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Modeling and Verification of a Vapor Compression Heat Pump

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MODELING AND VERIFICATION OF A VAPOR COMPRESSION HEAT PUMP

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ABSTRACT

The computer modeling of a water-to-water heat pump using a first principles approach is presented. State equations were developed for each component of the heat pump system using conservation laws. Both spatial and time response of the refrigerant mass inventory and flow, pressure and enthalpy are predicted throughout the heat pump circuit.

Good agreement is shown between the model and lab data for both transient and steady state performance. The cause and effect of various performance parameters during a start up are discussed.

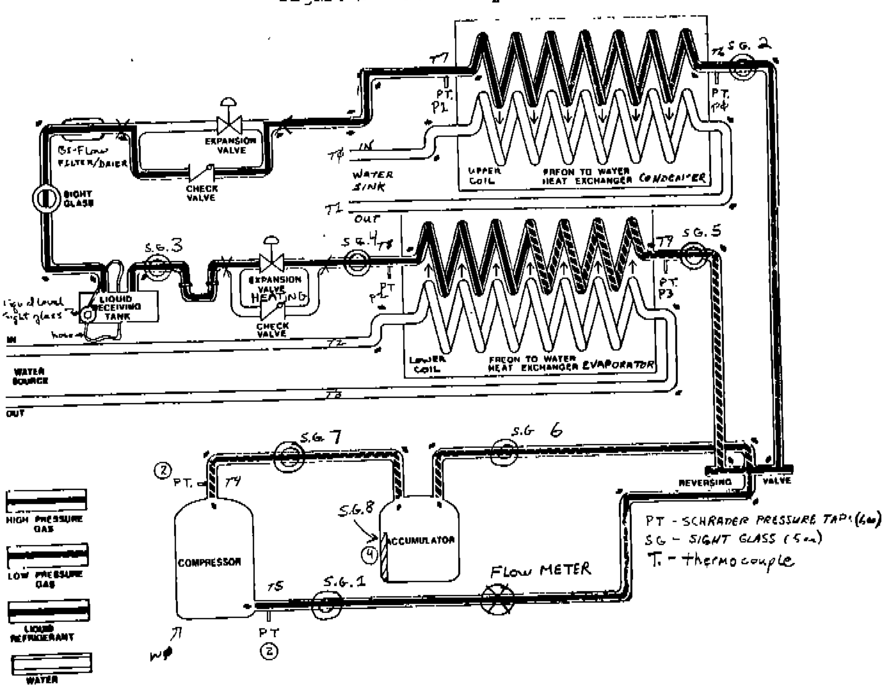
INTRODUCTION

As HVAC energy costs have continued to rise, so has the effort to increase the efficiency and reliability of heat pumps. The key to this effort lies in understanding the fundamental dynamics of the heat pump. This has been accomplished with the aid of a truly dynamic heat pump model. With this model, one may examine the time response of many variables that would otherwise be difficult or impossible to measure in a laboratory situation. The modeled equations allow one to understand the physical processes involved, while providing direction for future research.

1. PHYSICAL DESCRIPTION OF THE HEAT PUMP AND LAB SET UP

A schematic representation of the heat pump is shown in Figure 1.

Figure 1 - Heat Pump Schematic



The heat pump is a three ton water-to-water unit. The compressor is a Copeland reciprocating type. The unit has a "J" tube type accumulator. For lab tests used to verify the computer model, the thermal expansion valves that came with the unit were replaced with needle valves. The location of the temperature and pressure sensors are shown in Figure 1. The refrigerant flow was measured at the exit of the compressor. The flow meter used can only measure single phase flow, that is, all liquid or all vapor. Thus it was put at this location because it is the only location where the refrigerant is mostly single phase (all vapor) during transient conditions. In order to make a qualitative determination of the state of the refrigerant during the test, sight glasses were placed at key locations. Data from all the runs was obtained and stored under computer control.

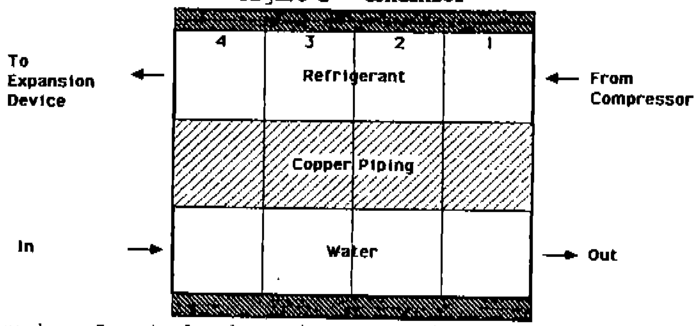
2. DESCRIPTION OF COMPUTER MODEL

The heat pump basically has five components; the compressor, the expansion device, two heat exchangers and the accumulator.

A. Heat Exchangers

Both heat exchangers (condenser and evaporator) are represented in a mathematically identical fashion. Each heat exchanger is broken up into control volumes with conservation of mass and energy used to express the relationship between the bulk parameter values for each control volume. Figure 2 shows a schematic representation of one of the heat exchangers (condenser) with the number of control volumes arbitrarily set to four.

Figure 2 - Condenser



As the number of control volumes increases, the bulk values represent more accurately the actual distributed values. However, the computational time increases with the number of control volumes. As a result, four control volumes are used for the condenser and three for the evaporator. This division proved adequate even though the model is set up to allow any division desired.

Mathematical representation /1/ of the condenser

a) Conservation of mass for each of the refrigerant control volumes takes the following form:

$$\dot m_1 = \dot m_{comp} - v_1 \frac{d\rho_1}{dt}, \qquad (1)$$

$$\dot m_2 = \dot m_1 - v_2 \frac{d\rho_2}{dt}, \qquad (2)$$

$$\vdots$$

$$\dot m_N = \dot m_{N-1} - v_N \frac{d\rho_N}{dt} \qquad (3)$$

b) Conservation of energy for each refrigerant control volume takes the following form:

$$m_1 \frac{dH_1}{dt} = \dot m_{comp} (H_{in} - H_1) - \dot m_{1out} (H_1 - H_{1out}) + hA (T_1 - T_{W1}) \qquad (4)$$

$$m_2 \frac{dH_2}{dt} = \dot{m}_{1out} (H_{1out} - H_2) - \dot{m}_{2out} (H_2 - H_{2out}) + hA (T_2 - T_{w2}) \tag{5}$$

$$\vdots$$

$$m_N \frac{dH_N}{dt} = \dot{m}_{N-1} (H_{N-1out} - H_N) - \dot{m}_N (H_N - H_{Nout}) + hA (T_N - T_{wN}) \tag{6}$$

Conservation of energy for both the piping and water control volumes are expressed in a similar fashion.

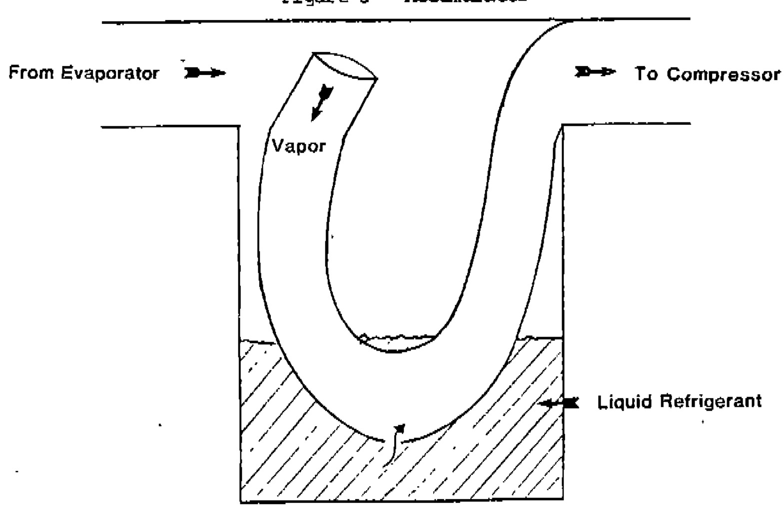
The pressure gradient along each heat exchanger was assumed negligible. Lab data indicated that the assumption of negligible pressure drop along each heat exchanger seemed reasonable.

The boundary conditions for the refrigerant side are determined by the mass flow and enthalpy entering from the compressor and the mass flow exiting to the expansion valve. For the water side, the boundary conditions are determined by the mass flow rate of water in addition to the entering and exiting water temperature. The heat transfer coefficient along the length of the heat exchanger for both the refrigerant and the water sides is assumed constant.

B. Accumulator

The accumulator was modeled basically as another control volume with the mass and energy balance equations having the same form as those in the heat exchanger. The influence of the "J" tube in the accumulator (see Figure 3) was assumed to affect only the enthalpy of the refrigerant leaving the control volume in the energy equation. In particular, in the heat exchanger, the enthalpy of the refrigerant leaving the control volume was assumed equal to the bulk enthalpy. For the accumulator, the following rationale was used: the bulk quality of the control volume was first determined using the pressure and the bulk enthalpy. If the quality was greater than or equal to one (all vapor), the enthalpy of the refrigerant leaving the control volume was set equal the bulk enthalpy. If the quality fell between a value of zero and one (part vapor and part liquid), the liquid was assumed to settle to the bottom of the control volume and the opening of the "J" tube at the top of the control volume would allow only vapor to leave. There is a hole in the "J" tube at its lowest point. Using the bulk quality, the liquid height from the bottom of the accumulator was determined. If the liquid height was below this hole, no liquid would be removed through it. In this case, the enthalpy of the leaving refrigerant would be set equal to the enthalpy of saturated vapor at that pressure. If the bulk quality decreased such that the liquid level rose above the hole in the "J" tube, liquid was assumed to enter through the hole. The rate of this liquid flow was proportional to the square root of the liquid height above the hole. In this case, the enthalpy of the leaving refrigerant was assumed to be equal to the adiabatic mixture of the vapor and liquid flows.

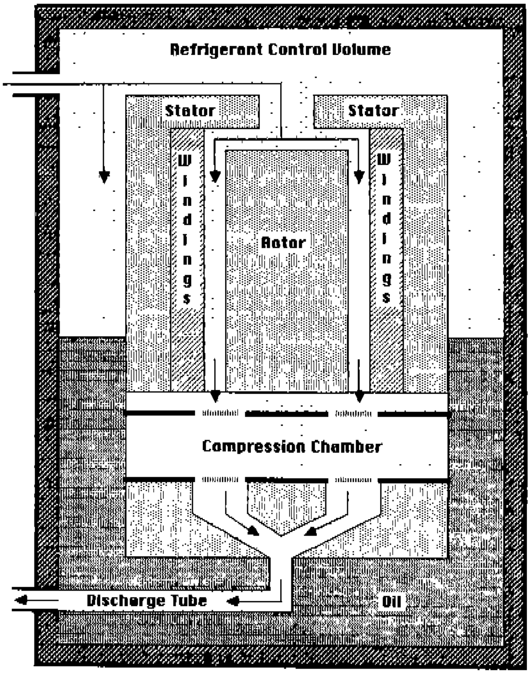
Figure 3 - Accumulator



C. Compressor

To better understand the internal workings and interactions in the compressor, an identical compressor was obtained and cut open for inspection. The volumes of various spaces which the refrigerant and oil would occupy as well as the refrigerant flow path, masses of the motor, compressor, shell and oil were obtained for use in the modeling equations. Figure 4 shows a schematic representation of the compressor that was used for the model.

Figure 4 - Compressor Model



The refrigerant flow through the compressor is as follows: The refrigerant enters through the upper left corner of the compressor shell. If liquid enters the control volume, a fraction of it is assumed to be carried along by the vapor into the motor. This liquid is assumed to mix adiabatically with the vapor flow entering the motor.

After the refrigerant enters the top of the motor, it flows through the annulus created by the stator and the rotor. The electric motor windings cover the outside surface of this annulus and so the refrigerant is in direct contact with the windings. This contact is intended as it helps cool the windings during operation.

After the refrigerant leaves this annulus, it goes through an identical pair of channels running through the compresion chamber leading to a set of reed valves. An isoenthalpic pressure drop is assumed across the reed valves.

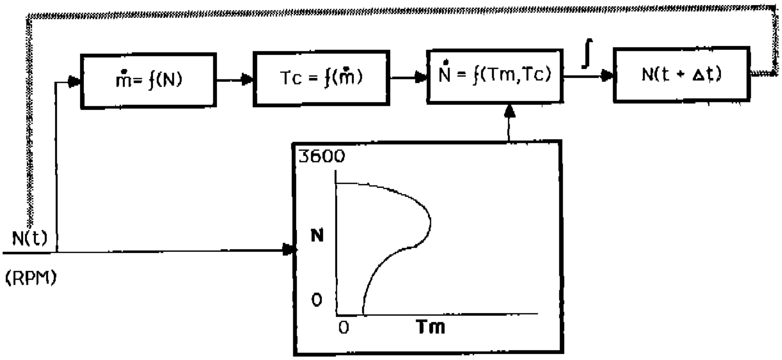
There is no restriction on the quality of the refrigerant entering the compression chamber. If liquid is present, as might happen at startup, it is assumed to remain liquid during the compression process and merely undergo a pressure increase. This liquid is assumed to mix adiabatically with the compressed vapor before leaving through the other set of reed valves. The power to pump the liquid is assumed negligible compared to the power to pump the vapor. The compression process is assumed to be polytropic. The refrigerant enthalpy increase can be evaluated by the equation derived from the expressions for isentropic and polytropic work of compression at the same compression ratio. The pressure drops across the reed valves are a function of the refrigerant mass flow.

The refrigerant then enters the discharge tube which runs through the oil. The reason for this is to evaporate out any refrigerant in the oil, returning it into the system. The refrigerant then leaves the exterior shell of the compressor. Heat exchange takes place between the component parts within the compressor shell and also with the environment. As a test of the compressor model performance, output results were compared to manufacturer supplied performance maps using the same inputs. The results show good agreement between the two.

To predict the refrigerant flow at any time, the following scheme (see Figure 5) was used: Knowing the present RPM value, the refrigerant mass flow rate was obtained and used to obtain the compressor braking torque needed. The compressor braking torque is obtained by equating the expression for the shaft power output to the expression for the power imparted to the refrigerant in pumping.

The available motor torque is obtained from a manufacturer's curve relating RPM to motor torque. The rate of change in RPM is obtained using the equation for conservation of angular momentum. This rate is integrated to find the predicted RPM value for use at the next time step.

Figure 5 - RPM



Note that this scheme will handle those cases where the motor will not turn due to high compressor braking torque. This may happen when the system is shut off and then restarted too soon before the pressures have equalized.

D. Expansion Device

The needle valve used was assumed to function as a basic orifice. The equation used to calculate mass flow rate was the standard orifice equation.

$$m = C \sqrt{\rho \Delta P} \qquad (7)$$

The pressure drop and density used in the above equation depend on the fluid inlet conditions.

3. QUALITATIVE INTERPRETATION OF RESULTS OBTAINED FROM LAB GENERATED DATA

Prior to the test, as much of the refrigerant as possible was moved to the evaporator by heating the accumulator, receiver, compressor, and condenser to evaporate any liquid residing there. This was done in order to get a known refrigerant mass inventory to be used as initial conditions for the test. The test proceeded as follows: The unit was left off for at least 24 hours at room temperature. The compressor was then turned on and the flows, temperatures, and pressures were taken until an equilibrium state was reached (approximately 1/2 hour).

Immediately after the compressor was started, it was noted that liquid refrigerant was seen both entering (sight glass 6, Fig. 1) and leaving (sight glass 7) the accumulator. The refrigerant at these two points appeared as a froth. This effect ended abruptly after about one half minute. At the end of this time, the liquid refrigerant level in the accumulator had built up to a height of approximately three and one half inches.

This accumulation is the due to the "J" tube in the accumulator. The fact that a significant amount of liquid did leave the accumulator and subsequently entered the compressor shell points to the inefficiency of the "J" tube in this situation to restrict liquid flow to the compressor. The reason for this flow of liquid at start up can be explained in the following way: At start up, most of the refrigerant charge resided as liquid in the evaporator. When the compressor was turned on, it quickly evacuated the vapor in the compressor shell. This localized sudden decrease in pressure pulled the refrigerant, both liquid and vapor, from the evaporator into the accumulator, and from there into the compressor shell. The sudden decrease in pressure caused the liquid to boil violently, creating the froth-like appearance.

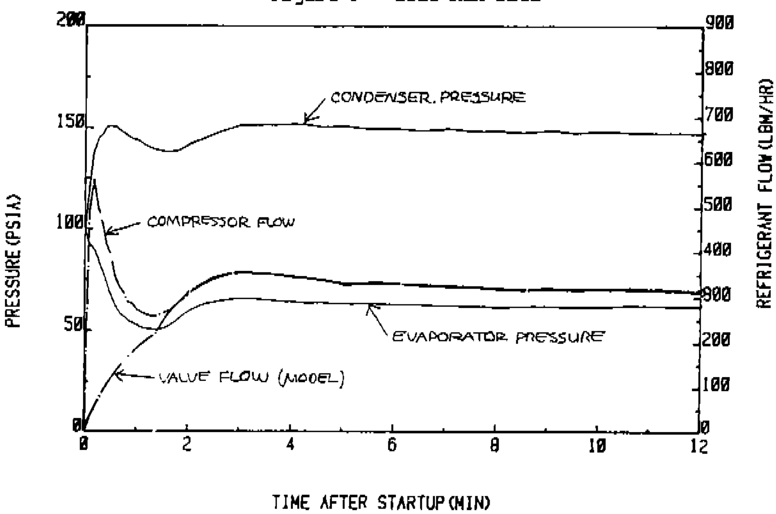
As the test proceeded, the liquid level in the accumulator (three and one half inches at one half minute) decreased slowly until it was all gone after about nine minutes. The removal of the liquid from the accumulator was due to a combination of the liquid being metered directly through the hole at the bottom of the "J" tube, heat transfer from the environment evaporating the liquid and also because of heat transfer from superheated vapor which entered the accumulator.

The refrigerant as seen just upstream of the needle valve (sight glass 3) started out clear (all vapor) decreasing in quality until after one and one half minutes it became all liquid.

Data evaluation

The pressure and flows obtained from this test run are shown in Figure 6.

Figure 6 - Test Run Data



The variation in condenser pressure can be explained using the fact that the rate of change of refrigerant pressure is a function of the rate of change of refrigerant vapor density, as follows:

The initial steep rise in the condenser pressure is due to the high rate of change of vapor density. The vapor density increase is due to the high flow of vapor coming from the compressor with very little vapor leaving through the needle valve. As the pressure rises, the refrigerant temperature rises, increasing the heat transfer and the rate of condensation. The increased condensation reduces the rate of change in vapor density, reducing the pressure rise.

As time proceeds, the quality of the refrigerant that leaves the condenser through the valve continues to decrease until it becomes all liquid. As the valve flow becomes more liquid it has less of an effect on the rate of change of vapor density and therefore the rate of change of condenser pressure. In fact, the rate of change of condenser pressure eventually only depends on the rate of change of vapor due to the flow of vapor in from the compressor and the vapor lost due to condensation.

From about one half minute to one and one half minutes, the rate of condensation is greater than the rate of vapor influx from the compressor. This caused the pressure to decrease. Note that if there was no condensation, the pressure would have continued to rise during this time because there is still a net influx of vapor.

After one and one half minutes, the compressor flow again increases, causing the vapor density to increase. This causes the pressure to rise until condensation catches up and equilibrium is reached.

The same rationale can be used to explain the pressure variation in the evaporator. Note that the initial fall in the evaporator pressure is less steep than in the condenser. This is because of the rapid evaporation of liquid into vapor in the evaporator. This effectively reduces the rate of change in vapor density which reduces the pressure fall. As time proceeds, the refrigerant flow entering the evaporator is less vapor due to the decrease in quality of the refrigerant entering the valve from the condenser. Eventually the valve flow becomes approximately 20% vapor at steady state. This means that the evaporator pressure change is not entirely decoupled from the valve flow as it is in the condenser, but, for the most part, follows the compressor flow and evaporation rate.

Since most of the liquid in the evaporator has been pulled out after one half minute and since the valve flow is now letting only a little liquid in, the rate of vapor increase due to evaporation is low. In fact, most of the refrigerant is superheated with little heat transfer taking place. This makes the rate of change of pressure mainly a function of the net outflow of vapor. This is evident where the evaporator pressure drops due to the net loss of vapor exiting to the compressor. As the compressor flow starts to bottom out at about one and one half minutes, so does the decrease in evaporator pressure. After one and one half minutes, the valve flow is seen to increase rapidly, approaching the compressor flow value. This causes the evaporator pressure to increase. The reason for this pressure increase is due to the sudden and rapid increase in flow of low quality refrigerant entering the evaporator from the valve. This "reflooding" of the evaporator greatly increases the heat transfer in the evaporator. As the pressure rises, the evaporation rate decreases until an equilibrium is reached.

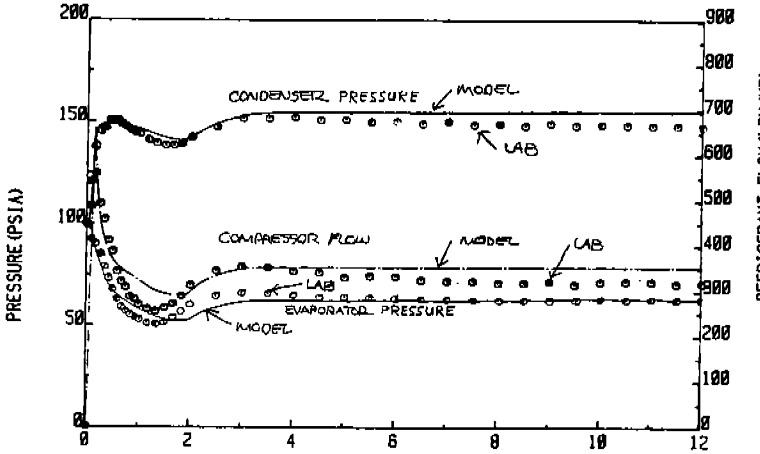
4. COMPARISON OF LABORATORY VERSUS COMPUTER PREDICTED RESULTS

Table 1 shows the values at steady state of various parameters obtained from laboratory data and the model indicating very good agreement. Figure 7 shows the model versus Laboratory data for pressure and flows, also indicating good agreement.

Table I. Comparison Between Model and Laboratory Data for Source and Sink Temperatures of 50 and 50°F, respectively.

PARAMETER	MODEL	DATA
Heat Rejection From Condenser (Btu/HR)	31650	32770
Input Power To Compressor (kW)	2.13	2.20
Refrigerant Mass Flow (lbm/hr)	322	336
Condenser Pressure (Psia)	160	156
Evaporator Pressure (Psia)	61	67
Compressor Discharge Temperature (°F)	160	179
Suction Superheat (°F)	2.6	2.6
Heat Delivery Time Constant (Min.)	0.75	0.75
System COP (No Auxiliary)	4.35	4.36

Figure 7 - Comparison of Model and Laboratory Data



CONCLUSION

A dynamic model of a heat pump has been developed and verified. The agreement between the model and laboratory data has been shown to be quite good even with the simplifying assumptions used in the model. As such, the model can be used for analyzing tradeoffs and trends using various advanced control scenarios and component modifications. The process of selecting and prioritizing potential control scenarios has been made easier by the understanding of heat pump dynamics gained through the modeling exercise.

SYMBOLS

m = mass flow rate
v = heat exchanger volume
ρ = refrigerant density
t = time
H = refrigerant enthalpy
h = heat transfer coefficient
A = heat exchanger area
T = refrigerant temperature
Tw = heat exchanger wall temperature
P = refrigerant pressure
c = constant

REFERENCES

1. Amin Patani: Dynamic Simulation of Electric Heat Pumps. Proprietary Honeywell Memo, HR-R-81-21:14-60, April 13, 1981.

MODELAGE ET VERIFICATION D'UNE POMPE À CHALEUR TYPE COMPRESSION - VAPEUR

On a présenté un modele mathématique d'une pompe à chaleur eau/eau, établi à partir des principes de base. On a développé les équations d'etat pour chacun des composants de la pompe en utilisant les lois de la conservation. On a calculé à l'avance, à travers le circuit de la pompe à chaleur, débit, pression, et enthalpie ainsi que les réponses dans le temps et dans l'espace de l' ensemble de la masse de réfrigérant. On a démontré une bonne correspondance entre le modele mathématique et les données du laboratoire à la fois pour les états transitoires et le régime continu. On a examiné les causes et les effets des differents paramètres durant la phase de démarrage.

IMPROVEMENT AND CHECKING OF A COMPRESSION STEAM HEAT PUMP

Starting from basic base a mathematical model for a water/water heat pump is proposed. Using conservation laws, state equations has been expanded for each component of the heat pump. Output, pressure, enthalpie, as space and time feedback answer for mass and refrigerant package as been calculated at first in each part of the heat pump circuit. For both unceasing and temporary states a good correlation between mathematical model and experimental results has been pointed out. During starting phase causes and effects of various parameters has been studied.