The Development and Testing

of a prototype Domestic Gas Fired Heat Pump

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Abstract

This paper describes a research project undertaken at the Glynwed Central Research Unit laboratory, Birmingham, UK, between 1976 and 1980 to develop a prototype gas fired domestic heat pump. The system is based on an Organic Rankine Cycle (ORC) directly coupled to a reverse Rankine Cycle Heat Pump.

A miniature high-speed turbo machine (180,000r.p.m.) is used with a compressor and expander located on a common shaft.

Gas bearings are utilised to allow oil-free operation with the same working fluid being used in both the power and heat pump cycles.

THE DEVELOPMENT AND LABORATORY TESTING OF A PROTOTYPE DOMESTIC GAS FIRED HEAT PUMP

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Summary

This paper describes work carried out over a six year period at the central research laboratories of Glynwed International.

The project involved the specification, design and development of a 10 kW output Rankine/Rankine cycle domestic heat pump.

The design adopted utilised miniature turbo-machinery having a rotational speed of 180,000 rpm. An expansion turbine coupled directly to a centrifugal compressor with gas lubricated bearings was incorporated to permit oil free operation. The same organic working fluid (R114) was used in both the power cycle and heat pump cycle, with the entire system being hermetically sealed.

Many design and development problems were encountered and these are discussed in this paper, together with a detailed analysis of the laboratory prototype test results.

1. Introduction

The original specification called for a gas fired domestic heat pump having the following features: the minimum number of moving parts, a low maintenance requirement, silent and vibration free operation together with the potential to offer a realistic payback time.

The design adopted utilised miniature turbo-machinery operating on a Rankine/Rankine cycle, with the same organic working fluid (R114) being used in both the power cycle and heat pump cycle. This solution allowed the design to be oil free which permitted a high working fluid temperature (250°C) to be achieved in the power cycle, which resulted in a satisfactory Rankine cycle thermal efficiency. The target PER (prime energy ratio) for the overall system was 1.35.

Other features of the proposed design included the potential for reliable low maintenance operation, with gas lubricated bearings being used to support the high speed turbo-machine.

Since both the power cycle and heat pump cycle were hermetically sealed, shaft leakage problems did not occur and the unit was almost silent when operational.

Of the many design, development and test rig problems encountered, differential thermal expansion and the need to maintain small clearances in the turbo-machine proved to be particularly difficult to resolve.

Despite having satisfactorily demonstrated performance with the laboratory prototype the project was abandoned. This was a result of higher than originally anticipated manufacturing costs for mass production of the turbo-machine, and technical problems associated with the maintenance of small running clearances in the journal bearings under all conditions of operation.

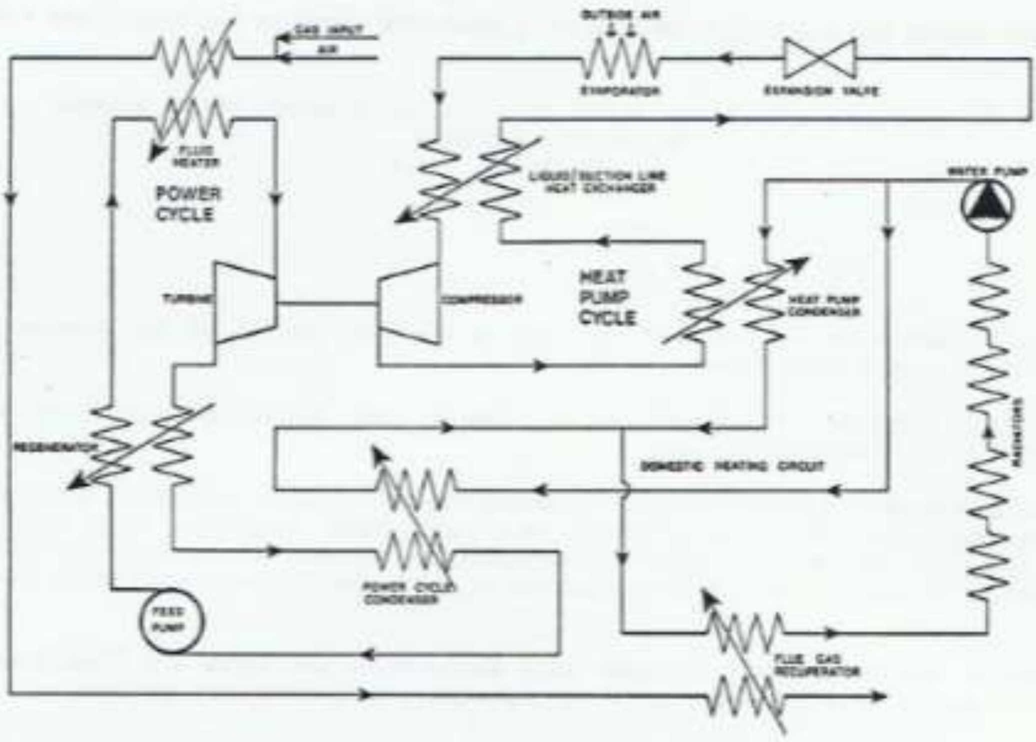

Fig 1. Schematic Diagram - Directly Fired Heat Pump

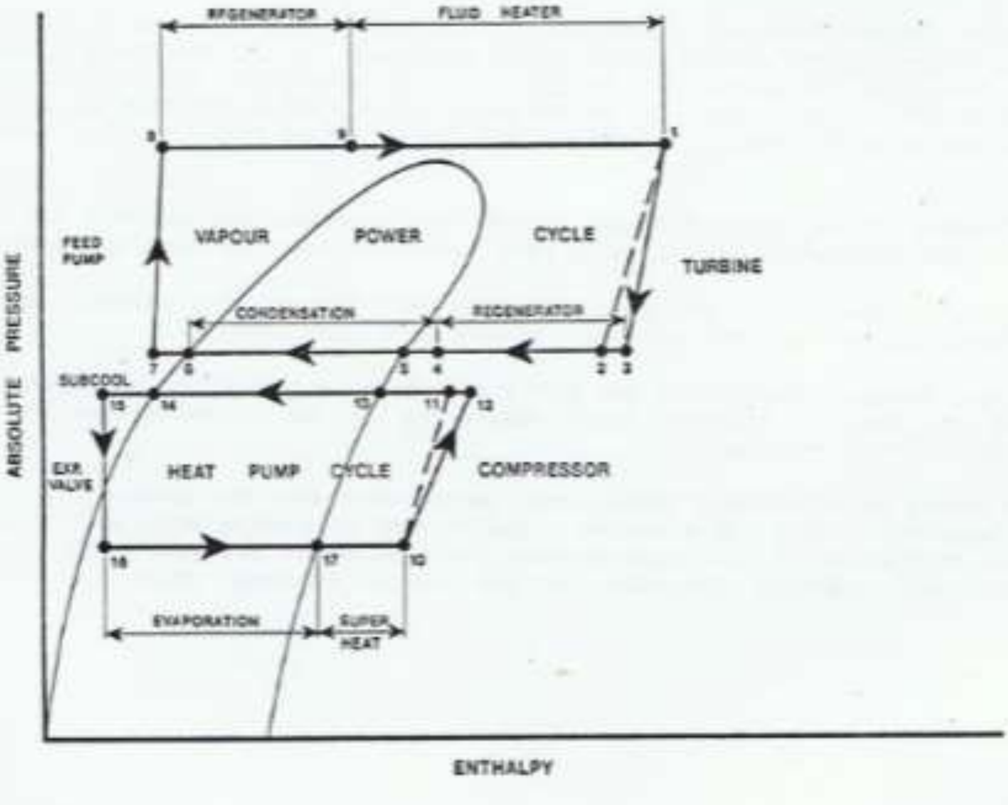

Fig 2. Pressure - Enthalpy Diagram

2. Description of the Rankine/Rankine cycle

A simplified schematic diagram of the power cycle and heat pump cycle is shown in Figure 1 The following gives a brief description of processes involved :-

a) A small feed pump raises the pressure of liquid in the Power cycle to a pressure above the critical pressure (see Figure 2).

b) Preheating of the fluid occurs in a regenerator, by energy transferred from the hot turbine exhaust gases.

c) The fluid is raised to turbine inlet temperature by input from an external high temperature energy source, at the fluid heater (it should be noted that no change of phase occurs in either regenerator fluid heater, since the fluid is in a supercritical condition).

d) The fluid is expanded through the turbine to produce shaft power for driving the centrifugal compressor.

e) Hot exhaust gases from the turbine are used to preheat the fluid at the regenerator.

f) Energy is recovered at the condenser to provide part of the energy input to the domestic heating circuit.

g) The heat pump cycle is a conventional air/water vapour compression heat pump cycle with the compressor being driven directly by the turbine.

3. Specification and optimisation of the power and heat pump cycle

The selection of optimum operating parameters and conditions for the combined cycles was carried out using a number of computer models.

A suite of computer programs were developed at the outset of the project to permit a detailed analysis to be made related to the following factors :-

a) choice of working fluids programs were developed to give thermodynamic and thermophysical properties at all conditions of interest).

b) cycle operating conditions and the influence of pressure, temperature, heat exchanger effectiveness and pressure drop.

c) the influence of turbine and compressor isentropic efficiency, expansion ratio and compression ratio etc.

The optimisation of the cycles proved to be extremely complex with the requirement to specify the design operating conditions of seven heat exchangers, a turbine, compressor and feed pump. These were assessed from an ideal thermodynamic aspect together with practical engineering constraints such as strength and compatibility of materials, safety, cost and manufacturing requirements. The interaction and effect on overall efficiency related to changes in cycle operating conditions and component performance were closely examined with the results being displayed in both tabular and graphical form.

3.1 System design parameters

The computer optimisation of the power cycle resulted in the following basic specification :-

Working fluid:	R114
Total heat output:	10 kW

	Minimum allowable	Design value
Turbine efficiency (isentropic)	65%	70%
Compressor efficiency (isentropic)	55%	70%
Feed pump efficiency (volumetric)	60%	95%
Regenerator effectiveness	75%	85%
Fluid heater effectiveness	70%	85%
Turbine inlet temperature	250°C	250°C
Turbine inlet pressure	5,000 kN/m^2	5,000 kN/m^2
Turbine outlet pressure	500 kN/m^2	500 kN/m^2
Compressor outlet pressure	500 kN/m^2	500 kN/m^2
Compressor inlet pressure	90 kN/m^2	90 kN/m^2
Power for auxiliary equipment	200 Watts	200 Watts

The predicted overall PER (prime energy ratio) for the design is 1.43 with the minimum allowable values giving 1.12.

4. Working fluid selection

The selection of a suitable working fluid for the power and heat pump cycle is of crucial importance.

4.1 Fundamental working fluid requirements

The requirement to maintain realistic rotational speeds and blade passage dimensions in low power turbo-machines implies the use of a high molecular weight working fluid. Of the thousands of potential fluids many can be eliminated on grounds of toxicity, flammability, cost, availability or unsatisfactory vapour pressure/temperature characteristics. Fluids were also eliminated on grounds of being corrosive or not compatible with materials of construction. A further fluid requirement was inertness and stability over the entire range of pressure and temperature conditions.

4.2 Candidate working fluids

A significant simplification in turbo-machine and system design occurs if the same working fluid is used in both the heat pump and power cycle. Unfortunately from a thermodynamic aspect the requirements of an 'ideal' power cycle working fluid differ in certain aspects from those required of a refrigerant in a heat pump cycle.

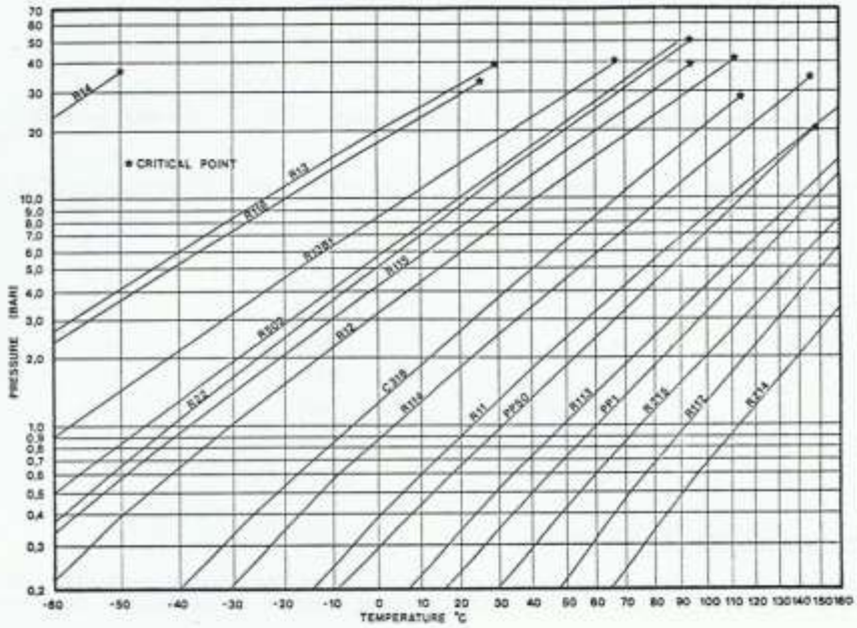

Fig 3. Pressure - Temperature Relationship of Fluorocarbon Working Fluids

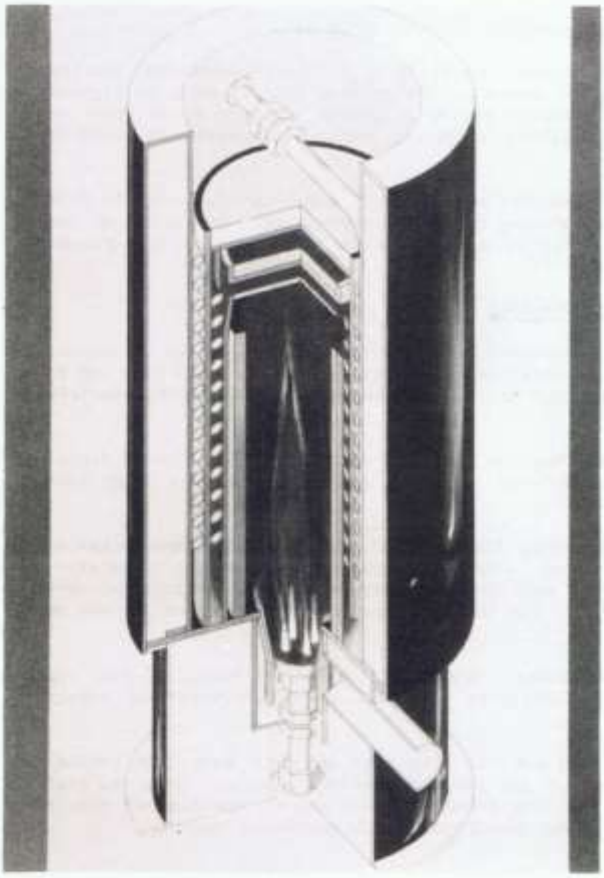

Fig 4. Cutaway view of Fluid Heater

With the above factors in mind the following short list of candidate working fluids was produced :-

Working fluid	Formula	Molecular weight	Boiling point (°C) (Atms. Press)
R11	CCl_3F	137.4	23.8
R12	CCl_2F_2	120.9	-29.8
R21	$CHCl_2F$	102.9	8.9
R22	$CHClF_2$	85.6	-40.8
R113	CCl_2FCClF_2	187.4	47.6
R114	$CClF_2$-$CClF_2$	170.9	3.8
C318	C_4F_8 (cyclic)	200.0	-5.8
PP50	C_5F_{12}	288.0	29.0

4.3 Thermal stability of working fluid

Since it is desirable to operate the power cycle at the highest possible turbine inlet temperature, thermal stability of the proposed working fluid is of fundamental importance.

A great deal has been published related to the thermal stability and chemistry of chlorinated and fluorinated hydrocarbons. Much of the published data and thermal stability test results are of a contradictory nature and in many cases must be treated with considerable caution.

However, a number of conclusions can be drawn from the published data :-

a) A relationship exists between the thermal stability of a chlorofluorocarbon and the number of chlorine and fluorine atoms present. The general rule is that the highest thermal stability occurs with compounds having a greater proportion of fluorine atoms than chlorine atoms, and highest stability occurs in totally fluorinated fluids.

b) Materials of construction and contaminants such as oil, air, water, grease or dirt can have a significant effect in promoting thermal decomposition. In particular the presence of oil will have a catalytic effect and reduce substantially the thermal stability of the working fluid.

4.4 Fluid specification and thermal stability testing

Of the eight candidate working fluids summarised in Section 3.2 several can be eliminated on grounds of unsatisfactory vapour pressure/temperature characteristics which would have led to extremely high or low pressures in the system and as a result expensive containment materials (see Figure 3).

The fluid C318 was of particular interest being a very stable, totally fluorinated fluid. Unfortunately the fluid is currently available only in small quantities at very high cost (£60/kg).

Of the commonly available chlorofluorocarbons R114 had all the hallmark characteristics required for good thermal stability together with a satisfactory pressure/temperature relationship. Also, R114 was well documented with full thermodynamic and thermophysical data (and was available at a moderate cost £3.00/kg). For these reasons R114 was selected for use in the prototype machine.

Before the turbine inlet temperature was finally specified, a thermal stability test rig was constructed to test R114 under cyclic conditions at 250°C in the presence of likely materials of construction.

The thermal stability testing was carried out oil free over an eight week test period. Fluid samples were taken at regular intervals for gas chromotographic analysis. Over the test period no discernible increase in volatile impurity level occurred and it was deduced that a turbine inlet temperature of 250°C could be safely specified for the prototype machine.

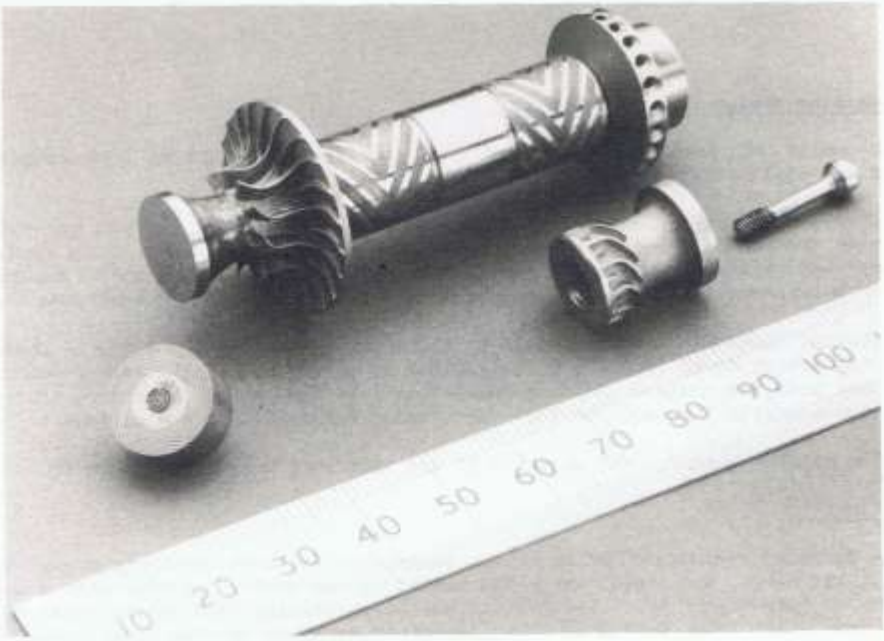

Fig 5. Turbomachine Rotor

Fig 6. Turbomachine Components

5. Turbo-machine design and specification

A wide range of possible types and configurations of turbine and compressor were considered at the design stage.

The prototype turbine had a rated power output of 1.53 kW at 180,000 rpm and the smallness of size of both the turbine and compressor gave much cause for concern. In particular the optimum shape and size of blade passages and problems associated with manufacture and maintenance of tolerances and running clearances presented a considerable technical challenge.

The prototype turbo-machines were made by machining metal from the solid by spark errosion and by utilising a high speed (0.030") dental cutter in an end mill. (In mass production it was anticipated that both the turbine and compressor would be cast using high precision investment casting techniques - with the required tolerances being achieved with the minimum of machining).

The turbo-machine used for the majority of the test work consisted of the following :-

a) Turbine

A shrouded radial inflow turbine, comprising an outer rotating shroud with radial inflow holes, and a seal which was bolted to the shaft, together with a bladed inner rotor (see Figure 5). Five partial admission nozzles feed the 22 holes in the outer rotor which lead to the 22 blade/splitter passages of the inner rotor - giving a flow path similar to a conventional radial turbine (the type of turbine is best described as a 'shrouded mixed flow turbine').

b) Compressor

The compressor is based upon a conventional design of outward flow radial impeller. Design analysis led to a compressor tip diameter of 26.24mm (1.033 ins) being specified with 11 full vanes and 11 splitter vanes having 40° of backsweep, discharging into a radial diffuser. The diffuser was of a straight wedge design having 29 vanes.

c) Bearings

Herringbone groove inward pumping self acting journal bearings were utilised. These were applied to the shaft by ion-beam etching to a depth of 100 micron - the herringbone profile being produced by a stainless steel mask made by photo-reduction and electro-chemical machining.

A spiral groove self acting thrust bearing was incorporated in the first design of turbo-machine but was abandoned in favour of an externally pressurised thrust bearing as a result of insufficient load carrying capacity.

A considerable effort was made to test the bearings at the design stage. Several test rigs were constructed to allow a number of gas bearing concepts to be examined. The herringbone groove bearing was selected since it gives excellent shaft stiffness and permitted close running clearances to be achieved in the turbine and compressor.

6. Fluid heater design

The gas fired fluid heater had to be designed with great care so that fluid decomposition should not occur as a result of hot spots or localised overheating. A high thermal efficiency was also required together with a need to be compact and potentially low cost in mass production. A further requirement was for a fast response and an ability to modulate energy input.

The prototype gas fired fluid heater is shown in Figure 4. A matrix type gas burner was incorporated together with spark ignition and a flame failure system. Dilution of combustion gases with secondary air reduces combustion gas temperature to 1160K before entering the first part of the heat exchanger.

The heat exchanger section employed a novel configuration to enable a high differential temperature to be maintained between the entering hot combustion gasses and the R114 so as to avoid excess internal wall temperatures or localised overheating. The first pass for the

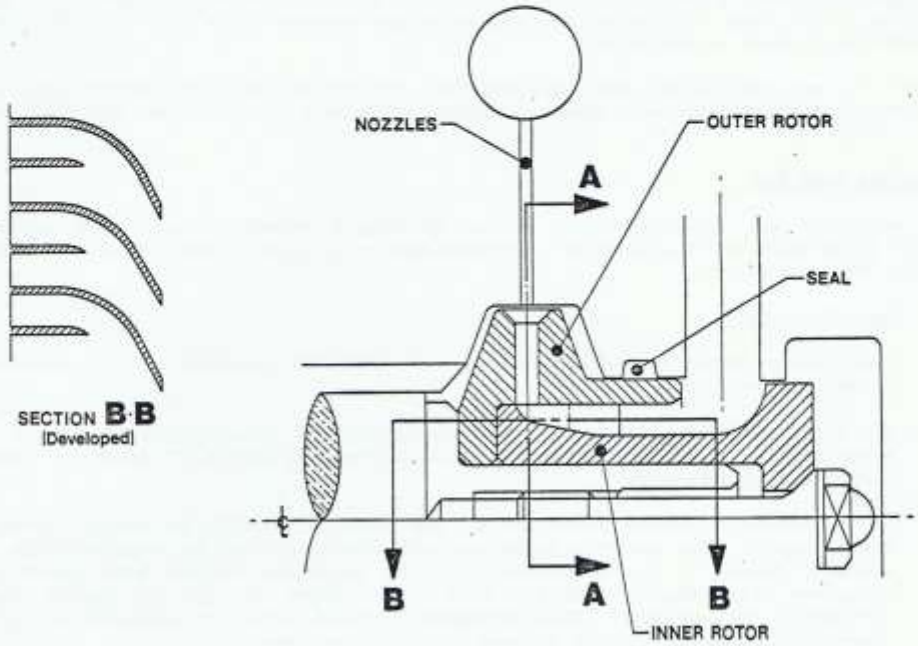

Fig 7. Shrouded Radial Inflow Turbine

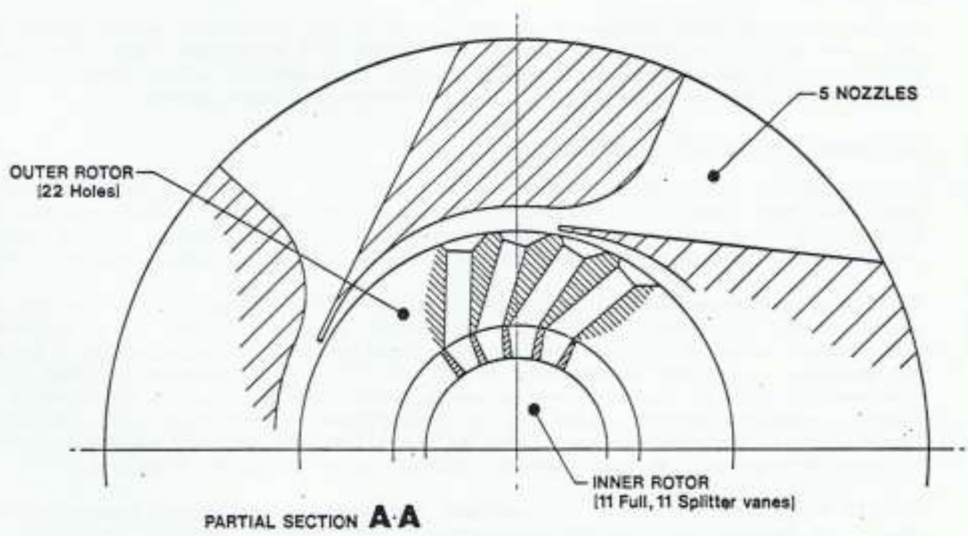

Fig 8. Turbine Nozzle and Rotor Detail

combustion gases is over a section of tubing through which the R114 is pumped in a contra-parallel flow. The final section of the heat exchanger consists of a simple spiral tube with R114 and combustion gases in contraflow.

A test rig was constructed which allowed full testing of the fluid heater under a wide range of operational conditions with measured thermal efficiency being between 80-85 percent.

7. Prototype Test Rig

The test rig was constructed so as to be highly versatile and allow performance measurements to be made over a wide range of operational conditions. The test rig consisted of the following main components:-

a) Heat Exchangers

Proprietary water cooled condensers were utilised together with a conventional finned tube evaporator.

The regenerator was constructed from externally and internally wire finned tubing with the high pressure fluid being pumped through the centre of concentric tubing in a contra-flow arrangement.

An electrically heated fluid heater was used to provide an easily controllable energy input. The electric fluid heater consisted of ten, 2m long cartridge heater elements having a low Watt-density. The cartridge heaters were inserted into stainless steel tubing with the R114 being pumped through the tubing over the outside of the heaters. This arrangement proved to be very satisfactory with the elements being controlled by a heavy duty Thyristor controller.

b) Feed Pump

The feed pump specified for use on the test rig was of the balanced diaphragm type (Wanner Engineering Hydra-cell). The pump performed well giving the required high outlet pressures and low mass flow rate. The pump was self priming and oil free (on the working fluid side) and gave low pulse flow. Since no shaft seals are used in this type of pump leakage of R114 did not occur. Variable flow rates were achieved by using a variable speed drive and thyristor controller to give flow rates in the range 40 - 200 litres/hour.

c) Ancillary and Safety Equipment

Standard refrigeration equipment (pipework, fittings, receivers, safety valves etc) were used on the low pressure side of the system with stainless steel pipework and fittings being used on the high pressure side. A number of safety interlocks were incorporated together with an over pressure/temperature alarm system.

d) Instrumentation and data-logging

A full computer data-logging system was incorporated. Pressures, temperatures and mass flows were measured at relevant positions on the test rig. Many problems were experienced with the instrumentation and data logging systems, with regular recalibration being required and difficulties with computer hardware and software.

Mass flow measurement caused particular problems. Three Venturi flow meters were connected via a series of computer controlled solenoid valves to a calibrated differential pressure cell. Problems were experienced with back-leakage of R114 at the solenoid valves and condensation of R114 in the static pressure tappings. The condensation problem was overcome by using tracer heater elements to maintain the static pressure tapping lines at a temperature higher than the condensing temperature. Back leakage was solved by installing the solenoid valves in a "back to back" arrangement, so that leakage could not occur in either direction.

Despite a significant number of problems the data-logging system proved to be a useful and potent tool allowing performance (shaft power, turbine, compressor and cycle efficiency, etc) to be quickly and continuously calculated and displayed together with temperatures and pressures at all intermediate conditions.

The computer was also used to examine and test a number of control algorithms to give automatic start up, shut down and running control.

8.0 Test Results

The testing occurred over a two year period with many modifications to the turbo-machine and test rig.

The following sections give a brief summary of the performance achieved by the various components.

8.1 Turbine Performance

As a result of the small physical size difficulties occurred in making accurate temperature and static pressure measurements within the turbo-machine. For this reason the performance estimates were treated with some caution - particularly at low speeds (below 120,000 rpm) and at low mass flow rates, because of the influence of internal heat transfer.

However, the following main conclusions can be drawn:-

i) The turbine exceeded the target isentropic efficiency under design operating conditions, with an average isentropic efficiency of 72% being recorded.

ii) Turbine power output was satisfactory with a peak output of 1.67 kW being measured (design power output = 1.53 kW.

iii) After modification to the nozzle profile outlet velocity from the nozzles was satisfactory.

8.2 Compressor Performance

Despite many modifications to the compressor and diffuser the target efficiency and pressure ratio were never achieved.

The maximum isentropic efficiency recorded was 60% (design = 70%) and a maximum pressure ratio of 5.2:1 (design = 6.7:1).

Reasons for the shortfall in compressor performance were high internal leakage losses, design errors related to scaling from larger machines, and the use of inaccurate working fluid density data at the design stage.

8.3 Bearing Performance

The journal bearings demonstrated excellent performance over a limited range of operating conditions.

The main problem associated with the journal bearings was the requirement for a very small running clearance, to give satisfactory bearing performance and shaft stiffness (max. diametral clearance = 7.62×10^{-3} mm (3.0×10^{-4} inch).

After several shaft seizures it became apparent that differential thermal expansion was the main cause of seizure. The required clearance could be achieved within tolerance in a closely controlled temperature environment. However, satisfactory running of the rotor would occur only if the difference in temperature between the shaft and sleeve did not exceed 50°C.

The rotor was run satisfactorily for many hours after installation of an electric heater to maintain the shaft sleeve at an appropriate temperature relative to the rotor. This caused particular problems under start up conditions when shaft and sleeve temperatures were changing rapidly.

8.4 Overall Performance measurement

Overall performance measurements gave a prime energy ratio (PER) of between 1.1 and 1.2 for an evaporating temperature of 1°C and a condensing temperature of 57°C.

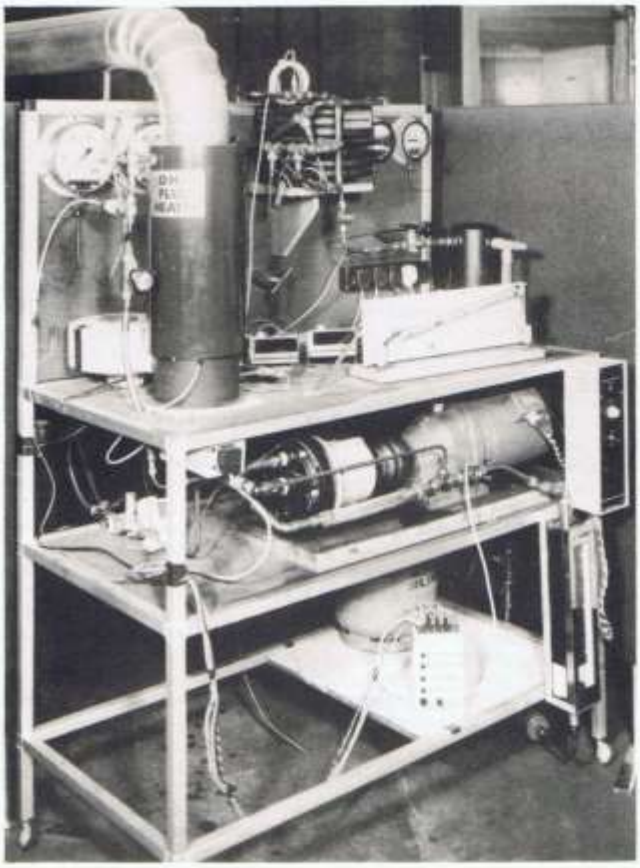

Fig 9. Fluid Heater Test Rig

Fig 10. Prototype Test Rig and Data Logger

Unfortunately low evaporating temperatures could not be achieved as a result of the lower than design compressor pressure ratio.

The reason for the low P.E.R. was poor compressorn efficiency and heat losses from the test rig (partially caused by instrumentation).

9.0 Conclusions

Like many good thermodynamic ideas the constraints and costs of practical engineering have conspired to prevent the concept becoming a reality!

Much of value has been achieved:-

- the demonstration that a chloroflurocarbon can be utilised at much higher temperatures than previously thought possible (provided no oil or contaminants are present).
- miniature high speed turbomachines can be designed and manufactured with realistic blade passage dimensions and practical tolerances.
- excellent isentropic efficiencies can be achieved from a shrouded mixed flow turbine.
- high rotational speeds can be achieved repeatedly on herringbone groove bearings (provided a close control is maintained of running clearances).

The project was abandoned as a result of a significantly higher manufacturing cost than had been anticipated at the outset of the project. The high manufacturing costs were a consequence of the small tolerances required within the turbomachine (this could possibly have been overcome by development of specialist production equipment).

The sophisticated control system, and problems associated with differential thermal expansion in the bearings also led to the decision to abandon the work.

It should be noted that many of the above problems were caused by the smallness of size, and a larger machine (possibly greater than 100 kW heat output) could have significant potential.

References

1. Strong, D.T.G. Development of a Directly Fired Domestic Heat Pump. D. Phil Thesis Oxford University 1980