Theoretical analysis of utilization of helical screw expanders

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Summary

The authors analyse the criteria for the possible use of the screw expander within the context of organic Rankine cycle systems. After a brief description of the screw expander, they justify the choice of optimization parameters and give an example of its use. They then present the criteria for the posssible use of this type of expander. The adaptation diagram is worked out for six different organic fluids Freon 11, 113 and 114, n-butane, toluene and perfluorohexane (Flutec PP1). This tool enables to establish a priori the possibility of using the screw expander for an organic fluid of a given nature as a function of the characteristics of the hot well, a step which has hitherto remained rather intuitive.

1. INTRODUCTION

The usefulness of power units which produce mechanical or electrical energy, which are based on Rankine cycle and which use as working fluid an organic fluid of high molecular mass, is no longer disputed. To appreciate this, one only has to consider the many applications which have already been developed throughout the world, in such different fields as petrochemicals and geothermal energy.

This study is the result of a wider research project which aimed to establish the criteria for choosing the main elements of organic Rankine cycle systems, using a thermodynamic modelling of the system as a whole. This study led us to consider the thermodynamic cycle, the working fluid and the prime mover. A comparison of the results of the model thus obtained with the characteristics of existing industrial installations enabled us to verify the latters'validity.

2. THE SCREW EXPANDER

The screw expander is based on an operating principle similar to that of the screw compressor developed at the beginning of the 1930's by the Swede A.Lysholm. There are very few conceptual differences between these two types of machine - except for the fact that the direction in which the rotors rotate and the fluid flows out are reversed.

The built in volume ratio ν_i, defined by the ratio between the specific volume of the fluid at the exit and entrance to the machine :

$$\nu_i = \frac{v_s}{v_e} , \tag{1}$$

is determined by the geometric characteristics of the machine and thus constitutes an invariant for the latter.

The built in expansion ratio π_i, defined as the ratio between pressure at the entrance and exit to the machine :

$$\pi_i = \frac{p_e}{p_s} \quad , \tag{2}$$

can easily be deduced from it, if one considers that the expansion is isentropic

$$\pi_i = \nu_i^{\gamma} \quad . \tag{3}$$

It depends on both the characteristics of the machine and the nature of the fluid. The internal expansion ratio has no upper limit, providing the pressure involved are compatible with the mechanical limitations of the machine (values of 20 bar, and exceptionally 40 bar can be achieved). The most common expansion ratios lie between 1.5 and 5.

The theoretical volume flow rate, expressed as m³/s and calculated at the entrance of a given geometric machine, can be assessed using the following formula :

$$\dot{V}_{th} = V_{CT} \; z_1 \; \frac{n_1}{60} \quad , \tag{4}$$

in which z_1 denotes the number of lobes, n_1 the rotational speed of the male rotor expressed in rpm and V_{CT} the volume of a working unit at the beginning of expansion, expressed in m³. One can show [1] that this formula can also be written as follows :

$$\dot{V}_{th} = C \; C' \; \frac{\ell}{d_1} \; d_1^3 \; \frac{n_1}{60} \quad , \tag{5}$$

in which C denotes a corrective coefficient expressing the ratio between the actual working volume and that corresponding to the theoretical maximum, C' a coefficient dependent on the geometry of the rotors, ℓ the length of the rotors and d_1 the diameter of the male rotor.

3. CALCULATION OF OPTIMIZATION PARAMETERS

The basic geometric parameter used for sizing the machine is the diameter of the rotors. It can be assessed using equation (5) in which the male rotor rotational speed can be expressed as a function of this same diameter, as follows :

$$n_1 = \frac{60 \; u_1}{\pi \; d_1} \quad , \tag{6}$$

in which u_1 denotes the peripheral speed at the tip of the lobe, which also constitutes a basic characteristic of the machine.

To the extent that we only take into consideration those rotors which have 4 lobes and 6 flutes, the product of coefficients C and C' introduced into equation (5) will be expressed as :

$$d_1 = \sqrt{\frac{\pi \dot{V}_{th}}{0.5\ u_1(\ell/d_1)}} \quad . \tag{7}$$

If we regard the diameters of the two rotors as identical, subscript 1 becomes superfluous and this common diameter can be denoted by d.

In relation (7), the volume flow rate is fixed by the characteristics of the Rankine cycle, while values u_1 and ℓ/d constitute optimization parameters which have to be optimally chosen in order to maximize the machine efficiency. We studied the sensitivity of these parameters. The curves in figures 1 and 2 show the evolution of the rotors' diameter and the male rotor rotational speed as a function of the theoretical volume flow rate at the entrance to the machine, for different values of the peripheral speed u_1 and of the ratio ℓ/d.

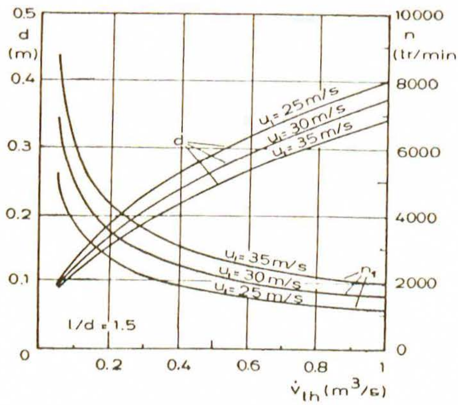

Fig.1

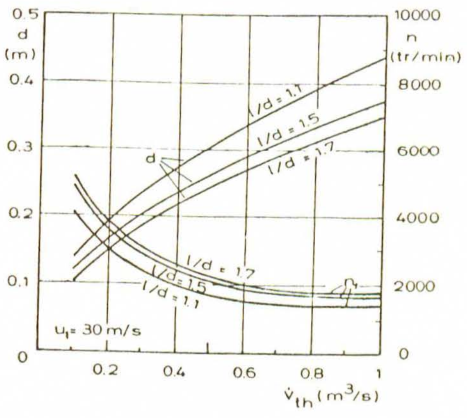

Fig.2

4. APPLICATION TO ORC INSTALLATIONS

Unlike axial turbines and to a lesser extent radial turbines, there are very few predictive efficiency correlations of a screw expander. We have found no reliable formulation which would enable us to perform a predictive efficiency calculation on the basis of the machine's geometric or operating characteristics. Nevertheless, several manufacturers and research institutes have carried out tests with air, steam or Freon. Unfortunately, these studies are only seldomly exhaustive, and can therefore not produce reliable correlations. In addition, the results of these tests are not generally valid beyond the machine or type of machine studied. A recent study [2] offers one of the rare efficiency curves, which one would be tempted to describe as "universal" and which gives the efficiency of a screw expander as a function of its specific speed.

We should nevertheless stress that there are no bibliographic references which would enable one to check the validity of the model proposed. However, a comparison with the characteristics of various existing installations has proved sufficiently valid for us to be able to accept this correlation.

We believe it would be of great interest to perform tests whose aim would be to establish correlations which would permit a predictive efficiency assessment of a screw expander using on the one hand air, and on the other hand superheated vapour (steam or high molecular mass organic fluid vapour).

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********** CARACTERISTIQUES DU CYCLE OPTIMISE ******************************

        1          2          2STAR       2PRIM        3          4

 T    30.0       30.9       30.9        52.4        52.4       37.7    deg.Cel
 P     2.88       5.37       5.37        5.37        5.37       2.88   bar
 H   276.3      277.2      277.2       338.2       671.4      650.2    kJ/kg
 S     1.262                              1.456       2.479      2.495 kJ/kg K
 V                                                   76.360    143.354 dm3/kg

********** PERFORMANCES DU SYSTEME O R C ***********************************

EFFLUENT CHAUD
**************
Debit-masse        2.00 KG/S
Chaleur massique moyenne : 4.20 kJ/kg K
Temperature limite de refroidissement : 20.0 deg.Cel.

FLUIDE MOTEUR
*************
Type : BUTA
Debit-masse :    0.558 kg/s

GENERATEUR DE VAPEUR
********************
Temperature d entree des effluents chauds : 85.0 deg.Cel
Temperature de sortie des effluents chauds : 58.3 deg.Cel.
Puissance thermique extraite de la source chaude :     224.6 kW
Pincement :10.0 deg.Cel.
Rendement des echangeurs :0.98

CONDENSEUR
**********
Temperature d entree de la source froide : 20.0 deg.Cel
Temperature de sortie de la source froide : 20.0 deg.Cel.
Debit de la source froide : -
Chaleur massique moyenne de la source froide : -
Temperature d entree du fluide moteur au condenseur : 37.7 deg.Cel.
Temperature de condensation du fluide moteur : 30.0 deg.Cel.
Pincement :10.0 deg.Cel.

POMPE
*****
Relevement d enthalpie :     0.9 kJ/kg
Rendement interne :0.50

PERFORMANCES GLOBALES
*********************
Rendement thermique du cycle :0.052
Efficacite du generateur de vapeur :0.494
Rendement global du cycle :0.026
Puissance motrice :      11.4 kW


******* CARACTERISTIQUES DE LA MACHINE DE DETENTE A VIS  *****************

GEOMETRIE:             l=0.100 m
*********              d=0.084 m
                       l/d=1.2
                       n1=  13712   tr/min
                       u1=60.  m/s
                       s1=0.120 m
                       a=0.066 m
                       ns=0.23

RENDEMENT:             ETAS1=0.815
*********
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Table I

Let us now examine a low-power electrical energy producing unit using as heat source a heat carrier fluid which recovers heat of gas effluents from a chemicals industry. Let us assume the hot fluid inlet temperature to the vapour generator to be 85°C and the temperature of the heat sink to be 20°C. If we consider a heat source mass flow rate of 2 kg/s, we reach a net power output of about 11 kW, as shown in table 1, which gives the main characteristics of an installation functioning on n-butane. These characteristics are derived from a systematic optimization performed in the document referred to under [3].

5. SCREW EXPANDER UTILIZATION DOMAINS

5.1. Utilization limits

Unlike expansion turbomachines, there are no limits to the use of the screw expander which are intrinsically linked to its enthalpy drop. There are, however, limits which are indirectly dependent of this enthalpy drop. Moreover, the mass flow rate of the fluid which expands through the machine has a considerable influence on its geometric characteristics.

The main limit to the use of the single stage screw expander lies in the ratio between the volume flow rate at the exit and at the entrance of the machine. This ratio cannot exceed a particular value. The geometric dimensions of the machine have to permit the three basic phases of its operation, i.e. the inlet of the high pressure fluid, its expansion and its outlet to the outside. This ratio between the volume flow rate at the exit and at the entrance to the machine is equal to the built in volume ratio and can be expressed as a function of the built in expansion ratio, providing the nature of the fluid is known and one acknowledges that the expansion is isentropic.

For a fluid of a given nature, which expands up to a given condensation pressure, to limit the built in volume ratio also leads to limiting the enthalpy drop, which can however only be determined if the prime mover's efficiency is known. A study of the characteristics of different machines made by various manufacturers led us to adopt a maximum value of 6 for the built in volume ratio ($\nu_i = 6$).

We have studied the influence of the increase in built in volume ratio on the fluid's exit velocity, and to this end we have established a correlation in order to calculate the area of the exit section of the machine on the basis of its rotor diameter. This correlation is derived from data on the complete range of machines of one manufacturer [4]. We observed that for the different fluids under consideration and for a built in volume ratio which does not exceed 6, the Mach numbers at the exit to the machine remained moderate.

A second limit to the use of the screw expander is related to the mass flow rate of the working fluid. For a given temperature of dry saturated vapour at the entrance to the machine, an increase in mass flow rate leads to an increase in the diameters of the rotors and a reduction in their rotational speed (see figures 1 and 2). As is common practice, we shall restrict the speed at which the male rotor rotates to the following range :

$$1500 \text{ rpm} \leq n_i \leq 20000 \text{ rpm} .$$

We feel that the rotor diameters which it is currently possible to construct lie between the following extremes :

$$0{,}04 \text{ m} \leq d \leq 0{,}80 \text{ m} .$$

We have also retained the following range of variations in the optimization parameters :

$$20 \text{ m/s} \leq u_1 \leq 70 \text{ m/s}$$
$$1.0 \leq \frac{\ell}{d} \leq 1.7 .$$

Finally, we fixed a lower limit to the efficiency below which we acknowledge that the screw expander does not suit :

$$\eta_T = 0.70 .$$

5.2. Adaptation diagrams

Using the modelling programme described in [3], we systematically calculated a series of installations and could thus show the limits to the use of the single stage screw expander in a diagram, for different working fluids, the enthalpy drop on the y-axis and the mass flow rate on the x-axis (figure 3). The organic fluids we considered are Freon 11, 113 and 114, n-butane, toluene and perfluorohexane (flutec PP1). A programmed thermodynamic vapour table, whose structure is independent from the nature of the working fluid, was developed to this end [5].

The condensation temperature was fixed at 30°C and the rotational speed was not set. The curves presented are therefore envelope curves for rotational speeds lying within the interval specified above. Curves of equal power are also plotted on Figure 3.

We should point out that the prime mover will be suited providing its geometric and aerothermodynamic characteristics correspond to those defined. The fact that the machine is said unsuitable does not necessarily mean that it is unbuildable, but rather that its characteristics depart from the norms we regard as standard.

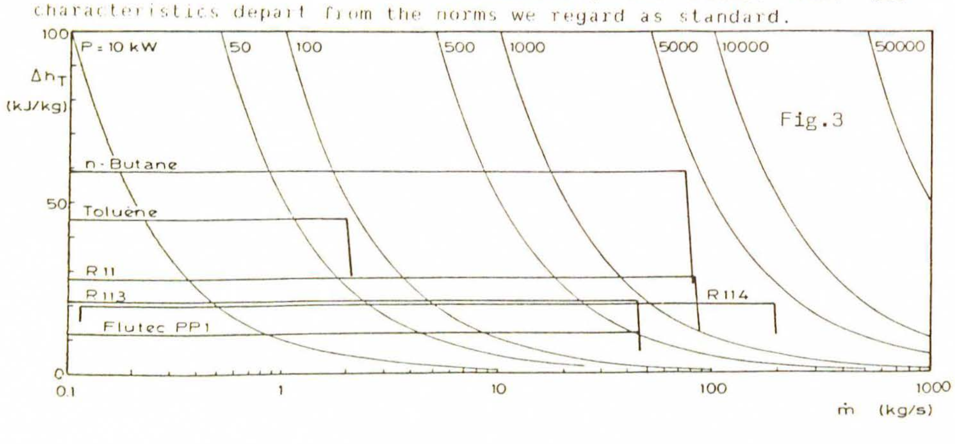


Fig.3

The shape of the boundary lines in figure 3 can be explained as follows. The mass flow rate lower limit is determined by an increase in rotational speed above the limit we settled on. Similarly, the mass flow rate upper limit is determined by the decrease in rotational speed below the limit set. Nevertheless, in certain cases, the increase in rotor diameter, a consequence of the increase in the mass flow rate, can make the latter go beyond the upper limit set before the rotational speed gets too low. For increasing enthalpy drops and a fixed mass flow rate, the volume flow rate increases. Using relation (7), we can infer that on the one hand, the rotor diameters also increases for a fixed ratio ℓ/d and a fixed peripheral speed u_1 and that on the other hand, the rotational speed falls (see figures 1 and 3). As a consequence, the mass flow rate upper limit decreases. This explains the shape of the right-hand ends of the curves in figure 3.

The upper limit to the enthalpy drop is, in turn, due to the maximum value of the volume built in ratio. As the latter is independent from the mass flow rate of the working fluid, this boundary line will be parallel to the x-axis.

If we take a look at figure 3, we notice first of all that the lower limit to the mass flow rate is, for all fluids except R114, outside the range of flow rates we settled on. On the other hand, the working range of the expanders using the three Freon considered are quite close to oneanother. However, expanders working on toluene only permit very low mass flow rates but higher enthalpy drops. N-butane, as for it, permits the use of a screw expander within a wide range of mass flow rate and enthalpy drops.

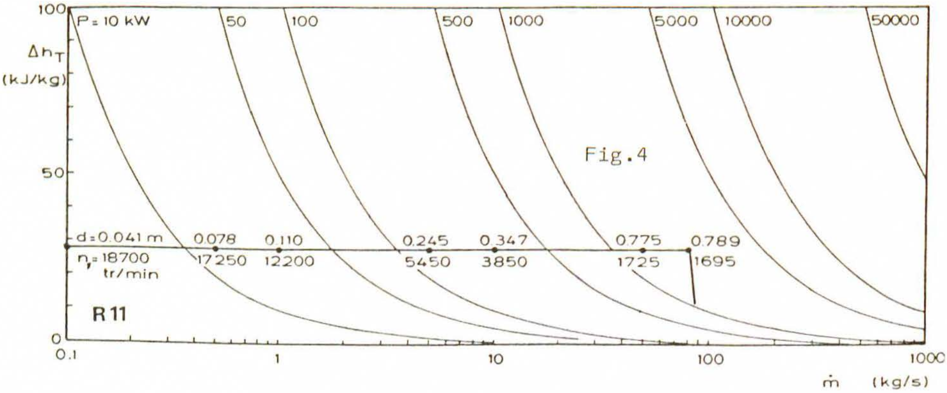
Figure 4, drawn up for Freon 11, shows the evolution of the rotor diameter and the rotational speed of the male rotor on those machines whose characteristics lie on the upper boundary line in figure 3.

In Table II we give the maximum evaporation temperatures of Rankine cycles permitting the use of the screw expander for the various fluids under consideration. As the condensation temperature was fixed at 30°C, the maximum temperatures given below can be likened to the maximum enthalpy drops Δh_T given in figure 3.

Note here that these values are given by way of example. The maximum permitted enthalpy drop for a given fluid can in fact vary as a function of the mass flow rate.

R11	R113	R114	Toluene	n-butane	Flutec PP1
97°C	88°C	96°C	74°C	100°C	80°C

Table II (t_c = 30°C)



6. CONCLUSIONS

The adaptation diagrams worked out for the different fluids enable us to guide the choice of prime mover and in particular the choice of screw expander. It is thus possible to establish a priori the possibility of using a screw expander in a given case. Until now, this step has remained a fairly intuitive one. In fact, manufacturers are currently far from unanimous about the choice of type of prime mover suited to an ORC installation. The fact that they know one machine better than another is often the only justification for their choice.

The modelisation we have carried out enables us to include the type of prime mover in the set of characteristics considered in the conceptual choices of an installation. Finally, let us add that this study has enabled us to confirm the promising future which awaits the use of low-power screw expanders in ORC installations.

ACKNOWLEDGEMENTS

The authors would like to thank Atlas Copco Airpower for their help in carrying out this study.

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List of symbols

d	rotor diameter
h	specific enthalpy
Δh	specific enthalpy drop
ℓ	rotor length
$\dot{m}$	mass flow rate
n	rotor rotational speed
n_s	specific speed
p	pressure
P	power
s	specific entropy
t	temperature °C
u	peripheral speed
v	specific volume
$\dot{V}$	volume flow rate
z	number of lobes
γ	isentropic exponent
ν	built in volume ratio
π	built in expansion ratio
η	efficiency

Subscripts

1	relating to male rotor
2	relating to female rotor
c	on condensation
e	relating to the entrance to the machine
s	relating to the exit to the machine
i	built in
T	relating to the prime mover
th	theoretical