International Journal of Engineering Innovation & Research
Volume 3, Issue 6, ISSN: 2277 – 5668

Investigating the Suitability of Air as a Refrigerant in an Open Cycle Air Cooling System – Using CFD

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***Abstract* – This paper investigates the feasibility of using air as a coolant media in an air-conditioning open cycle system. Recent research in this area has been mainly in using closed cycle systems, using relatively higher compression levels, when compared with currently used refrigerant gas systems. This investigation focuses on an open cycle based on a series of compression and expansion cycles. Thus reducing the size of air compressor. Subsequently reducing the consumption of energy.**

Methodology employed was using CFD software to model the system. Airflow stream lines and air flow pressure contours were recorded. Data was collected and tabulated for discussions. Several models were generatedby increasing the number of compression/expansion cycles, searching for optimum solution.

CFD generated results and calculations were compared with commercially available air-conditioning system. It was concluded that CFD techniques can be used to develop effective sustainable cooling systems, based on a simplified easy to manufacture geometric model. Air can be used in an open cycle cooling system with an acceptable level of sensible cooling and low energy consumption. Thus avoiding the need for manufactured refrigerant gases which are costly to produce and can be harmful to the environment.

***Keywords* – Air Refrigerant, Air-Conditioning, Sustainable Development, CFD.**

I. Introduction

Recent research in environmentally friendly cooling systems investigated the use of air as a refrigerant gas. However, due to the limitations of the air pressure cycle associated with air when used as a refrigerant, Progress in this field has remained slow. The challenge continues to investigate the possibilities of using air as a refrigerant. With the aim of reducing fuel consumptions, and reducing the carbon footprint.

In this paper CFD software was used to investigate the practicality of using air as a refrigerant in an open cycle system. The aim was to remain as practically as possible when considering pressure levels at the inlet compression stage, keeping the size of the air compressor down. Similarly, at the heat rejection stage, the lowest possible forced air convection air flow rate was considered. Keeping the size of the forced air fan down. The aim of this research was based on searching for the optimal combination of low air compression/expansion and temperature of cooled air output. The paper unlike previous works, relies on a series of compressions/expansion cycles, where heat is rejected during each compression stage, with the minimum possible pressure applied at the first compression stage.

II. Drive to Develop Environmental Cooling Systems – Brief Discussion

Recent demand for sustainable development has prompted a number of institutions in researching and developing the usage of air as a refrigerant media in an open cycle system. A paper by C. M. Bartolini (1992) [1] discusses this subject in details as briefly shown in the following five paragraphs.

The aims of the scientific community in the 1990s in replacing CFC refrigerant gasses in air-conditioning equipment, in order to solve the well-known serious pollution problems. One of the solutions which was considered was the usage of air cycle refrigeration systems that are more commonly used in the air-conditioning of aircraft than in surface and stationary applications because the lightweight compact equipment. Reason being that such systems known for their lightweight usually offset its inherently low efficiency.

The air cycle refrigeration units have been deeply investigated in the literature with both dry and wet air working fluid; the thermodynamic analysis pointed out the influence of the compressor, turbine, and heat exchanger efficiencies on the global performance and the influence of the expansion procession the final thermodynamic conditions of air in open cycle conditions. An expansion that makes possible to maintain the steam in equilibrium with dry air allows a lower temperature at the end of transformation especially with high pressure ratios, affecting improvements in the coefficient of performance; such expansion can be more likely obtained in a turbine rotating at lower speeds.

Open air cycle systems were considered uneconomical in many applications because of the high power required. However, they are used in specialized applications where efficiency is not the primary factor, such as in remotely located military bases. Another example of the usage in aircraft air conditioning systems where the compression of the outside low pressure in flight ambient conditions to the near standard atmospheric pressure is in any case is operated. All these reasons suggest to look for new ways to reduce the cost/benefit ratio, maintaining the compactness and reliability.

In relatively small room volumes requiring lower air flow rates open cycle systems using volumetric machines may not be suitable due to the possibility of their internal lubrication contaminating the airflow. Until now in such applications, centrifugal compressors and radial turbines have found a wide application; the low flow rates involve relatively smaller impeller dimensions which require higher angular velocity speeds in order to get the necessary value of the tip velocity.

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Volume 3, Issue 6, ISSN: 2277 – 5668

The earliest recorded studies on such systems dates back to the thirties and started in Germany and Japan. Since then several theoretical and experimental researchers have been presented in literature.

Examples of similar recent work are given below;

Established companies such GEA (2014) [2] do offer cooling systems using air rather than a refrigerant gas.

A research paper by D J G Butler (2001)[3] based on a closed cycle system discusses the limitations associated with the usage of such a system due to the low perceived energy efficiency of air cycle systems. An example of where such a technology is currently utilized is Germany's ICE-3 high speed trains. A suggestion in Mr Butler's [3] paper has been made referring to the possibility of using an open cycle system. This is the research topic of this paper, an open cycle system.

The above examples clearly indicate that there is an interest in using such cooling systems and that there is room for further development. With the ultimate aim of driving capital and operation costs down, while maintaining the performance and reliability at an acceptable level or even better when compared with currently available vapor compression systems.

Recent drives in phasing out hydro chlorofluorocarbon or HCFC are progressing, the main reason being their well-known impact on the depletion of the Ozone layer. Refrigerant gases such as R410, as an example, are continuously being introduced worldwide. However R410 refrigerants are known to have a global warming potential, a negative point against R410 offset by the associated system higher operating efficiency.

The drive remains in the air-conditioning industry in developing further more efficient environmentally friendly air cooling systems. This paper investigates the possibility of using air as a "refrigerant" in an open cycle system as suggested by D J G Butler (2014) [3]. Addressing technical and environmental issues. Such a scheme is clearly environmentally friendly with no refrigerants used, and simple to manufacture, therefore economical to use. Such a scheme can contribute to sustainable development in air-conditioning.

This paper uses CFD tools way to search for ways in improving such a system. Multiple compression and decompression stages will be considered in optimizing the cooling power ratio to power consumed.

III. Description of the CFD Model

Description of model can be seen in Fig 1. The aim was to develop a simple geometry where air can be compressed, reject heat, and then allowed to expand back to atmospheric pressure levels. The remaining CFD generated images in Figs 2 to 7 show graphical results. Fluent software was used to develop the model. An example of how such a model can be put into practice is as shown in Fig 8.

The analysis investigated; one, two, three and four stages of compression and expansions cycles. Probing for an optimal solution, low compression, hence energy, versus an acceptable air cooling. The heat rejection segment of the model was investigated by varying the forced air convection rate. Thus investigating what is the minimal forced air convection rate required to remove heat from the heat rejection surface of the pipe wall.

War air entering the tube inlet was applied at the CFD model tube inlet in the form of mass flow rate, the associated pressure values generated by the CFD analysis were then used in the air compressor calculations shown in Appendix 1.

Assumptions:

- Air as an ideal gas. Note: According to E Rathakrishnan (2006) [4], many of the familiar gases such as air, nitrogen, oxygen, hydrogen, helium, argon, neon, krypton, and even heavier gases such as carbon dioxide can be treated as ideal gas with negligible error, often less than 1%.
- K-Omega turbulence model in Fluent database was selected. See Appendix 2 for k- Omega model description, mesh set up and specific CFD set ups.
- Tube wall roughness constant; 0.5. -Pipe wall material selected copper.
- The heat rejection surface area was investigated using various heat flux rates. With the ultimate aim of using the lowest possible heat flux rejection rate. Simplifying heat ejection surface and lowering the forced convection air. Heat flux figures investigated 40 to 150 W/m^2(0.003522 Btu/ft^2 to 0.013208 Btu/ft^2).
- Tube entry air temperature or room temperature in this case was assumed to be 24 °C (75.2 °F). Refer to Fig 1.
- Mesh details, software automatic meshing tool was used. Number of elements was set at 10 million cells for single stage, and 30 million cells for three compression stages, a full size model. Model was then reduced to a 36 degrees revolved volume rather than the 360 degrees revolved tube volume. The reduction in tube volume reduces the amount of computing time. See Appendix 2 for specific mesh details.-Initial analysis assumed dry air as per the software database. Final analysis was based on an assumption of 24 °C (75.2 °F), 60% relative humidity at the tube inlet. Simulating a near true room condition. See Fig 7 for details.
- A taper angle of 2 degrees inwards and towards the outlet was introduced. The tapering was allowed for the purpose of modeling to minimize the fluid reverse flow at the outlet assisting the CFD analysis, see Fig 1.

IV. Method

The first step involved building a model with a single stage compression cycle. Geometry was kept simple, keeping manufacturing requirements simple and thus lowering costs see Fig 1. A mass flow rate of 0.1 kg/s (0.22 lb/s) was applied at the inlet, air was then forced to pass through a narrow passage, and then allowed to expand at the outlet. The compression stage downstream of the inlet is as shown in Fig 1 generates heat.

The generated heat as a result of air compression is rejected by the tube wall. Rejected heat at the tube wall is

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removed by forced air convection. This has been simulated in the CFD boundary wall condition as heat flux. Assigned values of heat flux are as shown in tables 1 to 3. This phase of rejected heat removal is similar to what happens in a vapor compression cycle, where the condenser allows the compressed gas refrigerant to reject heat.

The first stage compression cycle can be described by the following relationship, $p1v1/T1 = p2v2/T2$, known as the perfect gas equation or the ideal gas equation. It is clear that if v1 and v2 are constant, T2 will increase as p increases. The opposite is also true, that is, starting from a high air pressure and then expands as in this example into a low pressure space, the temperature then drops.

The analysis was run for a number of increased inlet air flow rates. Results were tabulated as shown in table 1. This was to investigate the physics involved and the optimum low pressure that may achieve the lowest temperature drop at the outlet. The analysis has revealed that a relatively high temperature drop can actually be achieved at low pressure differentials, as shown in graph Fig 9. The methodology of this paper was therefore focused on analyzing the model by applying low pressure at the tube inlet. This analysis was developed to investigate a; two, three, and four stages of air compression/expansion models. Temperature drop from 24 to 20 °C (68 to 75.2 °F) was achievable with low relatively low pressure applied at tube inlet. The reason for investigating a number of compression/expansion stages was to explore how the SHR criteria aimed for in this paper compares with acceptable SHR values. The results will be discussed in section 5.

The two, three, and four stage model geometry was created by copying the initial single stage compression model and rejoining in the software modeler. Figure 5 shows the three stage model. The single stage model was meshed with 10 million cells, while the three stage compression model was meshed with 30 million cells. The aim was to create an extreme fine mesh and obtain good reliable results. Inflated mesh layers were created along the tube boundary wall, using the software standard settings for inflated mesh layers.

The four stage compression model was generated to cater for the added 60% relative humidity. This model represents near true operating conditions. See Fig 6 for details on what the 60% relative humidity represents in terms of water mass per a 1 kg (2.2 lb) of air mass.

V. CFD Results

Graphical simulations were generated as shown in Figs 1 to 7. Table 1 shows the data entry columns produced by the software package and the result columns. Appendix 1 shows calculations used to produce the results shown in tables 1 to 4. Graphs were plotted relating temperature output against the tube inlet pressure.

Results from table 1were plotted as shown in Fig 9(a). For better equation development, the curve was split into two parts as shown in Figs 9(b) & 9(c). Curve fittings were used to generate equations.

Notes: Curve approximation-fitting techniques were

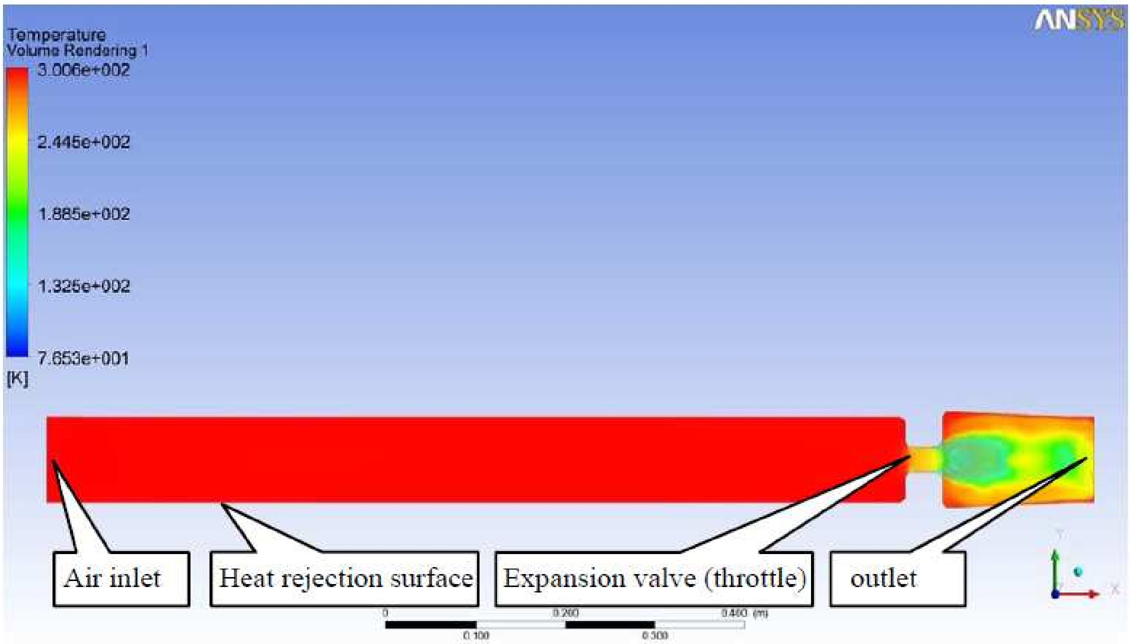

Fig.1. Single stage graphical results. The left segment of tube represents the heat rejection part. The wall segment was assigned with a heat flux value at the software boundary wall conditions. Model dimensions; - Left segment the heat rejection part; Diameter = 100 mm (4 inches), length = 950 mm (38 inches). - Chamfers near the expansion valve = 3mm (0.12 inch). - Expansion valve; Diameter = 30 mm (1.2 inch), length = 25 mm (1 inch). - Right tube segment, outlet; Diameter at the start = 100 mm (4 inch), length = 150 mm (6 inches). Tapered inwards towards the outlet by 2 degrees.

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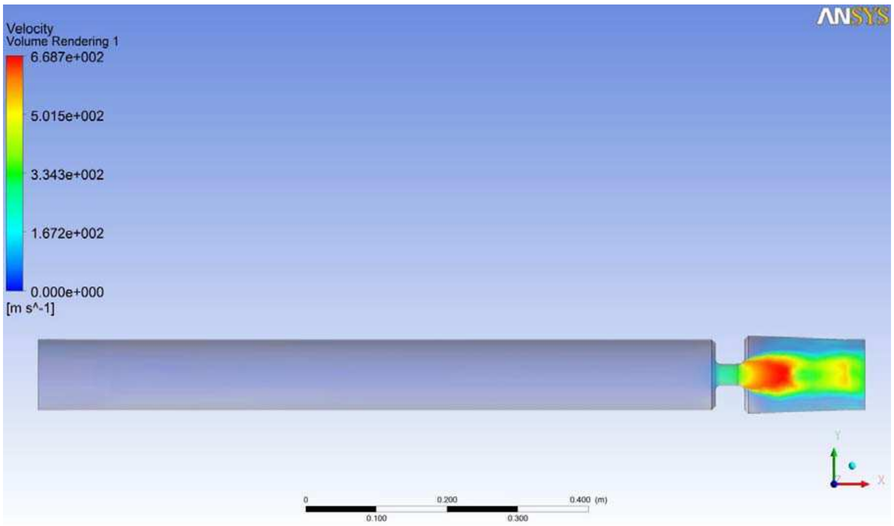

Fig.2. Single stage. Velocity volume rendering, where the left segment shows a relative low velocity, which the expansion valve (smaller diameter segment), and the tube outlet show a higher velocity.

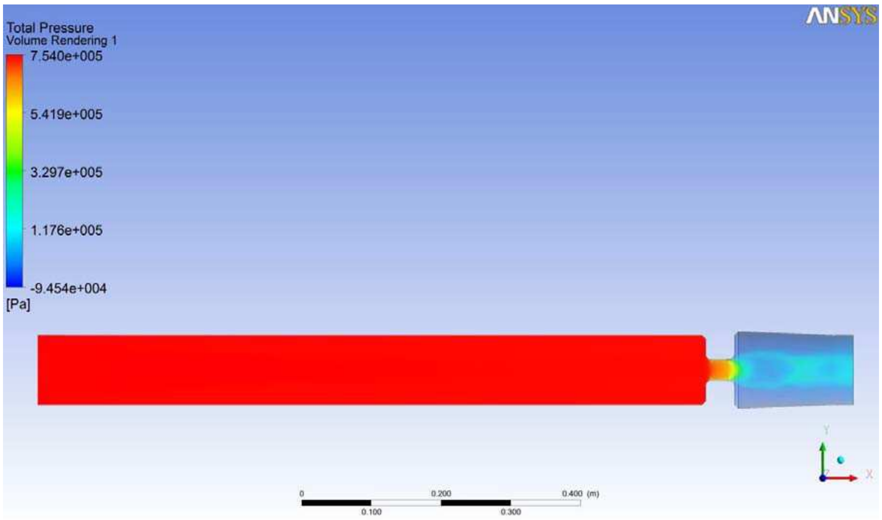

Fig.3. Single stage. Total pressure volume rendering, left segment where air enters the tube, and gets compressed shows a higher pressure when compared with the outlet relatively lower pressure.

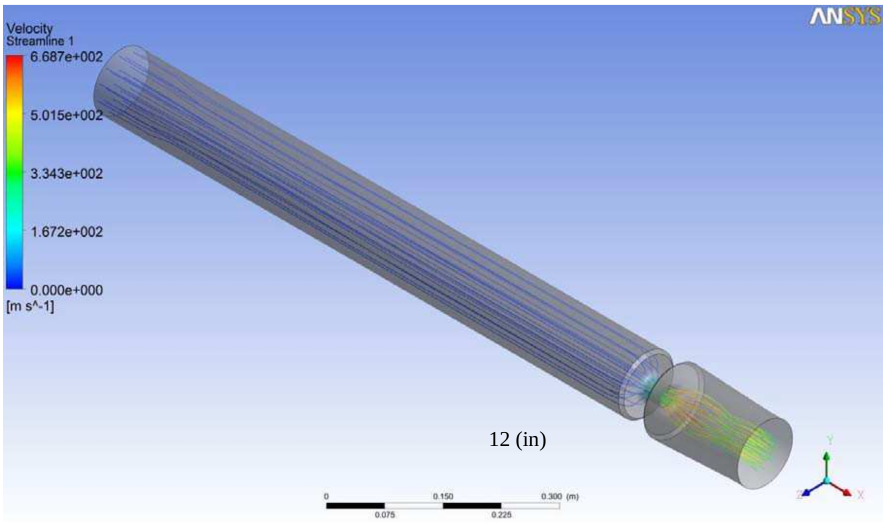

Fig.4.Singlestage.Picturesfromthesoftwarepostprocessor.Velocitystreamlinesattheleft segment of tube are relatively low, and higher at the outlet to the right of picture.

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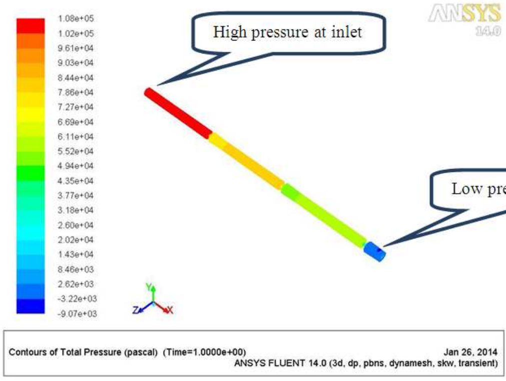


Fig.5. Picture shows a three stage open cycle model. The overall length now is 3000 mm (120inches). The above model represents detail Fig 8b, with the exception of the air compressor and the drain valves.

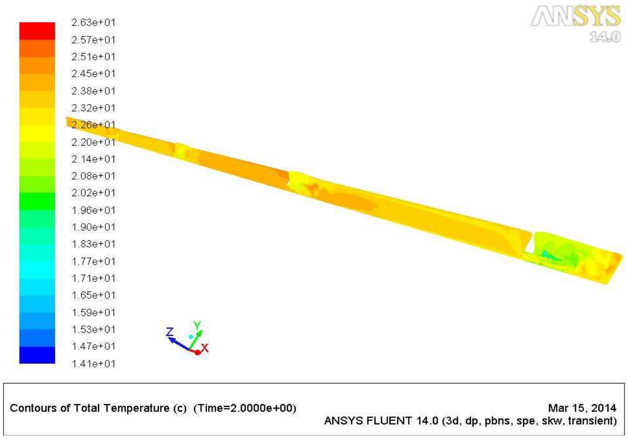


Fig.6. Picture shows a 3D image of a four compression stage open cycle model. With 60 % relative humidity at 24 °C (80 °F). Temperature contour plot shown for a 36 degrees segment was modeled to reduce computing time. Tube outlet shown to the right of picture.

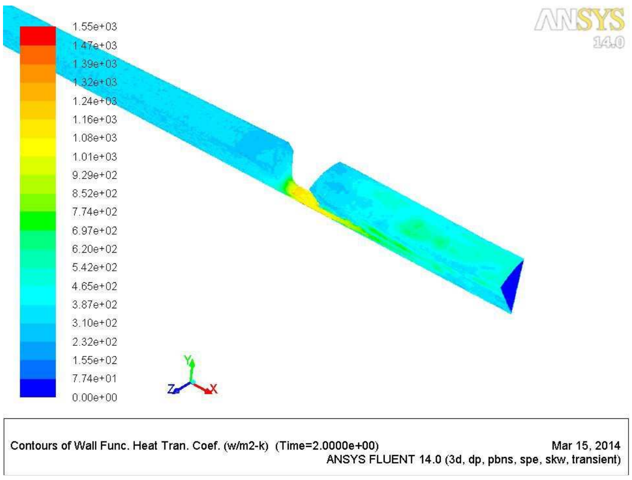
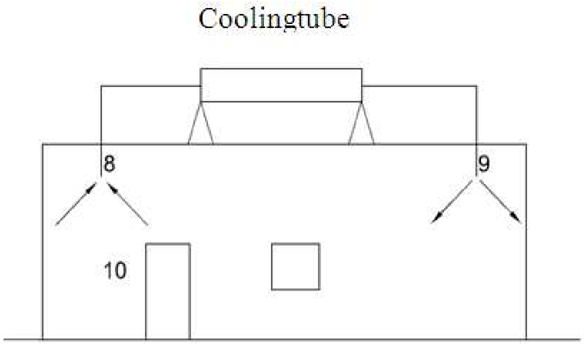
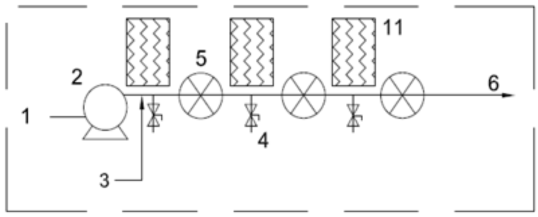


Fig.7. Picture shows a 3D image of a zoomed-in image near the tube outlet section of a four compression stage open cycle model. With 60 % relative humidity at 24 °C (80 °F). Contours of wall function heat transfer coefficient.



8(a)



8(b)

Fig.8. Schematic showing how an open cycle air cooling system works. Room air enters the return air grill 8, moves along the air duct 1, enters an oil free air compressor 2, enters the first compression stage at point 3, heat is rejected along the tube surface, heat removed by forced air convection 11, and then air expands once it leaves the expansion valve 5. The expansion valves downstream of expansion valve 5 are a repeated heat rejection cycles. In this example, three compression stages are shown. At the last expansion stage, air retunes to room via tube or air duct 6 towards. Air discharges at room space via air diffuser 6.

Humidity expected to condensate during compression; therefore, drain valve item 4 was introduced to allow for water drainage. Drain valve control can be manual or automated.

The above example shows three compression stages in a straight line, this layout can be in any configuration, example inverted U shaped tubes in series. Detail 8(a) shown above building`s roof, this can be installed vertically along a wall, or even within the building with the exhaust heat removing air is ducted. Detail 8(b) shows details of item 8(a).

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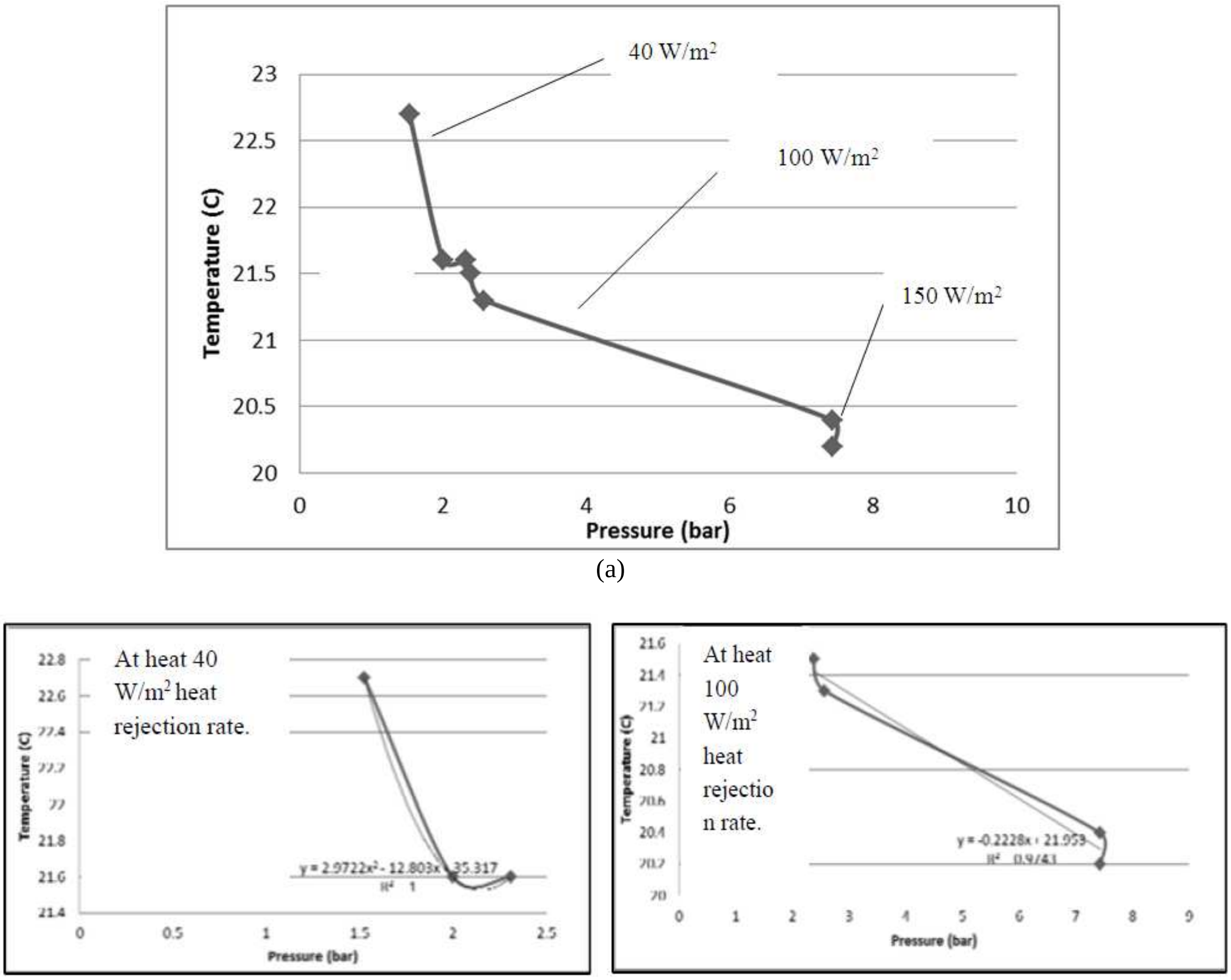

(a)

(b) (c)

Fig.9. (a) Data plotted from table 1. Representing a single stage compression cycle, with various heat flux ratings assumed at the heat rejection segment of tube. Aim of this graph was to establish the physics. The graph show the most significant pressure drop obtained with a low inlet pressure applied, below 2 bar. Curve (a) was split into two curves for curve fitting, curves (b) and (c). Equations were generated by the spread sheet as shown above, and translated to reflect the actual physical parameters as shown in section V.

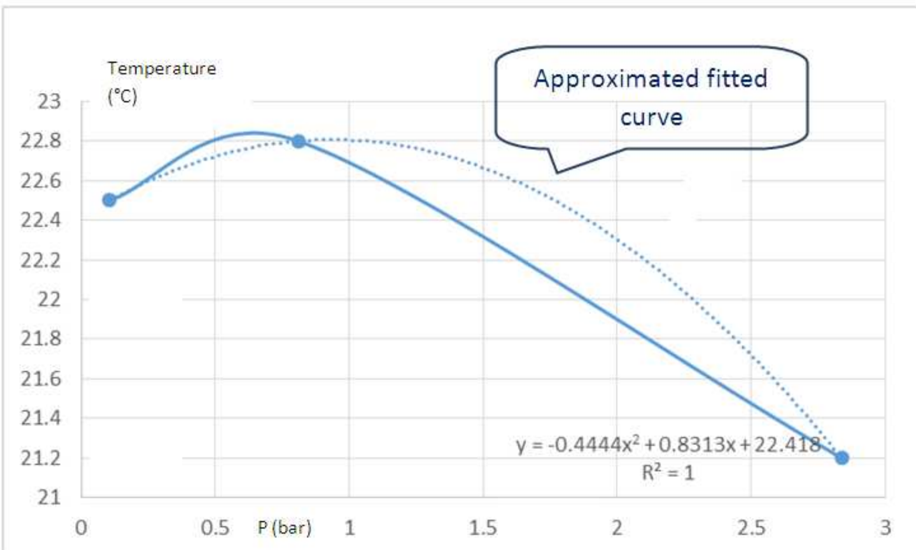

Fig.10. Graph representing a two stage compression open air cycle. Graph indicates lower temperatures achieved at low inlet pressure. Equations were developed as shown in section V.

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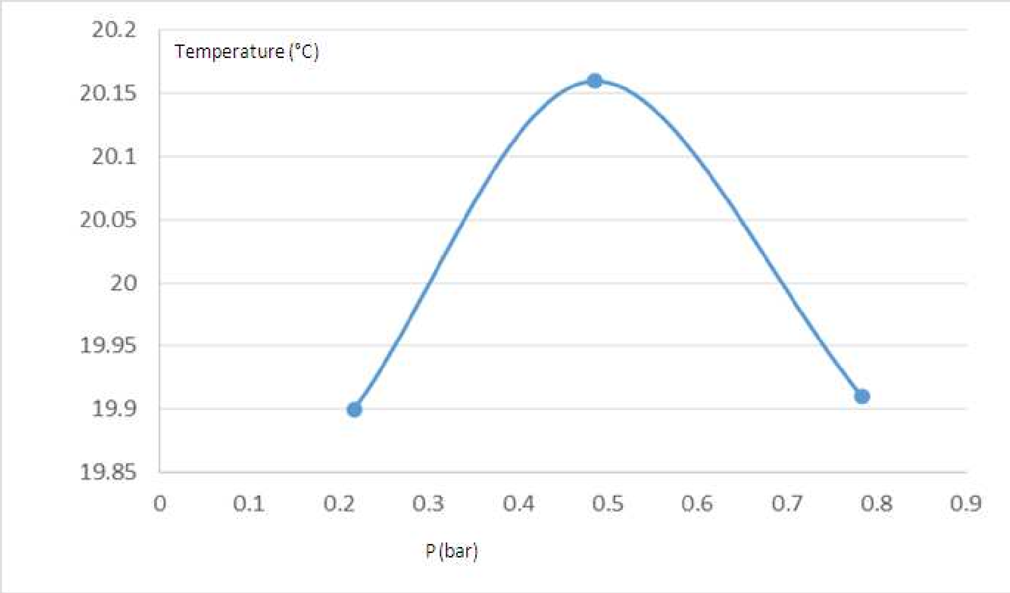

Fig.11. Graph representing a four stage compression expansion open air cycle. Graph indicates lower temperature achieved at the lowest inlet pressure. Equations were developed as shown in section V. Inlet air condition simulated at 24 °C (80 °F) 60% relative humidity.

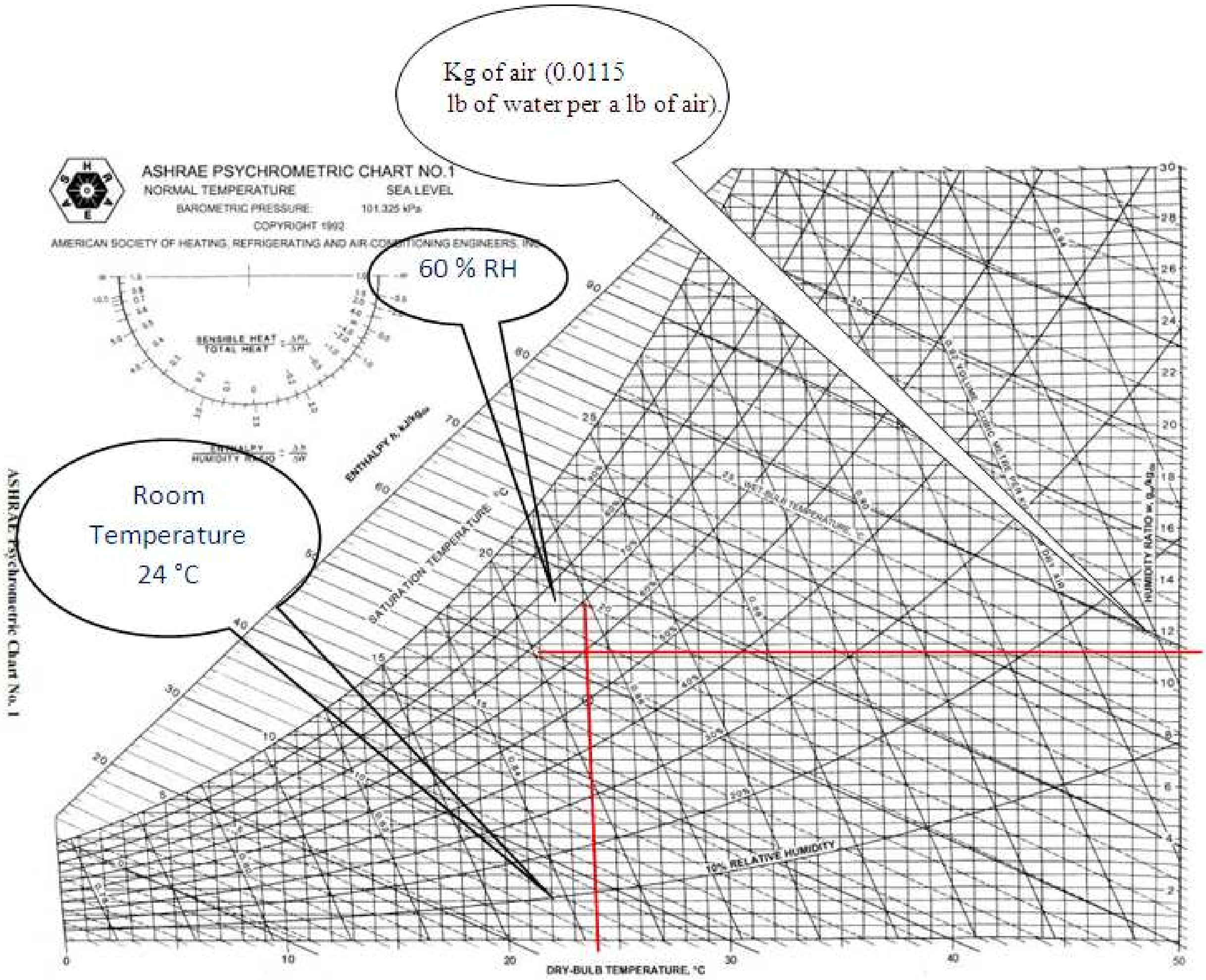

Fig.12. Psychometric chart showing an assumption of room conditions. Properties of air considered; 11.5g of water per a 1 kg of air.

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Table1: CFDData Output and Calculated Results–one stage compression.

Description	Single compression stage					
Air flow mass, kg/s (lb/s)	0.4 (0.88)	0.5 (1.1)	0.51 (1.12)	0.52 (1.144)	0.55 (1.21)	1.3 (2.86)
Pressure in (bar)	1.53	2	2.31	2.37	2.57	7.43
Pressure out (bar)	0.049	0.121	0.187	0.190	0.205	0.403
ΔP (bar)	1.481	1.879	2.123	2.180	2.365	7.0267
Temperature out, °C (°F)	22.7 (77.4)	21.6 (75.2)	21.6 (75.2)	21.5 (75)	21.3 (74.6)	20.2 (72.4)
Air flow, l/s (cfm)	400.2 (848)	500.3 (1060)	510.3 (1081.2)	520.3 (1102.4)	550.3 (1166)	1300.7 (2756)
Heat flux W/m2 (Btu/ft2)	40 (0.003522)	40 (0.003522)	40 (0.003522)	100 (0.008805)	100 (0.008805)	150 (0.013208)
Enthalpy in, J/kg, (Btu/lb)	1722.7 (0.740)	1722.7 (0.740)	1722.6 (0.740)	1722.8 (0.740)	1722.9 (0.740)	1722.3 (0.740)
Enthalpy out, J/kg, (Btu/lb)	-16268.6 (-6.994)	-16268.6 (-6.994)	-16446.9 (-7.071)	-16695.9 (-7.178)	-17598 (-7.566)	-26170 (-11.251)
Δ Enthalpy, J/kg, Btu/lb)	17991.3 (7.735)	17991.3 (7.735)	18169.5 (7.811)	18418.7 (7.918)	19320.9 (8.306)	27892.3 (11.991)
Ratio of cfmto TR	160.25	200.32	202.32	203.50	205.19	335.95
SHR	0.09	0.17	0.16	0.17	0.17	0.16
Air flow mass, kg/s (lb/s)	0.1 (0.22)	0.2 (0.44)	0.5 (1.1)	0.1 (1.1)	0.2 (0.44)	0.3 (0.66)
Pressurein (bar)		0.1		0.81		2.84
	2.37		2.57		7.43	
Pressure out (bar)	0	0.121	0.187	0.190	0.205	0.403
ΔP (bar)	0.006	0.018	0.066	2.180	2.365	7.027
Temperature out, C (°F)	22.5 (77)	22.8 (77.6)	21.2 (74.4)	20.4 (72.8)	22 (76)	22 (76)

The calculated SHR values show that acceptable sensible cooling can be obtained when using multi stage compression. *= Compression power was estimated to be 1.223 kW, calculated in section a. Allow 20% extra power friction losses gives 1.47 kW. See Appendix 1 & 2 for calculations on SHR & compressor power.

Table 2: CFD Data Output and Calculated Results – two & three stages compression.

Description	Two compression stages			Three compression stages		
Air flow mass, kg/s (lb/s)	0.1 (0.22)	0.2 (0.44)	0.5 (1.1)	0.1 (1.1)	0.2 (0.44)	0.3 (0.66)
Pressure in (bar)	0.1	0.81	2.84	2.37	2.57	7.43
Pressure out (bar)	0	0.121	0.187	0.190	0.205	0.403
ΔP (bar)	0.006	0.018	0.066	2.180	2.365	7.027
Temperature out, ℃ (°F)	22.5 (77)	22.8 (77.6)	21.2 (74.4)	20.4 (72.8)	22 (76)	22 (76)
Air flow, l/s (cfm)	100 (212)	200 (424)	500 (1060)	100 (212)	200 (424)	300 (636)
Heat flux, W/m2 (Btu/ft2)	40 (0.003522)	40 (0.003522)	40 (0.003522)	100 (0.008805)	100 (0.008805)	150 (0.013208)
Enthalpy in, J/kg (Btu/lb)	-1054 (-0.453)	-1077.6 (-0.463)	-1106 (-0.475)	-1042.6 (-0.448)	-1060.8 (-0.456)	-1077.2 (-0.463)
Enthalpy out, J/kg (Btu/lb)	-3066.6 (-1.318)	-3904.5 (-1.678)	-8842.3 (-3.801)	-4855.4 (-2.087)	-4192.7 (-1.802)	-3739.6 (-1.607)
Δ Enthalpy, J/kg (Btu/lb)	2012.6 (0.865)	2827 (1.215)	7736.3 (3.326)	3812.8 (1.639)	3131.9 (1.346)	2662.4 (1.144)

Ratio of cfm to TR	358.14	509.94	465.86	189.05	460.26	812.20
SHR	0.75 *	0.426	0.364	0.95	0.64	0.75

The calculated SHR values show that acceptable sensible cooling can be obtained when using multi stage compression. *= Compression power was estimated to be 1.223 kW, calculated in section a. Allow 20%extra power friction losses gives 1.47 kW.See Appendix 1 & 2 for calculations on SHR & compressor power.

Table 3: CFD Data Output and Calculated Results – four stages compression. With 60% relative humidity added.

Description	Four compression stages		
Air flow mass, kg/s	0.1	0.15	0.2
Pressure in (bar)	0.217	0.485	0.784
Pressure out (bar)	0.002	0.003	0.003
ΔP (bar)	0.215	0.482	0.778
Temperature out, °C (°F)	19.9	20.16	19.91
Air flow, l/s (cfm)	100 (212)	150 (318)	200 (424)
Heat flux, W/m2 (Btu/ft2)	40 (0.003522)	40 (0.003522)	40 (0.003522)
Enthalpy in, J/kg (Btu/lb)	-1056.1 (-0.454)	-1077.6 (-0.463)	-1090.6 (-0.468)
Enthalpy out, J/kg Btu/lb)	-5425.46 (-2.332)	-5272 (-2.266)	-2142.88 (-0.921)
Δ Enthalpy, J/kg (Btu/lb)	4369.35 (1.878)	4195.00 (1.803)	1052.24 (0.452)
Ratio of cfm to TR	164.97	257.73	1370.02
SHR	0.94	0.92	3.9

Results and calculations for a four stage compression. High SHR values were produced, with air temperature output at 20 °C (72 °F) and below.

1- Employed using Excel 2010 software. R^2, or regression description. Assessing integrity of formulas developed by regression. With a low $R^2 = 0$ none of the variances on the Y axis can be explained against values on shown on the X axis. With high $R^2=1$, all of the variances on the Y axis can be explained against values shown on the X axis.

2- Assumptions were made on forced air convection at the heat rejection section of the CFD model, indicated in Fig 8. Table 1 shows the assumed forced air convection figure at every pressure input level. As pressure at the inlet was increased, the forced air convection was increased to cater for the increased heat flux. Since this is an assumption actual heat rejection geometry and possible use of fins may change. The aim in this investigation is to demonstrate that the system can work.

The following equations were developed based on the curves showing Figs 9, 10, and 11. Curves can be used to predict the air temperature out-put in response to the input pressure.

$R^2 = 1$. Equation derived from Fig 10.

• Temperature output against applied inlet pressure – equation – four stage compression with 60% relative humidity 24 °C room temperature.

$$AT\ (°C) = -3.1857(Pin)^2 + 3.2065(Pin) + 19.354 \qquad (i)$$

$R^2 = 1$. Equation derived from Fig 11.

• Temperature output against applied inlet pressure – equation – single stage compression.

$$AT\ (°C) = 2.9722(Pin)^2 - 12.803(Pin) + 35.317 \qquad (ii)$$

$R^2 = 1$ for the condition 1.5< Pin > 2.3 bar Equation derived from Fig 9 (b).

$$AT(°C)=-0.2228(Pin) + 21.953 \qquad (iii)$$

$R^2 = 0.9743$ for the condition 2.6< Pin > 7.4 bar Equation derived from Fig 9 (c).

• Temperature output against applied inlet pressure – equation – two stage compression.

$$AT\ (°C) = -0.4444(Pin)^2 + 0.8313(Pin) + 22.418 \qquad (iv)$$

VI. Discussions

The aim of this paper as stated in the introduction section was to investigate the practicality of such a system, with sustainability in mind. The CFD analysis have demonstrated that it is possible to create an open cycle cooling system to produce an air temperature down to 20 °C (68 °F), starting from a room temperature of 24 °C (75.2 °F). Achieved with a single stage compression/expansion cycle. With a heat removal rate of over 27 kW, per a 1.3 kg/s (2.86 lb/s) of air mass. The pressure at the tube inlet was at 7.5 bars as shown in table 1. However the SHR values shown in table 1 indicated a value of SHR = 0.15, which is considered as a low value. Research with CFD modeling was continued to investigate ways to achieve a value of SHR = 0.7.

According to Faye McQuiston (2000) [5], electric vapor compression systems produce 165.2 l/s to 188.8 l/s (350 to

400 cfm) per a 3.4 kW (1 TR) at an SHR value of 0.7.

Two stage, three, and four stage compression cycles were modeled as shown in Figs 1 to 7. Figure 5 shows a three stage compression cycles, and Fig 6 shows a four stage compression cycles with relative humidity set at 60%. The analysis have shown that the model can be developed to approach a value of SHR = 0.7 and even higher with multistage compression. Also, the multi stage compression results showed that such SHR values can be obtained with relatively lower inlet air pressure. Therefore, lower energy consumption using smaller air compression equipment. Table 2 shows at what mass air flow rates a value of SHR 0.7 can be achieved, and how this value compares with what is given by Faye McQuiston [5] (2000); 165.2 l/s to 188.8 l/s (350 to 400 cfm) per a 3.4 kW (1 TR) at an SHR value of 0.7. Table 3 shows how near realistic results can be achieved with acceptable SHR values at 60 % relative humidity.

The advantages of using such an open cycle system can be summarized as follows;

a- Tube geometry is easy to manufacture, with relatively lower consumption of manufacturing materials when compared with an electric vapor compression system.

b- Using air as a refrigerant media, avoids the manufacturing of refrigerant gases. Currently available refrigerant gases including gases which are regarded environmentally friendly gases do require energy to; manufacture, store, distribute, and maintenance by refilling. This system uses available space air. Low energy consumption during the compression stage. Table 2 and 3 show what cooling power can be achieved with low compression rates and achieve SHR values of 0.7produced between 0.2 and items;a, b, and c, above.

Comparing the open air cycle cooling with electric air cooled vapor compression can be done by referring to table 4, and as follows;

The ratios of cooling power to electric power are 2.31 and 2.42 for air open cycle and electric air cooled vapor compression respectively. A difference of 4.7 % in favor of the latter. -Comparing the carbon foot print between the above mentioned systems. The carbon footprints per a 1 kW of cooling power are 0.19 and 0.13 for air open cycle and electric air cooled vapor compression respectively. A difference of 31.6 % in favor of the latter. However, considering the facts that refrigerant gases consume energy during their production phase, need storage facilities, consume energy during transportation from storage to point of usage, and require resources to refill systems with refrigerant gas. The balance of preference could be tilted in favor of the open air cycle system. The risks associated with escaping refrigerant gases are eliminated when using an open air cycle.

The methodology employed of having a series of compression and expansion cycles, simplifies the currently available systems discussed in section II, paper by C. M. Bartolini (1992) [1]. Figure 13 appendix 3, shows description sketches of what is described in the referred to paper. The differences between what is described in this paper and what is described in C. M. Bartolini [1] paper are; the removal for the need of the turbine component, and the differences in the number of heat exchangers. See Appendix 3. The TS diagram in Fig 13 also shows as an example how a two stage model can be represented.

A paper by L. Beatrice and F. Fiorelli [8] investigates the feasibility of using such systems in automotive applications. Where a system is powered by the engine exhaust. Paper concludes that such system when compared with conventional systems; consumes 1.7 kW (similar to conventional systems), weighs1.0 kg more, cost 56 % more, and requires 24 % less maintenance costs. With the advantage of not using refrigerant gasses. The costs may come down if mass production starts. Therefor, the 56 % cost difference can be reduced.

VII. Conclusion

CFD packages can be used to further develop HVAC systems, improving efficiency, reducing manufacturing costs, and carbon footprint. Open air cycle cooling systems simplify the production of HVAC cooling systems. A simple tube was used with a relatively small air compressor to produce an acceptable SHR value of 0.7.
Open cycle air cooling can be used in other applications such as automotive air-conditioning, replacing manufactured refrigerant gasses which can harm the environment, and consume energy to produce.

Recommendations

1. Experimental work to be conducted.

Acknowledgments

The author would like to thank -Dr. Sim for checking CFD model and providing access to the ANSYS Fluent software. University of Qatar.

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International Journal of Engineering Innovation & Research
Volume 3, Issue 6, ISSN: 2277 – 5668

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NOTATION

CFD = Computational Fluid Dynamics. ANSYS CFX = CFD software brand. m/s or m ^s-1 = meters per a second. m^3/s = cubic meters per a second. M^2 = square meter. m = meters. mm = millimeter. Pa = Pascal W = Watts TR = Ton of Refrigeration. W/m2 = Watts per a square meter. Btu/ft^2 = British Thermal Units per a square foot. Pin = Pressure at the tube inlet, (bar). AT = Air temperature at the tube outlet, (°C). HP = horsepower N = number of compression stages 1 stage compressor. k = 1.41 = adiabatic expansion coefficient. P1 = Pin = absolute initial atmospheric pressure (psi) (14.1 psi at sea level). P2 = Pout = absolute final pressure after compression (psi). V = volume of air at atmospheric pressure (cfm). c_p= specific heat capacity of air. γ= heat capacity ratio for ideal gas. COP = Coefficient of Performance.

APPENDIX I

Calculations associated with results in tables 1, 2, & 3;

Data produced in table 1 was generated by the CFD package, with the exception of the following information; - The value of air flow in cfm was generated by multiplying 1 l/s by 2.119 cfm. Assuming 1 kg/s = 1 l/s of air flow. -Δp = pressure at inlet – pressure at outlet. –Δ Total Energy = Total Energy at inlet – Total Energy at outlet. –Δ Enthalpy = Enthalpy at inlet – Enthalpy at outlet. In this example this equals the change in the internal energy of the system, plus the
work that the system has done on its surroundings. -Ratio of cfm to 1

TR, divide cfm by (Δ Enthalpy ÷ 3400). Note the 3400 convert energy in Joules to 1 TR of cooling power. -SHR = the ratio of sensible heat transfer to the total heat transfer of the process. Expressed as follows; SHR = 1005 × [(outlet temperature – 24 C) ÷ (Δ Enthalpy)]. Where 1005 is the specific heat capacity of air.

Power required for air compression

Figures from tables 2 as*an example* can be entered in equation (1) to calculate compressor power.

N =assume1stagecompressor.

k =1.41.

$$HP = \frac{144\,NP1Vk}{33e3\,(k-1)} \cdot \left[\left(\frac{P2}{P1}\right)^{\frac{k-1}{Nk}} - 1\right] \qquad \text{-- -equation (1)}$$

P1=14.1psi at sea level.

P2= 15.95 psi (14.5 psi+ 1.45 psi) for air flow 0.1kg/s (0.22 lb/s)in a two stage compression flow, shown in table 2.

V= 212 cfm, from table2.

HP = 1.64 horse power or1.223kW

Assuming adiabatic compression (or expansion) takes place without transmission of heat. It is clear from equation (1) that as compressor output pressure P2 increases compressor power requirements increases, and therefore more overall pollution emissions. Aim is to work with minimum pressure levels.

APPENDIX II

Specific CFD software setup is shown in this section. Other setups not described here shall be as per software default settings.

Turbulence Model used

A brief description of k-omega or k-ω turbulence model is given by Professor Brian Spalding (2014)[6] is shown below;

The first two-equation turbulence model was the KE-OMEGA model of Kolmogorov [1942]. This model, which is also known as the KE-F and k-w model, involves the solution of transport equations for the turbulent kinetic energy KE and the turbulence frequency F. It should be mentioned that other workers define F as the ratio EP to KE, where EP is the dissipation rate of KE.

Several different and improved versions of Kolmogrov's model have been proposed, including those of: Saiy [1974], Spalding [1979], Wilcox [1988], Speziale et al [1990] and Menter [1992].

The KE-F model of Wilcox [1988] has been selected for use in PHOENICS, mainly because it is the most extensively tested and it includes a low-Reynolds-number extension for near-wall turbulence.

Although the KE-F model is not as popular as the KE-EP model, it does have several advantages, namely that:

a. the model is reported to perform better in transitional flows and in flows with adverse pressure gradients;

b. the model is numerically very stable, especially the low-Re version, as it tends to produce converged solutions more rapidly than the k-epsilon turbulence models; and

c. the low-Re version is more economical and elegant than the low-Re k-epsilon models, in that it does not require the calculation of wall distances, additional source terms and/or damping functions based on the friction velocity.

The main weakness of the KE-F model is that unlike the KE-EP model, it is sensitive to the free-stream boundary condition for F in free-shear flows. Modified variants exist which claim to remove this sensitivity, but Wilcox [1988] notes that it is a desirable feature for transitional applications.

Equations and description shown below, also given in Wikipedia K-Omega [7] turbulence model (2014) section;

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho u_j k)}{\partial x_j} - P - \beta^* \rho \omega k + \frac{\partial}{\partial x_j}\left[\left(\mu + \sigma_k \frac{\rho k}{\omega}\right)\frac{\partial k}{\partial x_j}\right]$$

$$\frac{\partial(\rho\omega)}{\partial t} + \frac{\partial(\rho u_j \omega)}{\partial x_j} = \frac{\gamma\omega}{k}P - \beta\rho\omega^2 + \frac{\partial}{\partial x_j}\left[\left(\mu + \sigma_\omega \frac{\rho k}{\omega}\right)\frac{\partial \omega}{\partial x_j}\right] + \frac{\rho\sigma_d}{\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j}$$

K-omega (k-ω) turbulence model is a common two-equation turbulence model in computational fluid dynamics that is used as a closure equation of the Reynolds-averaged Navier–Stokes equations. The model attempts to predict turbulence by two partial differential equations with the first variable being the turbulence kinetic energy while the second being the specific rate of dissipation.

The first transported variable is turbulent kinetic energy, k. The second transported variable in this case is the specific dissipation, ω. It is the variable that determines the scale of the turbulence, whereas the first variable, k, determines the energy in the turbulence.

Specific model settings were based on k-omega for turbulence models and inlet and outlet conditions. This selection was based on the description given above, and was found to perform exceptionally well.

A transient analysis was run, with 10 time steps, at 0.1 second per a time step. Each step consists of 20 iterations.

Mesh statistics

Meshing was created taking acre of boundary conditions, with inflation layers applied as per software default settings. Minimum element size = default 1.7106e-4 m, max face size = 1e-3 m, and maximum size = default 3.421e-2 m.

Mesh Metrics; Minimum aspect ratio = 1.1575, maximum = 10.24, average = 1.865, and standard deviation = 0.470.

Processes decryption of Fig 13; 1-2 adiabatic (compression), 2-3 isobaric (heat addition), 3-4 adiabatic (expansion), & 4-1 isobaric (heat rejection). A reversed Brayton cycle, known as the Bell Coleman cycle. As discussed, the difference between current practices (where an expander is used) and this paper, where no expander is used, but working with a series of compression and expansions stages, where heat is being rejected at each stage.

Based on Fig13 T-s diagram and assuming air an ideal gas, the isentropic relationship equations are generated (per a unit mass);

Refrigeration effect = Cp(T1-T4)

Heat rejected = Cp(T2-T3)

Compressor work = $\frac{\gamma}{\gamma-1}(p2v2 - p1v1)$ = Cp(T2-T1)

Expander work = $= \frac{\gamma}{\gamma-1}(p3v3 - p4v4)$ = Cp(T3-T4)

According to C.P. Arora(2000)[9] discusses the above equations and shows a reduced equation of COP;

COP = $\frac{1}{\frac{T2-T3}{T1-T4} - 1}$

C.P. Arora notes (2000)[9] that such HVAC systems are currently only employed in aircraft. The rotational speed of compressor and expander turbines is in the order of 100,000 rpm. An advantage of an open air cycle over a closed cycle, is that no heat exchanger is required for the refrigeration process, therefore, a weight and cost saving. However, the disadvantage of an open air system when compared with a closed cycle, is that if the return air from the cooled space is humid, the expanded at the outlet air may develop fog or ice and possible clog the outlet line.

APPENDIX III

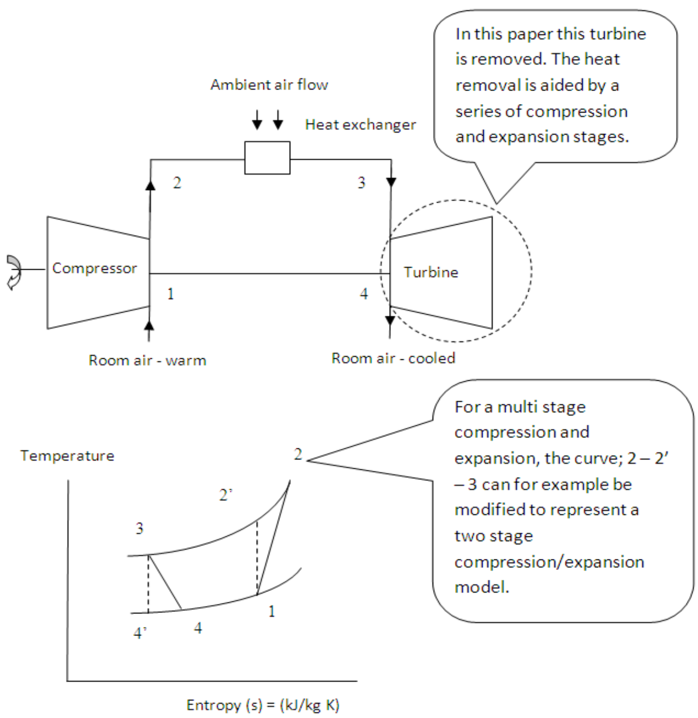

Fig13. Above sketch shows a typical layout for a simple open air cycle system as shown in a paper by C. M. Bartolini (1992) [1]. While the lower sketch is a typical thermodynamic air cycle T-s diagram representing the above mentioned

sketch.