1996

A Rotary Positive Displacement Heat Pump Compressor and Turbine Combined in One Rotor

R. W. Driver
Driver Technology Ltd.

D. P. Davidson
Driver Technology Ltd.

A ROTARY POSITIVE DISPLACEMENT HEAT PUMP COMPRESSOR AND TURBINE COMBINED IN ONE ROTOR.

R. W. Driver,
Engineering Director
D.P.Davidson,
Chief Executive
DriverTechnology Ltd. UK

ABSTRACT

A novel positive displacement turbine and compressor has the unique capability to efficiently compress and expand in a continuous cycle in one rotor at 3600 r/min or less. In a heat pump cycle the first 180 degrees provides compression, whilst the second 180 degrees between the condenser and the evaporator inlet provide controlled expansion. The slow speed positive displacement turbine has a low tip speed and therefore low intake losses. The combined effect of low intake loss and positive displacement expansion gives a high intrinsic coefficient of performance.

The unique mechanism avoids friction losses and wear problems, and the rotational speed allows conventional plain bearings to be used, giving low unit costs in sizes down to 30,000 BTU/Hr. Performance measurements confirm high isentropic and mechanical efficiencies .

NOTATION

A	=	leakage area
cp	=	specific heat at constant pressure
cv	=	specific heat at constant volume
g	=	gravitational acceleration
*m*l1	=	fluid mass leakage from '1' to 'f'
ma	=	inlet fluid mass at temperature *Ta*
*m*al	=	fluid mass leakage from inlet to '1'
n	=	cp/cv
nc	=	number of vane compartments
*P*1	=	total pressure in compartment '1'
Pa	=	condenser pressure
*P*f	=	total pressure in final compartment
r/sec	=	revolutions per second
R	=	gas constant
*T*1	=	total temperature in compartment '1'
Ta	=	inlet temperature
*T*f	=	temperature in final compartment
*T*fi	=	ideal total temperature in final compartment

INTRODUCTION

Domestic heat pumps for heating and cooling buildings are relatively small and normally use positive displacement reciprocating motion to avoid high tip speeds. Safeguards are needed to prevent slugs of liquid from entering the cylinder and causing damage during the compression stroke. At other than large sizes it is not usually practical to do work when expanding the fluid at exit from the condenser and so a simple throttle is used in which thermodynamic losses occur when expansion takes place. Cylinder head clearances produce additional thermodynamic losses.

Sliding vane turbines have been used for many years and are well known. The problems associated with these devices are friction, sealing and lubrication. Adequate vane tip sealing at low speed conditions results in excessive tip loading and wear at high speed. Inertia and pressure loads produce friction losses from the faces of the vanes as they slide in their grooves. Oil lubrication sufficient to reduce wear can reduce heat exchanger performance while dry lubricants have relatively high coefficients of friction giving short lives. Turbine isentropic efficiency is likely to be less than 50% with very poor mechanical efficiency. A Roots type machine is unsuitable as a turbine since there is no internal expansion between inlet and outlet. They also have poor compression efficiency above a compression ratio of about 2:1.

This paper describes a unique positive displacement device, that is inexpensive and has all of the desired characteristics but in which the seemingly inevitable efficiency and rotor tip wear problems are eliminated by the use of a novel mechanical arrangement.

POSITIVE DISPLACEMENT TURBINE

Turbine construction

The current device has six cam-shaped vanes which rotate inside a static circular casing and avoid contact with the casing by pivoting as they rotate (Figure 4). The vanes are connected via crank arms to connecting rods arranged like the spokes of a wheel (Figure 5). The little end of the conrod is centred on line with the vane tip and the big end centred on the centre of the static casing. This makes the vane tips rotate in a circle concentric to, and nearly coincident with the bore of the static casing. Close to the side ends of the vanes are discs which rotate with the vanes and in which the vanes pivot. These side discs extend along the sides of the static casing beyond the vane tip diameter. The spoke conrods and pivoting shafts are all carried on plain journal bearings. Lubrication for the bearings is outside the plane of the turbine and divorced from it. The bearing loads are relatively light which allow the bearings to have the optimum length/diameter ratio for minimum friction losses. All the moving parts use standard plain bearings and the device has one location ball bearing. Intake mass flow variation can be provided if necessary by sliding a portion of the casing in a circular movement about the centre of rotation, thus exposing the required intake cross-sectional area. Apart from bearings, all constituent parts of the heat pump are able to be made from pressure diecast magnesium or aluminum. The weight of the turbine assembly and drive is between 5 and 8 kg.

Turbine Operation

In the heat pump cycle the first 180 degrees of rotation of the combined compressor and turbine takes vapor from the evaporator and, as the mechanism closes, it compresses the fluid and delivers it to the condenser. The mechanism, now fully closed, begins to open as the second 180 degrees of rotation commences and in doing so is exposed to the return fluid from the condenser. The energy in the fluid drives the turbine around and the centrifugal effect separates the liquid and vapor. At exit from the turbine the liquid is deflected away from the vapor and fresh vapor from the evaporator is added when a slight pressure drop in the vapour provides acceleration to the relatively low speed of rotation. The cycle then repeats.

At entry to the turbine from the condenser the fluid is almost saturated liquid but the opening turbine mechanism reduces the fluid pressure until there is sufficient momentum in the expanding vapor to accelerate the liquid to rotor speed: this is typically achieved when the dryness fraction is less than 20%. The increase in heat pump coefficient of performance that can be achieved by providing a turbine expansion in the cycle is about 25%.

COMPRESSOR AND TURBINE EFFICIENCIES AND PERFORMANCE

Assumptions for the rotary positive device

With a knowledge of predicted bearing loads and industry-accepted bearing theory, the friction losses can be calculated. Turbo-machinery and Roots compressor isentropic efficiency can be accounted for if it is assumed that compressable fluid expands isothermally in accordance with Joule's law when leaking from high to lower pressure. Fluid leakage between adjacent vane compartments can be calculated using well known theory. The final fluid conditions, when stabity is achieved, can be used to calculate the isentropic efficiency of the turbine. In the present turbine there are two vanes between inlet and outlet. The equation for the mass of condenser pressure fluid leaked (m_{a1}) past one vane to adjacent compartment '1' is:

$$m_{a1} = \sqrt{\frac{n\ g\ R}{\frac{n-1}{2}}\ (T_a - T_1)}\quad \frac{P_1\ A}{T_1\ R\ (r/sec\ \ nc)}$$

and for compartment '1' the fluid leaked (m_{11}) past one vane to the outlet 'f' is:

$$m_{11} = \sqrt{\frac{n\ g\ R}{\frac{n-1}{2}}\ (T_1 - T_f)}\quad \frac{P_f\ A}{T_f\ R\ (r/sec\ \ nc)}$$

After a number of iterations a solution is found when T_f becomes constant.

$$\text{Turbine isentropic efficiency} = \frac{T_a - T_f}{T_a - T_{fi}}$$

Turbine performance

With Tf and bearing loads calculated the turbine efficiencies can be found.

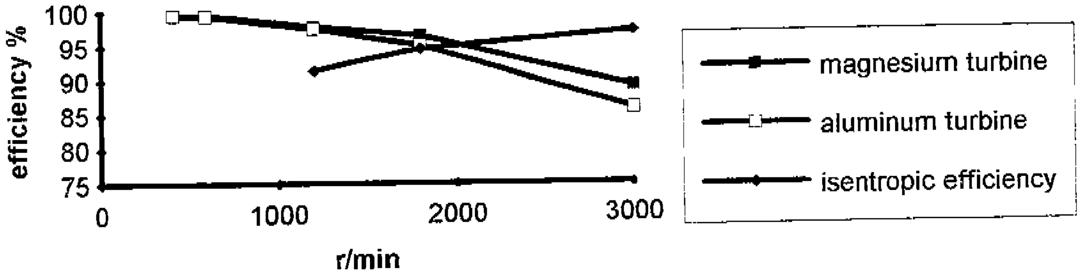

Figure 1 Turbine isentropic efficiency and mechanical efficiency.

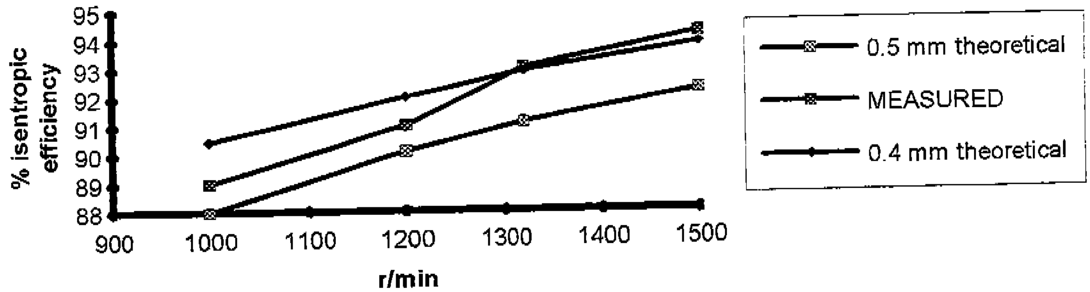

Figure 2 Measured efficiency for 0.4 mm vane tip clearance with air as fluid compared to theoretical vane tip clearance of 0.5 mm and 0.4 mm measured efficiency

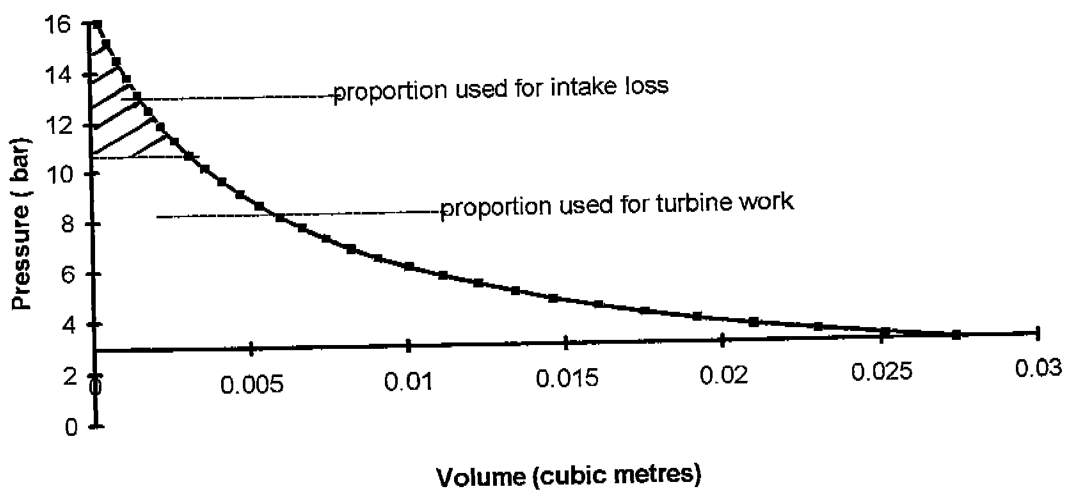

Figure 3 Pressure - Volume diagram for fluid returning to evaporator

CONCLUSIONS

The rotary positive displacement heat pump described in this paper recovers energy from the fluid returning to the evaporator at speeds that can be 3600 r/min and less. The return fluid energy recovered improves the coefficient of performance by over 25%. Low intake loss, low operational speed, and high thermodynamic and mechanical efficiency combine to give a high overall coefficient of performance. The simplicity of the novel design and pressure diecast aluminium components provide a low cost, high efficiency, heat pump.

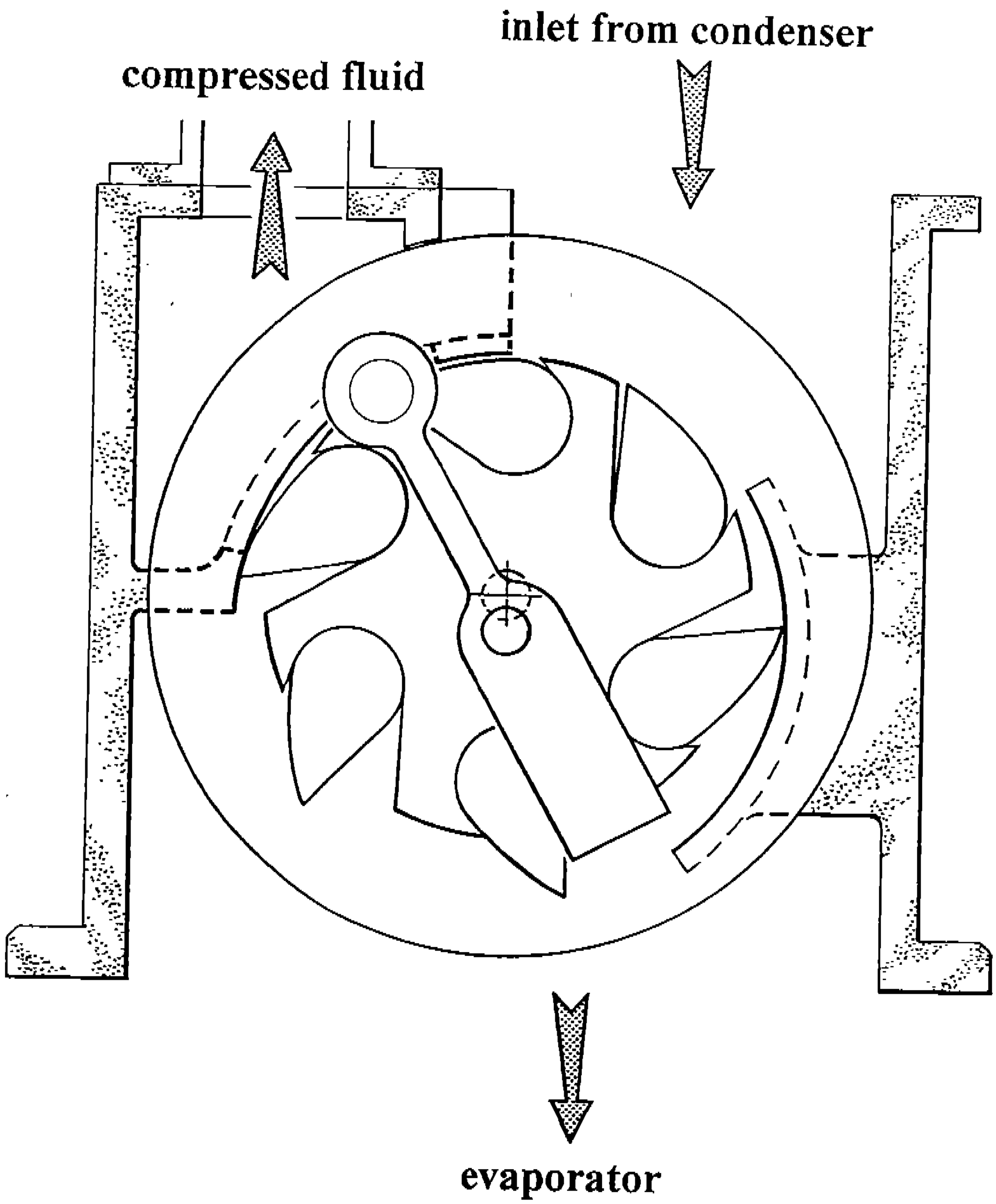

FIG 4

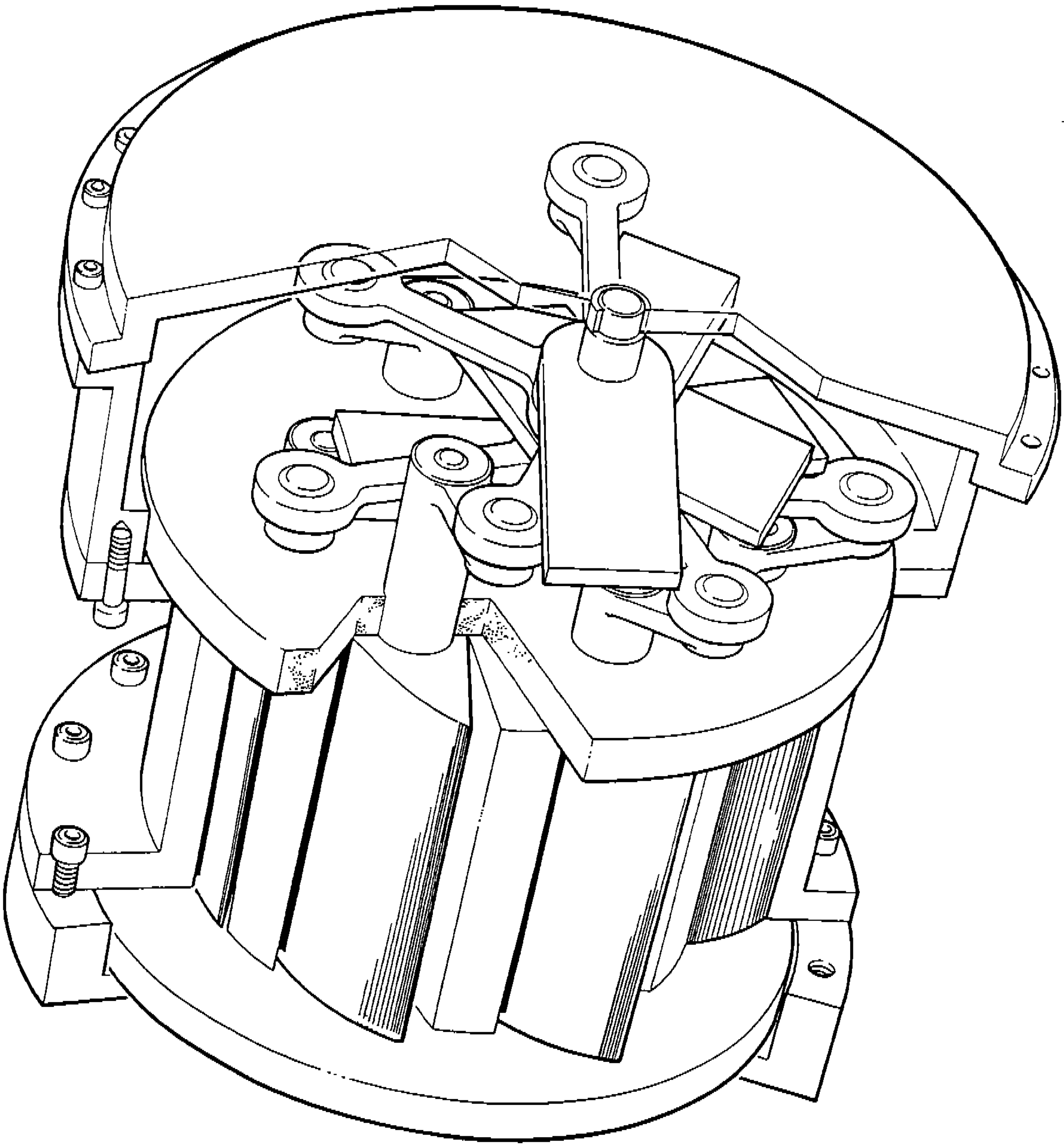

FIG 5