ABUAD Journal of Engineering Research and Development (AJERD)
ISSN (online): 2645-2685; ISSN (print): 2756-6811

Volume 6, Issue 2, 218-229

Performance Evaluation of Vapour Compression Cascade Refrigeration System for Storing Blood Plasma

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Date Submitted: 20/11/2023
Date Accepted: 19/12/2023
Date Published: 30/12/2023

Abstract: *A cascade refrigeration system consists of two cycles to get the greatest possible refrigeration effect for applications requiring extremely low temperatures. The experimental examination of the cascade refrigeration system utilized for quickly freezing and storing blood products, notably plasma, was the main emphasis of this paper. This system is able to manage and apply cold storage for plasma at temperatures as low as -35^{o}C. R410A (low temperature cycle) and R404A (high temperature cycle) are the working fluids in this system respectively, and their selection was based on their reasonable boiling point differences and rapid freezing times. These working fluids have very little ozone depletion and global warming potential, and they are non-toxic to the environment. This system performance evaluation determines the cascade refrigeration system's parameters ranging effects of variation in evaporator temperature, condenser temperature and temperature difference in cascade condenser. As the evaporator temperature increases from –35^{o}C to -7^{0}C at interval of 7^{o}C, the suction pressure increased from 0.22 MPa to 0.64 MPa. The decrease in compressor discharge temperature of 30^{o}C to 26^{o}C resulted to decrease in discharge pressure ranging 1.62 to 1.49MPa in the high temperature cycle at constant condenser temperature of 40 ^{0}C. At varying heat exchanger temperature, all parameters reduced with exception of discharge temperature in both cycles and evaporator heat. A substantial increase in the coefficient of performance of comparable systems was found to be 3.04 (LTC) and 7.7 (HTC). At the high temperature cycle condenser, there was a total heat rejection of 6228 kJ, which could then be recycled for heating.*

Keywords: *Performance Analysis, Refrigerating Effect, System Parameters, Plasma and Heat Rejection.*

1. INTRODUCTION

Food, pharmaceuticals, and medical supplies are frequently stored and distributed using refrigeration systems to preserve their quality and freshness for a long time before usage. The quality and retention of coagulation protein in blood products are significantly influenced by post-collection management. Blood products are kept in refrigerators, which also serve as storage and freezing units. The cascade refrigeration system was essentially invented as a result of the limitations of intended ultra-low temperature applications. Pictet utilized cascade refrigeration for the first time in 1877 to liquefy O_2, SO_2, and CO_2 as intermediate refrigerants. Three-stage cascade refrigeration systems with ammonia, ethylene, and methane are also often used to liquefy gases [1]. The cascade refrigeration system is among the most promising low-temperature technologies because it has an energy-efficient cycle that lowers operating costs while achieving rapid and homogenous freezing and maintaining desired storage temperatures [2]. It operates at temperatures ranging from -18^{0}C to -86^{o}C depending on the refrigerants selected and design calculations. The cascade system can be used to handle a variety of heat-sensitive temperature challenges, including the absolutely necessary handling of vaccinations and blood products, but it requires a reliable storage container and a potent refrigeration system. One refrigerant in the system condenses the other primary refrigerant to the necessary evaporator temperature [3]. Some advantages of cascade refrigeration systems include keeping things at the proper temperature, increases freezing capacity, results in long-term and secure storage, and maintain high-quality products, aiding continued storage at extremely low temperatures. Hence, refrigerator storage is used to extend product shelf lives rather than discarding them [4]. The purposeful aim of quick freezing and storage of blood plasma proteins for an extension of shelf life led to the choice of R410A and R404A refrigerants in this investigation. Among other things, the following considerations were taken into account by a better mass flow rate, stronger cooling effect, higher discharge pressure, a shorter time to reach low temperatures, better coefficient of performance, cheapness and availability, as well as favourable existing refrigeration components R410A and R404A [5, 6]. The reviewed literatures showed that there were high demand and scarcity of plasma for life-saving transfusions in case of emergency. The paper provided a portable and motorized plasma processing and storage machine for fresh plasma management. This keeps the qualities of plasma intact (coagulation factors and total plasma qualities) after frozen which could be utilized at all levels of health centres as required by the world health organisation to address casualty cases.

1.1 The Two-Stage Cascade Refrigeration System's Operation

Figure 1 depicts the cascade refrigeration system and its components. The thermodynamic link between the two phases of the lower temperature (pressure) cycle and higher temperature (pressure) cycle is a cascade condenser. The main components of the cascade system are the Low Temperature Compressor, High Temperature Compressor, Condenser, Evaporator, Cascade Condenser (Heat Exchanger), and Throttling Devices for the Low Temperature Cycle (LTC) and the High Temperature Cycle (HTC). The low temperature cycle refrigerant is isentropically compressed in processes 1-2, which is then passed through a cascade condenser where it heats the higher temperature cycle refrigerant (processes 3-4), expands in a throttling device (processes 3-4), and then continues to the evaporator (processes 4-1) to produce the necessary refrigerating effect. The compression of the high temperature cycle is a part of processes 5 and 6 [7].

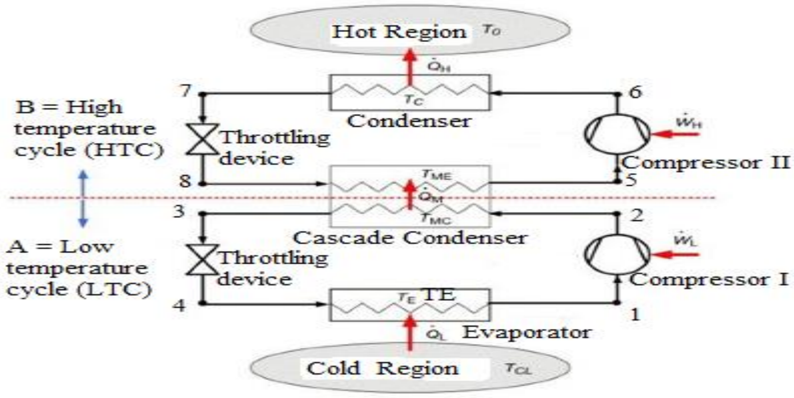

Figure 1: A two-stage cascade refrigeration [5]

1.2 Effects of Refrigerants

Refrigerants play a crucial and unique role in the performance and efficiency of cascade refrigeration systems. The physiochemical and thermodynamic characteristics of refrigerants R410A and R404A, such as boiling point, freezing point, molecular weight, critical temperature, critical pressure, critical volume, and design target in industries warranted the choice. Due to qualities like minimum GWP and ODP non-toxic to the environment refrigerants that help to construct a compact system as well as increase compressor efficiency, the pharmaceutical and medical industries are very concerned about short-term freezing refrigerants and on natural refrigerants [8]. In order to compare natural refrigerants with synthetic ones in terms of performance and environmental considerations, a theoretical evaluation of real-world natural refrigerant alternatives for ultra-low temperature applications was conducted. The natural refrigerant R1270/R170 pair has a 5% higher coefficient of performance (COP) and about 50% lower CO_2 emissions when compared to the synthetic refrigerant R404A/R508B pair. The performance of the cascade systems for Ultra Low Temperature (ULT) between -50^{0}C and -100^{0}C was evaluated that gave optimal value using a theoretical model and constructed Engineering Equation Solver (EES) software [9].

An innovative theoretical analysis of an ACCRS—an absorption-compression cascade refrigeration system—was carried out for low temperature cooling applications. The lower temperature segment features vapour compression refrigeration (VCR) system using R1234yf refrigerant, and the higher temperature section has a triple-effect H2O-LiBr series flow vapour absorption refrigeration system. An algorithmic model built on the EES software has been created to calculate performance parameters [10].

1.3 Working Fluids

The working fluids (refrigerants), R410A and R404A, were carefully chosen due to their affordability, availability, and environmental friendliness. They also have satisfactory and appealing thermal stability. The time it takes to reach the correct temperature is very important in many industries and medical fields. In this quick homogenous freezing and storing of fresh blood plasma, the refrigerants R410A and R404A [11, 12] were chosen out for the reason that they could;

i. R410A performs marginally better in terms of cooling effect and discharge pressure than R22.
ii. R404A is advised for achieving low temperatures quickly.
iii. R410A is chosen above R22 for domestic refrigeration purposes.
iv. R404A has a higher mass flow rate than R410A and R22

1.4 Thermodynamic Properties and Justification for the Selected Refrigerants

The thermodynamic properties of the working fluids are presented in Table 1. The choice of refrigerants R410A and R404A for low temperature cycle (LTC) and high temperature cycle (HTC) were their freezing points of -51.6^{o}C and -46.6^{o}C respectively. Other useful properties such as latent heat, specific heat liquid and vapour, critical temperature and pressures were stated in Table 1 for suitable system modelling. The zeotropic HFC refrigerant R404A is a 44/52/4 weight percent mixture of R-125, R143a, and R-134a. It is extensively used in low- and medium-temperature refrigeration applications. R404A is a mixture of R125 - R143a - R134a for the purpose of both domestic and industrial refrigeration applications. Refrigeration applications that require ultra-low temperatures use R404A due to its compatibility with existing refrigeration components and it can be retrofitted into R502 systems [13]. The chosen refrigerant mixture also has distinctive qualities, design temperature, boiling temperature point, freezing temperature point, ODP and GWP satisfaction, moderate cost and availability and stable and safety.

Table 1: Thermodynamic properties of R404A and R410A as working fluids

Refrigerant	**Units**	**R410A**		**R404A**	
Type		HFC		HFC	
CAS Number		AZ-20, Puron, Suva 9100		HP-62, FX-70	
Chemical name		R-32/125		R-125/143/134A	
Molecular Formula		$CH_2F_2+CHF_2CF_3$		$C_2HF_5+C_2H_3F_3+C_2H_2F_4$	
Component		R125-R32		R125 - R143a - R134a	
Composition	%by weight	50 % - 50 %		52 % - 44 % - 4 %	
Molecular mass		72.6		97.6	
OEL ppm (v/v)		1000		1000	
RCL (ppm(v/v))		140,000		130,000	
DLH (g/m^3)		420		500	
ODP		0		0	
Atmospheric life time	Years	16.95		40.36	
Critical temperature	^oC	71.8		72.1	
Critical pressure	MPa	48.9		37.3	
Boiling point	^oC	-51.6		-46.6/-45.8	
Temperature	^oC	-30	30	-30	30
Temperature glide	K	0.06	0.11	0.71	0.42
Sat. pressure	Kpa	270.32	1876.91	204.95	1425.91
Specific heat liquid	kJ/kg-K	1.40	1.75	1.25	1.56
Specific heat Vapour	kJ/kg-K	0.91	1.51	0.82	1.24
Latent heat	kJ/kg	252.52	197.25	183.56	130.19
Density liquid	kJ/m^3	1280.6	1035.4	1282.3	1040.7
Density sat. Vapour	kJ/m^3	10.57	76.11	10.98	78.15
Volumetric capacity	kJ/m^3	2816.4	-	2155.8	-

1.5 Blood Plasma Storage

Red blood cells, white blood cells, and platelets are suspended in blood plasma, a clear, yellowish fluid component of blood or lymph. Glucose, hormones, proteins, mineral salts, lipids, vitamins, waste products, clotting factors, immunoglobulins, and carbon dioxide are among the 8%–9% of solids found in plasma, which also contains between 91% and 92% water. Red, white, and platelets make up the majority of the remaining 45% of blood. About 60 to 80 mg/mL of protein, of which 50 to 60 mg/mL are albumins and 40 mg/mL are globulins (including 10 to 20 mg/mL of immunoglobulin) are present in plasma and serum [14, 15]. Each of these is essential to maintaining the blood's efficient operation. The focus of interest for transfusion medicine has been on proteins [16].

Fresh frozen plasma is plasma that has been spun, separated, and solidified at -18^0C or colder within eight hours of being obtained through whole blood donation or by using an apheresis apparatus. Plasma coagulation factors are blood-borne proteins that help control bleeding. Figure 2 depicts the blood and its constituents.

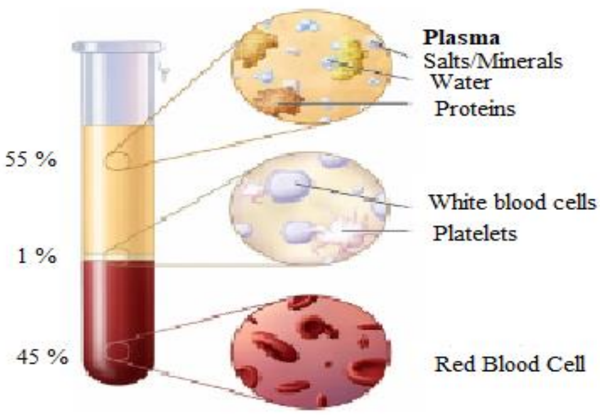

Figure 2: Blood components [16]

2. MATERIALS AND METHODS

2.1 Materials Selection

The main material of the experiment that tested the system's dependability and reproducibility was animal blood. 16 pants, each holding 250 ml of the 4.1 kg of cattle blood plasma, were created. Each pair of pants is suspended using a hanger to ensure equal cooling and homogenous freezing within the cooling chamber using a blast evaporator.

2.2 Input Parameters

Table 2 presents the needed input parameters for the design and construction of a two-stage cascade refrigeration system.

Table 2: Input parameters for low and high temperature cycles

Low Temperature Cycle (process 1-2-3-4)		
Symbols	**Parameters/meanings**	**Values**
	Refrigerant	R410A
T_{EL}	Evaporator temperature	-35°C
P_{EL}	Evaporator pressure	0.2196 MPa
T_{CL}	Condenser temperature	18°C
P_{CL}	Condenser pressure	1.3613 MPa
T_{AP}	Ambient temperature	35°C
Q_R	Refrigeration capacity	1 kW
High Temperature Cycle (process 5-6-7-8)		
	The Refrigerant	R404A
T_{EH}	Evaporator temperature	12°C
P_{EH}	Evaporator pressure	0.8772 MPa
T_{CH}	Condenser temperature	40°C
P_{CH}	Condenser pressure	1.8148 MPa
T_O	Temperature overlap	6°C

2.3 Determination of Heat Load

The five categories used in this study to classify the heat load are heat conducted through the walls of cooling space, product heat load, infiltration heat gain, packaging heat gain, and service loads.

1. **Heat conducted through the walls of evaporator**

The amount of heat that is transferred through an evaporator's walls is primarily influenced by the temperature differential between the interior and exterior of the chamber, as well as by the walls' thermal resistance, thickness, and surface area. As a result, the following are maintained in this design:

Outer dimension = 1.03 m x 0.65 m x 0.50 m (length x breadth x depth)
Inner dimension = 0.62 m x 0.49 m x 0.42 m (length x breadth x depth)
Ambient product temperature, T_{AP} = 308 K
Temperature of frozen plasma, T_{FP} = 238 K
T surface area (A_{ins}) covered by the insulator is:

$$A = [lb + lh + bh] \quad (1)$$

$$A_{ins} = \{[(1.03 \times 0.65) + (1.03 \times 0.50) + (0.65 \times 0.50)] \times 2\}\ m^2$$

$$A_{ins} = 3.02\ m^2$$

Therefore, the rate of heat conduction through the walls of the refrigerator space, $\dot{Q}_{cond}$ is determined as follows:

$$\dot{Q}_{cond} = UA\Delta T \quad (2)$$

$$\dot{Q}_{cond} = (0.314 \times 3.02 \times 70) = 0.0664\ kW$$

A specific quantity of heat transfer, Q_c is 717.12 kJ through the evaporator's walls is anticipated to occur throughout the course of three hours of continuous operation.

2. **The product heat load**

The paediatrics bag used in this study for plasma package 250 ml. A total of 16 bags of blood plasma can be stored in the chilling chamber at once. Therefore, the cooling chamber load is as follows:

$$m_p = \rho_p v_p \quad (3)$$

$$m_{tp} = 0.256 \times 16 = 4.1 \text{ kg}$$

i. **Chilling load above freezing, Q_c:** This is the heat added to the blood plasma during the process of lowering it from 35°C to -0.59°C, the temperature at which it begins to freeze.

$$Q_c = m_{tp}\, c_p\, \Delta T_1 \quad (4)$$

$$Q_c = (4.1\ x\ 3.93\ x\ 35.59) = 573.46 \text{ kJ}$$

ii. **Cooling load below freezing, Q_{bf}:** This is the additional heat acquired by chilling from -0.59°C to -35°C. The cooling load below freezing is determined by the product mass, mean specific heat below freezing, actual storage temperature, cooling duration, and desired freezing temperature.

$$Q_{bf} = m_{fp\ x}\, c_{fp\ x}\, \Delta T_2 \quad (5)$$

$$Q_{bf} = 4.1\ x\ 2.0\ x\ 34.41 = 282.16 \text{ kJ}$$

iii. **Freezing Load, Q_f:** This is the heat gained during the transition of blood plasma from the freeze plasma phase at -0.59°C to the latent heat of the freezing phase.

$$Q_f = m_{fp} L_{fp} \quad (6)$$

$$Q_f = 4.1 \text{ x } 307 = 1258.70 \text{ kJ}$$

The product load of the cascade system is 2114.32 kJ as computed from the sum of all the calculated heat gains.

3. **Infiltration heat gain**

The air that seeps into a frigid room through gaps and open doors is known as infiltration air.

$$Q_{inf} = \rho\ V_f\ Cp\ (T_o - T_i) \quad (7)$$

$$Q_{inf} = 0.0798 \text{ kg/hr} * 1000 \text{ J /kg °C} * [35 - (-35)] \text{ °C} = 5.586 \text{ kJ/hr}$$

4. **Packaging heat gain**

Over 10 % of the weight of the items are in packages while they are refrigerated.

$$Q_{pk} = m_{pk}\, C_{pk}\, (T_o - T_i)\ x\ 10^3 \quad (8)$$

$$Q_{pk} = 0.1 \text{ x } 4.1 \text{ x } 1.6 \text{ x } 70 \text{ x } 10^3 = 45.92 \text{ kJ}$$

5. **Service load**

The term "service load" refers to the quantity of heat produced by system operations such as lighting, opening, and similar activities. As a result, the service load, Q_{serv}, is provided by:

$$Q_T = Q_{cond} + Q_{prod} + Q_{inf} + Q_{pk} + Q_{serv} \quad (9)$$

$$Q_T = (66.4 + 195.77 + 1.55 + 4.25 + 2.68) = 270.65 \text{ W}$$

2.4 System Operation

As shown in Figure 3, unit A is the LTC, cycle 1–2–3–4 containing R410A, and unit B is the HTC, cycle 5–6–7–8 supplied with R404A. The LTC compressor (process 1-2) compresses the vapourized refrigerant to a high-pressure and high-temperature after it leaves the unit A evaporator (cycles 1-2-3-4). After that, a cascade condenser, a concentric counter-flow heat exchanger, is used to transfer the high-pressure and high-temperature refrigerant. By rejecting heat to unit B, cycles 5-6-7-8 result in the refrigerant condensing from a vapour into a liquid. Therefore, the cascade condenser has two functions in the system. It functions as an evaporator (process 8-5) for unit B, cycle 5-6-7-8, and first as a condenser (process 2-3) for unit A, cycle 1-2-3-4. Therefore, unit B receives heat from unit A's refrigerant and subsequently releases the resulting heat into the environment via the HTC condenser.

After that, unit B's high-pressure, high-temperature liquid refrigerant exits the cascade condenser and is discharged at low pressure and temperature, passing through a dryer to be purified of any contaminants. The refrigerant (filtered) was then transferred via an expansion valve (process 3-4), which throttled the refrigerant to minimize pressure and temperature while metering the proper amount. The refrigerant exits the expansion device at low pressure and temperature and flows to the evaporator (process 4-1), where it evaporates and transforms into vapour by removing heat from the cold region. It then flows back into the compressor at stage A for a continuous cycle of operation until the desired refrigeration effect is achieved in stage 1 alone.

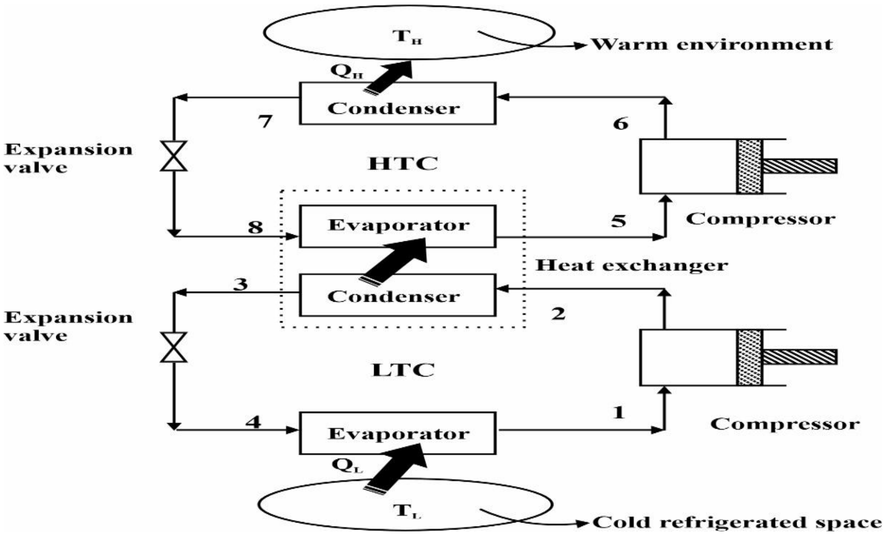

Figure 3: The schematic diagram of the two-stage cascade refrigeration system

Similar to this, the vaporized refrigerant exits the unit B evaporator, cycle 5-6-7-8, and flows to the HTC compressor (process 5-6) where it is compressed to a high pressure and high temperature. The highly vaporized refrigerant enters the condenser (process 6-7), where it condenses from the vapour to the liquid phase by rejecting heat to the environment. The condensate passes through a dryer where dirt is removed and pressure and temperature are lowered, while refrigerant flows through an expansion valve (process 7-8) that controls the amount of refrigerant needed in the cascade condenser unit, which functions as an evaporator (process 8-5) to cycle 5-6-7-8. Thus, the condensate draws in heat from the product kept in the cooling chamber of cycle 1-2-3-4 [condenser (process 3-4)] to chill. Solely in cycles 5-6-7-8 did this refrigerant vapourized before flowing back to compressor II in unit B (process 5-6).

Heat is removed from the cooling chamber (evaporator) by Unit A (LTC), while Unit B (HTC) removes heat from the LTC refrigerant and releases it into the environment. The cascade condenser provides thermal coupling between the two refrigeration cycles (LTC and HTC). As a result, the unit A evaporator's cycle 1-2-3-4 achieves the needed cooling effect. The compressors, which are used to push the refrigerants through the systems, provide the necessary mechanical effort for the closed-loop operation of the entire system.

2.5 System Experimentation

The temperature meter positioned on the evaporator is checked for flawless operation before being set up for test running. Each low temperature circuit (LTC) and high temperature circuit (HTC) system's evaporator and condenser have fixed thermocouples at their inlet and exit points. Every compressor has a pressure gauge installed at the input and exit to monitor the suction and discharge pressure. The system computer aided design (CAD) and experimented refrigeration machine are shown in Figure 4 and 5.

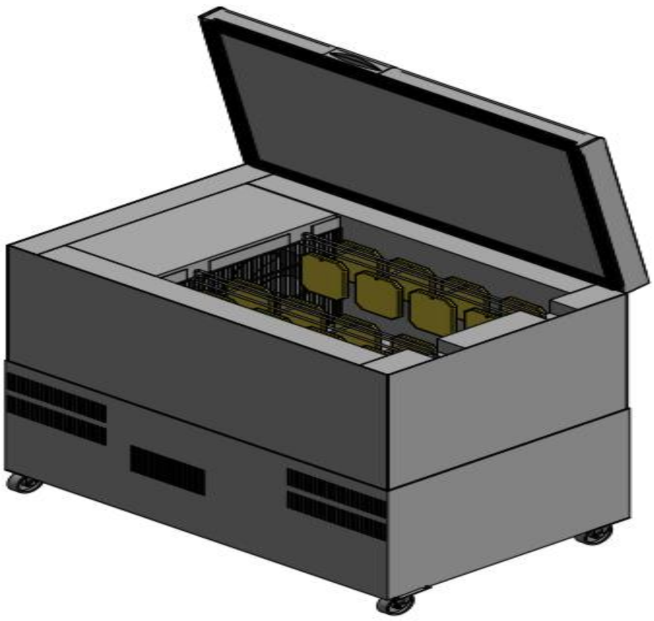

Figure 4: System computer aided design

Figure 5: Cascade refrigeration machine

The apparatus was then tested under a variety of load settings, including 25%, 50%, 75%, and 100% of 4.1 kg of centrifuged plasma, at various intervals. In a nutshell, the refrigerating effect, coefficient of performance, and system efficiency were the main foci of the analysis of the machine performance.

2.6 System Analysis

The temperature and entropy (T-S) diagram for a real two-stage vapour compression cycle is shown in Figure 6. It consists of two thermally connected cycles (VCRS) called the LT and HT stages. According to the first law of thermodynamics, the primary designed region for cooling capacity is indicated, along with the various major components' heat input and rejection into the system and on its surroundings. Through the HL cycles, the compressor work input dropped during the superheated phase. The T-S diagram analytically aids in determining the temperature and entropy at each stage of the processes and in evaluating the system coefficient of performance using the temperatures.

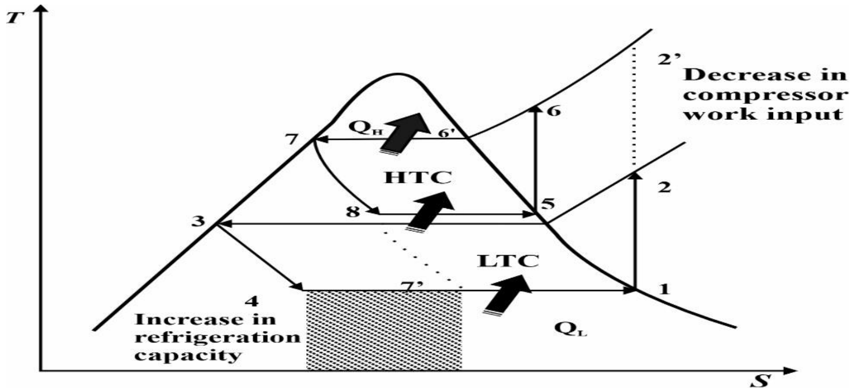

Figure 6: T-S diagram for actual two- stage vapour compression cascade cycle

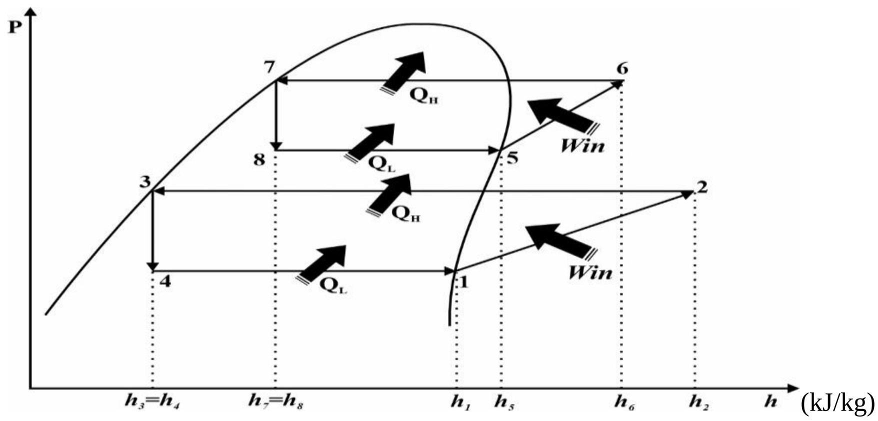

Figure 7: P-h diagram for two-stage vapour-compression refrigeration cycle

A representation of the pressure and enthalpy (P-h) diagram for the paths of heat flow across the two cycles of the cascade refrigeration system is shown in Figure 7. For the purpose of making, it simple to understand and do computational analysis, the work input into the system through the LTC and HTC via the compressors was indicated.

The P-h diagram (Figure 7) is a useful tool for estimating the enthalpy at each point when pressures are known. The P-h diagram is crucial for determining the refrigerating effect or capacity of each cycle (LTC and HTC) and the system as a whole, the work performed by the compressor in each cycle, the heat rejection by the condenser, and ultimately the performance coefficient of the designed overall system.

2.7 Calculations of Important Cascade System Parameters

The coefficient of performance, COP_L at LTC is determined using equation 10.

$$COP_L = \frac{Re}{W_{CL}} \quad (10)$$

Suppose the system is a Carnot cycle, the coefficient of performance for an LTC system is determined by Equation 11.

$$COP_{ideal(L)} = \frac{T_{EL}}{T_{CL} - T_{EL}} \quad (11)$$

The system's relative effectiveness η_L is determined using Equation 12.

$$COP_H = \frac{R_E}{W_{CH}} \quad (12)$$

The high temperature cascade cycle's COP_H coefficient of performance is computed by Equation 13.

$$COP_H = \frac{R_E}{W_{CH}} \quad (13)$$

Assuming the system is a Carnot cycle, the coefficient of performance is given by Equation 14.

$$COP_{ideal(H)} = \frac{T_{EH}}{T_{CH} + T_{EH}} \quad (14)$$

The HTC System Efficiency, sis calculated by Equation 15.

$$\eta_H = \frac{COP_{actual(H)}}{COP_{idea(H)}} \quad (15)$$

The coefficient of performance for the whole system, COP_w is calculated by Equation 16.

$$COP_{CAS} = \frac{Re}{W_{CL} + W_{CH}} \quad (16)$$

3. RESULTS AND DISCUSSIONS

Table 3 presented the outcome performance analysis of the experimental refrigeration freezer. The effect of variation in evaporator indicates that, keeping condenser unit constant, all other parameters decreased except suction pressure at lower cycle ranging 0.23 MPa to 0.64 MPa. It signifies that the evaporator pressure ratio is higher than the discharge ration via high temperature cycle due to high refrigerating effect in favour of chilling and storing plasma quickly.

Table 3: The effect of variation in evaporator temperature

T_{EL} (°C)	P_{SL} (MPa)	T_{DL} (°C)	P_{DL} (MPa)	T_{CL} (°C)	T_{EH} (°C)	P_{SH} (MPa)	T_{DH} (°C)	P_{DH} (MPa)	T_{CH} (°C)	△ TCC (°C)
-35	0.23	30	1.62	24	17	1.01	58	1.82	40	6
-28	0.30	29	1.62	24	15	0.96	57	1.78	40	6
-21	0.39	28	1.57	23	13	0.90	55	1.74	40	6
-14	0.50	27	1.52	22	11	0.85	53	1.72	40	6
-7	0.64	26	1.49	21	8	0.98	51	1.70	40	6

Table 4 presented outcome performance analysis of the experimental refrigeration system. The effect of variation in condenser increases both suction and discharge pressure on the system performance at HTC and increase in LTC discharge pressure. This implies that, an increase in condenser temperature results in high compressor discharge at HTC while the evaporator pressure ratio keeps increasing. This in turns favour a better cooling rate and system coefficient of performance. Table 4 shows the outcome performance analysis of the experimental refrigeration system.

The effect of variation in temperature difference in cascade condenser is presented in Table 5. This shows the outcome performance analysis of the developed refrigeration system. The effect of variation in temperature difference in cascade condenser showed a better performance of LTC compressor and subsequently increases in heat rejection at HTC. The amount of heat rejection could further be recycled and used for heating.

Table 4: The effect of variation in condenser temperature

T_{EL} (°C)	P_{SL} (MPa)	T_{DL} (°C)	P_{DL} (MPa)	T_{CL} (°C)	T_{EH} (°C)	P_{SH} (MPa)	T_{DH} (°C)	P_{DH} (MPa)	T_{CH} (°C)	△ TCC (°C)
-35	0.23	30	1.62	24	12	0.88	58	1.82	40	6
-35	0.23	28	1.66	25	13	0.91	61	1.91	42	6
-35	0.23	27	1.70	26	14	0.93	63	2.00	44	6
-35	0.22	25	1.75	27	15	0.96	66	2.10	46	6
-35	0.22	23	1.79	28	16	0.99	70	2.20	48	6

Table 5: The effect of variation in temperature difference in cascade condenser

T_{EL} (°C)	P_{SL} (MPa)	T_{DL} (°C)	P_{DL} (MPa)	T_{CL} (°C)	T_{EH} (°C)	P_{SH} (MPa)	T_{DH} (°C)	P_{DH} (MPa)	T_{CH} (°C)	△ TCC (°C)
-35	0.23	29	1.58	23	12	0.88	59	1.69	40	6
-35	0.23	28	1.56	22	12	0.90	60	1.69	40	5
-35	0.22	27	1.49	21	12	0.93	62	1.69	40	4
-35	0.22	26	1.45	20	12	0.95	64	1.69	40	3
-35	0.22	25	1.42	19	12	0.97	65	1.69	40	2

The coefficient of performance, COP_L at LTC is determined as follows:

$$COP_L = \frac{180.7\ kJ/kg}{59.45kJ/kg} = 3.04$$

Supposing the system is a Carnot cycle, the coefficient of performance (ideal) for an LTC system is found by:

$$COP_{ideal(L)} = \frac{238}{291-238} = 4.49$$

The system's relative effectiveness η_L is calculated by:

$$\eta_L = \frac{3.04}{4.49} \times 100 = 68\ \%$$

The high temperature cascade cycle's COP_H coefficient of performance is computed by:

$$COP_H = \frac{(374.6-263.8)}{(389.06-374.6)} = 7.66$$

Assuming the system is a Carnot cycle, the coefficient of performance, $COP_{ideal(H)}$ is given by:

$$COP_{ideal(H)} = \frac{285}{(313-285)} = 10.18$$

The HTC System Efficiency, η_H is calculated by:

$$\eta_H = \frac{7.66}{10.18} \times 100\% = 75.25\ \%$$

The coefficient of performance for the whole system, COP_w is calculated as follows:

$$COP_W = \frac{(409.4-228.7)}{(59.45+14.46)} = 2.45$$

Figure 8 presents a comparison of the operational conditional parameters. The results of several tests indicate a very high gain in efficiency, and the design target was met within the allotted setup time, demonstrating the high efficacy and dependability of the planned system for managing cold storage. It also demonstrates how the quantity stored in the refrigerator affects how long blood products like plasma must freeze in a fixed system. The outcome also showed that the designed equipment, when compared to current freezers, has a highly amazing transformation capacity in terms of timing and homogenous product freezing. The machine's overall efficiency was 70%, and the amount of work completed was calculated to be 70 kJ/kg. The high temperature cycle condenser's heat rejection capacity was 6228 kJ. Figure 9 presents the outcomes of the R410A-R404A refrigerant performance analysis for low temperature and high temperature cycles difference as 190 kJ/kg.

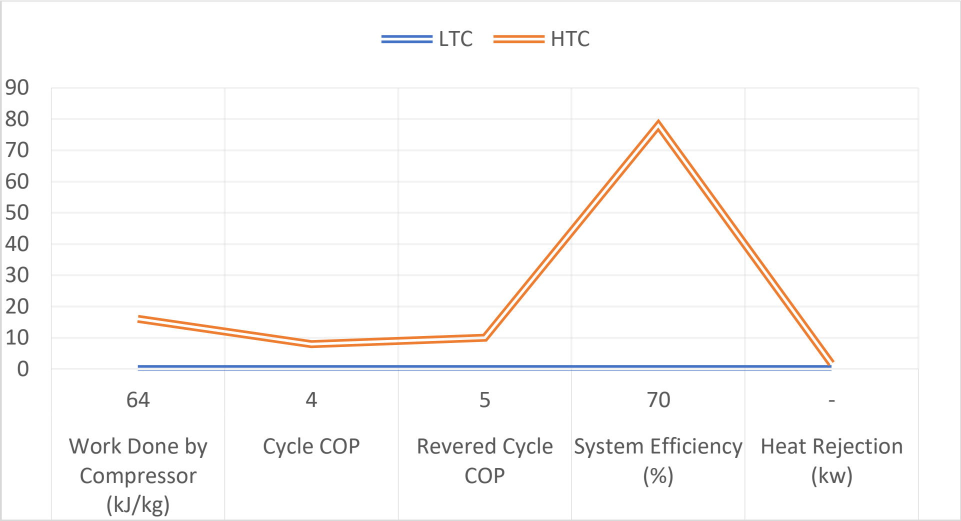

Figure 8: System parameters analysis

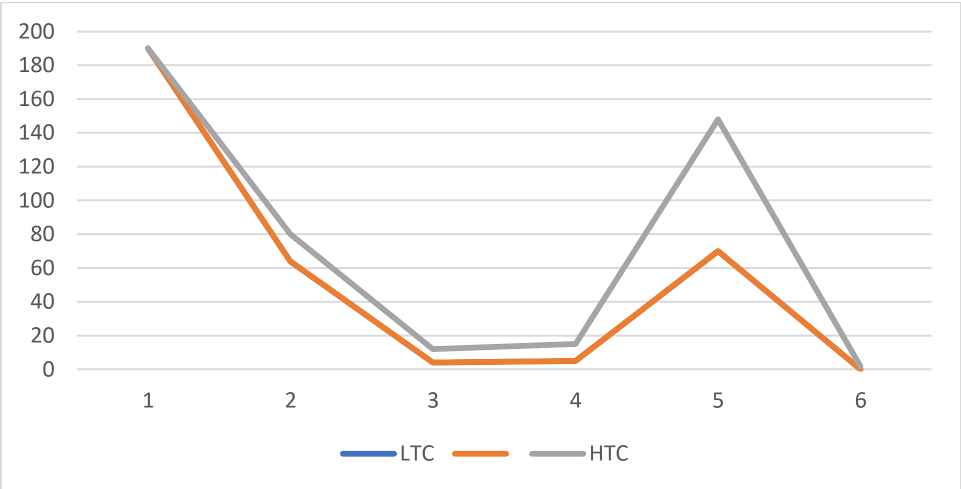

Figure 9: LTC and HTC performance

4. CONCLUSION

The system was evaluated at different loading conditions for refrigerating effects and storage time. Sequential pre- and post- plasma protein tests for integrity were carried out at Ekiti State Teaching Hospital (EKSUTH) for comparison. The results produced a reliable and sustainable new model management for preserving, storing fresh frozen plasma integrity, safety, and continual life-saving transfusions in an emergency. In the Mechanical Engineering Laboratory, Ikole Campus of the Federation University Oye Ekiti, Nigeria, testing and simulations of the R410A-R404A cascade refrigeration system were conducted. T-S diagrams for cascade refrigeration systems were used to conduct a thorough investigation of this system. With a total work done by the LTC system of 64 kJ/kg, it was determined that the system's refrigerating impact is 190 k 43J/kg. In comparison to its efficiency of 75.25%, the system's performance coefficient was estimated to be 7.66. A substantial increase in the COP of comparable systems was found in the results. At the high temperature cycle condenser, there was a total heat rejection of 6228 kJ, which could then be recycled for heating. The work novelty lies in the new model plasma management for increasing its shelf-life and quality viability for saving millions of lives and serve as important resource of every health-care system worldwide. In future research, the cascade refrigeration system's spent heat rejection at the condenser may be converted, stored, and used to heat other devices like boilers.

LIST OF SYMBOLS

Symbols	Meanings	Units
ρ_p	Density of blood plasma	g/ml
v_p	Volume of a unit blood plasma	ml
Q_c	Chilling product load (heat gained) above freezing	kJ
m_{tp}	Total mass of loaded blood plasma	kg
C_p	Specific heat capacity of blood plasma above freezing	kJ/kg-
ΔT_1	Temperature change in cooling blood plasma to -0.59°C	K
$t_{ap} - t_{fp}$	Change in chilling product temperature	K
t_{ap}	Entering blood plasma temperature	K
t_{fp}	Freezing blood plasma temperature	K
Q_{bf}	Cooling load below freezing	kJ
m_u	Unit mass of a blood plasma bag	kg
m_{fp}	Total mass of the frozen product	kg
C_{fp}	specific heat of frozen plasma below freezing	kJ/kg-K
ΔT_2	Temperature change in cooling blood plasma	K
T_c	Freezing product temperature	K
T_r	Refrigerated temperature	K
Q_f	Freezing load	kJ
m_{fp}	Total mass of the freezing plasma	Kg
L_{fp}	Latent heat of blood plasma changing to freezing plasma	kJ/kg-K
C_p	Specific heat capacity of blood plasma above freezing	kJ/kg-K
m_{inf}	Mass of in-flow air	Kg
ρ	Air density	kg/m^3
C_a	Specific heat of the air	J /kg °C
V_f	Volumetric flow rate of infiltrated air	m^3/s
T_o	Outside temperature	K
T_i	Inside temperature	K
Q_{pk}	Packaging heat load	Kg
m_{pk}	Mass of product	Kg
C_{pk}	Packaging material specific heat	J/ kg ^{0}C
T_i	Temperature of the refrigerated space	K
τ	Desired cooling time	s
R_e	Total heat absorbed from evaporator	kJ/kg
W_{CL}	Compressor work	kJ/kg
$COP_{ideal(L)}$	Coefficient of performance (ideal) for LTC system	
T_{CL}	Condenser temperature for low temperature cascade system	K
T_{EL}	Evaporator temperature for low temperature cascade system	K
W_{CH}	HTC's compressor work	Kw
$COP_{ideal(H)}$	Coefficient (ideal) for HTC system	K
T_{EH}	Evaporator temperature for HTC system	K
T_{CH}	Condenser temperature for LTC stage	K
R_e	Heat absorbed from evaporator	[kJ/kg
U	Overall heat transfer	W/m^2K
A	Surface area of the insulator	m^2
T	Freezing time	s
m_p	Blood plasma mass	kg
COP_H	Coefficient of performance at HTC	

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