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A REVIEW OF ORGANIC RANKINE CYCLES (ORCs) FOR THE RECOVERY OF LOW-GRADE WASTE HEAT

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Abstract—The efficiencies of ORCs using cryogens such as benzene, ammonia, R11, R12, R134a and R113 as working fluids have been analyzed parametrically and compared. For operation between two isobaric curves, the system efficiency increases and decreases for wet and dry fluids, respectively, and the isentropic fluid achieves an approximately constant value for high turbine-inlet temperatures. These effects are primarily due to the different slopes and shapes of the saturation vapor curves of the fluids. Isentropic fluids are most suitable for recovering low-temperature waste heat. Freons and their alternatives have been studied and shown similar system responses in ORCs. © 1997 Elsevier Science Ltd.

INTRODUCTION

In the energy-to-power conversion industry, the thermal efficiency becomes uneconomically low when the exhaust-stream temperature drops below 370°C. However, recovering low-grade waste heat in power generation becomes economically feasible when using ORCs. For countries with low fuel prices such as the U.S., waste-heat recovery with ORCs may yield economic benefits. The recovery value is greater for countries with high fuel prices.

Statistical investigations indicate that low-grade waste heat accounts for 50% or more of the total heat generated in industry. Due to lack of efficient recovery methods, low-grade waste heat has generally been discarded by industry and has become an environmental concern because of thermal pollution. Therefore, recovery of low-grade waste heat with production of electricity has become a challenging task for the power industry. Recovery of waste heat has recently become popular. Generation of electricity in a bottoming cycle is a promising option since it consumes virtually no additional fuel. For a steam cycle, a non-condensation steam turbine can be used in a topping cycle along with the waste-heat recovery system. A Rankine cycle using water as working fluid does not allow efficient recovery of waste heat below 370°C.

ORCs have been investigated for power production [1,2]. However, wide applications have not been achieved because of concerns about economic feasibility and safety. There is a wide range of the heat sources which can be applied to the ORC systems such as waste heat from the condenser of a conventional or a nuclear power plant, waste heat from industrial processes, solar radiation, and geothermal energy [3,4]. Regarding the working fluids, the following fluids have been considered in the literature [5–7]: benzene, toluene, pyridine, and the azeotropic mixtures fluorinol 85, 2-methyl pyridine/water, and some cryogens such as the freons R114, R113, R11. Chaudoir et al [8] used benzene as the working fluid over a medium temperature range to generate electrical power of 37.5 kW. This design was claimed to be applicable to space stations in which solar radiation serves as the heat source to generate electrical power up to several hundred kW.

ORC systems with capacities ranging from 750 to 1500 kWe were examined by Koebbman [9]. An ORC system combined with a space nuclear reactor in order to achieve higher efficiency was proposed by Niggeman [10]. Hant and Coles [11] studied various methods of recovering the waste heat at temperatures around 700–1000°F in the chimney exhaust of a glass/ceramic furnace. Utilization of waste heat rejected from a condenser was examined by Angelino et al [12], who concluded that a combined

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cycle with an ORC system as bottoming cycle that utilizes the waste heat at a temperature greater than 200°C from the condenser has a return-on-investment (ROI) of 2 years less than for a conventional cycle.

CONCEPTS OF ORC SYSTEMS

A simple ORC system for converting waste heat from a gas turbine or other heat sources into useful electrical power is depicted in Fig. 1. There are four ideal processes and basic pieces of equipment for the cycle. ORC systems are usually used in the bottoming cycle of a gas cycle or a steam cycle to form a combined cycle in order to meet various patterns of electrical or thermal power requirements.

Categories of working fluids

The saturation vapor curve is the most crucial characteristic of a working fluid in an ORC. This characteristic affects the fluid applicability, cycle efficiency, and arrangement of associated equipment in a power-generation system. There are generally three types of vapor saturation curves in the temperature–entropy (T–S) diagram: a dry fluid with positive slopes (dT/dS), a wet fluid with negative slopes, and an isentropic fluid with nearly infinitely large slopes. Wet fluids usually have low molecular weights; examples are water and ammonia. It is observed from the T–S diagram that a superheater is employed to superheat the vapor. Superheated fluids are benzene and R113. The saturated vapor phase of a dry fluid becomes superheated after isentropic expansion. An isentropic fluid has a nearly vertical vapor saturation curve, e.g. R11 and fluorinal 85. Since the vapor expands along a vertical line on the T–S diagram, vapor saturated at the turbine inlet will remain saturated throughout the turbine exhaust without condensation. The features of persistent saturation throughout expansion and the fact that there is no need for installing a regenerator make isentropic fluids ideal working fluids for ORCs.

Other considerations for the working fluids

The thermophysical properties of working fluids are compared and presented in Table 1. It is apparent that dry and isentropic organic fluids generally have much lower relative enthalpy drops during expan-

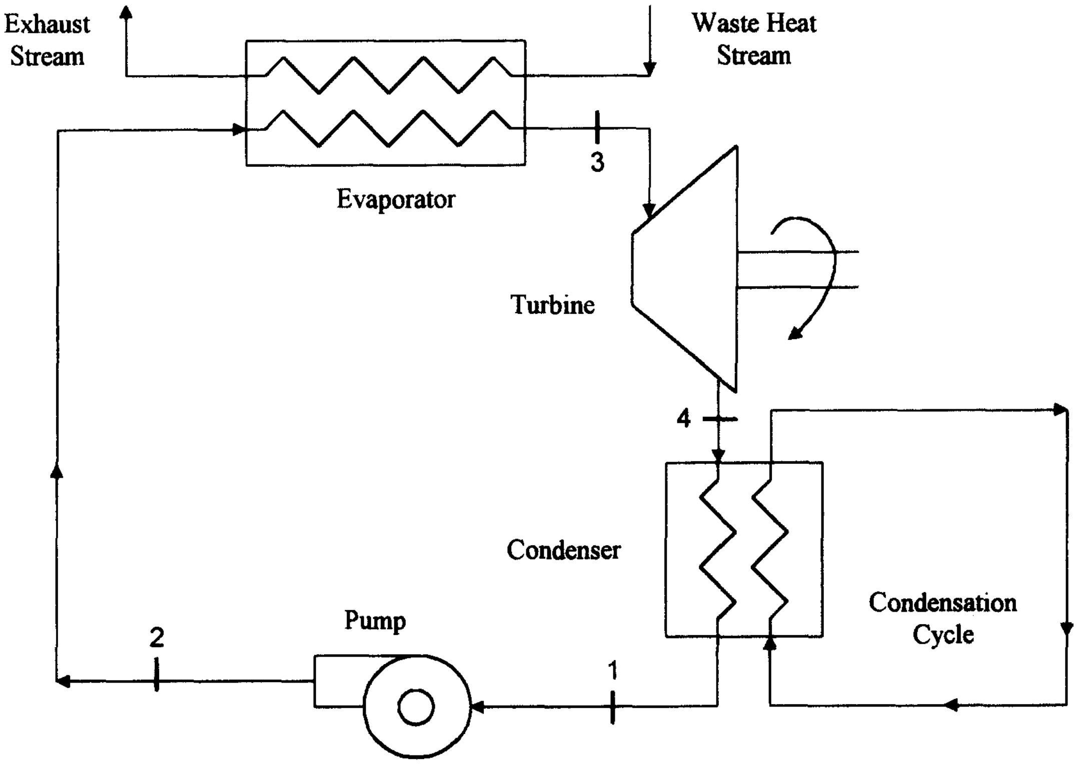

Fig. 1. A simple schematic of an ORC for use as a bottoming cycle.

Table 1. Thermophysical properties of working fluids.

Parameters	H_2O	NH_3	Benzene	R134a	R12	R11	R113
Molecular weight	18	17	78.14	102	121	137	187
Slope of the saturation vapor line	Negative	Negative	Positive	Isentropic	Isentropic	Isentropic	Positive
Enthalpy drop across the turbine	1570 ~ 900 (kJ/kg)	725 ~ 70 (kJ/kg)	120 ~ 230 (kJ/kg)	55 ~ 22 (kJ/kg)	43 ~ 20 (kJ/kg)	80 ~ 40 (kJ/kg)	85 ~ 60 (kJ/kg)
Max. stability temp. (K)	None	750	600	450	450	420 ~ 450	450 ~ 500
Turbine stage(s)	3 (or more)	3 (or more)	Single	Single	Single	Single	Single
Critical point	647 K 22.06 MPa	405.3 K 11.33 MPa	562.2 K 4.9 MPa	374.15 K 4.06 MPa	385 K 4.13 MPa	471 K 4.41 MPa	487.3 K 3.41 MPa
Boiling point at 1 atm (K)	373	239.7	353	248	243.2	296.2	320.38
Latent heat at 1 atm (kJ/kg)	2256.6	1347	438.648	215.52	166.1	178.8	143.9

sion than the water–steam mixture. Therefore, a single-stage turbine is usually used in an ORC, whereas a multi-stage steam turbine is used in a water–steam cycle. Unlike water, most organic fluids suffer chemical decomposition and deterioration at high temperatures and pressures. Therefore, an ORC system must be operated well below the temperature and pressure at which the fluids are chemically unstable. Most organic fluids have relatively low critical pressures and are therefore usually operated under low pressures and with much smaller heat capacities than water–vapor cycles. A suitable organic fluid must have a relatively high boiling point. Based on these features, the following fluids have been selected in the current study: benzene, ammonia, R11, R12, R113 and R134a. Water is used as reference for comparison.

ANALYSIS AND RESULTS

The performance of ideal ORC systems has been analyzed by using the appropriate thermodynamic properties for the various organic fluids. Energy losses due to irreversible processes occurring in the cycle and heat-transfer losses are ignored. A software program was developed to perform data interpolation for the thermodynamic properties and calculate system efficiencies. Once a working fluid has been selected, variations of system efficiency with turbine-inlet temperature and pressure and condenser-outlet temperature and pressure were calculated using iterations.

As is shown in Fig. 2, operations of the organic fluids are restricted to narrow temperature ranges and low pressures, primarily due to restrictions imposed by their thermodynamic properties. These also include stability and safety of the organic fluids under high temperatures and/or pressures. Of the fluids investigated, benzene was found to provide the highest efficiency, followed sequentially by R113, R11, R12, R134a, and ammonia. The efficiency is closely related to the latent heat of the fluid at low pressure; a greater latent heat at low pressure yields a lower efficiency since a larger portion of the energy carried by the fluid is rejected via the condenser. As shown in Fig. 2, the efficiency is a weak function of the turbine-inlet temperature, i.e. an increase of superheat in the turbine inlet does not result in a significant increase in efficiency.

Figure 3 shows the efficiencies versus the turbine-inlet temperatures at various evaporation pressures for water. It is obvious that the efficiency increases nearly linearly with the turbine-inlet temperature. Like water, ammonia is also a wet fluid and therefore shows trends similar to water. Unlike wet fluids, dry fluids show decreased efficiency as the turbine-inlet temperature is increased, except when the system pressure is very high (see Fig. 4 for R113). This result indicates that the optimum efficiency occurs if R113 is operated along the saturation curve without being superheated. Isentropic fluids exhibit trends similar to wet fluids except that the increase of efficiency levels off as the temperature is increased, as is shown in Fig. 5 for R11. The system efficiency also increases as the system pressure increases. It is obvious that the working fluid yields more work during the isentropic expansion process if the turbine-inlet pressure is raised. However, raising the system pressure is not always feasible for economic reasons since the capital costs for the waste-heat boiler and piping system, as well as system complexity and material selection of the components, must also be considered.

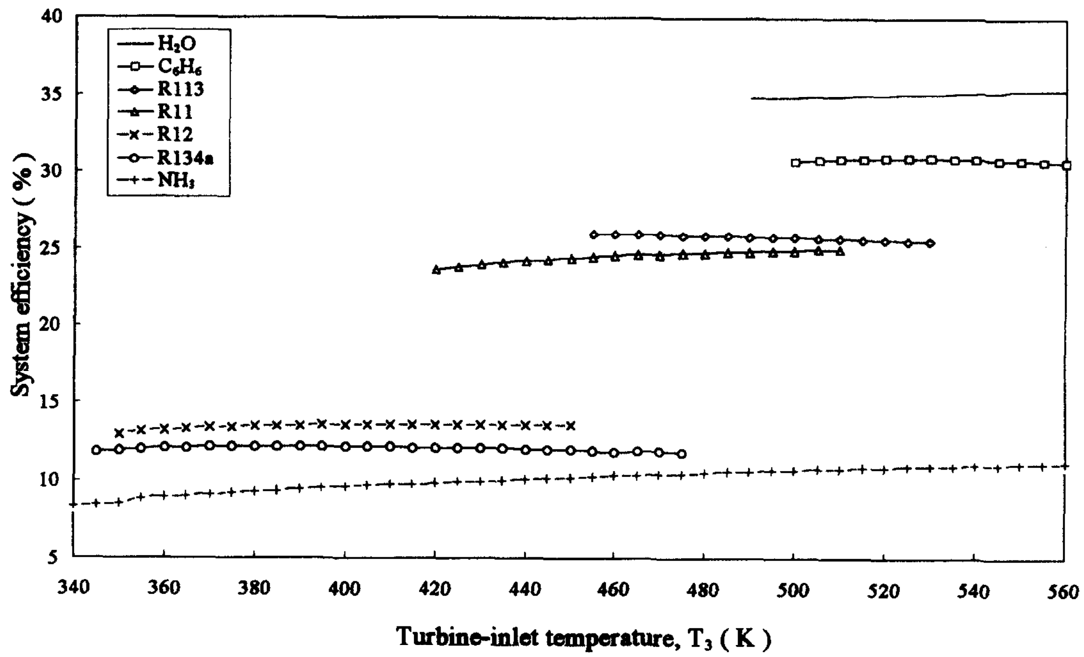

Fig. 2. Variations of system efficiency with turbine-inlet temperature for various working fluids ($P_3 = 2.5$ MPa, $T_1 = 293$ K).

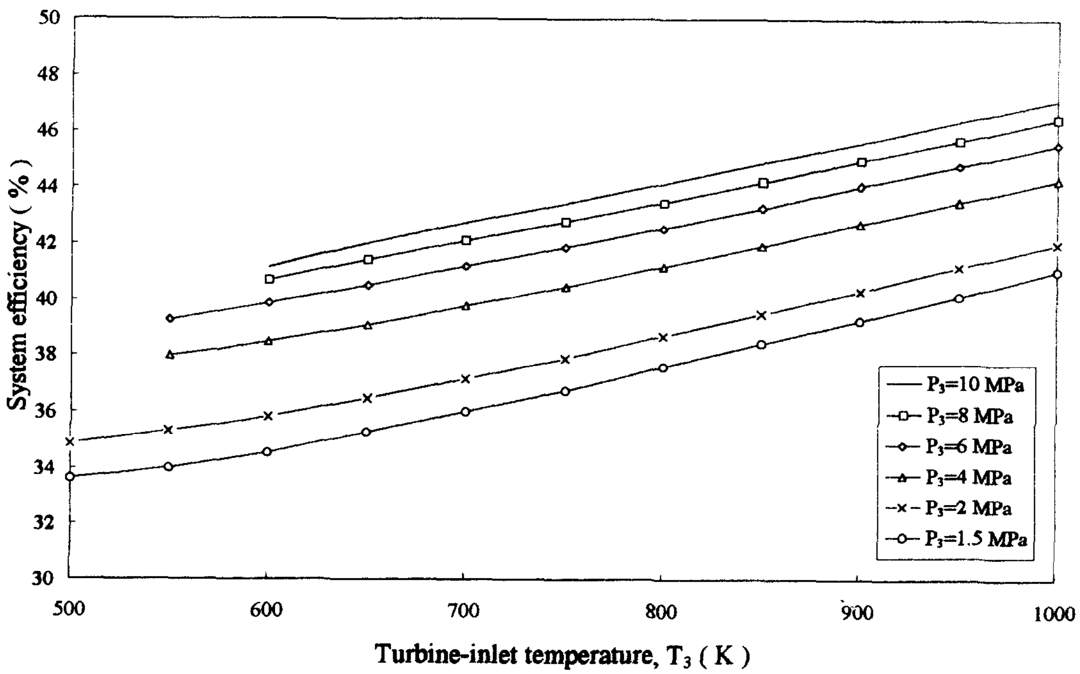

Fig. 3. Variations of system efficiency with turbine-inlet temperature for wet water at $T_1 = 293$ K.

The turbine-inlet temperatures exerts a wide range of effects on system efficiencies, depending on the slopes of the isobaric curves in the superheated region of the T–S diagram. If a fluid exhibits a steeper slope for the isobaric curve of the high-pressure region than in the low-pressure region, the system efficiency increases as the turbine inlet temperature is raised. This result is explained by Fig. 6, which shows that R113, for example, has a steeper high-pressure isobaric curve than at low pressure. As a consequence, the efficiency of the cycle using R113 as the working fluid decreases as the turbine-inlet temperature increases. The effect of turbine-inlet temperature on system efficiency for other fluids may be analyzed similarly. These features may also be explained from the viewpoint of irreversibility.

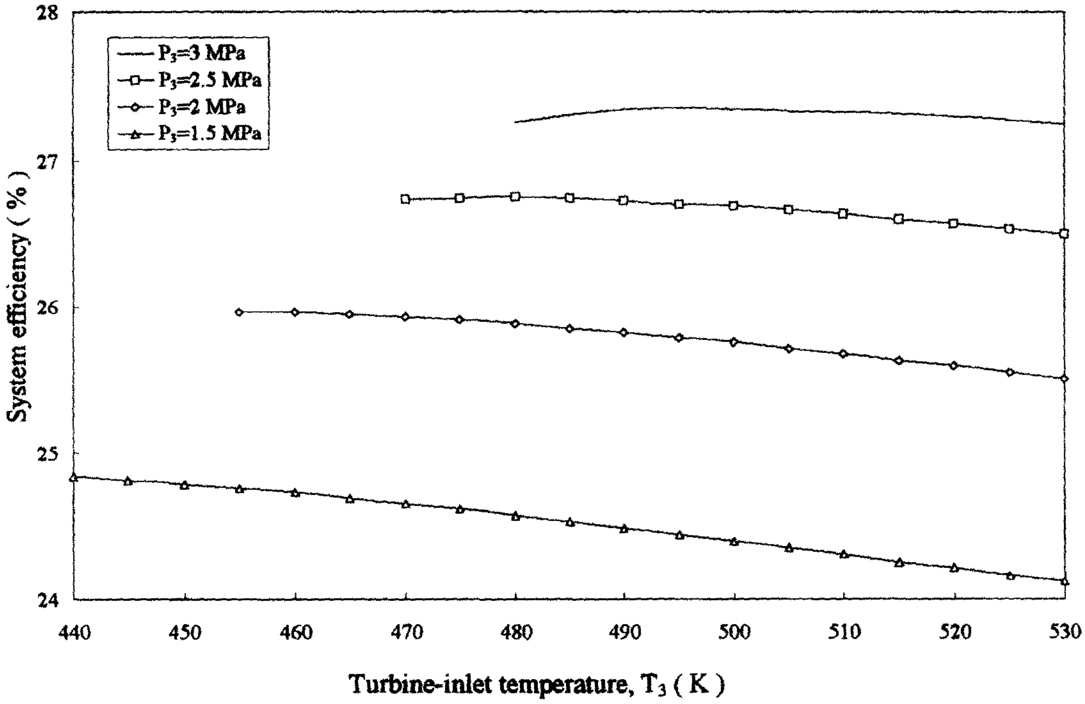

Fig. 4. Variations of system efficiency with turbine-inlet temperature for dry R113 at $T_1 = 293$ K.

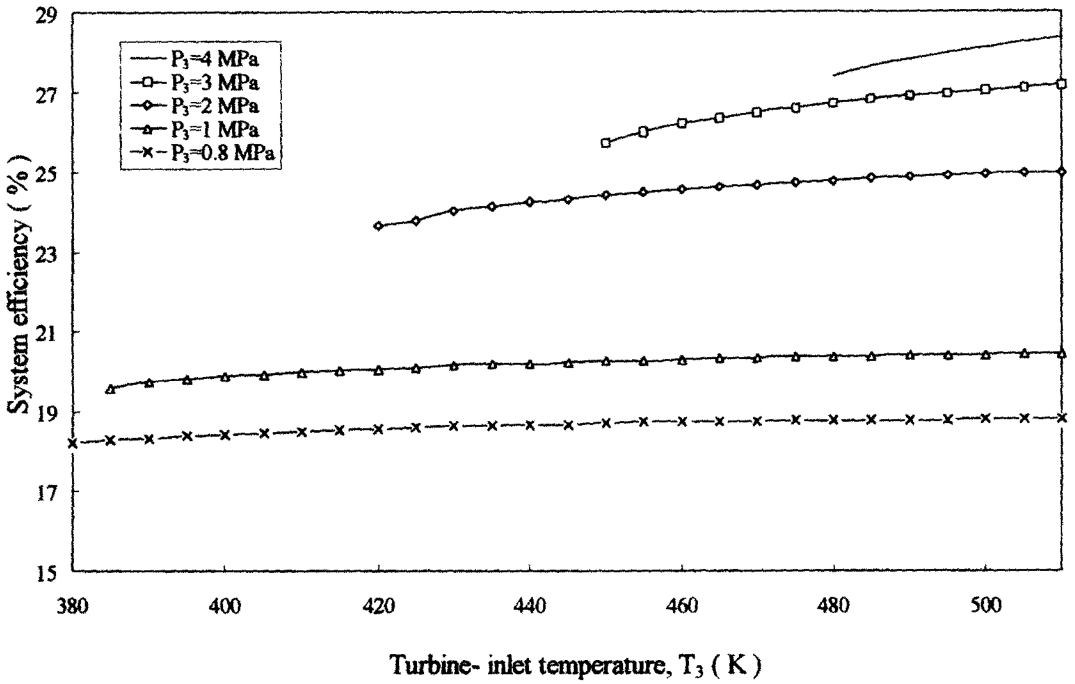

Fig. 5. Variations of system efficiency with turbine-inlet temperature for isentropic R11 at $T_1 = 293$ K.

For wet fluids with less steep low-pressure isobaric curves on the T–S diagram, a benefit results from the decrease in the average temperature of the fluid during cooling, which results in a decrease in the irreversibility for the condensing process.

The condenser exit temperature is affected and limited by the environmental temperature. Therefore, the system efficiency is also affected by environmental conditions. As is shown in Fig. 7, the system efficiency decreases as the condenser-exit temperature increases. This result means that an ORC system installed at a location of lower annual temperature will have a higher system efficiency. The shape of the curves is affected by the shape of the saturation curve of the fluid.

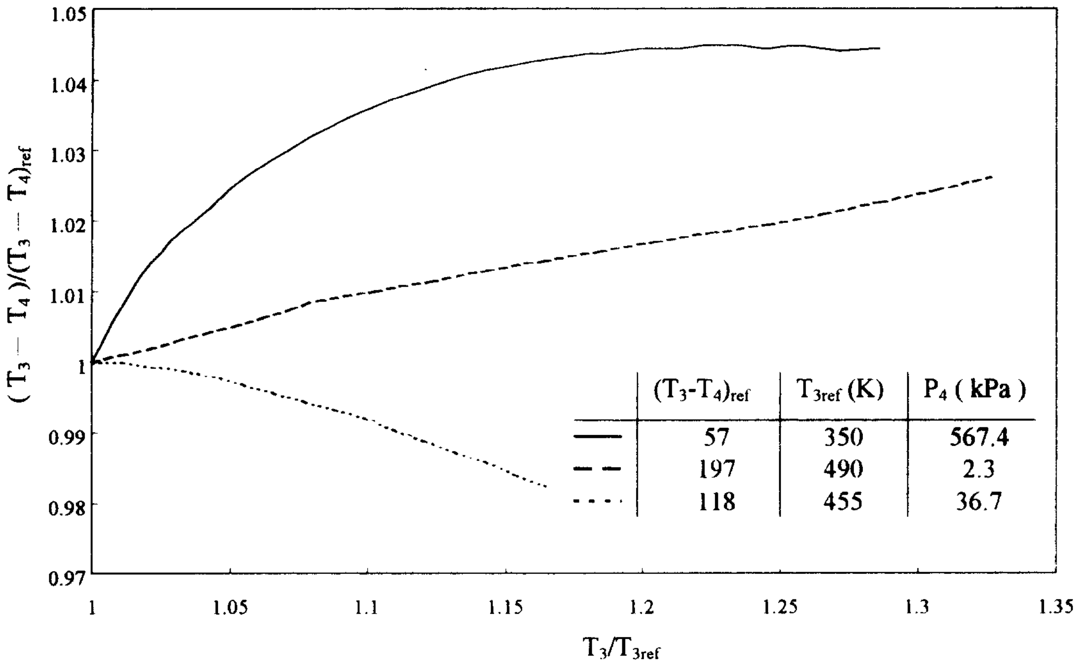

Fig. 6. (T_3–T_4) vs T_3 in the superheat region for water (wet), R12 (isentropic), and R113 (dry); $P_3 = 2000$ kPa.

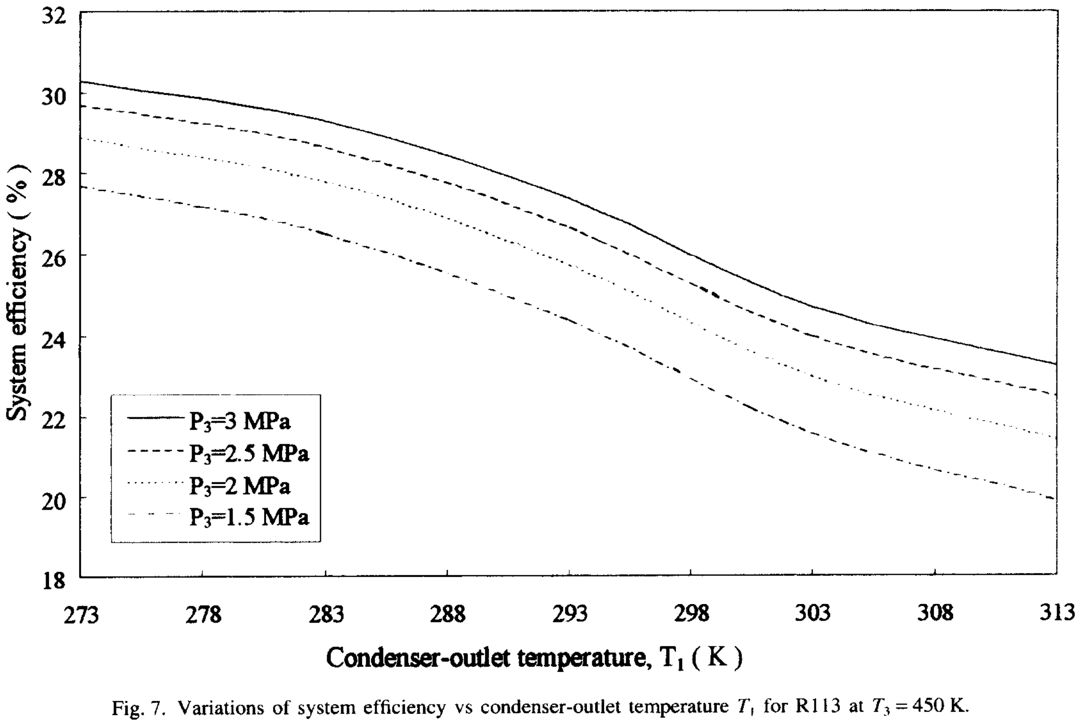

Fig. 7. Variations of system efficiency vs condenser-outlet temperature T_1 for R113 at $T_3 = 450$ K.

We find that many organic fluids are suitable as refrigerants, e.g. R113 which shows a relatively high thermal efficiency. However, most cryogenic fluids have a high content of CFCs, which are believed to be the main sources causing ozone depletion and contribute to changes of the world climate. Their use is to be discontinued by international treaty agreements. Therefore, substitutes are now introduced to replace CFC-containing cryogen fluids. Substitutes are R134a (HFC134) for R12 and HCFC123 for R11. As is shown in Fig. 8, the system efficiency of the cycle using R12 as the working fluid is generally higher than that of the cycle using R134a as the working fluid under the same working conditions. The difference in thermal efficiency is closely related to applicable values at the critical points and to the

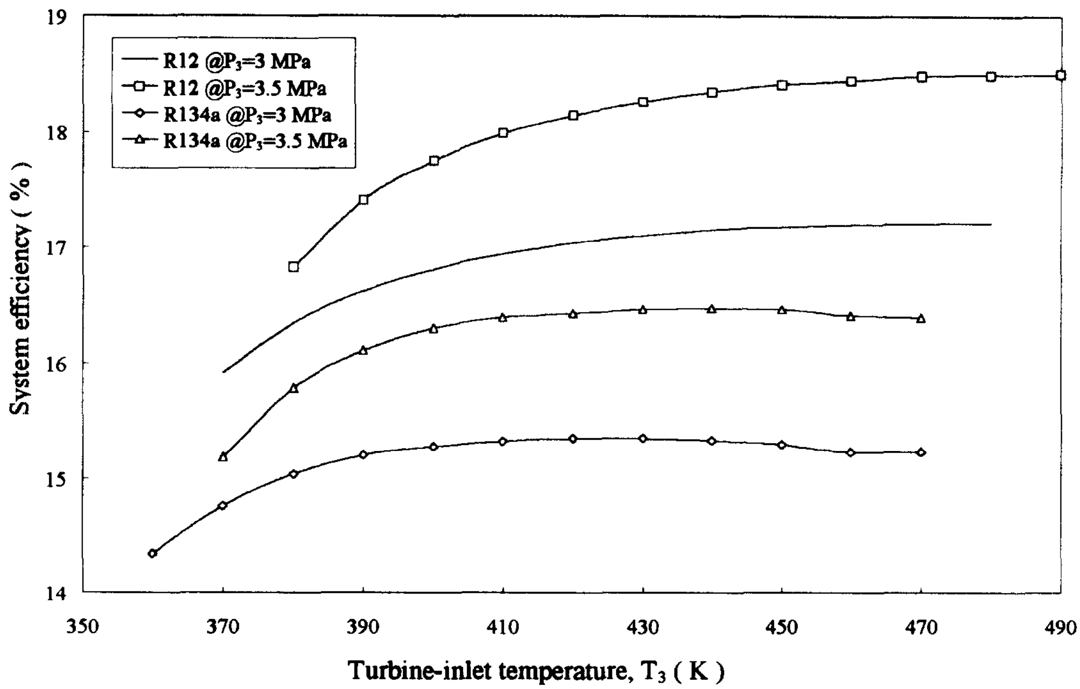

Fig. 8. System efficiencies vs turbine-inlet temperature for R12 and R134a at $T_1 = 293$ K.

latent heats at low pressures. Nevertheless, the effects of turbine-inlet temperatures on system efficiency are almost identical for these fluids.

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