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Modelling and Simulation of a R744 based Air Conditioning Unit

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ABSTRACT

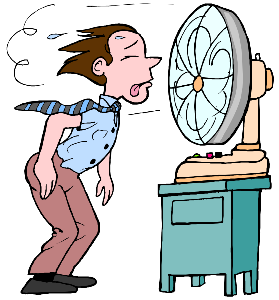
- Performance of transcritical R744 systems is highly sensitive to the high side pressure. An appropriate strategy is required to control the high side pressure.

- In the present study, a R744 based air conditioning cycle with two expansion valves is considered for proper control of high side pressure and quality of refrigerant at evaporator exit.

- A system simulation model is developed for this cycle considering detailed model for each component.

- Using the developed model, the effects of variation of various important environmental parameters on system performance is analysed.

INTRODUCTION



- **Increasing Installation of Heating, Ventilation and Air Conditioning equipments globally**

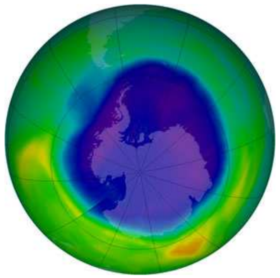
- **Use of synthetic refrigerants is harmful to environment**

Source: http://static8.depositphotos.com/1012832/844/v/950/depositphotos_8447532-Earth-Coughing.jpg

- **Use natural refrigerants like air, water, CO_2, ammonia etc.**

Source: http://www.belvederehotels.net/?page_id=477

Why CO_2 ?



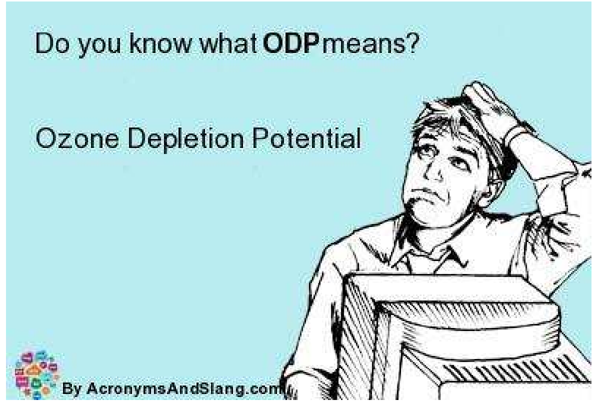
Source: ozone-hole-
2007-NASA

Source: http://english.tempoframing.com/our-products-are-eco-friendly/

CO_2 is a promising refrigerant -

<u>Advantages</u>:

- **Zero ODP and unity GWP.**
- **A non-flammable, non-toxic and naturally available refrigerant.**
- **Compatible with normal lubricants and common machine construction materials.**
- **Weight and space requirement is low due to high volumetric refrigerant capacity.**
- **No recycling required.**



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http://acronymsandslang.com/acronym_image/297/141798fcf29279d75bb5dd236538e2bd.jpg

Transcritical CO_2 system

- Low critical temperature
- High critical pressure
- High operating pressure
- Heat rejection at supercritical pressure

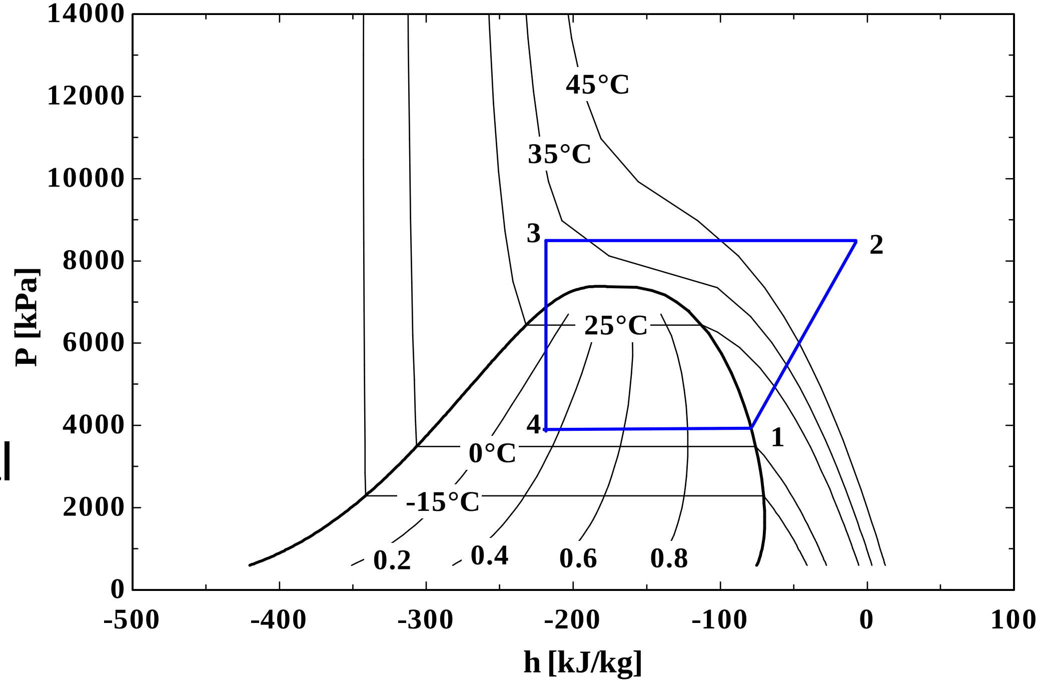


Fig. 1: p-h plot of transcritical CO_2 cycle

Control of high side pressure

- Performance of transcritical system is strongly dependent on **high side pressure** and **gas cooler exit temperature**

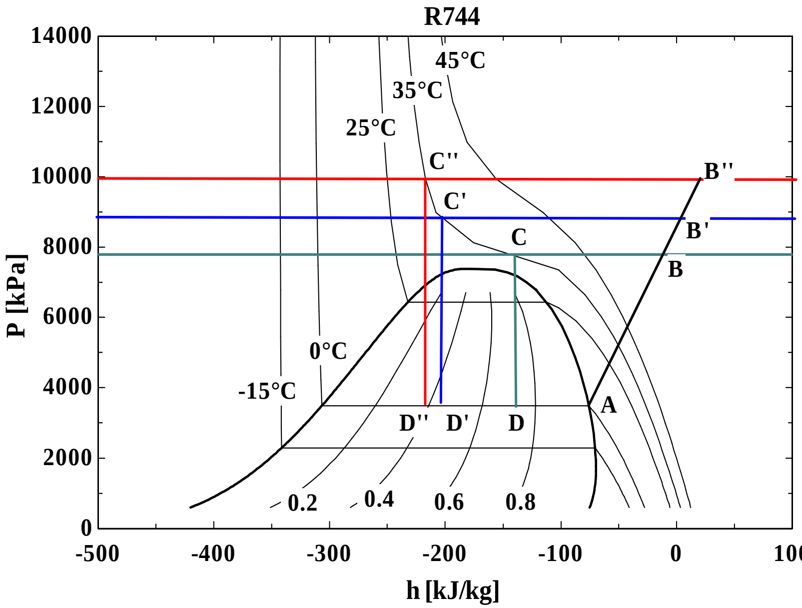


Fig. 2: p-h plot of transcritical CO_2 cycle for gas cooler exit temperature of 35°C

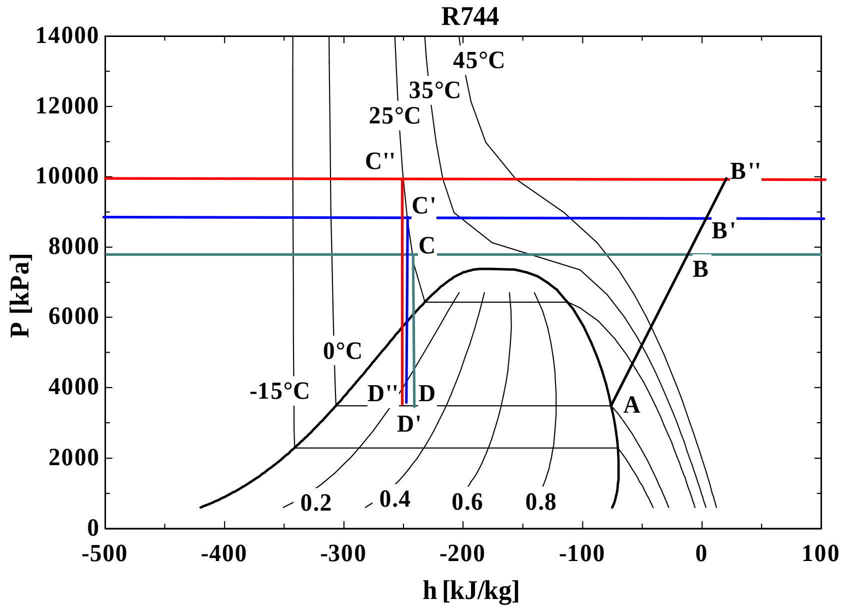


Fig. 3: p-h plot of transcritical CO_2 cycle for gas cooler exit temperature of 25°C

Objectives of present work

- A comprehensive mathematical model is developed for R744 based air conditioning cycle with two expansion valves.
- Design dimensions for different components are predicted using the developed model.
- Optimum operating parameters are identified from the results of numerical simulations.
- The effects of various important environmental parameters on system performance are analyzed.

System description

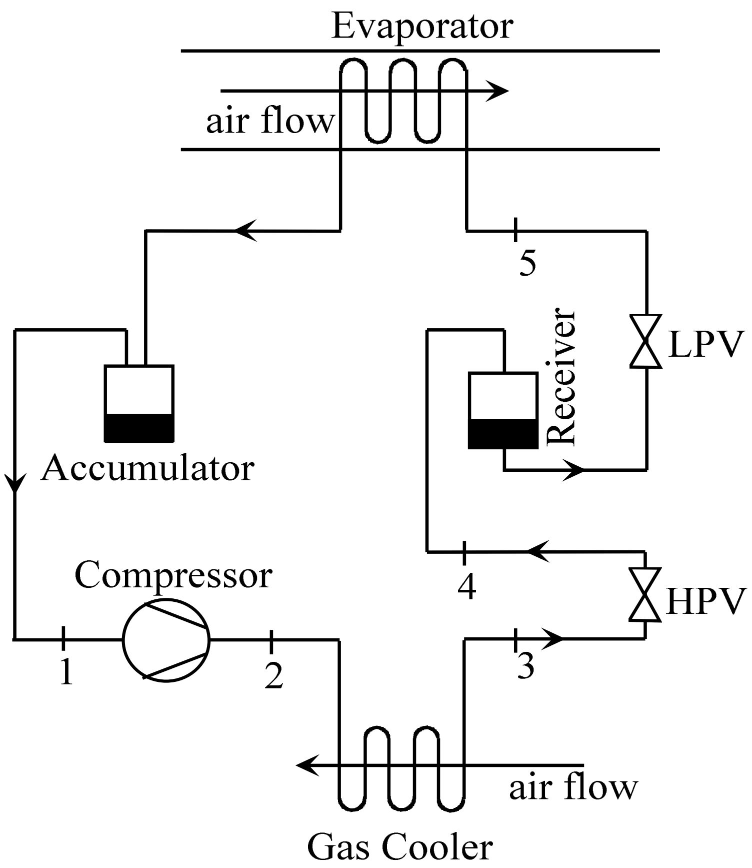


Fig. 4: Schematic of the R744 based air conditioning unit

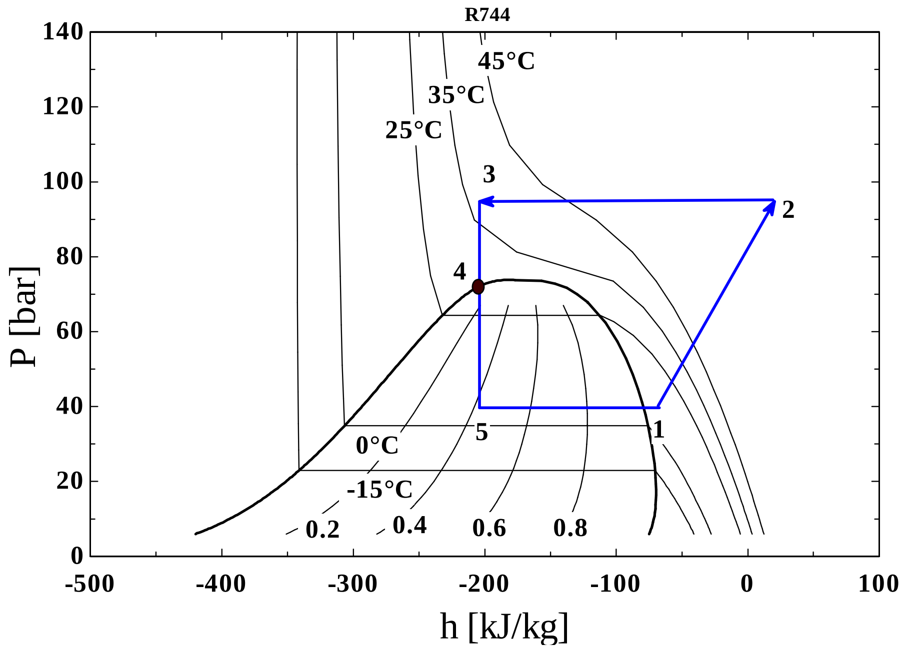


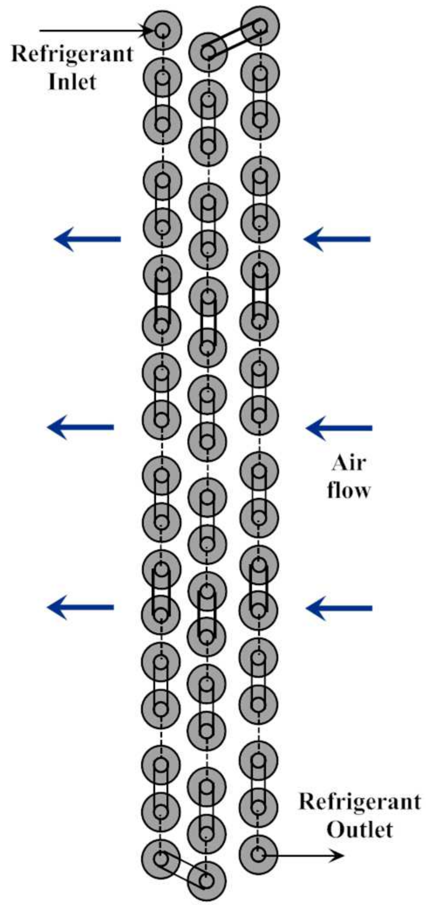
Fig. 5: p-h plot of transcritical R744 based cycle

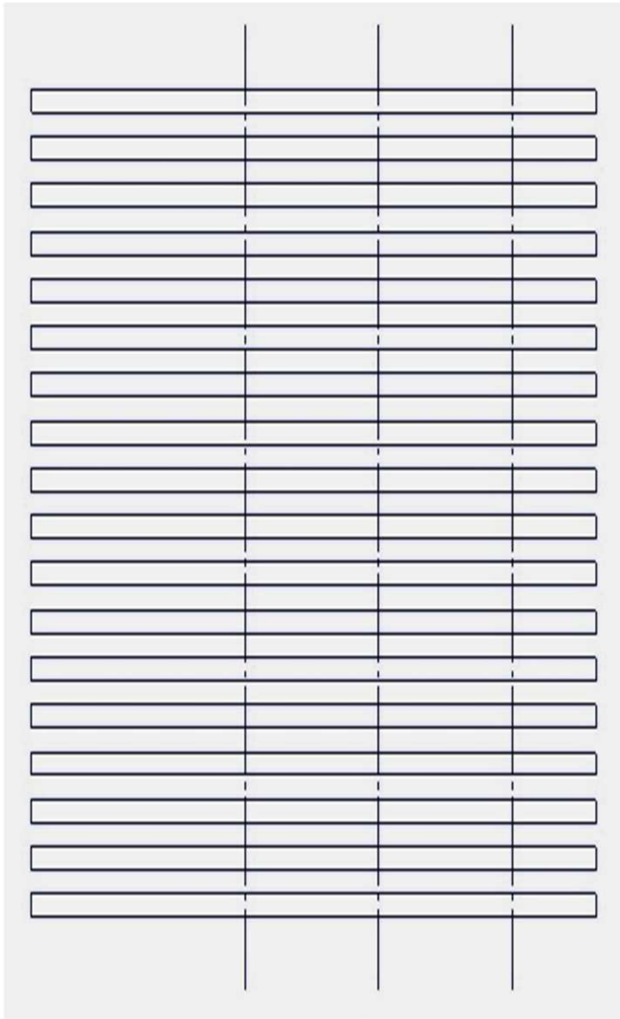
Design procedures adopted

- Elemental log mean temperature difference approach for designing gas cooler
- Elemental log mean enthalpy difference approach for designing evaporator
- Semi empirical model for compressor

Gas cooler

- Counter cross flow arrangement
- Discretized approach to design the gas cooler
- Guess Pressure and temperature at exit
- Node by node marching from exit to inlet
- Iterative procedure to match pressure and temperature obtained at the end of the march

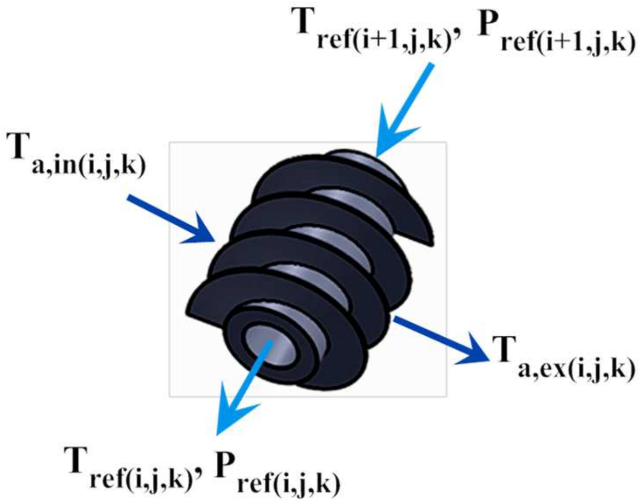




Gas cooler (Mathematical model)

Procedure adopted for designing gas cooler:

- Calculation of surface geometrical characteristics for each node
- Calculation of air side heat transfer coefficient and pressure drop
- Calculation of overall fin efficiency
- Calculation of refrigerant side heat transfer coefficient and pressure drop
- Reduced governing equations for each discretized node



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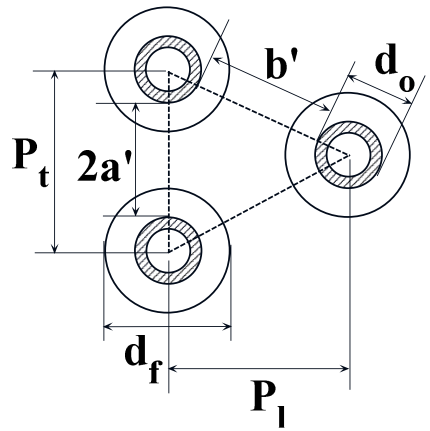
Surface geometrical characteristics [**Shah et al. (2003)**]:

Unfinned base surface area,

$$A_b = n_t N_{row}\left(\pi d_o L - \left(\sqrt{f_p^2 + (\pi d_o)^2}\right)\times f_t\left(\frac{L}{f_p}\right)\right)$$

Fin surface area,

$$A_f = n_t N_{row}\left(\frac{L}{f_p}\right)\times\left(\frac{1}{2}\pi\left(d_f^2 - d_o^2\right) + \pi d_f d_t\right)$$



Total surface area,

$$A_o = A_b + A_f$$

Minimum free flow area,

$$A_{\min} = \left[(n_t - 1)c' + (P_t - d_o) - \left(d_f - d_o\right) f_t\left(\frac{1}{f_p}\right)\right]\times L$$

$$c' = \begin{cases} 2a' & if \quad 2a' < 2b' \\ 2b' & if \quad 2b' < 2a' \end{cases}$$

$$2a' = (P_t - d_o) - \left(d_f - d_o\right) f_t\left(\frac{1}{f_p}\right) \qquad b' = \left(\left(\left(\frac{P_t}{2}\right)^2 + P_l^2\right)^{1/2} - d_o\right) - \left(d_f - d_o\right) f_t\left(\frac{1}{f_p}\right)$$

Contd.

Air side heat transfer coefficient and air side pressure drop [**Pongsoi et al. (2013)**]:

$$\alpha_a = \frac{j \rho_a V_{\max} C_{p,a}}{\Pr^{2/3}} \qquad \Delta P = f \times \left(\rho_a V_{\max}^2 / 2 \right) \times \left(\frac{A_o}{A_{\min}} \right)$$

$$j = 0.215 \mathrm{Re}_{do}^{-0.4059} \qquad f = 0.4852 \mathrm{Re}_{do}^{-0.2156} \left(\frac{f_p}{d_o} \right)^{0.4771}$$

Fin efficiency [**Pongsoi et al. (2013)**]:

$$\eta_f = \frac{2\psi}{\phi(1+\psi)} \frac{I_1(\phi R_o) K_1(\phi R_i) - I_1(\phi R_i) K_1(\phi R_o)}{I_0(\phi R_i) K_1(\phi R_o) + I_1(\phi R_o) K_0(\phi R_i)}$$

$$\phi = (r_o - r_i)^{3/2} \left(\frac{2\alpha_o}{k_f A_p} \right)^{1/2} \qquad A_p = f_t (r_o - r_i) \qquad \psi = \frac{r_i}{r_o} \qquad R_o = \frac{1}{1-\psi} \qquad R_i = \frac{\psi}{1-\psi}$$

Overall fin efficiency:

$$\eta_o = 1 - \left(\frac{a_f}{a_o} \right) (1 - \eta_f)$$

Contd.

Refrigerant side heat transfer coefficient and pressure drop are predicted using correlation proposed by **Pitla et al. (2002)**:

$$Nu = \left(\frac{Nu_{wall} + Nu_{bulk}}{2} \right) \frac{k_{wall}}{k_{bulk}}$$

Nu_{wall} and Nu_{bulk} are calculated using **Gnielinski correlation** within the range $2300<Re<10^6$ and $0.6<Pr<10^5$:

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3}-1\right)+1.07}$$

For $Re>10^6$, Nu_{wall} and Nu_{bulk} are calculated using **Petukhov-Popov-Kirilov correlation**:

$$Nu = \frac{(f/8)\mathrm{Re}\,\mathrm{Pr}}{12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3}-1\right)+1.07}$$

Friction factor *'f'* is given by: $f = (0.79\ln(\mathrm{Re})-1.64)^{-2}$

Contd.

Governing equations for each node [**Yin et al. (2001)**]:

Refrigerant side: $$Q_{node} = m_{ref}\left(h_{i+1,j,k} - h_{i,j,k}\right)$$

$$Q_{node} = \alpha_{ref}\, a_{ref}\, LMTD_{ref}$$

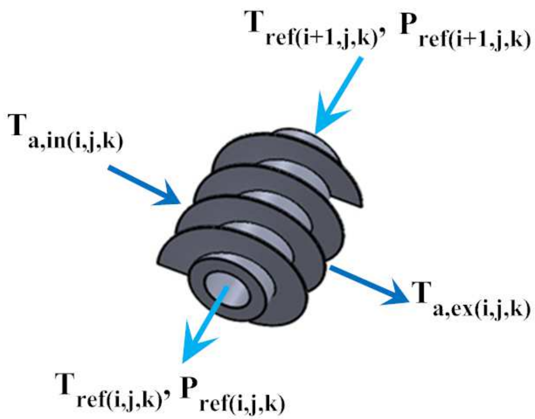
$$LMTD_{ref} = \frac{\left(T_{ref(i+1,j,k)} - T_{s(i,j,k)}\right) - \left(T_{ref(i,j,k)} - T_{s(i,j,k)}\right)}{\ln\left(\frac{T_{ref(i+1,j,k)} - T_{s(i,j,k)}}{T_{ref(i,j,k)} - T_{s(i,j,k)}}\right)}$$

$$P_{ref(i+1,j,k)} - P_{ref(i,j,k)} = \Delta P_f = f\,\frac{G^2 \times dL}{2 \times \rho \times d_i}$$

Air side: $$Q_{node} = m_a C_{pa}\left(T_{a,ex(i,j,k)} - T_{a,in(i,j,k)}\right)$$

$$Q_{node} = \alpha_a \eta_o a_{node} LMTD_a$$

$$LMTD_a = \frac{\left(T_{s(i,j,k)} - T_{a,in(i,j,k)}\right) - \left(T_{s(i,j,k)} - T_{a,ex(i,j,k)}\right)}{\ln\left(\frac{T_{s(i,j,k)} - T_{a,in(i,j,k)}}{T_{s(i,j,k)} - T_{a,ex(i,j,k)}}\right)}$$



Gas cooler (Dimensions predicted)

Tube	
Material	Copper
Inner diameter	5.5 mm
Outer diameter	9.5 mm
Longitudinal tube pitch (P_l)	30 mm
Transverse tube pitch (P_t)	27 mm
Finning length of tube	480 mm
Number of tubes in each row	18
Number of tube rows	3
Fin	
Material	Copper
Outer diameter	26 mm
Fin thickness	0.19 mm
Fin pitch	3 mm
Surface area	
Total fin surface area	8.0837 m^2
Unfinned base surface area	0.7243 m^2
Total surface area, A	8.8080 m^2

Evaporator

Like gas cooler similar discretized approach is adopted for designing the evaporator

Evaporator coil is divided into two different zones based on the quality of refrigerant:

- Two phase zone
- Single phase vapour zone

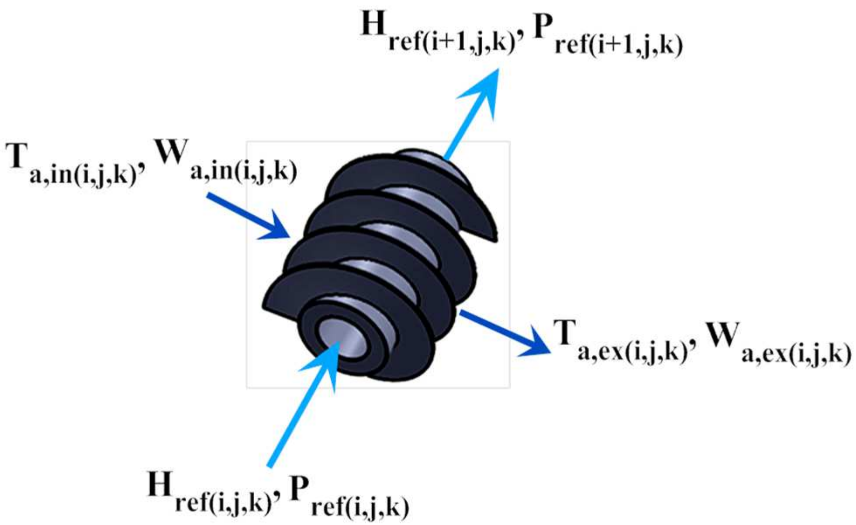
Each zone is then further subdivided based on the occurrence of condensation of moisture on coil surface:

- Wet surface (for water film temperature less than DPT of air)
- Dry surface (for water film temperature greater than DPT of air)

Evaporator (Mathematical model)

Procedure adopted for designing the evaporator

- Calculation of surface geometrical characteristics for each node
- Calculation of air side heat transfer coefficient and pressure drop
- Prediction of zone based on quality of refrigerant
- Prediction of Wet/dry region based on condensation of moisture on surface



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For refrigerant in **two phase region**, heat transfer coefficient is estimated using the correlation proposed by **Yoon et al. (2004)**.

Critical quality at which liquid film breaks down is calculated from:

$$x_{cr,t} = 38.27\,\mathrm{Re}_l^{2.12}(1000Bo)^{1.64}\,Bd^{-4.7}$$

For $x<x_{cr,t}$	For $x\geq x_{cr,t}$
$\alpha_i = \left[(S.\alpha_{NB})^2 + (E.\alpha_l)^2\right]^{1/2}$	$\alpha_i = \dfrac{\theta_{dry}\alpha_g + (2\pi - \theta_{dry})\alpha_{wet}}{2\pi}$
$\alpha_{NB} = 55P^{*0.12}(-\log P^*)^{-0.55}M^{-0.5}q''^{0.67}$	$\alpha_g = 0.023\dfrac{k_g}{d_i}\mathrm{Re}_g^{0.8}\,\mathrm{Pr}_g^{0.4}$
$S = \dfrac{1}{1+1.62\times10^{-6}E^{0.69}\,\mathrm{Re}_l^{1.11}}$	$\alpha_{wet} = E.\alpha_l$
$E = \left[1+9.36\times10^3\,x\,\mathrm{Pr}_l\left(\dfrac{\rho_l}{\rho_g}-1\right)\right]^{0.11}$	$E = 1+3000Bo^{0.86}+1.12\left(\dfrac{x}{1-x}\right)^{0.75}\left(\dfrac{\rho_l}{\rho_g}\right)^{0.41}$
$\alpha_l = 0.023\dfrac{k_l}{d_i}\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}$	$\dfrac{\theta_{dry}}{2\pi} = 36.23\,\mathrm{Re}^{3.47}\,Bo^{4.84}Bd^{-0.27}\left(\dfrac{1}{X_{tt}}\right)^{2.6}$

Contd.

For **wet zone**, where condensation occurs on external surface, heat as well as mass transfer takes place. Method proposed by **Threlkeld (1998)** is adopted here to incorporate the mass transfer effects.

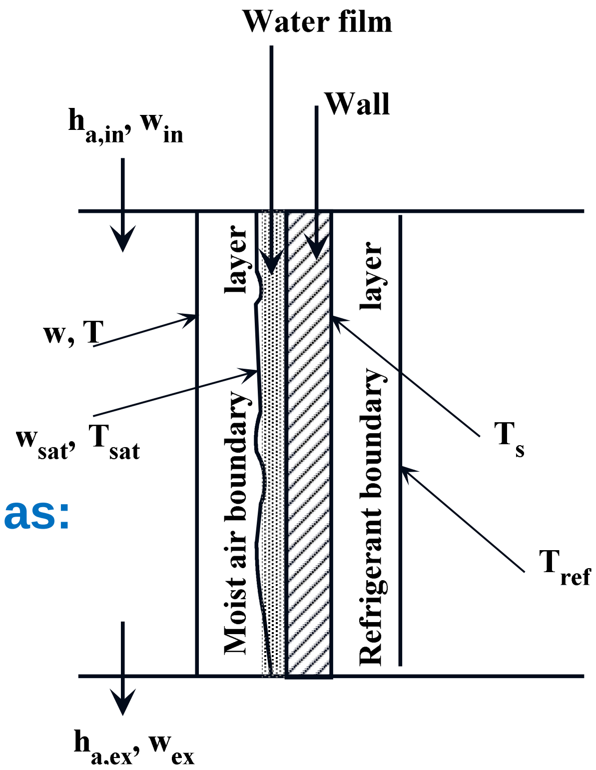
Governing equations for each node of evaporator:

$$m_a h_{a,in} = m_a h_{a,ex} + m_a (w_{in} - w_{ex}) h_f + dQ$$

$$\Rightarrow dQ = -m_a (h_{a,ex} - h_{a,in}) + (m_a dw) h_f$$

$$dQ = \alpha_D A (w - w_{sat}) h_{fg} + \alpha_{c,o} A (T - T_{sat})$$

$$-m_a (w_{ex} - w_{in}) = \alpha_D A (w - w_{sat})$$



In reduced form, the above equations are expressed as:

$$dQ = \frac{\alpha_{c,o} A}{C_{p,a}} (h - h_{sat})$$

$$\frac{dh}{dw} = Le \left(\frac{h - h_{sat}}{w - w_{sat}} \right) + (h_g - 2501 \times Le)$$

Contd.

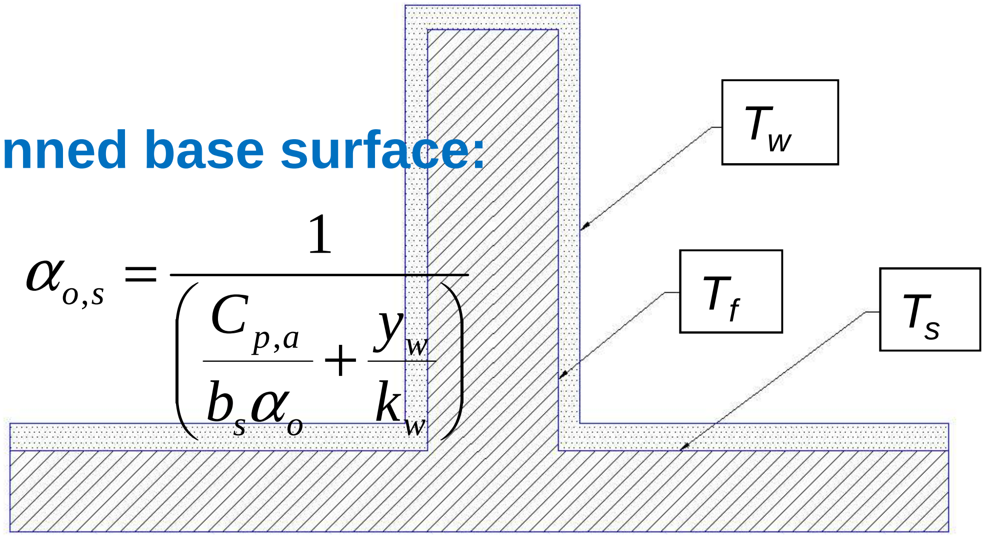
Heat transfer rate through fin surface:

$$dQ_f = \frac{\alpha_o a_f}{C_{p,a}}(h-h_w) = \frac{k_w a_f}{y_w}(T_w - T_f) = \frac{k_w a_f}{y_w b_f}(h_w - h_f)$$

$$= \left(\frac{1}{\frac{C_{p,a}}{\alpha_o a_f} + \frac{y_w b_f}{k_w a_f}} \right) \times (h - h_f) = \left(\frac{\alpha_{o,f} a_f}{b_f} \right) \times (h - h_f)$$

Heat transfer rate through unfinned base surface:

$$dQ_s = \left(\frac{\alpha_{o,s} a_s}{b_s} \right) \times (h - h_s) \quad \text{where,} \quad \alpha_{o,s} = \frac{1}{\left(\frac{C_{p,a}}{b_s \alpha_o} + \frac{y_w}{k_w} \right)}$$



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Total heat transfer rate for each node:

$$dQ_{node} = dQ_s + dQ_f = \left(\frac{\alpha_{o,s} a_s}{b_s}\right) \times (h - h_s) + \left(\frac{\alpha_{o,f} a_f}{b_f}\right) \times (h - h_f)$$

$$= \left(\frac{\alpha_{o,s} a_s}{b_s}\right) \times (h - h_s) + \left(\frac{\alpha_{o,f} a_f \eta_f}{b_f}\right) \times (h - h_s) = \left(\frac{\alpha_{o,w}}{b_w}\right) \times (a_s + a_f \eta_f) \times (h - h_s)$$

$$= \left(\frac{\alpha_{o,w} \eta_o a_{node}}{b_w}\right) \times (h - h_s)$$

where, $b_s = b_f = b_w$ and $\alpha_{o,s} = \alpha_{o,f} = \alpha_{o,w}$

Refrigerant side heat transfer rate:

$$dQ_{node} = \alpha_i a_{ref} (T_s - T_{ref}) = \frac{\alpha_i a_{ref}}{b_{ref}} \times (h_s - h_{ref})$$ where, $b_{ref} = \dfrac{(h_s - h_{ref})}{(T_s - T_{ref})}$

Contd.

Now equating air side heat transfer rate and refrigerant side heat transfer rate we get:

$$dQ_{node} = \left(\frac{\alpha_{o,w}\eta_o a_{node}}{b_w} \right) \times (h - h_s) = \frac{\alpha_i a_{ref}}{b_{ref}} \times (h_s - h_{ref})$$

$$= \left(\frac{1}{\dfrac{b_w}{\alpha_{o,w}\eta_o a_{node}} + \dfrac{b_{ref}}{\alpha_i a_{ref}}} \right) \times (h - h_{ref}) = U_o a_{node} \times (h - h_{ref})$$

where,

$$U_o = \left(\frac{1}{\dfrac{b_w}{\alpha_{o,w}\eta_o} + \dfrac{b_{ref} a_{node}}{\alpha_i a_{ref}}} \right)$$

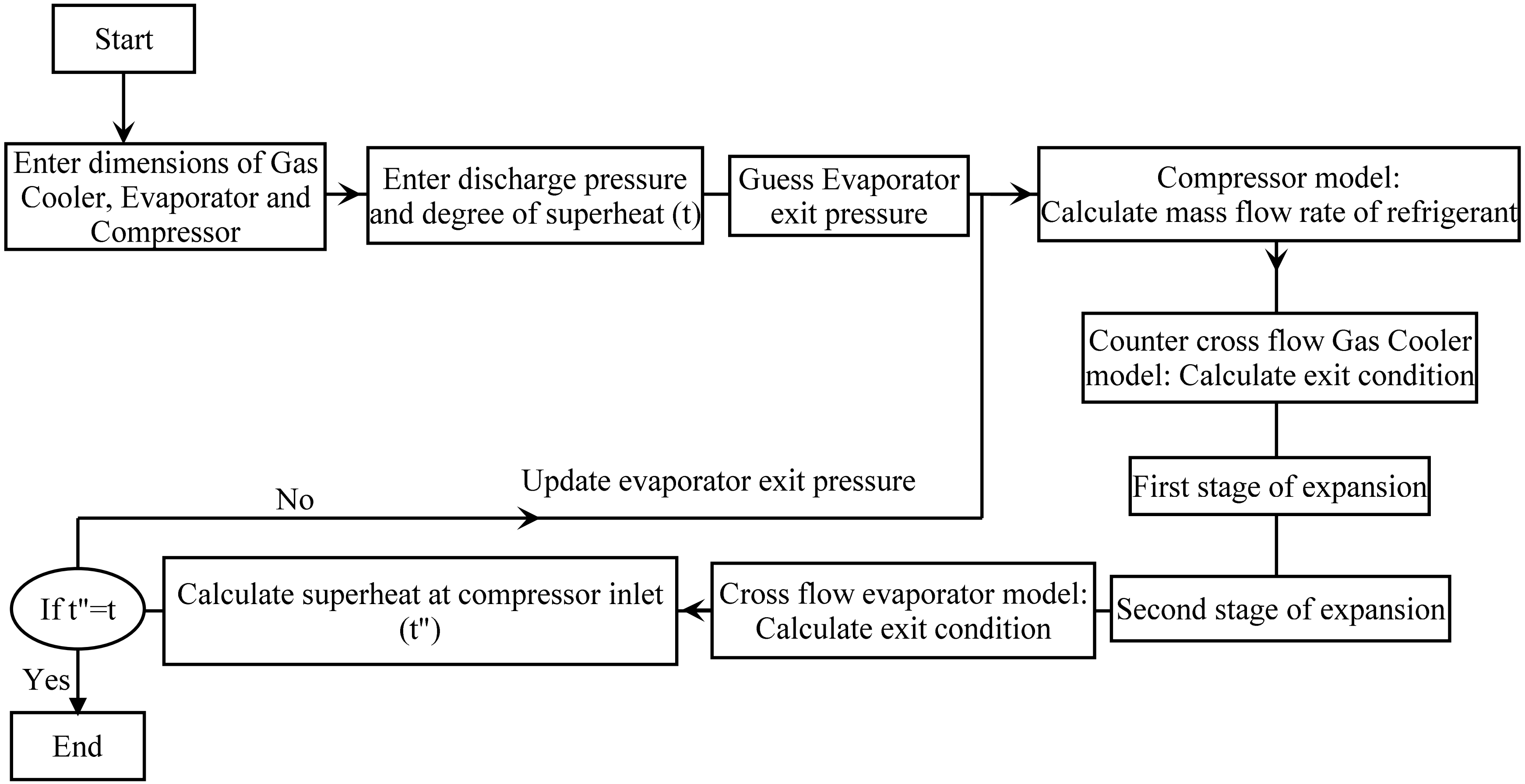
In order to account for the variation in air enthalpy, logarithmic enthalpy difference is introduced in the above equation:

$$dQ_{node} = U_o a_{node} \times \Delta h = U_o a_{node} \times \frac{h_{a,in} - h_{a,ex}}{\log \dfrac{h_{a,in} - h_{ref}}{h_{a,ex} - h_{ref}}}$$

Evaporator (Predicted dimensions)

Tube	
Material	Copper
Inner diameter	5.5 mm
Outer diameter	9.5 mm
Longitudinal tube pitch (P_l)	30 mm
Transverse tube pitch (P_t)	30 mm
Finning length of tube	300 mm
Number of tubes in each row	13
Number of tube rows	3
Fin	
Material	Copper
Outer diameter	26 mm
Fin thickness	0.19 mm
Fin pitch	4 mm
Surface area	
Total fin surface area	2.7367 m^2
Unfinned base surface area	0.3324 m^2
Total surface area, A	3.0691 m^2

Flowchart of the model



Results & discussions

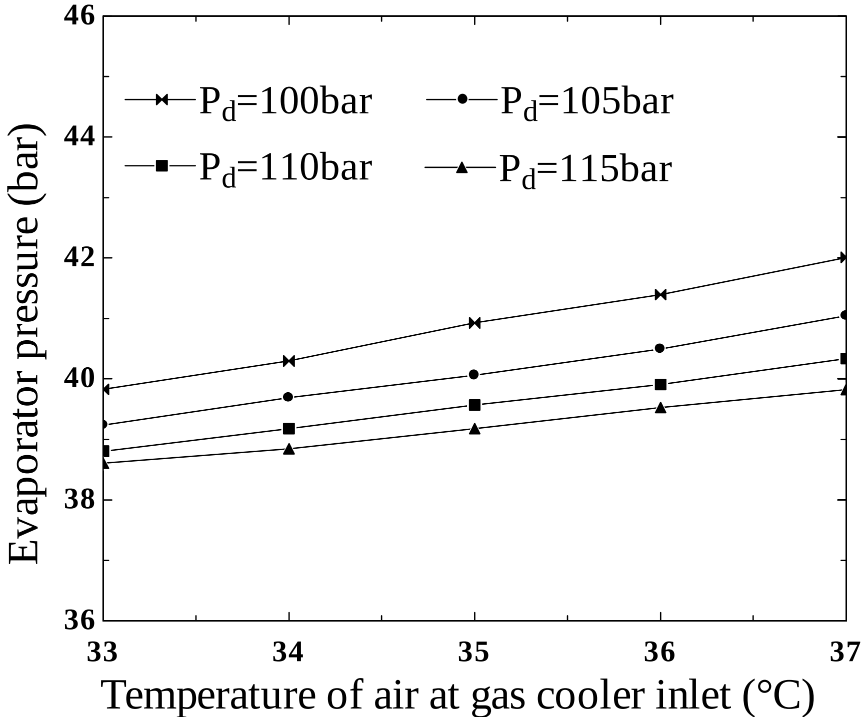


Figure 6: Variation of evaporator pressure with change in temperature at gas cooler inlet

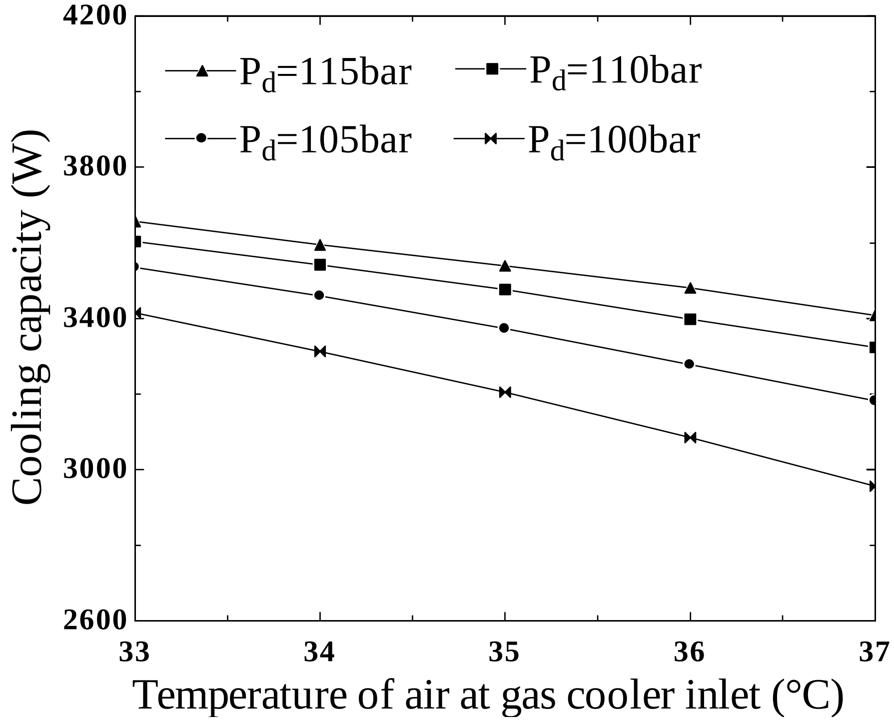


Figure 7: Variation of evaporator capacity with change in temperature at gas cooler inlet

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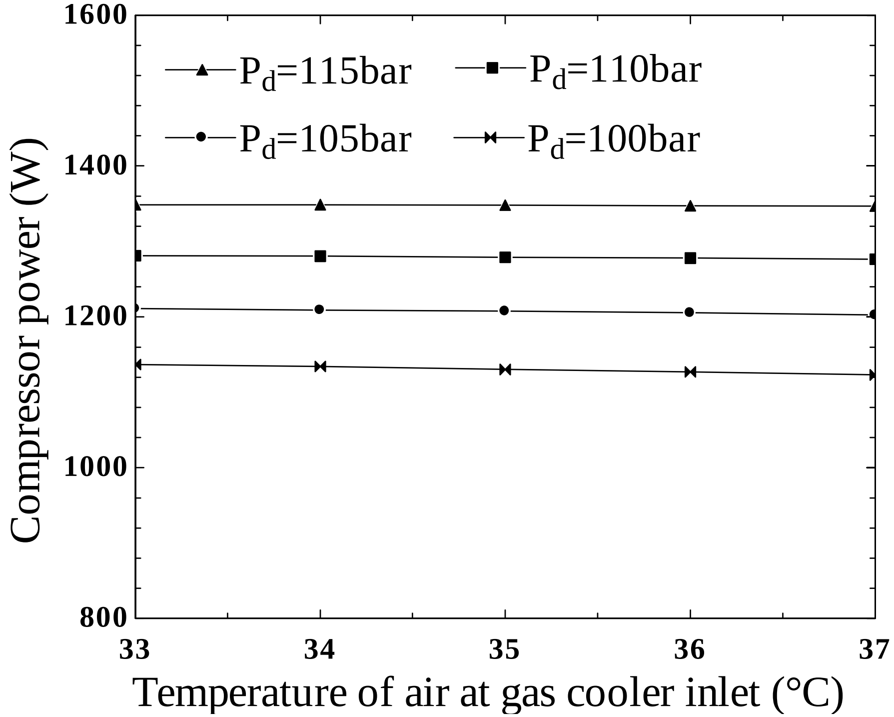

Figure 8: Variation of compressor power with change in temperature at gas cooler inlet

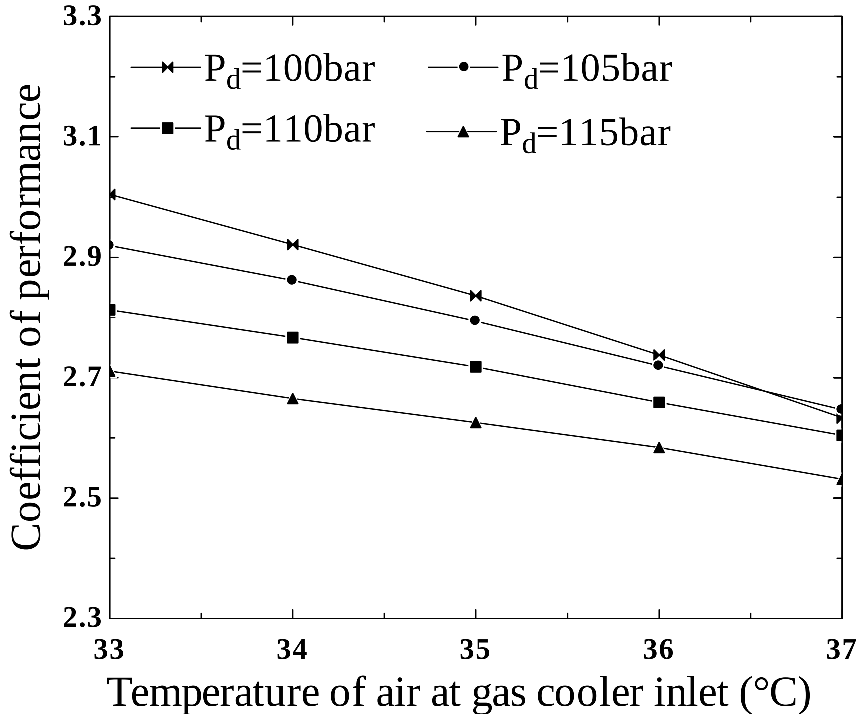

Figure 9: Variation of COP with change in temperature at gas cooler inlet

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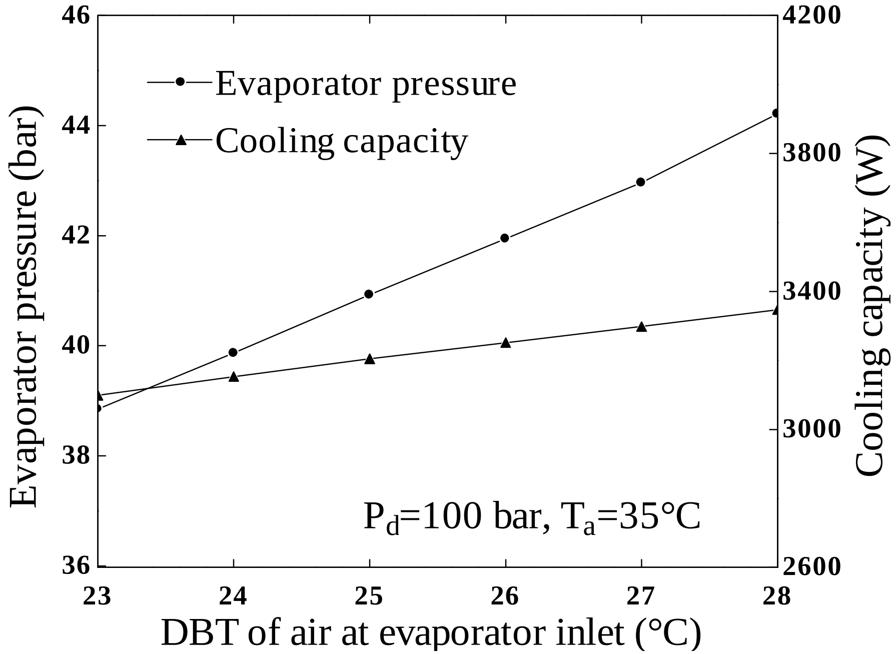

Figure 10: Variation of evaporator pressure and cooling capacity with changes in DBT of air at evaporator inlet

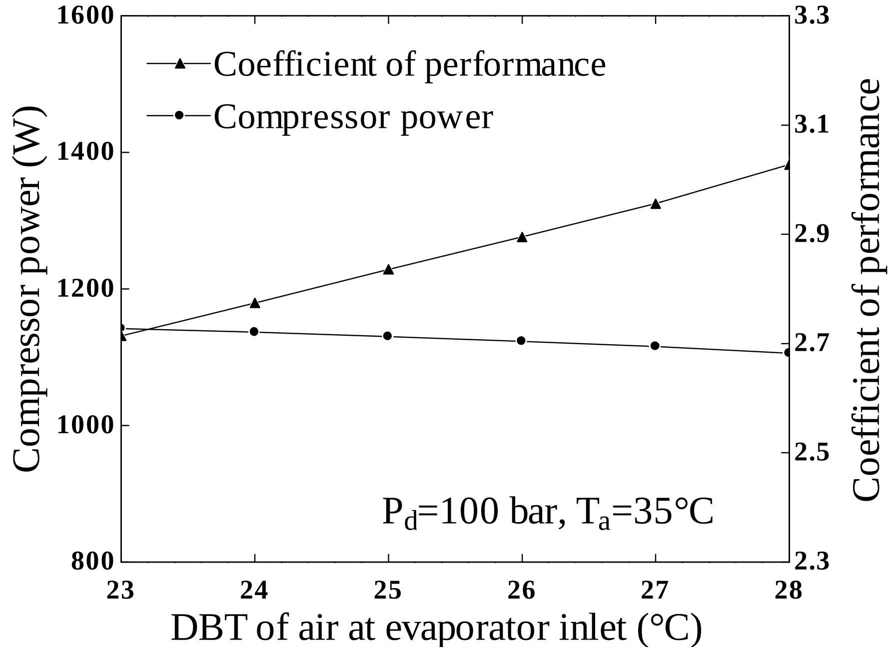

Figure 11: Variation of compressor power and COP with changes in DBT of air at evaporator inlet

Optimum operating parameters

Ambient temperature (°C)	Performance of system with double stage expansion			
	Optimum discharge pressure (bar)	**Differential pressure drop (bar)**	**Cooling capacity (W)**	**System COP**
33	95	20.97	3205.8	3.04
34	95	20.33	3067.8	2.92
35	100	26.07	3204.5	2.84
36	100	25.48	3084.4	2.74
37	105	31.00	3182.0	2.65
38	105	30.49	3074.7	2.57
39	110	35.86	3146.1	2.47

Conclusions

- Comprehensive mathematical model has been developed for R744 based air conditioning cycle with two expansion valves.
- Numerical simulations are carried out using the model.
- From the results obtained, the effects of variation of various important environmental parameters on system performance are analyzed.
- Optimum operating conditions are also identified for which the system attains maximum COP.

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