1983 CEC Conference

fermilab-conf-83-148-TD

FERMILAB'S SATELLITE REFRIGERATOR EXPANSION ENGINES

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INTRODUCTION

Each of Fermilab's 24 satellite refrigerators includes two reciprocating expanders, a "wet" engine and a "dry" engine. The wet engines and all but eleven of the dry engines were manufactured by Koch Process Systems (Westboro, Massachusetts). These are basically Koch Model 1400 expanders installed in cryostats designed by Fermilab. The other eleven dry engines are an in-house design referred to as "Gardner-Fermi" engines since they evolved from the GX3-2500 engines purchased from Gardner Cryogenics. Table I summarizes the features of our three types of expanders.

Every engine is instrumented at Fermilab with inlet and exhaust pressure gauges, a cylinder pressure transducer, inlet and exhaust vapor pressure thermometers with gauges, and an hour meter. Engine speed is controlled by a General Electric dc drive which has a tachometer with digital speed readout and a "power out" meter which indicates the electrical power generated or used by the system. Fermilab also adds a mechanically actuated safety overspeed brake and electrical interlocks to protect personnel and equipment.

THERMODYNAMIC PERFORMANCE

Efficiency

Figure 1 illustrates the behavior of cylinder pressure as

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Table I. Some Features of Fermilab's Expansion Engines

GINE	KOCH WET	KOCH DRY	GARDNER-FERMI DRY
ston	2 phenolic pistons 2 inch diameter 2 inch stroke	2 phenolic pistons 3 inch diameter 2 inch stroke	1 stainless piston 3.19 inch diameter 3 inch stroke
als	warm O-rings, grease lubricated	warm O-rings, grease lubricated	cold piston rings, graphite-Teflon
lves	pullrod operated, Kel-F seal	pullrod operated, Kel-F seal	pushrod operated, cold rocker arms, all stainless steel
take cutoff	0.70	0.42	0.38
eed range (RPM)	50-230	50-230	75-350
fective dead volume c/piston)	29	41	24
splacement volume c/piston)	103	232	392
ical inlet essure (ata)	18	18	18
ical inlet nperature (K)	5.0	30	30
ical exhaust essure (ata)	2.0	1.4	1.4
w rate at typical nditions and maximum eed (g/sec)	60	20	22

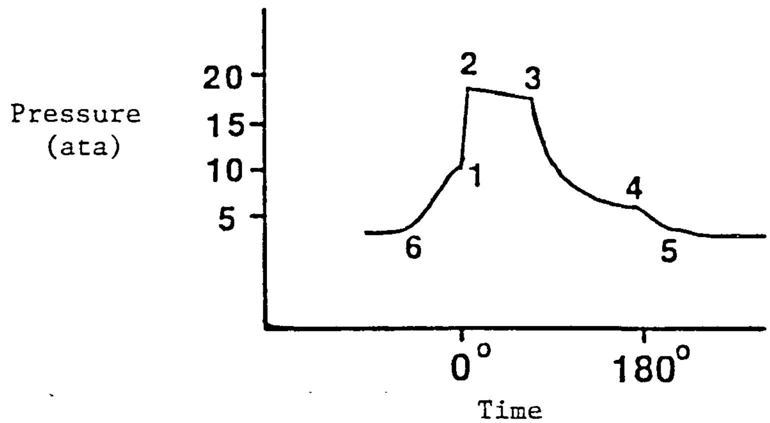

Figure 1. Koch dry engine cylinder pressure during one cycle. Supply is 18 ata, 24K, engine speed is 100 rpm.

measured by a pressure transducer over one cycle of a typical Koch dry engine. Our expansion process may be considered to consist of the following events occuring between the labeled points on the pressure trace for the dry engine.

Point 1 to point 2 is the filling of the dead space with high pressure gas; a "tank filling" process[1] and mixing with the gas in the dead space at point 1. Point 2 to point 3 is the filling of the cylinder as the piston moves up with the inlet valve open. Point 3 to point 4 is the isentropic expansion of a constant mass. Point 4 to point 5 is the release of the expanded gas to the exhaust pressure a "tank-discharge" process.[2] W.A. Morain and J.W. Holmes[3] modeled blowdown as what Reference 1 calls a "tank-discharge" process. Point 5 to point 6 is the emptying of the cylinder as the piston moves down with the exhaust valve open. Point 6 to point 1 is isentropic compression of a constant mass into the dead space.

This model of our expansion process implicitly contains assumptions that the valves and seals do not leak and that heat conduction through the head and down the piston, cylinder, and valves is negligible and heat convection to the gas from the tubes, valves, cylinder walls, and head is negligible.

If we measure supply temperature and the pressure for this process and know the cylinder volume at each point, the equations for tank-filling and discharge and isentropic expansion and compression can be used with some equation of state or tables of helium properties to determine the state at any point in this process. With the additional assumption of the ideal gas equation of state, I have derived the following equation for efficiency.

$$\eta = \frac{1 - \dfrac{\frac{1}{k}\left(\frac{V_4}{V_3}\right)^{1-k} + \left(\frac{P_5}{P_3}\right)\left(\frac{V_4}{V_3}\right)\left(1-\frac{1}{K}\right) - \left(\frac{P_6}{P_3}\right)\left(\frac{V_6}{V_3}\right)}{1 + \left(\frac{P_2}{P_3}\right)\left(\frac{V_2}{V_3}\right)\left(\frac{1}{k}-1\right) - \frac{1}{k}\left(\frac{P_6}{P_3}\right)\left(\frac{V_6}{V_3}\right)\left[\left(\frac{V_6}{V_1}\right)^{k-1}\right]}}{1 - \left(\frac{P_e}{P_s}\right)^{1-\frac{1}{k}}} \tag{1}$$

Efficiency η is defined in this paper as the change in helium enthalpy from the supply line to the exhaust line divided by the change in enthalpy which would result from isentropic expansion from the supply line pressure to the exhaust line pressure. The ratio of specific heats is k; P_e is exhaust line pressure; P_s is supply line pressure; P_i and V_i are pressure and volume respectively, at point i in the process in Fig. 1. Thus, V_1 is the dead volume; $(V_3 - V_1)/(V_4 - V_1)$ is the intake cutoff.

With this equation one can predict the effect of varying the intake cutoff, varying the amount of dead space, or varying the amount of recompression. If a pressure drop is seen during intake or discharge and the pressures at each point in Fig. 1 are known, the effect of these pressure drops on efficiency can be accommodated by this equation.

For both our types of dry engines there is significant dead space (see Table I) and significant but incomplete recompression. (Recompression is defined as complete when $P_1 = P_2$.) Using the volumes for our dry engines and assuming no pressure drops across the valves, the above equation yields a plot of theoretical efficiency versus intake cutoff (Fig. 2). We sacrifice some efficiency to gain mass throughput by operating with a large cutoff. Since Koch dry engines have occasionally operated near the theoretical efficiency in Fig. 2, it appears that the effect on efficiency of heat transfer to the gas is at most only a few percent.

Most instances of low efficiency in Koch dry and wet engines have been confirmed to be due to valve leaks. I suspect that the difference between the measured efficiencies (see Table II) and the prediction in Fig. 2 is primarily due to valve leaks. Efficiency seems very sensitive to particles such as metal dust or dirt as well as contaminants such as nitrogen which may cause valve leaks.

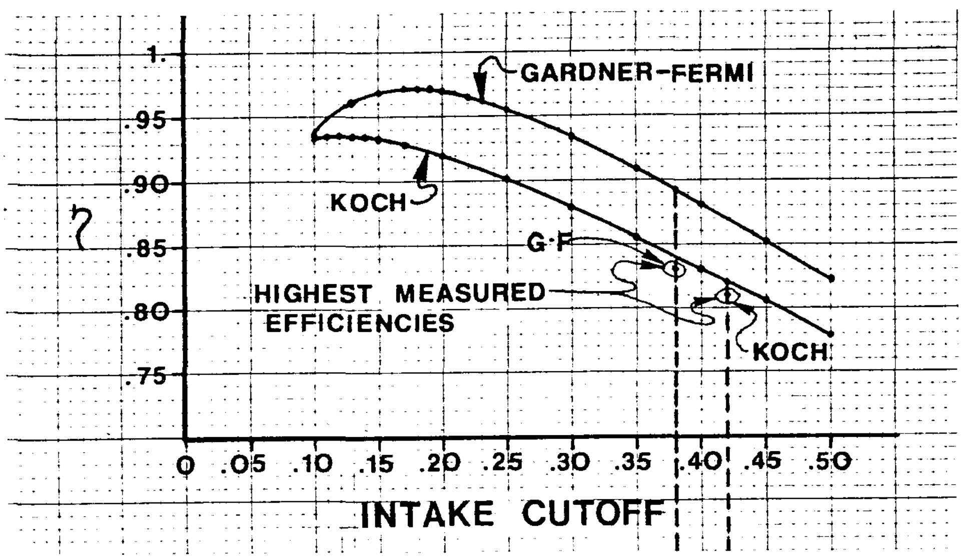

Figure 2. Theoretical dry engine efficiency versus intake cutoff

Gardner-Fermi engines have not operated as close to the curve in Fig. 2. Piston ring blowby has been measured at a rate which should reduce the efficiency by 2%. Perhaps frictional heating at the piston rings is also significant. The result is that both the best and typical Gardner-Fermi efficiencies are not significantly different from the Koch dry engine values.

Wet engines only operate in the ideal gas range during the early stages of cooldown, so the above equation for efficiency generally does not apply to them. Pressure traces show that recompression is nearly complete for inlet temperatures of 5.5 K or less, and expansion is theoretically complete ($P_4 = P_5$) at 5.5 K. Since the inlet is typically 5 K, we normally overexpand a little ($P_4 < P_5$) and recompress almost completely. Therefore, as one would expect, wet engine efficiencies are higher than dry engine efficiencies (see Table II).

No trends of efficiency with time have been observed in our engines and except for cases with relatively low efficiency where valve leaks were confirmed, no significant dependence of efficiency on speed has been observed.

Mass Flow Rate and Power

The refrigeration provided by an expander depends not only on its efficiency but also on the mass flow through the engine. Cylinder pressure traces such as Fig. 1 with corresponding data were analyzed for all three kinds of engines in order to estimate gas density in the cylinder and the effect of inlet pressure drop on mass flow. The smaller Koch engine valves restrict the flow somewhat more at higher speeds than the Gardner-Fermi valves.

Since mass flow times enthalpy change is the rate of energy removal from the gas, the mass flow can be used with the corresponding temperatures and pressures to predict engine power. For dry engines the predicted and measured horsepowers agree with an average error of 3%. Horsepower versus speed for two inlet temperatures is shown in Fig. 3 for a Koch dry engine. The shape of the horsepower curve is the shape of the mass flow curve in this model since efficiencies do not vary significantly with speed.

Our maximum dc drive speeds of 1200 to 1400 rpm were established because of the leveling off of performance indicated by the horsepower curves and the fact that refrigerator performance confirms this region of dimishing returns.

Wet engine efficiency and power predictions are complicated by not having use of the ideal gas relationships. Neverthless, wet engine mass flow below 6 K is almost linear with speed since inlet helium density is not a strong function of pressure.

MECHANICAL PERFORMANCE

Table II summarizes engine operations for the 24 satellite refrigerators since December 1982. Both types of dry engines had

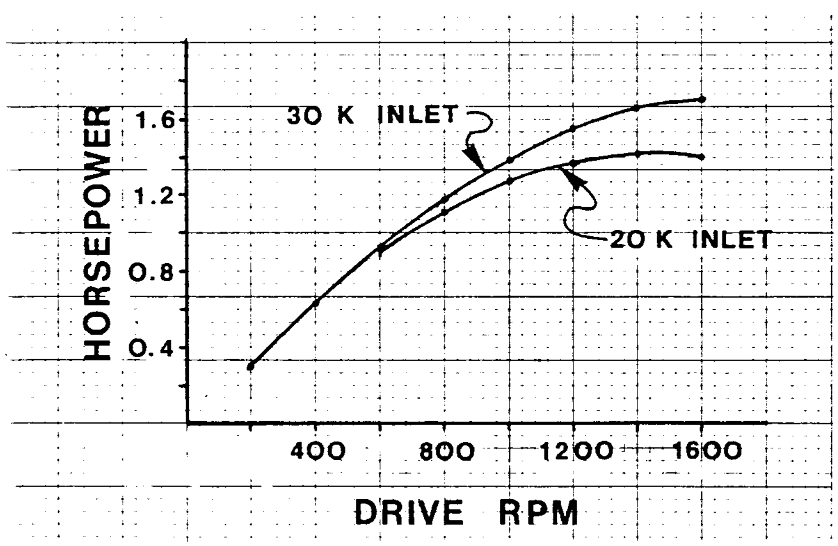

Figure 3. Koch dry engine power versus speed. Koch engine rpm is drive rpm divided by 6.

Table II. Performance Summary (through July 20, 1983)

ENGINE	KOCH WET	KOCH DRY	GARDNER-FERMI DRY
Number of engines	24	15	9
Total engine hours	72700	28600	22200
Most hours on an engine	7170	3082	4534
Mean time between major maintenance (engine warmup) (hours)	3040	2020	1450
Most frequent cause of major maintenance	piston O-ring blowby	piston O-ring blowby	piston shaft scored and pushrod seal leaks
Mean time between minor repairs (engine cold) (hours)	3256	2520	------
% time engine down for maintenance	0.3%	------	------
Average efficiency	.75	.70	.70
Greatest measured efficiency	.88	.81	.83
Confirmed causes of lower than normal efficiency	1. Valve leaks due to damaged or dirty valves 2. Valve leaks due to poor lifter clearance adjustment	1. Valve leaks due to eccentrically rotating exhaust cam taking up clearance 2. Valve leaks due to damaged or dirty valves 3. Poor lifter clearance	1. High-pressure tie-down seal leak

mechanical problems before September 1982, which limited the mean time between major repairs to less than 1000 hours.

The Koch dry engine drive-shaft failed by fatigue near its point of attachment to the crank, but a redesign of this area by Koch Process Systems appears to have solved that problem. Piston O-ring lifetime is extremely variable but appears to be between 2000 and 4000 hours. Different styles and materials of rings are being tried. Wearing and scoring of the crosshead guide, which can ultimately lead to piston O-ring failure has been somewhat controlled by weekly greasing and attention to cleanliness. But this and wearing of the piston O-ring groove are still problems in engines with over 3000 hours. In response to these problems, some engines with bronze sleeved crosshead guides and hardcoated O-ring groves in the aluminum crosshead are in operation.

Wet engine problems are like those described above for the Koch dry engine except that only one wet engine drive-shaft has failed on an engine with 3300 hours. Whether this indicates a problem in the long run is not yet known.

Gardner-Fermi dry engine lifetimes were limited by piston shaft seal failures. A complete redesign of the shaft seal area during 1982 has eliminated that problem, but some pushrod seal problems and scoring of the piston shaft by the linear bearing early in 1983 caused frequent major maintenance. Those problems are now solved and lifetimes of well over 3000 hours are anticipated.

SUMMARY

Efficiencies have generally been good and a few Koch dry engines have operated near the theoretical value after the effects of dead space, incomplete recompression, and incomplete expansion are accounted for by the model presented in this paper. Where efficiencies have been below average, valve leaks have usually been the cause. Engine efficiencies are extremely sensitive to contaminants in the helium.

Mechanical problems have been reduced to the point that, with a well-organized engine monitoring and maintenance program and one eight-hour shift each week for inspections, greasing, and whatever maintenance is required, engine reliability has not been a problem for accelerator operations. Modifications already being tested and other improvements such as automatic greasing devices, should result in engines which routinely last over 4000 hours between major overhauls and only require minor maintenance (greasing and inspection) every other week rather than every week as is now the case.

REFERENCES

1. D. A. Mooney, "Mechanical Engineering Thermodynamics," Prentice-Hall, Inc., New York (1953), p. 83.
2. Ibid.
3. W. A. Morain and J. W. Holmes, in "Advances in Cryo. Engr., Vol. 8," Plenum Press, New York (1963), p. 228.