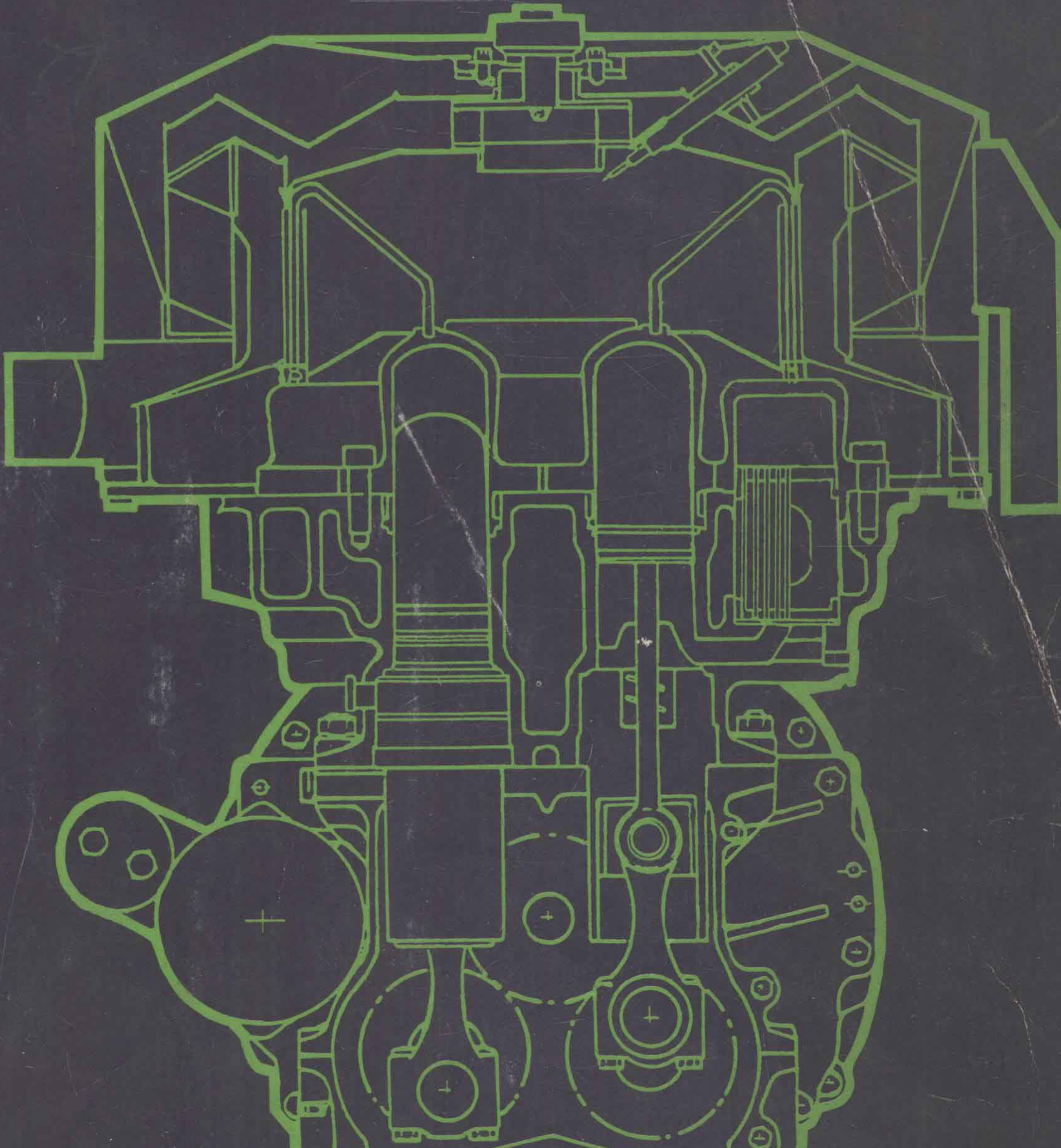
Stirling Engines

~ progress towards reality



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STIRLING ENGINES—PROGRESS TOWARDS REALITY

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CONTENTS

Future coal-burning Stirling engines

G WALKER, BSc, PhD, R FAUVEL, BSc, PhD, S SRINIVASAN, R GUSTAFSON
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SYNOPSIS The high price and limited availability of oil fuel forces a return to coal as the primary fuel. This provides unprecedented opportunities for applications of Stirling engines to stationary power and automotive application over a broad spectrum. For heavy automotive application in railroad locomotives and the larger off-highway mining and earth moving vehicles, it appears possible for Stirling engines of corresponding power and efficiency to replace diesels in the same installation envelope. Proposals are presented for the conversion of diesel engines to Ringbom-Stirling engines using air as the working fluid, water as the lubricant and with high pressure steam injection for power boost at peak load conditions. An experimental proof-of-principle Ringbom-Stirling diesel conversion presently under development at the University of Calgary is briefly described.

INTRODUCTION

The high price and limited availability of oil fuel is forcing a return to coal as the primary fuel. As this occurs the opportunities for application of Stirling engines are unlimited. Just as the last century belonged to the steam engine, and the present one to the internal combustion engine, so the next will belong to the Stirling engine.

Much of the coal will be used, along with nuclear energy, in baseload power stations to produce electric power. Cars, trucks and buses will use that power for propulsion with overnight charging by resistance heating of a lithium fluoride thermal battery to energize the Stirling automotive engine. Following its introduction by Meijer (1) subsequent studies (most recently Morgan (2)) have confirmed the feasibility and attraction of this concept as an alternative to electric storage batteries.

Other Stirling engines will use coal directly for stationary power generation, for heavy automotive applications in locomotives, surface mining equipment, off-highway vehicles, large, long-distance highway trucks and for marine use. There are no limits to maximum and minimum power capacities achievable with coal-fired Stirling engines. Given the will, virtually every oil-burning diesel engine is fair game for replacement by a coal-burning Stirling engine.

Limitations of space preclude the comprehensive discussion given elsewhere (Walker (3)) and we shall confine our subsequent discussion to large coal-burning Stirling engines (arbitrarily defined as greater than 1 MW output) applied to heavy automotive use for railroad locomotives and the larger off-highway vehicles.

LARGE STIRLING ENGINES

At the start it should be made clear there is no established experience of large Stirling engines. Most work to date has focussed on smaller engines up to 150 kW capacity. Walker (4) has briefly described the four-cylinder engine of 265 kW (360 horsepower) made by Philips for General Motors and the U.S. Navy in the early 1960's. Later the Electromotive Division of General Motors made a 300 kW (400 horsepower) 4-cylinder in-line, engine representing one bank of a Vee-8 engine of 800 horsepower, and did the design work for other larger Stirling engines. Hearsay has it that the German licensees MAN/MWM of the Philips Company have developed Stirling engines in the 600 kW (800 horsepower range) but no details of this work are known. Ishizaki (5) has briefly described a Japanese 750 kW (1000 horsepower) prototype engine.

Meijer (1) presented the results of a study made at the Philips laboratories for a 680 kW (900 horsepower) four-cylinder rhombic drive engine for stationary power having a capacity of 170 kW (225 horsepower) per cylinder, using helium as the working fluid.

Later Hoagland *et al* (6) extrapolated the established experience of the Philips licensee United Stirling with the well-developed P75 Siemens-Stirling engine. An outline design and the technology evaluated for a 1490 kW (2000 horsepower) Siemens-Stirling engine having 16 cylinders, four modules of the four-cylinder Siemens-Stirling 'square-four' arrangement shown in Figure 1. These encouraging results prompted the U.S. Department of Energy to sponsor further engineering studies of large coal-fired stationary power Stirling engines (AMT (7), General Electric (8)) in a programme reviewed by Holtz *et al* (9).

More recently, Walker (10) used the results of all these studies to prepare Table 1 containing comparative data for 2200 kW (3000 horsepower) diesel and Stirling railroad locomotive engines.

Data for the ALCO Series 251 diesel was taken from the manufacturers (Bombardier, MLW) literature. Data for the Stirling engines were taken from the studies referenced above and adjusted for common heater and cooler temperatures of 800° and 70°C, respectively, output of 2200 kW (3000 horsepower) and a speed of 16.6 Hz (1000 revs. per min.) for direct comparability with the diesel unit. The techniques for scaling and adjusting are fully described by Walker (10). Data for the rhombic engine was developed simply by adding more cylinders to the four-cylinder, 900 horsepower, engine described by Meijer (1). The result was a monstrous distortion to a long-thin snake of an engine nearly twice the length of the 16-cylinder Vee diesel. In practice a 3000 horsepower rhombic engine would likely consist of four or six large cylinders. The important point to note is that even with such gross distortion the installation volume per unit power output is *exactly the same* for the diesel and rhombic Stirling engine.

The Siemens-Stirling engines derived from the Hoagland and General Electric studies both had 16 cylinders arranged in four modules of square-four cylinders. Hoagland followed the familiar United Stirling parallel-cylinder, double crank arrangement and General Electric favoured a Vee arrangement.

COAL COMBUSTION AND HEAT TRANSPORT SYSTEM

Fluidized bed coal combustors are preferred for large Stirling applications. They operate at relatively low temperatures so that nitrous oxide production is negligible. Limestone added to the bed prevents the emission of sulphur dioxide. Coal can therefore be readily burned with little or no pretreatment apart from cleaning, crushing and screening to a uniform manageable size. Very high and predictable rates of heat transfer can be gained with fluidized bed combustors. Dunn *et al* (11) indicates heat transfer rates of 100 kW/m^2 of heat transfer surface for the boiling section of heat pipes immersed in the fluidized bed. Dunn *et al* (11) further suggests the maximum power density of fluidized bed combustors to be 650 kW/m^3. In round numbers, this corresponds to 4 m^3 per 750 kW (1000 horsepower) of engine output assuming 80 percent of the heat released in the bed reaches the engine, the engine conversion efficiency is 40 percent, and allowing 10 percent of the bed volume for heater tubes or heat pipe boiling sections.

It is possible for the engine heater tubes to be immersed directly in the bed, but the use of a sodium heat pipe is preferred to transport heat from the bed to the engine. Sodium boils in the bed section and the vapour condenses on the engine heater tubes. Very high rates of heat transfer in the boiling and condensing processes and heat fluxes as high as 15 kW/cm^2 can be achieved with little temperature difference between the combustor and the engine.

Use of the heat pipe heat transfer system permits the fluidized bed to be relatively remote from the engine. Furthermore, there are no hot spots on the engine heater tubes on which the sodium vapour is condensing so the *mean* temperature of the tubes is also the *maximum* temperature, a 50 to 100°C increase in the mean temperature compared with directly heated tubes with the inevitable 'hot spots'. Moreover, the very high rates of heat transfer permit the tubes to be optimized from considerations of the engine internal geometry and to be smaller and shorter than those required for direct heating. The increase in heater temperature and decrease in internal dead volume increases the efficiency and power output of the engine to a significant degree.

COST

Reliable data on the cost of Stirling engines is simply non-available. Hoagland *et al* (6) suggest that stationary Stirling engines burning distillate fuels would be 25 to 30 percent more than diesel engines of the same size, and, further, that Stirling engines using solid fuels would likely be 30 to 50 percent more expensive than on distillate fuels. The limiting rate of costs is then 1.6 to 2.25 the cost of diesels. General Electric estimated costs of $230 to $297 per horsepower for a 1000 horsepower engine at a production rate of 1000 engines per year and compared this with the cost of $117 per horsepower for a 2000 horsepower diesel engine in current production.

IMPEDIMENTS TO APPLICATION

A coal-burning Stirling engine costing two or three times the oil-burning diesel it replaces can be readily accepted with a large difference in the cost of oil and coal, about five times per unit of heat at 1981 world prices. Furthermore, Table 1 shows the Stirling engine could likely fit the same installation envelope and have the same power and efficiency as the diesel it replaces. It would run without noise, has excellent part-load performance and favourable torque/speed characteristics.

Yet despite all this, the prospect of Stirling engines of the type discussed above supplanting diesels is not very bright. The reason, in a word, is seals, the Achilles heel of all high performance Stirling engines.

Seals are used on the piston rods to prevent leakage of the high pressure working fluid from the working space. For high performance, high power density engines, helium or hydrogen at high pressures must be used. These gases are extremely difficult to contain and so replenishment must be carried out at intervals. The seals also prevent ingress of lubricating oil to the working space. This is even more important than preventing gas egress for the oil accumulates in the fine interstices of the regenerator, carbonizes and blocks the regenerator with consequent loss of power, overheating and burnout of the heater tubes.

Other seals on the pistons are used to separate the different working spaces of the Siemens-Stirling arrangement operating at cyclically different pressures. It is difficult to prevent some preferential 'blow-by' and as a consequence one space becomes starved of working fluid while its neighbour is gorged to excess.

Over the past 40 years a king's ransom has been invested in research and development on seals yet in 1978 Hoagland *et al (6)* wrote '... *Using state of the art seals United Stirling obtains leakage of hydrogen from the 4-piston rod seals of a 100 brake horsepower double-acting engine ranging from 0.1 to 0.2 grams per hour at full load. This life of their piston rings as well as the rod seals is currently about 4000 hours...*'.

In North America the anticipated, and routinely achieved, life of a railroad diesel electric locomotive before major maintenance is 500 000 miles. The prospect of regularly replenishing helium working fluid (assuming it would be preferred to hydrogen for reasons of safety despite some loss of power) and changing seals every 4000 or even 10 000 hours is simply unacceptable.

In this desperate situation new approaches are clearly required and we present these below.

NOVEL APPROACHES: THE CONVERSION OF DIESELS TO STIRLING ENGINES

We have seen how large coal-burning Stirling engines could be designed and constructed to replace the oil-fuelled diesel engines in railroad locomotives and so permit the gradual conversion and continued usage of existing locomotive fleets. The greatest impediment to implementation is the limited life of seals on the piston and piston rod and the need to replenish unfamiliar working fluids such as helium.

We propose to overcome these problems by the approaches outlined below and, to go further, to show how the existing railroad locomotive diesel engines may be converted to Stirling engines of a special type called Ringbom-Stirling engines.

Use Single-Acting Engines

Abandon the use of double-acting engines and revert to ensembles of single-acting engines on a common crankcase used in the Philips prototype engines of the 1960's.

Use Air as the Working Fluid

Use air as the working fluid instead of helium or hydrogen to eliminate the need for a hermetic gas seal. Reasonable leakage from the working space can be replenished from a reservoir fed by an engine driven air compressor.

Figure 2 shows the maximum possible thermal efficiency as a function of power density (brake horsepower per litre of engine displacement) for Stirling engines of 225 brake horsepower per cylinder using air, helium and hydrogen. This data was presented by Meijer (1) and resulted from the study of the 900 horsepower stationary power generator discussed above. The important point to note is that the endpoint of the curve for air indicates a power density of only 17 brake horsepower per litre of engine displacement. Yet this is precisely the same power density of 17.4 brake horsepower per litre engine displacement of the ALCO Series 251 diesel engine given in Table 1. Furthermore, the Stirling engine has a power stroke every revolution whereas the diesel is a four-stroke engine and has a power stroke every two revolutions. The speed of Stirling engines at 400 revolutions per minute is then not too dissimilar to the 525 working strokes per minute of the diesel engine. However to provide a direct replacement of the same speed, the Stirling engine could be accelerated to 1000 revolutions per minute without significant change in power and efficiency by relaxing the pressure to compensate for the increase in speed. Air was always used as the working fluid in regenerative engines until the early 1950's when Philips perceived the need to develop very high output engines comparable to small automotive gasoline and diesel engines.

The lower efficiency (30 percent) at maximum power of the Stirling engine is not significant for a locomotive engine spends little time at maximum power. Ephraim *et al* (12) published the typical duty cycle for a North American railroad locomotive which calls for the engine to be operating above 50 percent maximum power output for only 40 percent of the time. The loss in efficiency at high power is therefore compensated by the increased efficiency of the Stirling engine at lower power. The diesel engine reaches its maximum efficiency close to the rated output and the efficiency declines at lower output.

Use Steam Injection for Maximum Power

Use steam injection in the hot expansion space to obtain a power boost during periods of high power operation. This corresponds to the process of water or methanol injection used in all Rolls-Royce Merlin engines for Spitfires in World War II when taking off or in combat.

In the steam injection Stirling engine superheated steam at high pressure would be injected to the expansion space from a monotube, once-through, super-critical boiler immersed in the fluidized bed or sodium vapour heat pipe near the expansion space. The steam would be admitted through a valve operated automatically by the control system when the displacer was approaching top dead centre. The steam injected would condense in the regenerator and accumulate as liquid in the cold compression space. It would act as the lubricant for the piston rings and be pumped out of the working space by the piston seals. The water naturally descends from the hot expansion space to the cold compression space according to the Law of Minimum Energy States. Its presence in the compression space is beneficial for it acts as the seal lubricant. Furthermore, it improves the efficiency of the seals since they are now hydraulic rather than pneumatic. Another advantageous feature is that the water improves the heat transfer process in the compression space so that compression is closer to the ideal isothermal process.

Preliminary studies at the University of Calgary of two-phase, two-component working fluids have shown that very substantial increases in output may be gained compared with a gaseous working fluid. Figure 4 shows the theoretical work diagrams obtained by Walker *et al* (13) for two-phase two-component and gaseous working fluids. Injection of significant amounts of steam during periods of high power demand could likely double the power output of the engine but with substantial loss of efficiency.

Use Water as the Lubricant

Prevention of oil contamination of the working space is difficult to achieve in conventional Stirling engines. The problem can be eliminated by substituting water instead of oil as the lubricant. This also eliminates the problem of separating oil from the 'return' water to be reused for steam injection Stirling engines.

Water is not as good as oil for lubrication but is tolerable in special circumstances. Its use in oxygen compressors and steel rolling mills has been routine for many years. Furthermore, the lubrication requirements in Stirling engines are very much less demanding than in diesel engines. This is particularly true for the piston rings which have a less arduous duty in Stirling engines. Moreover, the extremely high rates of pressure change experienced during combustion in diesel engines giving impact-like loading to the crankshaft and connecting rod bearings do not exist in Stirling engines. The pressure-time curves are virtually sinusoidal and the maximum rates of pressure change are very low in comparison with diesel engines.

Convert Diesel Engines to Stirling Engines

Figure 5 shows a Ringbom-Stirling engine. The piston is coupled to a crank and connecting rod and a free-displacer moves in an adjacent cylinder under the action of fluidic and inertia forces acting upon it. This type of engine was invented in 1905 by Ossian Ringbom, a Russian living in Finland. A complete discussion of the engine is given in the book by Walker and West (14). This arrangement permits existing internal combustion engines to be converted to Ringbom-Stirling engines. Figure 6 shows an early Ringbom-Stirling engine made at the University of Calgary (Walker (15)) by converting a Honda industrial engine. More recently, a prototype 1 kW Ringbom-Stirling engine has been made by Sunpower Inc. (Beale (16)) pioneers of free-piston Stirling engines.

Applying the principle of conversion the ALCO Series 251 diesel engine shown in Figure 7 might appear as the Ringbom-Stirling engine shown in Figure 8. A more comprehensive discussion of various aspects of this conversion is given by Walker (3) and (10).

THE COAL-BURNING STIRLING LOCOMOTIVE

Application of the Ringbom-Stirling engine described above to a railroad locomotive is illustrated in Figure 9. The locomotive is modelled after the Bombardier MLW M630 six-axle 3000 horsepower diesel electric locomotive in current usage.

Twin-fluidized bed coal combustors are located beneath the frame of the locomotive between the bogies. There is space also for a small buffer coal store but for long hauls the locomotive would tow a coal tender in the same way diesel locomotives tow fuel tankers.

Heat from the fluidized beds are conveyed to the engine by sodium heat pipes clustered in well-insulated conduits. The additional cooling required for Stirling engines is obtained by enlarging and double-banking the existing cooling water 'radiators'.

An electrically driven air compressor and reservoir provides compressed air for initially fluidizing and heating the combustors, for starting and for charging or recharging the Stirling engine. A large engine driven fan supplies air to the fluidized bed when the system is operating. A large recuperative exhaust gas/inlet air preheater replaces the customary diesel engine muffler.

Apart from the above changes, replacements or additions, the rest of the Stirling engine locomotive is exactly the same as the present diesel electric locomotive. The principal difference to the environment of the conversion to coal-burning Stirling locomotives is the absence of noise and a very much reduced level of air pollution.

Kneiling (17) suggests that conventional diesel electric units on long runs with an oil tank car attached consume 2 (U.S.) gallons of diesel oil per 1000 gross ton miles. Two power units hauling a 5000 ton train uses 10 (U.S.) gallons per loaded ton mile. A 1000 mile haul and empty return trip consumes 15000 (U.S.) gallons. The same road trip with two coal-burning Ringbom Stirling engines would consume 75-80 tons of coal (Walker (10)).

EXPERIMENTAL PROOF-OF-PRINCIPLE ENGINE

An experimental proof-of-principle Ringbom-Stirling diesel engine conversion is presently under development at the University of Calgary. The engine is illustrated in Figure 10. The conversion is based on the Lister 8/1 single cylinder diesel engine of 8 horsepower at 850 revolutions per min. The diesel engine piston is used as the crosshead direct coupled to a double-acting or pressure balanced piston. The space above the piston, the compression space, is connected by a duct to the Ringbom cylinder mounted coaxial to the diesel cylinder.

The Phase 1 prototype will be heated by natural gas/air blow torches but the heater has been designed for eventual combination with a condensing sodium vapour heat pipe and with the cylinder horizontal. A more complete description of the design is given elsewhere (Walker (18)).

CONCLUSION

Given the substantial price differential between coal and oil, the future for coal-burning Stirling engines in heavy traction applications appears brighter than ever before. It appears feasible to convert existing diesel-electric railroad locomotives to coal-burning Stirling-electric locomotives. Moreover, it appears possible to modify the diesel engines to Ringbom-Stirling engines. Given the large inventory of existing diesel-electric locomotives the potential savings of conversion compared with outright replacement are enormous.

ACKNOWLEDGEMENTS

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The proof-of-principle experimental engine is being developed under contract to the Department of Supply and Services, Government of Canada.

REFERENCES

1. MEIJER, R. Prospects for the Stirling Engine for Vehicular Propulsion. Philips Tech. Rev., Vol. 31, No. 5/6, pp. 168-185, 1970.

2. MORGAN, D.T. Thermal Energy Storage for the Stirling Powered Automobile. Dept. of Energy Report, No. ANL-K-78-4135, 1978, (available N.T.I.S., Springfield, Va.).

3. WALKER, G. Large Stirling Engines. (monograph in preparation), University of Calgary, Alberta, Canada.

4. Walker, G. Stirling Engines. Oxford University Press, 1980.

5. ISHIZAKI, Y. Stirling Engine Development in Japan. Paper No. 799437, Proc. 14th Inter. Soc. Energy Conv. Eng. Conf., pp. 2035-38, Boston, Aug. 1979.

6. HOAGLAND, L.C. and PERCIVAL, W.H. A Technology Evaluation of the Stirling Engine for Stationary Power Generation in the 500 to 2000 Horsepower Range. Dept. of Energy, Report No. ORO/5392-01, 1978, (available N.T.I.S., Springfield, Va.).

7. ANON. (Advanced Mechanical Technology Inc.). Design and Development of Stirling Engines for Stationary Power Generation in the 500 to 3000 Horsepower Range. Dept. of Energy, Report No. DOE/ET/15207-T1, 1980, (available from N.T.I.S., Springfield, Va.).

8. ANON. (General Electric). Design and Development of Stirling Engines for Stationary Power Generation Applications in the 500 to 3000 Horsepower Range. Dept. of Energy, Report No. DOE/ET/15209-T1, 1980, (available from N.T.I.S., Springfield, Va.).

9. HOLTZ, R.E., MARCINIAK, T.J. and UHERKA, K.L. DOE Stationary External Engine Programme: Status Report. Paper No. 799242, pp. 1120-1123, Proc. 14th Inter. Soc. Energy Conv. Eng. Conf., Boston, August. 1979.

10. WALKER, G. The Stirling Engine for Railroad Locomotive Application. Report for Transport Canada, Transportation Development Centre, Montreal, 1981, (available from G. Walker, Dept. of Mechanical Engineering, University of Calgary, Alberta, Canada).

11. DUNN, P.D., RICE, G. and THRING, R.H. Hydraulic and Rotary Drive Stirling Engines with Fluidized Bed Combustion/Heat Pipe Systems. Paper No. 759141, pp. 942-947, Proc. 10th Inter. Soc. Energy Conv. Conf., Newark, Del., Aug. 1975.

12. EPHRAIM, M. and ADDIE, A.N. Diesel Traction in North America. Jnl. Railway Eng., Inst. of Mechanical Eng., London, Vol. 14, pp. 73-84, 1975.

13. WALKER, G. and AGBI, B. Thermodynamic Aspects of Stirling Engines with Two-Phase Two-Component Working Fluids. Trans. Can. Soc. Mech. Eng., Vol. 2, No. 1, pp. 1-8, 1974.

14. WALKER, G. and WEST, C. Free-Piston and Liquid-Piston Stirling Engines. (monograph in press) (to be published 1982).

15. WALKER, G. Stirling Cycle Machines. Oxford University Press, 1973.

16. BEALE, W. Private Communication. Sunpower Inc., Athens, Ohio.

17. KNEILING, J.G. Trains for Tomorrow. Trains, July, pp. 28-33, August 1977, pp. 44-50.

18. WALKER, G. Design of Ringbom-Stirling Diesel Engine Conversion. Proc. 17th Inter. Soc. Energy Conv. Conf., San Diego, Calif., Aug. 1982.

Table 1

Comparative Data for 3000 Horsepower Diesel and Stirling Locomotive Engines

Column No.	1	2	3	4	5	6	7
	Series M251 Diesel	Meijer Rhom-bic Drive Engine B-Fig. 11	Revised Meijer Rhombic	Hoagland *et al* (1978) Engine	Revised Hoag-land Engine	General Elec-tric (1980)	Revised General Electric
Horsepower (gross)	3050	900	3094	2000	3000	1000	3000
No Cylinders	16	4	14	16	16	8	16
HP/Cylinder	190.6	225	221	125	187.5	125	187.5
Speed RPM	1050	810	1000	850	1000	1200	1053
Cylinder Bore (ins)	9	-	-	7.18	9.55	7	7.97
(mm)	*228.5*	-	-	*182*	*242.5*	*177.8*	*202.4*
Piston Stroke (ins)	10.5	-	-	4.0	5.32	4.0	4.55
(mm)	*266.7*	-	-	*102*	*135.1*	*101.6*	*115.5*
Swept Vol/Cyl (in^3)	668	-	-	162	380	153.94	227
(litre)	*10.95*	-	-	*2.65*	*6.22*	*2.52*	*3.72*
Total Swept Vol (in^3)	10688	-	-	2592	6080	1231	3632
(litre)	*175.20*	-	-	*42.4*	*99.52*	*20.16*	*59.5*
Engine Weight Drag (lbs)	40800	-	-	32000	39336	12222	36600
(kg)	*18507*	-	-	*14500*	*17880*	*5555*	*16636*
Weight/HP (lbs/HP)	13.37	-	-	16	13.1	12.2	12.2
(kg/HP)	*6.06*	-	-	*7.25*	*7.25*	*5.55*	*5.55*
Length (ins)	191	90	317	143	164	72	166.5
(m)	*4.89*	*2.3*	*8.05*	*3.632*	*4.16*	*1.82*	*4.22*
Breadth (ins)	60.5	39.4	39.4	50	57.5	84	96.6
(m)	*1.55*	*1.0*	*1.0*	*1.270*	*1.45*	*2.13*	*2.45*
Depth (ins)	96.5	94.5	94.5	68	78.2	78	89.7
(m)	*2.47*	*2.4*	*2.4*	*1.727*	*1.98*	*1.37*	*2.27*
Installation Volume (ft^3)	645.3	193.9	683	281.4	421.8	273	834.9
(m^3)	*18.75*	*5.52*	*19.32*	*7.96*	*11.94*	*5.31*	*23.46*
Vol. per HP (ft^3/HP)	0.215	0.215	0.215	0.1406	0.1406	0.273	0.273
(m^3/HP)	*.0061*	*0.006*	*0.006*	*0.0039*	*0.0039*	*0.0053*	*0.00782*
HP/Unit Swept Volume (HP/in^3)	.285	.655	.655	.7716	.493	.813	-
(HP/litre)	*17.40*	*40*	*40*	*47.16*	*30.14*	*49.58*	*50.4*
Max. Thermal Efficiency (%)	37	39	39	39.6	35	40	40
Heater Temperature (°C)	-	700	800	800	800	760	800
Cooling Water (°C)	-	25	70	30	70	80	70
Working Gas	Air/Fuel	Helium	Helium	Helium	Helium	Helium	Helium
Mean Operating Pressure (psi)	-	1552	1261	2030	1297	1470	1470
(MPa)	-	*10.7*	*8.7*	*14.00*	*8.94*	*10.1*	*10.1*
Max. Pressure (psi)	1450	-	-	-	1.733	1984	1984
(MPa)	*1.0*	-	-	-	*11.95*	*13.68*	*13.68*
Min. Pressure (psi)	-	-	-	-	866	955	955
(MPa)	-	-	-	-	*5.97*	*6.58*	*6.58*
Mean. Eff. Pressure (psi)	215	-	-	-	-	-	-
(MPa)	*1.48*	-	-	-	-	-	-
Engine Type	Diesel	Stirl. Rhomb.	Stirl. Rhomb.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.	Stirl. Siem.
Column No.	1	2	3	4	5	6	7

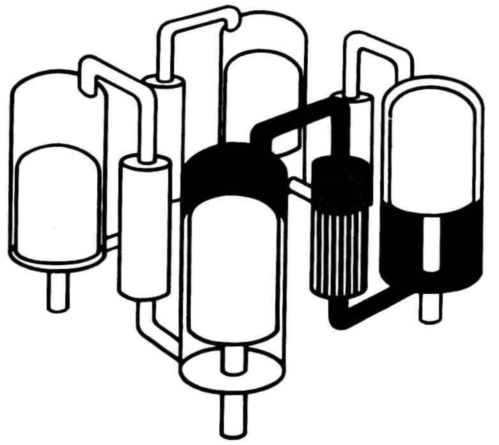

Fig 1   The square-four arrangement of the Siemens-Stirling engine

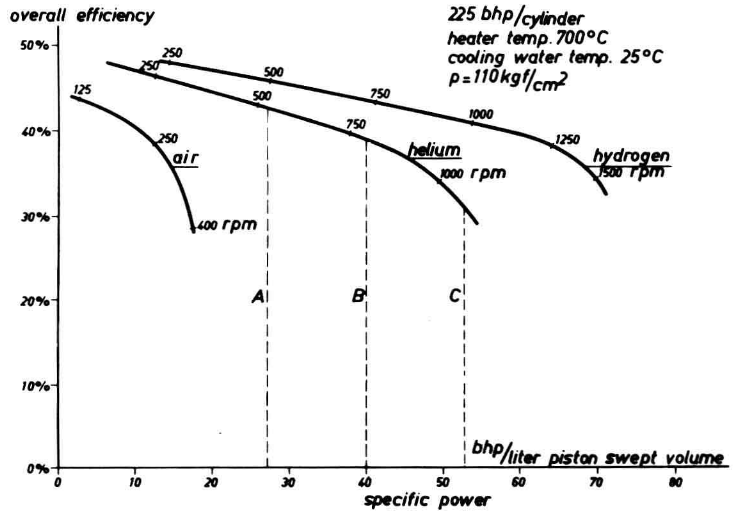

Fig2   Maximum possible efficiency of a Stirling engine as a function of the power density with air, helium and hydrogen working fluids, an output of 225 brake horsepower per cylinder and a mean working fluid pressure of 10.7 MPa (110 kg_f/cm^2).

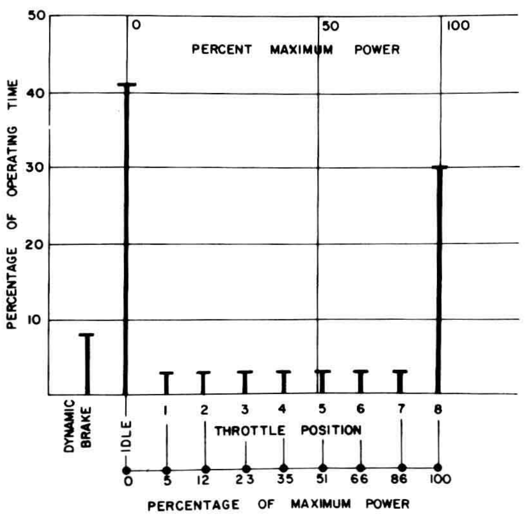

Fig 3   Representative duty cycle for North American railroad locomotive, (after Ephraim *et al* (12))

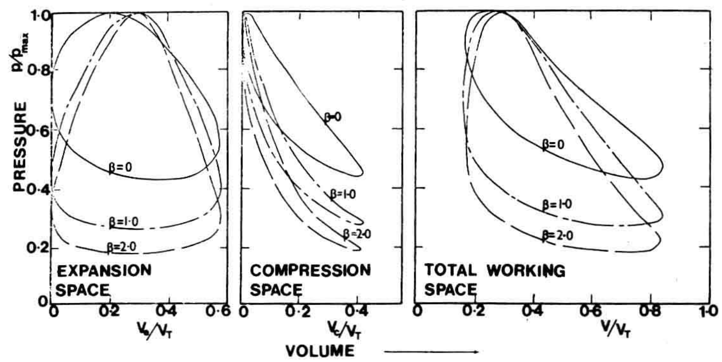

Fig 4   Comparison of theoretical work diagrams for Stirling engines with gaseous and two-phase two-component working fluids. β is the mass ratio of phase change component, (after Walker(13))

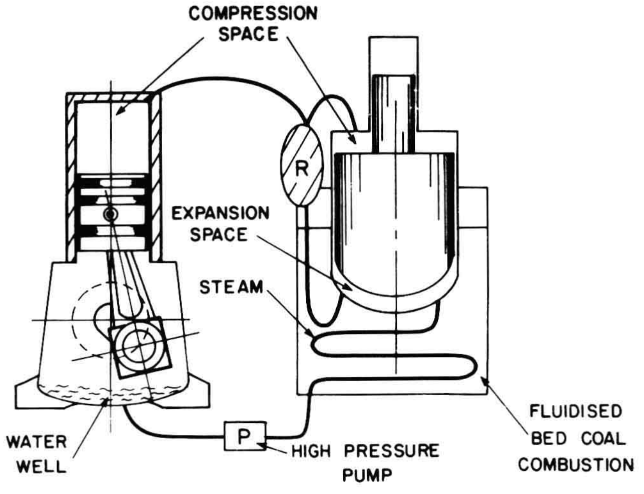

Fig 5 The Ringbom-Stirling engine

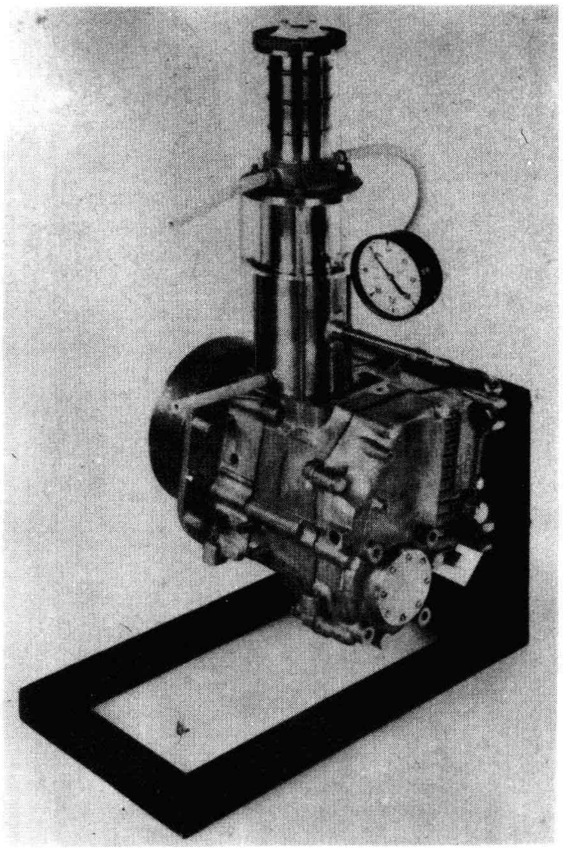

Fig 6 Early Ringbom-Stirling engine made by converting a Honda industrial engine

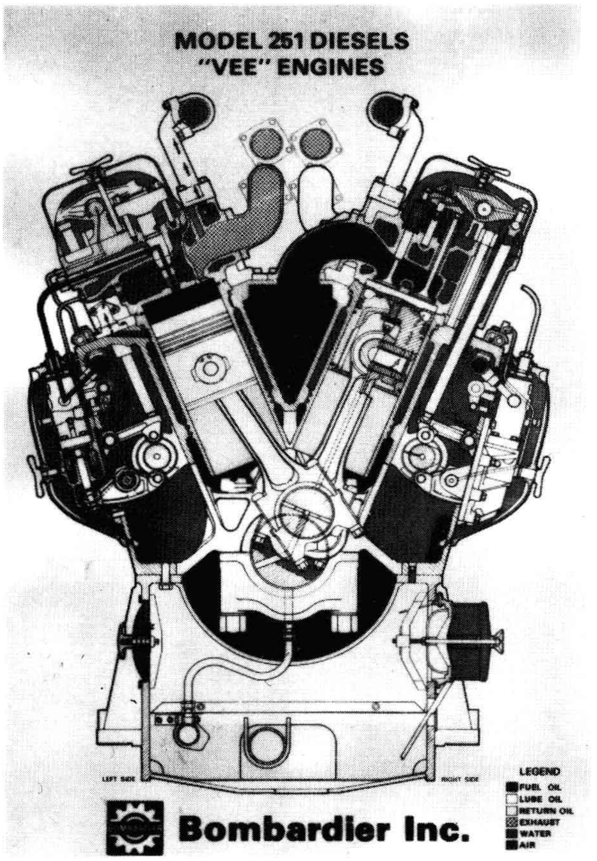

Fig 7 Cross-section of MLWSeries 251 four-stroke diesel vee engine, (after Bombardier MLW)

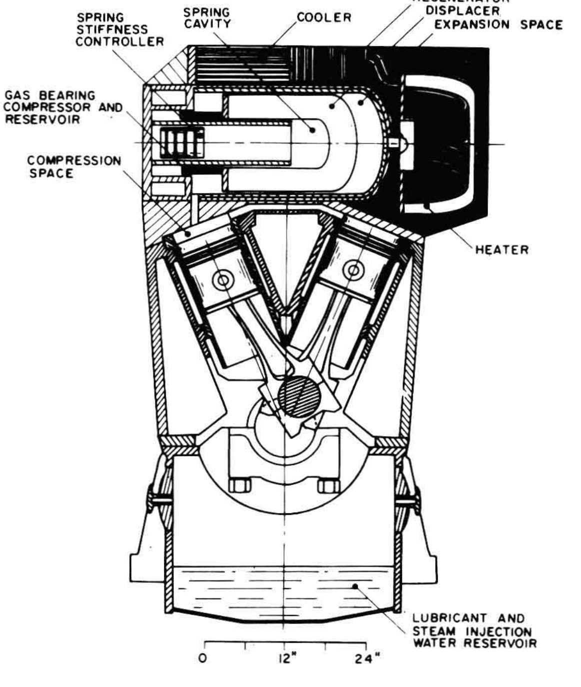

Fig 8 Ringbom-Stirling engine based on conversion of the MLW Series 251 diesel engine with one displacer cylinder per pair of engine cylinders

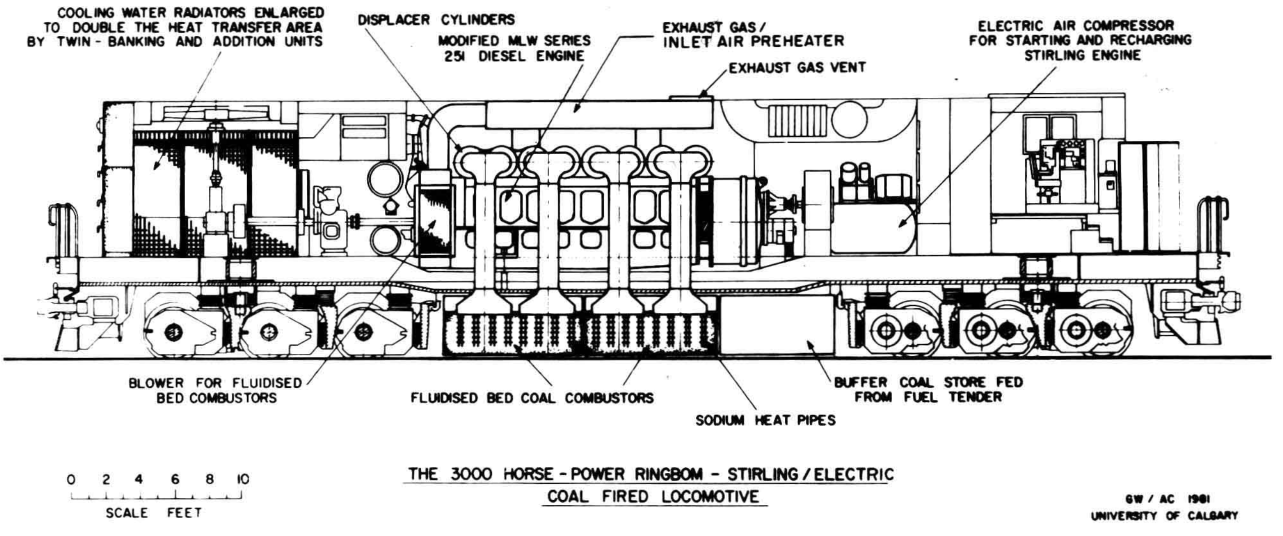

Fig 9 The coal-burning Ringbom-Stirling powered railroad locomotive

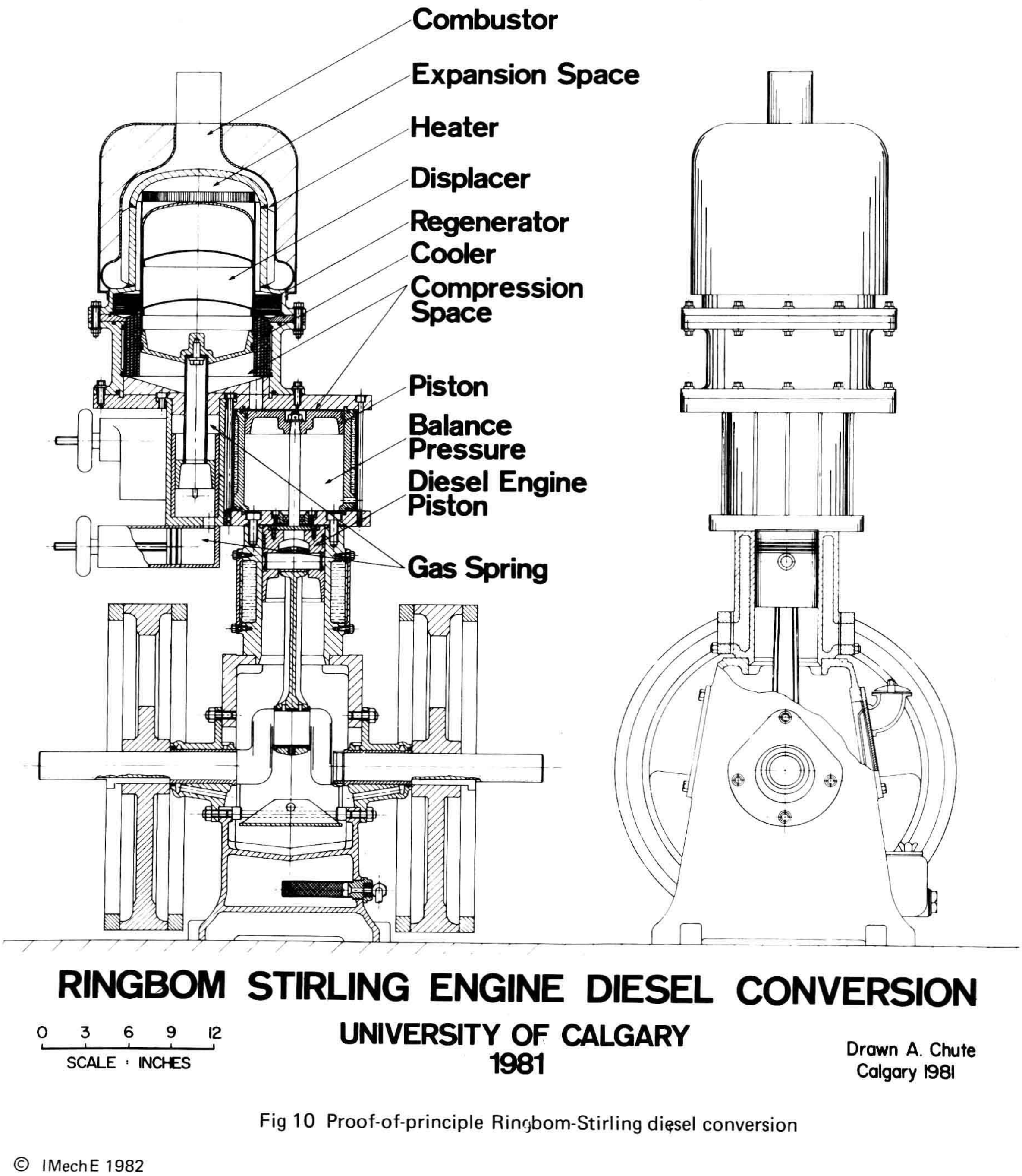

Fig 10 Proof-of-principle Ringbom-Stirling diesel conversion