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PERFORMANCE OPTIMIZATION OF A LARDER TYPE REFRIGERATOR UNIT USING COMPUTER AIDED ANALYSIS TOOLS

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ABSTRACT

This paper describes the utilisation of computer aided analysis tools in the design and performance optimisation process of domestic refrigerators. A larder type table-top refrigerator unit is considered in the present study. In the first phase of the project, the existing refrigerator using CFC-12 as a working fluid was converted into a prototype with environmental benign refrigerant isobutane, R600a. The conversion process involved compressor stroke volume and refrigerant charge adjustments to achieve the same cooling performance. In the second phase, in-house developed refrigeration cycle simulation code and commercial Computational Fluid Dynamics (CFD) packages were used for the optimisation with respect to energy consumption. Design parameters of the main components of the refrigeration circuit, namely compressor, condenser, evaporator and capillary tube-suction line intercooler assembly were modified based on the results of computer simulations. Three prototypes were subsequently built and tested under standard climatic conditions. Redesigned unit has been found to have significantly lower energy consumption compared with the existing refrigerator.

INTRODUCTION

The ongoing phaseout process of ozone depleting chlorofluorocarbons (CFCs) and the increasing emphasis on global warming considerations which demand the production of more energy efficient appliances have been forcing domestic refrigerator manufacturers for the design changes and improvements on their existing product lines. To help to facilitate this need, a flexible component based computer simulation package for the performance prediction of the household refrigerator/freezer units has been developed. Extensive use of this in-house developed analysis capacity is being made in the product development environment. Commercial CFD packages are also being used routinely to support the refrigerator design process.

In this study, utilisation of the established capacity is described for the redesign case of a larder type refrigerator unit. The redesign objective was twofold: (i) conversion to an environmentally friendly hydrocarbon refrigerant, R600a from CFC-12 and (ii) improvement in energy consumption.

Since the beginning of 1990's, appliance manufacturers in Europe have started using hydrocarbons refrigerants in domestic refrigerators. Hydrocarbons are environmentally safe substances since they have no ozone depletion and negligible global warming potential. Among their advantages, superior transport properties (high thermal conductivity and lower viscosity), requirement for lower amount of charge compared with CFC and HFC type refrigerants, solubility in traditionally used lubricant mineral oil can be stated. These advantages are so attractive for refrigeration system designers and the people who care for nature that the efforts spent for the design of a risk-free system are easily justified. Although it requires some design changes in the cooling circuit components, isobutane became the choice of domestic refrigeration producers among other hydrocarbon refrigerants. The reason for its selection is that it allows the compressor to operate under less loads thus causing more reliable and a quieter system.

In the first part of the paper, the establish simulation capabilities are reviewed. The refrigerant replacement and the energy optimisation methodology followed in the redesign process are presented in the remainder of the paper after a brief description of the existing refrigerator unit.

Conclusions

Our investigation allows to infer that the mixture forms positive azeotrope and that azeotropic concentration changes its value from $x=0.39$ at 180 K to $x=0.33$ at 285 K.

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Experimental apparatus

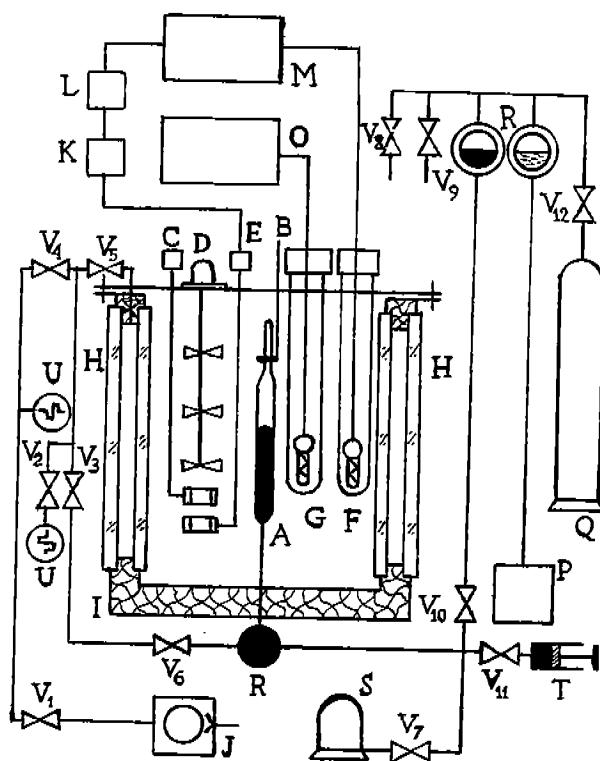


Fig.

A, optical cell; B, magnetic stirrer; C, main heater; D, stirrer; E, subheater; F, G, platinum resistance thermometers; H, windows; I, thermostat; J, vacuum pump; K, thyristor amplifier; L, PID controller; M, thermometer bridge; O, potentiometer; P, dead-weight pressure gauge; Q, nitrogen gas; R, separators; S, Hg-cylinder; T, precise volume controller; U, vacuum gauge; V_1 - V_{12} , valves.

Table 2

Thermodynamic properties on the saturation line for composition $X = 0.36$

T K	P' MPa	P'' MPa	ρ' kgm^{-3}	ρ'' kgm^{-3}	h' kJkg^{-1}	h'' kJkg^{-1}	s' $\text{kJ}(\text{kg K})^{-1}$	s'' $\text{kJ}(\text{kgK})^{-1}$
180	0.0799	0.0785	1560.7	5.18	155.9	317.5	0.7594	1.6423
185	0.1044	0.1042	1540.5	6.73	168.1	320.3	0.8248	1.6339
190	0.1352	0.1352	1519.7	8.58	175.2	323.1	0.8603	1.6266
195	0.1734	0.1734	1498.5	10.81	180.4	325.6	0.8850	1.6193
200	0.2203	0.2203	1476.8	13.54	184.7	328.0	0.9046	1.6118
205	0.2773	0.2772	1454.5	16.82	188.4	330.2	0.9211	1.6039
210	0.3455	0.3455	1431.8	20.73	191.8	332.1	0.9358	1.5955
215	0.4263	0.4264	1408.3	25.36	194.8	333.7	0.9490	1.5864
220	0.5212	0.5213	1384.1	30.82	197.6	334.8	0.9609	1.5765
225	0.6316	0.6316	1358.9	37.21	200.1	335.5	0.9717	1.5655
230	0.7588	0.7588	1332.6	44.68	202.3	335.7	0.9814	1.5531
235	0.9045	0.9043	1304.8	53.37	204.2	335.1	0.9899	1.5390
240	1.0702	1.0698	1275.5	63.52	205.8	333.8	0.9970	1.5228
245	1.2577	1.2569	1244.2	75.37	207.0	331.5	1.0025	1.5038
250	1.4688	1.4675	1210.6	89.30	207.6	328.0	1.0061	1.4814
255	1.7056	1.7037	1174.4	105.79	207.5	323.0	1.0072	1.4546
260	1.9703	1.9678	1135.1	125.52	206.4	316.1	1.0052	1.4220
265	2.2654	2.2622	1091.9	149.52	204.2	306.6	0.9992	1.3817
270	2.5936	2.5899	1043.9	179.38	200.3	293.5	0.9876	1.3305
275	2.9582	2.9545	989.8	217.77	193.9	275.1	0.9679	1.2631
280	3.3634	3.3607	928.2	269.62	183.7	248.3	0.9355	1.1695
285	3.7230	3.6539	836.8	293.18	169.7	236.8	0.8921	1.1339

Table 3

Parameters used in Eqs. (7)

i	t_i	P_i	ρ_i
0	299.06	48.086	525.06
1	-54.03869	-23.65859	147.6902
2	63.49612	6.05004	-56.04627
3	-15.40773	-	-

Table 1 (continued)

Coefficient	R23	R116
B ₀	-0.016	-0.009
C ₀	0.073255	0.064110
C ₁	5.275804·10 ⁻²	-7.054309·10 ⁻¹
C ₂	2.717245·10 ⁻¹	1.264610·10 ⁻¹
D ₀	0.00852	0.02010
D ₁	-7.009538·10 ⁻³	1.042710·10 ⁻¹
E ₁	7.17218	7.02955
E ₂	5.2398	5.6778
E ₃	2.64	2.64
F ₁	1.7152	1.6495
F ₂	0.3294	0.3261
T _c , K	299.0	293.0
P _c , MPa	4.8200	3.0386
ρ _c , kg·m ⁻³	525.0	616.8

The thermal properties of the mixture on the vapor-liquid coexistence curve were calculated by the following equations:

$$\ln(p_c/p_s) = E_1 \ln(T_c/T) + E_2 (\ln(T_c/T))^{E_3} \quad (5) \quad \ln(\rho'/\rho_c) = F_1 (\ln(T_c/T))^{F_2 f}, \quad (6)$$

were $f = 1 - 1.113 (\ln(T_c/T))^{0.4} (\ln(\ln(T_c/T)))^{-1}$;

E₁, E₂, E₃, and F₁, F₂ coefficients dependent on individual properties of substances.

The coefficients and critical parameters used for R23 and R116 are listed in Table 1.

Thermodynamic properties on the saturation line for composition of 36 mol.% R116 were calculated by using Eq.(4) and listed in Table 2.

The results for the critical parameters of the mixture can be represented by the following equations:

$$T_c = \sum t_i X^i; \quad P_c = \sum P_i X^i; \quad \rho_c = \sum \rho_i X^i. \quad (7)$$

The coefficients of Eqs.(7) are listed in Table 3.

The numerical constants in Eq.(2) are tabulated in Table 1 for pure components of the mixture. The mixing rules of the coefficients of Eq.(1) were given in the following equations:

$$\begin{aligned} a &= \sum \sum X_i X_j (1 - K_{ij}^a) (a_i a_j)^{0.5}; & b &= \sum \sum X_i X_j (b_i b_j)^{0.5} \\ c &= \sum \sum X_i X_j (1 - K_{ij}^c) (c_i c_j)^{0.5}; & d &= \sum \sum X_i X_j (1 + K_{ij}^d) (d_i d_j)^{0.5} \end{aligned} \quad (3)$$

The binary interaction parameters K_{ij} for the present system were determined to be:

$$\begin{aligned} K_{ij}^a &= 0.305 + 0.1 (X_2 - 0.36), [X_2 \leq 0.36]; & K_{ij}^a &= 0.305 + 0.27 (X_2 - 0.36), [X_2 > 0.36]; \\ K_{ij}^c &= 0.4; & K_{ij}^d &= 0.4. \end{aligned}$$

Deviation of the experimental data from those calculated by means of Eq.(1) did not exceed 0.6%.

Data [2] (P_s of composition $X = 36$ mol. % R116 in temperature range from 187 to 280 K) and our experimental VLE data of the mixture were used to determine parameters of the cubic equation of state (the equation used to calculate vapor - liquid equilibria):

$$P = RT[1/(V - b) - a/V(V + c)], \quad (4)$$

where a , b , c are coefficients determined from the conditions of vapor - liquid equilibria of the components of the mixture for each value of the temperature on the saturation line.

Using the same mixing rules (Eq.(3)), the binary interaction parameters K_{ij} were determined to be:

$$K_{ij}^a = 0.102 + 0.45X_2; \quad K_{ij}^c = 0.$$

The binary interaction parameters for the calculation of the saturated-liquid densities of mixture were determined to be:

$$K_{ij}^a = 0.1, [X_2 \leq 0.44]; \quad K_{ij}^a = 0.1 + 0.8(X_2 - 0.44), [X_2 > 0.44]; \quad K_{ij}^c = 0.$$

Deviation of the measured bubble-point and dew-point pressures from values calculated using Eq.(2) did not exceed 1%. The maximum deviation between the measured saturated -liquid and saturated-vapor densities of mixture and those calculated from Eq.(2) was 0.4% and 1% respectively.

Table 1

Critical parameters and constants in Eqs.(2),(5),(6)

Coefficient	R23	R116
A_0	0.171583	0.257681
A_1	3.490379	$1.119119 \cdot 10^1$
A_2	-1.362514	-7.700742
A_3	$2.619432 \cdot 10^{-1}$	1.303123

another 100ohm platinum resistance thermometer (F) was mounted into the thermostated bath. The temperature fluctuation was detected by thermometer (F) and temperature control was carried out by the 1 kW main heater (C), 40 W subheater (E), thermometer bridge (M) (Model MO-62), PID controller (L) (Model VRT-2) and thyristor amplifier (K). The pressure inside the balloon (Q) had been transmitted to mercury and to oil through a visual separators (R). The dead-weight pressure gauge (P) (Model MP-60) was directly used to measure the pressure. The level of mercury and vapor-liquid interface was determined by visual observation. The sample density was calculated from the inner volume of the cell and the mass of the confined mixture. The composition of the mixture was determined by weighting the mass of each pure components before mixing.

PVTx properties were measured on the isotherms.

Determination of the saturation state of a mixture with a prescribed composition was performed by observing the appearance of a bubble in the liquid-phase sample confined in the cell.

The parameters on the dew curve were determined in the points, where isotherms of gaseous phase crossed the isotherms of double-phase region. For analysis of the results of measurements near the dew curve a method of investigation the PVT properties in the region of blurred phase transition considered in [1] was used.

The critical point was determined by observing disappearance and reappearance of the meniscus.

The purities of each component supplied was 99.79% R23 and 99.80% R116. The experimental uncertainty in determining the concentration was estimated as 0.0008 mol/mol, temperature - at 15 mK. Taking into account the systematic and accidental errors, full error in pressure value was equal to 0.1-0.3% in all pressure region, the one of liquid density was about 0.2-0.3%, for vapor density - 0.15-0.25%. The uncertainties in the critical temperature, critical pressure and critical density have been estimated to be within 20 mK, 8 kPa and 6 kg/m³, respectively.

Results

The measurements were performed at compositions of x: 0.1015; 0.2232; 0.3600; 0.4736; 0.5624; 0.7010; and 0.8767 mol/mol R116 in the following ranges: for temperature from 243.15 to 298.15 K, for pressure from 1.0 to 3.4 MPa and for density from 50 to 250 kg/m³ in the vapor phase. In addition, the bubble-point pressures and saturated-liquid densities were measured at these compositions in the temperature range from 243.15 to 280.15 K. The shape of the critical curves was studied at all compositions studied.

Discussion

We developed an equation of state in the vapor region based on the present measurements using for it a functional form derived from the cubic equation of state:

$$P = RT[1/(V-b) - a/(V(V+c)+d)]. \quad (1)$$

Temperature dependences of the coefficients of Eq.(1) are following:

$$\begin{aligned} a &= A_0(1 + A_1(1/\tau - 1) + A_2(1/\tau^2 - 1) + A_3(1/\tau^4 - 1)); & b &= \text{const}; \\ c &= C_0 + C_1(1 - \tau) + C_2(1 - \tau)^2; & d &= D_0 + D_1(1 - \tau). \end{aligned} \quad (2)$$

VAPOR - LIQUID EQUILIBRIA IN R23/R116 SYSTEM AND ITS THERMODYNAMIC PROPERTIES

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Abstract

Vapor - liquid equilibria and PVTx properties of binary refrigerant mixture R23/R116 have been measured by means of a variable volume method. The measurements were performed at compositions of 10.15, 22.32, 36.00, 47.36, 56.24, 70.10 and 87.67 mol. % R116. The uncertainties of the temperature, pressure and density measurements were estimated to be not greater than 15mK, 0.3% and 0.3% respectively. Also the critical parameters (T_c , P_c and ρ_c) for this mixture have been measured. The parameters of the cubic equation of state calculated from our measurements are presented too.

Nomenclature

K_{ij}	binary interaction parameter
P	pressure, MPa
R	$= 8.3145 \cdot 10^{-3}$, universal gas constant, $\text{kJ}(\text{molK})^{-1}$
T	temperature, K
V	molar volume, $\text{m}^3 \text{mol}^{-1}$
X	mole fraction of R116
ρ	density, kg m^{-3}
τ	$= T/T_c$, reduced temperature

Introduction

The binary system R23/R116 may be suitable replacement for the refrigerants R13 and R503 which are widely used as working fluids in refrigerant equipment. The aim of this work is to present experimental vapor-liquid equilibria (VLE) and PVTx data and provide parameters for the cubic equation of state for this mixture.

Experimental

The experimental apparatus is shown schematically in Fig. The main part of the apparatus is an optical cell (A). The body of this cell was made of the molybdenum glass (300 mm long, 3.4 and 9.6 mm in inner diameters). The inner volume calibrated by using mercury is about 14 cubic cm. The optical cell was installed in a thermostatic bath (I) with heat-transfer medium that was circulated by stirrer (D). The temperature measurement was made by calibrated 10 ohm platinum resistance thermometer (G) (Model PTS-10) with the aid of potentiometer (O) (Model R348). The thermometer was placed in the vicinity of the optical cell. In order to keep the temperature constant

temperature $275 \leq T \leq 345$ K and concentration $0.1 \leq x \leq 1$ regions with the accuracy sufficient for practice.

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Cabinet Model: The entire internal volume of the cabinet is modelled as a single node. The heat gain through the cabinet insulation is expressed in terms of overall conductance value ($W/^\circ C$) and the temperature difference between the cabinet air and the ambient. ERA1.2, public domain refrigeration cycle simulation software [4], was employed to estimate the cabinet conductance value by using measured thermal conductivity values. Reverse heat leak tests were also carried out to confirm the predictions of ERA1.2. The results were found to agree within a reasonable tolerance.

Refrigerant Model: REFPROP4.0 software [5] is integrated into the REFSIM package for the calculation of refrigerant properties.

COMPUTATIONAL ANALYSIS OF THE FLOW FIELD INSIDE THE CABINET

In the present study, state-of-the-art CFD methodology is applied for the prediction of the flow field and temperature distribution inside the refrigerator cabinet. The air motion inside the cabinet is produced by buoyancy forces acting on a fluid of non-uniform density resulting from the non-uniform temperature field. The governing non-dimensional parameter for the buoyancy driven flow inside the cabinet is a Rayleigh number:

$$Ra = \frac{g \beta \Delta T L^3}{\alpha \gamma} \quad (1)$$

where, g is the gravitational acceleration, β is the coefficient of volume expansion, L is the characteristic length scale, α is the thermal diffusivity, γ is the kinematic viscosity, and ΔT is the temperature scale.

The Ra number indicates the type of flow to be expected, whether flow is laminar or turbulent. Since the geometry of the enclosure plays a significant role in influencing the flow regimes, it is difficult to estimate a value for the Ra number for which transitional and turbulence flow regime starts. In the present study, the Ra number was estimated based on the height of the cold-wall evaporator and the temperature difference between the cabinet air and the cold-wall surface with the analogy to vertical free convection boundary layers. Substantially laminar flow can be observed up to a Ra number of an order of magnitude of 10^9 in vertical free convection boundary layers. The calculated Ra number for the given cabinet geometry is in the order of magnitude of 10^8 . Laminar flow assumption was made for the flow regime.

All these modes of heat transfer, i.e. conduction, convection and radiation are acting simultaneously resulting in a complex physical situation. The radiant exchange between the internal surfaces of the cabinet was modelled by making ideal grey diffuse surface assumption.

In the present study, a commercial CFD package was employed for the coupled flow and heat transfer analysis. The CFD package used in this analysis is a general-purpose Navier Stokes solver. The code employs efficient finite-volume solution methodology in conjunction with a highly flexible body-fitted unstructured mesh system including local embedded mesh refinement. The analysis procedure includes geometry modeling, mesh generation and numerical solution tasks. The 3-D solid model of the refrigerator unit was created by the designers using a CAD package. The model was then modified to include only those details which are considered to be important in term of fluid flow and heat transfer. In the next step, the computational grid was prepared for the solution phase. The flow situation inside the cabinet is such that the flow field contains different length scales such as the boundary layer next to the cold-wall surface. Different mesh intensities were applied in different parts of the domain to capture the flow details. After defining appropriate boundary conditions and material properties, the model became ready for the numerical solution phase. Transient PISO algorithm was employed for the numerical solution of governing non-linear flow equations. This involves approaching the steady-state solution in time-marching fashion with the fixed time steps.

A variety of flow situations were considered corresponding to different operating conditions and geometric parameters, including the relative position and the height of the cold-wall evaporator. One representative case is presented in this paper to show the scope of the CFD analysis. Figure 1 shows the calculated velocity field on the

mid-plane between the side wall. The maximum velocity of 0.29 m/s occurs in the thin boundary layer next to the evaporator surface while the fluid in the upper core region of the cabinet remains almost at rest. A stratified temperature field is observed inside the cabinet. The temperature difference between the upper and lower parts of the cabinet is nearly 5.0 °C. The average air-side heat transfer coefficient in the boundary layer next to the cold-wall surface were determined from the results of the CFD analysis and used as an input parameter in the evaporator module of the REFSIM program.

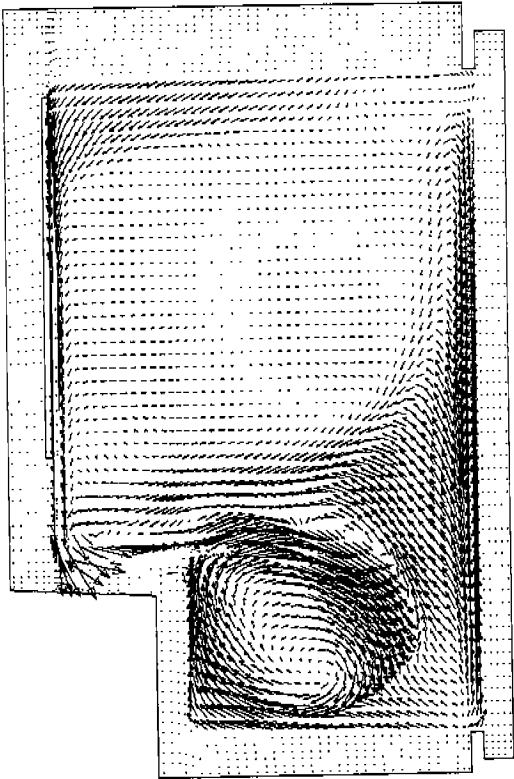


Figure 1. Calculated Velocity Field inside the Refrigerator Cabinet

PHASE I: CONVERSION FROM R12 TO R600A

The basic design characteristics of the existing larger unit is specified in the following table.

Table 1. Description of the Existing Refrigerator Unit

Cabinet	
Outer Dimensions	800 x 600 x 550 mm
Net Volume	135 lt
Blowing Agent	50% reduced R11
Appliance Constant	1.00 W/°C
Condenser	Wire & Tube Type
Evaporator	Tube on Sheet type
Compressor	
Refrigeration Capacity	63 kcal/hr
COP	1.02
Refrigerant	R12

Necessary Design Changes for R600a Conversion: The saturation pressure and the vapor density of R600a corresponding to the operating temperature range of the evaporator are significantly lower than those of R12. If the same compressor stroke volume is used, a substantial loss in cooling capacity would occur due to the decrease in the refrigerant mass flow rate pumped by the compressor. The stroke volume of the compressor needs to be enlarged to overcome this capacity loss. About 80% increase in the stroke volume was predicted to achieve a similar cooling performance. Electric motor of the existing compressor has been assembled with the cylinder-piston group of a larger capacity to accommodate this need. The resulting prototype compressor, with a swept volume increase of about 90%, was found to have a cooling capacity of 62 kcal/hr and a COP of 0.90. Lubricant of the compressor was not changed since both R600a and mineral oil belong to hydrocarbon family of substances and have very good solubility in each other.

Since the total heat transfer resistance is governed mostly by the air side, heat transfer performance of exchangers were considered to be practically independent of the refrigerant type. Therefore, design parameters of condenser and evaporator units were not modified

The calculations carried out on the REFSIM package indicated that the same capillary-suction line heat exchanger can also be used for the R600a gas with a slight changes in system performance. From the practicality point of view, capillary-suction line assembly has not been modified in the new prototype refrigerator.

Necessary amount of R600a charge was calculated based on the liquid density difference of the refrigerants at the same evaporation temperature. The predicted R600a charge requirement for the new prototype is nearly %40 of the R12 charge.

Performance of the Initial Prototype with R600a: Besides redesigning and constructing the refrigeration circuit, a prototype refrigerator cabinet has been rebuilt by blowing the insulation with another hydrocarbon c-Pentane. The appliance constant for the new unit was 1.025 W/°C. This indicates slightly higher heat gain value compared with the existing cabinet. The new prototype unit was tested for energy consumption and cooling performance. The measured energy consumption was 0.771 KWh/24h. This value corresponds to the average cabinet temperature of 5 °C under the 25 °C ambient temperature. The average cabinet temperature of 8.3 °C was obtained under the steady-state regime corresponding to 43 °C ambient temperature.

PHASE II: ENERGY OPTIMISATION

The energy consumption and the cooling performance values of the initial R600a prototype was similar to the existing refrigerator unit. In an effort to minimise the energy consumption of this initial prototype, the following design modifications were proposed based on the results of REFSIM program and realised without a cost consideration.

Condenser: The number of the tube passes was increased by 2, utilising the available space in the back of the refrigerator. This corresponds to 11% increase in the total area of the condenser. A further step was the adjustment of inclination of the condenser. The existing stoper design allowed inclination angle of 4 degrees. Even this value was found to enhance the natural convection heat transfer coefficient significantly.

Evaporator: A new aluminium sheet having a surface area 21% greater than the original design was used in the same evaporator location. The number of tube passes was increased by 2 while spacing of the tubes were kept the same.

Capillary Suction Line Assembly : Length of the capillary tube was adjusted for the new geometries of the condenser and evaporator while inner diameter was the same. A relatively small gain in energy consumption was expected from increased capillary length.

Compressor: Based on the operating conditions observed in the performance tests, electric motor of the initial R600a compressor prototype has been optimised. This yielded a final COP of 1.02. This value is comparable to the COP of R12 compressor.

Prototypes and Testing: Three prototypes have been built and tested to see the effectiveness of proposed design changes on cooling performance and energy consumption. R600a charge in the circuit was increased slightly to accommodate the increased volumes of the evaporator and condenser. Test results are summarised in following table.

Table 2. Performance Comparison between the Optimised Unit and the Initial Prototype

	New Prototype	Initial Prototype
At 43 °C ambient temperature		
Average cabinet temperature (°C)	5.2	8.3
Condensation Temperature (°C)	58.6	58.8
Evaporation Temperature (°C)	-23.6	-22.6
At 25 °C ambient temperature		
Average cabinet temperature (°C)	5.0	5.0
Condensation Temperature (°C)	41.2	42.4
Evaporation Temperature (°C)	-26.0	-29.7
Run Time of the Unit (%)	30.3	38.3
Daily Energy Consumption (KWh/24h)	0.551	0.771

CONCLUSION

This study was intended to prove the applicability of computer aided analysis tools to the refrigerator design process. The established methodology was tested in a real design situation where the objective was to improve the energy consumption level of a larger type unit while undergoing a refrigerant replacement. Nearly 28% reduction in energy consumption was achieved by optimising refrigeration circuit components based on the results of simulation models.

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