioning was interspersed with the normal break in heat checks.

EPC site staff operated the engine according to their standard practice. The Hoerbiger lead engineer was responsible for directing all aspects of the *feasibility* testing including start, idle, load and expansion of the operating range.

Globe and ScavengeTech staff were on-site for the feasibility portion of the test to compare the collected data with NGML test stand data. They were responsible for assessing the likelihood of the turbocharger self-sustaining and offer recommendations for turbocharger modifications as required.

A representative of EPC Plant Services observed the testing, set final priorities and resolved unanticipated issues which arose.

AETC engineers were responsible for collecting all data and compiling it for immediate review by the team. AETC's engine experts were available via phone to assist with data review and analysis as required.

Optimization Test

AETC directed the optimization test. AETC provided an engineer to manage the testing and a second individual to operate the transportable CEMS and record manual data. A Hoerbiger engineer coordinated with the plant and provided assistance as required. EPC reliability staff collected compressor output data. A description of the mapping conditions follows.

<u>Mapping and Optimization – OCC</u>

The team mapped the engine over the full range of operation according to AETC's standard practices. The map included (not necessarily in order) the conditions shown in Table V.

Points 1-3 established the NO_x versus trapped equivalence ratio (Φ) curve. Points 1-5 established the engine air flow line and scavenging efficiency. Points 6 and 7 verified timing sensitivity. Points 8 and 9 further verified and established the uncertainty in the NO_x versus Φ curve. Points 10 and 11 confirmed the scavenging efficiency accurately captured AMT effects.

<u>Mapping and Optimization –SIP</u>

The test plan included SIP's as a contingency, particularly if the turbocharger could not self-sustain at <10inHg AMP forcing operation at higher boost levels resulting in equivalence ratios which were too lean to light with conventional OCC operation. While turbocharger self sustenance did not prove to be a problem, the CO levels remained higher than desired. Therefore, the team elected to install the SIP's and test over a limited map to evaluate the CO impact. In addition to the normal parameters, the map included tests of varying SIP fuel supply pressure.

Table V: Mapping test conditions.

Point	Speed	Torque	Equivalence Ratio	AMT	IT	Comments
1	100%	100%	Nominal	Nominal	Nominal	~8 g/BHP-HR NO_x or normal
2	100%	100%	Richer	Nominal	Nominal	~12-15 g/BHP-HR NO_x
3	100%	100%	Leaner (Max Boost)	Nominal	Nominal	~6-8 g/BHP-HR NO_x
4	~95%	100%	Nominal	Nominal	Nominal	Or halfway between rated and
5	~90%	100%	Nominal	Nominal	Nominal	Or minimum
6	100%	100%	Nominal	Nominal	±1-2°	IT test
7	100%	100%	Nominal	Nominal	±3-4°	IT test
8	100%	100%	Intermediate Rich	Nominal	Nominal	Between points 1 & 2
9	100%	100%	Intermediate Lean	Nominal	Nominal	Between points 1 & 3
10	100%	100%	Nominal	High	Nominal	AMT test
11	100%	100%	Nominal	Low	Nominal	AMT test

Field Test Results

The pure turbocharger test data was compared with the January and February 2003 testing with the blower since this best reflects the effects of the reduced parasitic load.

Unit Operability

Unlike Digicon's verbally reported experience, the team encountered no combustion performance problem operating the unit in the pure turbocharged mode with the configuration based on the T-RECS simulation. The unit started, idled and accepted load with no problem. The engine neither backfired nor stalled. None of the contingency testing prepared for turbocharger performance was required.

Data Quality and Integrity

The air flow ratio[4] exhibited excellent monotonic trends as a function of AMP with reasonable exponential coefficients (~0.6). This validates the fuel flow measurement and the oxygen measurement. The SIP data exhibited slightly greater scatter probably reflecting changes in fuel consumption and/or fuel slip for varying SIP supply pressures.

The OCC CO versus NO_x trade-off for the March 2006 data exhibits an unusual non-asymptotic second-order shape with minimum at about 3 g/bhp-hr NO_x. The January 2003 data exhibited a

[4] Based on an oxygen balance.

similar shape, but over a much larger range of NO_x levels, typical of swinging from “rich[5]” to “lean” combustion on 2SC engines. The March 2006 OCC data may indicate the addition of the turbocharger and the associated reduction of scavenging ratio has shifted and shrunk the range of operable air/fuel ratios.

Data scatter for the OCC test is a slightly higher than normal and significantly higher than normal for the SIP testing. Presumably this also reflects the effects of changing SIP supply pressure. The BSFC versus NO_x trade-off curve for the OCC data exhibits the same unusual second order shape with some scatter. The SIP data exhibits less scatter but an inverted shape.

In summary, the data quality is acceptable to excellent with some scatter in the SIP data probably due to changing fuel supply pressure. Some characteristics are unusual, but as will be seen probably reflect the effects of sub-unity scavenging ratio.

Trapped Equivalence Ratio Effects

Due to the substantial change in back pressure when adding the turbocharger, AETC could not utilize its standard methods to develop a credible estimate of scavenging efficiency. Therefore, only data for the “theoretical” trapped equivalence ratio is shown. While this significantly overestimates the leanness of the mixture, it does offer some insights into engine performance.

NOx

When trended as a function of trapped equivalence ratio (Φ), NO_x data for the March 2006 OCC and SIP tests largely coincide with some significant scatter. This indicates the same main chamber phenomenon governs NO_x emissions for both configurations. This is typical. For a given Φ, NO_x levels in the pure turbo configuration are about a third lower than the pure blower configuration. This might suggest scavenging efficiency improved in the pure turbo configuration. In view of the sub unity scavenging ratio, this seems unlikely. More probably, the low scavenging ratio significantly increases the retained residual (internal EGR) reducing the flame speed and peak temperature thereby reducing NO_x for the same theoretical Φ.

CO

CO exhibited an unusual second order trend with Φ with a local minimum. CO for the OCC pure turbo configuration was about 75% higher than for the pure blower configuration and the SIP data was higher yet. These results primarily indicate the Φ is not governing CO emissions for this particular engine configuration. It also shows SIPS have a detrimental impact on CO for this engine configuration.

Scavenging Ratio Effects

As noted, the subject engine operates at a low scavenging ratio. A description of scavenging ratio and then a discussion of the effects follows.

[5] While 2SC pipeline engines always have excess O_2 in the exhaust, operation above ~14 g/BHP-HR NO_x often exhibits characteristics of rich combustion like increased CO, increased BSFC and the tendency to detontate.

Theoretical Basis

The scavenging ratio compares the total air mass flow through the engine to the mass the engine can theoretically trap at cylinder filling conditions.

Most researchers quantify scavenging ratio (SR[6]) as:

$$SR = \frac{\dot{V}}{N \times V_{trap} \times \rho_{trap}} \tag{18}$$

where:

$\dot{V}$ = Air flow mass rate

N = Engine speed

V_{trap} = Volume at trapping conditions

ρ_{trap} = Maximum theoretical density at trapping conditions (based on the measured AMP and AMT).

In contrast to 4SC engines which "pump" fresh charge into the cylinder, 2SC engines must "blow" fresh air through the cylinder and rely on a combination of displacement and dilution to minimize the residual combustion gases trapped in the cylinder. This requires the engine receive considerably more fresh air than it can theoretically trap. Typical pipeline engines operate at *SR* ratios of ~120-150% and up to 200%.

Due to this low rating in the pure turbocharger mode, the subject engines operate at less than unity scavenging ratios. This means considerable residual gases remain trapped in the cylinder after port closer. While there sufficient oxygen is available to complete combustion, the residual gases alter the thermodynamics of the process. In addition, sub unity scavenging ratios make it extremely difficult to estimate the scavenging efficiency. Therefore AETC compared the emissions data directly with scavenging ratio.

NOx

NO_x does not trend well with scavenging ratio. However, the pure blower data trend suggests NO_x should decrease with decreasing scavenging ratio which is exactly were the NO_x for the pure turbocharger data falls. The overall trend indicates the reduced scavenging ratio contributes to the NO_x reduction, presumably due to the scavenging effect noted above.

CO

The March 2006 CO data trend reasonably well as a function of scavenging ratio. The OCC and SIP data nearly coincide, with the SIP data slightly higher. The pure turbo data almost appears to follow the trajectory of the pure blower data, which suggests that scavenging ratio controls CO for this engine. That is, the high fraction of retained residual slows flame speed reducing the time for CO to "cook" to completion and/or prevents local pockets of fuel from receiving sufficient oxygen to burn to completion.

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[6] Various researchers also call this quantity the delivery ratio denoted as Λ (lambda), however many utilize displaced mass flow instead of trapped mass flow, resulting in slightly different value ranges.

NO_x versusCO Trade-off

For the same NO_x level the pure turbocharged engine generates less CO in the OCC configuration than the SIP. The SIP relies on the main chamber for fresh air. Presumably the high fraction of residual in the main chamber forces the pre-chamber to operate richer than normal resulting in higher CO emissions for the same NO_x levels. For this particular situation, with a critical CO limit, the SIPS are surprisingly a detriment.

NO_x versus BSFC Trade-off

For the same NO_x level the pure turbocharged engine exhibits slightly better BSFC in the OCC configuration than the SIP. This could suggest the SIPS run too rich and possibly slip unburned fuel during the scavenging process. However the SIP air/flow data falls slightly below the OCC data indicating the SIP's are slipping less fuel if anything. Perhaps the enriching of the SIP's due to the low main chamber scavenging ratio results in a higher than normal NO_x contribution by the SIPS.

NO_x versus Combustion Stability Trade-off

"Combustion stability", that is the "smoothness" is an important factor in assessing an engine's reliability and range of operation. "Stability" is admittedly difficult to quantify and most measures are rather subjective. AETC has found the covariance (COV[7]) of peak pressure quantifies this subjective assessment reasonably well. In general, a COV above ~12% is often considered "rough" and COV's above 15% are considerable unacceptable. In general COV increases as NO_x decreases.

AETC does not have any pre-Digicon conversion COV data. Post conversion COV data recorded in October 2005 ranged from 9% at 10 g/BHP-hr NO_x to 15% at 7 g/BHP-HR NO_x. Stability data recorded in March and April 2006 remained surprising flat ranging from 12-14%. Installation of PCC's dropped the COV's to 10%. Rather surprisingly, plant staff did not consider stability during OCC operation at all objectionable. They considered the stability far superior to that obtained after the initial Digicon conversion and equal to, if not better than, OEM operation.

Meaningful comparisons of COV assume typical distributions of misfires, low fires and hard hits. AETC has previously noted that COV measurements made on conventionally fueled engines may not be directly comparable to those made on engine fitted with enhanced mixing. Conventional fueling seems to result it more misfires and hard hits. Enhanced mixing results in a slower, later burn but rarely results in a total misfire. Perhaps the low scavenging ratio and resultant high residual similarly results in a slower but more consistent burn. Either way, plant staff found OCC oepraiton acceptable and did not feel the improved stability offered by PCC's justified the maintenance headaches.

Miscellaneous

Part Load Performance

The team did operate the engine down to ~930 BHP (805 load). NO_x and CO were ~1 g/BHP HR and 2 g/BHP-HR respectively. BSFC was high (~9,500 BTU/BHP-HR) but acceptable and

[7] COV is the standard deviation of a parameter value divided by the average value.

COV's were quite acceptable at ~12%. Boost was about 11-12"Hg and probably represented the limit of turbocharger sustainability. However exhaust temperatures were ~925°F so some additional operating range might be available (see below).

TuMS Air Flow Measurement

This test provided the first opportunity to conduct a field validation of the TuMS pitot tube air flow measurement calibrated on the NGML test stand against the standard oxygen balance based air flow calculation. For the OCC data the oxygen balance based flow measurement coincided exactly with the TuMS pitot tube measurement. The oxygen balance based flow for the SIP test fell slightly below the pitot measurement. The SIP pitot measurement coincided with both OCC flow rates. Since the air flow characteristic for the engine cannot change there must be something wrong with the SIP oxygen based flow calculation. There discrepancy is within the experimental uncertainty. More importantly the pitot tube based measurement gives excellent results in the field.

Digicon Nozzles

During the test, the team attempted to operate the engine with a different Digicon design and the OEM nozzles. The purpose was confirm the validity of the current design and see if any further benefit could be gained. The engine operated poorly with either nozzle confirming the currently installed Digicon nozzle was the best optimized for this configuration.

Discussion

A discussion of the results follows.

Relevance of Findings

Pure Turbocharger Challenges.

The results clearly demonstrate the challenges of operating low BMEP engines in the pure turbocharger mode. In order to self sustain, the turbocharger must operate at higher boost levels than in a mechnaiclly scavenged mode (~12" Hg vs 6" in this case). The combination of natural orifice effects of the engine and available power for the turbocharger limit the air flow to approximately the level as obtained with mechanical scavenging. As a consequence, the low BMEP pure turbocharged engine runs at sub unity scavenge ratios. This means greater than normal residual remains in the cylinder. This residual[8] does not appear to pose a major operability or NO_x problems and in fact may help reduce NO_x. The low scavenging ratio does appear to limit the minimal achievable CO. Adding SIP's appears to compound the problem. Normally SIP fitted engines operate at even higher scavenging ratios than OCC engines to insure the SIPS receive sufficient air from the main chamber. The current results suggest if the main chamber contains too much residual the SIP's will run too rich, adding CO rather than reducing it.

It is not clear why Digicon could not successfully pure turbocharger the engine with OEM liners and pistons. Their sole attempt was probably with a turbocharger that was not properly specified

[8] Sometimes called internal EGR.

Attainable CO

The sub 1 g/BHP-HR CO levels measured in the "baseline with Digicon valves" configuration in November 2002 were recorded with a portable analyzer. The BSFC measurements indicate the readings taken at less than 0.5 g/BHP-HR CO reflect rich-like operation (15 g/BHP-HR NO_x). Recent testing with portables on rich burn engines fitted with NSCR catalysts suggests some form of measurement interference results in a 50% low CO reading under rich conditions. The 1 g/BHP-HR CO values recorded under leaner conditions (~7 g/BHP-HR NO_x) are probably more accurate. These are the levels achieved by the best mixing conventionally fueled Cooper engines operating at best economy. The January and February 2003 blower testing yielded similar 1-1.5 g/BHP-hr results.

Bo Mikkelsen reported achieving 2.5 g/BHP-HR CO, attributing the high levels to the overly lean condition required to achieve minimal NO_x (~1.5 g/BHP-HR) and the limitations of the MPCC's. In retrospect the even higher boost on his engine (18'Hg) reduced the scavenging ratio even lower than this Jasper conversion. His MPCC's probably ended up even richer resulting in the somewhat higher CO levels.

Review of Alternative Final Solutions

Upon completion and review of the March test results the team conducted several telephone conversations in which they reviewed all options to further reduce CO. A summary of those evaluations follows.

Re-Engineered Pistons and Liners

The team rejected development of new pistons and liners during the November 2005 review due to schedule, cost and risk. The test results validate that decision. The engine successfully operates in the pure turbocharged condition at well below the NO_x target. Re-optimized pistons and liners may improve the scavenging efficiency. However the low BMEP still precludes achieving even a unity scavenge ratio. The net improvement in CO, if any, would not justify the cost.

Series Blower-Turbocharger

In November 2005 the team deemed the series blower-turbocharger options "proven" based on Bo Mikkelsen's results in comparison to the riskier option of pure turbocharging. The team elected to pursue pure turbocharged first since it best leveraged the installed infrastructure and hardware purchased to date. This proved the correct choice. Bo's data indicates Series Blower-Turbocharging resulted in higher CO levels than achieved with pure turbocharging, probably due to the higher boost required for his option. It deserves no further consideration.

SIP's

Conventional SIP's rely on the main chamber for scavenging air. The high residual fraction in the main chamber due to the less than unity scavenging ratio for these low BMEP engines appears to result in inadequate SIP scavenging. Consequently adding SIP's increases CO. In theory, supplying the SIP with premixed air and fuel could correct the problem. OEM's have experimented with and provided such systems for high output four stroke cycle engines. The development costs for the Jasper engines would probably exceed $100,000 not including the hardware required for each unit without any certainty of successful CO reductions. This deserves no consideration.

Alternate Enhanced Mixing Combustion Technology

Prior work has demonstrated enhanced mixing can significantly reduce CO emissions. The current mechanical Digicon valves already offer some of these benefits. In the baseline configuration prior to conversion to the engine emitted 2.5-3.5 g/BHP-HR CO. Ignoring the specious November 2002 data, the Digicon valves probably reduced CO to ~1-1.5 g/BHP as reflected in the blower test. In addition, the team could not load the engine in the pure turbocharger mode with the OEM valve. These results suggest the Digcon valve generated some improvement. Switching to High Pressure Electronic Fuel Injection might result in some further improvement. On the other hand, the low scavenge ratio with its high retained residual might limit the further benefit of any enhanced mixing system. The hardware costs would be ~$250,000 per engine and deserves no further consideration.

Alternative Turbocharger Specification

A modern higher efficiency “marine” turbocharger would probably offer up to 5 percentage points of efficiency improvement. The increased efficiency would manifest itself as lower pressure drop across the turbine which might allow higher air flows at the same boost level. This would improve scavenging ratio, reduce the retained residual and possible reduce CO. The costs for new turbochargers and manifolding would exceed $250,000 per unit not including commissioning without any certainty of successful CO reductions. This deserves no further consideration.

The current turbocharger does exhibit limited performance margin, particularly on a hot day. A slight change in nozzle ring area (~1in^2 reduction) might improve the operating range without further compromising scavenging ratio. The incremental cost of the analysis and testing is probably less than $5,000. This option deserves analysis and, if favorable, on engine testing.

Catalyst

Utilization of oxidation catalysts to reduce CO mass emissions rates from current levels to below permitted levels would be an extremely low technical risk option. Catalytic reduction of CO, up to 93%, is a well proven technology for lean-burn engines.

Drawbacks to this option not only include increased operational maintenance costs associated with periodic hardware replacement and RICE MACT – type monitoring for ensuring proper catalyst operation, but also include additional compliance risks associated with utilization of an after-treatment for emission control. The incremental, turn-key capital costs of installation of oxidation catalysts are anticipated to be $75,000 per engine. This option remains open as the primary manner for further CO control, if needed.

Concluding Remarks

The key result from this project is that the software and hardware engineering tools that have been developed over the last several years by AETC, K-State’s NGML, and ScavengeTech, many through research supported by PRCI, can be decisively used to develop an engineered solution for a very difficult engine upgrade project. The project team brought to bear a multitude of tools to ensure that the final engine system specification was accurate and met the emissions requirements. While some may consider the use of these tools and the design time costly, the end result is that the engine operated the first time just as predicted by the project team. There were no hardware/field iterations necessary. The cost-savings in reducing the time spent on-site by the contractors greatly outweighs the cost of creating the engine system design.

A second key result is that the project time treated the engine, turbocharger, intake, and exhaust systems as one coupled engine system. For example, modifying a cylinder port impacts the performance of the entire system, not just the engine itself. Modifying a port affects the trapped mass inside the cylinder, which impacts the power produced by the engine as well as the NO_X and CO emissions. Additionally, the impact on the trapped mass affects the turbocharger turbine inlet turbine, which in turn affects the ability of the turbocharger compressor to provide air to the engine.

The moral of this story is that something as simple as grinding a small portion off the cylinder port can easily impact the entire engine system in a way the exceeds our intuitive abilities and rules-of-thumb pseudo-engineering techniques. To reach today's emissions target, we have to rely on fundamental physics-based engineering tools. When these types of tools are employed, there is no reason to suffer through field hardware iterations – and everyone sleeps much better at night.

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