ENERGY AUDIT – IIT(ISM) DHANBAD LIBRARY REFRIGERATION SYSTEM

- Narendra Gavali
- Hardik Sharma
- Dharmesh Meena

Contents

List of figures

ABSTRACT

Refrigeration systems are integral part of modern-day buildings. As life style standard is increasing with overall development across the world, the need for better understanding of refrigeration systems is crucial in energy auditing and management. Centralized air conditioning systems gives us opportunity to monitor and operate a single system efficiently which is quite easier instead of performing the same task for smaller ACs at individual cabins or offices. Operating such huge systems efficiently is necessary, as even minute rise in efficiency can amount to huge savings due to gigantic size in terms of heat duty control to an entire building.

This report performs energy audit for centralized air conditioning system of IIT ISM Dhanbad library. It explains the concept of vapour compression cycle which governs the refrigeration systems of buildings. Using ASPEN simulation software heat balance for the system was calculated. After thoroughly understanding the operation, we managed to find some critical gaps. They were mainly due to operational and continued maintenance lapses which is the need of any machinery. We quantified potential savings which can result from rectifying one of the gaps. In this the system's cooling water pump was being operated at higher pressure drop than the standard value. After quantifying the savings understandable suggestions were proposed.

INTRODUCTION

WORKING OF VAPOUR COMPRESSION CYCLE

The vapor-compression refrigeration cycle is represented in Fig. 1, along with a TS diagram showing the four steps of the process. A liquid refrigerant evaporating at constant T and P absorbs heat (line 1 → 2), producing the refrigeration effect. The vapor produced is compressed via dashed line 2 → 3′ for isentropic compression (Fig. 7.6), and via line 2 → 3, sloping in the direction of increasing entropy, for an actual compression process, reflecting inherent irreversibility's. At this higher T and P, it is cooled and condensed (line 3 → 4) with rejection of heat to the surroundings. Liquid from the condenser expands (line 4 → 1) to its original pressure. In principle, this can be carried out in a turbine from which work is obtained. However, for practical reasons it is usually accomplished by throttling through a partly open control valve. The pressure drop in this irreversible process results from fluid friction in the valve. The throttling process occurs at constant enthalpy.

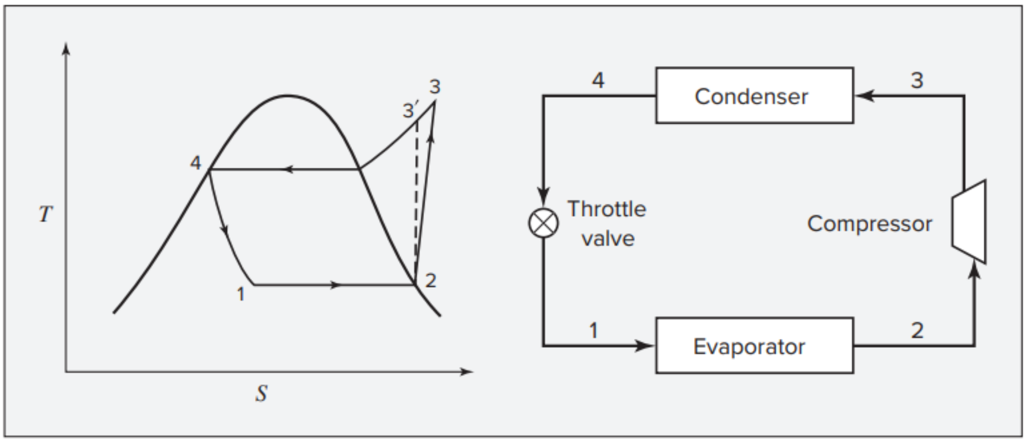

Figure 1 Vapor-compression refrigeration cycle

Based on a unit mass of fluid, the equations for the heat absorbed in the evaporator and the heat rejected in the condenser are:

Equation 1

$Q_C = H_2 - H_1$ and $Q_H = H_4 - H_3$

The small changes in potential and kinetic energy are neglected. The work of compression is simply: $W = H_3 - H_2$, and by Eq. 3, the coefficient of performance is:

Equation 2

$$\omega = (H_2 - H_1)/(H_3 - H_2)$$

Equation 3

$$W = -(Q_C + Q_H)$$

Equation 4

$$\omega = \frac{\text{heat absorbed at the lower temperature}}{\text{net work}} = \frac{QC}{W}$$

To design the evaporator, compressor, condenser, and auxiliary equipment one must know the rate of circulation of refrigerant $\dot{m}$. This is determined from the rate of heat absorption in the evaporator by the equation:

Equation 5

$$\dot{m} = \dot{Q}_C / (H2 - H1)$$

The vapor-compression cycle of Fig. 1 is shown on a PH diagram in Fig. 2, a diagram commonly used in the description of refrigeration processes, because it shows the required enthalpies directly. Although the evaporation and condensation processes are represented by constant-pressure paths, small pressure drops do occur because of fluid friction. For given T_C and T_H, vapor-compression refrigeration results in lower values of ω than the Carnot cycle because of irreversibility's in expansion and compression.

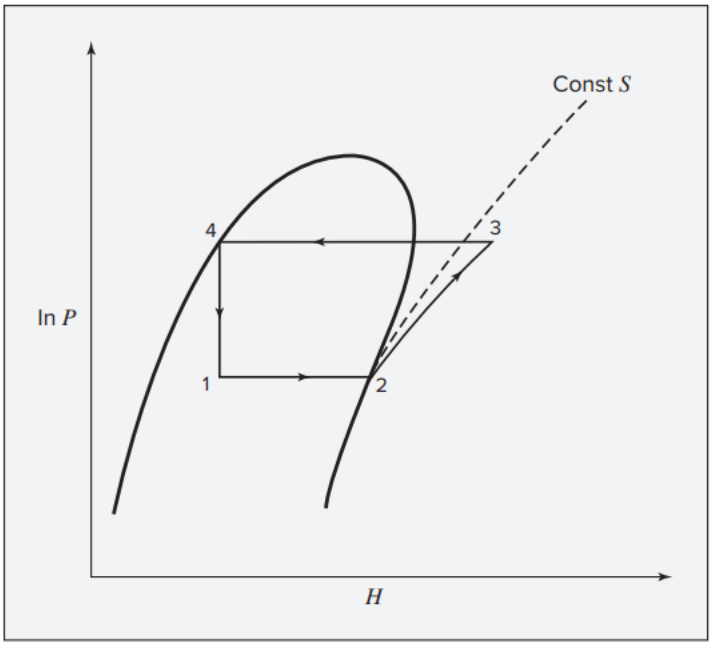

Figure 2 Vapor-compression refrigeration cycle on a PH diagram

LIBRARY REFRIGERATION SYSTEM DESCRIPTION

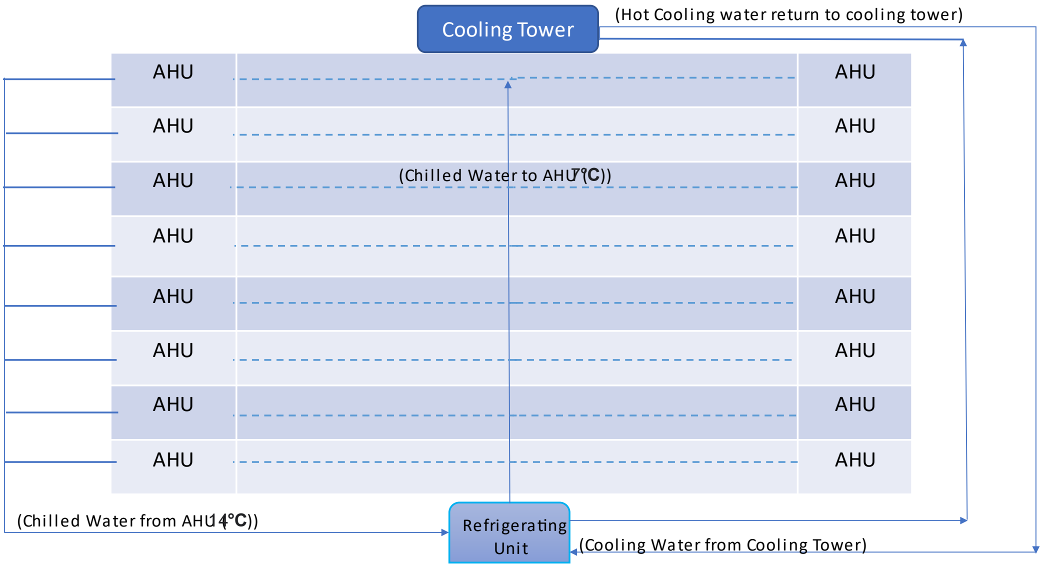

Figure 3 - Block diagram of refrigeration system

A systematic process in the library consists of Air-handling unit (AHU) rooms, Cooling towers and Refrigeration unit at ground floor. Cooling towers are placed on the roof of the library that is used to supply cooling water to the refrigeration units.

The conditioned air inside the library comes from the AHU rooms present on every floor. At each floor's ceilings, there are two types of vents; one is to suck the hot air from the floors that go to the heat exchanger as the pressure of the AHU room is low due to negative pressure created by suction of Air to water heat exchanger. Chilled water from the refrigerating system cools the air in an Air to water heat exchanger, and we pump back the conditioned air to the floors through the vents.

The Hot, chilled water from the AHU at 12°C returned to the evaporator, a Shell and tube heat exchanger. We use R-134a or 1,1,1,2-Tetrafluoroethane as refrigerant to cool the water. Evaporator cools the chilled water to 7°C and returns it to the AHU units.

After that, the refrigerant goes to Semi Hermitic Screw compressor, compressing up to 12 bar and 60°C. The compressed refrigerant along with lubricant oil FVC68D of compressor goes to an oil separator. The oil-free & superheated refrigerant transfers heat to cooling water in a condenser, which helps it to drop the temperature up to 38°C. Then pressurise and condensed refrigerant at around 9 bar throttles to 1.5bar. Due to Joule Thomson expansion the refrigerant reaches to -17°C at the outlet of Throttling valve. The cold working fluid or refrigerant again goes to evaporator to continue the vapour compression cycle.

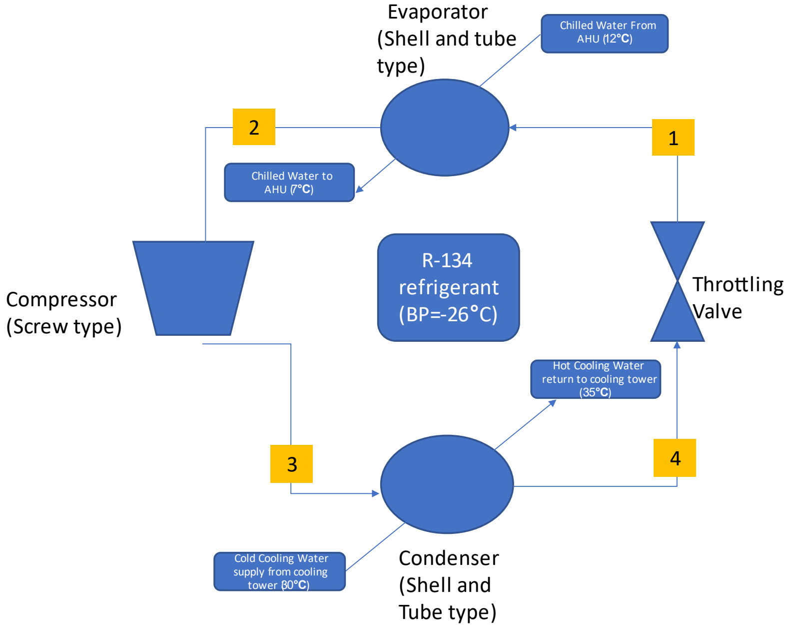

Figure 4 - Vapour compression cycle

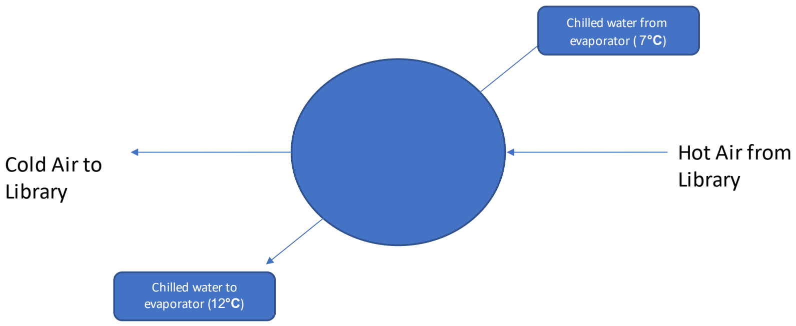

Figure 5 – Air to water heat exchanger or Air Handling Unit (AHU)

ASPEN SIMULATION

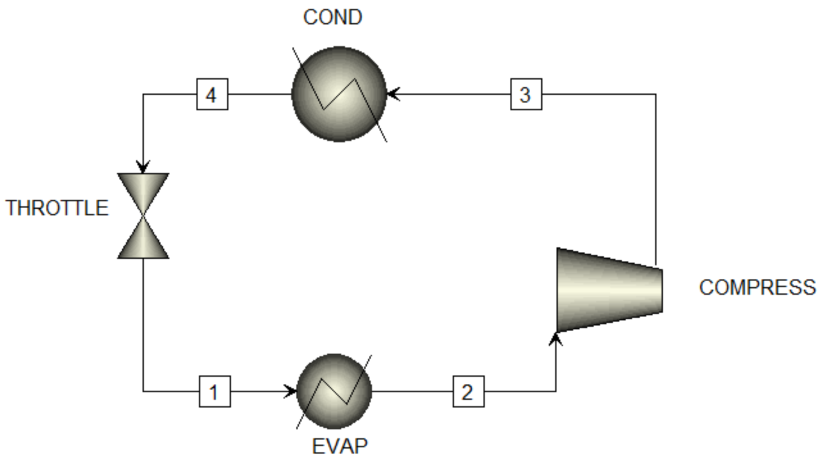

Figure 6 - ASPEN Flowsheet

The main flowsheet was created in ASPEN version 14 using blocks of Compressor, Condenser, Evaporator and Throttling valve. REFPROP thermodynamics was used to as method, as it is commonly used thermodynamic model for simulations of commercial refrigerants.

Properties specified for equipments in Simulation window:

- Evaporator
  - Pressure = 268Kpa – Value taken from system display.
  - Vapor fraction = 1
- Compressor
  - Discharge pressure = 1200Kpa – We measured the temperature of discharge line from compressor to be 60°C. we varied the pressure in ASPEN and it was at 1200Kpa that temperature came out to be 60°C.
  - Compressor model - Polytropic using ASME method. As this is the most rigorous model to explain the compressor behaviour.
- Condenser
  - Pressure = 968Kpa – Value taken from system display.
  - Vapor fraction = 0
- Throttling valve
  - Outlet pressure = 150KPa. We assumed this value. As the evaporator pressure is 268Kpa the process line leading to it should have less pressure.

After entering these values, the flowsheet's degree of freedom was zero, meaning all the required inputs were complete to define the system and flowsheet is ready to run for simulation.

SIMULATION RESULTS

		Material			
Stream Name	**Units**	**1**	**2**	**3**	**4**
Description					
From		THROTTLE	EVAP	COMPRESS	COND
To		EVAP	COMPRESS	COND	THROTTLE
Stream Class		CONVEN	CONVEN	CONVEN	CONVEN
Phase			Vapor Phase	Vapor Phase	Liquid Phase
Temperature	C	-17.13222719	-2.418242941	64.10736601	38.18484697
Pressure	bar	1.5	2.68	12	9.68
Molar Vapor Fraction		0.361908164	1	1	0
Molar Liquid Fraction		0.638091836	0	0	1
Molar Enthalpy	cal/mol	-218163.1687	-214666.6162	-213566.5118	-218163.1687

Result summary window solves for simulation and the required properties of each stream were displayed. Molar enthalpy row shows enthalpy of each stream in Cal/moles of refrigerant. This completes the Energy balance of the system.

CARNOT COP CALCULATIONS

- $COP = \frac{Heat\ absorbed\ at\ lower\ temperature}{Net\ Work} = \frac{Qc}{W} = \frac{Tc}{Th - Tc}$

Operating Temperature

- Condenser = 38 °C = 311K = Th
- Evaporator = -2°C = 270K = Tc

$$COP = \frac{270K}{(311 - 270)K} = 6.58$$

GAP ANALYSIS

After understanding the overall operation of the refrigeration system, we identified gaps that are as follows,

- Each AHU or Air Water Heat exchanger has temperature controller, which is coupled with Chilled water flowrate from the refrigeration unit. Meaning that the flowrate of Chilled water to warm air is regulated using a Solenoid valve. But on 1st and 2nd floor the controller work not operation as conveyed by the operator. Hence Whenever temperature varies for the conditioned air from requirement, Operators manually switch off/on the condenser water flow at the refrigeration cycle of ground floor. Due to that entire system which could have been well regulated is being operated manually.
- AHU blower belts on some floors need maintenance. But to avoid the maintenance, the operators again manually operate the system in the similar fashion explained in previous point. Hence proper maintenance can totally avoid the scenario to result in smooth operation of the machinery.
- The rated pressure drop across the condenser is given in system manual as 0.7Kg/cm^2. But on field it is at 1.1 Kg/cm^2. Resulting in extra pumping cost for a pressure drop of 0.4 Kg/cm^2. We have shown the potential savings if we rectify this gap in next section of the report.

PUMPING COST DUE TO INCREASED PRESSURE DROP

$$Power\ consumed\ by\ centrifugal\ pumps(Watts) = \frac{Flowrate(\frac{m3}{s})\ *pressure\ drop(Pa)}{Efficiency\ of\ the\ pump}$$

Assumptions –

1. Typical velocities of Process cooling water = 1.5 – 2.5m/s
2. Typical Efficiencies of centrifugal pumps = 55 – 70%
3. Days of operation per annum = 300
4. Working hours per day = 15
5. Cost of electricity = 12 Rs/Unit

Assumptions 1 and 2 were made by referring to the textbook of Chemical Engineering titled as "Unit Operations of Chemical Engineering" by Warren McCabe, Julian Smith , Peter Harriott.

Measurements –

1. Pressure drop readings across Supply and Return lines of cooling water = 1.1Kg/cm^2
2. Rated pressure drop = 0.7kg/cm^2
3. Cooling water process line diameter = 10 inches

Equation 6

$$Cost\ (Rs) = Power\ (Watts) * Working\ hours\ \left(\frac{Hrs}{day}\right) * Operational\ days\ \left(\frac{Days}{annum}\right) * Electricity\ rate(\frac{Rs}{Unit\ of\ electricity})$$

SENSITIVITY ANALYSIS

All the values were added in MS Excel. A sensitivity analysis was done to calculate the savings at varied velocities from 1.5 – 2.5m/s. The table is given below.

Velocity	**Power**	**Energy**	**Cost**
m/s	KW	Units	Lakhs/annum
1.5	4.257072	19156.8	2.3
1.7	4.824681	21711.1	2.6
1.9	5.392291	24265.3	2.9
2.1	5.959901	26819.6	3.2
2.3	6.52751	29373.8	3.5
2.5	7.09512	31928.0	3.8

Suggestions

1. Condenser descaling interval should be increased from current state. Currently once a year is the period of descaling. It should be increased to 2 to 3 times a year.
2. Continued maintenance of all the machinery should be ensured so as carry out smooth operations. As mentioned in gap analysis, Controller at each AHU and Blower belts should be properly maintained.

THANK YOU.