TITLE:

U. S. DEPARTMENT OF ENERGY
AND
AIR PRODUCTS AND CHEMICALS, INC.

METHANOL FROM COAL END USE DEMONSTRATION PROJECT

WEST VIRGINIA UNIVERSITY

EMISSIONS AND OPERATIONAL ASPECTS OF METHANOL AS AN ALTERNATIVE FUEL IN A STATIONARY GAS TURBINE (LOTH/CLARK, 2000)

Final Contract Report

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ABSTRACT

During the past thirty years two major concerns have developed with our current fuels. These concerns are reliable supplies and pollution. Because of these problems there has been a great interest in alternate fuels such as alcohol and natural gas. Since 1997 research has been conducted at West Virginia University on methanol as an alternate fuel for gas turbines. There have been two main areas of study in this research, the problems associated with operating a gas turbine on methanol and exhaust gas emissions. There are two major differences between methanol and aviation kerosene that affect the operation of a gas turbine. The first is methanol's poor lubricating properties and the second is methanol's lower heating value. During this research techniques have been developed to measure the lubricating properties of methanol and various additives. Suitable lubricant additives were found to improve methanol's lubricity to equal that of aviation kerosene, with as little as 1% additive. The lower heating value of methanol required modifications to the WVU gas turbine's fuel system and atomizer, to provide higher flow rate of fuel then required with aviation kerosene. The gas turbine was modified and operated on methanol for an extended period, without failure. Exhaust gas emissions were tested for carbon monoxide (CO), carbon dioxide (CO_2), oxides of nitrogen (NO_x), total hydrocarbons (HC), and particulate matter (PM). During operation on methanol significant reductions in NO_x and HC emissions were observed. Without significant change in turbine inlet temperature, this observation can only be explained by a significant reduction in primary combustion zone peak temperature. Combustion completion with methanol must then extend into the secondary dilution air zone. Start-up at idle and even at low bleed air power levels, proved to be impossible on methanol. At these low power levels, engine flame-out was experienced during fuel change over from aviation kerosene to methanol.

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NOMENCLATURE

A	Combustor cross-sectional area
A, C	Experimental constants
a, b, c	Algebraic constants
f	Fraction of total air participating in combustion
L	Length of combustion zone
m, n	Reaction orders
$\dot{m}_A$	Air mass flow rate
P	Pressure
ΔP	Pressure drop
R	Gas constant
T	Ambient air temperature
T_3	Combustion chamber air inlet temperature
t	Time
V	Volume
V_c	Combustion-zone volume
x, y, z	Algebraic constants
η_c	Combustion efficiency
ρ	Density
ϕ	Equivalence ratio
ϕ_{pz}	Primary-zone equivalence ratio

I. INTRODUCTION

A. Methanol as an Alternative Fuel

During the past thirty years two major problems have developed, associated with our currently used fuels. These problems are increasing demand for a limited supply and its harmful emissions, which make air quality intolerable in heavy traffic areas. Political problems and instability in many oil producing regions of the world have made most fossil fuel supplies unreliable and expensive. Harmful emissions from fossil fueled cars, trucks, aircraft and power generation facilities have been shown to have profound effects on the environment we live in. As a result there is an increased need for cleaner burning alternate fuels such as alcohols and natural gas.

Methanol or Methyl Alcohol (CH_3OH) is a liquid petrochemical made from natural gas, wood or coal. Methanol is used to manufacture the gasoline additive methyl tertiary butyl ether (MTBE), acetic acid and many other chemicals. It can also be used as a low emissions alternative fuel. Fuel properties of methanol and aviation kerosene are shown in Table 1.1.

Until recently, the manufacturing capability to produce methanol has just kept up with the demand of the chemical industry and has been insufficient to supply methanol as an alternative fuel. Currently there are 18 methanol production plants in the United States with a total annual capacity of over 2.6 billion gallons per year. Worldwide, over 90 methanol plants have the capacity to produce over 11 billion gallons of methanol annually.

The U.S. Department of Energy (DOE), Air Products and Chemicals and Eastman Chemical Company have constructed a facility in Kingsport, Tennessee to take advantage of a new process (liquid phase methanol (LPMEOH™)) to produce methanol from coal-derived synthesis gas. LPMEOH™ was used exclusively throughout this research and will be referred to as methanol from this point forward. This demonstration project has shown that methanol could be produced at much higher volumes and at lower cost.

Table 1.1 – Fuel Properties

Property	Aviation Kerosene (Avtur)	Methanol
Chemical formula	$C_{12}H_{26}$	CH_3OH
Relative Density @ 15.5C	0.80	0.797
Lower Specific Energy MJ/kg	42.80	22.67
Molecular Mass	170.3	32.04
Boiling point K (F)	423-573(301-571)	338(148)
Stoichiometric Fuel /Air ratio	0.0676	0.155
Surface Tension N/m	0.02767	0.0226
Viscosity @ 293 K, m^2/s	$1.65x10^{-6}$	$0.75x10^{-6}$

B. Previous Work

Large fluctuations in conventional fuel costs make methanol and its reduced NO_x emissions, an attractive alternate fuel when its cost per unit energy becomes competitive. The U.S. Department of Energy and many state agencies have sponsored a number of methanol demonstration projects, which have included methanol-fueled automobiles, buses, trucks and gas turbines. International car and truck companies have also conducted demonstration projects using methanol.

1. United States Department of Energy and WVU Methanol Demonstration Project.

A number of alternate fuels DOE sponsored operational and emissions tests have been conducted at WVU in internal combustion piston engines for cars, buses and trucks. Test fuels included methanol, ethanol, and compressed natural gas. The program involved collecting operational and maintenance data from over 100 buses across the country. The WVU mobile emission lab and transportable dynamometer were used to perform power and emissions testing.

Corrosion and lubricity additives proved to be essential for reliable piston engine operation on alcohol fuels such as methanol and ethanol as the lubricating quality of these fuels is much lower than diesel fuel. The associated excessive wear of fuel injectors can be reduced when methanol is treated with a lubricity additive. Fuel filter fouling, associated with poor fuel quality at the test sites, was another problem that was easily remedied.

Emissions testing showed significant reductions in oxides of nitrogen (NO_x) emissions and in particulate matter (PM) when compared to diesel fueled trucks and buses, which were not equipped with particulate traps. NO_x concentrations ranged between 6 and 12 ppm when fueled with methanol and ranged from 25 to 27 ppm when using diesel fuel. PM concentrations ranged between 0.1 to .4

mg/m^3 with methanol and ranged between 0.72 and 2.6 when using diesel fuel without particulate traps. Total Hydrocarbon (HC) and Carbon Monoxide (CO) appeared to be higher in fleets operating on methanol. HC concentrations were between 2 and 38 ppm when fueled by methanol and between 2 and 4 ppm when fueled by diesel. CO concentrations were between 8 and 26 on methanol and between 6 and 16 when using diesel fuel. It was noted that the large diversity in data obtained on the alcohol fueled buses may be attributed to differences in engines and maintenance (Motta et al, 1996). See emissions data in Table 1.2.

Table 1.2. Alternate Fuel Transit Bus Emissions.

U.S. EPA Heavy-Duty Engine Emissions Certification Standards for Urban Transit Buses

	CO	**NO_x**	**HC**	**PM**
1991-92	15.5	5.0	1.3	0.25
1993	15.5	5.0	1.3	0.10
1994-95	15.5	5.0	1.3	0.07
1996-97	15.5	5.0	1.3	0.05
1998	15.5	4.0	1.3	0.05
	Units = g/bhp-hr			

2. Department of the Environment California Research

During 1980 and 1981, The Electric Power Research Institute and Southern California Edison Company (SCE) conducted a test to compare the operational and emissions characteristics of two 26 MW power generation gas turbines running on methanol and aviation kerosene fuel with and without water injection.

These tests were conducted at SCE's Ellwood Energy Support Facility in Goleta California.

The heating value of methanol is approximately half that of aviation kerosene turbine fuel. To maintain the same electrical output, the fuel flow rate had to be doubled. Approximately 30,000 gallons of methanol were burned daily.

To supply the large volumes of methanol needed and to deal with corrosion and lubricity issues the original fuel pumps were replaced with electric centrifugal pumps made of methanol compatible materials. The fuel nozzle orifices had to be enlarged, to accommodate the increased flow rate.

In the first tests a fuel heater and Mobilead F800 lubricant additive were used when operation on methanol. These precautionary steps were discontinued for most of the later tests without any problems. Examinations of the fuel pump showed minor wear on the pump shaft after testing.

On-line fuel change-overs were conducted, but not without some difficulties. The fuel system could not adjust fast enough for the higher fuel flow rate necessary when operating on methanol. This problem may have been avoided if a large volume fuel mixer loop had been placed inside the fuel line.

Emissions testing showed significant reductions in both oxides of nitrogen (NO_x) and particulate matter (PM). NO_x emissions were further reduced with the use of water injection. Hydrocarbons (HC) were slightly higher when running on methanol (Weir, et al, 1981). Emissions results are summarized in Table 1.3.

Table 1.3. Gas Turbine Emissions Results from Department of the Environment California Research.

Emission Species	**Aviation Kerosene (Avtur)**	**Methanol**
NOx, engine A, 6/27/79, 15MW Load (ppm)	90	19.1
CO2, engine A, 6/27/79, 15MW Load (%)	2.9	2.78
CO, engine A, 6/27/79, 15MW Load (ppm)	66	108
HC, engine A, Baseload, (ppm)	3	5
Solid Particulates EPA-5, lb/106 Btu	0.018	0.003
Total POM, 15MW Load, (μg/SCM)	7.98	3.44

3. Gas Turbine / Methanol Future

Volvo has introduced two new demonstration projects the Environmental Bus (ECB) and the Environmental Concept Truck (ECT). Both projects are alcohol fueled gas turbine electric hybrids. The ECT has shown over 90 percent reductions in NO_x (Borg, M., 1998).

General Motors has introduced its new gas turbine electric hybrid car. It is powered by a Williams micro auxiliary power unit (APU) gas turbine and GM's EV1 electric drive train. Gas turbine/electric school buses have been suggested as an offshoot of this technology to reduce pollution. The Southern Coalition for Advanced Transportation (SCAT), based in Atlanta reports that America's 425,000 school buses produce pollution equivalent to the emissions of 68 million cars. Incorporating General Motors (GM) series hybrid technology could pay off big environmental dividends just in this one transportation category alone, making the school buses of the future quiet, clean and efficient (EV World and Digital Revolution, 1998).

Ford introduced it's methanol Taurus FFVs in the 1980's which with the help of the State of California has done well. Today, in California over 14,000 methanol FFVs serve in federal, state and municipal governments fleets, corporate fleets, rental car fleets, and are driven by hundreds of individual consumers.

To serve these vehicles, an extensive network of 55 public methanol-refueling stations stretches from Los Angeles to Sacramento, including a station in Yosemite National Park. This methanol-fueling infrastructure was established by the California Energy Commission in cooperation with the State's major gasoline retailers. In addition, more than 50 private fueling stations are operated in California by individual fleet operators (Dolan, G. A., 1996).

Currently, the largest market for methanol in the U.S. is for the production of methyl tertiary butyl ether or (MTBE). Methanol production capacity is expanding (AMI, 1996). MTBE has recently been linked with large-scale ground water contamination and has been outlawed in some states. Therefore large quantities of methanol may become available for use as an alternate fuel, if current legislation continues.

II. WVU UNMODIFIED GAS TURBINE OPERATION ON METHANOL

A. Test Set-Up

The GTC-85-72 gas turbine, which is installed in the West Virginia University STOL research aircraft had to be modified for use in this research project. For safety reasons, a shield was installed on one side of the airplane to protect the operator in the event of a gas turbine failures. The operational controls and instruments for the turbine, including the starter switch, air bleed switch, tachometer and compressor pressure gage were relocated from the cockpit to the operator side of the airplane. In addition to these controls, engine performance measuring equipment had to be installed including K-type thermocouples to read the exhaust gas temperature, bleed air temperature and venturi inlet air temperature.

Additional hardware required for testing includes a motor-generator set gas turbine start cart used to supply the required 26 volts for start-up. To measure the total air mass flow into the turbine, a venturi with a 7-inch throat diameter was installed in-line with the turbine intake. The vacuum reading in the venturi throat was used in addition to the atmospheric pressure and temperature to calculate the engine airflow rate. Power loading was accomplished through the use of a bleed air manifold containing various numbers of choked flow metering nozzles. Bleed air power was calculated from the total nozzle area, bleed air pressure and temperature, which were read from a pressure gage and K-type thermocouple respectively. The test fuel was pumped directly from a 55-gallon drum to the fuel selector valve system, described in section B-6.

B. General Description of the Gas Turbine

The GTC-85-72 engine is a gas turbine auxiliary power unit (APU) that is primarily used to provide pneumatic jet engine start-up power at airports. This particular engine was manufactured by AiResearch/Garrett in the late 1960's. In 1972 the engine was installed in the West Virginia University STOL research aircraft, Figures 2.1 & 2.2. This aircraft is no longer airworthy and therefore grounded.

There are six basic engine assemblies, which include: the compressor section, the turbine section, the combustion chamber, the lubrication system, the electrical system and the fuel flow and RPM controller, Figure 2.3.

1. Compressor Section

The centrifugal compressor provides about 40 psig compressed air for the turbine and the bleed air for pneumatic power. The compressor is a two stage centrifugal type with a pressure ratio of 3.4: 1 and a total air mass flow of 5.5 lb./sec, at 40,800 rpm.

Figure 2.1 -Photographs of Instrumentation, Controls and Bleed Air Manifold Shown with Operator Protective Shield

Figure 2.2 - WVU STOL Research Aircraft Containing the GTC-85-72 Gas Turbine Engine and View Looking Down its Exhaust Stack

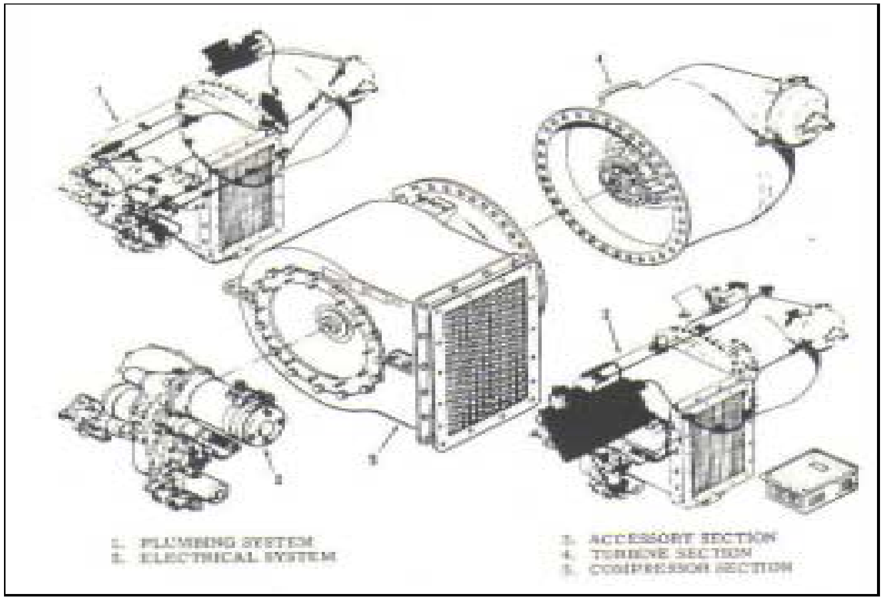


Figure 2.3 - Six Basic GTC-85-72 Gas Turbine Assemblies

2. Turbine Section

The turbine section provides power to the compressor and the accessories and is designed to operate at inlet temperatures up to 1200° F.

3. Combustion Chamber

The combustion chamber is a reverse flow can type, which is comprised of a cylindrical liner mounted concentrically inside a cylindrical casing. The chamber's key components include an air casing, diffuser, liner, fuel atomizer, glow plugs and spark igniter, Figure 2.4.

4. Lubrication System

The lubrication system is a self-contained positive pressure, dry sump type. This system provides pressurized splash lubrication to all gears, shafts and bearings.

5. Electrical System and Instrumentation

The electrical system requires approximately 26 volts DC to operate the starter, solenoid, instrumentation and the ignitions system. The ignition system is a high-energy step up transformer charging capacitors, which build up voltage across the igniter plug. In addition to the igniter, a pair of 8 amp glow plugs, Figure 2.5, and their voltage regulator from a PT-6 jet engine have been added to provide a higher energy ignition source. Power is

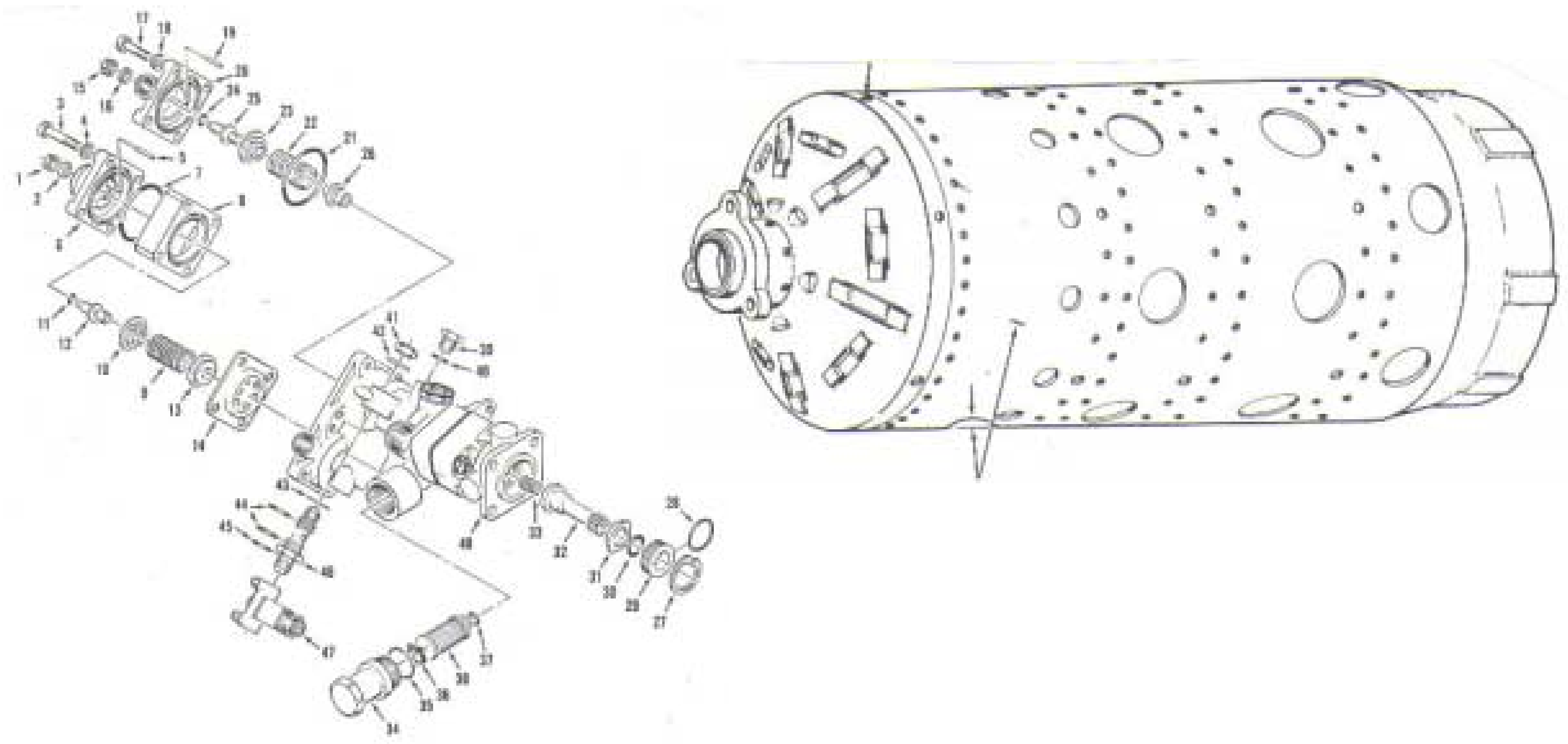


Figure 2.4 - Fuel Controller and Combustor Can of the GTC-85-72 Gas Turbine

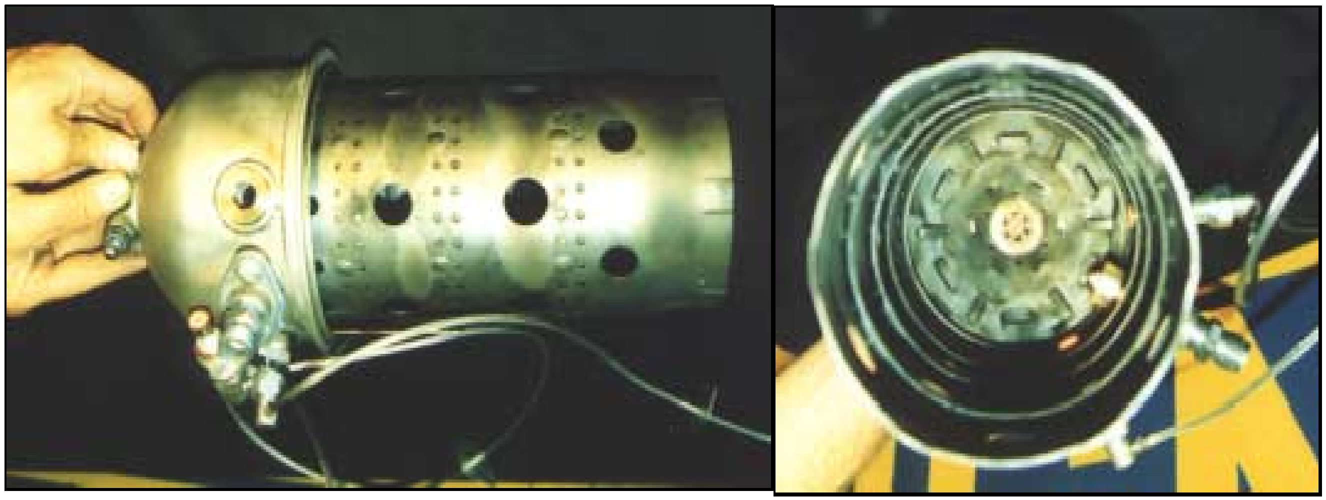


Figure 2.5 - Combustion Chamber View with Fuel Atomizer, Igniter, Glow Plugs and Holes for Secondary Cooling

supplied to this system by a 26 volt DC generator for the main engine circuits and a 24 volt battery for the glow plug voltage regulator.

Instrumentation for the engine's operation and for testing include three K-type thermocouples located to measure exhaust gas temperature, bleed air temperature and ambient air temperature, one gear driven tachometer, one compressor outlet pressure gauge, one bleed-air pressure gauge, one fuel pressure gauge, fuel flow meter, and one charging voltage gauge.

6. Fuel/RPM Controller and Bleed Air Valve

The fuel and bleed air control system automatically adjusts fuel flow to maintain a near constant turbine operating RPM under the varying load conditions, which depends on the amount of bleed-air extracted. A gear in the accessory section drives the fuel pump and control unit, Figure 2.6. This gear type fuel pump capable of 230 psi incorporates a fuel filter, acceleration limiting valve, fuel pressure relief valve, fuel solenoid, and connections for the pneumatic control, and electric control. A constant operating speed is achieved through a combination of an acceleration limiting flyweight-type governor bypass fuel dump valve and a diaphragm bypass valve activated by the bleed air pressure. Fuel is transferred under pressure to the fuel atomizer located in the end of the combustor cap. The fuel atomizer consists of a screen, a flow divider valve, distributor head and housing. The distributor head divides the fuel passageway within the core.

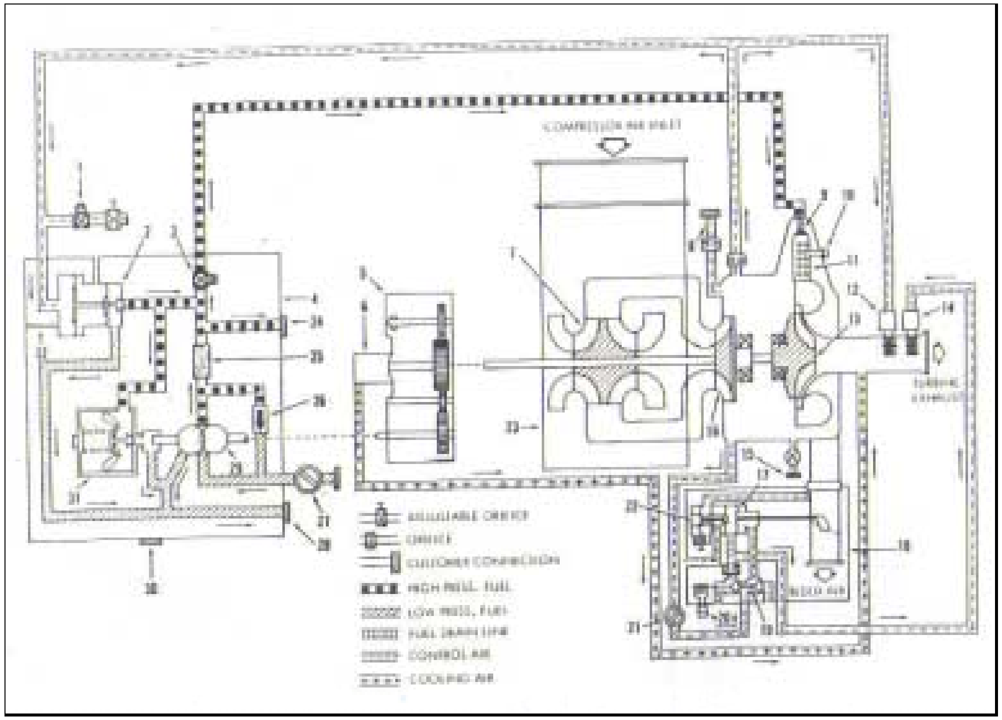


Figure 2.6 - Schematic of Fuel/RPM Control System

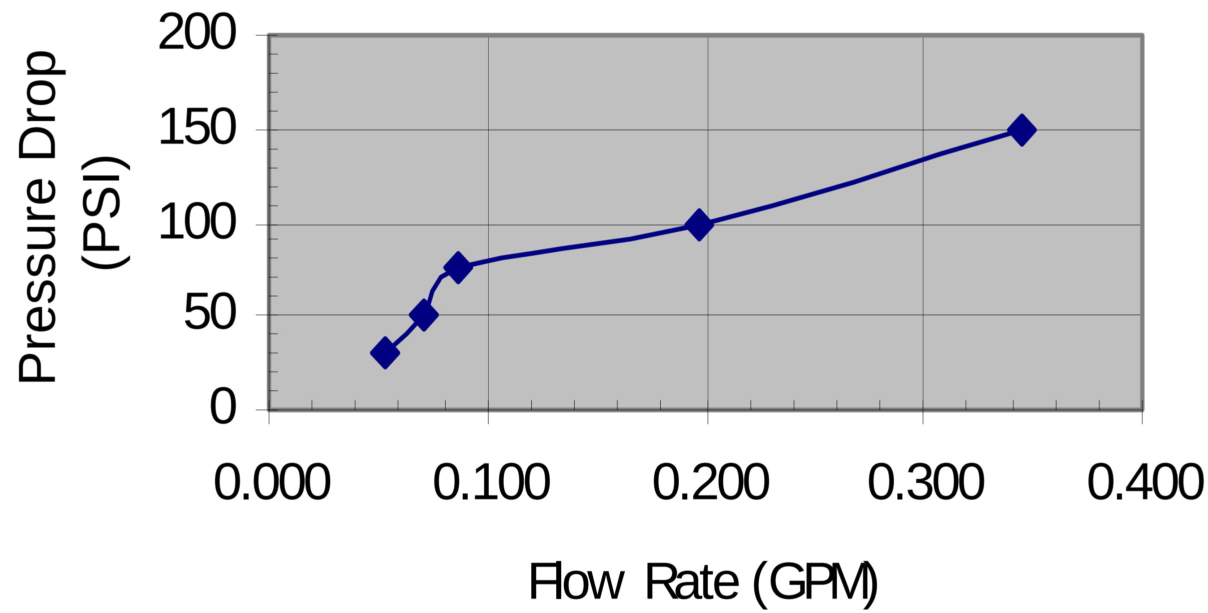


Figure 2.7 - GTC-85-72 Gas Turbine Spray Nozzle Calibration

The center passage leading to a small orifice plate and an annulus leading to a large orifice. The flow divider valve directs fuel at low pressure through the small center orifice and at high pressure to both the small and large orifice.

During May 1998 the fuel atomizer was calibrated in a spray booth at Pratt and Whitney Engine Services in Bridgeport, West Virginia. This calibration was necessary to ensure that there would be adequate atomization and correct spray cone geometry under all the operating pressures expected during operation of the engine with aviation kerosene and methanol, Figure 2.7.

C. Fuel System Design

For safety reasons a separate fuel system was designed so that it could be disconnected at the end of each test and stored in an approved storage facility. Because of the corrosive nature of methanol, and to eliminate cold starting problems, it was necessary to perform engine start-up and shutdown using conventional aviation kerosene (Jet A). The gas turbine is started on aviation kerosene, operated under load to bring the combustor up to operating temperature before gradually changing over to methanol. After the tests are completed, the fuel type was changed back to aviation kerosene prior engine shutdown.

To accomplish the desired fuel change-over procedure, a special fuel supply system was developed. It consists of two 55 gallon DOT #17 fuel drums one containing methanol and the other containing aviation kerosene, Figure 2.8. Each of these drums was equipped with a separate pneumatic powered fuel pump, capable of 4.6 gpm, which discharges to the fuel type selector valve, Figure 2.9. The selected fuel then traveled to the fuel emulsifier. This allows a gradual change in mixture concentration during fuel type change-over. The components of this emulsifier are shown in Figure 2.9 they consist of a small orifice, a clear sight glass and a recirculating pneumatic fuel pump. During fuel change-over, this sight glass becomes cloudy with the emulsified aviation kerosene/methanol mixture. Downstream of the fuel emulsifier, a fuel pressure spike damper was installed, Figure 2.9. This damper consists of a volume of captured air in a clear sight glass to compensate for the pulsating nature of the pneumatic fuel supply pumps. Following the pressure spike damper, the fuel was routed to a volumetric flow meter and on to the gas turbine fuel controller.

Figure 2.8 - Fuel Supply System

Figure 2.9 - Fuel Selector and Emulsifier

D. Problems Encountered During Turbine Operation

During the course of this project, various unforeseen problems were encountered. The first of which was engine flameout due to the too sudden fuel type change over. This problem was solved by the addition of the fuel emulsifier recirculating pump described in section C.

With the modified fuel supply system, another problem surfaced, in that the gas turbine would not operate at idle or even at very low power settings on methanol. This is believed to be due to the nearly 5 time greater heat of vaporization of methanol when compared to aviation kerosene. Because of this, methanol requires more ignition energy upstream of the point where the dilution air enters the burner.

A second, and predictable, operation limitation was uncovered whereby the gas turbine could not be operated on methanol at high power levels. This is due to the inability of the fuel system to double the volumetric fuel rate flow for the same combustion temperature when operating on methanol. If fuel type change-over from aviation kerosene to methanol was attempted at a high power setting, then the turbine experiences a gradual loss in RPM, which terminates in combustor flame-out. Operation of this turbine on methanol at these elevated power settings, requires a new fuel controller system capable of higher flow rates.

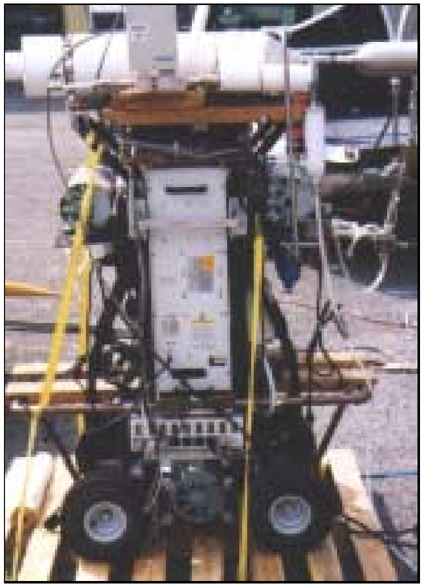
In addition to the power operation limitations found when operating on methanol, additional durability issues were encountered. The first of these was the quick destruction of the aged rubber diaphragms in the gas turbine fuel controller. These diaphragms failed after only a short exposure to the methanol fuel. As a result, this fuel controller was rebuilt using all new diaphragms and seals. After overhaul this seals performed flawlessly throughout the remainder of the tests. However, one additional problem was experienced. This was the destruction of the brass gear pump housing and the fuel controller RPM governor both caused by the poor lubricating properties of methanol.

E. Emissions Testing Equipment

WVU Mechanical and Aerospace Engineering (MAE) has designed and built two mobile emissions testing labs that are capable of testing vehicles up to 30,000 kg. (66,000 lbs.) in the field. WVU has tested over 700 buses and trucks from more than thirty-five locations throughout the United States. Much of the data collected from the buses and trucks are available in database from WVU.

The mobile emission lab is comprised of two tractors, an emission measuring instrument trailer and a flat-bed chassis dynamometer with the rollers, flywheels and power absorbers, Figure 2.10. Inside the instrument trailer there is an environmental chamber for preparation of the particulate filters and a microgram scale for measuring them, there are also precision gases for calibrating the analyzers, racks of data acquisition and dynamometer control equipment, emissions analyzers etc. The trailer also has a blower and the power supply for the sonic flow venturi constant volume sampling (CVS) system and the stainless steel dilution tunnel on top of the instrument trailer.

Figure 2.10 - WVU Mobile Emissions Lab and Mobile Testing Equipment

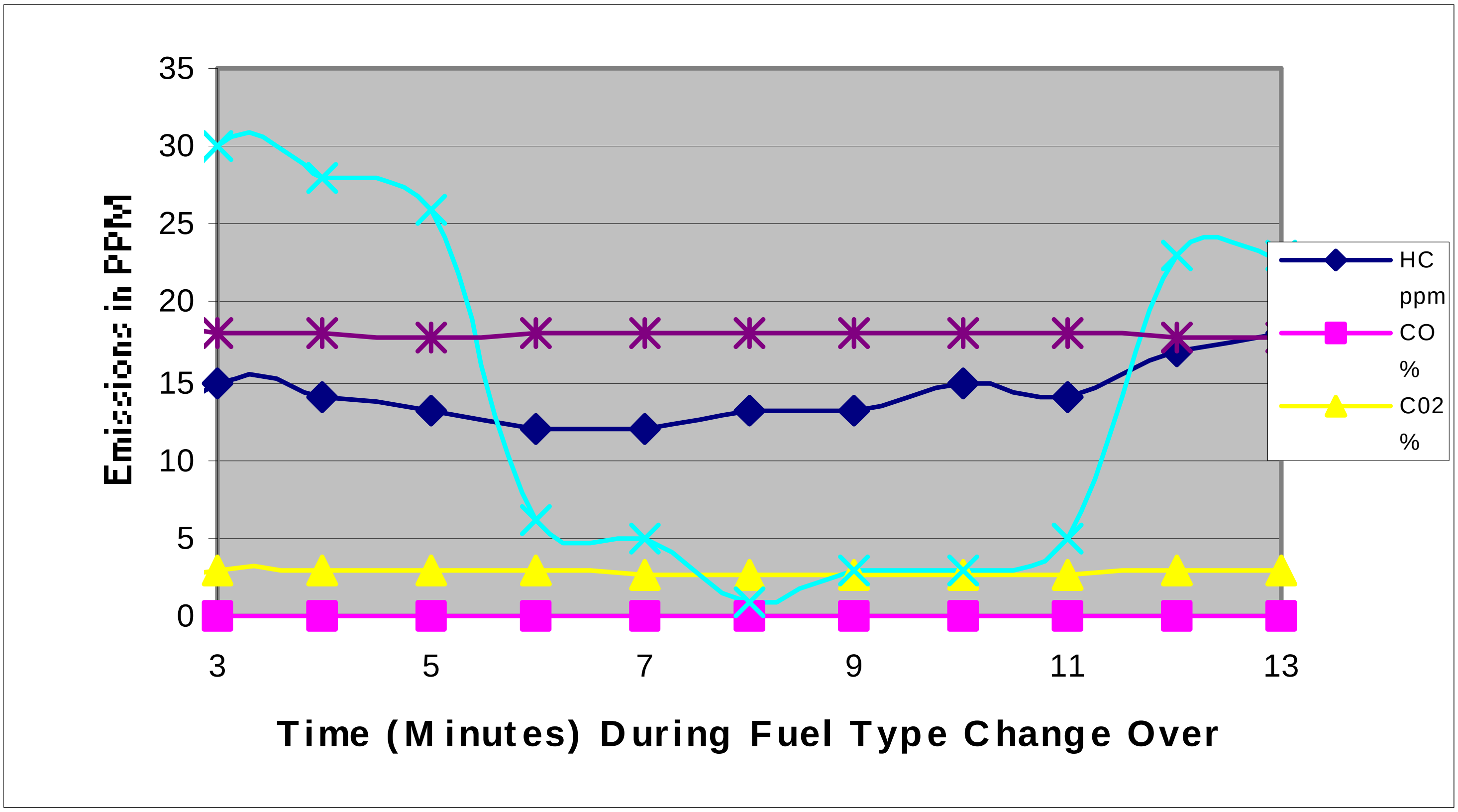


The emissions lab can measure and characterize emissions from a wide range of vehicles that use various types of fuels. However, most of the vehicles tested use alternative fuels. The exhaust emissions from vehicles are measured using a dilution tunnel and full exhaust gas emissions measurement instrumentation. Each test is run three times to ensure repeatability and data quality. The laboratories measure carbon monoxide, carbon dioxide, oxides of nitrogen or NO_x, methane, total hydrocarbons, aldehydes and particulate as per USEPA standards. Figure 2.11 shows the emissions lab set up for testing of the GTC-85-72 gas turbine installed in the WVU STOL airplane. Because of the exhaust gas flow rates and dilution ratios, the dilution tunnel was removed in favor of a slip stream sampling probe.

F. Data Collection and Reduction

The self regulating gas turbine operates in a near steady state flow rate condition with the exception of the fuel flow rate, which varied depending on output power level and varied slowly during fuel type change over. All turbine operating parameters measured, varied slowly enough, that the data could be collected manually by reading gages, see Figure 2.12. The transportable laboratory comes equipped with a standard 18 inch diameter dilution tunnel. It has choked flow metering nozzles, which are sized for various flow rates up to 3000 CFM. As its flow should be diluted to below 290°F, about two-thirds of the dilution tunnel flow

Figure 2.11- Emissions Testing Setup



Reduction in Emission When Switching from Kerosene to Methanol

Power KW M100-2CB/ KW Jet	Jet Nox g/s	M100 Nox g/s	Reduction %	Jet CO2 g/s	M100 Co2 g/s	Reduction %	Jet CO g/s	M100 CO g/s	Reduction %	Jet HC g/s
59/60	0.04	0.01	72.50	58.20	56.09	3.62	0.45	0.49	-8.89	0.29
59/61	0.05	0.01	73.33	59.87	51.01	14.79	0.91	0.47	48.68	1.67

Figure 2.12 - Emissions during Fuel Type Change Over from Kerose
to Methanol 8/18/98 and Percent Reduction of Emissions from 9/15/

must come from outside air. The GTC-85-72 gas turbine exhaust flow rate is about 3.5 pounds/second = 2700 SCFM at more than 700°F. Therefore the standard 18 inch diameter dilution tunnel cannot process this much exhaust flow. Instead a 3/8th cooled copper tube slip stream probe was inserted in the exhaust stack.

A sampling pump draws a metered steady flow through the analysis equipment inside the transportable emission laboratory.

Carbon monoxide is measured by infrared absorption, nitrogen oxides are measured by chemical luminescence and unburned hydrocarbons are measured by flame ionization detection. From these the fuel/air ratio could be calculated. However in the gas turbine tests this is not necessary. From the measured turbine air inflow rate and compressor bleed air flow rate together with fuel flow rate, this ratio is determined. This is done in a simple computer program, for example see test #2, shown in Table 2.1., and other test data as shown in Appendix 7. Program formulas are also listed in this Appendix. For example in test 2J on aviation kerosene the stoichiometric air/fuel ratio by mass is 14.7. The burner air flow rate is 3.48 lbm/s and the burner fuel flow rate is 0.0456 lbm/s this results in an actual air/fuel ratio 3.48/0.0456=76.31 or equivalence ratio Φ =14.7/76.31=0.19. From an emission point of view this very lean equivalence ratio is meaningless as the combustion takes place near stoichiometric at the burner inlet.

Table 2.1- Sample Computer Program for Power and Emissions

Data Reduction from GTC85-72 Gas Turbine Test #2J on Jet-A, September 15, 1998				
Bleed air load setting: manifold equiped with bleed air nozzles, Dia.Dn inch=		0.625		
Number of bleed air nozzle installed in the bleed air manifold is given by Nn=		**3.5**		
Engine inlet air flow is metered with a venturi with throat diameter Dv inch =		7		
Recorded test data running on jet-A fuel is indicated by (A pi=		3.141593		
1A) Outside air temperature OAT is measured in degrees F, called OATF=		**80**		
local sealevel barometer reading reported by airport tower in " mercury Hg=		**30.13**		
Vacuum measured in throat of intake air flow metering venturi in " water H2(		**11.8**		
Bleed air nozzle total temperature measured in the manifold Tt3 in degree F		**400**		
Bleed air nozzle total pressure measured in the manifold Pn in psi =		**36.75**		
Turbine RPM during test under load with flow control bleed air valve on =		**42700**		
Compressor outlet pressure in psi gage		**38**		
2A) Turbine engine air inlet flow calculation in units of pound mass per second				
Test altitude is 1250 ft ASL or local barometer is 1.33" below sealevel and =		28.8		
Ambient air absolute pressure in units of PSFA is from local barometer		2036.791		
Ambient air absolute temperature in degrees Rankine =		**540**		
Ambient air density = venturi throat density is calculated rho (slug/ft^3)=		0.002198		
Venturi throat area calculated in square feet Av=		0.267254		
Venturi throat velocity calculated from measured vacuum Vv in ft/s =		236.287		
Venturi mass flow rate as measured by the intake air venturi in lbm/s=		4.469457		
3A) Turbine bleed air output power load calculation				
Calculate combined bleed air nozzle throat in square feet defined as An=		0.007457		
Calculate nozzle flow absolute total pressure inside manifold PSFAn=		7328.791		
Calculate nozzle flow absolute total temperature inside manifold TRankine=		860		
Calculate total bleed air nozzle flow rate in lbm/s		0.990099		
Bleed air compressor power required in BTU/s = flow rate*temp. rise*Cp=		76.03957		
Bleed air compressor horse power input (1 HP=0.707 BTU/s) or HPbleed=		107.5524		
4A) Turbine inlet temperature calculation from exhaust gas temperature EGT				
Turbine type fuel flow meter reading in gallon per minute		**0.41**		
Fuel flow rate in lbm/s from flow meter reading, at spec grav. =	0.8	0.04563		
Turbine combustion chamber pressure Pt3, not measured assume= bleed air Pn				
Total Compressor power input = HP(bleed air)*(lbm/s venturi)/(lbm/s bleed)=		485.5082		
Assume Turbine shaft HP power output =1.1*compressor power input=		534.059		
Turbine shaft power output in BTU/s =		377.5797		
Turbine exhaust gas temperature Tt9 in degrees F is called EGT and =		**909**		
Turbine inlet temperature= EGT+(BTU/s output power)/(lbm/s*Cp) degree F=		1305.723		
5A) Turbine inlet temp. calculation from fuel/air ratio and fuel heating value				
Stoichiometric reaction equation, assuming complete combustion				
Aviation kerosine has formula CH1.93 and LHV=18400BTU/lbm=10222cal/g		42800		
1 mole fuel gives:1CH1.93+1.4825(O2+3.76N2) produces 1CO2+0.965H2O-		13.93	=kg fuel per mole	
Then moles product to moles of stoichiometric air ratio is		1.068346	= (1+0.965+5.574)/(1.4825*(1+3.76))	
Product molecular weight Mprod is (1*44+0.965*18+5.57*28)/(1*0.965+5.57		28.842		
Stoichiometric air/fuel ratio in moles= A/F by volume = 1.4825(1+3.76)/1=		7.057		
Stoichiometric A/F ratio by weight (using Mair=28.97) =7.057*28.97/13.93=		14.677		
Burner air flow rate in lbm/s= venturi air flow rate - bleed air flow rate		3.479359		
Burner fuel flow rate in lbm/s, assume entering at reference temp.=25 C		0.04563		
Excess air (ex)=ratio of: (burner air flow-stoichiometric air)/stoichiometric ai		4.195347	this is both mass and volume or mol	
Exhaust gas molecular weight Mex=(air(ex)*28.97+prod*28.84)/(air (ex)+prc		28.94361		
Actual fuel/air ratio F/A by weight=burner fuel flow/burner air flow		0.013114		
Equivalence ratio at turbine inlet =(Stoichiometric A/F)*(actual F/A)		0.19248		
Total exhaust gas mass flow rate is air flow + fuel flow, in units of lbm/s		3.524988	in kg/s =	1.59893
Jet A heating value LHV in BTU/lbm (at comb. efficiency eta=1.0) =		18400	air in specific heat Cpc=0.24 BTU/lb	
Calc. turb. inlet temp.Tt4(in F)=((F/A)*LHV*eta+Cpc*Tt3)/(Cpt*(1+F/A))=		**1280.533**	air out specific heat Cpt=0.27 BTU/	
6A) Alternate calculation of turbine inlet Tt4 using average Cp in KJ/KmoleK				
Cold dual molecule gas has specific heat =3.5R which on mole basis is same for O2 and N2 with Cpc=3.5*8.314=29				
Using burner average specific heat for O2 with Cpt=34.0, for N2 with Cpt=31.6				
Using burner average specific heat for CO2 with Cpt=51.9 and for H2O with Cpt=40.6				
Product (nP) composition: 1CO2+0.956H2O+ex*1.4825O2+(1+ex)*5.574N2				
Reactant (nR) composition: 1CH1.93+(1+ex)*1.4825O2+(1+ex)*5.574N2			**Emissions data conversion factors**	
Sum of product: moles*specific heat = sum(nP*Cpt)=		1216.623		
Sum of reactant: moles*spec. heat* temperature rise=sum(nR*Cpc*(Tin-298		191139.4	from ppm to gm/1000 kg use conve	
Heat of formation Hfo for JP-8=jet A in KJ/KmoleK=		-30681	ppm*(M/Mex). To get gm/s multiply	
Heat of reaction for Jet A in KJ/KmoleK=-393522+0.965*(-241827)-(-30681)=		-596204	(kg/s(exhaust)/1000)	
(Tt4-298) in K=(sum of reactants-heat of reaction)/sum of products		647.1545	To get std. cc/s multiply by (22400/M	
Turb. inlet temp Tt4 (in F)=((Tt4-298)(in K)+298)*9/5-460=		**1241.278**	To get gm/Joule fuel at 42800kJ/kg, r	
			gm/s/(lbm/s fuel*0.4536*42800000 J/	

Therefore, NO_x and unburned hydrocarbons HC are formed as a function of an unknown equivalence ratio during combustion. After reaching peak flame temperature, the combustion products are diluted with secondary air to the allowable turbine inlet temperature. Only by measuring or modeling the temperature profile along the length of the combustor can one analyze the effect of dilution air on the NO_x and HC concentrations in the exhaust. Chemical kinetics show that the concentration of NO_x increases rapidly with flame temperature, and is greater than predicted by equilibrium thermodynamics. The rate of forward reaction is different from the backward reaction, and there is insufficient time for equilibrium to be reached.

In Table 2.1, the turbine inlet temperature has been calculated three ways. First the compressor bleed air power is calculated from the temperature rise and flow rate, this is 102 HP in test #2. From the measured bleed air and total inlet airflow the compressor power is calculated to be 485 HP. Equating this to the turbine power, allows one to calculate the turbine temperature drop. This added to EGT of 824°F in test #2J results in a turbine inlet temperature 1221°F. The second and third methods are based on assuming 100% adiabatic combustion and neglecting emissions other than CO_2, H_2O, O_2 and N_2. The expected results will be slightly higher. They are 1280°F using a mean specific heat and 1241°F using individual specie specific heats.

Measurements were recorded on a concentration basis. The emissions data were recorded by computer at 1 second intervals during 10 minute periods for single-fuel steady state operation. If these tests were conducted on an engine for a car, then the emissions would be reported in grams per mile. For a stationary engine it would be reported in grams per HP. As this turbine does not provide shaft HP but only compressed air, it is more appropriate to report emissions in units of standard cc per second. First reduce the turbine exhaust gas flow rate to a room temperature volume flow rate, using density 0.0765 FT^3/lbm. For test #2J the exhaust gas flow rate is 3.48+0.0456=3.5256 lbm/s= 46 STD FT^3/s = 46*28317 cc/s.=1.3*10^6 cc/s. Thus, in test 2J if the ppm values are multiplied by 1.3 then one gets the emissions in cc/s.

During the transient fuel type change-over maneuver, another automotive type emission test apparatus was employed. This one was capable of printing data in five seconds intervals. Such high speed data acquisition was essential, as the fuel-change-over lasts less than 0.5 minutes, depending on the power setting. In that time the fuel concentration ratio changes gradually from 0% to 100%. Data were collected continuously and printed out in 5 seconds intervals. Because the equipment used for this test was designed for simple automotive testing, the data presented here should only be used for relative comparisons. These data are plotted as a function of time in Figure 2.12.

For this test, the turbine was operated on aviation kerosene until steady state was reached at which time the data acquisition was initiated at t = 0. Because of the steady nature of the data on aviation kerosene, data were only plotted starting at t = 4 minutes. At t = 5 minutes, fuel type change-over was initiated from aviation kerosene to Air Products methanol. Immediately following this change-over, Figure 2.12 shows a dramatic decrease in NO_x production as methanol replaces aviation kerosene. At approximately t = 6 minutes, one minute after the initiation of the fuel change over, the NO_x data approach the pure methanol equilibrium value. At t = 11 minutes the reverse fuel type change-over, from methanol to aviation kerosene, is initiated. Following this procedure, the NO_x production rapidly approaches the aviation kerosene steady state value as represented by value at t = 4 minutes. It can be seen from Figure 2.12. that while fuel type has a strong effect on the NO_x production, it has little effect on the other species sampled.

G. Conclusions and Recommendations About Emissions When Operating on Methanol.

The GTC-85-72 gas turbine was successfully operated on both aviation kerosene fuel and on fuel grade methanol produced by Air Products and Chemicals Inc. Emission data were collected on each fuel during steady state (defined as unchanged during at least 6 minutes). In addition emission data were collected during the transient fuel-change over procedure which lasted about less than 0.5 minutes. Some alcohols like ethanol are entirely miscible with jet fuel, but methanol is only partially miscible. The miscibility reduces with the presence of water and at lower temperatures. To prevent separation, chemicals such as benzene and acetone can be added. Engine starting proved to be only possible on aviation kerosene, due to the low volatility of methanol and the high heat of vaporization. To minimize corrosion and diaphragm deterioration during storage, and permit starting, it was decided to change over to methanol only after the engine was warmed up and return to aviation kerosene prior to engine shut-down. A sight-glass in the fuel supply manifold clearly demonstrated the poor miscibility between aviation kerosene and methanol. They can only be forcibly mixed, just like oil and vinegar. After a fuel emulsifier pump was installed, the transition from one type of fuel to the other becomes visible like a milky cloud, which only clears up after change-over is completed. To achieve fuel change-over without engine flame-out, it proved to be essential to raise the EGT to more than 750°F, which is done by applying at least 25% bleed air load. In an attempt to improve this low power flame-out problem, two PT-6 engine glow plugs were added to the existing spark plug. The continued inability to operate on methanol at idle and below 25% bleed air load, is most likely due to the cooling effect from the high heat of vaporization. This delays ignition and moves the flame front to further downstream in the burner. Because the mixture is diluted by secondary air and becomes too lean to ignite. Giving more separation between the primary and secondary air supply zones might solve this problem.

Unfortunately the fuel controller was unable to supply enough methanol to permit operation at more than 50% bleed air. This problem was later solved by installing a fuel controller and burner nozzle of a larger model.

The lack of methanol lubricating properties destroyed the bearings and the cylindrical RPM control fuel valve inside the fuel controller. It is imperative that all future turbine tests on methanol must incorporate a suitable lubricant additive.

The significant change in NO_x level from about 25 ppm on aviation kerosene down to about 5 ppm on methanol, is most likely caused by the before mentioned burning of the methanol spray at a location further downstream, where the mixture gets cooled by secondary air flow, thereby lowereing the peak flame temperature and thus reducing the production of thermal NO_x.

III. METHANOL LUBRICITY PROBLEMS AND SOLUTIONS

A. Introduction to Lubricity

During the 1998 initial gas turbine tests at WVU on fuel grade methanol, without additives, the poor lubricating properties of methanol caused repeated mechanical failures of the gas turbine's RPM governor and fuel pump. It became obvious that a suitable methanol additive must be used to improve its lubricating properties to equal or better than that of aviation kerosene fuel. Such an additive is also needed for corrosion inhibition, and must be readily miscible with methanol and be able to remain in solution during storage inside fuel barrels. Further it had to be readily available and be economical. Measuring the lubricity of methanol as a function of percent of additive, turned out to be the most challenging portion of this research project.

The wear of lubricated bearing surfaces depends not only on the lubricant, but also on the materials used, the bearing load, and surface finish. Lack of sufficient lubricating properties results in wear, which alters the surface finish and produces loss of material from the surface. One can experience four types of wear: corrosion, adhesive wear, abrasive wear and surface fatigue. Wear can be reduced by the presence of lubricants and corrosion inhibitors at the point of contact of the wear bodies.

One distinguishes two types of fluid lubrication "Boundary Lubrication" and "Hydrodynamic Lubrication". Boundary Lubrication occurs when the lubricant surface tension maintains a boundary between the solid surfaces, thereby reducing the frictional forces between them. Hydrodynamic lubrication is when a lubricant is forced or pumped in between the two surfaces, to limit their interaction. Many tests have been developed to characterize lubricating fluids. The three most common test methods are: BOCLE (Ball-on-Cylinder Lubricity Evaluator), the HFRR (High Frequency Reciprocating Rig), and field-testing. Each of these tests uses test specific criteria, as measures of lubricity, to compare different fluids.

1. **Ball on Cylinder Lubricity Evaluator (BOCLE) (ASTM 5001)**

The BOCLE (American Society for Testing and Materials, 1999) test was designed for testing the lubricity of diesel and jet fuel. The test consists of placing a ½” diameter ball on cylinder rotating at 244 RPM, submerged in the test fluid at 25°C. Each test starts with a new ball loaded with a 9.81 Newton force and lasts 30 minutes. Upon completion of the test, the scar on the ball is measured to the nearest 0.01 mm

A variation of this test is called the Lubrizol Scuffing BOCLE[2]. This test is similar to the before mentioned test but applies a steady load provided by a 7 kilogram mass. The test is run on the cylinder for 2 minutes. The average scar diameter is then measured and used to compare lubricating qualities.

2. High Frequency Reciprocating Rig (HFRR)

The HFRR[3] test uses a ½" ball, which is rapidly vibrated back and forth over a flat surface. A load of 200 grams is placed on the ball and moved back and forth with a 1-mm stroke. The time necessary to wear a scar into the ball is measured; the size of the scar gives the lubrication qualities of the fuel being tested.

3. Field Testing

Field-tests are the most reliable tests, because all of the operating conditions are duplicated exactly. However, this type of testing is usually very expensive and can be impractical. The WVU methanol fueled GTC-85-72 gas turbine, experienced two fuel controller/gear-pump failures, which proved to be very expensive to repair. This emphasizes the importance of fuel additives to provide the required lubricity.

B. WVU Lubricity Tests

1. Ball on Flat Disc (Type 1)

One lubricity test apparatus was available at WVU. It was a variation of the Lubrizol Scuffing BOCLE method. Here the cup, containing the sample material is filled with the test fluid and rotated. A stationary ½" steel ball is lowered onto the sample at a distance from the center of rotation. This test is designed to quantify fluid lubricity by measuring changes in wear rate, either from mass loss or from scarring.

When used with methanol, it was found that once wear had begun, the data collected over different time intervals, keeps on changing, rendering it difficult or impossible to produce repeatable data. This erratic performance was due to a changing wear pattern.

2. Cylinder on Washer (Type 2)

To get repeatable data, a new fluid lubricity test machine was developed at WVU. It is like a thrust bearing, submerged in the fluid to be tested. It measures torque due to friction at the points of contact, instead of measuring wear related to mass loss. This test was developed to measure the friction coefficient at a specific bearing load, the justification being that friction is ultimately responsible for wear. The new apparatus was designed for operation in a vertical milling machine with a digital position readout. This assured a vibration free drive system with accurate and steady RPM control. The first design was based on a rotating steel cylinder on a stationary washer made of brass. Force was applied to the cylinder by a free-floating 5 kg. mass. The region of contact between the two surfaces was submerged in the fluid mixture to be tested. RPM of the disc, normal load, and the torque imparted to the stationary disc were all measured. Using the load, RPM and torque data a coefficient of friction for the apparatus and the specific fuel mixture being test was calculated.

The contact surface area between the discs was approximately 0.002 m^2, which is relatively large when compared to other test methods. The 0.002 m^2 area disc is shown in Figure 3.1. Any irregularities in the steel cylinder, the brass wear washer or any particles from wear created unreliable torque. A .001 m^2 wear disc was constructed to remedy this, but demonstrated the same inherent problems. The high noise to signal ratio can be seen in the typical raw torque data in Figure 3.2.

3. Armature with One Ball on a Stationary Washer (Type 3)

This Type 3 configuration combines features of the Type 1 and Type 2. This Armature with One Ball on a Stationary Washer configuration used the Type 1-½" steel ball rotating in an armature on a stationary brass washer to measure torque. A force was initially applied to the ball with a spring, but this was changed to a brass dead weight to avoid changes in force during rotation.

This configuration was an improvement over the first two, but the repeatability of data was poor. When the force, applied to the ball was low, between 5 and 10 N it was difficult to distinguish between two different lubricating fluids. When the force was increased above 10 N wear began to occur between the ball and the brass washer. The contact surface area needed to be increased to prevent wear without the problems associated with the previous method. These problems were eliminated in the final (Type 4) configuration of the WVU lubrication evaluator. Tests results from this research can by found in Appendix 4.

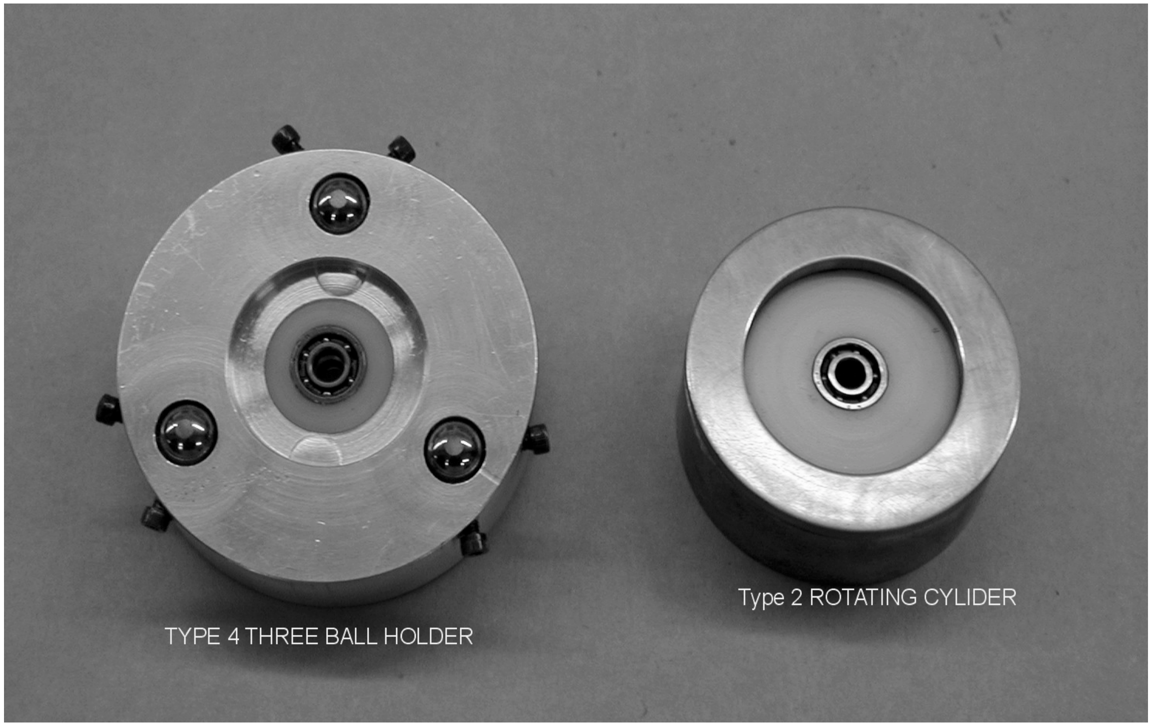


Figure 3.1 - Photograph of Type 4 - 3 Ball Holder and Steel Type 2 Cylinder

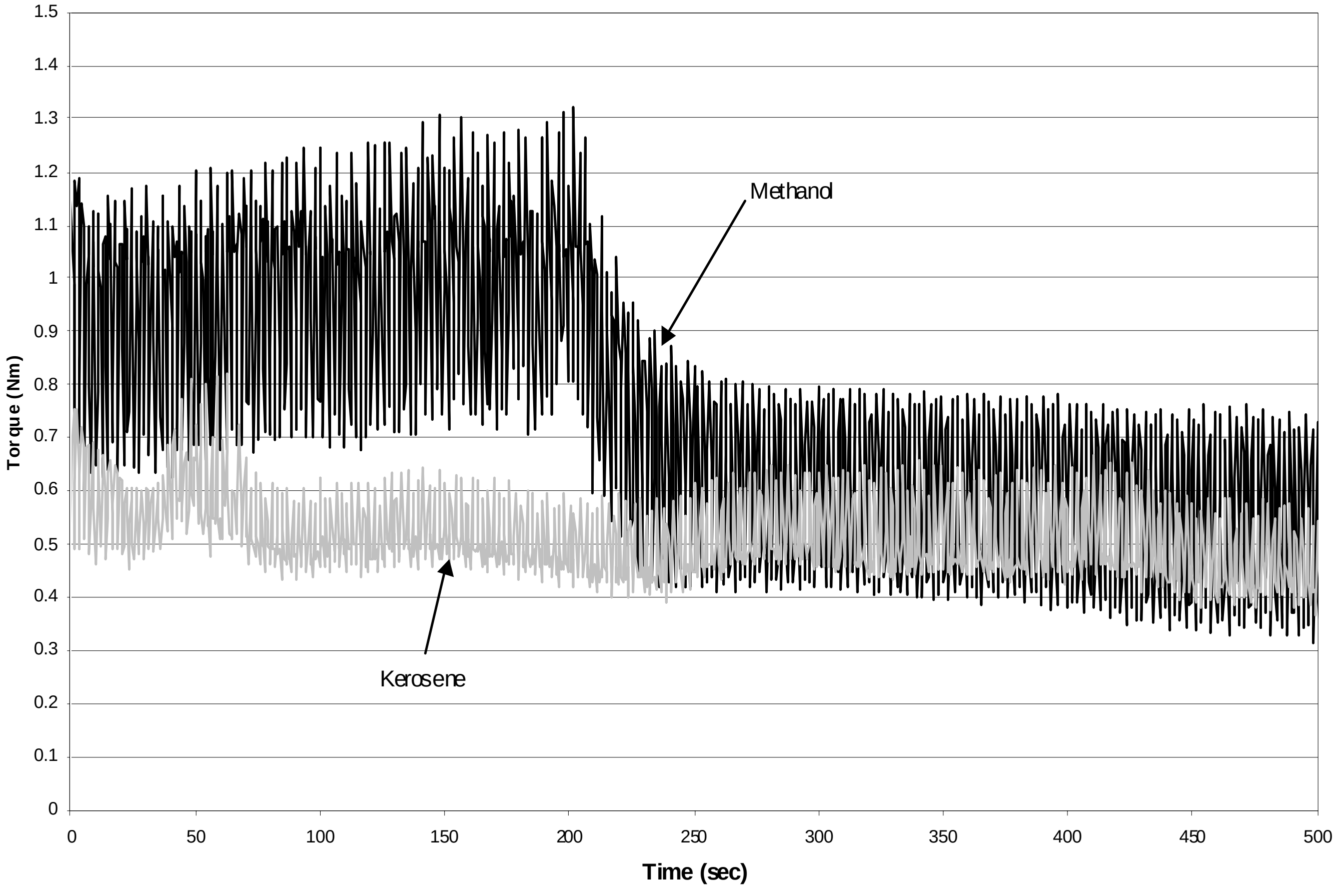


Figure 3.2 - High Signal to Noise Ratio in Raw Torque Data for Methanol and Aviation Kerosene Using the Type 2 Lubricity Evaluator

4. Armature with Three Flattened Balls on a Washer (Type 4)

The final WVU lubricity apparatus was designed to operate at normally used bearing pressures on a rotating disc containing three balls (Figure 3.1). The three balls transferred the load onto a fixed brass washer and were mounted at a distance of 31mm from the centerline of the disc holder. The three balls were ground to form flats of 3.81-mm diameter. With the 56.501 N dead weight load in use these flats reduced the lubricated contact pressure to 1.65 M Pa, which is 3.5% of the maximum design load limit for a well-lubricated lead-bronze bearing. This contact pressure reduction proved to be necessary to prevent marring the surface when operating on methanol. To guarantee that the disc rotates smoothly about its axis, it was guided by a ball bearing installed on a centering pin in the middle of the fixed washer. The wear disc and the bearing holder were mounted inside an aluminum cup, which was 100 mm in diameter and 50 mm in depth. This cup was filled with the test fluid so that the contact surface between the load balls and brass disc was fully submerged. The cup was mounted on a 76mm ball bearing, which allows it to rotate freely. Torque measurements were taken with strain gauges, mounted on a 197 mm aluminum arm, which extends from the cup. Using the contact area, the load, and the measured torque, coefficients of friction were calculated for each fuel/lubricant mixture. The data were very stable when the load ball holder is rotated at 200 RPM. An exploded view of the complete testing apparatus is shown in Figure 3.3. Shown here is the disc three-ball drive head, to be installed on a vertical mill. A disc drive shaft extends from the end of the mill attachment, passes through the dead weight, and is connected

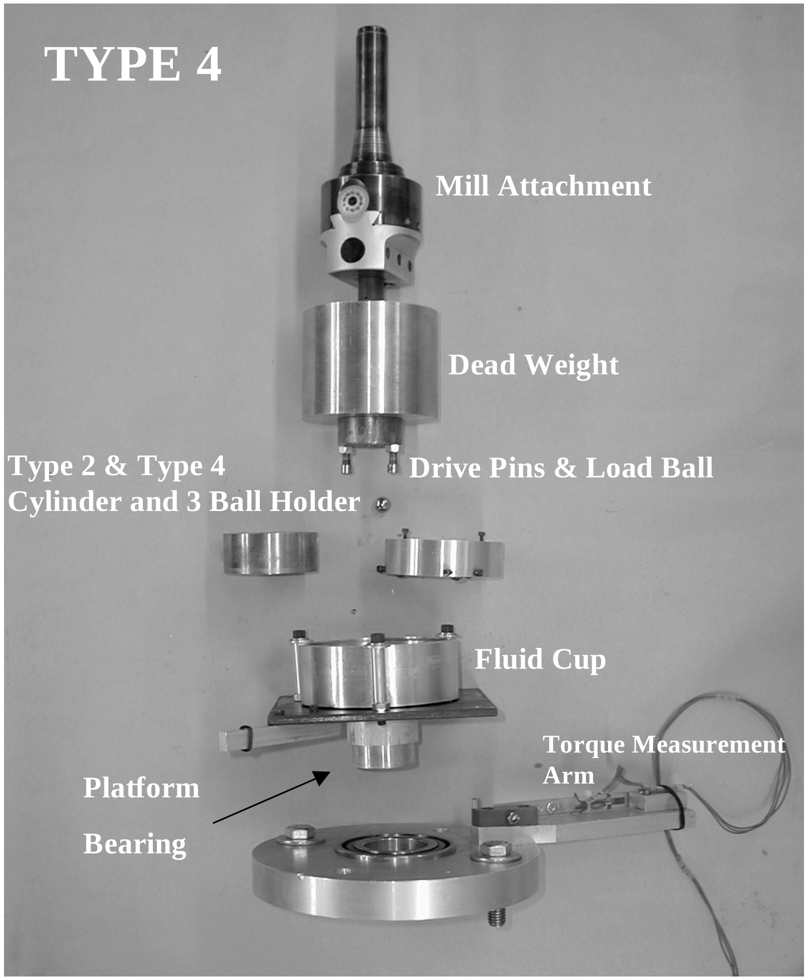


Figure 3.3 - Exploded View of the Type 4 Lubricity Evaluator

to the disc in a manner that allows only rotational forces to be transferred from the mill. The dead weight slides on the shaft, so that its weight is entirely supported by the balls in the driven disc. Torque is transferred from the drive shaft to the dead weight by a pin and from there to the driven disc by two pins, which protrude from the bottom of the weight. The dead weight normal force is transferred to the driven disc through a ½ inch steel ball on the system centerline. This configuration insured that the driven disc was loaded at the center, so that all three flattened balls transfer the same normal force.

a) Test Procedure

Prior to testing, great care was taken to prepare the contact surfaces for testing. The washer was machined to insure that its surface was perfectly flat and both contact surfaces, balls and washer, were hand finished by wet sanding using 1500 grit abrasive paper on a flat steel surface. No matter how fine both of these surfaces were ground, the system required additional rotational polishing before the surface finish was good enough to provide steady and repeatable friction coefficient data. This was accomplished by running the system at 200 rpm using aviation kerosene fuel as a lubricant. During this procedure, the friction coefficient data were monitored until a steady-state value was reached usually requiring 45 minutes of run time. A data set obtained during the first 30 minutes of the 45-minute "break-in" period can be seen in Figure 3.4.

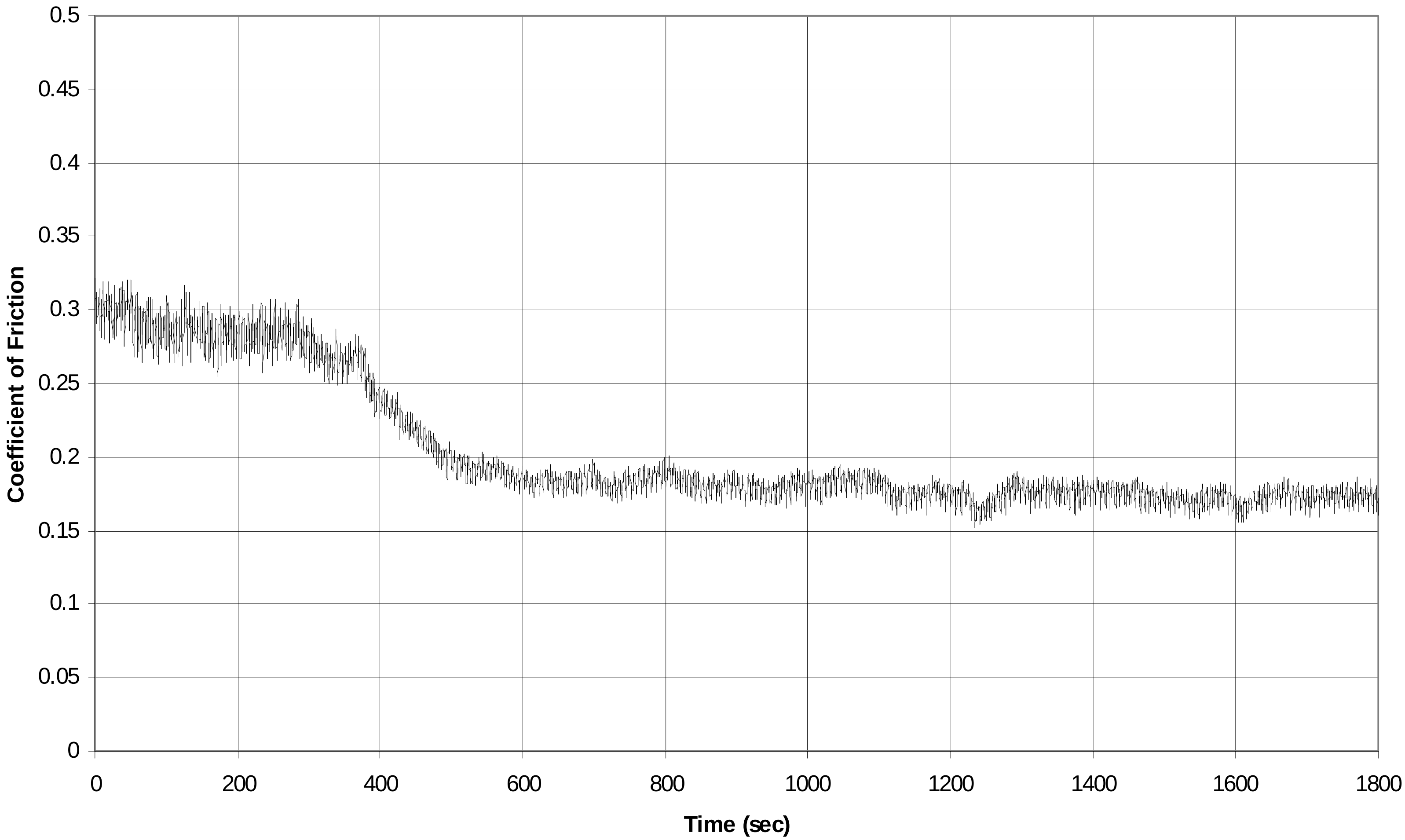


Figure 3.4 - Type 4 Lubricity Evaluator,
Aviation Kerosene Break-in Period, Test 123

Following the break-in procedure, testing was accomplished by filling the test cup with fluid to be tested, such that the contact surfaces between the load balls and brass disc were fully submerged. The system was operated at 200 rpm and friction torque data were collected at approximately 2 Hz for a period of 10 minutes. When a lubricant, such as castor oil, was tested at various concentrations, tests were run starting with pure methanol followed by ever increasing oil concentrations. This prevented the possibility of oil deposits from higher oil/methanol concentrations, to introduce errors at the lower concentrations.

Following the 30 minute “break-in” period, time dependent data acquired during six of the aviation kerosene and M100 tests are shown in Figures 3.5 and 3.6. Because of the starting transients experienced during many of these tests, the first two minutes of data were discarded prior to data averaging. A Quick Basic computer program was written to process the raw data. This program is included in Appendix 5.

b) Test Procedure

Measurement of the lubricating qualities of both aviation kerosene and methanol were necessary prior to evaluating the performance of the different methanol-lubricant solutions. Of the 74 tests conducted with the Type 4 evaluator, 22 of them were with either aviation kerosene and methanol.

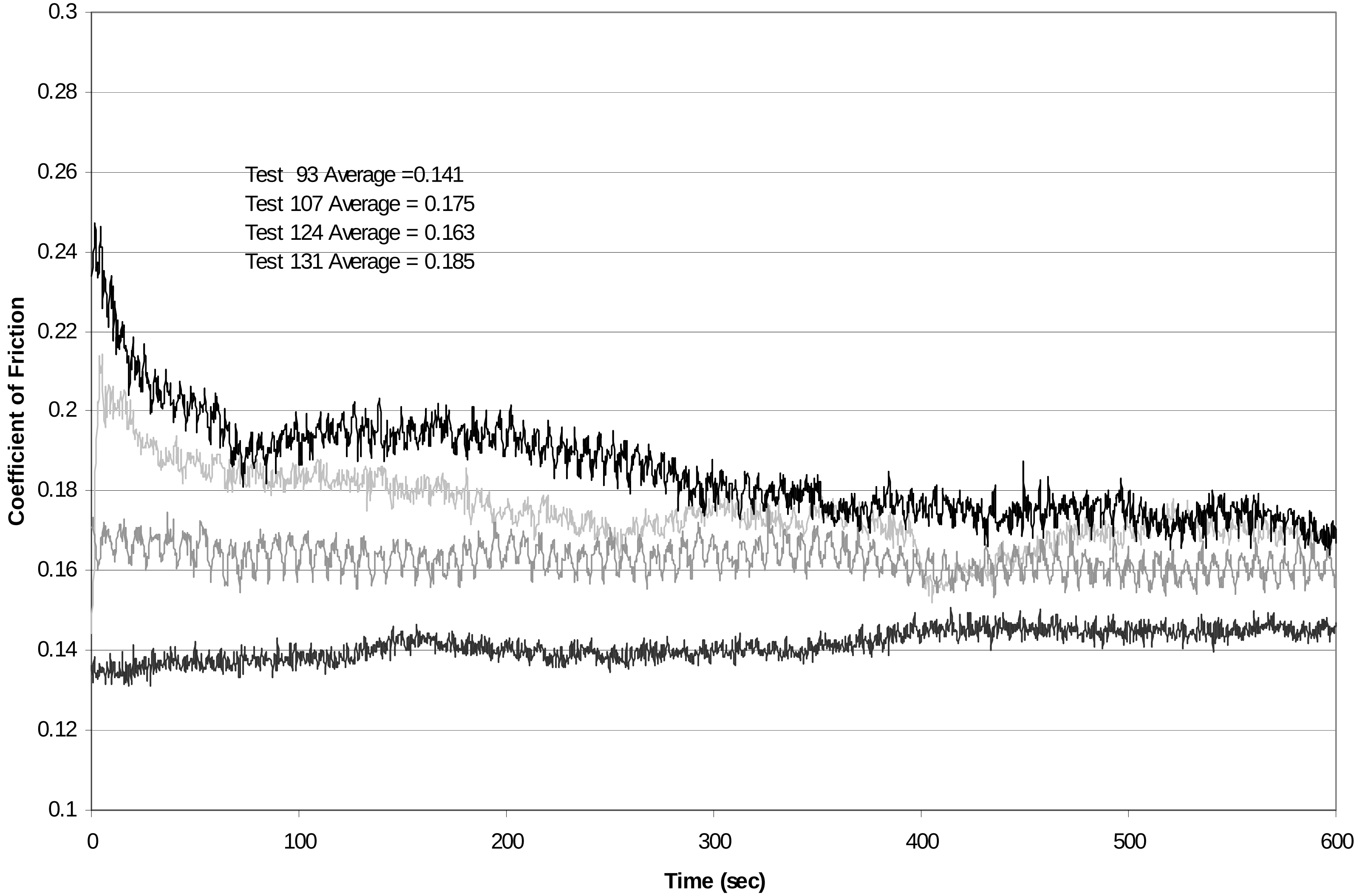


Figure 3.5 - Type 4 Lubricity Evaluator (Aviation Kerosene Tests), Raw Coefficient of Friction Data

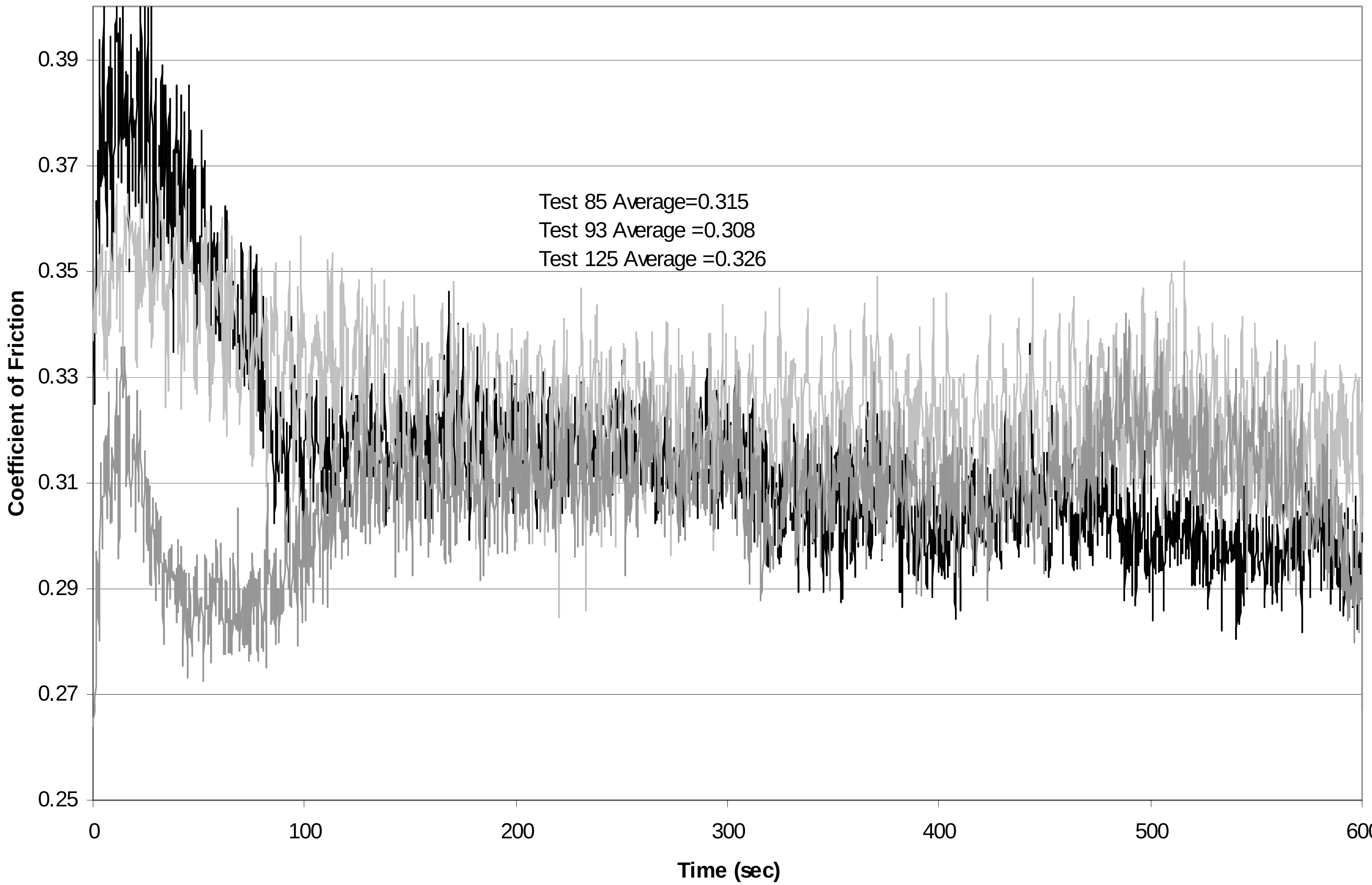


Figure 3.6 - Type 4 Lubricity Evaluator (Methanol Tests), Raw Coefficient of Friction Data

Six methanol and aviation kerosene tests, which had the lowest standard deviation, were chosen to calculate the statistically averaged coefficients of friction for each. The coefficient of friction for aviation kerosene was found to be 0.167 and 0.309 for methanol. Table 3.1 contains the experimental friction coefficients obtained experimentally for both methanol and aviation kerosene as compared to various handbook data. The six statistically averaged measurements for the coefficients of friction for aviation kerosene and methanol are shown in Figure 3.7.

Table 3.1. Friction Coefficient Data from Engineering Handbooks and WVU Data.

System	**Friction Coefficient**
Metal on Metal, Dry*	0.15 – 0.20
Metal on Metal, Wet*	0.3
Occasionally Greased*	0.07 – 0.08
Continuously Greased*	0.05
Mild Steel on Brass**	0.44
Methanol (WVU)	0.309
Aviation Kerosene (WVU)	0.167

* - Oberg et al. (1962)
** - Avallone and Baumeister III (1987)

Experimentation indicated that the coefficient of friction depended on the velocity or the revolutions per minute of the test apparatus. In general, as the velocity increased the coefficient of friction would decrease. Tests at various RPM, between 75 and 250, were conducted with aviation kerosene as shown in Figure 3.8.

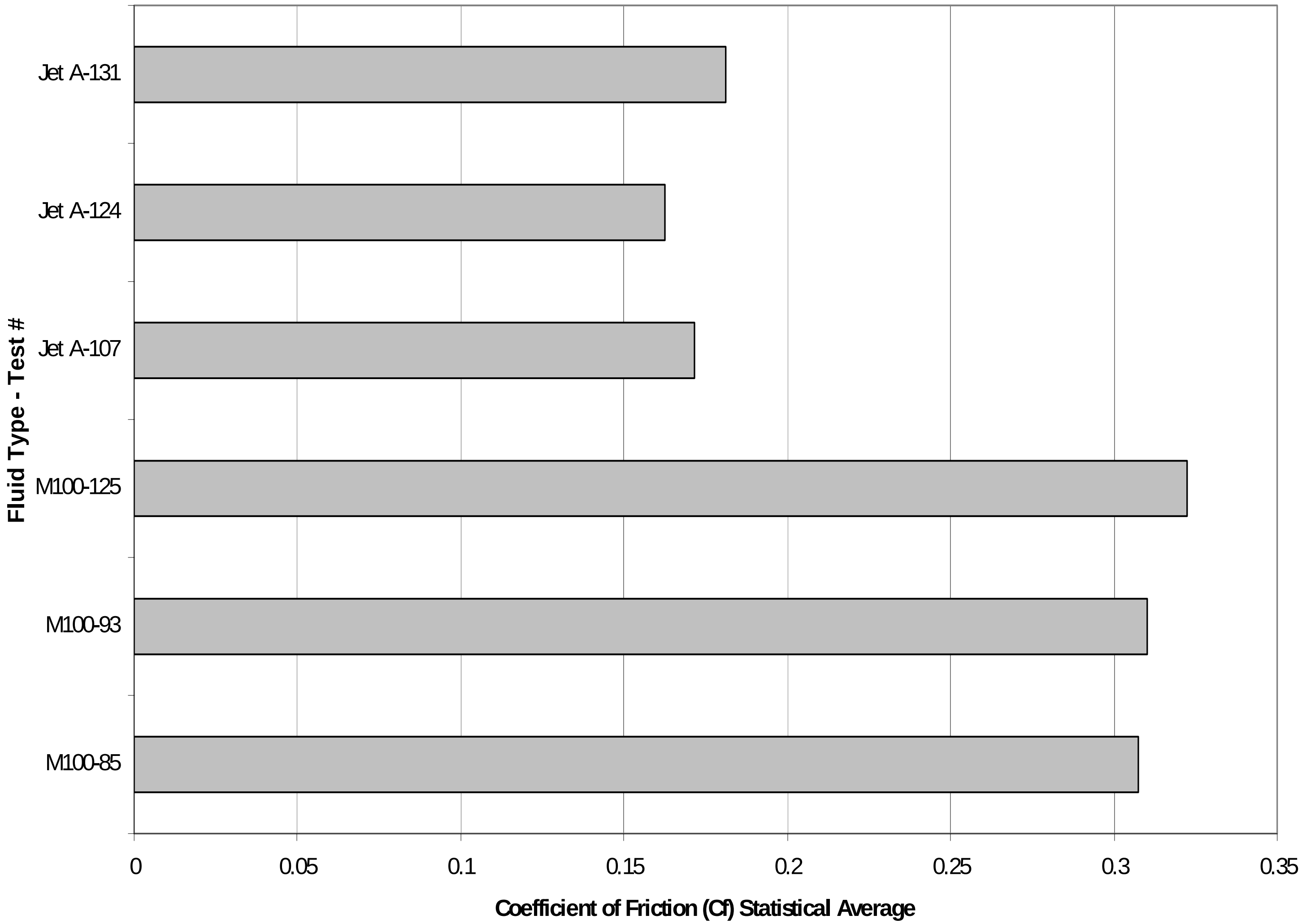


Figure 3.7 - Statistically Averaged Coefficients of Friction for Aviation Kerosene (Jet A) and M100 from Torques measured by Type 4 Lubricity Evaluator

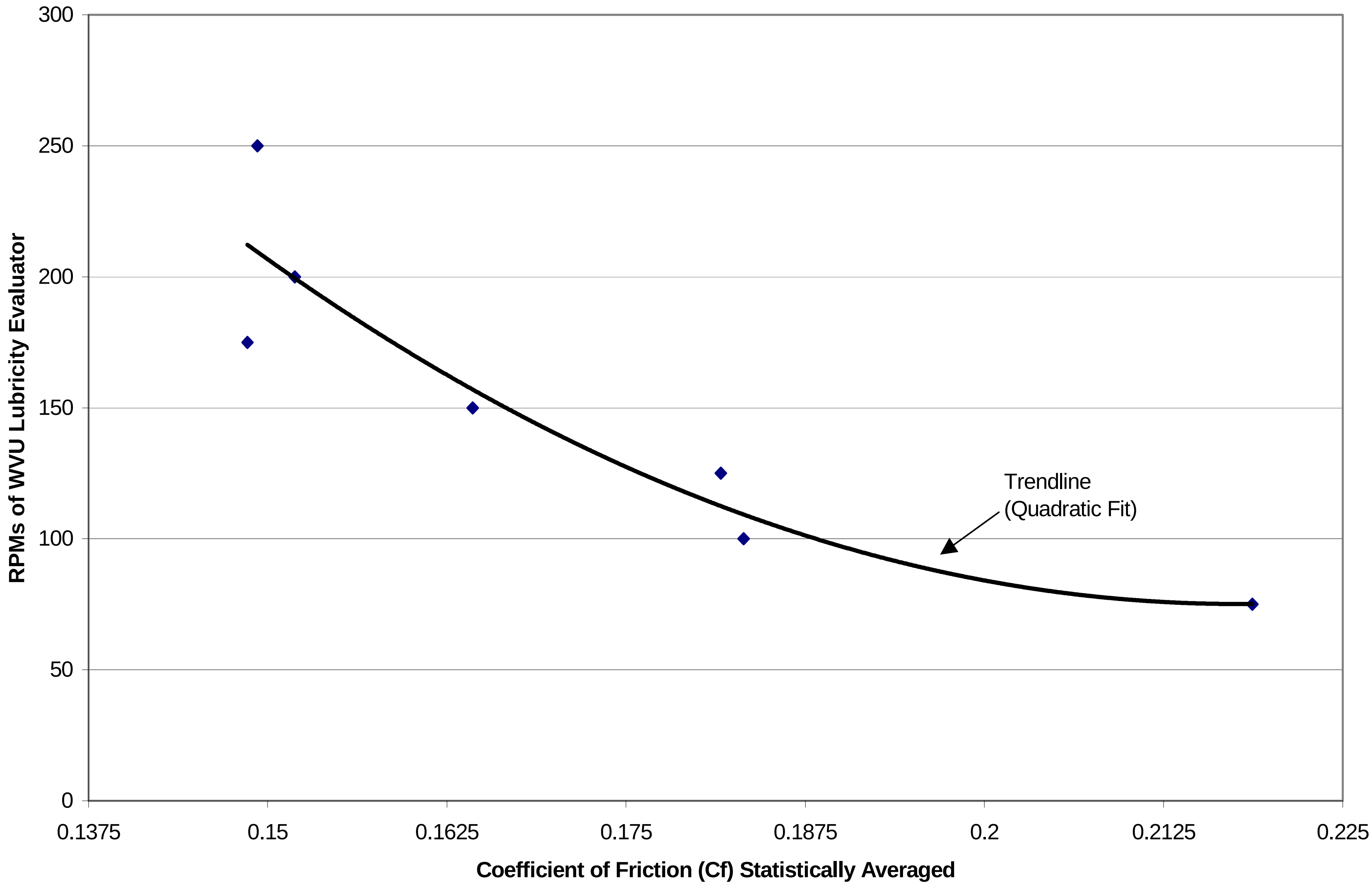


Figure 3.8 - Coefficient of Friction versus RPMs

Few lubricity additives proved to be both: effective in reducing friction and readily soluble in methanol. Only three of all the additives tested had the required properties and produced lubricity in excess of that of aviation kerosene fuel. They were readily soluble with methanol in quantities far in excess of that needed and remained in uniform suspension during storage. One satisfactory additive was pure castor oil and the other two were Morgan Fuels *Two Cycle Blue* and Manhattan Oil Company's *Power Plus Cherry Bomb* racing fuel additives. Both of these are primarily synthetic commercial methanol fuel additives for use in racing applications.

Friction coefficient data obtained for methanol containing varying concentrations of castor oil can be seen in Figure 3.9. At low concentrations, the addition of an additive has a large effect on friction coefficient. However, once a level of approximately 5% has been reached, there is little gained by increasing the castor oil concentration. Also shown in figure 3.9 are two horizontal lines indicating the friction coefficients of both pure methanol and aviation kerosene. Using the aviation kerosene line, it can be seen that a castor oil/methanol concentration of approximately 2.5% is required to achieve the same friction coefficient as aviation kerosene.

Using the same method two commercial products were evaluated. The manufacturer recommended ratio for the *Two Cycle Blue* additive is

0.04% in racing applications. However, to achieve the same friction factor as aviation kerosene, a 1% concentration was required. Manhattan Oil Company's *Power Plus Cherry Bomb* additive required approximately 1.6%. Coefficients of friction versus additive concentrations are shown in Figure 3.8 and are included with data statistics in Table 3.2.

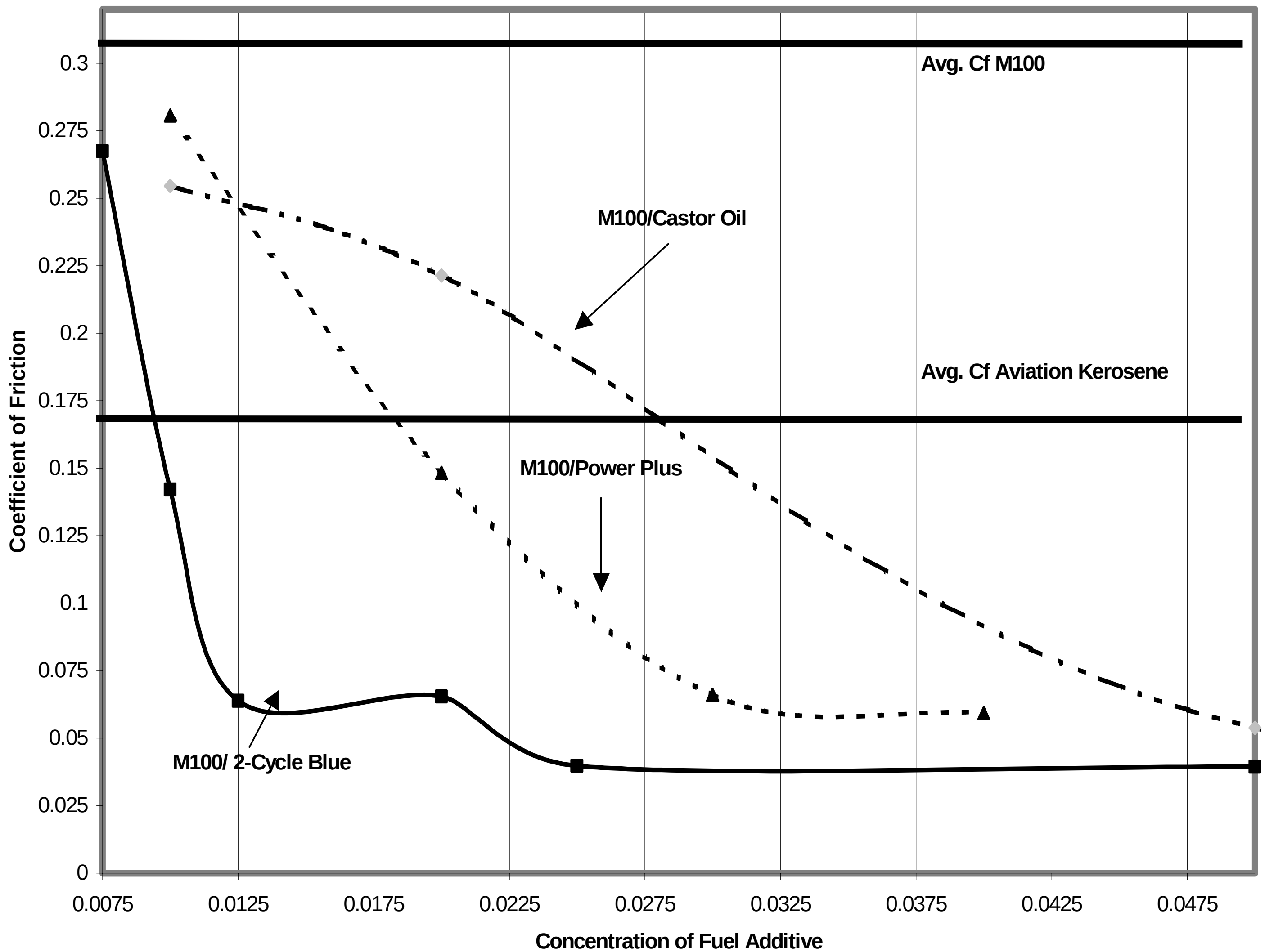


Figure 3.9 - Coefficient of Friction versus Fraction of Fuel Additive

Table 3.2

Statistical Data and Coefficients of Friction for Significant WVU Lubricity Tests

(not available electronically)

IV. WVU MODIFICATIONS TO OPERATE GAS TURBINE ON METHANOL PLUS ADDITIVES

A. Fuel System Modifications

During the unmodified gas turbine operation on pure methanol, flame out occurred when power dropped below 25% of rated level, and again when power demand exceeded the 50% of rated level. The upper limit was due to insufficient fuel flow rate. Its original gear type fuel pump was only capable of supplying fuel at a maximum of 230 psi and 300 lbs of fuel per hour. This fuel system is adequate for all power levels possible with aviation kerosene. Approximately twice the volume of methanol is required for the same energy flow rate as with aviation kerosene. Therefore, the original fuel control system and atomizer had to be replaced. The objective is to minimize modifications to this engine but some were essential to be able to prove that using methanol in existing stationary gas turbines, is practical and significantly reduces harmful emissions. A similar APU gas turbine, the GTPC-180L, has twice the power of the GTC-85-72 and is also manufactured by Allied Signal. The fuel controller and atomizer of the GTCP-180L are capable of supplying fuel at 600 psi and 750 lbs per hour. This engine's fuel controller tolerates a similar fuel bleed air control mechanism and therefore, requires very little modification to be installed on the GTC - 85 - 72. During July 1999, Piedmont Aviation of Melfa Virginia modified a GTCP - 180L fuel controller and installed it on one of their test GTC-85s to prove to us that it would be compatible with our engine. After some minor adjustments the new fuel controller functioned flawlessly on aviation kerosene over the entire power range and was still capable of supplying over twice the fuel flow of the original controller. WVU purchased the modified fuel controller and atomizer and installed them on the WVU gas turbine. The GTCP – 180L fuel controller and atomizer both function the same as described in chapter 2 and as shown in Figure 2.8.

When running this turbine on methanol the fuel supply pressure exceeded the design pressure of the original fuel lines. Therefore, it was necessary to upgrade them to ¼” MSHA 84/19 1500 psi fuel lines. The higher fuel pressures also exceeded the original 300 psi gauges in the system, so they were replaced with 600 psi gauges.

B. Ignition System and Instrumentation Modifications

In the lower power range, below 25% of rated power, flameout occurred when using methanol. In an attempt to solve that problem, and additional ignition source was added in the form of two glow plugs of eight Amps. each at 28 V. These, with their voltage supply, were taken from a PT-6 turboprop engine, as shown in Figure 2.5. Adding these glow plugs to the existing spark plug reduced the flameout power limit slightly, from 70 KW to 48 KW compressor bleed air power.

All engine controls have been mounted below a ¼” steel protective plate to allow safe monitoring during emissions testing. All instrumentation including a 10 channel thermocouple reader, engine RPM indicator, a compressor pressure gauge, bleed air pressure gauge, fuel pressure gauge, electrical power indicator and an intake venturi pressure gauge were installed together for safe monitoring.

C. Combustor Can Modifications

Along with the installation of PT-6 glow plugs described above a second combustor can was modified by Pratt Whitney Engines Services Division in Bridgeport, West Virginia. This consisted of the installation of four thermocouples (K type – 2000°F), which can be rotated by the operator, while the engine is running. The rotational arc is more than 90 degrees in the plane perpendicular to the flow. When not in use, they are rotated into the lower temperature region near the walls of the combustor can. This modification can be seen in Figure 4.1.

Figure 4.1 - Photographs of Thermocouple Equipped Combustor Can as Modified by WVU and Pratt and Whitney Canada

V. GAS TURBINE OPERATION WITH METHANOL AND 2% OF TWO CYCLE BLUE ADDED

A. Modified Gas Turbine Operation with Methanol Plus Additives

To provide an adequate factor of safety, all the second phase emissions testing on the WVU gas turbine were conducted using a 2% methanol-*Two Cycle Blue* solution. The gas turbine was operated for an extended period using this mixture, without problems.

Modifications to expand the operating range of the GTC-85 on methanol were completed. These modifications included the installation of a fuel controller and atomizer of the GTCP-180L gas turbine. These new modifications increased the operating range of the gas turbine when fueled by methanol from between 48 and 58 KW compressor bleed air power to between 48 and 103 KW compressor bleed air power.

B. Emissions Test Set-Up

When the weather improved sufficiently the outdoor emission testing of the WVU gas turbine, which is installed inside the STOL research airplane, was resumed. During March 2000 the WVU gas turbine and the mobile emissions lab were ready for a series of emissions tests for different power and fuel combinations. The mobile emissions lab was positioned approximately 15 feet away from the WVU STOL aircraft and the GTC-85-72 gas turbine. Figure 2.11 shows the emissions lab set up for testing. Two 3/8-inch stainless steel tubes were placed in the center of the gas turbine's exhaust stream as slipstream sampling probes and were run 15 feet into the mobile lab. One tube was for gaseous emissions and the other was for solid particulates.

Table 5.1 - WVU Mobile Emissions Analyzers

Test	Analyzer	Type of Analysis
Total Hydrocarbons	Rosemont Analytical Model 402 High Temperature	Flame Ionization Detector
Carbon Monoxide	Rosemont Industrial Models 880A and 868	Non-dispersive Infrared Detector
Carbon Dioxide	Rosemont Industrial Models 880A	Non-dispersive Infrared Detector
Oxides of Nitrogen	Rosemont Analytical Model 955 NO/Nox	Chemical Luminescent Detector
Particulate Matter	TEOM Series 1105 Diesel Particulate Mass Monitor	TEOM Filter and Microbalance

C. Analytical Tests

1. Gaseous Emissions

The gas analysis equipment detects the concentration of each gas in ppm and relays a signal to the computer at a 10 Hz frequency. Carbon monoxide (CO) and carbon dioxide (CO_2) concentrations are measured using non-dispersive infrared absorption. Oxides of nitrogen (NO_x) concentrations are measured using chemical luminescence and total hydrocarbons (HC) concentrations are measured using a Flame Ionization Detector (FID). See Table 5.1 for specific analyzers used by the mobile lab

2. Particulate Matter (PM)

Particulate matter was analyzed using a Diesel Particulate Mass Monitor. This mass monitor provides real-time measurements on the particulate mass generated by the exhaust stream. This microbalance instrument measures the mass of a series of TEOM filters every 0.83 seconds. This real time data allows the comparison of particulate mass flow rate and engine performance.

D. Emissions Data and Data Reduction

1. Gaseous Emissions

CO, CO_2, NO_x and HC data from the mobile emissions lab were recorded at 10 samples per second. Reduced emissions data are provided in Table 5.2 and raw data are included in Appendix 6.

Carbon monoxide concentrations over the series of power ranges tested varied between 350 ppm and 490 ppm for aviation kerosene and between 330 ppm and 390 ppm for methanol. Figure 5.1 is a graph showing CO emissions in g/s versus compressor bleed air power. When running on aviation kerosene fuel CO emission increased from 380 ppm at idle to 487 ppm at 60 KW. Then CO emission fell off to 355 ppm at 107 KW. Emissions results when burning methanol were similar. CO emissions climbed from 378 ppm at 60 KW to 389 ppm at 82 KW. Then concentrations fell off to 338 ppm at 107 KW.

Table 5.2A

Gas Turbine Emissions Testing 9/15/98 & 3/14/00

(not available electronically)

Table 5.2B

Gas Turbine Emissions Testing 9/15/98 & 3/14/00

(not available electronically)

Carbon dioxide concentrations varied between 13,000 ppm to 30,000 ppm on tests using aviation kerosene and between 20,000 ppm and 28,000 ppm . CO_2 concentrations for both aviation kerosene and methanol increased as power increased. Testing on aviation kerosene indicated a CO_2 concentration of 14,650 ppm at idle which increased to 29,534 ppm at 107 KW. methanol test results indicated a concentration of 20,665 at 60 KW which increased to 27, 670 ppm at 107 KW. (Figure 5.2).

Oxides of nitrogen concentrations varied between 15 and 37 ppm when using aviation kerosene and between 7 and 12 ppm when using methanol. Tests done when running aviation kerosene show NO_x concentrations gently increase from 15.79 ppm at idle to 25.41 ppm at 70 KW and then increased more rapidly to 37.35 ppm at 107 KW. Concentrations when using methanol increased rapidly from 7.52 to 12.31 ppm between 60 KW and 82 KW.

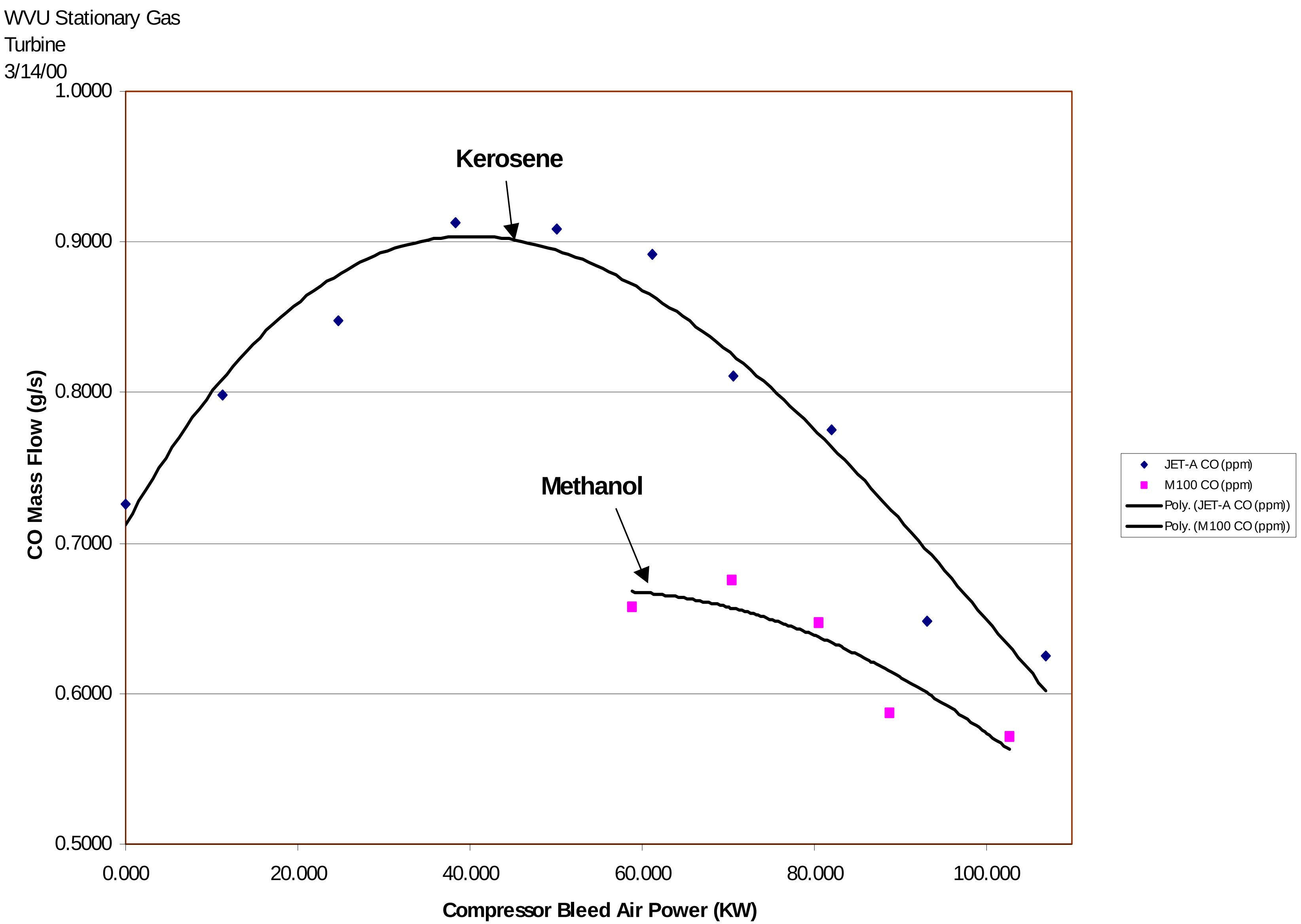


Figure 5.1 - CO Emissions in g/s versus Compressor Bleed Air Power

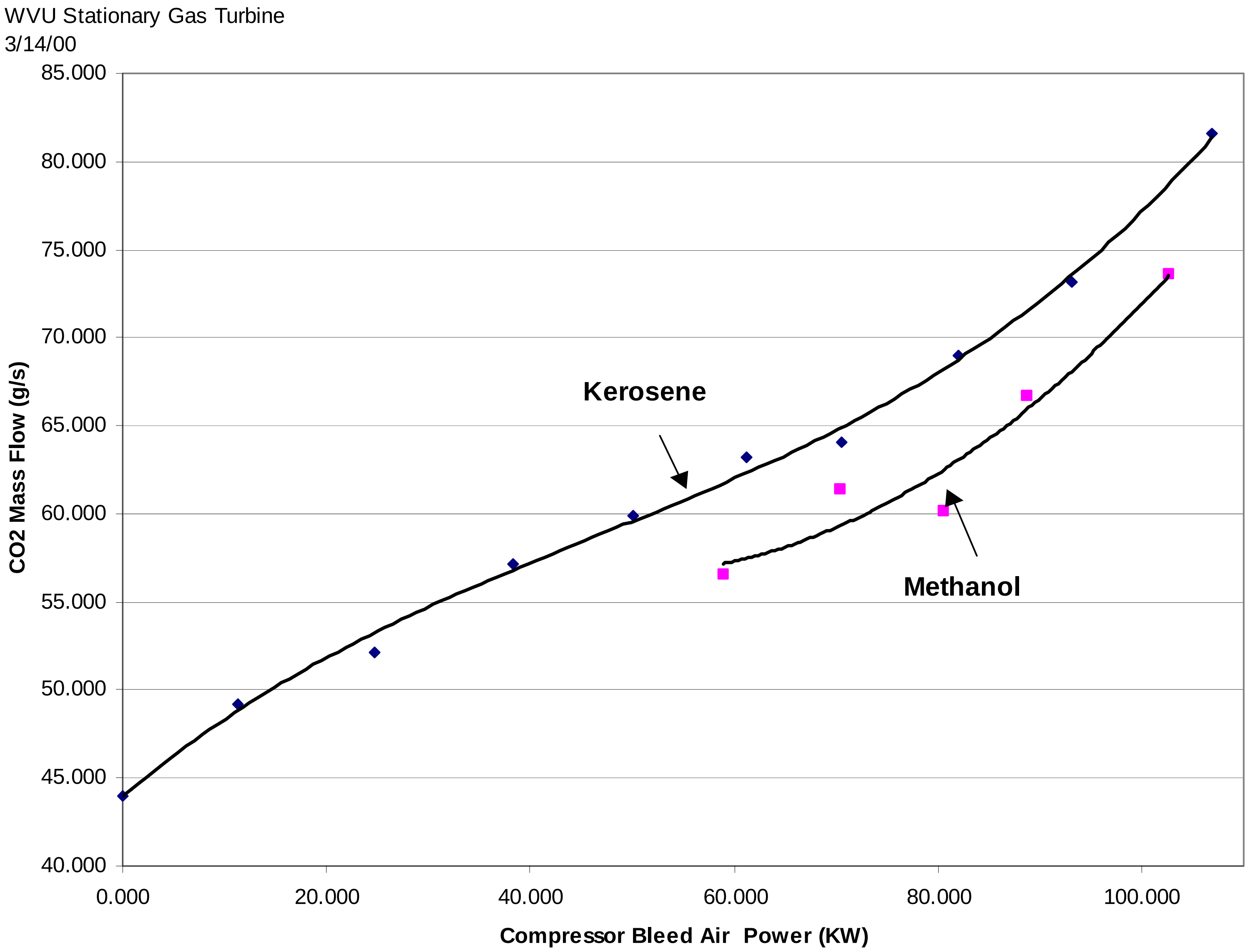


Figure 5.2 - CO2 Emissions vs. Compressor Bleed Air Power

Then concentrations fell to 8.80 ppm at 82 KW an then gradually increased to 10.44 ppm at 107 KW. (Figure 5.3).

Total hydrocarbon concentrations varied between 84 and 184 ppm for tests done on aviation kerosene and between 69 and 148 ppm when burning methanol. HC concentration when compared with power showed erratic behavior with some of the lowest reading occurring at the highest powers (Figure 5.4).

2. Particulate Emissions

Particulate mass concentrations varied between 2 and 10 ppm during aviation kerosene tests and 0.74 and 6.74 ppm during methanol tests. Both sets of data show a very general trend of lower concentrations at higher temperature and power. See Figure 5.5 and Table 5.2.

3. Data Reduction

The air fuel ratio is calculated using the measured turbine air inflow rate and compressor bleed air flow rate together with fuel flow rate. This is done in a simple computer program, for example see test 2J, shown in Table 2.1, and other test data as shown in Appendix 7. Program formulas are also listed in Appendix 7. For example in test 2J on aviation kerosene the stoichiometric air/fuel ratio by mass is 14.7. The burner airflow rate is 3.48 lbm/s and the burner fuel flow rate is 0.0456 lbm/s. This results in an actual air/fuel ratio 3.48/0.0456=76.31 or equivalence ratio Φ =14.7/76.31=0.19.

WVU Stationary Gas Turbine
3/14/00

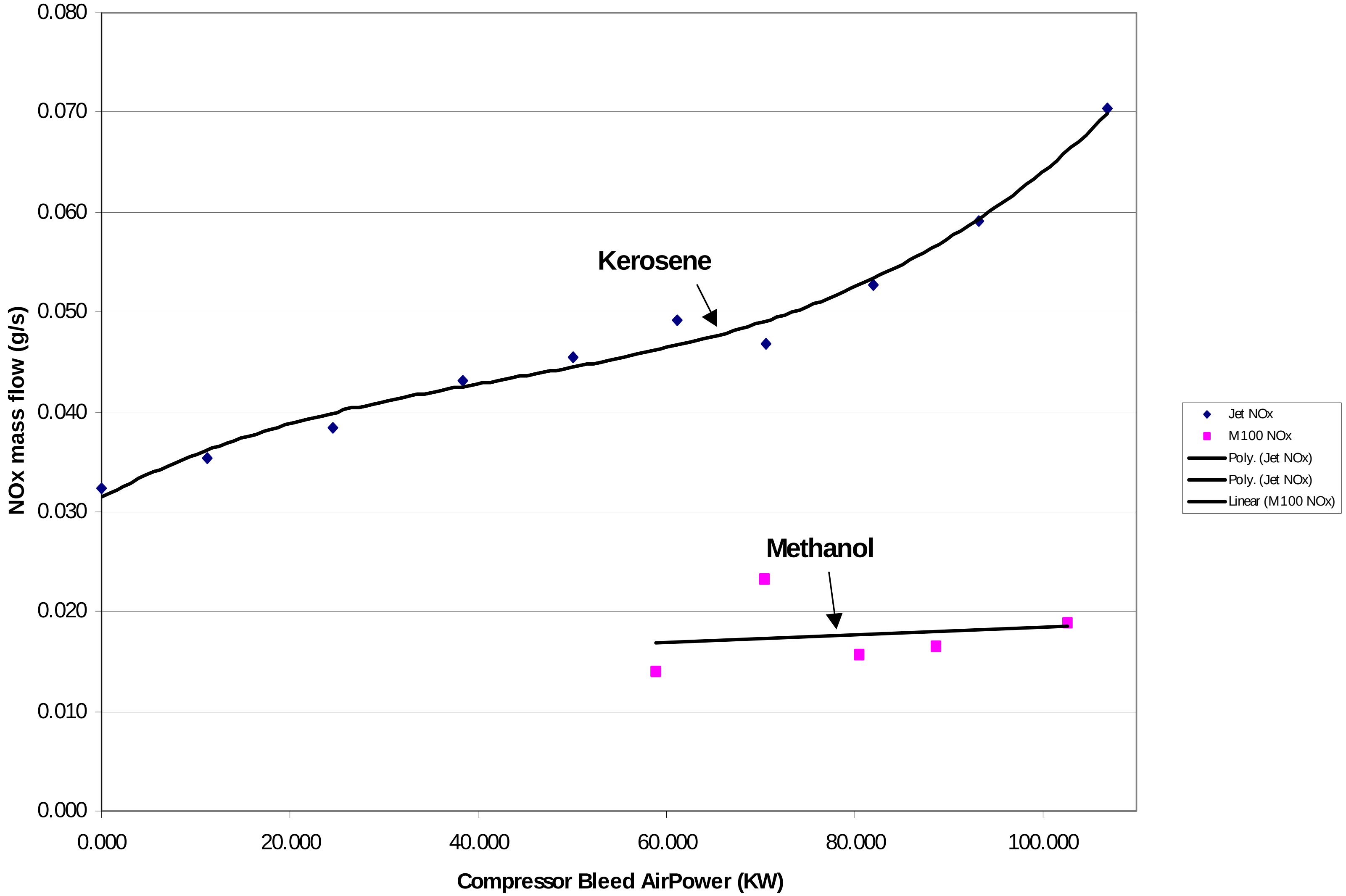


Figure 5.3 - NOx Emissions versus Compressor Bleed Air Power

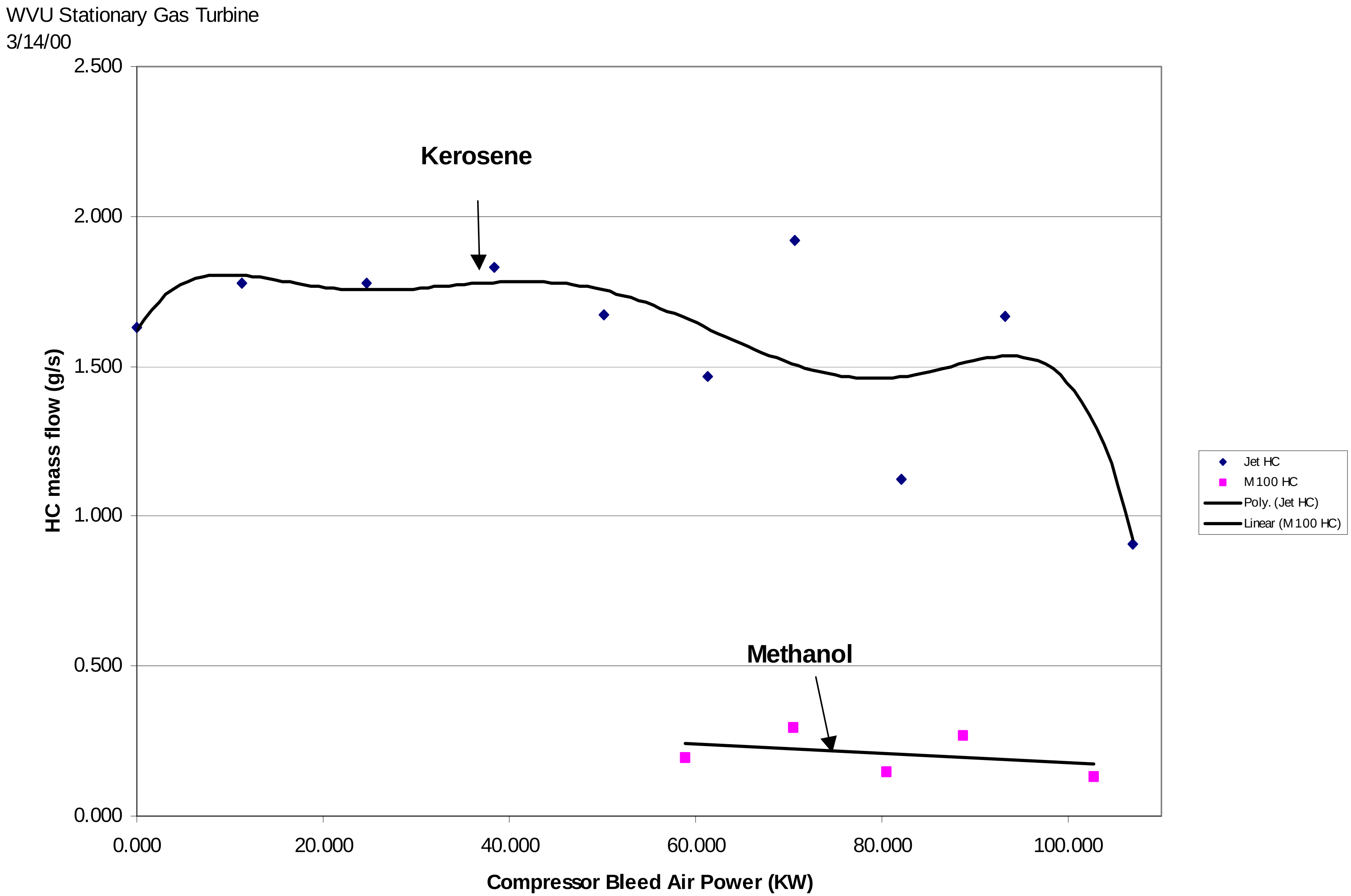


Figure 5.4 - HC Emissions versus Compressor Bleed Air Power

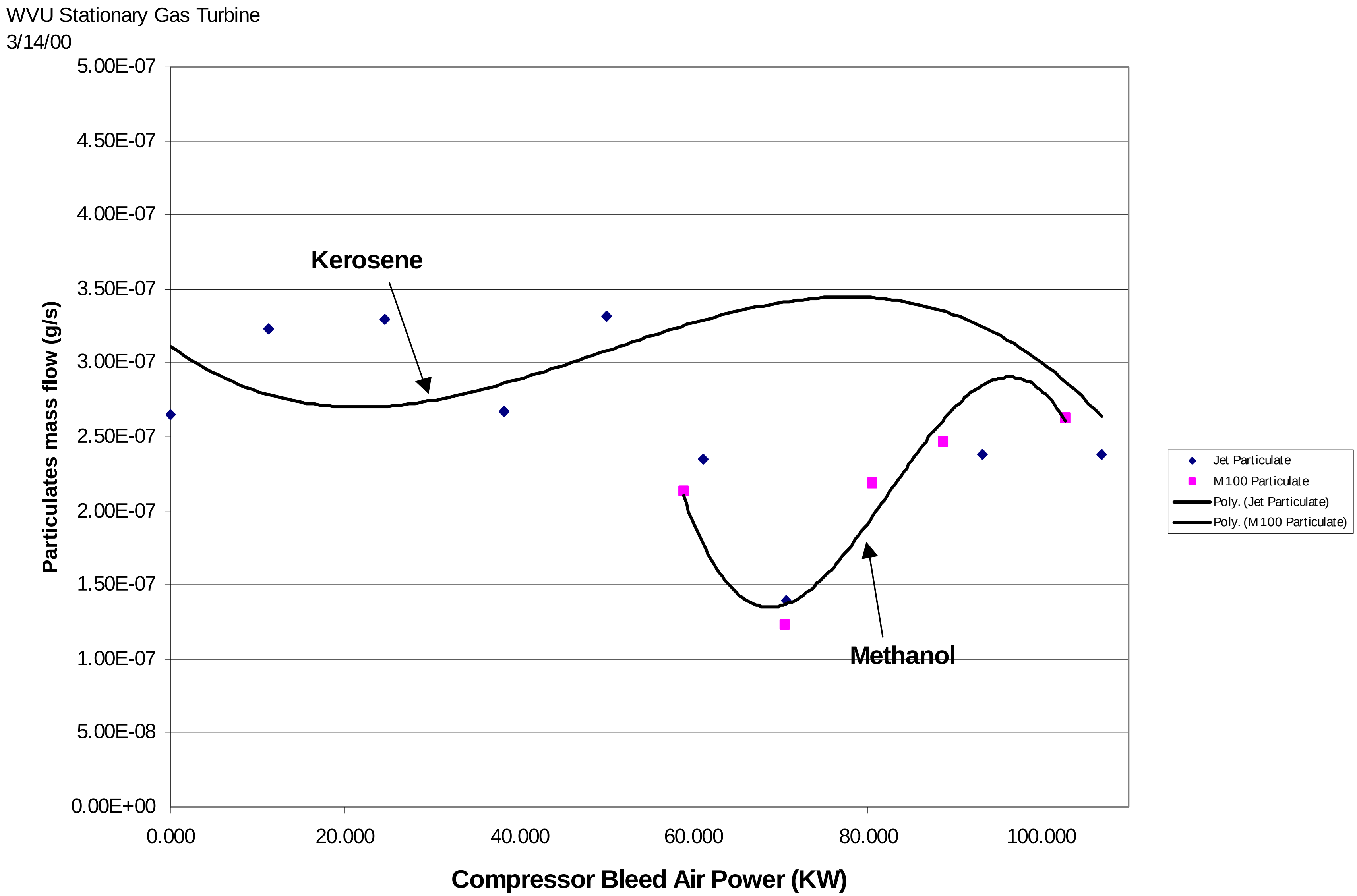


Figure 5.5 - Solid Particulate Emissions versus Compressor Bleed Air Power

In table 2.1, the turbine inlet temperature has been calculated three ways. First, the compressor bleed air power is calculated from the temperature rise and flow rate, this is 102 HP in test 2J. From the measured bleed air and total inlet airflow the compressor power is calculated, this is 485 HP. Equating this to the turbine power, allows one to calculate the turbine temperature drop. This added to EGT of 824°F in test 2J, provides the turbine inlet temperature 1221°F. The second and third methods are based on assuming 100% adiabatic combustion and neglecting emissions other than CO_2, H_2O, O_2 and N_2. The expected results will be slightly higher. They are 1280°F using a mean specific heat and 1241°F using individual specific heats. Turbine inlet temperatures can be seen in Figure 5.6 and Appendix 7.

During the transient fuel type change-over maneuver, data collection was continued at 10 Hz. Such high-speed data acquisition was essential, as the fuel-change-over lasts less than 0.5 minutes, depending on the power setting. In that time the fuel concentration ratio changes gradually from 0% to 100%. These data are reported in ppm and are plotted as a function of time in Figure 5.7. This shows tests 14J and 15M. The turbine was operated on aviation kerosene until a steady state idle condition was reached at which the data acquisition was initiated at t = 0. The compressor power was increased to 70.62 KW at 3.5 minutes.

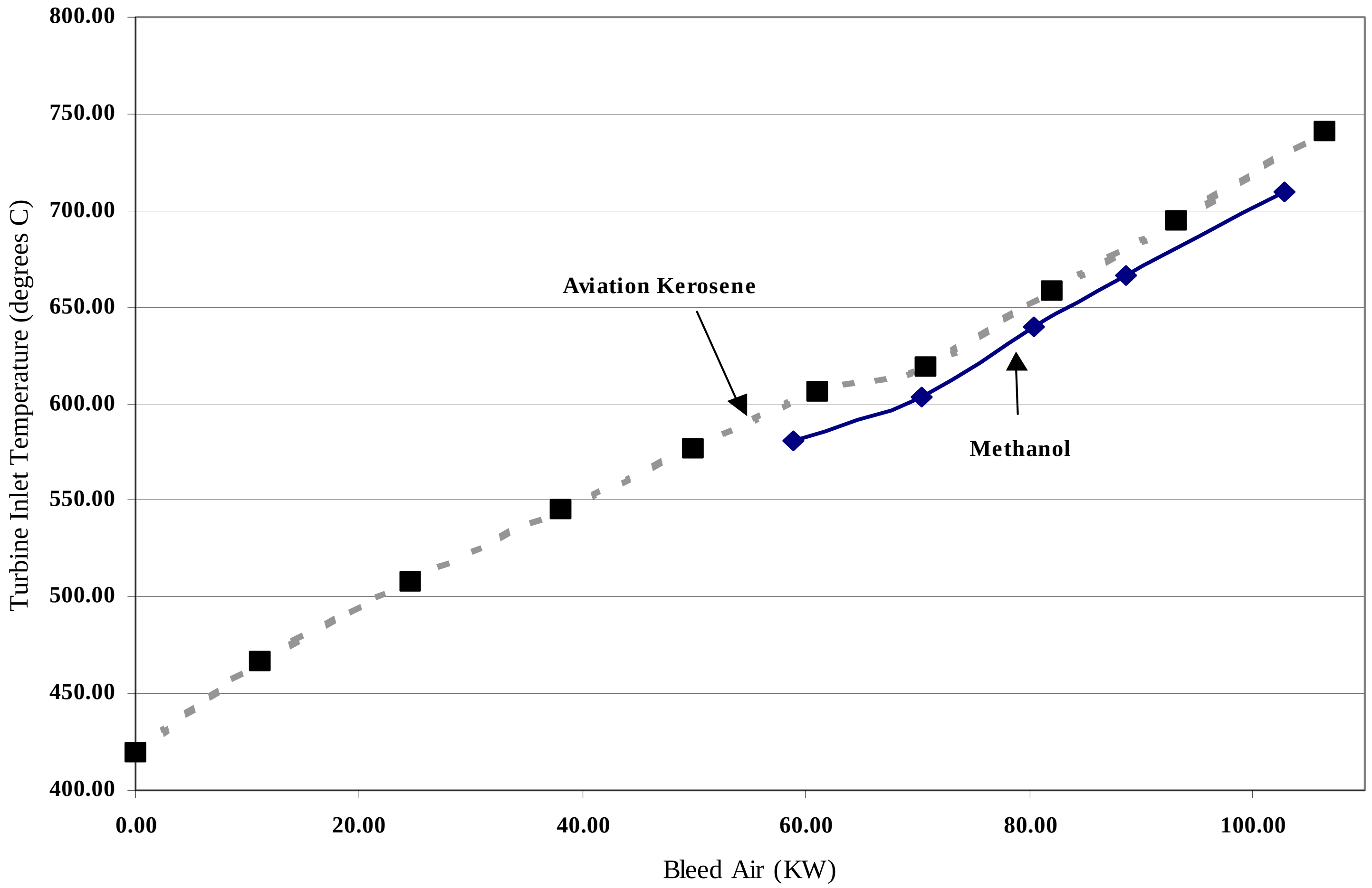


Figure 5.6 - Turbine Inlet Temperature versus Bleed Air Power

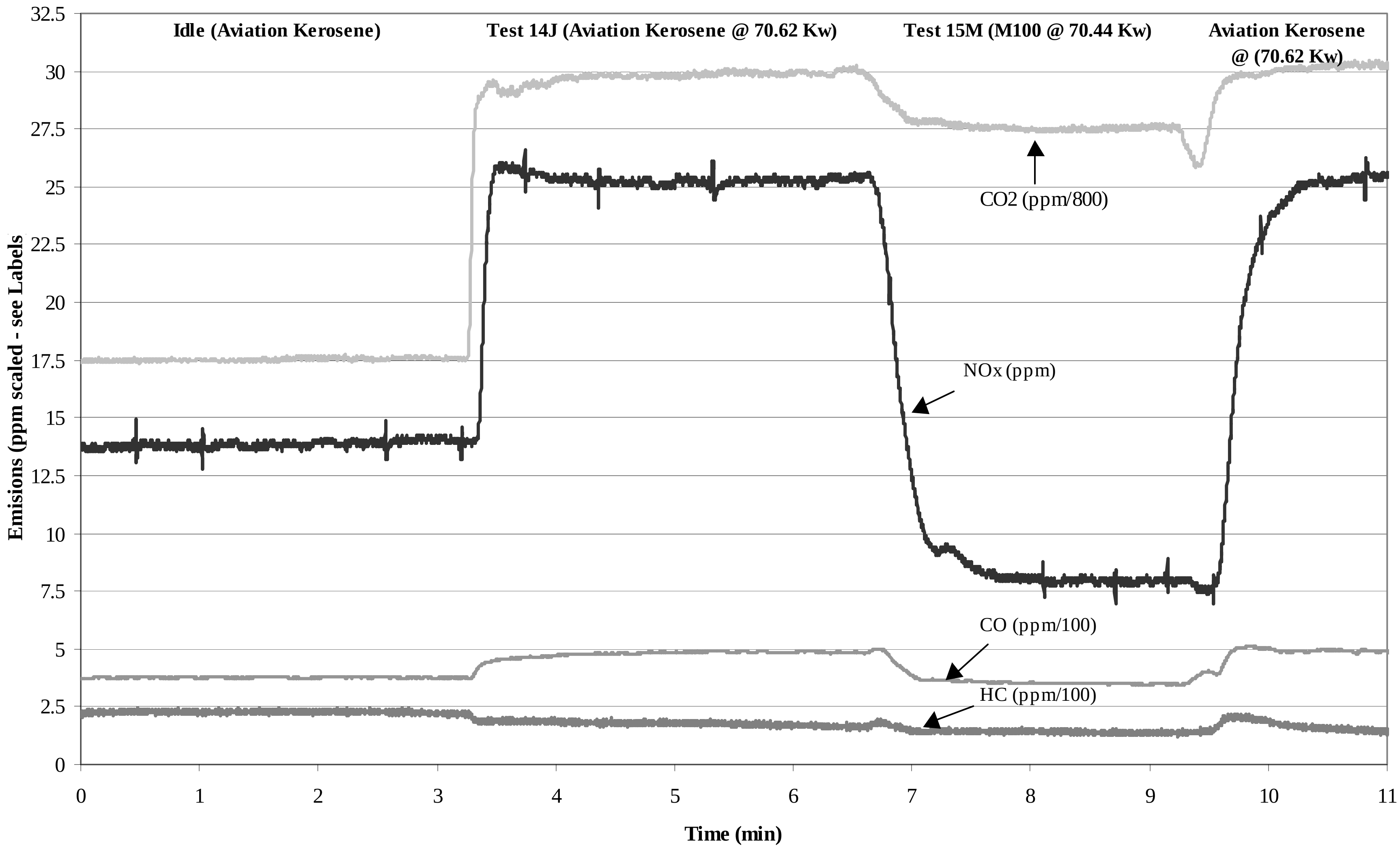


Figure 5.7 - Emissions During Fuel Type Change Over From Aviation Kerosene To Methanol, 3/14/00 Tests - Idle, Test 14J (Aviation Kerosene @ 70.62 Kw), 15M (70.44 Kw) And Back To Aviation Kerosene @ 70.62 Kw

Table 5.3

Reduction in Emission When Switching from Aviation Kerosene to Methanol/2 Cycle Blue Solution

(not available electronically)

CO, CO_2, NOx and HC increased with power, NO_x increasing the most, rising from approximately 14 ppm to approximately 25 ppm. Fuel type change-over was then initiated, from aviation kerosene to Air Products methanol. During fuel change-over all emissions species concentrations tested, decreased with CO_2 and NO_x decreasing the most. CO_2 decreased from approximately 24000 to 21500 ppm and NO_x decreased from approximately 25 to 8 ppm. Immediately following this change-over, Figure 5.6 shows this dramatic decrease in NO_x production during aviation kerosene dilution process. At approximately t = 7 minutes, one minute after the initiation of the fuel change over, the NO_x data approached the pure methanol equilibrium value. At t = 9.5 minutes the reverse fuel type change-over, from methanol to aviation kerosene, is initiated. Following this procedure, the NO_x production rapidly approaches the aviation kerosene steady state value as represented by value at t = 4 minutes. It can be seen from Figure 5.7 that while fuel type has a strong effect on the NO_x production, it has a much lesser effect on the other species sampled. A compressor power of between 70.4 and 70.6 KW was maintained during the tests shown in Figure 5.7. The results shown from tests 14J and 15M are typical of the other tests conducted.

VI. CONCLUSIONS

The initial test objectives of this research were accomplished using the GTC-85-72 gas turbine It was successfully operated on both kerosene and on fuel grade methanol as produced by Air Products and Chemicals Inc. Emission data were collected on each fuel during steady state (defined as unchanged during at least 6 minutes). In addition emission data were collected during the transient fuel-change over procedure, which lasted about less than 0.5 minutes. Engine starting proved to be only possible on kerosene. It is suspected that this is due to the combination of the low volatility of methanol and the high heat of vaporization. This engine is designed for compressor bleed air, so it runs leaner than shaft power output engines.

To minimize corrosion and diaphragm deterioration during storage, and permit starting, it was decided to conduct a change over to methanol only after the engine was warmed up and return to kerosene prior to engine shutdown. To achieve successful fuel changeover it proved to be essential to raise the EGT to more than 750°F, which is done by applying at least, 25% bleed air load. Even at this elevated EGT value was it necessary to increase the ignition power. This was achieved by adding two glow plugs from a PT-6 aircraft gas turbine to the existing spark plug. The inability to operate on methanol at idle could also be due to the cooling effect from the high heat of vaporization. This stretches the flame further downstream in the burner where the mixture is diluted by secondary air, and may become too lean to ignite. This problem might be solved by extending the burner by four inches in length in between the primary and secondary air supply zones.

The unmodified fuel controller and atomizer were unable to supply enough methanol to permit operation at more than 50% bleed air. The lack of lubrication when using methanol caused the ball bearing and cylindrical valve of the RPM controller to seize up which resulted in loss of RPM control. Also the gear type fuel pump housing was so badly worn that the fuel pressure dropped below rated values.

These were some of the significant operational problems encountered during the first phase of testing on pure methanol. The emission testing presented no difficulties. The ppm emission data are readily convertible to units of cc/s. The conversion coefficients, calculated in the program for each test, are in the order of 1 to 3 cc/s per ppm.

The significant change in NO_x level when running on pure methanol from about 25 ppm on kerosene down to about 5 ppm on methanol, is most likely caused by the reduced flame temperature using nearly double the fuel flow rate and taking advantage of it's high latent heat of vaporization. The graph of the transient test emissions is show in Figure 2.13.

While operating on pure methanol, the lack of lubricating properties destroyed the bearings and the cylindrical RPM control fuel valve inside the fuel controller. It became imperative that all future tests on methanol in the GTC-85-72 gas turbine must incorporate a suitable lubricant additive.

The lubricating properties of kerosene, methanol and methanol additive mixtures needed to be measured in order to choose a suitable additive. Measuring lubricity with an existing WVU lubricity-testing machine proved to be unproductive. The scatter in the data gave us only trends, but no exact values. Assessment of suitable additives would only be possible after a lubricity tester was developed, as no existing equipment was available. Three types of test apparatus were designed and tested at WVU and hundreds of tests were run without satisfying results. Finally the last configuration (Type 4), shown herein provided repeatable and accurate data. Three suitable additives were found and an index of the final series of successful tests is shown in Appendix 2.

The new lubricity test apparatus designed and tested at WVU was relatively easy to use and provided the needed repeatable data. Each run was conducted over a 10-minute period. Conducting the tests at 3.5% of a lubricated bearing design load proved to be the most successful. It was also found that this system yielded an experimental repeatability far greater than that possible with the wear based lubricity-testing methods. Test results indicate that all three additives tested would be satisfactory for use in the WVU gas turbine. The *Two Cycle Blue* additive appeared to offer the best lubricity for the lowest concentrations when mixed with methanol. A 1% solution was sufficient to match the lubricating properties of kerosene.

To provide an adequate factor of safety, all the second phase emissions testing on the WVU gas turbine were conducted using a 2% methanol-*Two Cycle Blue* solution. The gas turbine was operated for an extended period, with out problems.

A new larger fuel controller and atomizer of the GTCP-180L gas turbine, which are capable of supplying fuel at 600 psi and 750 lbs per hour were installed in the WVU turbine. Along with the use of glow plugs these modifications increased the operating range of the gas turbine when fueled by methanol from 48 and 58 KW compressor bleed air power in the 1998 tests to 48 and 103 KW compressor bleed air power in the year 2000 tests.

Emissions testing while operating with the new methanol-*2 Cycle Blue* additive solution shows reductions in all species tested when compared to gas turbine operation when using kerosene. These reductions in emissions are apparent all power ranges tested. Most significant of these reduced emissions was NOx, which in percent reduction decreased according to the power level between 47 to 72 % when switching from kerosene to methanol. HC emissions between 83 to 87 %, CO2 - 4 to 11 %, and HC - 9 to 26%. Table 5.4 summarizes emissions reductions when switching from kerosene to methanol.

Operation over the entire rated power range of the GTC-85 is still not possible when running on methanol. To understand this problem, the full combustion process needs to be studied. A combustor instrumented with 4 thermocouple was tested in March 2000 and proved that accurate temperature profiles can be taken in the combustion region. A numerical model of the methanol combustion process inside the GTC-85-72 gas turbine combustor is needed. This model then needs to be compared to the available data and when in agreement may explain the flameouts experienced at idle and also the reduction in rated power. Such understanding is invaluable for future designs of combustors capable of operating on methanol over the entire intended power range.

VII. RECOMMENDATIONS

- The lubricity of methanol produced with the LPMEOH^{TM} can be made equal to that of aviation kerosene by the addition of less than 1% of the commercially available racing fuel additive *Two Cycle Blue*. However it is recommended that a more economical additive be found to make methanol more cost effective as an alternate fuel.
- The combustion chamber of the GTC-85-72 used at West Virginia University for emission testing was designed for operation on aviation kerosene. The fuel pump pressure, flow rate and spray nozzle size had to be doubled to be able to develop full power when switching to methanol. Further all seals have to be methanol resistant. Potential customers should be made aware of the need to make these modifications before switching to methanol.
- The problem of flame-out at less than 25% power, when switching from aviation kerosene to methanol at an EGT below 714°F needs to be solved. This problem also makes operation on methanol at idle or starting on methanol impossible. The turbine inlet and exit temperatures are only a function of bleed air power setting and are nearly independent of the fuel type in use. Therefore, the flame-out problem must be created upstream in the primary combustion zone of the combustion chamber. The cause of this problem needs to be studied and eliminated in future combustion chamber designs
- It is recommended to write a CFD code to determine the difference in required length of the primary combustion zone with methanol and with aviation kerosene. Further, it is recommended to verify the CFD code output experimentally by

installing an aspiration probe, capable of an axial survey of the combustion products along the length of the WVU gas turbine combustion chamber. Also, create radial temperature profiles for CFD validation with the four thermocouples, currently installed in the combustion chamber.

- It is anticipated that an additional separation zone, installed between the primary combustion zone and the secondary dilution air entry, will solve the flame-out problem, without having to resort to dangerous fuel pre-heaters to solve this problem. Such design information seems essential before the wide spread adoption of methanol fuel in small gas turbines. Note the available radiation heat in large gas turbines may mask this problem.

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APPENDIX 1

updated Oct 9
U. S. DEPARTMENT OF ENERGY
AND
AIR PRODUCTS AND CHEMICALS, INC.

METHANOL FROM COAL END USE DEMONSTRATION PROJECT

West Virginia University
Gas Turbine Emissions Study
(LOTH/CLARK, 1998)

Final Contract Report

October 14, 1998

Prepared for Contract Monitor

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West Virginia University
Gas Turbine Emissions Study

Summary

The object of this study is to demonstrate operation of a stationary gas turbine on Fuel Grade methanol, produced in La Porte Texas by Air Products and Chemicals Inc. Of interest is a comparison of the operational aspects and emissions between Jet A and Fuel Grade methanol. The gas turbine selected was a GTC-85-72 normally used at airports to start large jet engines with its 235 HP of compressed bleed air. This gas turbine is currently installed in the WVU experimental "Circulation Control High Lift Technology Demonstrator" aircraft. To operate this unit on methanol the following items needed to be modified: instrumentation, fuel supply system, fuel controller, ignition system and bleed air load control. One of the two WVU portable emission analysis laboratories was brought in for the gaseous and particle emissions study.
Jet A and Fuel Grade methanol are pumped directly from 55 gallon drums into a common manifold with fuel flow meter. A gradual change-over in fuel mixture ratio is desirable, to allow the gas turbine fuel controller time to adjust the fuel flow rate by up to 85% when changing over to methanol. By installing a fuel emulsifier loop in the fuel selection manifold, a gradual mixture ratio changeover can be obtained. By making the volume of the emulsifier loop equal 1/3rd of the GPM fuel flow rate, the change-over can be made to take about 20 seconds for completion. Another significant difference between these two fuels is that methanol has a five times higher heat of vaporization than jet A. The associated cooling effect required the combustor can spark plug ignition source to be augmented. For this purpose, two glow plugs from a PT-6 gas turbine were installed. Even then, flameout during fuel change over, could only be prevented by operating under at least 25% bleed air load. At this power level the exhaust gas temperature (EGT) is at least 750°F. At bleed air power level above 50%, the fuel controller with associated burner nozzle size, was unable to supply the required methanol flow rate and a gradual decrease in turbine RPM resulted followed by flame-out. Within the 25% to 50% bleed air power load level, the fuel change over is perfectly smooth. The NO_x emissions dropped from about 25 ppm on jet A to below 5 ppm on methanol. The EGT is about 75°F lower on methanol than on jet A. This alone does not explain this significant reduction in NO_x. The most likely reason is that methanol burner nozzle spray evaporates so slowly that it extends into the burner region were the secondary dilution air reduces the flame temperature thereby reducing the thermal NO_X production.

Other than associated problems with idling this may be a beneficial aspect of operating on methanol. The GTC gas turbine fuel flow rate controller contains a cylindrical valve, which is activated by flywheel weights. The surrounding fuel is supposed to lubricate the components and their bearings that are submerged in the fuel. The lack of lubrication in the methanol caused the ball bearings and the cylindrical valve to seize up during the emissions testing. The operational problems delayed and somewhat limited the emissions testing. These operational problems can be solved by adding a methanol lubricant such as Lubrizol. By increasing the size of the fuel controller and the size of the combustor nozzle this GTC-85-72 gas turbine can be modified to operate at 100% power on methanol. In order to operate at low load levels down to idle, one has to extend the burner can. This will allow completion of the methanol combustion, prior to secondary air dilution. By surveying the burner axial temperature distribution, one can calculate the required burner extension.

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Appendices

1) Introduction

The fuel grade methanol, storable gas turbine fuel, produced in La Porte, Texas by Air Products and Chemical Inc. is being tested for a variety of applications. This includes its application in diesel engines, gas turbines and fuel cells. Air Products and Chemical Inc. under partial support by the Department of Energy has contracted out various demonstration projects to evaluate their fuel. Two of these projects were conducted at West Virginia University, one on diesel engine emissions and the other on gas turbine engine emissions.

West Virginia University faculty and students have received national recognition for their work on transportation engine conversion to alternative fuels. These include compressed natural gas (CNG), liquified natural gas (LNG), methanol, ethanol and others. A large number of alternative fuel transportation engines in: cars, trucks, busses, marine engines and aircraft are in use throughout the country. As a service to fleet owners, operating alternative fuel heavy duty trucks and busses, WVU operates two mobile emission testing laboratories throughout the US and Canada. WVU has converted a Cessna 150 aircraft, which now operates on either aviation gasoline or E95 ethanol. Its excellent performance and in flight fuel change over capability contributed to the Department of Energy dedicating the Morgantown Airport as the "2nd Clean Airport in the USA", in 1997. WVU's experience in converting engines to operate on alternative fuels resulted in this demonstration project contract. Allied Signals Aerospace, formerly AiResearch/Garett, manufactured the WVU GTC-85-72 gas turbine. Their technical representative Mr. Jessup Hunt did not anticipate any problems with operation on alcohol fuels, other than deterioration of the rubber fuel hoses and diaphragms in the fuel controller and solenoid valves. He anticipated the need to increase the size of the fuel controller and pump as well as the burner nozzle. To minimize corrosion due to long term exposure to alcohol fuels it was decided to always start and shut down our gas turbine using jet A fuel. Only after the engine was properly warmed up would the operator change over to methanol. As these two fuels do not mix readily and have widely different heating values, the fuel controller must have time to gradually alter the fuel flow rate so as to maintain the near constant turbine RPM. These requirements determined the unique fuel supply system, which had to be designed for the methanol demonstration tests.

The 235 HP WVU GTC-85-72 gas turbine was acquired in the early 70's to provide compressed air at the rate of 2 pound per second to develop an experimental aircraft high lift system. The 40 psi air supply was distributed along the wings trailing edge. The air flow rate was doubled in a supersonic ejector which provided boundary layer control by suction and the retractable flap hinge and circulation control by blowing over the rounded trailing edge. The aircraft was successfully test flown in 1974, and capable of operating at a wing lift coefficient CL=6.

It was decided to leave the gas turbine in the airplane for the test, as this test did not require a dynamometer for engine loading. The compressor bleed air flow represents the load. A special manifold with up to eight calibrated 5/8th of an inch diameter choked flow nozzles was used to increase the load in 12.5% increments from zero to maximum 2 pounds per second flow rate.

For exact power and turbine inlet temperature calculations the inlet air flow rate had to be measured. This was achieved by installing a 7 inch throat diameter venturi. This report describes in detail the test equipment, operating procedures used, data collected and data reduction techniques used. Photographs and drawings are used to explain the set-up and instrumentation.

2) Test Set-Up

The GTC-85-72 gas turbine which is installed in the West Virginia University STOL research aircraft had to be modified for use in this research project. For safety reasons, a shield was installed on one side of the airplane to protect the operator from the gas turbine. The operational controls and instruments for the turbine, including the starter switch, air bleed switch, tachometer and compressor pressure gage were relocated from the cockpit to the operator side of the airplane. In addition to these controls, engine performance measuring equipment had to be installed. This includes K-type thermocouples to read the: exhaust gas temperature, bleed air temperature and venture inlet air temperature.

Additional hardware required for testing includes a motor-generator set gas turbine start cart used to supply the required 26 volts for start-up. To measure the total mass flow into the turbine, a venture with a 7 inch throat diameter was installed in-line with the turbine intake. The vacuum reading in the venturi throat was used in addition to the atmospheric pressure and temperature to calculate the engine air flow rate. Power loading was accomplished through the use of a bleed air manifold containing various numbers of choked flow metering nozzles. Bleed air power was calculated from the total nozzle area, bleed air pressure and temperature, which were read from a pressure gage and K-type thermocouple respectively. The test fuel was pumped directly from a 55 gallon drum to the fuel selector valve system, described in section 4.

3) General Description of the Gas Turbine

The GTC-85-72 engine is a gas turbine auxiliary power unit (APU) which is mostly used to provide pneumatic jet engine start-up power at airports. This particular engine was manufactured by AiResearch/Garrett in the late 1960's. In 1972 the engine was installed in the West Virginia University STOL research aircraft, Figures 3.1 & 3.2. This aircraft is currently no longer airworthy and therefore grounded.

There are six basic engine assemblies, which include: the compressor section, the turbine section, the combustion chamber, the lubrication system, the electrical system and the fuel flow and RPM controller, Figure 3.3.

Compressor Section

The centrifugal compressor provides about 40 psig compressed air for the turbine and the bleed air for pneumatic power. The compressor is a two stage centrifugal type with a pressure ratio of 3.4: 1 and a total air mass flow of 5.5 lb./sec, at 40,800 rpm.

Turbine Section

The turbine section provides power to the compressor and the accessories and is designed to operate up at temperatures up to 1200° F.

Combustion Chamber

The combustion chamber is a reverse flow can type, which is comprised of a cylindrical liner mounted concentrically inside a cylindrical casing. The chamber's key components include an air casing, diffuser, liner, fuel atomizer, glow plugs and spark igniter, Figure 3.4.

Lubrication System

The lubrication system is a self-contained positive pressure, dry sump type. This system provides pressurized splash lubrication to all gears, shafts and bearings.

Electrical System and Instrumentation

The electrical system requires approximately 26 volts DC to operate the starter, solenoid, instrumentation and the ignitions system. The ignition system is a high-energy step up transformer charging capacitors, which build up voltage across the igniter plug. In addition to the igniter, a pair of 8 amp glow plugs, Figure 3.5, and their voltage regulator from a PT-6 jet engine have been added to provide a higher energy ignition source. Power is supplied to this system by a 26 volt DC generator for the main engine circuits and a 24 volt battery for the glow plug voltage regulator.

Instrumentation for the engine's operation and for testing include three K-type thermocouples located to measure exhaust gas temperature, bleed air temperature and ambient air temperature, one gear driven tachometer, one compressor outlet pressure gauge, one bleed-air pressure gauge, one fuel pressure gauge, and one charging voltage gauge.

Fuel/RPM Controller and Bleed Air Valve

The fuel and bleed air control system automatically adjusts fuel flow to maintain a near constant turbine operating RPM under the varying load conditions, which depends on the amount of bleed-air extracted. A gear in the accessory section drives the fuel pump and control unit, Figure 3.6. This system incorporates a gear fuel pump capable of 230 psi, fuel filter, acceleration limiting valve, fuel pressure relief valve, fuel solenoid, and connections for the pneumatic control, and electric control. A constant operating speed is achieved through a combination of an acceleration limiting flyweight-type governor bypass fuel dump valve and a diaphragm bypass valve activated by the bleed air pressure. Fuel is transferred under pressure to the fuel atomizer located in the end of the combustor cap. The fuel atomizer consists of a screen, a flow divider valve, distributor head and housing. The distributor head divides the fuel passageway within the core. The center passage leading to a small orifice plate and a annulus leading to a large orifice. The flow divider valve directs fuel at low pressure through the small center orifice and at high pressure to both the small and large orifice. During May 1998 the fuel atomizer was calibrated in a spray booth at Pratt and Whitney Engine Services in Bridgeport West Virginia. This calibration was necessary to ensure that there would be adequate atomization and correct spray cone geometry under all the operating pressures expected during operation of the engine with Jet A and methanol, Figure 3.7.

4) Fuel System Design

For safety reasons a separate fuel system was designed so that it could be disconnected at the end of each test and stored in an approved storage facility. Because of the corrosive nature of methanol, and to eliminate cold starting problems, it was necessary to perform engine start-up and shut-down using conventional Jet A. The gas turbine is started on Jet A, operated under load to bring the combustor up to operating temperature before gradually changing over to methanol. After the tests are completed, the fuel type was changed back to Jet A prior engine shut-down.

To accomplish the desired fuel change-over procedure, a special fuel supply system was developed. It consists of two 55 gallon DOT #17 fuel drums one containing methanol and the other containing Jet A, Figure 4.1. Each of these drums was equipped with a separate pneumatic powered fuel pump, capable of 4.6 gpm, which discharges to the fuel type selector valve, Figure 4.2. The selected fuel then traveled to the fuel emulsifier. This allows a gradual change in mixture concentration during fuel type change-over. The components of this emulsifier are shown in Figure 4.2 they consist of a small orifice, a clear sight glass and a recirculating pneumatic fuel pump. During fuel change-over, this sight glass becomes cloudy with the emulsified Jet A/methanol mixture. Downstream of the fuel emulsifier, a fuel pressure spike damper was installed, Figure 4.2. This damper consists of a volume of captured air in a clear sight glass to compensate for the pulsating nature of the pneumatic fuel supply pumps. Following the pressure spike damper, the fuel was routed to a volumetric flow meter and from here on to the gas turbine fuel controller.

5) Problems Encountered During Turbine Operation

During the course of this project, various unforeseen problems were encountered. The first of which was engine flame-out due to the too sudden fuel type change over. This problem was solved by the addition of the fuel emulsifier recirculating pump described in section 4.

With the modified fuel supply system, another problem surfaced, in that the gas turbine would not operate at idle or even at very low power settings on methanol. This is believed to be due to the nearly 5 time greater heat of vaporization of methanol when compared to Jet A. Because of this, methanol requires more ignition energy upstream of the point where the dilution air enters the burner.

A second, and predictable, operation limitation was uncovered whereby the gas turbine could not be operated on methanol at high power levels. This is due to the inability of the fuel system to double the volumetric fuel rate flow for the same combustion temperature when operating on methanol. If fuel type change-over from Jet A to methanol was attempted at a high power setting, then the turbine experiences a gradual loss in RPM, which terminates in combustor flame-out. To operate this turbine on methanol at these elevated power settings, a new fuel controller system capable of higher flow rates will need to be installed.

In addition to the power operation limitations found when operating on methanol, additional durability issues were encountered. The first of these was the quick destruction of the aged rubber diaphragms in the gas turbine fuel controller. These diaphragms failed after only a short exposure to the methanol fuel. As a result, this fuel controller was rebuilt using all new diaphragms and seals. After overhauled it performed flawlessly throughout the remainder of the tests. However, one additional problem was experienced. This was the destruction of fuel controllers RPM governor due to the lack of lubricating property of methanol. It is proposed that the use of a lubrication additive such as Lubrizol be used to eliminate this type of problem.

6) Emissions Testing Equipment

WVU Mechanical and Aerospace Engineering (MAE) has designed and built two mobile emissions testing labs that are capable of testing vehicles up to 30,000 kg (66,000 lbs.) in the field. WVU has tested over 700 buses and trucks from more than thirty-five locations throughout the United States. Much of the data collected from the buses and trucks are available in database form.

The mobile emission lab is comprised of two tractors, an emissions measuring instrument trailer and a flat-bed with the rollers, flywheels and power absorbers, Figure 6.1. Inside the instrument trailer there is an environmental chamber for preparation of the particulate filters and a microgram scale for measuring them, there are also precision gases for calibrating the analyzers , racks of data acquisition and dynamometer control equipment, emissions analyzers etc. The trailer also has a blower and the power supply for the sonic flow venturi constant volume sampling (CVS) system and the stainless steel dilution tunnel on top of the instrument trailer. The emissions lab can measure and characterize emissions from a wide range of vehicles that use various types of fuels. Most of the vehicles tested use alternative fuels. The exhaust emissions from the vehicle are measured using a dilution tunnel and full exhaust gas emissions measurement instrumentation. Each test is run three times to ensure repeatability and data quality. The laboratories measure carbon monoxide, carbon dioxide, oxides of nitrogen or NO_x, methane, total hydrocarbons, aldehydes and particulate as per USEPA standards. Figure 6.2 shows the emissions lab set up for testing of the GTC-85-72 gas turbine installed in the WVU STOL airplane.

7) Data Collection and Reduction

The gas turbine was operated in near steady state conditions except for the fuel flow rate which varied slowly during fuel type change over. All turbine operating parameters to be measured, varied slowly enough, that the data could be collected manually by reading gages, see Figures 7.1. The transportable laboratory comes equipped with a standard 18 inch diameter dilution tunnel. It has choked flow metering nozzles, which are sized for various flow rates up to 3000 CFM. As its flow should be diluted to below 290°F, about two-thirds of the dilution tunnel flow must come from outside air. The GTC-85-72 gas turbine exhaust flow rate is about 3.5 pounds/second = 2700 SCFM at more than 700°F. Therefore the standard 18 inch diameter dilution tunnel cannot process this much exhaust flow. Instead a 3/8th cooled copper tube was inserted in the exhaust stack. A sampling pump draws a metered steady flow through the analysis equipment inside the transportable emission laboratory.

Carbon monoxide is measured by infrared absorption, nitrogen oxides are measured by chemical luminescence and unburned hydrocarbons are measured by flame ionization detection. From these the fuel/air ratio could be calculated. However in the gas turbine tests this is not necessary. From the measured turbine air inflow rate and compressor bleed air flow rate together with fuel flow rate, this ratio is determined. This is done in a simple computer program. For example in test #2 on jet A the stoichiometric air/fuel ratio by mass is 14.7. The burner air flow rate is 3.48 lbm/s and the burner fuel flow rate is 0.0456 lbm/s this results in an actual air/fuel ratio 3.48/0.0456=76.31 or equivalence ratio Φ =14.7/76.31=0.19. From an emission point of view this very lean equivalence ratio is meaningless as the combustion takes place near stoichiometric at the burner inlet. There NO_x and unburned hydrocarbons HC are formed as a function of an unknown equivalence ratio during combustion. After reaching peak flame temperature, the combustion products are diluted with secondary air to the allowable turbine inlet temperature. Only by measuring or modeling the temperature profile along the length of the combustor can one analyze the effect of dilution air on the NO_x and HC concentrations in the exhaust. Chemical kinetics show that the concentration of NO_x increases rapidly with flame temperature, and is greater than predicted by equilibrium thermodynamics. The rate of forward reaction is different from the backward reaction, and there is insufficient time for equilibrium to be reached.

The turbine inlet temperature has been calculated three ways. First the compressor bleed air power is calculated from the temperature rise and flow rate, this is 102 HP in test #2. From the measured bleed air and total inlet air flow the compressor power is calculated, this is 485 HP. Equating this to the turbine power, allows one to calculate the turbine temperature drop. This added to EGT of 824°F in test #2, provides the turbine inlet temperature 1221°F. The second and third methods are based on assuming 100% adiabatic combustion and neglecting emissions other than CO_2, H_2O, O_2 and N_2. The expected results will be slightly higher. They are 1280°F using a mean specific heat and 1241°F using individual specie specific heats.

Measurements were recorded on a parts per million basis. The emission data were recorded by computer at 1 second intervals during 10 minute periods for single-fuel steady state operation. If these test were conducted on an engine for a car, then the emissions would be reported in grams per mile. For a stationary engine it would be reported in grams per HP. As this turbine does not provide shaft HP, only compressed air it seems more appropriate to report emissions in units of standard cc per second. First reduce the turbine exhaust gas flow rate to a room temperature volume flow rate, using density 0.0765 FT^3/lbm. For test #2 the exhaust gas flow rate is 3.48+0.0456=3.5256 lbm/s= 46 STD FT^3/s = 46*28317 cc/s.=1.3*10^6 cc/s. Thus in test # 2 if the ppm values are multiplied by 1.3 then one gets the emissions in cc/s.

During the transient fuel type change-over maneuver, another automotive type emission test apparatus was employed. This one was capable of printing data in five seconds intervals. Such high speed data acquisition was essential, as the fuel-change-over lasts less than 0.5 minutes, depending on the power setting. In that time the fuel concentration ratio changes gradually from 0% to 100%. Data were collected continuously and printed out in 5 seconds intervals. Because the equipment used for this test was designed for simple automotive testing, the data presented here should only be used for relative comparisons. These data are plotted as a function of time in Figure 7.1. For this test, the turbine was operated on Jet A until steady state was reached at which the data acquisition was initiated at t = 0. Because of the steady nature of the data on Jet A, data were only plotted starting at t = 4 minutes. At t = 5 minutes, fuel type change-over was initiated from Jet A to Air Products methanol. Immediately following this change-over, Figure 7.1. shows a dramatic decrease in NO_x production during Jet A dilution process. At approximately t = 6 minutes, one minute after the initiation of the fuel change over, the NO_x data approach the pure methanol equilibrium value. At t = 11 minutes the reverse fuel type change-over, from methanol to Jet A, is initiated. Following this procedure, the NO_x production rapidly approaches the Jet A steady state value as represented by value at t = 4 minutes. It can be seen from Figure 7.1. that while fuel type has a strong effect on the NO_x production, it little effect on the other species sampled.

8) Conclusions and Recommendations

The principal objective of this contract has been accomplished. The GTC-85-72 gas turbine was successfully operated on both jet A fuel and on fuel grade methanol produced by Air Products and Chemicals Inc. Emission data were collected on each fuel during steady state (defined as unchanged during at least 6 minutes). In addition emission data were collected during the transient fuel-change over procedure which lasted about less than 0.5 minutes. Some alcohols like ethanol are entirely miscible with jet fuel, but methanol is only partially miscible. The miscibility reduces with the presence of water and at lower temperatures. To prevent separation, chemicals such as benzene and acetone can be added. Engine starting proved to be only possible on Jet A, due to the low volatility of methanol and the high heat of vaporization. To minimize corrosion and diaphragm deterioration during storage, and permit starting, it was decided to change over to methanol only after the engine was warmed up and return to jet A prior to engine shut-down. A sight-glass in the fuel supply manifold clearly demonstrated the lack of miscibility between Jet A and methanol. They do not freely mix, just like oil and vinegar. After a fuel emulsifier pump was installed, the transition from one type of fuel to the other becomes visible like a milky cloud, which only clears up after change-over is completed. To achieve successful fuel change-over it proved to be essential to raise the EGT to more than 750°F which is done by applying at least 25% bleed air load. Even at this elevated EGT value was it necessary to increase the ignition power. This was achieved by adding two glow plugs from a PT-6 aircraft gas turbine to the existing spark plug. The inability to operate on methanol at idle, is most likely due to the cooling effect from the high heat of vaporization. This delays ignition to further downstream in the burner. Because there the mixture is diluted by secondary air, the mixture there becomes too lean to ignite. This problem might be solved by extending the burner by four inches in length in between the primary and secondary air supply zones.

Unfortunately the fuel controller was unable to supply enough methanol to permit operation at more than 50% bleed air. This problem can probably be solved by installing a fuel controller of a larger model turbine and by opening up the burner high pressure nozzle hole size.

The lack of methanol lubricating properties destroyed the bearings and the cylindrical RPM control fuel valve inside the fuel controller. It is imperative that all future turbine tests on methanol must incorporate a suitable lubricant additive such as "Lubrizol".

These were the only significant operational problems encountered. The emission testing presented no difficulties. The ppm emission data are readily convertible to units of cc/s. The conversion coefficients are calculated in the program for each test and are in the order of 1.3.

The significant change in NO_x level from about 25 ppm on jet A down to about 5 ppm on methanol, is most likely caused by the before mentioned burning of the methanol spray at a location further downstream, where the mixture gets already cooled by secondary air flow.

This demonstration project has proven that Air Products methanol can be operated safely in gas turbines when the necessary modifications have been made.

APPENDIX 2

(not available electronically)

APPENDIX 3

Paper submitted to IMECE Nov. 5-10, 2000 conference in Orlando, Florida (6/9/00)

Lubricity Problems and Solutions for a Methanol Fueled Gas Turbine

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Abstract

The Liquid Phase Methanol (LPMEOH^{TM}) process, which was developed by Air Products and Chemicals Inc., can be used to convert coal-derived synthesis gas into a fuel or chemical grade methanol product. This technology is being demonstrated under sponsorship of the U. S. Department of Energy's Clean Coal Technology program at Eastman Chemical Company's chemicals-from-coal complex in Kingsport, TN. In 1998, fuel-grade methanol was used at WVU to operate a small unmodified (235 HP) gas turbine. During these tests, the fuel system gear pump and rpm controller failed due to the lack of lubricity of the methanol fuel. To remedy this problem, a pint (over an order of magnitude larger than the recommended amount) of a commercially available fuel additive was dissolved in half a barrel of methanol, and the fuel controller/pump was replaced. The next series of runs produced a similar failure. This prompted the WVU team to search for a suitable methanol additive, which can provide lubricity equal or better than that of jet fuel. To minimize the amount and thus cost of such an additive, it was essential to accurately measure lubricity of methanol/additive solutions, at various concentration levels. Conventional lubricity measuring apparatus are based on measuring wear. When used with methanol, the data were erratic due to a changing wear pattern. To get repeatable steady data, a new lubricity test apparatus was developed, based on comparing friction coefficients, at a typical bearing design load. After many modifications this apparatus provided satisfactory and consistent results. A few percent castor oil or even fewer percent racing fuel additives provided the needed lubricity to operate the WVU gas turbine safely on methanol.

Introduction

The wear of lubricated bearing surfaces depends not only on the lubricant, but also on the materials used, the bearing load, velocity and surface finish. Lack of sufficient lubricating properties results in wear, which alters the surface finish and produces loss of material from the surface. One can experience four types of wear: corrosion, adhesive wear, abrasive wear and surface fatigue. Wear can be reduced by the presence of lubricants and corrosion inhibitors at the point of contact of the wear bodies.

One distinguishes two types of fluid lubrication "Boundary Lubrication" and "Hydrodynamic Lubrication". Boundary Lubrication occurs when the lubricant surface tension maintains a boundary between the solid surfaces, thereby reducing the frictional forces between them. Hydrodynamic lubrication is when a lubricant is forced or pumped in between the two surfaces, to limit their interaction. Many tests have been developed to characterize lubricating fluids. The three most common test methods are: BOCLE (Ball-on-Cylinder Lubricity Evaluator), the HFRR (High Frequency Reciprocating Rig), and field-testing.

1) The BOCLE (American Society for Testing and Materials, 1999) test was designed for testing the lubricity of diesel and jet fuel. The test consists of placing a ½" diameter ball on cylinder rotating at 244 RPM, submerged in the test fluid at 25°C. Each test starts with a new ball loaded with a 9.81 Newton force and lasts 30 minutes. Upon completion of the test, the scar on the ball is measured to the nearest 0.01 mm.

2) A variation of this test is called the Lubrizol Scuffing BOCLE (Lubrizol Corporation, 2000). This test is similar to the before mention test but applies a steady load with a 7 kilogram mass. The test is run on the cylinder for 2 minutes. The average scar diameter is then measured and used to compare lubricating qualities.

3) The HFRR (Rabinowicz, 1995) test uses a ½" ball, which is rapidly vibrated back and forth over a flat surface. A load of 200 grams is placed on the ball and moved back and forth with a 1-mm stroke. The time necessary to wear a scar into the ball is measured; the size of the scar gives the lubrication qualities of the fuel being tested.

4) Field-tests (Rabinowicz, 1995) are the most reliable tests, because all of the operating conditions are duplicated exactly. However, this type of testing is usually very expensive and can be impractical.

The BOCLE has been used for some time, but there are few of these machines available at specialty fuel testing labs. HFRR has been accepted by ISO, SAE and is commonly in Europe for testing diesel fuel lubricity. The drawback is, there are very few of those testing machines available in North America. Field-testing is good but very expensive. The methanol fueled WVU model GTC-85-72 gas turbine, experienced two fuel controller/gear-pump failures, which costs approximately $20,000 each to replace. This emphasizes the importance of fuel additives to provide the required lubricity.

<u>Lubricity Tests at WVU</u>

One lubricity test apparatus was available at WVU. It was a variation of the Lubrizol Scuffing BOCLE method. Here a cup, containing the sample material is filled with the test fluid and rotated. A stationary ½" steel ball is lowered onto the sample at a distance from the center of rotation. This test is designed to quantify fluid lubricity by measuring changes in wear rate, either from mass loss or from scarring.

When used with methanol, it was found that once wear had begun, the data collected over different time intervals, keeps on changing, rendering it difficult or impossible to produce repeatable data. This erratic performance was due to a changing wear pattern. To get repeatable data, a new lubricity comparison test apparatus was developed. This one was based on comparing the friction coefficient, at typical bearing loads. The reason being that friction is ultimately responsible for wear.

The WVU lubricity comparison apparatus was designed to operate at near normal bearing pressures using a 60 N dead weight. This weight was placed on a rotating disc containing three balls, as shown in Fig. 1. The three balls transferred the load onto a fixed brass washer and were mounted at a distance of 31mm from the centerline of the disc holder. The three balls were ground to form flats of 3.81-mm diameter. This reduces the lubricated contact pressure to 1.65 M Pa, which is 3.5% of the maximum design load limit for a well-lubricated lead-bronze bearing. This load reduction proved to be necessary to prevent marring the surface when operating on methanol. To guarantee that the disc rotates smoothly about its axis, it was guided by a ball bearing installed on the centering pin in the middle of the fixed washer.

To achieve high accuracy in rpm control and rotating disc position, the apparatus was installed on a vertical mill with numerical position read-out. An exploded view of the complete testing apparatus is shown in Fig. 2. Shown here is the disc three-ball drive head, to be installed on a vertical mill. A disc drive shaft extends from the end of the mill head, passes through the dead weight, and is connected to the disc in a manner that allows only rotational forces to be transferred from the mill. The dead weight slides on the shaft, so that its weight is entirely supported by the balls in the driven disc. Torque is transferred from the drive shaft to the dead weight by a pin and from there to the driven disc by two pins, which protrude from the bottom of the weight. The dead weight normal force is transferred to the driven disc through a ½ inch steel ball on the system centerline. This system insured that the driven disc was loaded at the center, so that all three flattened balls transfer the same normal force. The next item shown in the exploded view, Fig. 2, is the fluid cup containing a fixed machined washer, submerged in the fluid to be tested. The cup system was placed on a bearing assembly, attached to the table of the mill, so that accurate torque measurements could be taken with an attached beam type load cell. The load cell data were used to calculate the friction coefficient between the washer and the driven disc.

Test Procedure

Prior to testing, great care was taken to prepare the contact surfaces for testing. The washer was machined to insure that its surface was perfectly flat and both contact surfaces, balls and washer, were hand finished by wet sanding using 1500 grit abrasive paper on a flat steel surface. No matter how fine both of these surfaces were ground, the system required additional rotational polishing before the surface finish was good enough to provide steady and repeatable friction coefficient data. This was accomplished by running the system at 200 rpm using Jet A fuel as a lubricant. During this procedure, the friction coefficient data was monitored until a steady-state value was reached. A data set obtained during the first 30 minutes of the 45-minute "break-in" period can be seen in Fig. 3.

Following the break-in procedure, testing was accomplished by filling the test cup with the fluid to be tested, such that the contact surfaces between the load balls and brass disc are fully submerged. The system was operated at 200 rpm and friction torque data were collected at approximately 2 Hz for a period of 10 minutes. When a lubricant, such as castor oil, was tested at various concentrations, tests were run starting with pure methanol followed by ever increasing oil concentrations. This prevented the possibility of oil deposits from higher oil/methanol concentrations, to introduce errors at the lower concentrations.

Time dependent data acquired during one of the Jet A and M100 tests are shown in Fig. 4. Because of the starting transients experienced during many of these tests, the first two minutes of data were discarded prior to data averaging in order to arrive at a representative friction coefficient.

Test Results

Very few lubricity additives were both: effective in reducing friction and are readily dissolved in methanol. Only three of all the additives tested had the required properties and produced lubricity in excess of that of jet-A fuel. They were readily soluble in methanol in quantities far in excess of that needed and remained in uniform suspension during storage. One satisfactory additive was pure castor oil and the other two were Morgan Fuels *Two Cycle Blue* and Manhattan Oil Company's *Power Plus Cherry Bomb* racing fuel additives. Both of these are synthetic commercial methanol fuel additives for use in racing applications.

Friction coefficient data obtained for methanol containing varying concentrations of castor oil can be seen in Fig. 5. From this plot, it can be seen that, at low concentrations, the addition of oil has a large effect on friction coefficient. However, once a level of approximately 5% has been reached, there is little gained by increasing the oil concentration. Also shown in Fig. 5 are two horizontal lines indicating the friction coefficients when using both pure methanol and Jet A. Using the Jet A line, it can be seen that a castor oil/methanol concentration of approximately 3% is required to achieve the same friction coefficient as Jet A.

Using the same method, the oil mixture ratio for the commercial additives was found. The manufacture recommended ratio for the *Two Cycle Blue* additive is 0.04% for use in racing applications. However, to achieve the same friction factor as Jet A, a 1% concentration was required.

Table 1 contains the experimental friction coefficients obtained experimentally for both Methanol and Jet A as compared to various handbook data.

Table 1: Friction Coefficient Data

System	Friction Coefficient
Metal on Metal, Dry*	0.15 – 0.20

Metal on Metal, Wet*	0.3
Occasionally Greased*	0.07 – 0.08
Continuously Greased*	0.05
Mild Steel on Brass**	0.44
Methanol (WVU)	0.309
Jet A (WVU)	0.167

* - Oberg et al. (1962)
** - Avallone and Baumeister III (1987)

Conclusions

The new lubricity test apparatus designed and tested at WVU was relatively easy to use and provided the needed steady state data. Each run was conducted over a 10-minute period. Conducting the tests at 3.5% of a lubricated bearing design load proved to be the most successful. It was also found that this system yielded an experimental repeatability far greater than that possible with the wear based lubricity-testing methods.

Following these lubricity tests, the WVU gas turbine was operated on methanol using one the *Two Cycle Blue* additive available additives for an extended period, with out failure.

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APPENDIX 4

(not available electronically)

APPENDIX 5

```
'File: LUBSTAT.BAS : 6/7/00
'-------------------------------------------------------'
'        Program to Reduce Raw Experimental Data        '
'         by Discarding Data Points Outside of          '
'         Plus or Minus 3 Standard Deviations           '
'           and removing staring effect data            '
'-------------------------------------------------------'
'                     Written by                        '
'                   Robert E Bond                       '
'-------------------------------------------------------'
Cls

Dim t(15000), Cf(15000)
Tdel = 120
Nforce = 56.506
Rad = 0.03185

'-------------------------------------------------------'
'                 Read Raw Data File                    '
'-------------------------------------------------------'
'-----Construct File Name-----'
Print "  On which drive is the data located: <A, C, E>"
Do
  Drive$ = INKEY$
Loop Until Drive$ = "A" Or Drive$ = "a" Or Drive$ = "E" Or Drive$ = "e" Or Drive$ = "C" Or Drive$ = "c"

10 INPUT "  Please input the test number"; file$
file$ = "test" + file$
FILEIN$ = Drive$ + ":" + file$ + ".dat"
FILEOUT$ = Drive$ + ":" + file$ + ".STA"

'-----Open and Read Data File-----'
Cls
Open FILEIN$ For Input As #1
  Print "Reading "; FILEIN$
  Input #1, junk$
  Input #1, junk$
  Input #1, junk$
  Input #1, junk$
  I = 0
  Do Until EOF(1)
    I = I + 1
    Input #1, t(I), junk, Torque
    Cf(I) = Torque / (Nforce * Rad)
```

```
    If t(I) < Tdel Then SAMPmin = I
  Loop
  SAMP = I
Close #1

'-------------------------------------------------------'
'          Average Data Sets and Determine SD           '
'-------------------------------------------------------'
Print "Averaging Data"
Sum = 0
SQSUM = 0
N = SAMP - SAMPmin + 1
For I = SAMPmin To SAMP
    Sum = Sum + Cf(I)
    SQSUM = SQSUM + Cf(I) ^ 2
Next I
FLAG = 0
DELTA = N * SQSUM - Sum ^ 2
If DELTA < 0 Then DELTA = 0: FLAG = 1
Ave = Sum / N
SD = Sqr(DELTA / (N * (N - 1)))

'-------------------------------------------------------'
'            Recalculate Average using data             '
'              within +- 3 std Deviations               '
'-------------------------------------------------------'
Sum = 0
N = 0
If FLAG = 1 Then
  Ave2 = Ave
  N = SAMP - SAMPmin + 1
Else
  For I = SAMPmin To SAMP
    If Abs(Cf(I)) < (Abs(Ave) + 3 * Abs(SD)) Then
      If Abs(Cf(I)) > (Abs(Ave) - 3 * Abs(SD)) Then
        Sum = Sum + Cf(I)
        N = N + 1
      End If
    End If
  Next I
    Ave2 = Sum / N
End If


'-------------------------------------------------------'
'                   Output Results                      '
'-------------------------------------------------------'
Cls

'-----Print Results-----'
Print "For the file "; FILEIN$
Print
Print USING; "After eliminating the first ### sec (#.# min) of data;"; Tdel; Tdel / 60
Print
Print USING; "  Raw Data Average (Cf) =#.####"; Ave
Print USING; "Stat. Data Average (Cf) =#.####"; Ave2
```

```
Print USING; "Standard Deviation (Cf) =#.####"; SD
Print USING; "         Data Rejection =###.#%"; 100 - 100 * N / (SAMP - SAMPmin + 1)

'-----Write Output File-----'
Print
Print "Would you like to write a data file? <Y/N>"
Do
  ANS$ = INKEY$
Loop Until ANS$ = "Y" Or ANS$ = "y" Or ANS$ = "N" Or ANS$ = "n"
If ANS$ = "Y" Or ANS$ = "y" Then
  Open FILEOUT$ For Output As #2
  Print #2, "T(min)     Cf      Cfave     SD     DR(%)    T       SDl      SDh      T     Cf"
  Print #2, USING; "##.###   #.####   #.####   #.####   ###.#    0     #.####   #.####   ##.##   0"; t(1) / 60; Cf(1);
Ave2; SD; 100 - 100 * N / (SAMP - SAMPmin); Ave2 - 3 * SD; Ave2 + 3 * SD; Tdel / 60
  Print #2, USING; "##.###   #.####                            ##.#   #.####   #.####   ##.##   1"; t(2) / 60; Cf(2); t(SAMP) /
60; Ave2 - 3 * SD; Ave2 + 3 * SD; Tdel / 60
  For I = 3 To SAMP
    Print #2, USING; "##.###   #.####"; t(I) / 60; Cf(I)
  Next I
  Close #2
End If

'-----Run Program Again?-----'
Print
Print "Would you like to run this program again? <Y/N>"
Do
  ANS$ = INKEY$
Loop Until ANS$ = "Y" Or ANS$ = "y" Or ANS$ = "N" Or ANS$ = "n"
If ANS$ = "Y" Or ANS$ = "y" Then GoTo 10

End
```

APPENDIX 6

(not available electronically)

APPENDIX 7

(not available electronically)