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EVALUATION OF EVAPORATOR CORRELATIONS FOR HYDROCARBON REFRIGERANT PROPANE (R290), ETHANE (R170), ETHYLENE (R1150) AS A REPLACEMENT IN HORIZONTAL TUBE DOMESTIC REFRIGERATOR

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ABSTRACT

Hydrocarbons are eco-friendly citing to their thermodynamic performance and non-pollutant criteria making them suitable alternatives. This paper presents study of environment friendly hydrocarbon refrigerant PROPANE(R-290), ETHANE (R170), ETHYLENE(R1150) best known for their zero ozone depletion potential and minimal global warming effects in domestic refrigerator. They provide additional benefits like high latent heat, lower discharge temperature, feasibility with copper and additives and provide better performance with minimum energy consumption comparatively. Based on theoretical results obtained from thermo physical properties and suitable correlation, it has been found that, Hydrocarbons can replace HFCs in the evaporator of a domestic refrigerator without any system modifications. This leads to capital savings owing to non-alteration of existing system. In this paper, three well known correlations have been evaluated to find heat transfer co-efficient (HTC) which are in turn used to calculate theoretical heat transfer area which are compared with actual heat transfer area. The least error for R290 is found with Kandlikar correlation, for R170 is found with Gungor, and for R1150 is found with Kandlikar and Gungor correlation which gives +10 % error. This upholds the replacement with hydrocarbon refrigerant in existing evaporator system.

KEYWORDS: Hydrocarbon refrigerants, R290, R170, R1150, correlations, horizontal tube refrigerator.

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INTRODUCTION:

In the early 1900s natural refrigerants such as ammonia, carbon dioxide, and sulphur dioxide were commonly used. They possessed toxicity and were found hazardous. During 1930s, chlorofluorocarbons (CFCs) and hydrochloro fluorocarbons (HCFCs) emerged as safer alternative refrigerants with additional benefits such as stability, non–toxicity, non–flammability, better material compatibility and thermodynamic properties. This revolutionized the Refrigeration and Air Conditioning sector leading to their wide spread use in domestic and industrial applications throughout the world. Researches have shown that earth's ozone layer, which protects the earth's surface from harmful effects of UV radiation is depleting because of chlorine presence in the stratosphere. The primary cause for such harm in CFCs and HCFCs are large class of chlorine containing chemicals, which rise up to the stratosphere where they react with ozone leading to convert more ozone to oxygen. This concern has effected in a number of international treaties urging for gradual phase out of halogenated fluids. Since 1996, developed countries have taken steps in banning of CFCs and is estimated that by 2030, production and usage of CFCs will be prohibited in the entire world. Also is the case for partially halogenated HCFCs [1].

The contribution of refrigeration system towards global warming can be viewed in two ways, namely direct and indirect. The direct effect occurs because of the leakage of refrigerant and whose effect can be connected directly with the global warming potential of the refrigerant used. Whereas carbon-dioxide emission from combustion of fossil fuels, which provides electricity to drive compressors can be viewed as indirect. In case of hermetic equipment where refrigerant leakage is low, energy consumption plays a key parameter in analyzing environmental impact of refrigerants. If refrigerant leakage is large as in car air-conditioners, the global warming potential dominates but energy costs are still of interest [2].

The quest for eco-friendly refrigerants which provide no harm to our environment and shielding ozone layer are moving in quick pace. Air, ammonia, carbon dioxide, hydrocarbons and water are accepted as the only class of refrigerants with no ozone depletion potential, low global warming trait and cleaner production ways by all national governments except the Environment Protection Agency of the United State. Noteworthy contributions are made in China and Germany in this direction while other European countries are now requiring phase-out of harmful refrigerants. P. J. vanderWeyde of Philadelphia invented hydrocarbon refrigerants in 1866 [3].

Ethane R170, propane R290 and isobutane R600a were successfully marketed by the Linde companies in the 1920s and 1930s. Since 1967 hydrocarbon refrigerants have liquified natural gas in plants which are now larger than 100 MW of cooling. Hydrocarbon refrigerants for domestic appliances were revived in Germany in 1992 and R600a refrigerators have spread worldwide [4].Research has shown that hydrocarbons are good alternative to existing refrigerants. Hydrocarbons, propane (R–290) and isobutane (R–600a) were among the first refrigerants, but due to their flammability and safety purposes, their use was abandoned and the direction of

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researches was shifted towards a safer and inert class of refrigerants [5].Propane (R–290) is widely being used in heat pumps, air conditioners and commercial refrigeration systems. Because of absence of fluorine content in Hydrocarbon refrigerants, they are viewed as alternatives to HFC refrigerants. Hydrocarbons (HCs) are the class of natural occurring substances comprising of propane, butaneand pentane. HCs are excellent refrigerants in many ways like energy efficiency, critical point, solubility, transport, heat transfer properties and environmentally sound but their major concern is their flammability [6].

The paper deals with evaluation of 3 well known flow boiling heat transfer correlation used to find the HTC at liquid phase, two phase and superheated phase and finally calculate the heat transfer area. The actual heat transfer area is calculated using evaporator dimensions. The percentage error between actual heat transfer area v/s theoretical heat transfer area serves to conclude the best correlation which accounts for all factors involved and thereby propose replacement with hydrocarbon refrigerant in the horizontal evaporator without any significant modifications.

METHODOLOGY:

The methodology carried out can be put up as finding the best empirical evaporator correlation for two-phase heat transfer co-efficient in order to obtain the overall heat transfer co-efficient in mixed phase thereby calculating the heat transfer area required for the given capacity. These results are validated by comparing with the evaporator dimensions of a 95W BOSCH domestic refrigerator.

In evaporator, the phase of refrigerant changes continuously leading to a continuous change in the heat transfer co-efficient. Thus we need to find out the heat transfer co-efficient in the two-phase region at a particular composition and region and incorporate it in the overall heat transfer coefficient and find the area using heat exchanger design relation.

In this paper we consider LMTD method to find the area of the evaporator required.

Q = U.A.LMTD

LMTD= Log mean temperaturedifference= $\frac{(Th1-Th2)-(Th1-Tc2)}{\ln[(Th2-Tc1)/(Th1-Tc2)]}$

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Heat transfer areas in finned tube evaporators:

Bare tube area, A_b= (tube perimeter)*(number of fin passages)*(number of tubes)*(width of each passage) = (πd_0) (1/D) (1/B) (D-t)

$A_b = \frac{D-t}{DB} \pi d_0$ m^2 per m^2 face per row

Fin area, A_f= (number of fins)*(two sides of fins)*(width of fin per row)*{number of tubes *(area of cross section of each tube)} = (1/D) 2{1*C-(1/B)$\pi(d_0^2/4)$}

$A_f = \frac{2}{D}[C - \frac{\pi d0^2}{4B}]$ m^2 per m^2 face per row

Total heat transfer area, A_0 = Bare tube area+Fin area

$A_{0t} = A_b + A_f$ m^2 per m^2 face per row

Air side heat transfer coefficient:

A simple expression has been proposed by Air conditioning and Refrigeration Institute, Arlington Va.(1972), which is as follows:

$$h_{air} = 38 * V_f^{0.5}$$

Overall heat transfer coefficient:

A general expression for overall heat transfer coefficient is given by:

$$U_o = \frac{1}{\left(\frac{Ao}{Ai}\right)*\left(\frac{1}{hi}\right) + Rfi*\left(\frac{Ao}{Ai}\right) + \left(\frac{Ao}{Ai}\right)*ri*\frac{\ln\left(\frac{ro}{ri}\right)}{Kw} + \left(\frac{Ao}{(ho(Af\eta f+Ab))}\right)}$$

In the above expression, h is the convective heat transfer coefficient, A_f and A_b are the finned and bare tube areas of the heat exchanger respectively, and 'η' is the fin efficiency. Subscripts "i" and "o" stand for inner and outer sides R_f is the resistance due to fouling if need to be considered. A_0 and A_i are the inside and outside heat transfer area respectively.

The evaporator particulars considered in the paper are as follows:

Refrigerant R290

Tube material = Copper
Thermal conductivity = 400W/mk
Mass flow rate of the refrigerant = 0.0003Kg/s

Table 1: Tube dimensions

Outer diameter of the tube (D_0)	9.53mm
Inner diameter of the tube (D_i)	7.53mm
Root diameter (D_r)	12.45mm
Fins pitch (P)	1.8mm
Fin height (h)	25mm
Fin thickness (t)	0.25mm
Fin spacing (s)	1.55mm
Fins/inch	14
Fins/metre	551
Transverse Pitch (B)	0.025mm
Longitudinal pitch (C)	0.0125mm
Length of the evaporator (L)	890mm
Height of the evaporator (H)	760mm

Following are the correlations employed to find the HTC and Heat transfer area:

- **Kandlikar's correlation (1983)**

$$\frac{htp}{hl}=C1C_0^{C2}(25Frlo)^{C5}+C3(Bo)^{C4}Ffl$$

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The values of constants $C1$ - $C5$ are given in Table 2

Table 2: Values of Constants

Constant	Convective region	Nucleate boiling Region
C1	1.1360	0.6683
C2	-0.9	-0.2
C3	667.2	1058
C4	0.7	0.7
C5	0.3	0.3

C5=0,for vertical tubes and for horizontal tubes with Frl>0.04

C_0<0.65 – Convective boiling region
C_0>0.65 - Nucleate boiling region
C_0= Convection number = $(\frac{\rho g}{\rho l})^{0.5}(\frac{1-x}{x})^{0.8}$
B_0 = Boiling number =$q/(Gh_{fg})$
G= m/A_i kg/m^2-s
q= $U*(T_H-T_S)W$
$U=(\frac{1}{\frac{1}{h(air)}+\frac{1}{h(cb)}})W/m^2\text{-}K$

$h_{cb}=(F_0*h_{l0})W/m^2\text{-}K$
$F_0=F*(1-x)$
$F=2.35(0.213+1/X_{tt})$
$1/X_{tt}= (\frac{x}{1-x})^{0.9} * (\frac{\rho g}{\rho l})^{0.5} * (\frac{\mu g}{\mu l})^{0.1}$

$F_{rl0}=\frac{G^2}{\rho l^2 * g * di}$

$C_5=0$ for $F_{rl0}>0.04$

- **Chaddock-Brunemann's Correlation:**

$Htp=1.91h_1*[Bo.10^4+1.5(1/Xtt)^{0.67}]^{0.6}$

Bo=Boiling Number= $\frac{\frac{Q}{A}}{hfg*(\frac{m}{A})}$

$X_{tt}=(\frac{1-x}{x})^{0.9}(\frac{\rho g}{\rho f})^{0.5}*(\frac{\mu f}{\mu g})^{0.1}$……Lockhart-Martenelli parameter

- **Gungor-Winterton Correlation:**

$H_{tp}=(SS_2+FF_2)h_{spl}$

Where $h_{spl,}$ is calculate with (2), and the factors S, S_2, F, and F_2 are calculated by

$$S=1+3000Bo^{0.86}$$

$$F=1.12(\frac{x}{1-x})^{0.75}(\frac{\rho l}{\rho g})^{0.41}$$

$S_2= Fr_l^{(0.1-2Frl)}$ if horizontal and $F_{rl}<0.05$
1 otherwise

$F_2= Fr_l^{1/2}$ if horizontal and $F_{rl}<0.05$
1 otherwise

RESULTS AND DISCUSSION:

The plot shows kandlikar correlation gives a higher value in case of R290 and lower value for R170.This is due to lower value of martenalli factor which in turn depend on higher density and viscosity values at liquid phase.

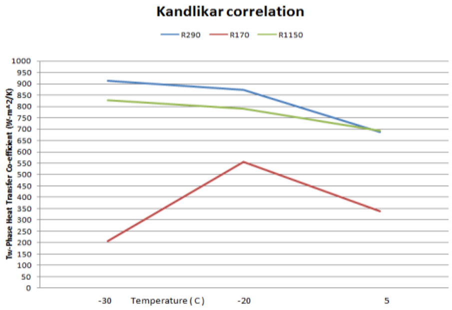

Fig.1:shows the variation of 2phase HTC with temperature for R290, R170, R1150 using Kandlikar correlation.

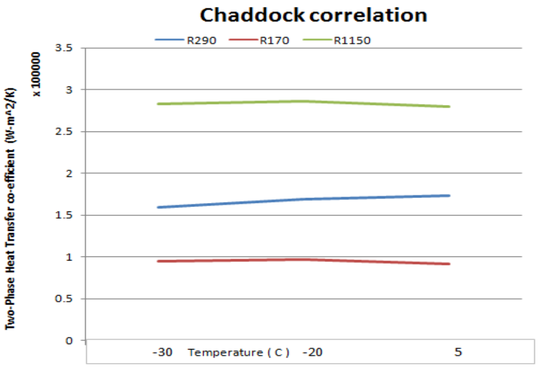

Fig.2:shows the variation of 2phase HTC with temperature for R290, R170 and R1150 using Chaddock correlation.

The plot shows Chaddock correlation gives a higher value in case of R1150 and lower value for R170 while the value for R290 lies in between them. This trend is obtained due to the influence of higher values of Latent heat of vapourisation.

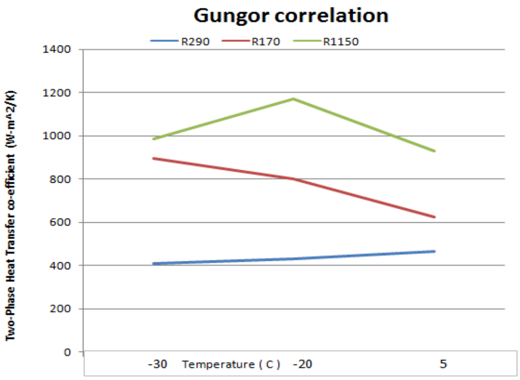

Fig.3: shows the variation of 2phase HTC with temperature for R290, R170, R1150 using Gungor correlation

The plot shows Gungor correlation gives a higher value in case of R1150 and lower value for R290. The higher values of liquid heat transfer co-efficient obtained from Dittus boelter correlation is found to be the reason for the variation of 2phase heat transfer co-efficient.

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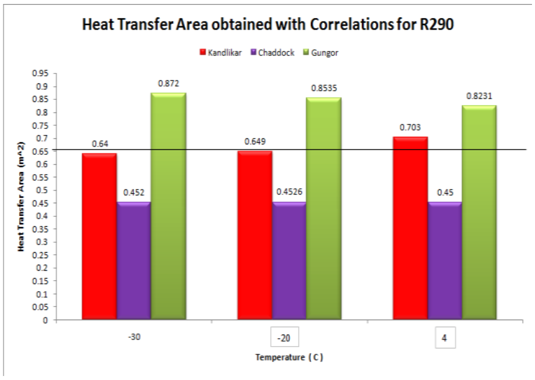

Fig.4: shows the Heat Transfer areas obtained from Kandlikar, Chaddock and Gungor correlation at working temperatures for R290.

Kandlikar correlation gives the area near to the actual heat transfer area obtained from system dimensions

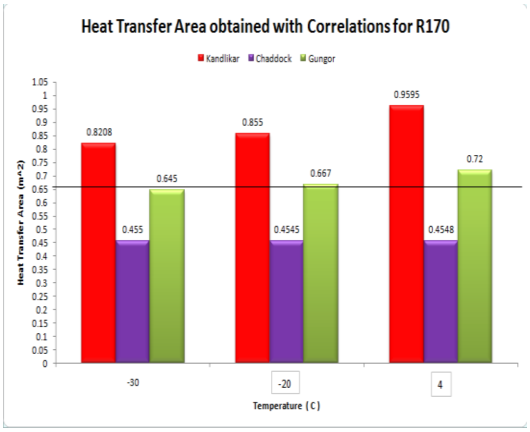

Fig.5:shows the Heat Transfer areas obtained from Kandlikar, Chaddock and Gungor correlation at working temperatures for R170

Gungor correlation gives the area near to the actual heat transfer area obtained from system dimensions.

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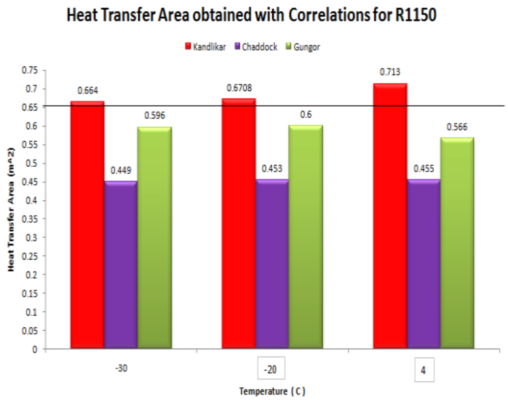

Fig.6: shows the Heat Transfer areas obtained from Kandlikar, Chaddock and Gungor correlation at working temperatures for R1150.

Gungor and Kandlikar correlations gives the area near to the actual heat transfer area obtained from system dimensions.

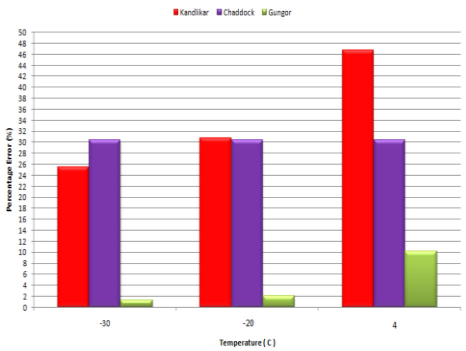

Fig 7a

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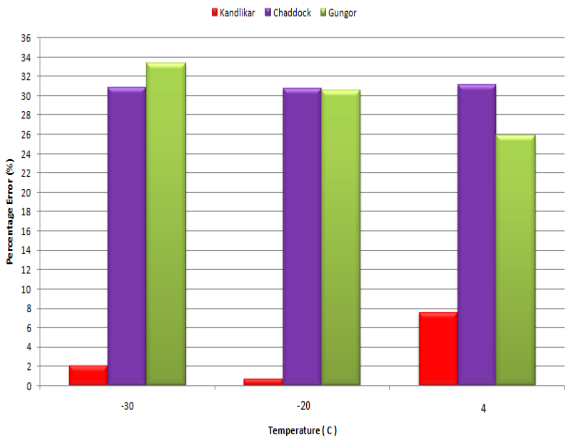

Fig: 7b

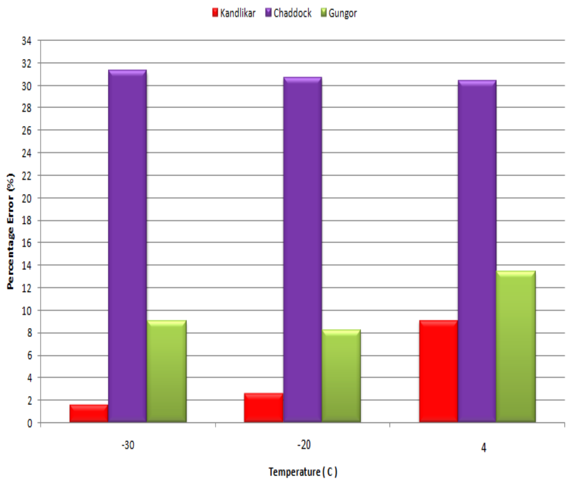

Fig: 7c

Fig.7a.b.c: shows the variation of percentage error in heat transfer area at working temperatures obtained from Kandlikar, Chaddock and Gungor correlation.

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The plot shows Kandlikar and Gungor gives least error with R1150 while Kandlikar alone best predicts for R170, Gungor for R170.

CONCLUSIONS:

As per the Kyoto and Montreal protocols, the harmful refrigerants are to be phased out and are to be replaced with alternate environmental friendly refrigerants. The objective of this paper is to evaluate environmental friendly hydrocarbon refrigerant as a replacement in the existing evaporator. On the basis of results, the following conclusions may be drawn.

Heat transfer co-efficient at two phase for three temperatures encountered in the evaporator have been calculated by three well known evaporator correlations and are in turn used to find the theoretical heat transfer area of the evaporator. This area is compared with actual heat transfer area obtained from evaporator dimensions. This paper shows that kandlikar(for R290, R1150) and Gungor(R170) correlation gives the least percentage error comparatively . Thus hydrocarbon refrigerants may replace HFCs(R-134a) without any system modifications in the evaporator.

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