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"Alternative Heat-Power Processes"
(AHP)-Processes
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Organic Rankine Cycle (ORC)

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A review of Italian activity in the field of Organic Rankine Cycles

G. Angelino, M. Gaia and **E. Macchi,** Milano/Italy

Summary

In the recent past, the authors have been involved in the development of a number of Organic Rankine Cycle engines, designed for a variety of heat sorces (solar energy, geothermal fluids, industrial waste heat), within a wide range of design power outputs (from 3 to 500 kW_{el}) and maximum operating temperatures (from 70 to 340°C).
A brief but comprehensive review of this activity which has been financed either by Italian University, Research Institutes and private Companies or by the Commission of the European Communities, is presented in the paper. In particular, the adopted criteria for the working fluid selection and the thermodynamic cycle optimization and the procedure followed in the design of the most critical plant components are discussed.

1 Introduction

Purpose of this paper is to present a review of the authors' activity in the design, construction and testing of Organic Rankine Cycles (ORC). This activity was financed by various Institutions (Italian University, Italian National Research Council (CNR), Commission of the European Communities (CEE), National Council for Alternative Energies (ENEA)), or by private Companies (Ansaldo, Belgonucleaire, Turboden). The engines described in the paper (see Tables 1 and 2) do not completely cover the Italian realizations in this field, since some prototypes were developed by Fiat /22/ and by other researchers, e.g. /23/.
The main engines'features are summarized in Tables 1 and 2, while an idea of the aspect, mechanical arrangement and dimensions of some representative units can be drawn by the photographs in Fig. 1 and 2. It can be seen that the engines were designed for a variety of heat sources (solar energy, geothermal fluids, industrial waste heat, fossil fuels), within a wide range of design power outputs (from 3 to 500 kW_{el})

ENGINE No.	DESIGN POWER (kW)	REFERENCES	YEAR	HEAT SOURCE	MANUFACTURER	WORKING FLUID	MAXIMUM CYCLE PARAMETERS T(°C)	P(kPa)	MINIMUM CYCLE PARAMETERS T(°C)	P(kPa)
1	4	/1/	1976	FLATE PLATE SOLAR COLLECTORS	G (1)	C_2Cl_4 (TETRACHLORO ETHYLENE)	75	23.5	30	3.1
2	4	/1,2/	1977	FLATE PLATE SOLAR COLLECTORS	G (1)	"	75	23.5	30	3.1
3	3	/3/	1977	FRESNEL LENS SOLAR COLLECTORS	G (1)	"	75	23.5	40	3.1
4	3	/20,21/	1978	GEOTHERMAL WATER	T (2)	"	75	23.5	40	5.1
5	3	/20,21/	1978	LOW TEMPERATURE PARABOLIC TROUGHS	T (2)	"	75	23.5	40	5.1
6	40	/4,5,6,7/	1979	EXHAUST GASES FROM A CERAMIC KILN	G (1)	"	110	72.4	40	5.1
7	50	/8,9,3/	1980	GEOTHERMAL WATER	G (1)	$CHCl_3$ (TRICHLOROETHYLENE)	70	60.5	40	22.0
8	45	/10,11,12/	1980	HIGH TEMPERATURE PARABOLIC TROUGHS	G (1)	C_8F_{16} (FLUTEC-PP5)	280	1160	40	10.0
9	25	/13,14/	1981	FOSSIL FUEL	G (1)	$C_{10}F_{18}$ (FLUTEC-PP3)	340	980	45	2.8
10	8	/15/	1982	LINEAR SOLAR COLLECTORS	G (1)	C_6H_5Cl (MONOCHLOROBENZENE)	177	300	33	2.5
11	500	/3,16/	1982	GEOTHERMAL FLUID	F.T.(3)	$C_2Cl_2F_4$ (R-114)	105	1530	40	3400
12	12	/17/	1982	FLATE PLATE SOLAR COLLECTORS	T (2)	C_2Cl_4 (TETRACHLORO ETHYLENE)	83	30.6	30	3.2
13	100	/18,19/	1984	EXHAUST GASES	T (2)	$C_6H_4Cl_2$ (DICHLOROBENZENE)	173	87.2	80	3.5
14	100	/19/	1984	EXHAUST GASES	T (2)	C_8H_{10} (M-XILOLO)	130	77.0	45	3.3

(1) Gemmindustria S.n.c., Milan, Italy - (2) Turboden S.r.l., Milan, Italy - (3) Franco Tosi S.p.A., Legnano, Italy

TAB. 1 Characteristics of developed ORC engines

ENGINE No.	NUMBER OF STAGES	R P M	MEAN RADIUS mm	FEED PUMP	PECULIAR FEATURES
1	1	13800	110	CENTRIFUGAL	VEE-BELT SPEED REDUCTION IN THE CONDENSER PLENUM; GREASE LUBRICATED BALL BEARINGS.
2	1	13800	110	GRAVITY FEED	MECHANICAL SOLUTIONS AS IN ENGINE No. 1.
3	1	12000	110	GRAVITY FEED	HIGH SPEED MECHANICAL SEAL; OIL LUBRICATED BALL BEARINGS.
4	1	12000	110	CENTRIFUGAL	AIR COOLED; HIGH SPEED GENERATOR; GREASE LUBRICATED BALL BEARINGS.
5	1	12000	110	CENTRIFUGAL	AIR COOLED; SAME MECHANICAL SOLUTIONS AS IN ENGINE No. 3.
6	1	6700	240	CENTRIFUGAL	HIGH SPEED MECHANICAL SEAL AND GEAR SPEED REDUCER TO SYNCHRONOUS ALTERNATOR.
7	1	6500	200	CENTRIFUGAL	INSTALLED IN A CONTAINER; MECHANICAL SOLUTIONS AS IN ENGINE No. 3.
8	4	6700	120	VOLUMETRIC	INSTALLED IN A CONTAINER; LARGE REGENERATOR; TILTING PAD OIL BEARINGS.
9	3	8500	130	VOLUMETRIC	HIGH SPEED MECHANICAL SEAL; REFRIGERATED SYSTEM FOR WORKING FLUID RECOVERY.
10	2	24000	90	VOLUMETRIC	HIGH SPEED MECHANICAL SEAL; OIL LUBRICATED BALL BEARINGS.
11	1	7500	160	CENTRIFUGAL	MAGNETIC DRIVE ELECTROPUMPS; OIL AND REFRIGERANT LOOPS WITH FLUID RECOVERY.
12	1	8200	200	GRAVITY FEED	MECHANICAL SOLUTION AS IN ENGINE No. 3; HIGH EFFICIENCY SYNCHRONOUS GENERATOR.
13	2	3030	430	CENTRIFUGAL	DIRECTLY COUPLED TO ASYNCHRONOUS GENERATOR; MECHANICAL SEAL.
14	2	3030	430	CENTRIFUGAL	SAME TURBOGENERATOR OF ENGINE No. 13 OPERATING WITH ANOTHER WORKING FLUID AND HEAT SOURCE.

TAB. 2 Characteristics of developed ORC engines

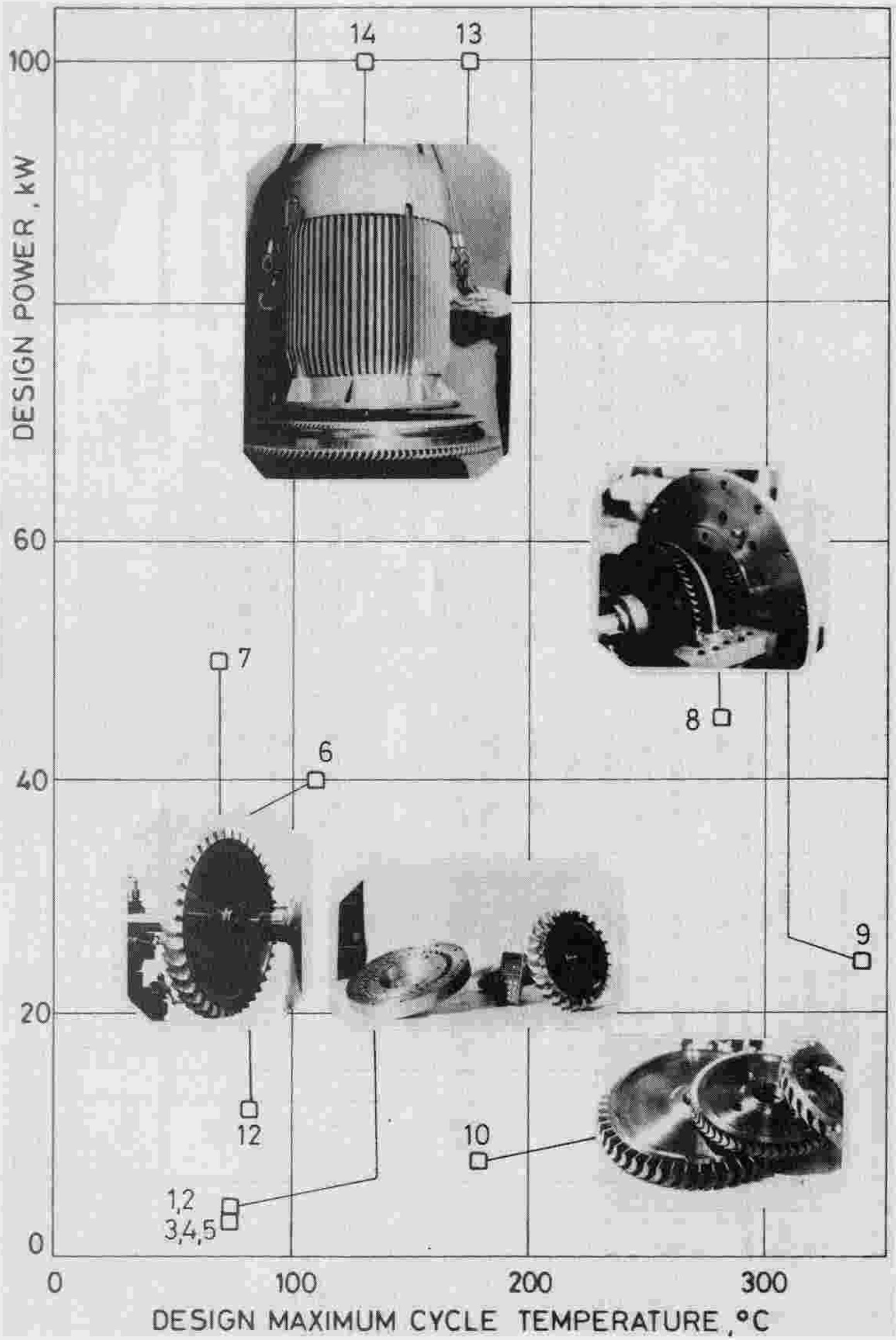


FIG. 1 Output power vs. maximum cycle temperature for the engines of TAB. 1.

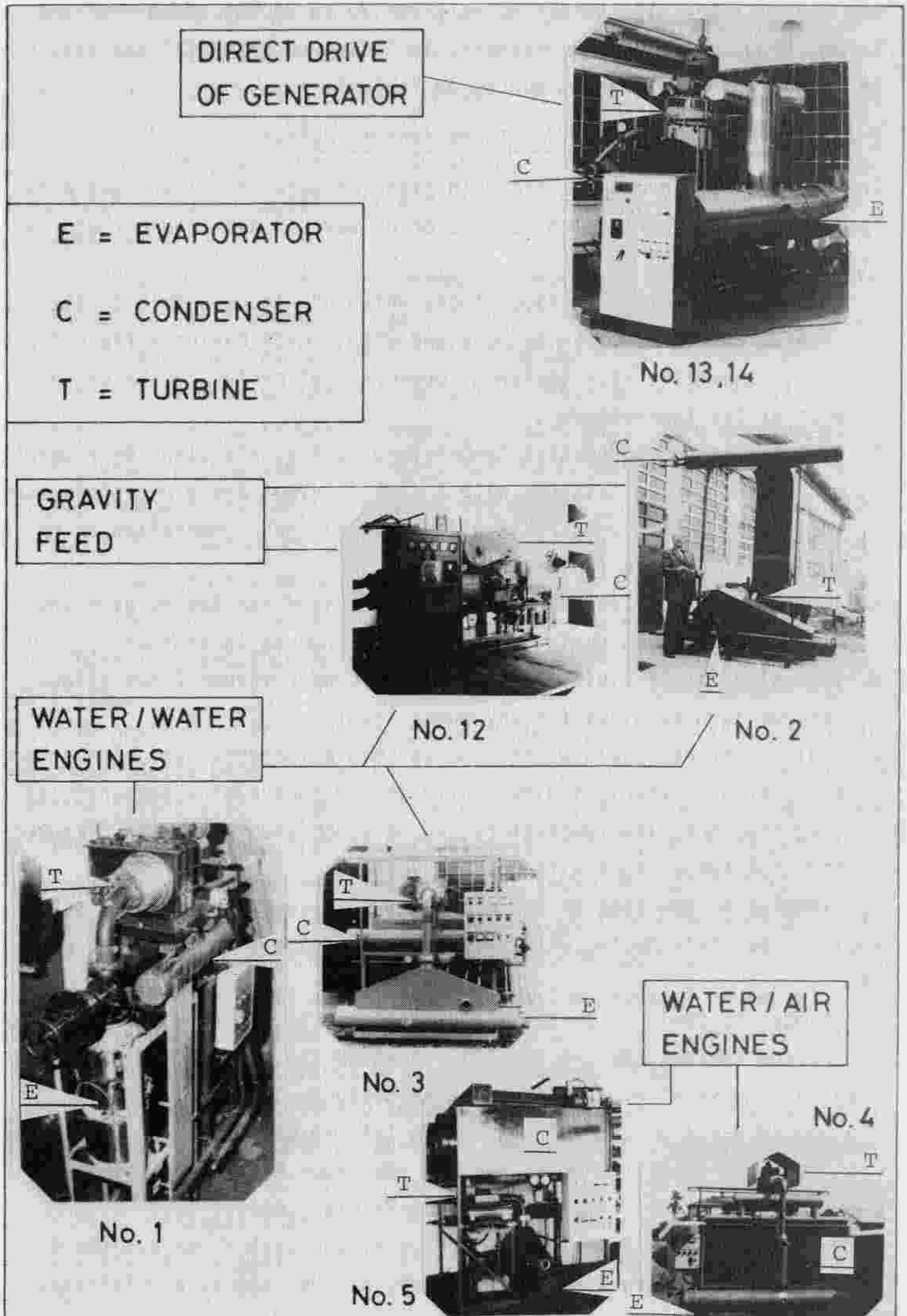


FIG. 2 View of some engines of TAB. 1.

and maximum operating temperatures (from 70 to 340°C). More detailed information on the various engines, including experimental results, can be found in the references quoted in Table 1.

2 Working Fluid Selection and Thermodynamic Cycles

The adoption of an organic fluid in place of water for a thermodynamic cycle allows the achievement of one or of several aims among those listed below:

a. thermodynamic cycle configurations which are inaccessible in the state diagram of water can be obtained with fluids having different critical parameters (for instance supercritical cycles can be designed even at low temperature);
b. even for large source/sink temperature ratios, efficient thermodynamic cycles can be obtained with a simple layout and a one stage expander, thanks to the non-extractive, desuperheating regeneration which is typical of organic fluids;
c. low peripheral speeds are in general required for the turbine and fluid condensation during the expansion process is avoided;
d. the selection of fluids giving rise to proper volume flows allows optimum turbine sizes for any power level;
e. fluid pressure levels within the various components can be selected, to a certain extent, indipendently of the source-sink temperatures (for instance, low temperature can be associated with high pressure and high temperature with low pressure).

As examples of how some of the aforesaid goals were achieved in actual engines, we shall quote, with reference to engine numbers in Tables 1 and 2:

. a number of small (3 to 12 kW) low temperature (75 to 83°C) perchloroethylene engines (No 1-5 and 12), in which the working fluid selection was at the service of a good turbine design;
. a 500 kW R-114 geothermal engine (No 11), in which the leading aim was the optimum exploitation of a given heat source through a fluid of optimized critical temperature and molecular complexity;
. two comparatively high temperature per-fluorocarbon engines (No 8-9) designed to achieve a high efficiency, at the cost of a multi-stage turbine and of a complex regenerator;
. a cogenerating, dichlorobenzene engine (No 13) featuring a direct-drive 3000 rpm turbo-alternator.

Aiming to achieve the main goal through an appropriate fluid selection, one must then accept a number of secondary characteristics which are connected with the wanted primary target. In other words, the change of the working fluid affects the design and performance of most cycle components so that any judgement about the merits of a particular solution must include a quantitative evaluation of many different technical details.

To show this, let us work out an example concerning the selection of the working fluid for a hypotetic low temperature engine (100^{o} - $40^{o}C$ evaporation-condensation temperature), of 100 kW capacity. Somewhat arbitrarily, it was decided to seek the solution within the class of linear hydrocarbons. Seven substances - from C_4H_{10} to $C_{10}H_{22}$ - are considered. The increase in molecular complexity has the effect of rising the fluid critical temperature (while reducing its critical pressure) so that the same 40-100^{o}C temperature span corresponds to regions of the state diagram of lower reduced temperatures ($T_r = T/T_{cr}$) as the number of atoms in the molecule is increased. This is clearly shown in Fig. 3, giving the saturation vapour curves and cycle configurations for the considered fluids.

The main physical difference among the various cycles is to be found in the condensation pressure, which range from 377 to 0.49 kPa (similar variations affect also the evaporation pressure). As a consequence, volume flows at condenser inlet exhibit wide variations, which directly affect the turbine design and dimensions. As shown in Fig. 4a), optimized /25/ stage rotating speeds range from 57000 rpm (for n-butane) to 1800 rpm (for n-decane), with mean diameters of 0.074 and 2.110 m respectively. Cycle pressure ratio steadily increases from about 4 (butane) to about 20 (decane), causing the turbine work to increase notwithstading the well known adverse effect of molecular weight (Fig. 5b).

Cycle efficiency is positively influenced by operation at low reduced temperatures (large number of atoms in the molecule), mainly if the cycle includes regeneration (Fig. 4b), in which case the amount of regenerated heat increases steadily from butane to decane: in the first case being sufficiently small to allow the suppression of the regenerator without excessive losses.

Finally, pump work and head are analyzed: starting from hexane or octane, the pump work becomes very small, so that also inefficient pumps are

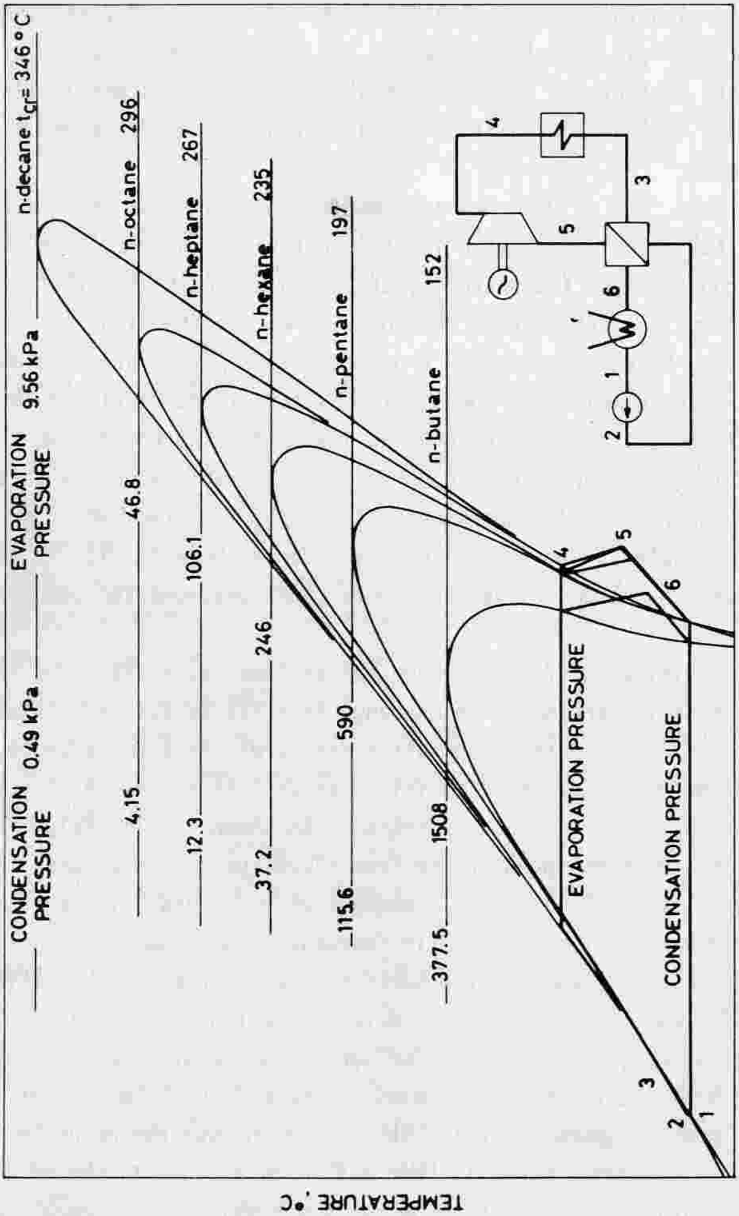


FIG. 3 Thermodynamic cycles configurations utilizing portions of the saturation curve at different reduced temperatures.

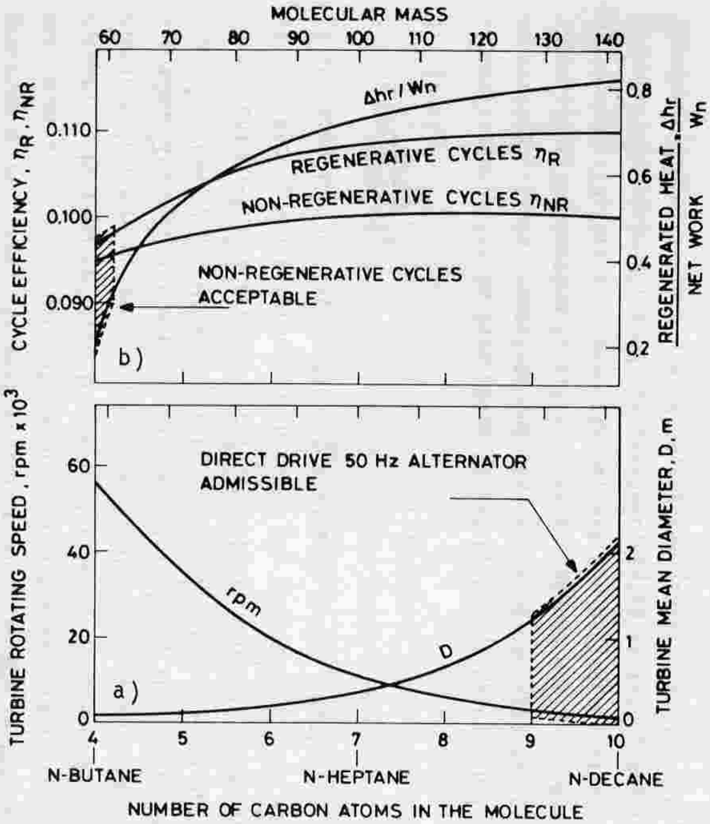


FIG. 4 Turbine and cycle characteristics as functions of the working fluid.

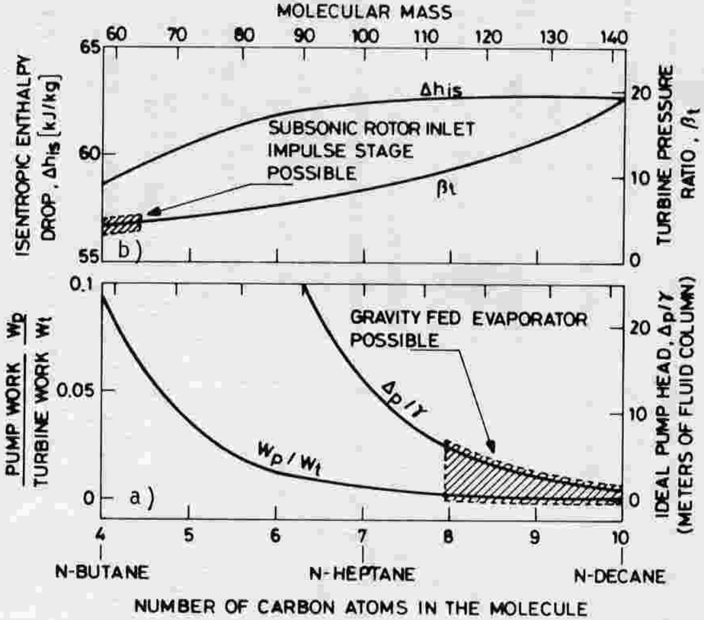


FIG. 5 Turbine and pump parameters as functions of the working fluid.

acceptable. Moreover, the evaporation pressure is so limited that condensation at a moderate height above the evaporator can supply, directly, the required head to the feed line. Even more pronounced variations in cycle parameters would be found if larger maximum to minimum temperature ratios were considered. Analogous, though different, considerations would be appropriate for other classes of fluids (for instance: chloro-fluorocarbons, perfluorocarbons, aromatic hydrocarbons, silanes etc.), which further highlights the effectiveness of working fluid selection in shaping the thermodynamic cycle characters.

3 Components Design

3.1 Turbines

The efficiency of a turbine stage is set by many factors, including blade profiles, fluid properties and manufacturing characteristics (clearances, surface roughness, etc.). However, according to the analysis developed in /24,25/, it can be stated that the efficiency of an axial flow turbine stage of good aerodynamic design is mostly determined by the following parameters, based on the similarity rules:

a. The isentropic head coefficient k_{is}, defined by:

$$k_{is} = \Delta h_{is}/(u^2/2)$$

where u(m/s) is the peripheral speed at the turbine mean radius and Δh_{is} (J/kg) is the isentropic enthalpy drop.

b. The specific speed N_s, defined by:

$$N_s = n\sqrt{\dot{V}_{out}}/\Delta h_{is}^{3/4}$$

where n is the speed of rotation (revolutions/s) and $\dot{V}_{out}$ (m³/s) is the volumetric flow rate at the stage exit.

c. The "size" parameter, defined by:

$$VH = \sqrt{\dot{V}_{out}}/\Delta h_{is}^{1/4}$$

The physical significance of VH is given by its proportionality to actual turbine dimensions.

d. The volumetric expansion ratio, defined by:

$$VR = \dot{V}_{out}/\dot{V}_{in}$$

where $\dot{V}_{in}$ is the volumetric flow rate at stage inlet. It was demonstrated in /25/ that VR accounts for the compressibility effects in a more generalized way than other equivalent parameters (pressure ratio, Mach numbers).

Two of these parameters, VH and VR, can be regarded as thermodynamic data, set only by the working fluid properties, the thermodynamic cycle and the power output. The other two, n_s and k_{is}, are to be selected by the turbine designer.
As far as k_{is} is concerned, it should be pointed out that it is generally possible to adopt optimum k_{is} values with organic fluids. In fact, they are compatible with conservative levels of peripheral speeds, a feature far to be achieved in single stage steam turbines, for which low k_{is} are a major cause of poor efficiency. A similar statement cannot be drawn for N_s, since the possibility of selecting optimum N_s depends on the particular application.
If we stipulate to select optimum values for N_s and k_{is}, it is possible to represent the achievable efficiency in the VH-VR plane, as done in Fig. 6 (from /24/): as obvious, highest efficiencies are found for large turbines and low expansion ratios.
Let's now consider the engines of Table 1. The operating conditions of various turbine stages can also be indicated in Fig. 6: it can be seen that the working fluid selection philosophy discussed in the previous section yields favourable VH even at very low power outputs (engines No 1-5). When high VR become unavoidable, as in engines operating under large temperature spans, a two (engines No 10 and 13) or multi-stage (engines No 8 and 9) solution is chosen.
As above said, the curves of Fig. 6 represent the limiting efficiency, achievable only by operation at optimum values of k_{is} and N_s. While optimum k_{is} could be actually maintained for all considered turbines, N_s quite far from optimum had to be selected in some cases, with and important efficiency decrease, as shown in Fig. 7 (from /25/). It is the case of the first stages of pluristage turbines, and of the two stages of engine No 13, where the speed of revolution was selected at a lower

FIG. 6

Constant efficiency lines for turbine stages at optimum values of k_{is} and N_S ~~/24/~~. The points refer to various turbine stages of the engines of Table 1.

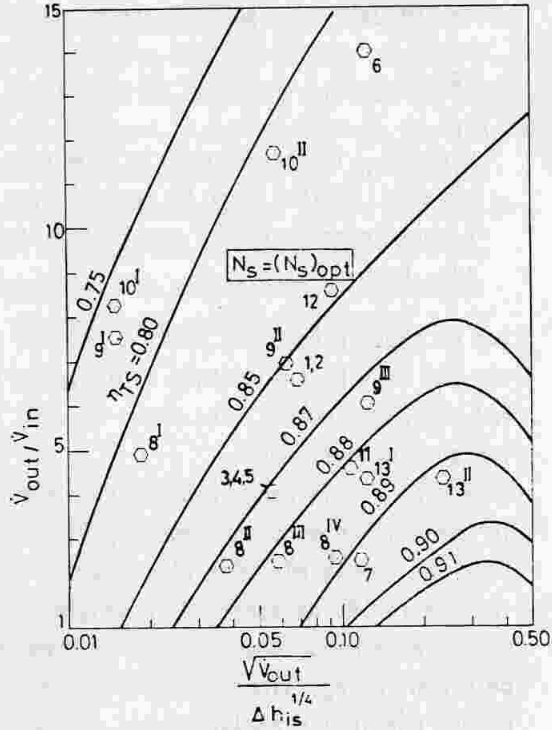
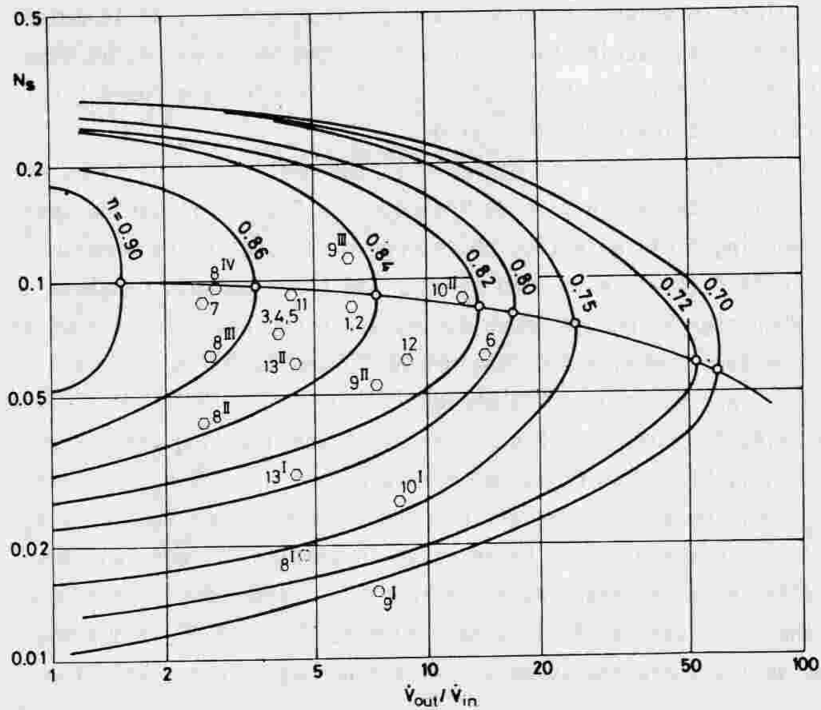


FIG. 7

Constant efficiency lines for full admission axial flow turbine stages operating at various N_S /25/. The points refer to various turbine stages of the engines of Table 1.



than optimum value for direct drive of a 50 Hz generator. It should be pointed out that the curves of Fig. 7 were obtained for a constant VH value ($\sim$ 0.06 m). Quite lower efficiencies are found for stages having lower VH, as shown in Table 3. The combined effects of low VH and N_s are so strong for the first stages of engines No 8, 9 and 10, that the adopted solution, an impulse stage with partial admission, becomes practically mandatory.

Engine No	VH, m	$\Delta\eta$
1,2	0.065	0
3,4,5	0.056	0
6	0.129	0
7	0.124	0
8 I	0.020	-0.21
II	0.039	-0.08
III	0.061	0
IV	0.098	0
9 I	0.016	-0.38
II	0.051	-0.02
III	0.114	0
10 I	0.016	-0.18
II	0.092	0
11	0.093	0
12	0.066	0
13 I	0.031	+0.01
II	0.066	+0.01

TAB. 3 Efficiency decrease from values of Fig.7 due to parameter VH.

3.2 Heat Exchangers

3.2.1 Evaporators

The following types of evaporators were employed during the experimental activity reported herewith: pool-boilers, reboilers and once through evaporators. Provided the evaporating pressure is not exceedingly low (in which case hydrostatic heads could adversely affect the heat transfer process), pool-boiling heat exchangers were found the simplest to design and to run, also owing to their intrinsic heat storing capability. Once through boilers, even at sub-critical pressures, were found the best solution to minimize working medium inventory in the case of costly fluids (engines No 8-9 of Table 1).

3.2.2 Condensers

Volume flow at condensers inlet being similar to turbine exhaust flow, the need of free flow area is related directly to turbine size. Low speed, large diameter turbines imply a free access of vapour to the whole boundary of the condenser tube bundle. Some low drag penetration lines to the bundle core were necessary in special cases. Tubes with low integral fins were found an advisable solution to reduce condensers' size and, to a lesser extent, cost.

3.2.3 Regenerators

Regenerator is a characteristic heat transfer equipment in organic fluid cycles. In the case of large source/sink temperature ratios and for fluids having complex molecules, cycle efficiency is extensively dependent on regenerator effectiveness. Furthermore, only counter-flow heat exchangers offer an adequate performance. For technological reasons, true counter-flow configurations are extremely difficult to be achieved in a heat exchanger in which volume flows in the cold and hot sides can be several order of magnitude different. Ordinary configurations feature local cross-flow, overall counter-flow arrangements, with finned surface vapour-side and tubular surface liquid-side. Eight to ten passages can be requested to achieve a satisfactory degree of counter-flow effectiveness. As in the case of condensers, turbine exhaust volume flow, often selected to obtain a desired turbine size and rotating speed, has a determining influence on regenerator configuration. Low speed large diameter turbines call for a large free-flow area regenerator which is sometimes difficult to install within a given engine frame.

4 Control items

The control of an Organic Rankine Cycle engine is conceptually analogous to the control of a conventional steam plant. However, a few practical differences are worth to be discussed:

a. Organic fluid engines are typically low power machines, to be used in adverse environment (industrial or rural) without skilled operators. Low cost and reliability are primary factors for the selection of the control subsystem.
b. The fluid inventory is generally as low as possible to minimize cost and hazard, while the mass flow in the expander is much higher then in a steam plant of the same power. Hence, the time for the hot well to be emptied or, conversely, for the evaporator to be overfilled during transients is generally lower and a faster level control system is often required.
c. High molecular mass fluids yield low peripheral and low fluid velocities. As a consequence, some degree of control inaccuracy, yielding for instance a temporary flow of low quality vapour in the turbine, can be tolerated.
d. Even in the engines having an impulse stage, the reduction of output

power is preferably obtained by throttling the admission instead of decreasing the arc of admission. The losses due to throttling are in part recuperated as thermal energy in the regenerator.

e. The number of items to be connected by fittings to the working fluid path has to be as low as possible, each connection being a potential source of leak. For this reason the number of controlled parameters is generally lower then in steam plants.

A general rule for control philosophy cannot be given due to the different uses of the power produced.
In Table 4 the admission valve mode of operation is presented for the various kind of generators adopted in the engines of Table 1.

USE OF POWER	MODE	TYPE OF GENERATOR	ADMISSION VALVE MODE OF OPERATION
ELECTRIC GENERATOR	STAND ALONE AC	SYNCHRONOUS	MODULATING (ISOCHRONOUS OR DROOP CONTROL)
	STAND ALONE WITH BATTERY STORAGE	ASYNCHRONOUS WITH CAPACITORS AND RECTIFIER	ON-OFF
	GRID OPERATION ONLY	ASYNCHRONOUS	ON-OFF
	BOTH GRID AND STAND ALONE	SYNCHRONOUS WITH ROTATING BRIDGE FOR SYNCHRONISATION	ON-OFF MODULATING (ISOCHRONOUS OR DROOP CONTROL)
MECHANICAL DRIVE OF PUMP OR COMPRESSOR			ON-OFF

TAB. 4 Admission valve operation vs. use of shaft power.

For industrial heat recovery applications in developed countries, the asynchronous generator with an on-off admission valve (slow opening/ fast closing) seems to be the best option. In fact, the advantages of supplying power during grid failure and of improving the power factor seldom compensate for the higher cost of the synchronous solution. Moreover, Italian experience with cogenerating gas turbines in the few hundred kW range has shown that it is difficult to avoid occasional counterphase operation of small generators in a disturbed grid /27/. The use of a high speed, 400 Hz asynchronous generator was tried by the authors as a way to provide, through a rectifier, DC current to lead-acid batteries. In this case the required reactive power is given by three capacitors with a solid state control to keep the voltage within an acceptable range. A more costly but probably more efficient option is to utilize a permanent magnet multi-pole synchronous generator. This solution was adopted e.g. by Barber /28/ for a 25 kW engine.

REFERENCES

/1/ Gaia, M., Macchi, E."A Comparison between Sun and Wind as Energy Sources in Irrigation Plants", Proceedings of International Solar Energy Society (ISES) Congress, Delhi (India), January 1978; "Sun, Mankind's Future Source of Energy", Pergamon Press, Vol. I, pp. 265-272.

/2/ Macciò, C., Tomei, G., Angelino, G., Gaia, M., Macchi,E. "Operational Experience of a 3.0 kW Solar Powered Water Pump", Proceedings of International Solar Energy Society (ISES) Congress, Atlanta (USA), in "Sun II", Vol. 2, pp. 1501-1505, Pergamon Press.

/3/ Angelino, G., Ferrari, P., Gaia, M., Gigliol i, G., Macchi, E. "Design and Performance of an Engine for Low Temperature Geothermal Water", May 1980, International Journal of Energy Systems, Vol. I, No. 2, 1981, pp. 187-191.

/4/ Gigliol i, G., Angelino, G., Ferrari, P., Gaia, M., Macchi, E. "Tetrachloroethylene Rankine Cycle for Waste Heat Recovery from Ceramic Tunnel Kilns", Proceedings of 13th Intersociety Energy Conversion Engineering Conference, San Diego (USA), August 1978, pp. 1488-1493, S.A.E. Paper No. 789603.

/5/ Casci, C., Angelino, G., Ferrari, P., Gaia, M., Gigliol i, G., Macchi, E. "Experimental Results and Economics of a Small (40 kW) Organic Rankine Cycle Engine", Proceedings of 15th Intersociety Energy Conversion Engineering Conference, Seattle (USA), August 1980, pp. 1008-1014, AIAA paper 809199.

/6/ Gaia, M., Macchi, E. "Heat Recovery in a Ceramic Kiln with an Organic Rankine Cycle Engine", in 'New Ways to Save Energy', Reidel Publishing Company, Dordrecht (Holland), 1980, pp. 799-808.

/7/ Casci, C., Angelino, G., Ferrari, P., Gaia, M., Gigliol i, G., Macchi, E. "Heat Recovery in a Ceramic Kiln with an Organic Rankine Cycle Engine", Heat Recovery Systems, 1981, Vol. 1, No. 2, pp. 125-131.

/8/ Angelino, G., Ferrari, P., Gaia, M., Gigliol i, G., Macchi, E. "Progetto e realizzazione di un motore geotermico a bassa temperatura da 50 kW", Proceedings of I° Seminario Informativo dell'attività del sottoprogetto 'Energia Geotermica', Pisa, 1979, pp. 647-655 (in Italian).

/9/ Angelino, G., Ferrari, P., Gaia, M., Gigliol i, G., Macchi, E. "Motori a fluido organico per acque calde geotermiche", Proceedings of I° Seminario Informativo dell'attività del sottoprogetto 'Energia Geotermica', Pisa, 1979, pp. 656-676 (in Italian).

/10/ Bado, G., Tomei, G., Angelino, G., Gaia, M., Macchi, E. "The Ansaldo 35 kW Solar Power System", Proceedings of International Solar Energy Society (ISES) Congress, Atlanta (USA), in "Sun II", Vol. 2, pp. 1090-1094, Pergamon Press, May 1979.

/11/ Angelino, G., Gaia, M., Macchi, E., Barutti, A., Macciò, C., Tomei, G. "Test Results of a Medium Temperature Solar Engine", International Journal of Ambient Energy, July 1982, Vol. 3, No. 3, pp. 115-126.

/12/ Barutti, A., Pedrick, W.G., Angelino, G., Gaia, M., Macchi, E. "The Ansaldo Solar Thermal and Photovoltaic Plants Located at Ballajura, Western Australia", Proceedings of International Energy Society (ISES) Congress, Perth, Australia, 1983 (to be published).

/13/ Angelino, G., Ferrari, P., Gaia, M., Giglioli, G., Macchi, E. "The CNPM Thermal Heat Pump - Part I - General Description and Thermodynamics Analysis", Revue International du Froid", Vol. 3, No. I, pp. 19-24, January 1980.

/14/ Macchi, E., Bassi, F., Bollina, E., Perdichizzi, A., Osnaghi, C. "The CNPM Thermal Heat Pump - Part II - Turbomachinery Development", Proceedings of XV International Congress of Refrigeration, Venezia, 1979, Paper EI-17.

/15/ Osnaghi, C., Angelino, G., Bassi, F., Bollina, E., Ferrari, P., Gaia, M., Giglioli, G., Macchi, E., Perdichizzi, A. "Progetto e realizzazione di un motore a fluido organico con espansione a turbina da accoppiarsi con un impianto solare a media concentrazione", Proceedings of II° Seminario Informativo dell'attività del sottoprogetto 'Energia Solare', Milano, 1979, pp. 432-443 (in Italian).

/16/ Salucci, B., Centemeri, L. and Carriato, G. "A 500 kWc ORC Plant for the Recovery of Heat from Geothermal Water", Proceedings of International Conference on "Geothermal Energy", pp. 249-263.

/17/ Gaia, M., Angelino, G., Macchi, E., De Heering, D., Fabri, J.P. "Experimental Results of the ORC Engine Developed for the Borj Cedria Solar Plant", Proceedings of International Energy Society (ISES) Congress, Perth, Australia, 1983 (to be published).

/18/ Angelino, G., Gaia, M. and Macchi, E. "Medium Temperature 100 kW ORC Engine for Total Energy System", Proceedings of EC. Contractors Meeting for Energy Conservation in Industry, Bruxelles, October 1982, D. Reidel Publishing Company, pp. 177-189.

/19/ Angelino, G., Gaia, M. and Macchi, E. "Medium Temperature 100 kW ORC Engine for Total Energy Systems - Experimental Results", to be published in "Energy Conservation in Industry, D. Reidel Publishing Company, 1984.

/20/ Gaia, M. "Experimental Engines for Small Scale Solar Thermal Power Generation", Intl. Solar Energy Society, UK Conference on Solar Thermal Power Generation, London, July 1978.

/21/ Gaia, M."Motori solari di piccola e media potenza", La Termotecnica, Vol. XXXIII No. 5 - 1979 pp. 296-302 (in Italian).

/22/ Cipolla, G. "Experimental Rankine Cycle Engine Designed for Utilization of Low Temperature, Low Pressure Heat", in "New Ways to Save Energy, D. Reidel Publishing Company, Dordrecht, 1980, pp. 809-822.

/23/ Gaudenzi, P. "I fluidi basso bollenti", Energie Alternative HTE, No. 23, 1983, pp. 229-234 (in Italian).

/24/ Macchi, E. "Design Criteria for Turbines Operating with Fluids having a Low-Speed of Sound", Lecture Series 100, on 'Closed Cycle Gas Turbines', Von Karman Institute for Fluid Dynamics, May 1977.

6 /25/ Macchi, E., Perdichizzi, A. "Efficiency Prediction for Axial Flow Turbines Operating with Non-Conventional Working Fluids", ASME Transactions, Journal of Engineering for Power, October 1981, Vol. 103, pp. 712-724.

/26/ Lozza, G., Macchi, E., Perdichizzi, A. "On the Influence of the Number of Stages on the Efficiency of Axial Flow Turbines", ASME Paper No. 82-GT-43, presented at 27th International Gas Turbine Conference and Exhibit, London, April 1982.

/27/ Tomassetti, G. (ENEA, Centro Studi Casaccia, Roma),Private communication .

/28/ Pons, R.L. and Dugan, A.F. "A Small Community Solar Thermal System", Solar Sciences, Spring 82, pp. 15-35.