REFRIGERANTS PERFORMANACE MEASUREMENT FOR COOLING EFFECT UNDER CASCADE SYSTEM: A CASE RESERACH

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Abstract

The refrigerant in the high temperature circuit gets evaporated in cascade system unit and passes through the HTC compressor. The refrigerant then passes through the HTC condenser where heat is released causing the refrigerant to cool. The refrigerant then passes through the HTC throttling valve, where the fluid expands losing temperature. In the presented research, the simulation model is used to simulate the performance of the system with some adaptations to match the case of the laboratory installation. The system's pressure boundaries, compressor speeds, and the set opinion for superheat in the freezers are input variables into the model, which are then inserted into the performance equations and efficiency curves to estimate energy consumption, cooling capacities, and the efficiencies of different components and the system as a whole. The model was used in calculations for the design and sizing of components, and while running the system the model is used to evaluate. The CO_2 compressor is a hermetic scroll with a volumetric efficiency of 90% which is assumed to be constant. The isentropic efficiency is correlated using an estimate from a similar compressor from the equivalent manufacturer the highest isentropic efficiency for the compressor is 64% at a pressure ratio of 2.5.

Keywords: Refrigerant, Cascade System, Performance Valuation.

I. INTRODUCTION:

The cascade refrigeration system is a freezing system that uses two kinds of refrigerants having different boiling points, which run through their own independent freezing cycle and are joined by a heat exchanger. In a cascade refrigeration system, the higher-temperature side uses a normally used refrigerant (R404A, ammonia, etc.), and the lower-temperature side uses R23, which is an HFC refrigerant. A cascade refrigeration system employs 2 or more individual refrigeration cycles operating at different pressure and temperature levels. The lower temperature cycle provides the desired refrigeration effect at a relatively low temperature. The condenser in the lower-temperature cycle is thermally coupled to the evaporator in the higher-temperature cycle. Thus, the evaporator in the higher cycle only serves to extract the heat released by the condenser in the lower cycle. Then this heat is rejected into the ambient air or a water stream in the condenser of the higher cycle. A cascade refrigeration system can operate with a lower evaporating temperature, smaller compression ratio and higher compressor volumetric efficiency when compared with a single-stage refrigeration system. Synthetic refrigerants were thought-about as safe for several decades, however, it well tried otherwise for the atmosphere. From this viewpoint, as a natural substance, CO2 is a perfect choice; it's a by-product of the chemical firm and exploitation it in refrigeration applications is thought-about as a further step before its inevitable discharge into the atmosphere.

As a naturally getable substance within the atmosphere, its long influence on the atmosphere is incredibly well scrutinized and that we will assume that there are not any unforeseen threats that CO2 poses for the atmosphere. This was the incident till the 1930's and 1940's once artificial refrigerants were introduced and so CO2 began to lose out tackled with competition from the new refrigerants and was bit by bit replaced altogether applications. The most reasons aimed toward the power phasing-out of CO2 are its high operative pressure (about 64.2 bars at a pair of 5°C) and its low precarious temperature of 31°C. This implicit that CO2 system had containment issues. These days technologies will give the tools to harness the extraordinary operating pressure of CO2, running and observation the system within the critical state.

II. LITERATURE REVIEW:

Proposed during the last 25 years automotive air conditioning (AAC) systems have significant development introduced by the industry and research institutes in the world to minimize the global warming threat to the environment. This paper reports the results of a study on the performance of an AAC system with measuring the compressor driving speed and the refrigerant leakage. For this purpose an experimental set up is designed and constructed to investigate the system performance. Although, the manufacturer's recommended amount for the tests with R-134a as refrigerant was 750 g, the experiments were also carried out by selecting different amount of the same refrigerant charges to analyze the coefficient of performance (COP), the cooling capacity and the compressor power change with respect to the rotating speed of the compressor. The evaluation of experimental data revealed that the best cooling capacity was achieved at 500 g refrigerant charge. Although, while the charge level decreased 40% below or increased 20% above the 500 g of the charge amount, cooling capacity loss increased up to 25% when optimum value of 500 g of the cooling refrigerant was utilized. The test results proved in each case that increasing the compressor driving speed cause almost a linear change hi the corresponding power level **[1]**. Evaluated performance merits CO_2 and R134a automotive air conditioning systems using semi-theoretical cycle models. The R134a system had a current-production configuration, which consisted of a compressor, condenser, expansion device, and evaporator. The CO_2 system was additionally equipped with liquid line/suction-line heat exchanger. Using these two systems, an effort was made to derive an equitable comparison of performance; the components in both systems were equivalent and deference in thermodynamic and transport properties were accounted for in the simulations. The analysis showed R134a having a better COP than CO_2 with the COP disparity being dependent on compressor speed (system capacity) and ambient temperature. For a compressor speed of 1000 RPM, the COP of CO_2 was lower by 21% at 32.2 C and by 34% at 48.9 C. At higher speeds and ambient temperatures, the COP disparity was even greater. The entropy generation calculations indicated that the large entropy generation in the gas cooler was the primary cause for the lower performance of CO_2 **[2].** Performed for the trans critical carbon dioxide refrigeration cycles with a throttling valve and with an expander, based on the first and second laws of thermodynamics. The effects of evaporating temperature and outlet temperature of gas cooler on the optimal heat rejection pressure, the coefficients of performance (COP), the energy losses, and the energy efficiencies are investigated. In order to identify the amounts and locations of irreversibility within the two cycles, energy analysis is employed to study the thermodynamics process in each component. It is found that in the throttling valve cycle, the largest energy loss occurs in the throttling valve, about 38% of the total cycle irreversibility. In the expander cycle, the irreversibility mainly comes from the gas cooler and the compressor, approximately 38% and 35%, respectively. The COP and energy efficiency of the expander cycle are on average 33% and 30% higher than those of the throttling valve cycle, respectively. It is also concluded that an optimal heat rejection pressure can be obtained for all the operating conditions to maximize the COP. **[3].** Expressed an analytical framework to predict the j and f factor for laminar and turbulent flow from experimental and analytical work **[4].** Suggested a liquid coolant instead of using previous air cooled models in an experimental set up to evaluate heat transfer and pressure drop of offset fin heat exchanger. It shows that the liquid cooled apparatus Prandtl number has a large effect on Nusselt number and numerical analysis examines the surface temperature distribution. Related to CFD work **[5].** Calculated the Colburn factor j and friction factor f for an Aluminumoil-air Plate Fin Heat Exchanger (PFHE) with serrated fins at low Reynolds No. (Between10-200) both experimentally, with constant air flow rate and 6 different oil flow rates and numerically, with 3D geometric analysis. One of the objectives of this paper is also to propose a procedure for the ANFIS model and an Artificial Neural Network (ANN) model alongside a few experiments so as to predict the performance of fins with new configuration in PFHE **[6].** Developed the successful utilization of Genetic Algorithm (GA) combined with the Back Propagation (BP) algorithm of Artificial Neural Network that is more efficient and advanced than the traditional GA method for the optimal design of PFHE, and showed that this method is also applicable to various PFHEs **[7].** The performance of the carbon dioxide Tran critical power cycle has been simulated and compared with the other supreme commonly employed power cycles in truncated grade heat source. At the beginning of this study, basic CO2 power cycles, namely carbon dioxide transcritical power cycle, carbon dioxide Brayton cycle and carbon dioxide cooling and power joined cycle were simulated and studied to see their potential in different applications (e.g. low-grade heat cradle applications, commercial applications and heat and power cogeneration applications). During the study, the work also involved studies in other parallel and related topics such as component design for COP. By identifying the strength and weakness facts in CO2 system solutions it is possible to apply and test modifications to optimize the system for its best cooling effect. By comparing the experimental and theoretical it is possible to point out potential improvements in the experimental rigs, and thereafter, conclude upon good CO2 system. The study subsequently focused essentially on carbon dioxide Tran critical power cycle, which has a wide range of applications **[8].** Library is given and the Modeling of CO2-Heat exchangers are described. A comparison with steady state results of heat exchangers is presented showing a very good agreement. The presented transient simulation results slow the expected trends, but the models have not yet been validated with transient experimental data **[9].** Focused on the evaluation of the performance of a single stage CO2 reciprocating compressor working on a beverage cooler application. A glass door merchandiser (GDM) was tested to develop a procedure to determine the best combination of capillary tube and refrigerant charge. Fin and tube heat exchangers were used both for the evaporator and the gas cooler. The criteria are choosing the combination was the total energy consumption of the system. The theoretical optimum discharge pressure was determined point by point during the "ON" period of the cycle and was compared to the experimental discharge pressure. The results showed that the closer profile to the optimum profile was the best in terms of energy consumption. The system was also tested with R134a and the results were compared showing 26% of energy savings in favor of the CO2 system **[10].** Obtained results allow the collection of detailed information on air and COs across the coil. The results have been compared with those obtained on our laboratory test bench and the agreement between the predictions and experimental data is very satisfactory. The analysis has been limited to the evaporator coil from the thermal hydraulics point of view. The recirculation ratio, N has been varied in the range 1 to 4 and corresponding heat transfer coefficients, internal pressure drop and saturation temperature variations have been obtained. Despite a substantial improvement in heat transfer due to recirculation (in the order of 180% for N=4), the coil capacity remained almost unchanged while pressure drop has considerably increased and the corresponding saturation temperature dropped. **[11]**

III. CASCADE SYSTEM DESCRIPTION

An excess of refrigerant is providing to the evaporator and simply a portion is evaporated, so both liquid and vapor exit the evaporator and there is no superheat. This is also mentioned to as a flooded coil, a solution which is based on natural refrigerants and a substitute candidate to conventional systems is the NH_3/CO_2 cascade concept. Fig. 4.4(b) is a schematic diagram of such a system solution which has been put up and tested in a laboratory environment. In this system concept CO_2 pressure levels are adequate; when condensing at -3℃ pressure is about 32 bars. At this temperature level, as can be seen in Fig. 1, the cooling load in the medium temperature level is provided by circulating CO_2 which accumulates in the tank.

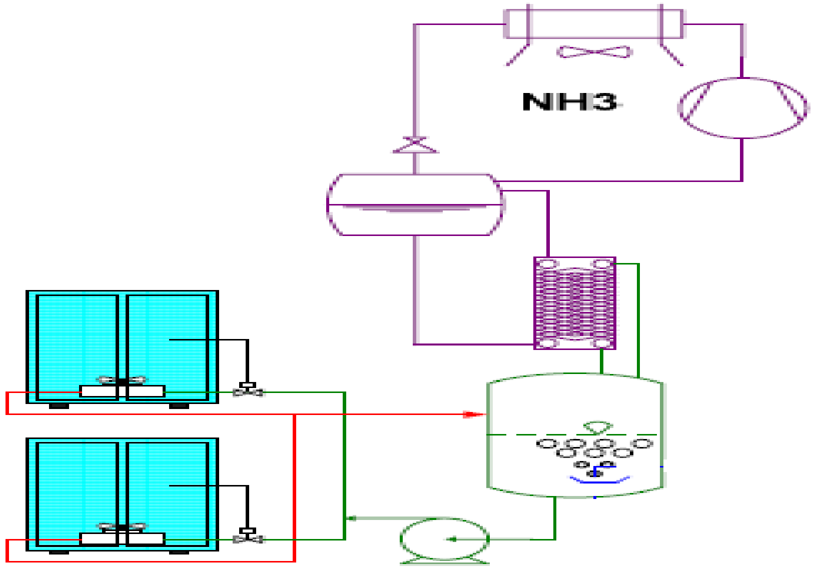


Figure.1 NH3/CO2 cascade system.

IV. CALCULATION MODEL

The model uses the required product temperatures and the ambient conditions as the boundaries of the system. At the medium temperature level the cooling capacity is 150 kW, with 50 kW for the freezing capacity; these capacities are typical for a medium size supermarket in Sweden which was defined based on analysis carried out among several major companies in the field. Load ratio is defined as the relation between cooling capabilities at the medium temperature level to individuals at the low temperature level; thus, the capacities selected result in load ratio of 2.The designed condensing temperature is 30℃. Some of the parameters inserted in the model have been a selection from investigational work on an NH_3/CO_2 cascade system. Product temperature at the medium level is +3℃ and -18℃ for frozen food.

Air temperature difference (ΔT product, air) is the difference between the warmest product temperature and the temperature of the air inlet into the display cabinet's heat exchanger. Experiments showed that the product temperature value falls between the air inlet and outlet temperatures of the display cabinet's heat exchanger; it is about 3K lower than the air inlet temperature. The air temperature change (ΔTair) is 7K and the temperature difference between the air and the refrigerant is 2K. Isentropic efficiency values of compressors were obtained from the following equation which is fit by Brown et al. (2002) for a CO_2 compressor in a mobile air conditioning system.

$$\eta_{CO_2} = 0.94 - 0.04478 * \left(\frac{P_1}{P_2}\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (3)$$

P_1 and P_2 are the discharge and suction pressures

V. SYSTEM OPTIMIZATION

Tube size selection took into consideration the possibility of running the system within a reasonable fringe of load variations and boundary changes. Some parameters, e.g. circulation ratio, can be changed while still sanctioning the system to give reasonable pressure drops at the selected sizes. The calculations are made for unlike tube diameters & circulation ratios of 2 and 3 at a saturation temperature of -8℃. A value of 2 is used as the design circulation ratio at the rated capacity. On the low temperature side, the calculation for two solutions direct development or flooded evaporators can be applied. However, higher value may be chosen in order to avoid dry out which may happen with at relatively low vapour fractions.

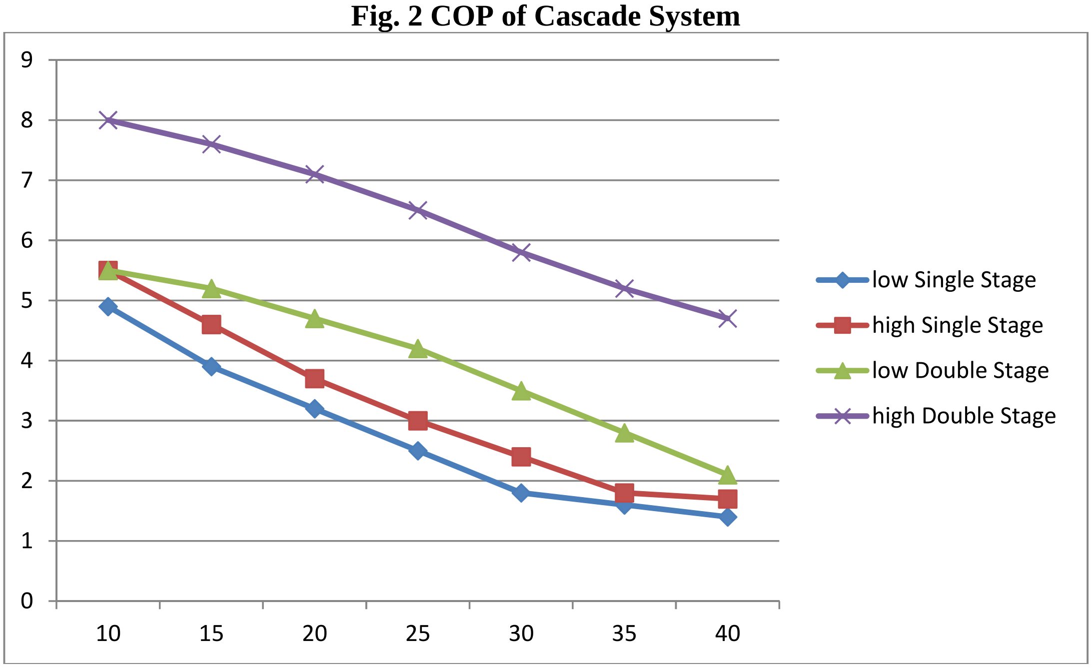
. VI. CONCLUSION:

The calculation of COP for the cascade system solution is carried out in the same way as for the parallel arrangement. The pump power apply on the medium temperature circuit is included in the system's energy consumption, which is usually very small compared to the total energy consumption; in this case 740W was calculated as the energy required running the pump at a circulation ratio of 2. The reference cascade system is defined as having single-stage compression in the high stage, as per the system in Figure 2. The improved parallel system solution with two stage compressions in both circuits becomes close to the reference centralized system resolution and at ambient temperatures higher than 30°C it gives up to a 6% higher COP. The total COP is connived for the reference centralized system solution, which reveals a 4-21% higher COP compared to the reference indirect system as shown in Table 1.

Table 1 COP of Cascade System

Ambient Temperature °C	COP of Single Stage Compression		COP of Double Stage Compression	
	Low	High	Low	High
10	4.9	5.5	5.5	8
15	3.9	4.6	5.2	7.6
20	3.2	3.7	4.7	7.1
25	2.5	3	4.2	6.5
30	1.8	2.4	3.5	5.8
35	1.6	1.8	2.8	5.2
40	1.4	1.7	2.1	4.7

Fig. 2 COP of Cascade System



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