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THERMAL CYCLE ANALYSIS FOR AN APPLICATION TO HEAT TRANSPORT SYSTEM
– TWO STAGE CONDENSATION SYSTEM–

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ABSTRACT

In the energy-saving research developments in recent years, many types of research project on high performance heat pump systems are under development. It is required to estimate the thermodynamic performance of these thermal cycles on a common scale by applying an adequate method. a basic cycle of two stage condensation heat pump system is considered. Steady state conditions are analyzed for the given parameters of structural component characteristics and ambient conditions. Solutions are obtained by numerical methods, and the temperature, pressure, flow rate, thermal input, output power required are calculated by means of an improved simulation program. these results were compared with experimental results, and also the exergy analysis was made for the thermodynamic loss evaluation.

INTRODUCTION

In this analysis, an basic cycle of two stage compression two stage condensation system[1] is considered. Steady state conditions are analyzed for the given parameters of component characteristics and ambient conditions. Solutions are obtained by numerical methods, and the temperature, pressure, flow rate of thermal input and output, and compressor power required are calculated by means of improved simulation program based on a single stage system analysis.[2, 3, 4, 5]

ANALYTICAL METHOD

1. Heat pump cycle balance points [6]

Evaporator water flow rate and inlet temperature, and condenser water flow rate and inlet temperature being set, heat pump system in a properly controlled state of compressor, condenser, expansion valve and evaporator, becomes to a steady state. This state is called as matching point or cycle balance point. Fig.1 shows the concept of two stage compression two stage condensation heat pump system analyzed in this report. Fig.2 shows a pressure–enthalpy diagram corresponding to the system in Fig.1. The properties of working medium at the inlet and outlet of system components are on display in order to calculate the operation of the system. Point 6 and 9 are the state after the adiabatic compression at each stage, and point 7 and 10 are the state after the polytropic compression. In the following, two stage compression two stage condensation cycle is abbreviated as two stage condensation cycle. Basic construction of two stage condensation cycle is assumed to be a superposed form of two single stage cycles whose evaporators are common. In Fig.2, the working fluid entering in the first stage compressor with a flow rate of m_1+m_2 are separated into flowrates m_1 and m_2 at the point 6, and discharges heat at the two condensers with the different temperature lebels, and meets at point 16, and constitutes a cycle. The flow rate ratio $m_2/(m_1+m_2)$ at the point of junction 6 and the two condenser pressure ratio P_{c2}/P_{c1} are given as initial parameters. Adiabatic compression and contant pressure variation at the condenser and evaporator are assumed, and the compressor work W_1, W_2, the thermal output from condensers, Q_{c1}, Q_{c2} and the thermal input to a evaporator Q_e are shown in the following,

$$W_1 = m_1 (h_6 - h_5) \tag{1}$$
$$W_2 = m_2 (h_9 - h_6) \tag{2}$$
$$Q_{c1} = h_6 - h_{16} \tag{3}$$
$$Q_{c2} = h_9 - h_{13} \tag{4}$$
$$Q_e = h_2 - h_1 \tag{5}$$

where, h denotes enthalpy.

When cycle balances, by presenting the inlet and outlet state of each components on pressure–enthalpy diagram, the working medium thermal properties neglecting a fluid momentum and energy losses are shown. These points are capable to be obtained experimentally. In the present analysis, 19 thermal characteristic points are obtained numerically from the components and operating conditions.

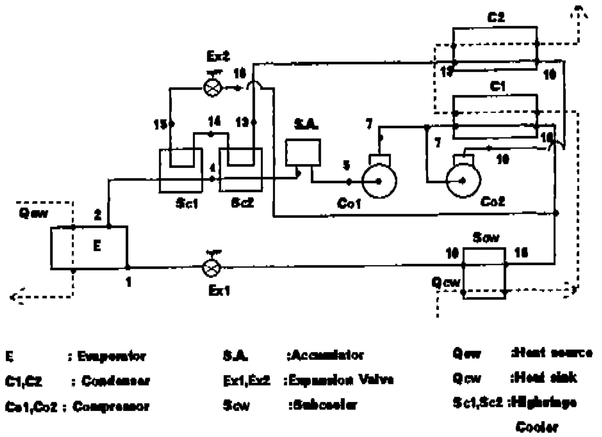


Fig.1 Two stage condensation heat pump system

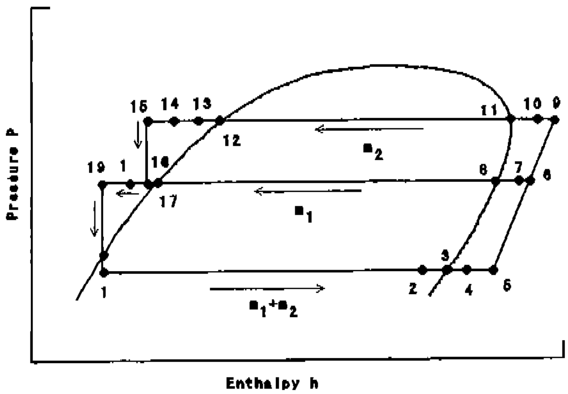


Fig.2 Two stage condensation heat pump cycle

2. Heat pump system

Component models in a heat pump system are considered. The parameters which require specifications are the rate of volumetric displacement, a compression efficiency, a clearance volume ratio and a volumetric efficiency. [7]

Theoretical volumetric efficiency is represented in Eq. (6) in terms of clearance volume ratio C, condensation and evaporation pressure P_c and P_e.

$$\eta_{ci}=1+C-C(P_c/P_e)^{1/n} \tag{6}$$

The ratio of actual and ideal volumetric efficiency is defined as η_r, then

$$\eta_v=\eta_r \cdot \eta_{vi} \tag{7}$$

The refrigerant mass flow rate is shown in terms of a rate of volumetric displacement V and a specific volume v.

$$m=\eta_r \cdot V/v \tag{8}$$

The compression efficiency is shown in the following.

$$\eta_c=(h_{oi}-h_i)/(h_{oa}-h_i) \tag{9}$$

The enthalpy h_{oi}, h_{oa}, h_i represent the properties at the end of an polytropic compression, adiabatic compression, and the start of compression respectively. The flow rate of a first stage compressor is given from Eq. (6), Eq. (7) and Eq (8), and the compressor work per unit weight is given from Eq. (9). The second stage flow rate is calculated from the flow rate dividing ratio: $m_2/(m_1+m_2)$ given as a initial parameter, and the first stage flow rate. Eq. (6), (7) and (9) are assumed to be applied to the either side of first and second stage compressors.

As for the condenser, an air-cooled or a water-cooled condenser may be simulated. The numerical value of parameters, (1) degree of subcooling, (2) heat exchanger effectiveness, (3) heat capacity of the space air or the storage water, (4) initial approximation of the condensation temperature require specification. The condenser performance is specified by the following equations.

The condenser heat transfer rate is

$$Q_c=\varepsilon_c \cdot H_c(T_c-T_{cwi}) \tag{10}$$

The heat transfer effectiveness [8] is

$$\varepsilon_c = (T_{cwo} - T_{cwi})/(T_c - T_{cwi}) \tag{11}$$

where, (1) H_c: heat capacity of space air or storage water, (2) T_c: condensation temperature of refrigerant, (3) T_{cwi}: inlet temperature of space air or storage water, (4) T_{cwo}: outlet temperature of space air or storage water.

The condensation temperature is calculated from Eq. (10), and the outlet temperature of space air or storage water leaving the condenser is obtained fromEq. (11). The mode of operation determines whether the heat sink is space air or storage water. In either cases, the same parameter and equations are used.

As for the evaporator heat source, (1) an ambient–air–cooled evaporator, (2) a storage–water–cooled evaporator with a valve to control the water flow rate, and (3) a storage–water–cooled evaporator in series with a tempering heat exchanger are considered. The evaporator performance extracting heat from ambient air is specified by the heat capacity of a heat source and the heat exchanger effectiveness. A water–cooled storage evaporator also have the same parameters. The performance of these evaporators is governed by Eq. (12).

$$Q_e = \varepsilon_e \cdot H_e (T_{ewi} - T_e) \tag{12}$$

where, ε_e: evaporator heat exchanger effectiveness, H_e: heat capacity of the heat source, T_{ewi}: heat source temperature, T_e: evaporation temperature. heat exchanger effectiveness ε_e is assumed to be constant.

3. Analytical program

The solution of the foregoing equations for the specified values of the initial parameters yields the steady state performance characteristics of the system. The computer program for this solution consists of a series of subroutines for analyzing the thermodynamic cycle for a heat pump, a series of subroutines for analyzing the evaporators, and the main program which determines the evaporation and condensation temperature for the specified heat source and sink temperatures. Thermodynamic analysis subroutine performs a thermodynamic analysis of a heat pump cycle, shown schematically in Fig.1 and Fig.2 . As for initial conditions, evaporation temperature, first stage condensation temperature, degree of superheat, degree of supercool, heat exchanger effectiveness and compression efficiency needs to be specified. The subroutine calculates the temperature, pressure and enthalpy at each key point on the cycle (shown in Fig.2), the energy transfered per unit mass for each heat exchanger, the compressor power required per unit mass, the specific volume at the compressor inlet and the coefficient of performance. Evaporator subroutine calculates the total rate of energy absorbed from the heat source and transfered to the refrigerant. Main program calculates the steady state performance characteristics for a specified mode of operation and heat source and sink conditions. Cycle subroutine is called to calculate the energy absorbed by the refrigerant in the evaporator per unit mass of refrigerant and the specific volume of the refrigerant at the inlet of the compressor. An estimate of the total rate of energy absorbed by the refrigerant in the evaporator is calculated as the product of the energy absorbed per unit mass, as determined by cycle subroutine and the mass flow rate of refrigerant. A new estimate of the first stage condensation temperature is calculated by Eq. (10), and is used in successive iteration. The second stage condensation temperature is obtained without iteration procedure, directly from the condenser initial parameters, the pressure ratio P_{c2}/P_{c1}, and the mass flow rate $m_2/(m_1+m_2)$.After the rate of energy absorbed in the evaporator and the condensation temperature are converged within a specified acceptable tolerance, the rate of energy transfer for all components, evaporator, condenser, liquid–subcooling heat exchangers, the power required by the compressor, the mass flow rate of refrigerant and other quantities of interest are calculated. Detailed analytical procedures are described in the reference [2].

EXPERIMENTAL PROCEDURE

An experimental device is made for the performance analysis of two stage condensation heat pump cycle under a specified condition. The schematic are shown in Fig.3. The devices is capable to experimentally investigate the effect of the variation of temperature, pressure and mass flow rate at several points in the pressure–enthalpy diagram upon the heat pump performance by changing the parameters of evaporation temperature, condensation pressure and the degree of superheat and supercool within a controllable specified

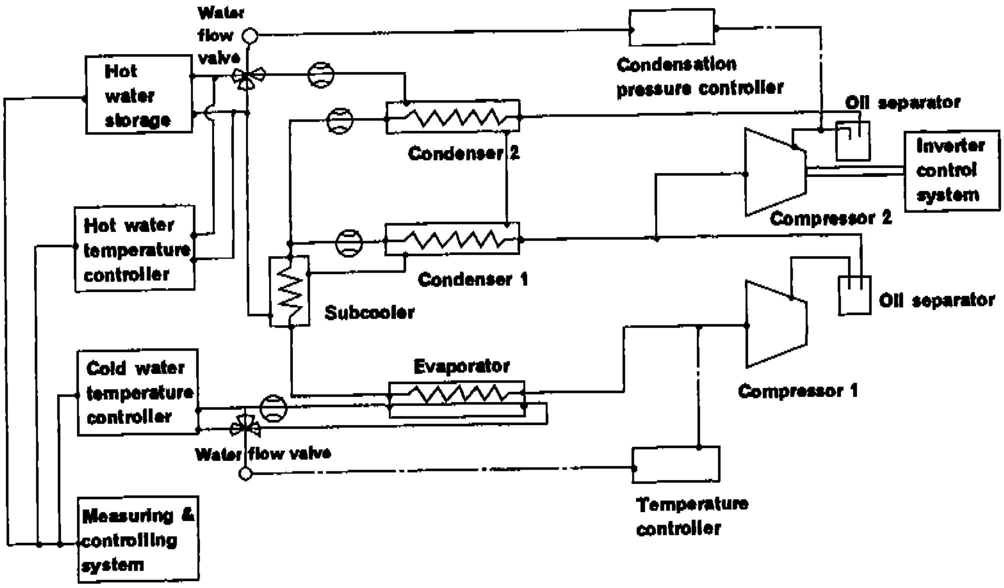


Fig.3 Schematic of experimental apparatus

range. R22 is used as a working fluids, and the compressor outputs are 1.9kW for the first stage, and 1.2 kW for the second stage. Manual type expansion valves are used for the first and second stages. The compressor r.p.m. for the second stage is made variable using a inverter control, in order to control the first stage condensation pressure within a certain range. Water supply is used for a heat source and sink. Some water flow content is extracted from the heat sink, and is utilized for the heating of a heat source. The heat sink consists of a constant temperature water supply of 6×10^{-2} m ^{3} water vessel with a automatic temperature controller equipment. Pressure, temperature and flowrate sensors are connected to a control and instrumentation system to make possible of online performance characteristics measurement.

ANALYTICAL RESULTS

Calculated examples from the analytical program are shown in Fig.4 ~ Fig.5. R22 [9] is used as a working fluids. In the figure, T_{c1}, T_{c2}, T_e, PR, Q_{co}, Q_{ev}, W, COP means first and second stage condensation temperature, evaporation temperature, first stage pressure ratio (P_{c1}/P_e), condensation energy flow rate, evaporation energy flow rate, output power required by compressor and coefficient of performance. Initial heat pump parameters used in calculations are obtained from experimental data.

Fig.4 shows a comparison of the performance characteristics for the five different working fluids. Only the result for R22 corresponds to the experiment, and results for other working fluids are the calculated results employing the corresponding similar experimental parameters based on the results for R22. The condition for the evaporator water inlet and the condenser water inlet temperature=283K, and also the constant flow rate are assumed. Employing R123 and R114 shows a considerable increase in COP, but a remarkable decrease in condensation energy flow rate. This means that the choice of R123 or R114 in a system designed for R22 results in only a small increase of COP, however in a decrease to the quarter of a thermal output for R22.

Fig.5 shows the effect of the pressure ratio P_{c2}/P_{c1} upon the performance characteristics under the condition of constant temperature difference at the condenser water inlet and outlet : $\triangle T_{cw}$=30K, and a constant temperature difference at the evaporator water inlet and outlet : $\triangle T_{ew}$ =5K. The particular case of P_{c2}/P_{c1}=1.0 corresponds to the single stage system, and increasing the ratio P_{c2}/P_{c1} results in the decrease of COP. The maximum value of COP is supposed to be realized around the ratio P_{c2}/P_{c1}=1.60. Decreasing the ratio P_{c2}/P_{c1} results in the increase of condensation temperature T_{c1} and the decrease of condensation temperature T_{c2}, and the effect of COP increase by the T_{c2} decrease is greater than the effect of COP decrease by the T_{c1} increase, and as a whole, COP tends to increase. If only T_{c1} can be varied keeping T_e and T_{c2} constant, reasonable results are expected to be obtained. In this analysis, this procedure is not possible, because of the input parameter is given in the form of ratio: P_{c2}/P_{c1}. This remains to be done in the future investigation.

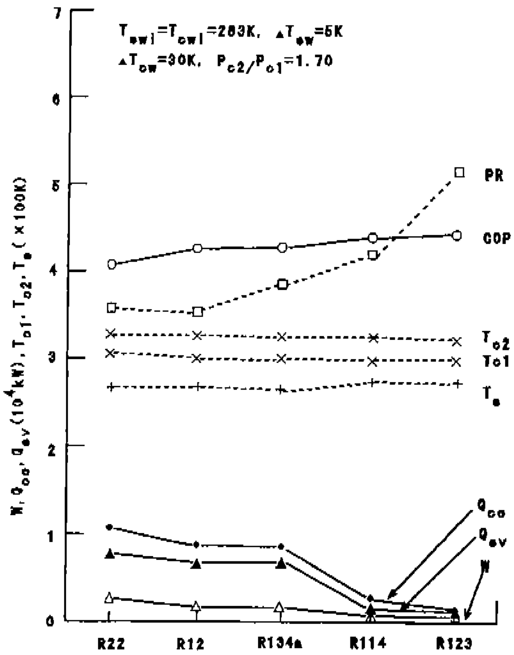


Fig.4 Effect of working fluid properties upon the thermal characteristics

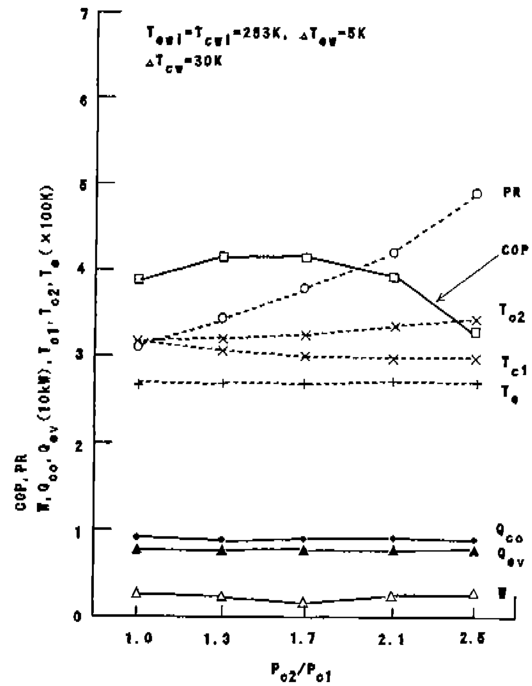


Fig.5 Effect of pressure ratio upon the thermal characteristics

EXERGY ANALYSIS

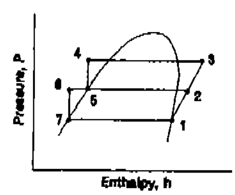
An example of exergy analysis is shown for the two stage condensation heat pump system analyzed in the previous section. Details for the analytical methods are based on the results in the reference [10]. The heat source temperatue T_{lm} =292K, the first heat sink temperature T_{h1} =294K, the second heat sink temperature T_{h2}=305K, the heat source exergy L_{ew}=8.08 KJ/kg, the low stage compressor exergy L_{c1} = 42.7 KJ/kg, the high stage compressor exergy L_{c2} =3.34 KJ/kg, the first heat sink exergy B_{h1} =5.53 KJ/kg, the second heat sink exergy B_{h2} =7.45 KJ/kg, flow quantity ratio $m_2/(m_1+m_2)$ =0.4 are given as an environmental conditions. Table 1 shows the set of formula to get the irreversible losses of heat pump components in the analysis. Fig.6 shows the exergy flow diagram obtained from the result for the two stage condensation heat pump system. From this figure, the irreversible losses for the first stage compressor and condenser ir_{cp1} & ir_{c1}, are shown to be considerably large. Improvements are expected to be done by optimizing the cycle balance points. It will be also effective to use a heat exchanger with a high performance heat transfer surface. The exergy efficiency η_E are given as follows.

$$1-\eta_E = \Sigma(ir_i)/(L_{c1}+L_{c2}+L_{ew}) \qquad (13)$$

$$\eta_E = 0.209$$

Table 1 Irreversible losses of heat pump system

evaporator	$ir_{71} = T_o(s_1-s_7)-Q_e(T_o/T_e)$
condenser	$ir_{25} = Q_{c1}(T_o/T_{c1})-T_o(s_2-s_6)$ $ir_{34} = Q_{c2}(T_o/T_{c2})-T_o(s_3-s_4)$
compressor	$ir_{12} = T_o(s_2-s_1)$ $ir_{23} = T_o(s_3-s_2)$
throttle valve	$ir_{45} = T_o(s_5-s_4)$ $ir_{67} = T_o(s_7-s_6)$



T_o : ambient temperature
T_e : evaporation temperature
T_{c1},T_{c2} :condensation temperature
s : entropy
h : enthalpy
Q_{c1},Q_{c2},Q_e : heat quantity

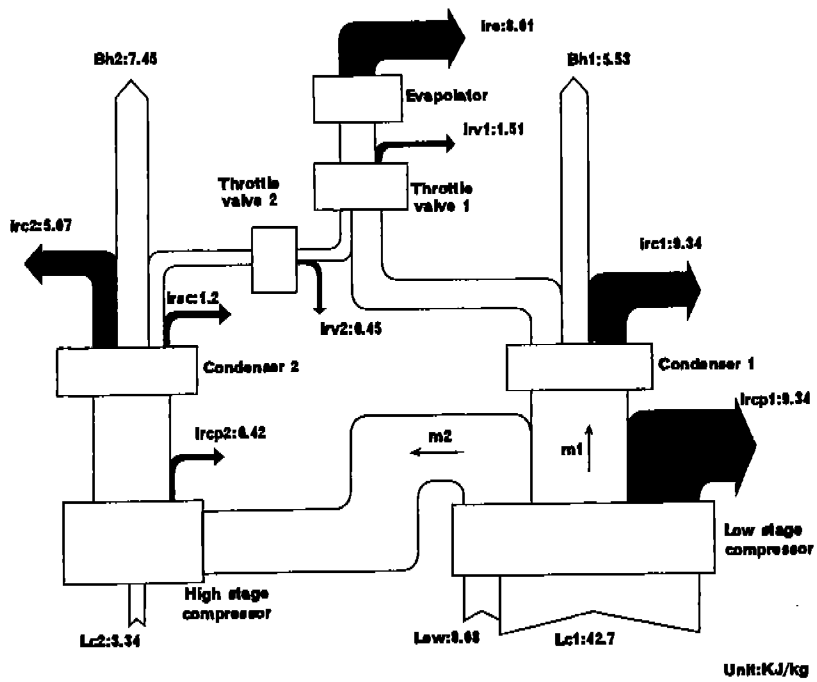


Fig.6 Exergy flow diagram of two stage condensation heat pump

CONCLUTIONS

The simulation analysis for the two stage compression two stage condensation cycle was undertaken, and the following results were obtained.

(1) Thermal performance characteristics of two stage compression two stage condensation cycle, one of the multistage heat pump cycle, is numerically analyzed in order to obtain a steady state solution of convergence. Cycle analysis is made by replacing the performance parameter of heat pump component into simplified modelling formula.

(2) The effect of condensation temperature, working fluid, and condensation pressure ratio upon the heat pump cycle performance are considered to obtain a useful information and knowledge in the design and evaluation of multistage heat pump systems.

(3) In the future developments, along with the consideration to apply to higher COP system simulation, namely multistage condensation heatpump systems, the analysis model will be used in computer program incorporating weather data and a building heat load calculation to determine heating output and the overall energy input required for various system configuration and controlled strategies.

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