Head-flow and npsh performance of an axial piston pump working with organic fluids at different temperatures

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A modified version of the standard axial piston pump, normally used with fuels, has been tested with organic fluids R11 and R113. Head-flow characteristics, volumetric and global efficiency, and npsh curves, have been determined at different speeds and fluid temperatures and the results compared with those obtained with kerosene. Pump efficiencies remain satisfactory with high density, very low viscosity and high vapour pressure fluids. In the absence of cavitation, pump performance seems to be a function of kinematic viscosity, while the npsh curves appear to be a complex function of density, viscosity and vapour pressure

Keywords: *Cavitation, axial piston pumps, performance*

Developments over the last few years in the organic Rankine cycle systems adopted for power generation from low–medium temperature heat sources have been remarkable. Applications of such power cycles are becoming increasingly common[1,2]. According to a generalised thermodynamic analysis[3], the cycle efficiency can be expressed as:

$$\eta = 1 - \frac{T_{\min}}{T_{\max}} - \frac{T_{\min}}{Q_1} \sum_{1}^{n} {}_{\mathrm{i}}\, \Delta S_{\mathrm{i}} \qquad (1)$$

where $T_{\min}$ and $T_{\max}$ are the minimum and maximum cycle temperatures respectively, Q_1 the heat input and ΔS_{i} the entropy increases generated in the five main system components: pump, evaporator, turbine, regenerator and condenser.

Focusing attention on the pump, pump losses are generally very low compared to the other components' losses. When certain working fluids are used, however, especially in regenerative cycles with high maximum pressure, the pumping work becomes a relevant percentage of the useful work and, thus, it can affect the cycle efficiency substantially. For saturated cycles the efficiency decrease can be expressed approximately by:

$$\Delta\eta_p = \frac{1-\eta_{\mathrm{p}}}{\eta_{\mathrm{p}}} \frac{1-\eta}{[\Delta h_{\mathrm{v}}]_{T_{\min}}} V_1 \Delta p \qquad (2)$$

where η_{p} is pump efficiency, η the cycle efficiency, $[\Delta h_{\mathrm{v}}]_{T_{\min}}$ the latent heat at the minimum cycle temperature, V_1 the liquid specific volume, and Δp the pump delivery-suction pressure difference. The relative incidence of such a loss on cycle efficiency is given by:

$$I = \frac{\Delta\eta_{\mathrm{p}}}{\eta} = \frac{1-\eta_{\mathrm{p}}}{\eta_{\mathrm{p}}} \cdot \frac{1-\eta}{\eta} \frac{V_1 \Delta p}{[\Delta h_{\mathrm{v}}]_{T_{\min}}} \qquad (3)$$

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It has been shown[3] that the influence of pump efficiency on cycle performance is more important when:

- the cycle maximum pressure is high;
- the cycle efficiency is low;
- the liquid specific volume is high;
- the latent heat of the working fluid is low.

Since the latent heat at a given temperature of fluids with similar critical temperatures is almost inversely proportional to their molecular mass, the pump efficiency must be particularly high for working fluids

Notation

I	$\Delta\eta_p/\eta$
n	Rotational speed, r/min
p_a	Suction pressure, bar
p_v	Fluid vapour pressure, bar
Q	Actual flow rate, m^3/h
Q_1	Cycle heat input, kJ/kg
T	Suction temperature, °C
T_{c}	Fluid critical temperature, °C
T_{cond}	Cycle condensing temperature, °C
T_{e}	Cycle evaporation temperature, °C
$T_{\max}$	Maximum cycle temperature, K
$T_{\min}$	Minimum cycle temperature, K
T_{R}	Fluid reduced temperature
V_1	Liquid specific volume, m^3/kg
γ	Density, kg/m^3
Δh_{v}	Latent heat, kJ/kg
Δp	Pump delivery-suction pressure difference, bar
ΔS_{i}	Entropy increase, kJ/kg °C
$\Delta\eta_{\mathrm{p}}$	Pump efficiency decrease
η	Cycle efficiency
η_{g}	Overall pump efficiency
η_{p}	Pump efficiency
η_{v}	Volumetric pump efficiency
ν	Kinematic viscosity, m^2/s

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having high molecular mass, as in the case of the organic Rankine cycles.

In Fig 1 the incidence of the pump efficiency on cycle efficiency can be read for various saturated vapour Rankine cycles working with different fluids at various evaporation temperatures. A constant condensing temperature of 40 °C was assumed with fixed values of turbine efficiency and cycle pressure losses, while two different values of pump efficiency were considered. From the figure, it can be seen easily how high such incidence is in the case of organic fluids, especially at the highest evaporative temperatures where cycle maximum pressure is high. The $\Delta\eta_p$ increases significantly with decreasing fluid critical temperature.

Pump performance is necessarily influenced by the physical and chemical characteristics of the pumped medium. For turbomachinery, similarity can be invoked to derive pump performance in different conditions; the performance of a volumetric pump working with organic fluids cannot be derived easily from the known performance obtained with traditional working fluids such as water or oil. For this reason, a test rig has been built in the laboratory of Energetics Department of Politecnico di Milano and an extensive series of experiments is being carried out to evaluate the head-flow characteristics, the efficiency performance and the npsh curves of pumps working with organic fluids. The very low viscosity and the high density and vapour pressure of these fluids are expected to be the most influential parameters on the pump's performance.

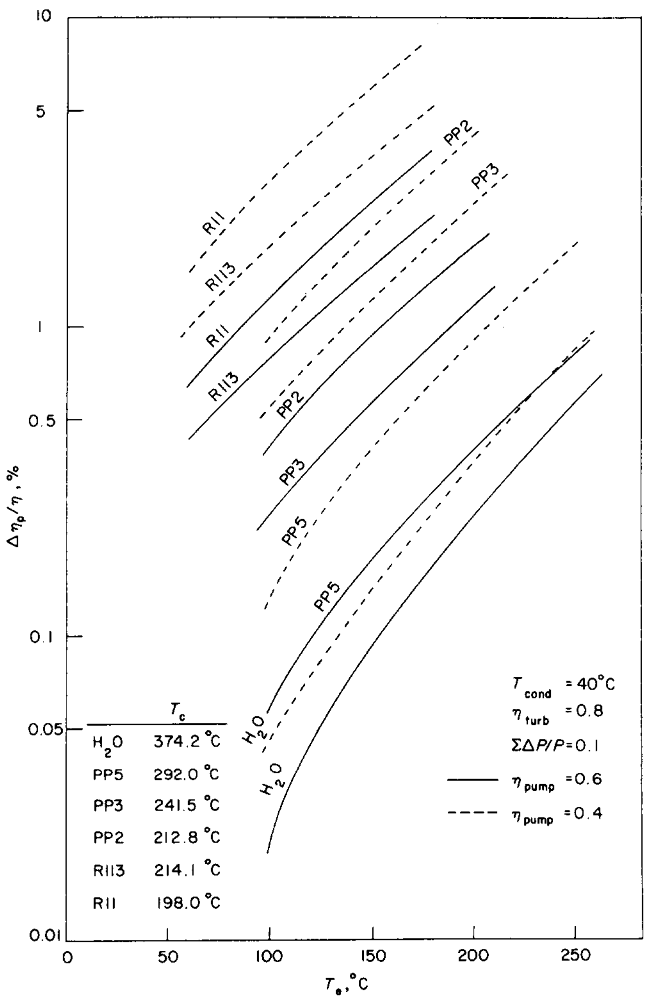

Fig 1 Pump efficiency incidence on cycle efficiency for different fluids

Test pump

The pump tested is a variable-stroke axial piston swashplate pump; the camplate can be tilted by a servo piston to control the stroke of the pistons. This type of pump was chosen because it should be able to satisfy the head-flow requirements of many organic Rankine cycles with expected high efficiency.

The pump has limited overall dimensions, is lightweight and can operate at relatively high rotational speeds. Another important feature is that all bearing surfaces are lubricated by the working fluid, thus avoiding any possible contamination of the working fluid with lubricant.

The standard version of this pump, the so called 'all metal' pump, was developed mainly as fuel pump for aircraft engines. An 'all carbon' version and a 'long life' version of this pump were developed to overcome problems arising from pumping low viscosity and low lubricity fuels, such as light virgin naphta. In the carbon version, carbon facings on one element of the major bearing areas, ie on slipper tops, rotor bores, and port insert, are fitted, thus reducing wear rates and increasing volumetric efficiency because of the closer clearances which can be obtained. In the 'long life' version, the carbon port, susceptible to erosion, is replaced by a tungsten carbide insert, running against a tungsten carbide disk on the end of the pump rotor.

As the organic fluid has very low viscosity, lower than that of the aircraft fuels, serious problems are expected to rise when such fluids must be pumped. In the absence of specific experience, the manufac-

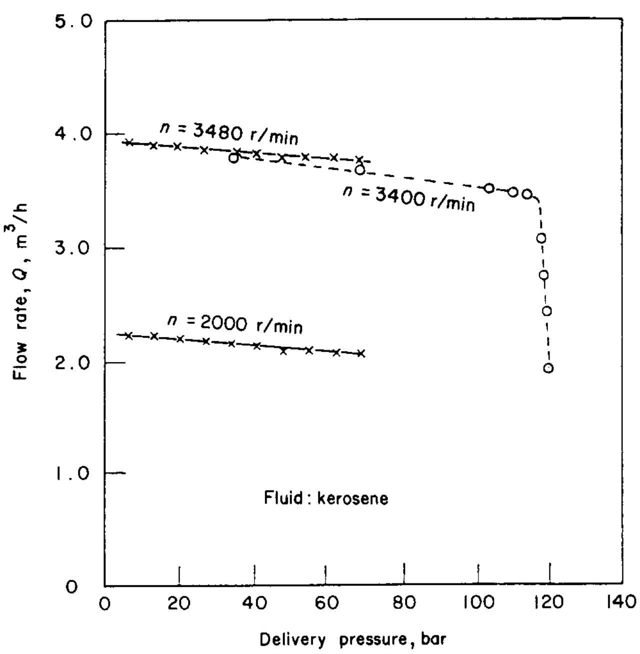

Fig 2 Pump calibration curve with kerosene.

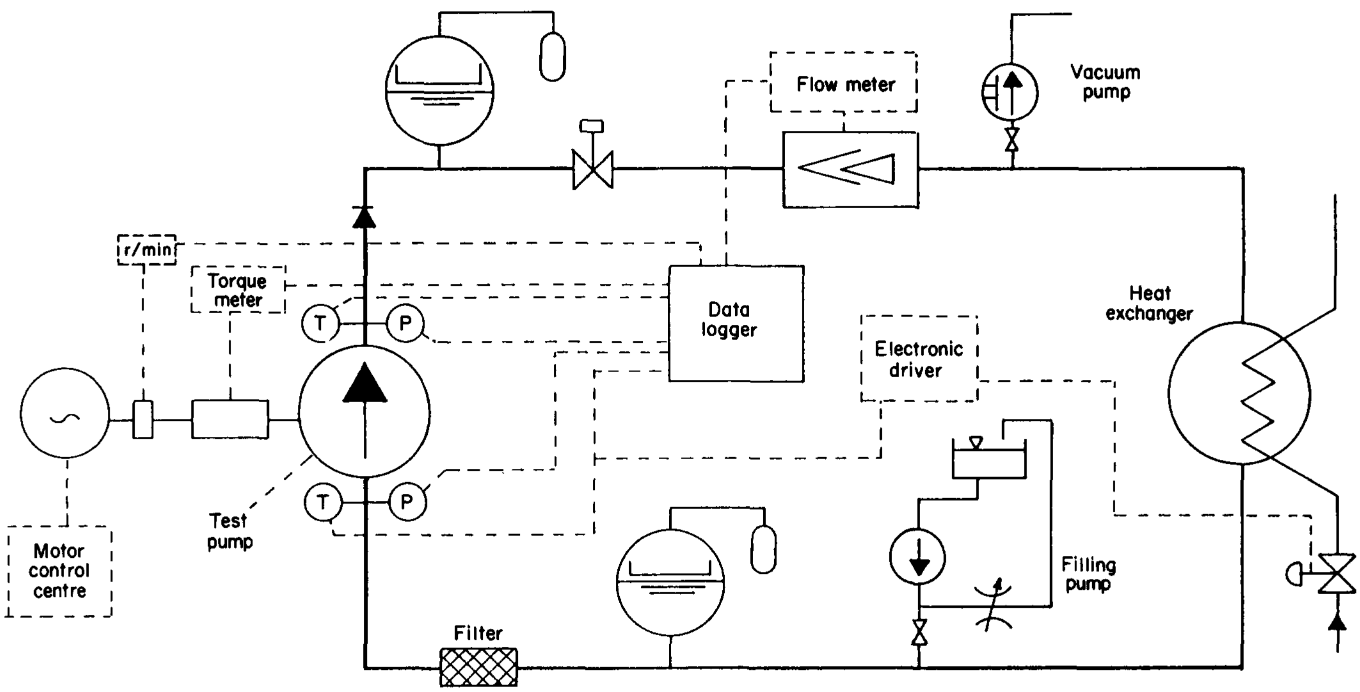

Fig 3 Test rig

turer provided for testing a special pump which incorporates a carbon sleeved rotor and silver plated nitralloy piston/slipper assemblies. Moreover, the pump overspeed governor was made inoperative by removing the diaphram tapped screw and the servo piston solenoid was removed and shorter studs fitted to the pump body to incorporate a standard servo cover plate. This pump was tested, by the manufacturer, using kerosene and performance curves given in Fig 2.

Experimental apparatus

A schematic view of the test rig is given in Fig 3. The apparatus is essentially an instrumented closed loop of small diameter stainless steel tubes. The main components of the circuit are the pump, the two accumulators on suction and delivery ducts, the filter, the Rotaflux flow meter, the heat exchanger with its motorised valve on the cooling fluid, the relief and the regulating valve.

The circuit has an auxiliary vacuum pump, used for venting the loop before filling and with a secondary auxiliary diaphram pump for feeding the circuit. The pump is driven by a variable speed electric motor; between the driven pulley and the pump shaft a brushless torque meter is mounted between two gear couplings. Rotational speed is measured by a digital tachometer.

The pumped fluid temperature is measured both at suction and delivery side of the test pump, close to its flanges, by two thermistor thermometers. The suction thermistor regulates the cooling fluid heat exchanger motor valve through an electronic device equipped with a feed-back signal, to maintain a ±0.5 °C constant flow temperature. Three LVDT pressure transducers are used to measure the pressure at the inlet and outlet test pump flanges and inside the pump casing close to the valve port. A minimum pressure switch mounted on the suction pipe and a maximum pressure switch mounted on the delivery pipe are connected to an alarm circuit to prevent casual cavitating conditions and over pressures.

All the measured parameters are monitored, recorded and then computer processed.

Test programme and method

The goals of the test programme were first:

- to verify the pump capability of working with high density, low viscosity and high vapour pressure organic fluids,
- to determine the head-flow characteristic,
- to evaluate the volumetric and overall efficiencies,
- to measure the npsh requirements of the pump working with these particular fluids at different temperature levels and at different rotational speeds;

and secondly:

- to compare the resulting pump performance with that measured with kerosene, a 'normal' fluid for the pump;
- to detect the influence of the most important parameters on pump performance.

Two commonly used refrigerants, R113 and R11, were chosen as test fluids among the organic fluids. Their physical properties are summarised in Table 1 with those of kerosene. Six rotational speeds were tested, from 1750 to 3000 r/min in 250 r/min steps. Experiments were performed at different fluid temperatures and at precisely 20, 40, 60, 80 °C for R113 and kerosene, and at 20, 40, 60 °C for R11.

During tests, for evaluation of the head-flow characteristic, different values of suction pressure were investigated to determine the possible influence of such a parameter on pump performance. Pressures of 1, 2 and 3 bar over the vapour pressure at the corresponding fluid test temperature were chosen in all the tests; in this manner an almost constant margin against cavitation was preserved at all test temperatures.

During testing to measure the npsh requirements, a constant pressure difference between delivery and suction side was maintained. Various pressure differences were tested. All the tests were performed at a constant camplate angle.

Experimental results: head-flow and efficiency performance

A preliminary important observation should be made: after more than 60 testing hours with kerosene, the pump operated for more than 350 hours with organic fluids showing a completely satisfactory behaviour. The modifications introduced in the pump body seem to be essential for correct pump running with low lubricity and low viscosity fluids. A previous 'all metal' version of the same pump tested in R113, had suffered severe damage after only 15 hours of testing.

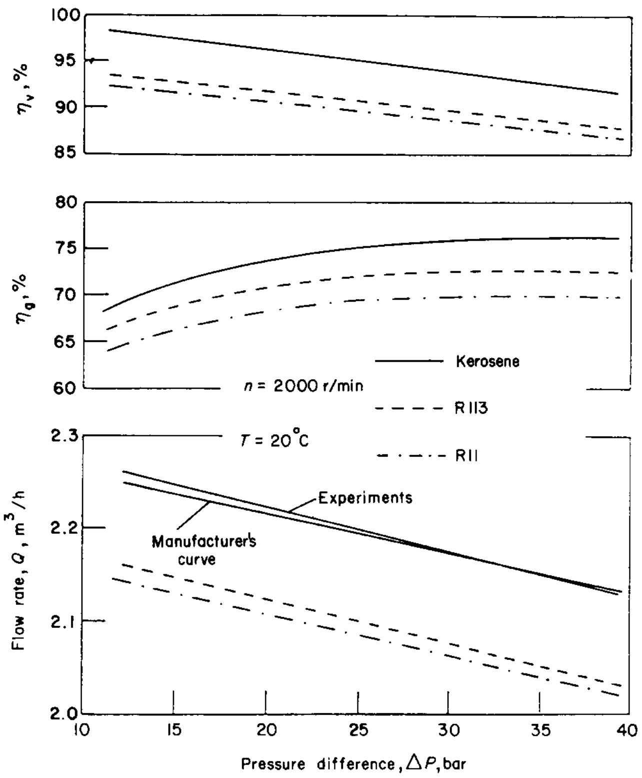

Fig 4 Head flow and efficiency curves at 2000 r/min for R11, R113 and kerosene at 20 °C

Since results of all the tests performed cannot be presented in detail, the influence of the main parameters on pump performance is presented separately in subsequent sections.

Performance with kerosene, R113 and R11

The experiments show a similar general pump behaviour working with organic fluids and with kerosene (Fig 4). A significant reduction in the flow delivery is recorded with R113 and R11, corresponding to a difference of about 5% in the volumetric

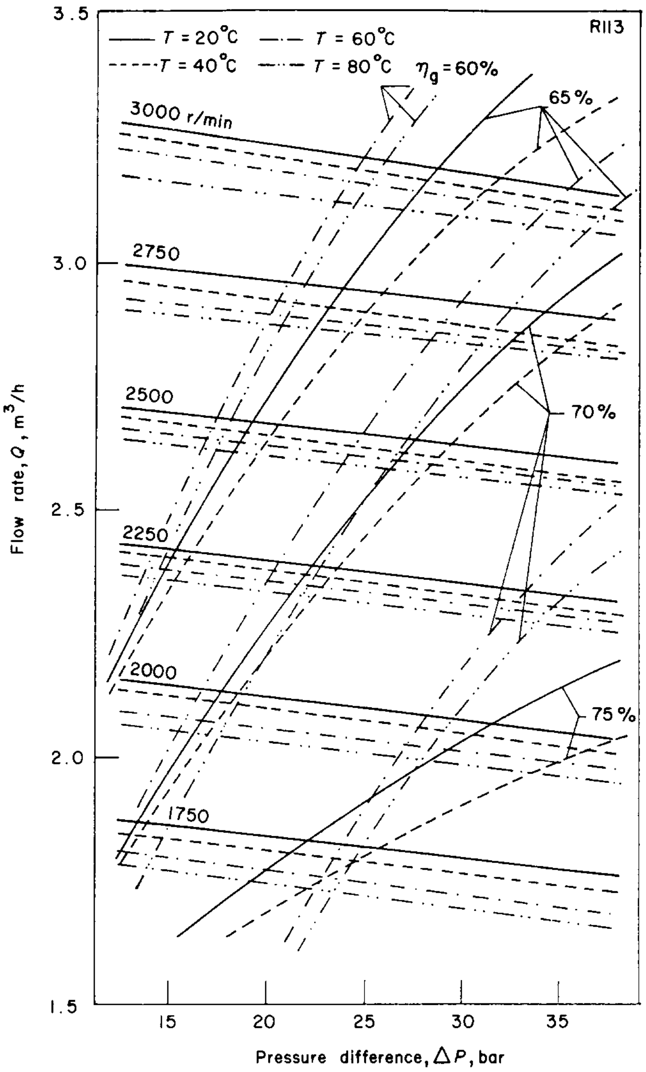

Fig 5 Pump map with R113 at 20, 40, 60, 80 °C

Table 1: Properties of the fluids tested

Temperature °C	Refrigerant R11			Refrigerant R113			Kerosene		
	γ, kg/m^3	p_v, bar	ν, 10^{-6} m^2/s	γ, kg/m^3	p_v, bar	ν, 10^{-6} m^2/s	γ, kg/m^3	p_v, bar	ν, 10^{-6} m^2/s
20	1486.63	0.88736	0.298	1576.46	0.364	0.458	790	0.0016	1.6
40	1439.25	1.7464	0.259	1529.06	0.783	0.365	775	0.0050	1.2
60	1388.64	3.13713	0.233	1479.02	1.513	0.300	761	0.0175	0.92
80	–	–	–	1426.31	2.684	0.254	746	0.04	0.75

efficiency. The differences between the two organic fluids are less significant.

In Fig 4, the pump calibration curve as obtained by the manufacturer is superimposed, showing good agreement: the maximum measured error is less than 0.5%. The overall performance maps of the pump working with R113 and R11 at different temperatures are given in Figs 5 and 6. Obviously, when no cavitation occurs, the boost pressure has no influence on the head-flow characteristic or on the efficiency curves.

Volumetric efficiency behaviour

Fig 7 shows the volumetric efficiency plotted against the rotational speed for different fluid temperatures and various pressure differences Δp. Note that:

* the volumetric efficiency decreases with increasing Δp.
* the volumetric efficiency increases with increasing rotational speed. This influence is more obvious at higher Δp.
* the volumetric efficiency decreases with increasing fluid temperature. In fact an increase in the fluid temperature produces a reduction in fluid viscosity with a consequent increase of flow leakage along the pistons.

Overall efficiency behaviour

In Figs 8 and 9 the pump overall efficiency is drawn in some particular cases as a function of the most important parameters: the speed, pressure difference and fluid temperature. Note that

* the overall efficiency increases with increasing pressure difference Δp. This behaviour can be

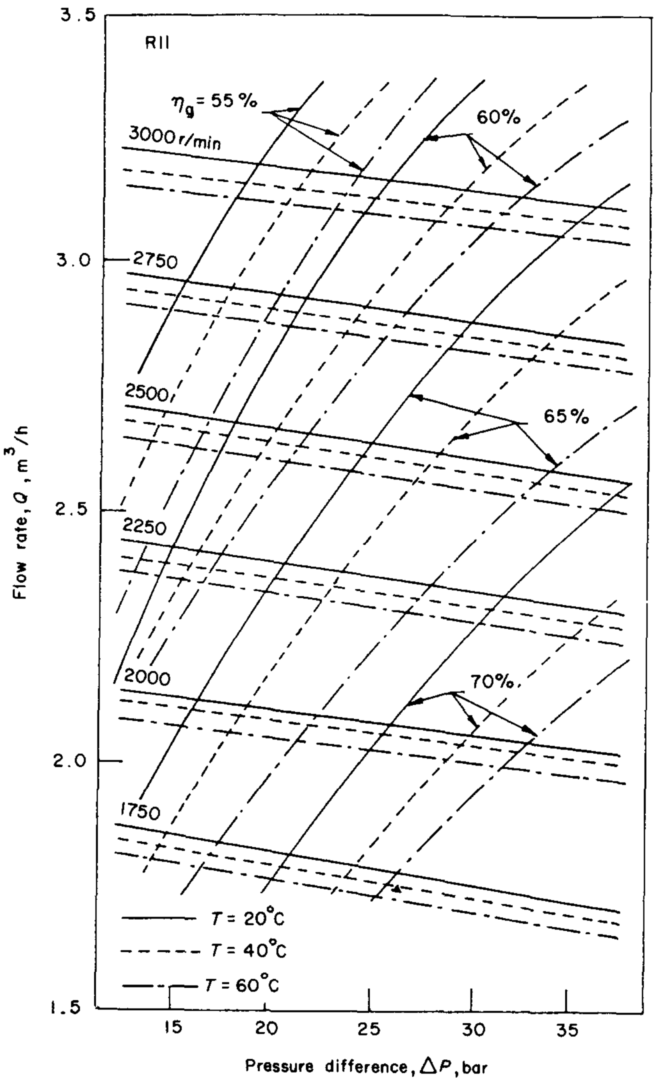

Fig 6 Pump map with R11 at 20, 40, 60 °C

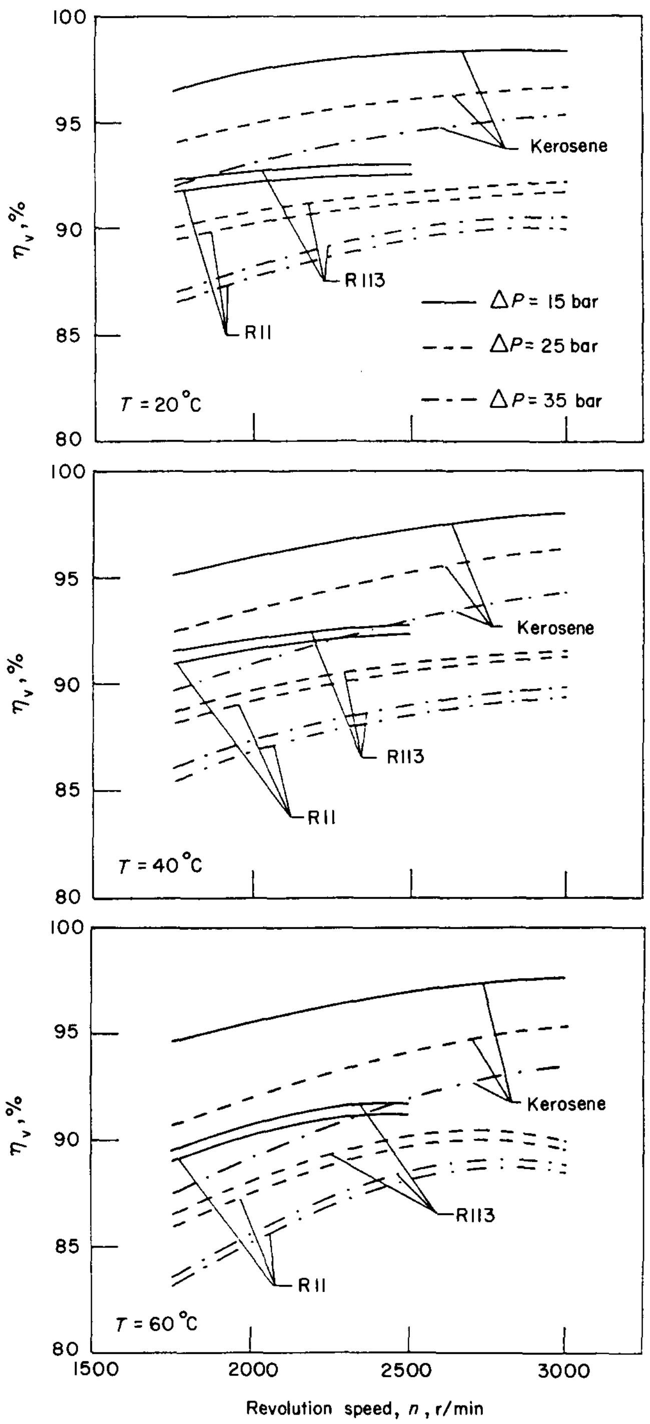

Fig 7 Pump volumetric efficiency at different speeds with R11, R113 and kerosene at various temperatures for different pressure difference

explained by considering the better lubrication that can be obtained between the piston slippers and the camplate due to the higher hydrodynamic fluid film pressure supported by the increased delivery pressure. This effect is obviously more important at low delivery pressures and practically disappears when complete hydrodynamic fluid film lubrication is obtained; moreover, it is much more significant at high speed because, in this case, the power absorption due to the friction losses is higher.

* the overall efficiency decreases with increasing speed because the lubrication conditions are worse and the friction losses and hence power absorption is higher. As noted above, this negative effect is enhanced at low Δp.
* the overall efficiency decreases with increasing fluid temperature (Fig 9). The fluid viscosity reduction, due to the temperature increase, is the main factor which affects the power absorption increase due to the higher friction losses resulting from worse lubrication conditions generated between all the mating sliding surfaces.

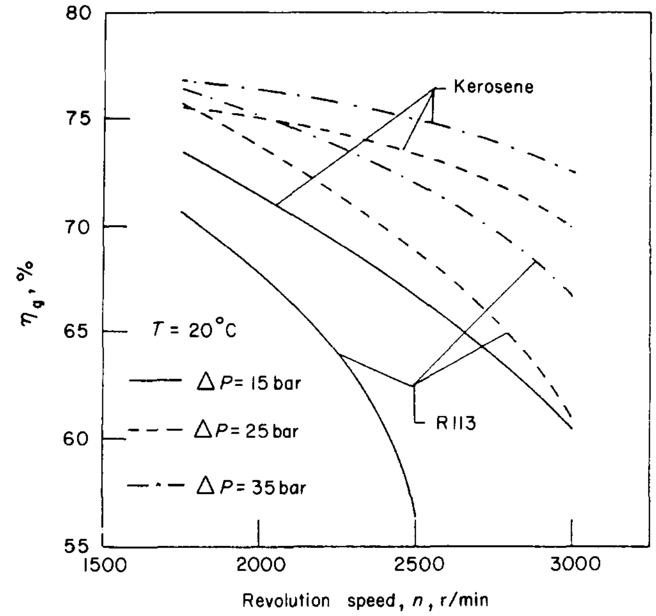

Fig 8 Pump overall efficiency as function of speed and delivery-suction pressure difference

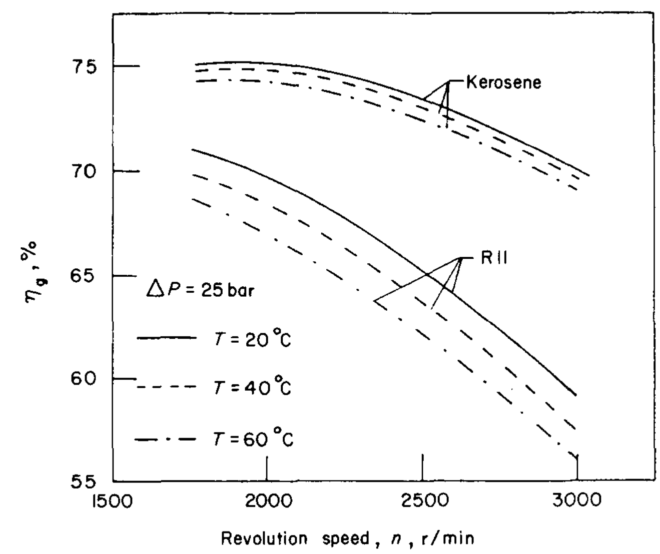

Fig 9 Pump overall efficiency as function of speed and fluid temperature

A correlation of the results of the experiments was attempted by assuming that the fluid viscosity is the main fluid physical property that influences the pump efficiency. Fig 10 supports that idea. It can be observed that both the volumetric and overall efficiencies are simply a function of the fluid kinematic viscosity; the scatter is within the experimental measurement errors.

Experimental results: npsh curves

It is known that axial piston pump cavitation performance is a function of the pressure drop at the cylinder port because, in order to maintain the force balance at the sliding surface between the cylinder block and the valve plate, the cross sectional area of the cylinder port is made smaller than the piston area. The pressure drop at the cylinder port arises from the pressure loss due to the axial flow velocity in the cylinder and the tangential velocity of cylinder port and to the pressure drop due to the centrifugal and inertia force[4,5]. The experimental results presented in this paper can be explained by considering the parameters affecting those pressure losses. The pump cavitation test results

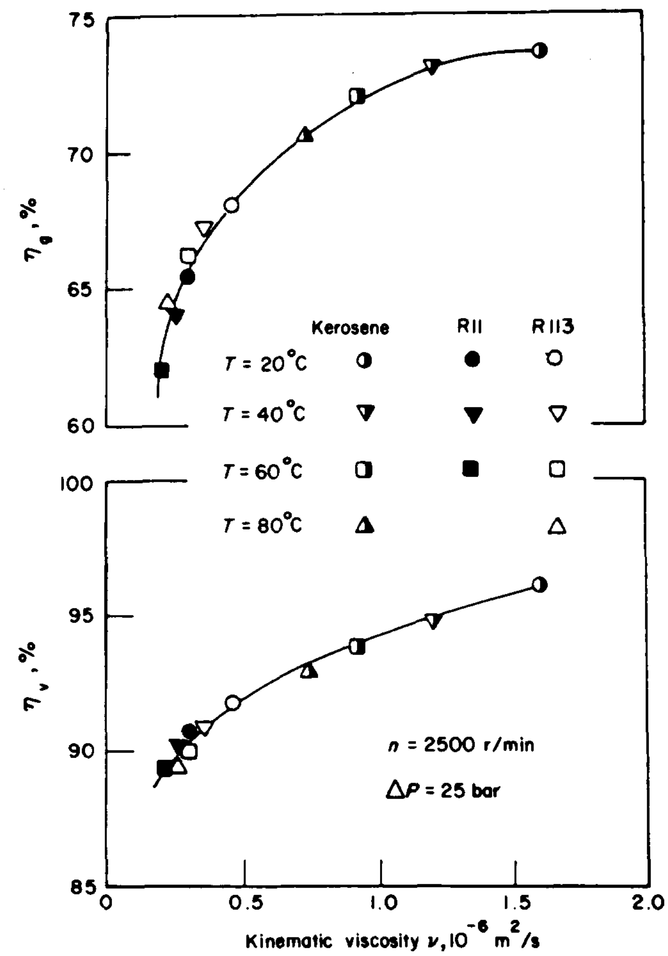

Fig. 10 Pump volumetric and overall efficiencies against kinematic viscosity

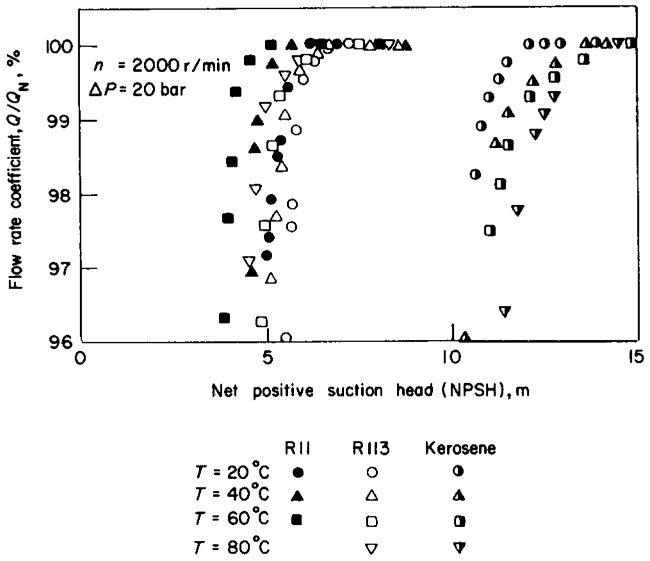

Fig 11 npsh values at 2000 r/min for R11, R113 and kerosene at various temperatures

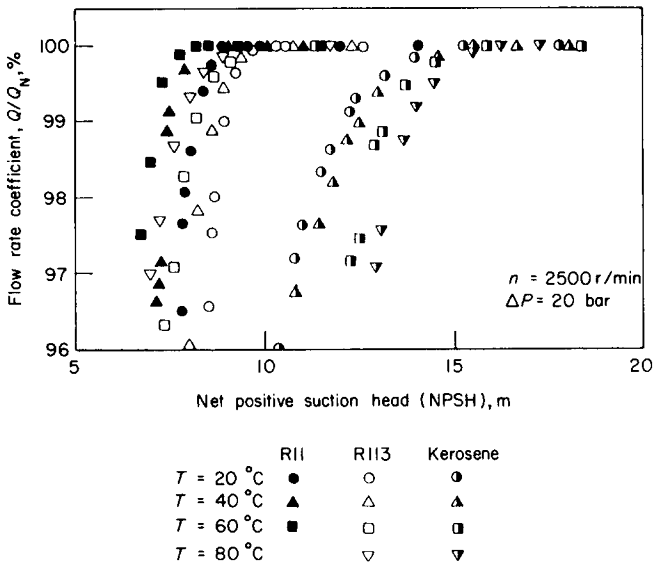

Fig 12 npsh values at 2500 r/min for R11, R113 and kerosene at various temperatures

are summarised in Figs 11–13 for various pump rotational speeds. The npsh values required by the pump at each working condition were always computed by considering only the fluid vapour pressure at fluid working temperature, disregarding the influence of dissolved gas pressure, which is, indeed, a value unknown and difficult to determine. This hypothesis is reasonable in the case of R113 and R11 due to their high vapour pressure. Moreover the closed circuit is completely degassed before loop filling and before every test.

From an analysis of Figs 11–13:

* an obvious influence of rotational speed on npsh values is recorded for each fluid tested. This expected trend is due to the valve plate tangential velocity increase which produces an increase in the pressure losses, as happens in a branched duct. Moreover, at constant camplate angle, any speed increase also produces an increase in the ideal flow rate with an increase in the cylinder axial flow velocity and subsequent higher pressure losses through the cylinder port. This speed influence is much more pronounced in the case of R11 and R113, where the npsh values were found to be an approximately quadratic function of the rotational speed.
* the different fluid physical properties affect the npsh values quite strongly. In particular, it can be seen that at low speed, the pump required npsh in R11 is almost half of that in kerosene; the npsh of the pump working in R113 is then closer to the value of R11 than to kerosene. The large differences in the density, viscosity and vapour pressure of the three fluids are the main causes of such great discrepancies (Table 1). It is very difficult to recognize separately and determine theoretically the influences of these three parameters because they interact with each other.
* the fluid temperature influence, always detectable, is rather discordant. In the case of R11 and R113,

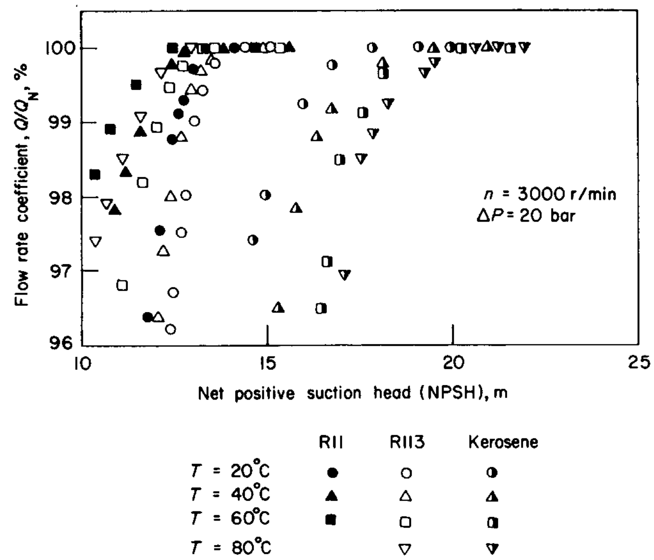

Fig 13 npsh values at 3000 r/min for R11, R113 and kerosene at various temperatures

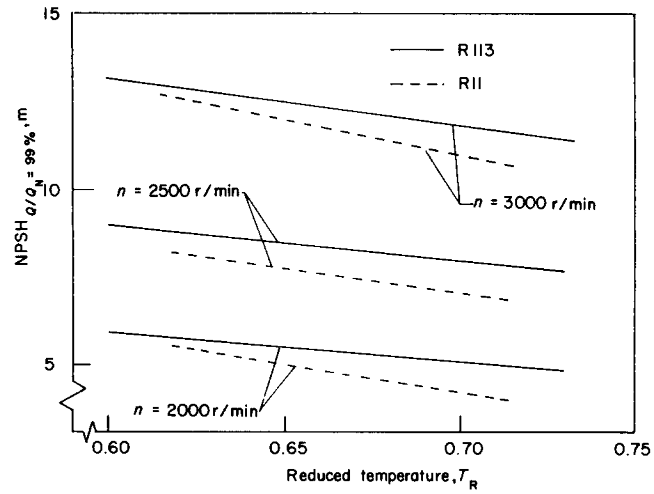

Fig 14 npsh values at $Q/Q_N = 99\%$ *against fluid reduced temperature* a temperature increase reduces the npsh as expected from the thermodynamic theory of cavitation. The attempt to correlate the npsh curves with a single thermodynamic parameter (reduced temperature in Fig 14 or other more complex formulations, as $K_{c,min}$ suggested for turbomachines[6]) was not successful. The opposite trend found with kerosene could be due to the influence of the dissolved gas pressure.

Concluding remarks

The tests performed have proved that the pump is capable of working with organic fluids at an efficiency which is still quite high, despite of the low viscosity and high density of these working fluids. Moreover the overall efficiency is generally higher than that obtainable with centrifugal pumps operating in the same headflow range. The kinematic viscosity is the main fluid physical property that influences the pump efficiency.

The cavitation tests demonstrate the presence of a thermodynamic effect. A simple function of npsh with respect to the reduced temperature is not sufficient to explain the phenomenon and a more complex function should therefore be sought.

References

1. *Proceedings of 'Conference on Organic Fluids for Waste Heat Recovery in Ships and Industry'. Trans. Inst. Mar. Engrs. 93–1981.*

2. **Angelino G., Gaia M., Macchi E., Barutti A., Macciò C., and Tomei G.** Test Results of a Medium Temperature Solar Engine *International Journal of Ambient Energy, 1982,* **3**, *3 pp 115–126*

3. **Macchi E.** Thermodynamic Cycles for Low Temperature Heat Sources – *Lecture Series 100 on 'Close-Cycle Gas Turbines' Von Karman Institute for Fluid Dynamics, Bruxelles, 1977*

4. **Ibuki T., Hibi A., Ichikava T. and Yokote H.,** Suction Performance of Axial Piston Pump, 1st report: Analysis and Fundamental Experiments. *Bulletin of JSME,* **20**, *139, January 1977*

5. **Ibuki T., Hibi A., Ichikava T. and Yokote H.** 'Suction Performance of Axial Piston Pump', 2nd report: Experimental Results. *Bulletin of JSME,* **20**, *145, July 1977*

6. **Ruggeri R. S. and Moore R. D.** 'Method for Prediction of Pump Cavitation Performance for Various Liquids, Liquid Temperatures, and Rotative Speeds'. *NASA Technical note. TN D-5292 – June 1969*