Fermi National Accelerator Laboratory

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TEST OF AN IMPROVED OIL INJECTED HELIUM SCREW COMPRESSOR AT FERMILAB

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ABSTRACT

Fermilab has tested a modified helium oil injected two-stage Mycom screw compressor for possible use in the Tevatron. The tests are part of a joint venture with Mycom. Modifications to the compressor include a new modified rotor profile and new generation lubricant which resulted in increased performance and efficiency. The effects of the modifications on shaft-power and isothermal efficiency are included. The results of these tests will determine the practicality of incorporating these modifications to the thirty-four existing screw compressors of the Tevatron.

INTRODUCTION

The Tevatron refrigeration system uses thirty-four Mycom screw compressors installed in compressor buildings at nine locations around the Fermilab ring. Each of the thirty-four compressors is an oil-injected, two-stage, Mycom screw compressor driven by a 400 hp motor. Since the power consumed by these compressors is a major operating expense for the Tevatron, we were interested in reducing the power consumption and increasing the efficiency of our compressors. In a joint venture with Mycom, Fermilab tested a compressor with modified features which allow for such performance increases. One of the modifications consists of a new rotor profile (O-profile) which is designed for improved performance and efficiency over the standard rotor profile which we currently use. Theoretical and empirical tests done by Mycom[1,2] have shown increases in isothermal efficiency of about 10% over machines with standard rotor design. Another modification to the compressor is the use of a new generation lubricant. Oil plays an integral part in the performance of a screw compressor. It is required to seal, cool, lubricate and actuate the capacity control valves. The main difference in the new oil is its lower viscosity and lower initial water content as compared to the Union Carbide LB-170X oil which we currently use in our compressor system. Each compressor has a reservoir of 90 gallons of processed oil. Fermilab processes this oil to remove the water content to below 2 ppm. The new oil (MH-41) is a synthetic polyalphaolefin base manufactured by Mobil USA. Table 1 summarizes some of the properties of the Mobile MH-41 and Union Carbide LB-170X oils. Use of lower viscosity oils should, in theory, reduce the shaft-power of the compressor due to the oil compression phenomenon within the compressor. This theory was verified by Mycom.[1,2]

* Operated by Universities Research Association, Inc. under contract with the U.S. Department of Energy

Table 1. Oil data, information provided by Mycom Corporation

	Test Method	MYCOM/Mobil MH-41	Union Carbide LB-170X
Viscosity @ 40 °C, cSt	ASTM D-445	16.9	36.5
@ 100 °C, cSt		3.9	6.8
Viscosity Index	ASTM D-2270	117	159
Pour Point, °C	ASTM D-92	-54	-42.8
Flash Point, °C	ASTM D-92	204	187
Specific Gravity	ASTM D-1298	0.821	0.988

TEST SETUP

The typical compressor package consists of a two stage oil flooded Mycom compound screw compressor, type 2016-C, with a 400 hp motor and associated oil removal system. The compressor consists of two mating helically grooved rotors set in bearings at each end of the compressor casing. The drive rotors are of male configuration and have 4 lobes each. The driven rotors are of female configuration and have 6 lobes each. Capacity control is accomplished by a slide valve which moves parallel to the rotor axis and changes the area of the opening in the bottom of the rotor casing. This, in effect, lengthens or shortens the compression zone of the rotor and further acts to return gas to the suction side. Actuation of the slide valves is by hydraulic pressure controlled through solenoid valves. A typical machine cross section is shown in Figure 1.

Figure 2 illustrates the setup configuration for the compressor tests. Helium mass flow was calculated from the measured pressure drop through a 0.781 inch diameter orifice located in the 2-inch compressor discharge pipe. The pressure drop was measured in inches of water using an ITT Barton Model 752 Electronic Differential Pressure Indicator. Instrumentation consisted of pressure taps using Setra Model 205-2 transducers and temperature sensors using National Semiconductor LM-135 integrated circuit temperature sensors. The power delivered to the motor was measured using a BMI 3060 Power Profiler power meter. All relevant parameters were datalogged for later retrieval and analysis.

The ideal isothermal work used in all the calculations was calculated using the formula

$$W = m\, R\, T_1 \ln (P_2 / P_1)$$

where P_2 is the measured discharge helium pressure, P_1 is the measured inlet helium pressure, T_1 is the measured helium inlet temperature, R is the ideal gas constant for helium (2.077 J/g K), and m is the calculated mass flow rate.

The isothermal efficiency is defined as the ideal isothermal work divided by the compressor shaft input power, which is the motor shaft horsepower. Thus, the isothermal efficiency is the efficiency of the compressor alone. The power consumed by the oil pump is not included in the isothermal efficiency. The efficiency of the motor was assumed to be 95.6% which is the NEMA rating for this motor. This value was used to arrive at the shaft power from the electric input power.

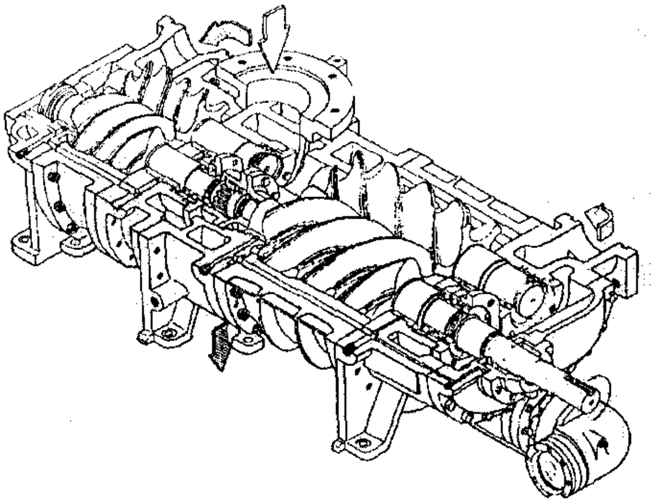
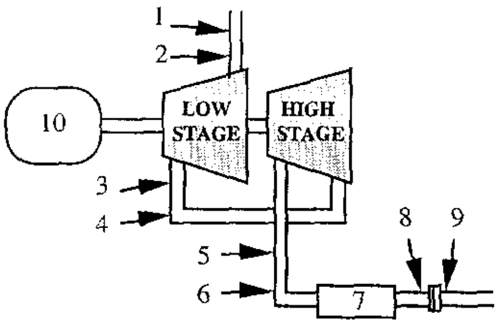


Figure 1. Compressor cross section, figure provided by Mycom Corporation



1 Inlet Pressure
2 Inlet Temperature
3 Interstage Pressure
4 Interstage Temperature
5 Exit Pressure
6 Exit Temperture
7 Aftercooler
8 Orifice Temperature
9 Orifice Delta P
10 400 HP Motor

Figure 2. Compressor instrumentation setup

TEST RESULTS

The compressor tests were performed in three phases:

Phase 1: Conventional compressor with conventional rotor profile and existing Union Carbide LB-170X oil. This test provides a baseline for comparison with the new modified features and also allows for comparison with earlier tests done by Peterson.[3]

Phase 2: Conventional compressor with conventional rotor profile and new Mobil MH-41 oil. This test, in comparison with the Phase 1 results, will determine the advantages of using lower viscosity lubricants and their effect on compressor performance.

Phase 3: New compressor with modified rotor profile and new Mobil MH-41 oil. This test provides a comparison between the conventional rotor profile and the modified 'O' profile rotors. Any performance increases between Phase 2 and Phase 3 cannot be solely attributed to the change in rotor profile because in Phase 3 we tested a completely different compressor with a different compressor casing. Each of these compressors have been modified to various degrees in the discharge porting. These modifications were done to reduce the hydraulic hammering effect by allowing oil relief through the discharge port to the discharge manifold as explained by Dreksler[1]. The reduction of the hydraulic hammering effect results in shaft-power reduction. These hydraulic effects occur due to trapping of oil between the rotors and the end wall. The forces created by the trapping of the oil are transmitted to the thrust bearings resulting in unpredictable failures.

Table 2 summarizes the results of the three phases at fully loaded conditions. Results of the Phase 1 test show an isothermal efficiency approximately 3% lower than measured by Peterson[3] for identically setup compressors. These differences can perhaps be attributed to differences in the instruments used to make the measurements, age of the compressors, differences in motor efficiency, etc..

Table 2. Compressor test results

	Helium Inlet Temp. (deg. F)	Electric Power (KW)	Motor Efficiency (%)	Motor Shaft Power (KW)	Lo Stage Slider Position (%)	Helium Flow (g/s)	Helium Inlet Press. (psia)	Helium Exit Press. (psia)	Isothermal Efficiency (%)
Phase 1	78.6	282.2	95.6	269.8	100	54.1	15.7	294.6	36.53
Phase 2	95.3	267.1	95.6	255.3	100	52.2	15.1	295.4	38.92
Phase 3	87.1	224.5	95.6	214.6	100	56.1	15.7	304.7	48.98

The Phase 2 results show a reduction in total shaft power of 14 KW as compared to the Phase 1 conventional machine. Table 2 indicates that for the Phase 2 tests oil flow rate was elevated to 75 gal/min compared to the 40 and 50 gal/min oil flow rates for Phase 1 and Phase 3, respectively. The lack of a constant flow rate for all the phases can be attributed to the absence of a throttling valve in the oil injection line. A throttling valve is the only means available to vary the oil flow rate since the oil pump is of fixed speed. This valve was added for the Phase 3 portion of the tests. The effects of elevated oil injection flow will cause a slight increase in shaft-power with a corresponding decrease in isothermal efficiency due to the oil compression phenomenon within the compressor. Mycom estimates that the increase in power is on the order of 5 to 6 KW for this change in oil flows.

Aside from lubricating the compressor and acting as a sealant, the oil acts as a coolant by removing the heat of the compression process. Thus, one of our main concerns in using the lower viscosity oil was its ability to keep the compressor at reasonable temperatures. The results of the tests as shown in Table 3 show no appreciable change in the helium discharge and oil manifold temperatures although more long-term tests need to be done to understand the full scope of these effects.

The Phase 3 results show a dramatic 55 KW shaft-power reduction compared to the Phase 1 tests. These results closely parallel the results obtained by Dreksler[1]. Figures 3 and 4 show the change in isothermal efficiency and shaft power, respectively, for all three phases at different flow rates.

Table 3. Compressor Temperatures and Oil Flow Conditions

	Low Stage Slider Position (%)	Helium Inlet Temp. (deg. F)	Helium Interstage Temp. (deg. F)	Helium Exit Temp. (deg. F)	Oil Manifold Temp. (deg. F)	Oil Supply Flow (gal/min)
Phase 1	100	78.6	148.2	169.0	106.7	40
Phase 2	100	95.3	142.4	156.7	108.6	75
Phase 3	100	87.1	145.2	167.0	106.7	50

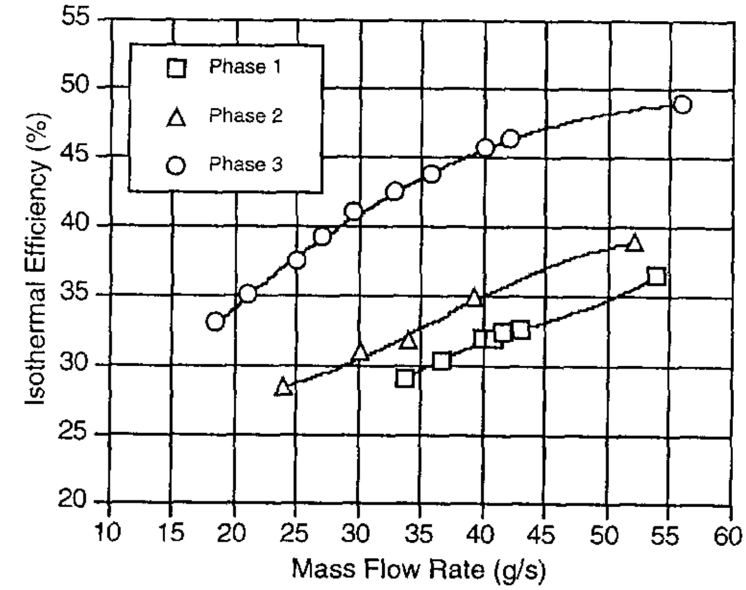


Figure 3. Comparison of isothermal efficiency

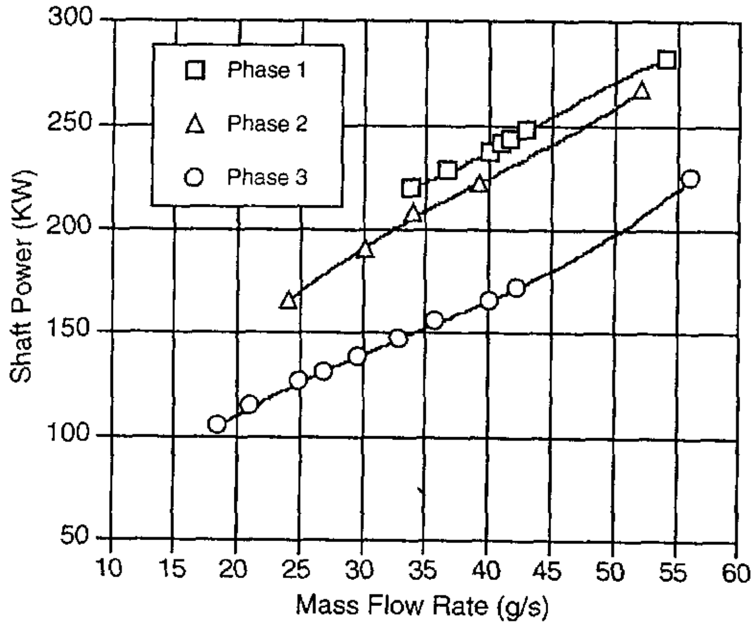


Figure 4. Comparison of total shaft power

CONCLUSIONS

The results of these tests indicate that a considerable increase in performance can be attained by using modified rotor profiles and new lower viscosity oils. The use of lower viscosity oil alone provides a reduction of approximately 14 KW in shaft power with a corresponding increase of over 2 points in isothermal efficiency representing a greater than 5% improvement in performance without considerably affecting compressor cooling. Further tests have to be done to investigate the impact of lower viscosity oil on our oil removal system[4]. The largest gain in performance is with the use of modified rotors. Isothermal efficiencies approaching 50% were obtained for a completely modified compressor (modified rotors and lower viscosity oil). This is an increase of more than 34% in isothermal efficiency compared to a conventional compressor. Our long-term goal is to install modified compressors (modified rotor profile and new generation lubricant) at one compressor house (4 compressors) in order to reaffirm our results and study the long-term effects on reliability and performance of these machines. Such tests may eventually pave the way to a complete retrofit of our entire compressor system using these modified features.

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