REDUCING AIR COMPRESSOR WORK BY USING INLET AIR COOLING AND DEHUMIDIFICATION

A Thesis

by

MARK JAMES HARDY, JR.

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

December 2010

Major Subject: Mechanical Engineering

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Approved by:

Chair of Committee,	Michael Pate
Committee Members,	Timothy Jacobs
	Yassin Hassan
Head of Department,	Dennis O'Neil

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ABSTRACT

Reducing Air Compressor Work by Using Inlet Air Cooling and Dehumidification.

(December 2010)

Mark James Hardy, Jr., B.S., University of Louisiana at Lafayette

Chair of Advisory Committee: Dr. Michael Pate

Air compressor systems play a large role in modern industry. These compressors can account for a significant portion of a manufacturing facility's electric consumption and any increase in efficiency can lead to economic benefits. Air compressors are sensitive to ambient conditions, as evidenced by the fact that compressing cooler and drier air decreases the amount of work required to compress the air.

A thermodynamic model of an air compressor system was developed and several cases were run by using both vapor compression and absorption cycle chillers to cool and dehumidify the inlet air. The results show that the performance increases as much as 8% for the compressor system with absorption inlet cooling and as much as 5% when using vapor compression inlet cooling. Climates with higher humidity and temperatures can see the most benefits from inlet air cooling and dehumidification.

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I would also like to thank Florent Hardy, Jr. and Mr. Jim Nelson for encouraging me to pursue my graduate studies here at Texas A&M University. Finally, thanks to my family and friends for their support and encouragement.

NOMENCLATURE

q	=	Specific heat transfer, Btu/lb_m
T	=	Temperature, F or R
P	=	Pressure, psia or psig
c_p	=	Specific heat of air at constant pressure, $Btu/(lb_m R)$
c_v	=	Specific heat of air at constant specific volume, $Btu/(lb_m R)$
k	=	Specific heat ratio, unitless
Φ	=	Relative humidity, %
ω	=	Humidity ratio, lb_{vapor}/lb_{air}
p_v	=	Partial pressure of vapor, psia
p_g	=	Saturation pressure, psia
w_C	=	Specific compressor work, Btu/lb_m
w_{AC}	=	Specific work of the refrigeration system, Btu/lb_m
h	=	Specific enthalpy, Btu/lb_m
h_{fg}	=	Heat of vaporization, Btu/lb_m
η_r	=	Refrigerator efficiency, unitless

*Subscripts refer to the state which the value was taken at

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1. INTRODUCTION

1.1 Introduction

The air compressor is an important and heavily utilized piece of machinery in today's world. Air compressors consume a large percent of the energy demand in all manufacturing facilities; therefore improving compressor efficiency can lead to cost savings by lowering the required energy input to achieve the desired final air pressure. The work required to compress a fluid is proportional to its specific volume. Since air acts as an ideal gas and atmospheric pressures are relatively constant, the work required to compress air is proportional to the air temperature. Thus, compressor work inputs are sensitive to the intake air temperature. For air being compressed, multiple compression stages are used. Cooling the air between these stages will decrease compressor work input. When intercooling is involved, humidity can also affect work input by limiting the temperature of which the air is cooled; the intercooling temperature is limited by the condensation temperature of water vapor at the higher pressures. Condensation in the compressor or compressed air storage and distribution network can result in system damage from corrosion, causing leaks, pressure losses, and ultimately failure of components. Drying of the air is necessary to reduce the risk of condensation damage when the air is cooled or stored at temperatures below the dew point. The work input of air compressors is thus sensitive to the inlet temperatures and humidity.

This thesis follows the style of Energy Policy.

At higher pressures, air has a decreased capacity for water vapor, and as a result, when compressed air is cooled, significant amounts of water can condense. For air at 75°F and 75% relative humidity, about 3.5-4 gallons of water per hour condenses out of a 500cfm compressor (Foszcz, 1997). As noted previously, liquid water can cause significant damage when present in the air compressor, air storage and delivery network, and in tools and equipment. Corrosion of pipes and tanks will be accelerated when liquid water is present and can lead to leaks and ultimately costly failures. Water can also wash out lubrication in seals and tools. Drying of air is therefore an important task in prolonging equipment life in a compressed air system.

There are several ways of removing moisture from the air before it enters the compressor. Because it is often the simplest and most economic method, the most common way to dry air is to use a refrigeration system to condense humidity out of the air. Other methods include physically removing water by adsorption and absorption processes such as desiccants and membranes. Since the refrigeration method serves the dual purpose of dehumidifying and pre-cooling, it will be the only method investigated in this analysis.

A simple way to reduce compressor work is to use cooler and dryer air by conditioning the air with a chiller before it is compressed. Cooler air is denser and allows the compressor to compress a greater mass of air per stroke while dehumidifying the air allows for lower intercooling temperatures in a multi-stage system. Cooling and drying the air can be achieved by a vapor-compression refrigeration cycle, Brayton refrigeration, or by an absorption refrigeration cycle where sufficient waste process heat is available. Regions with high temperatures and/or high relative humidity can see decreased compressor performance when compared to cooler regions and may benefit more from air inlet cooling.

Absorption cooling is a refrigeration method similar to vapor compression cooling, the difference being the way the refrigerant is compressed. In an absorption chiller, the source of input energy is waste heat rather than mechanical work. The most common absorption refrigeration cycles use an ammonia-water working fluid.

After leaving the evaporator section, the gaseous ammonia enters an absorber where it is mixed with cool liquid water. The ammonia reacts with water and is absorbed into an aqueous solution. The amount of ammonia that can be absorbed by water is inversely proportional to the temperature of the solution. The liquid solution then enters a pump where it is compressed. Since the fluid being compressed is liquid, the pump requires very little input work when compared to work required to compress a vapor. Heat is then transferred to the solution, causing the ammonia to boil out and separate from the water. The gaseous ammonia then passes through a rectifier where it is separated from the water before it enters the condenser. The water is then passed through an expansion valve before returning to the absorber.

1.2 Background

Inlet air cooling is a well known way to reduce compressor work in order to boost gas turbine performance in hotter regions(Alhazmy and Najjar, 2004; Najjar, 1996; Zaki et al., 2007). The power output of gas turbines can fall as low as 20% below their rated generation capacity (standard ISO power output at 59°F ambient) when temperatures are over 95°F. Further, warmer days result in increased electric power demand due to the need for household cooling. In order to maximize power output from gas turbines in these conditions, a variety of tactics are used to chill inlet air. Evaporative cooling (fogging) is most commonly used because of its low installation and operating costs. However, the wet bulb temperature limits the amount of possible evaporative cooling. Also, overspray of water into the turbine inlet can cause damage due to erosion, corrosion, and aerodynamic instabilities in compressor blades (Kalyanaraman, 2006). Absorption cooling driven by the turbine exhaust gases is another method that is commonly used. Absorption chillers are not sensitive to humidity and can cool inlet air to temperatures below dew point. Gas turbines using one or multistage absorption chillers have been shown to have power increases of up to 18% (when compared to an equivalent uncooled turbine) and are economically viable (De Lucia et al., 1995).

Although inlet air cooling is a well known way to improve the overall efficiency of gas turbines, there is little evidence in the literature of investigations of the effects of inlet air conditioning for improved performance of air compressor systems alone. This study investigates the work savings that can be achieved by chilling the inlet air of a two-stage compressor system for a range of operating ambient air conditions. A theoretical thermodynamic model is developed to determine the total work savings for a compressor system when an inlet air precooler is added.

1.3 Applications – Manufacturing and Industrial Facilities

Industrial facilities oftentimes refer to compressed air as a fourth utility alongside electricity, water, and gas. It is estimated that in the US alone, the total installed horsepower of air compressors excedes 17 million (Cengel et al., 2000). Oftentimes, these compressed air systems are the primary consumer of energy in a plant, requiring more than 30% of the total power consumption. Compressed air is used to power tools, move products, in drying applications, and for a variety of pneumatic control systems. Finding economic methods to reduce the cost of providing compressed air to machinery and processes can lower the operating costs of manufacturing facilities worldwide.

1.4 Applications – Compressed Air Energy Storage

As world energy consumption increases, there is a growing demand to meet these needs by stable and renewable sources. Some common renewable energy sources are intermittent and may not be available during times of high demand. Energy storage during times when production exceeds demand is an important issue to consider when using intermittent sources of generation such as wind and solar energy. Efficient storage is also beneficial to store cheaper power available at off-peak demand times for use during high demands. According to the Department of Energy, some areas such as west Texas, wind energy generation capacity has grown at a faster rate than the capacity to transmit the power to consumers.

Current proven ways to store power include pumped hydro storage, battery storage, thermal storage, and compressed air storage (Ibrahim et al., 2008). Pumped hydropower storage is economically proven and one of the most utilized methods of energy storage, but is limited by local topography and by political issues caused by the large surface areas that the reservoirs occupy. Battery storage is relatively expensive and is generally not feasible on large utility systems.

Compressed air energy storage (CAES) systems operate on a modified version of a conventional Brayton cycle gas turbine (GT) where the turbine and compressor elements are separated (Schainker et al., 1993). Electricity from low demand times is used to compress air and store it in underground geologic formations such as salt caverns or aquifers. During times of higher demand, the air is extracted, heated, and expanded in a turbine. There are currently two full scale operational CAES facilities in the world, one in Alabama and one in Germany with 110MW and 290MW turbine capacities, respectively (Shepard and Linden, 2001). Other sites are presently being planned in Texas, Iowa, and Europe, and these future CAES facilities may be prime candidates for inlet air precooling systems because their size and rate of energy consumption during the compressing phases could justify even marginal gains in compressor work reduction.

2. THERMODYNAMIC MODEL OF SYSTEM

2.1 Introduction and Assumptions

In order to examine the effects of inlet air precooling, a thermodynamic model of the air compressor system was developed. By making justifiable engineering assumptions and applying basic principles of thermodynamics, an analytical model was developed to determine the specific work required to compress air to the desired output. These assumptions are presented below:

Except for the case where humidity is condensed in the precooling section, both air and water vapor are treated as ideal gases. Air has a critical temperature of 238.5 R which is well below the temperature range that will be considered in the analysis. Water vapor has a critical pressure of 3200 psia and partial pressures of water vapor will be much lower (on the order of nearly 100 times lower) than this value.

Compression of air is a steady, isentropic process. Each compression stage is assumed to be reversible, adiabatic, and running at steady flow conditions with constant inlet and exit states.

Pressure drop through equipment (inlet chiller, intercooler, aftercooler, etc.) is assumed to be negligible.

At the intercooling stage, air is cooled to 5°F above dew point temperature, which is to ensure that no condensation occurs in the intercooler stage.

A constant atmospheric pressure of 14.7psia is used throughout the analysis. The first compressor stage compresses air to 47psia and the second compressor stage compresses air to 150psia. These pressures are based on optimizing the system by having equal compression ratios in both stages.

Specific heat and specific heat ratio, *k*, of the air are assumed to be constant. For air, constant pressure specific heat rises approximately 3% as temperature is increased from 40°F to 600°F. Specific heat ratio *k* decreases by approximately 2% over the same temperature range. Assuming these values as constant will result in errors of approximately 1% or less when calculating total work of the air compressors.

The work to power cooling water pumps in intercooling stage and aftercooling stages is negligible. Even in large compressor system (order of several MWs), the heat exchanger pump work will be negligibly small compared to air compressor work input.

The assumed parameters for the air compressor system are summarized in Table 1:

Table 1: Specifications and Parameters

Stage 1 Pressure	47 psia
Stage 2 (Final) Pressure	150 psia
Compressor efficiency	0.8
Inlet air temperature for cooled system	45°F
Refrigerator Efficiency	0.4 – 0.8
Ambient Temperature	70-110°F
Ambient Relative Humidity	20-100%

In this study, three separate cases are considered, and a separate model has been developed for each case. The first case serves as a base case to examine the work input of the compressor system without inlet air cooling over a range of ambient air conditions. The system is comprised of a two-stage air compressor system with intercooling between the stages and an aftercooler section to cool the discharge air. In the intercooling section, the air is cooled to a temperature which is 5°F above its dew point temperature. The aftercooler section is cooled to 120°F above ambient. The model is run for a range of ambient air conditions, and the theoretical work required to compress the air is determined at each inlet condition.

The second model consists of a similar system as the model in the first case, with the addition of an air conditioning system to chill the inlet air. The cooling process for the inlet air first removes sensible heat of the air until it reaches the dew point, and then removes the latent and sensible heat as humidity is condensed out the air during further cooling. For each ambient condition considered, the inlet air will be cooled to a range of specified inlet temperatures. Further, for each case, the total work required to compress the air is determined and compared against the results from the first model at similar ambient conditions. The difference between the two cases is the gross work savings. The work required to chill the air will also be determined and subtracted from the gross work savings to determine the net work savings for the system (if any). For each ambient condition, a maximum net savings will be determined by selecting the highest net saving from the range of inlet air temperatures.

The third model is similar to the second model with the exception of the source of energy required to cool the inlet air chiller. The heat rejected from the intercooler and aftercooler will be used to drive an absorption chiller system to cool the inlet air to the desired temperature. If the waste heat is not sufficient to cool the air to the desired value, a vapor compression chiller will provide the rest of the cooling power. When determining net savings for the system, only energy required to run the vapor compression system will be deducted from the gross savings. Waste heat rejected from the compressor will be considered “free energy”, otherwise known as waste heat recovery since this energy would otherwise be rejected into the ambient environtment.

2.2 Case 1 – Air Compressor System with No Inlet Cooling

A theoretical air compressor system is modeled as two isentropic air compressors with intercooling between the stages (Figure 1). The specific work for the compressor can be determined by using simple thermodynamic principals (Cengel, 2008).

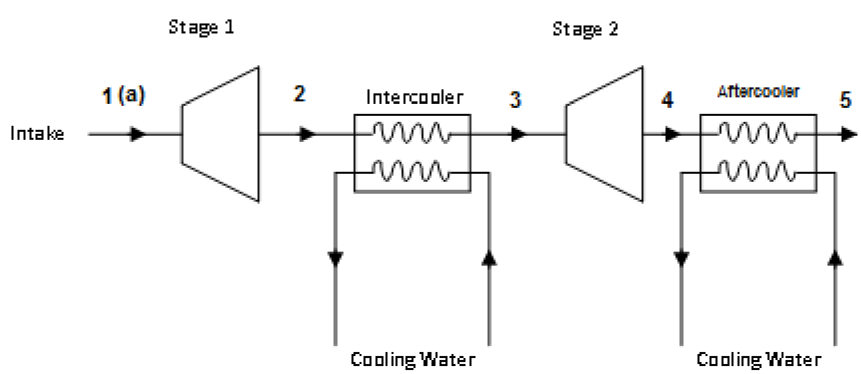

Fig. 1. Schematic for Base System, a Two-stage Compressor with Intercooling and Aftercooling.

For the given inlet temperature and relative humidity, specific humidity can be found by the following:

$$P_{v1}=\phi_1 P_{sat} \tag{1}$$

$$\omega_1 = 0.622\left[P_{v1}/(P_1 - P_{v1})\right] \quad (2)$$

The isentropic specific work for the first stage compression is found using:

$$w_{c,1\text{-}2} = (k/k\text{-}1)RT_1\left[1\text{-}(P_2/P_1)^{k-1/k}\right] \quad (3)$$

The temperature of air at stage 2 is found by:

$$T_2 = T_1(P_2/P_1)^{k-1/k} \quad (4)$$

Since no condensation occurs during the compression stage:

$$\omega_2 = \omega_2 \quad (5)$$

The partial pressure of water vapor at state 2 is found by:

$$P_{v2} = P_2/[(0.622/\omega_2) + 1] \quad (6)$$

Using steam tables and the partial pressure of the vapor, the dew point temperature at state 2 can be found. The air temperature is then cooled state 3 which is to 5°F above the dew point:

$$T_3 = T_{sat@Pv2} + 5°F \quad (7)$$

The isentropic specific work for the second stage compression can now be found using:

$$w_{c,3\text{-}4} = (k/k\text{-}1)RT_3\left[1\text{-}(P_4/P_3)^{k-1/k}\right] \quad (8)$$

The total specific work for the system can be found by summing the work of the two stages with the inclusion of a mechanical efficiency for the compressor.

$$w_{compressor,uncooled} = (w_{c,1\text{-}2} + w_{c,3\text{-}4})/\eta_{compressor} \quad (9)$$

2.3 Case 2 –System with Vapor Compression Inlet Cooling

An air chiller is used to condition the air before it is compressed. In the analysis, we investigate cooling the air to 45°F, which assures some moisture is removed from most ambient temperatures and humidities. Inlet air temperatures approaching 32°F can cause problems with the frosting of equipment due to freezing condensate. As noted previously, if the air is cooled below the dew point, water is condensed and removed from the air. The schematic of the system can be seen in Figure 2.

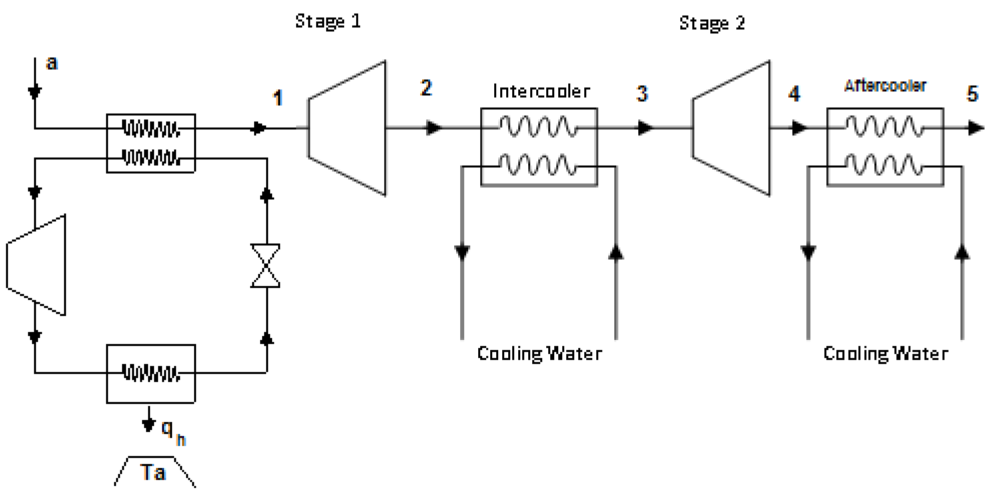

Fig. 2. Schematic of System with Vapor Compression Inlet Precooler

The same methodology used in the base system was used to determine the total work input for the air compressor system in case 2. Work input for the chiller system was determined by the total heat removed from the air and the efficiency of the refrigeration cycle. Heat removed by the AC system was determined summing the sensible heat removed by cooling the air and the latent heat from condensing water vapor:

$$q_{sensible}=c_p(T_a\text{-}T_1) \tag{10}$$

$$q_{latent}=h_{fg}(\omega_a\text{-}\omega_1) \tag{11}$$

$$q_{AC}=q_{sensible}+q_{latent} \tag{12}$$

To determine the work input for the refrigeration cycle, the COP is determined by a refrigerator efficiency based upon the Carnot efficiency of the cycle:

$$COP_{carnot}=1/[(T_a/T_1)-1] \tag{13}$$

A refrigerator efficiency is defined as the ratio of the actual COP to the theoretical COP:

$$\eta_{refrigerator}= COP/COP_{carnot}=COP*[(T_a/T_1)\text{-}1] \tag{14}$$

$$w_{AC}=q_{AC}/COP \tag{15}$$

The net work savings of the refrigeration cycle is the difference between the work input for an uncooled system and the total work input of the refrigerated compressor cycle:

$$\text{Work Savings}=(w_{compressor,\ cooled}+ w_{AC})\text{-}w_{compressor,uncooled} \tag{16}$$

2.4 Case 3 - Absorption Refrigeration Supplied by Intercooler and Aftercooler Heat

The heat removed from the air in the intercooling and aftercooling stages can be utilized in an absorption refrigeration cycle to aid in the precooling of inlet air. The input heat for the absorption chiller will come from the intercooler and aftercooler of the cycle. For calculation of the COP,rev of the chiller, the source temperature of the input heat will be selected as a value somewhere between the temperature of the air entering the intercooler and the temperature of the air after exiting (most likely taken as the halfway point). The schematic of the system can be seen in Figure 3.

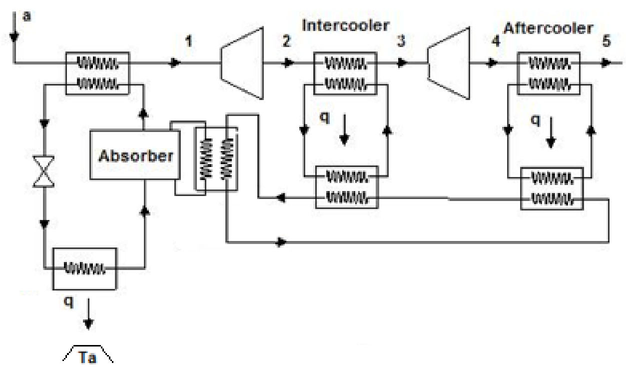

Fig. 3. Schematic of System with Absorption Chiller

The heat removed from the intercooler and aftercooler can be found by:

$$q_{intercooler}=(T_3-T_2)*c_p \tag{17}$$

$$q_{aftercooler}=(T_5-T_4)*c_p \tag{18}$$

The COP of an ideal absorption cycle can be found by:

$$COP_{ideal}=(1-T_a/T_s)*(T_1/T_a-T_1)=q_{cool}/q_{waste} \tag{19}$$

Where Ts is the temperature of the waste heat source. This temperature will be defined as the average of the air temperature before and after the respective cooling stage. Conservatively, this value can be chosen as the exit temperature of the cooling stage.

A refrigeration efficiency is again used and it is defined as:

$$\eta_{refrigerator} = COP/COP_{ideal} \tag{20}$$

The analysis of compressor work can now be performed by using a method similar to that used in the vapor compression refrigeration analysis. For this method, however, all available absorption cooling is used to cool inlet air. If the absorption cycle is unable to cool or dehumidify the air to the desired inlet state, a vapor compression system supplements the cooling power. If required, the work input of the vapor compression system will be deducted from the compressor work savings.

3. RESULTS OF THERMODYNAMIC MODEL

3.1 Case 1 Results

For Case 1, the compressor work varied linearly with temperature as seen in Figure 4 due to increased specific volume with temperatures. Compressor work increased by approximately 3.5-4% as relative humidity was increased from 20% to 100% at any given temperature. The required work increases due to higher relative humidity is caused by higher temperatures in the intercooling phase. More humid air cannot be intercooled as much as dryer air due to the possibility of condensation.

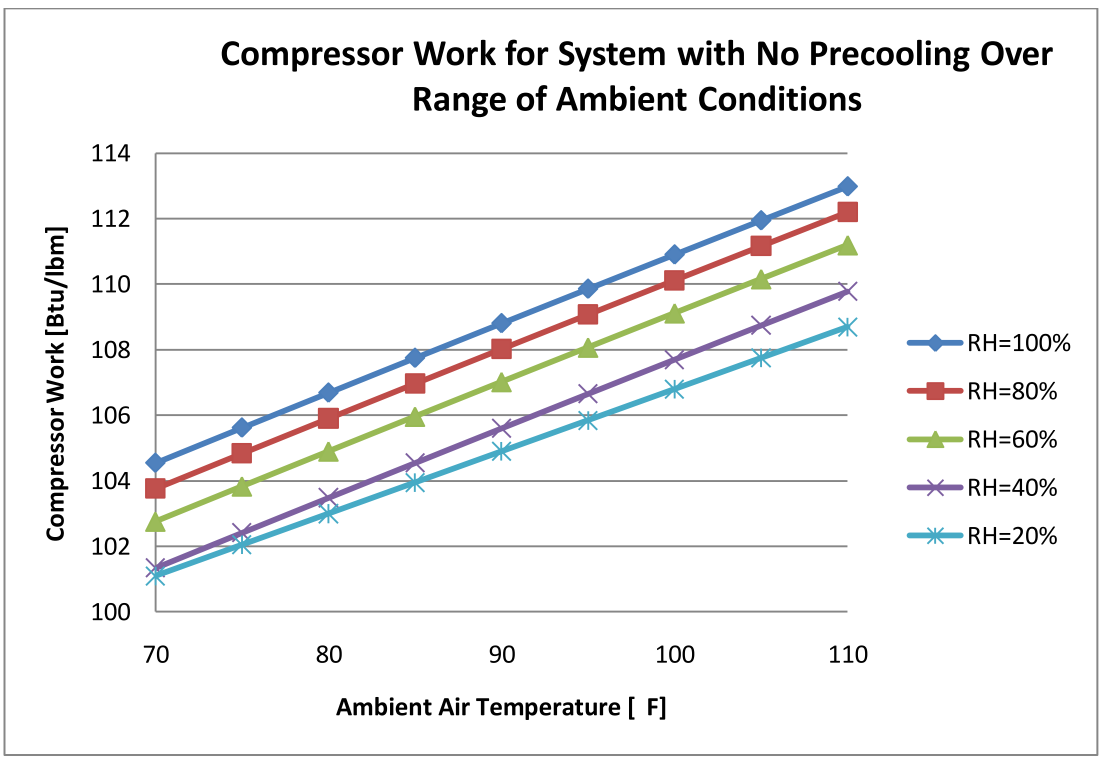

Fig. 4. Results of Case 1

3.2 Case 2 Results

Figure 5 shows the results of Case 2 with an assumed refrigeration efficiency of 0.4. The highest savings were seen at the lower ambient temperatures with work savings of about 1.8-3.5% for temperatures lower than 85°F, and at the higher humidities. At temperatures above 85°F, work savings drop off sharply and total work input becomes higher than for the case of a system with no cooling.

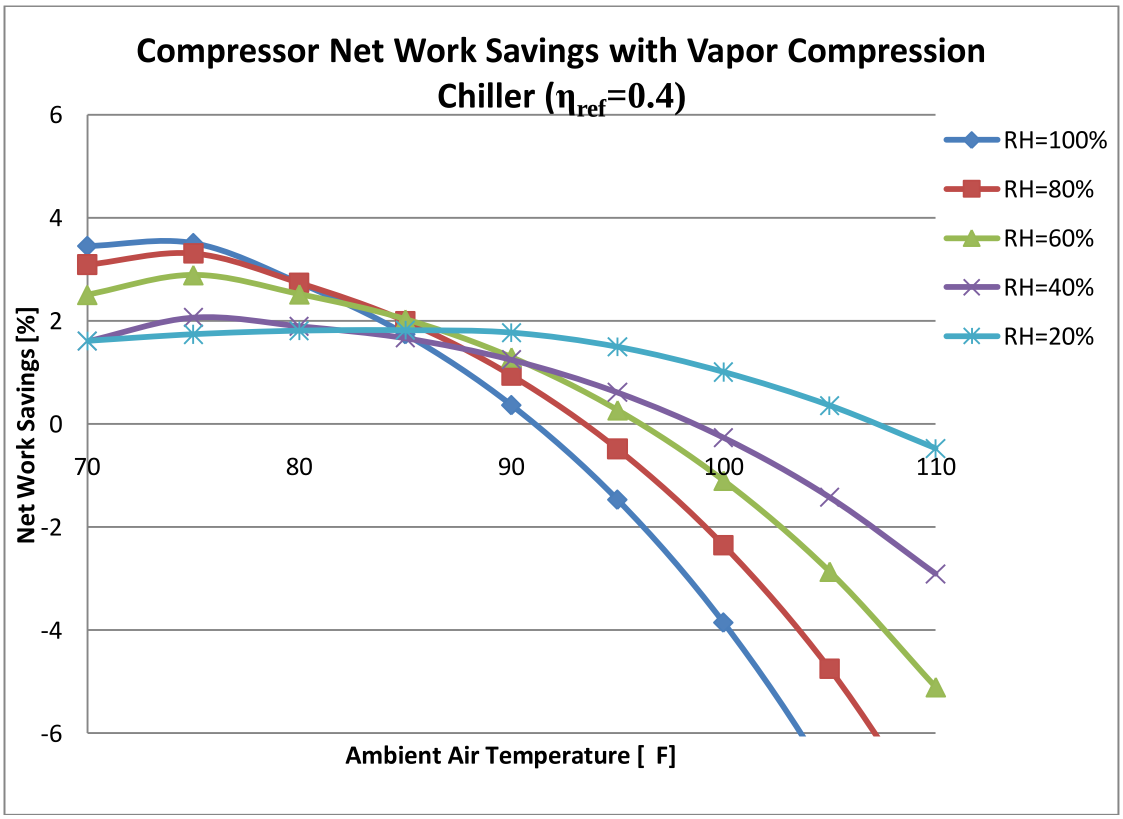

Fig. 5. Results of Case 2 with η_{ref}=0.4

Figure 6 shows the results of Case 2 with an assumed refrigeration efficiency of 0.5. The highest savings were seen at the lower ambient temperatures with work savings of about 2-4% for temperatures lower than 90°F, with and at the higher humidities. At temperatures above 90°F, work savings drop off sharply and total work input becomes higher than for a system with no cooling for conditions >100°F and >60% RH.

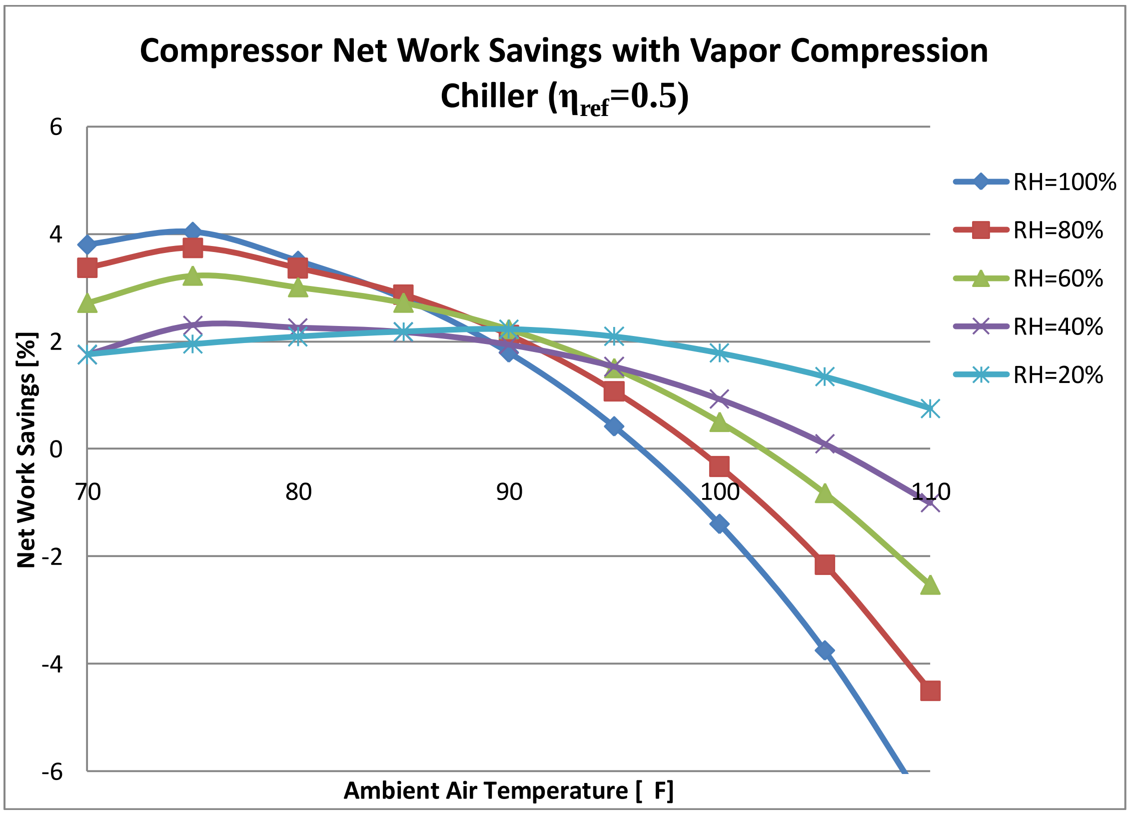

Fig. 6. Results of Case 2 with η_{ref}=0.5

Figure 7 shows the results of Case 2 with an assumed refrigeration efficiency of 0.6. The highest savings were seen at the lower ambient temperatures with work savings of about 2-4.2% for temperatures lower than 95°F, and at the higher humidities. At temperatures above 95°F, work savings begin to drop off sharply with most cases falling to less than 2% savings than for the case of no cooling.

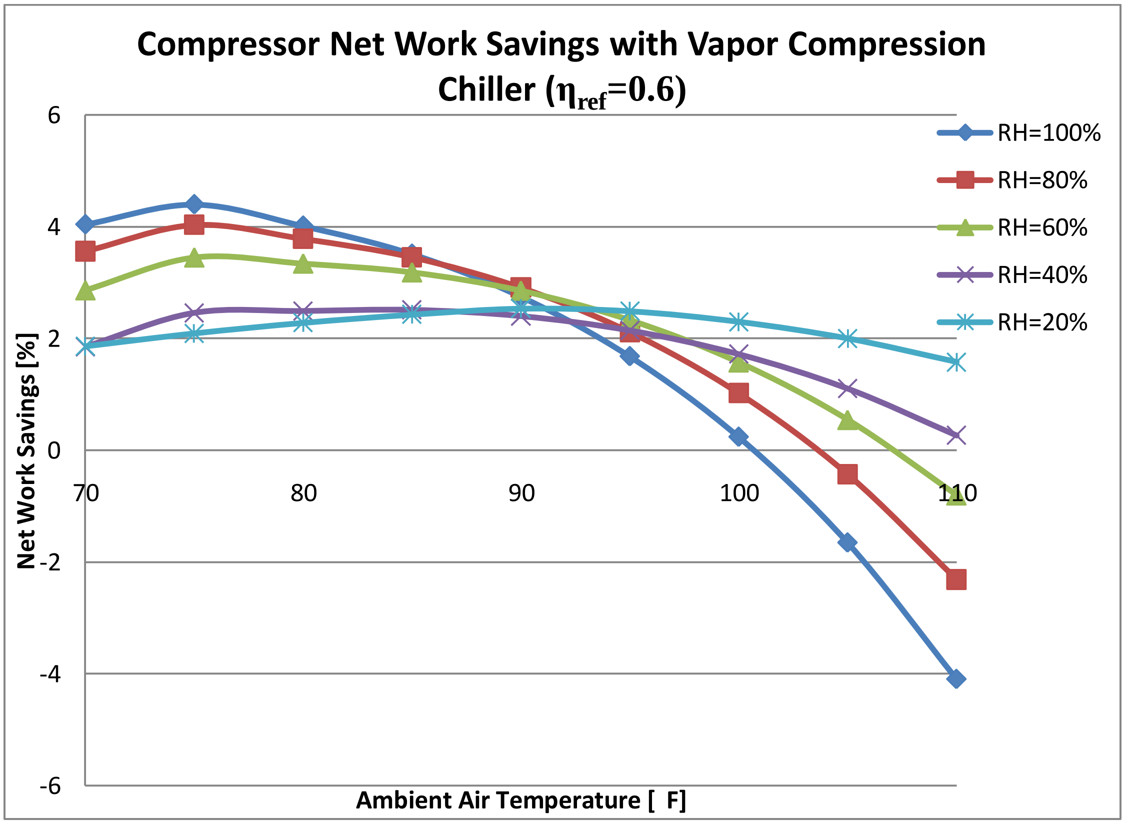

Fig. 7. Results of Case 2 with η_{ref}=0.6

Figure 8 shows the results of Case 2 with an assumed refrigeration efficiency of 0.7. The highest savings were seen at the lower ambient temperatures with work savings of about 2-4.4% for temperatures lower than 95°F and at the higher humidities. At temperatures above 95°F, work savings begin to drop off sharply for relative humidity >40%. Work savings of 1-2% still exist for temperatures up to the maximum investigated temperature of 110°F when RH<40%.

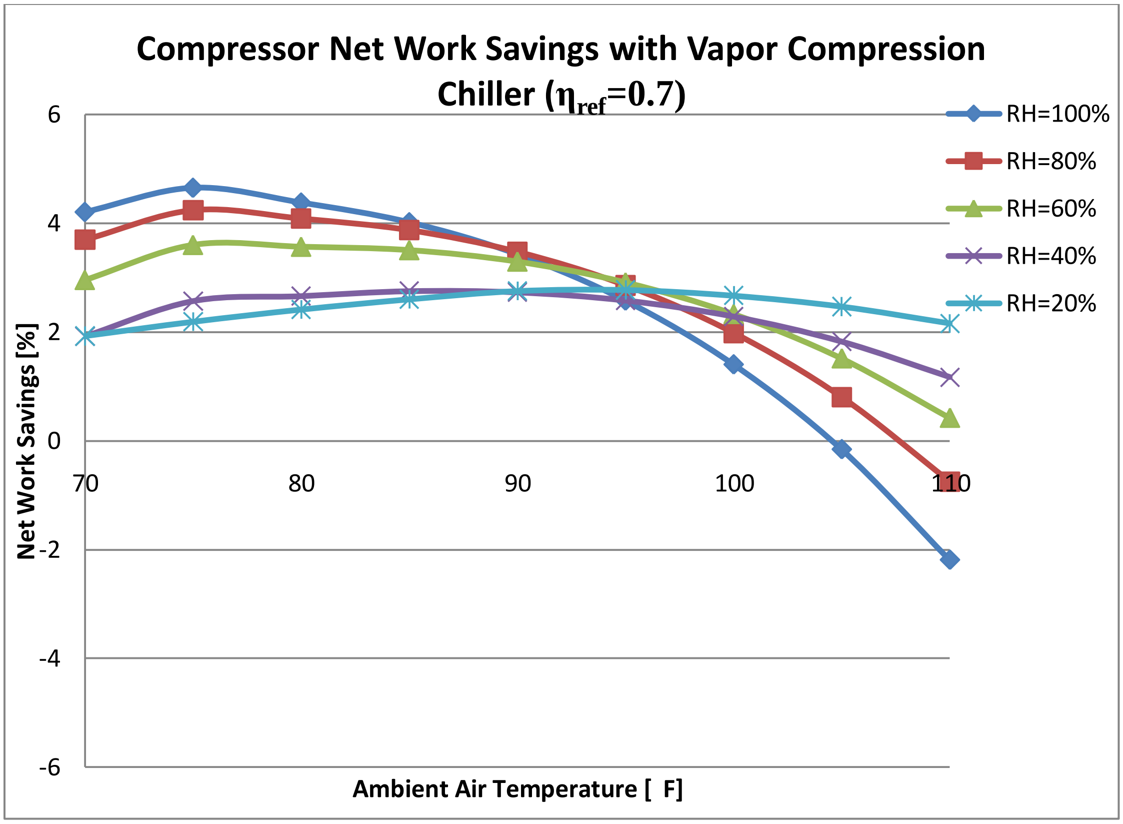

Fig. 8. Results of Case 2 with η_{ref}=0.7

Figure 9 shows the results of Case 2 with an assumed refrigeration efficiency of 0.8. The highest savings were seen at the lower ambient temperatures with work savings of about 3-5% for temperatures lower than 100°F and RH>60% and at the higher humidities. For temperatures up to 100°F and RH<40%, work savings were relatively stable at about 2-3% across the temperature range. At temperatures above 100°F, work savings begin to drop off sharply for relative humidity >40% while work savings of 2-3% still exist for temperatures up to the maximum investigated temperature of 110°F when RH<40%.

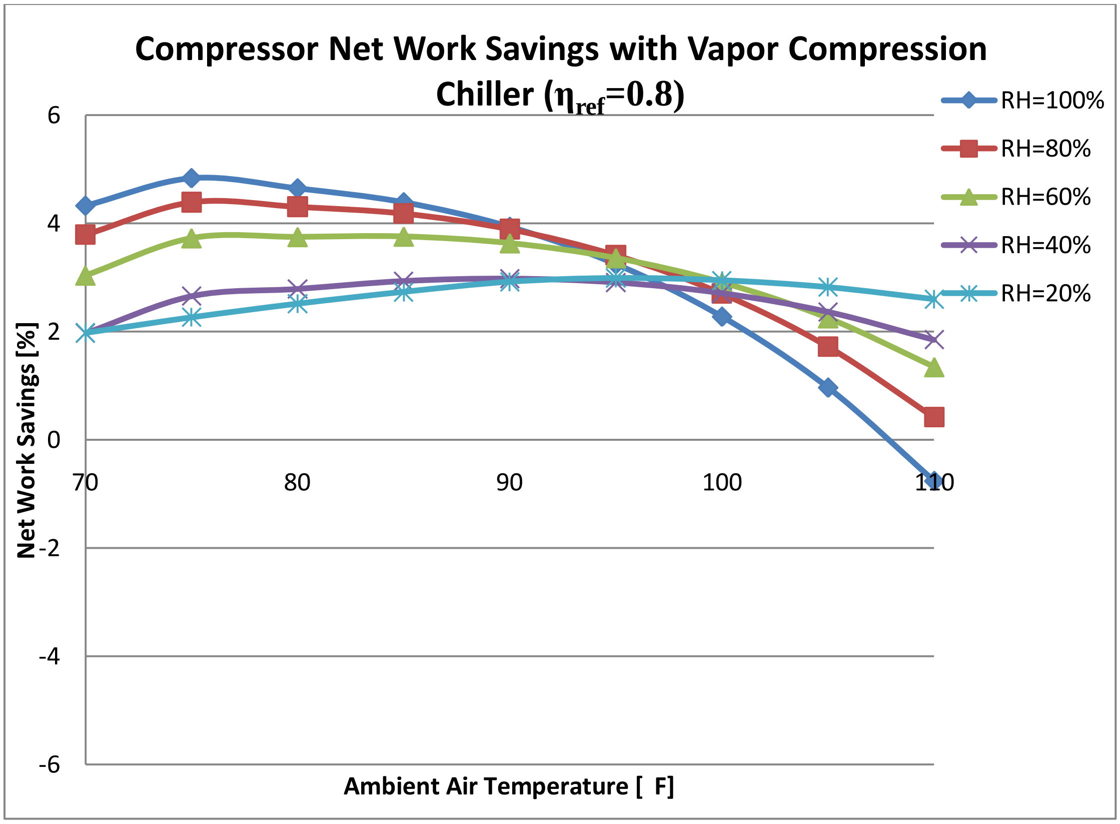

Fig. 9. Results of Case 2 with η_{ref}=0.8

3.3 Case 3 Results

Figure 10 shows the results of Case 3 with an assumed refrigeration efficiency of 0.4. Work savings increased linearly for all cases up to 85°F, with savings of up to 6-7% in the high humidity range (80-100% RH) and savings of 2-3% in the low humidity range (20-40%). Savings began to decrease for each humidity case once temperatures reached a certain temperature. For 100% RH, the decreasing trend begins at 85°F. At lower humidity, this trend begins at a temperature roughly 5°F higher for each 20% drop in RH.

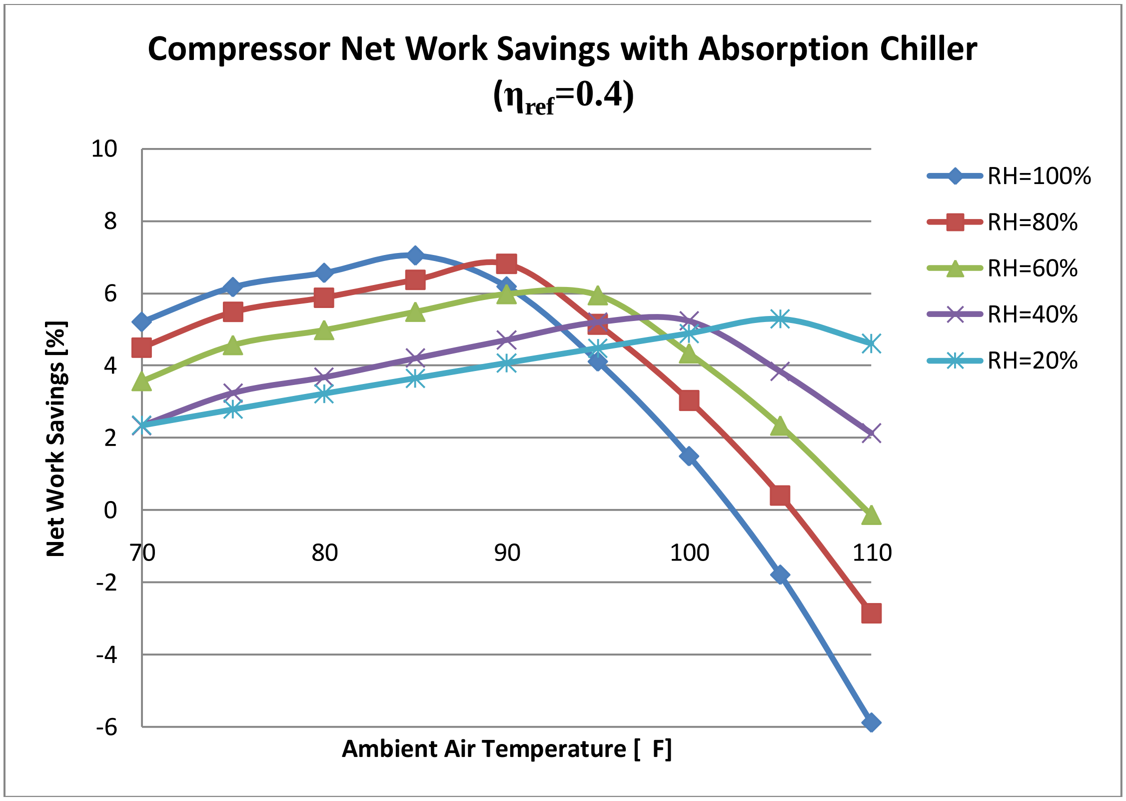

Fig. 10. Results of Case 3 with η_{ref}=0.4

Figure 11 shows the results of Case 3 with an assumed refrigeration efficiency of 0.5. Work savings increased linearly for all cases up to 90°F (with the exception of the 100% RH case which increased up until 85°F), with savings of up to 6-7.7% in the high humidity range (80-100% RH) and savings of 2.2-4.2% in the low humidity range (20-40%). Below 90°F, there was a higher work savings for higher humidity cases. Above 90°F, savings began to decrease, with higher humidity cases decreasing more sharply. The temperature at which savings began to decline was higher for lower humidity cases.

Above 100°F, savings of 4-5% were seen for lower (20-40%) humidity. Higher humidities (>60%) at temperatures above 100°F saw little to no gain.

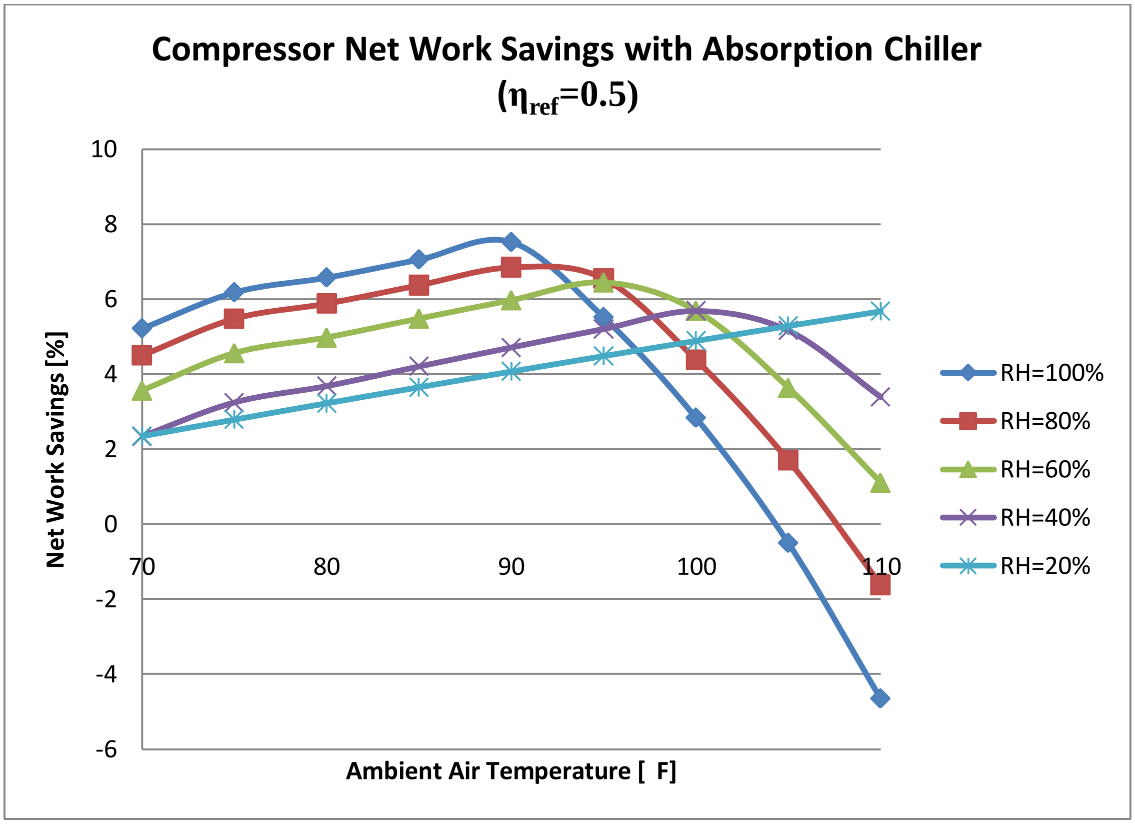

Fig. 11. Results of Case 3 with η_{ref}=0.5

Figure 12 shows the results of Case 3 with an assumed refrigeration efficiency of 0.6. Work savings increased linearly for all cases up to 95°F (with the exception of the 100% RH case which increased up until 90°F), with savings of up to 6-7.7% in the high humidity range (80-100% RH) and savings of 2.2-4.2% in the low humidity range (20-40%). Below 95°F, there was a higher work savings for higher humidity cases. Above 95°F, savings began to decrease, with higher humidity cases decreasing more sharply.

The temperature at which savings began to decline was higher for lower humidity cases. Above 100°F, savings of 4-5% were seen for lower (20-40%) humidities. Higher humidities (>60%) at temperatures above 100°F saw little to no gain.

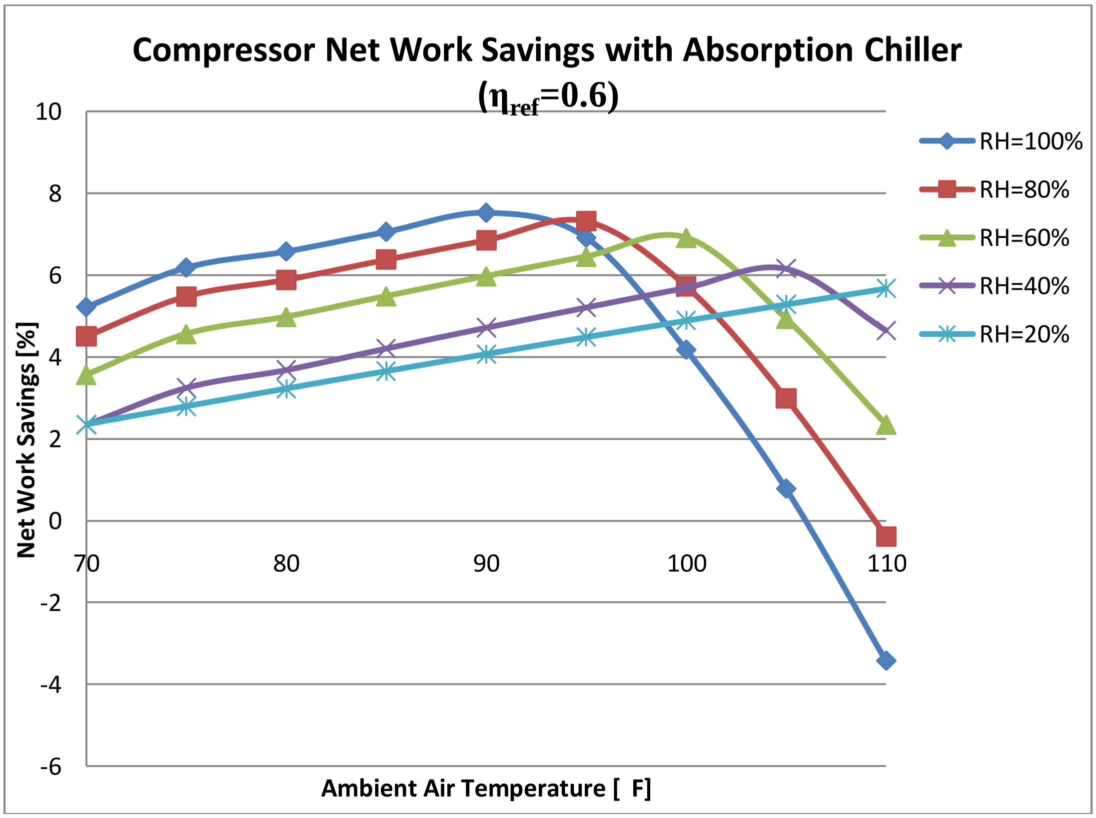

Fig. 12. Results of Case 3 with η_{ref}=0.6

Figure 13 shows the results of Case 3 with an assumed refrigeration efficiency of 0.7. Work savings increased linearly for all cases up to 100°F (with the exception of the 100% RH case which increased up until 95°F), with savings of up to 7-8% in the high humidity range (80-100% RH) and savings of 4-5% in the low humidity range (20-40%). Below 95°F, there was a higher work savings for higher humidity cases. Above 95°F, savings began to decrease for RH>60%, with RH<40% continuing to increase. Above

100°F, savings of 5-6% were seen for lower (20-40%) humidities. Higher humidities (>60%) at temperatures of 100°F saw gains of 5.8-7.2% rapidly diminish with increasing temperature.

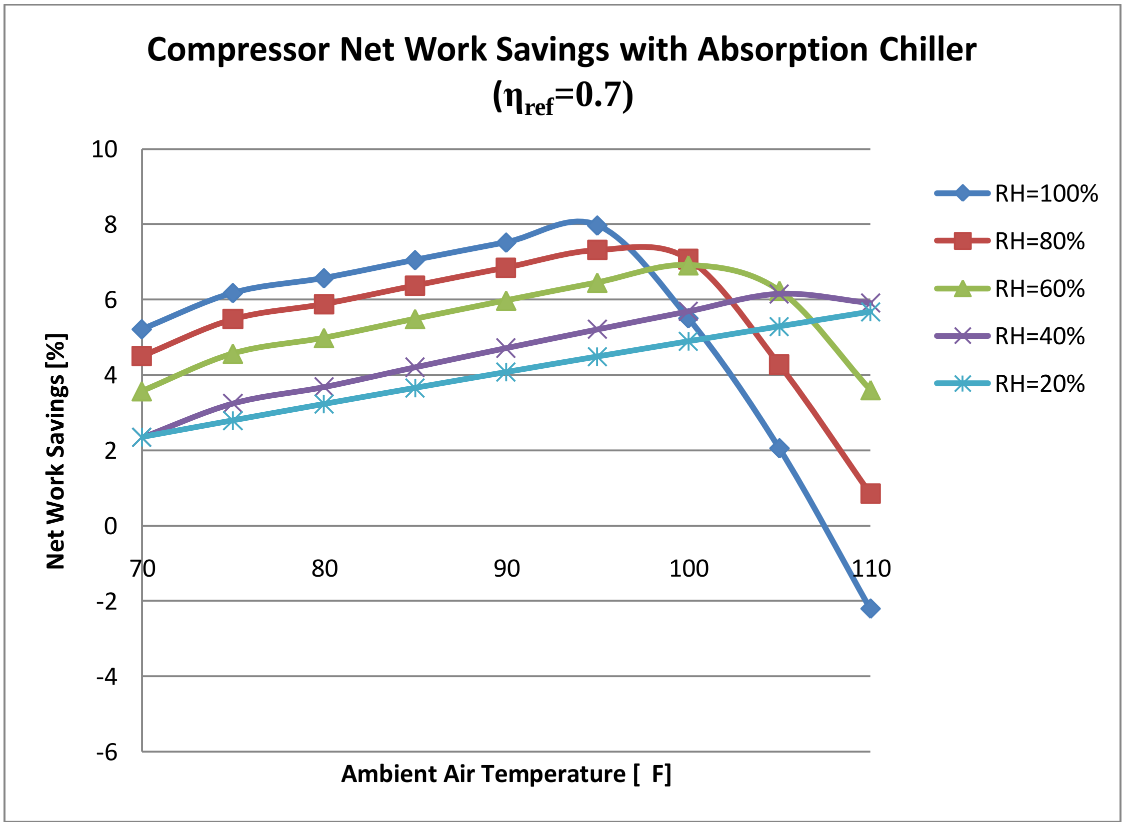

Fig. 13. Results of Case 3 with η_{ref}=0.7

Figure 14 shows the results of Case 3 with an assumed refrigeration efficiency of 0.8. Work savings increased linearly for all cases up to 95°F, with savings of up to 7-8% in the high humidity range (80-100% RH) over a broad range of temperatures and savings of 4-5% in the low humidity range (20-40%). Below 95°F, there was a higher work savings for higher humidity cases. Above 95°F, savings began to decrease for RH>60%, with RH<40% continuing to increase. Above 100°F, savings of 5-6% were seen for lower (20-40%) humidities. The higher humidities (>60%) at temperatures of 100°F saw gains of 5.9-7.4% rapidly diminish with increasing temperature.

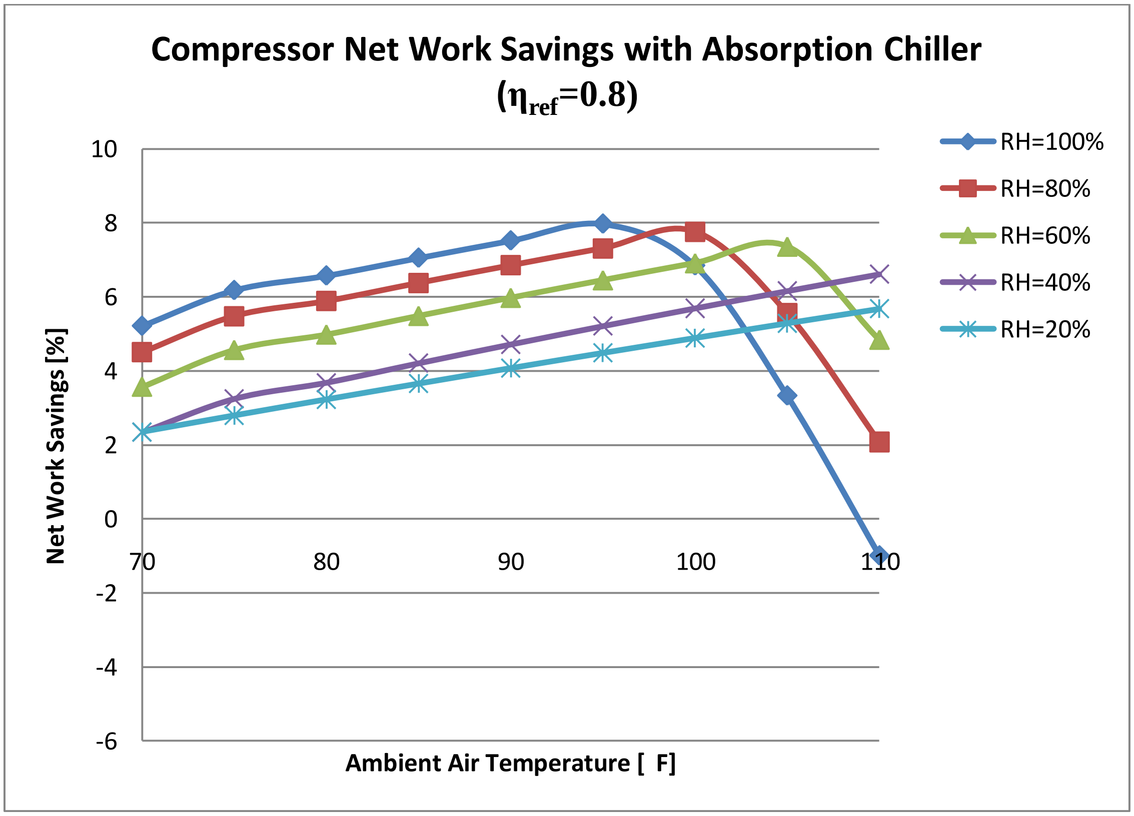

Fig. 14. Results of Case 3 with η_{ref}=0.8

The "turning point" in the results of Case 3 represents the point in which the heat rejected by the compressor is not enough to provide all of the cooling power by the absorption chiller. At this point additional work input for a vapor compression chiller is needed, thus increasing our work input and causing the performance gains to fall.

Results from Case 3 show that significant air compressor work savings can be realized by the addition of even poor efficiency absorption chillers, utilizing the "free energy" of the intercooling and aftercooling stages.

In Figure 15, the results of Cases 2 (dashed lines) and 3 (solid lines) are compared at a constant RH of 40%. It is seen that the absorption chiller used in Case 3 results in higher net work savings over the range of ambient temperatures. The difference in air compressor work savings between the two cases is small (<0.5%) at lower temperatures, but increase to about 4% as ambient air temperatures rise.

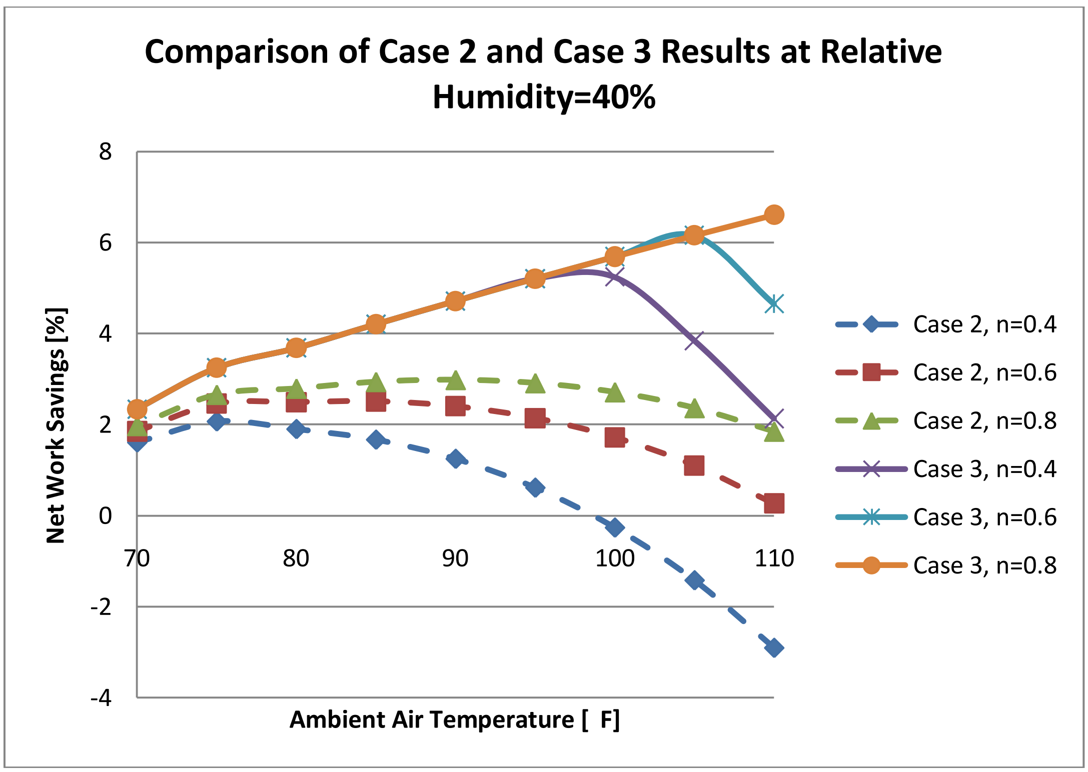

Fig. 15. Comparison of Cases 2 and 3 at RH=40%

In Figure 16, the results of Cases 2 (dashed lines) and 3 (solid lines) are compared at a constant RH of 80%. It is seen that the absorption chiller used in Case 3 results in higher net work savings over the range of ambient temperatures. The difference in air compressor work savings between the two cases is small (<0.5%) at lower temperatures, but increase to about 5% as ambient air temperatures rise.

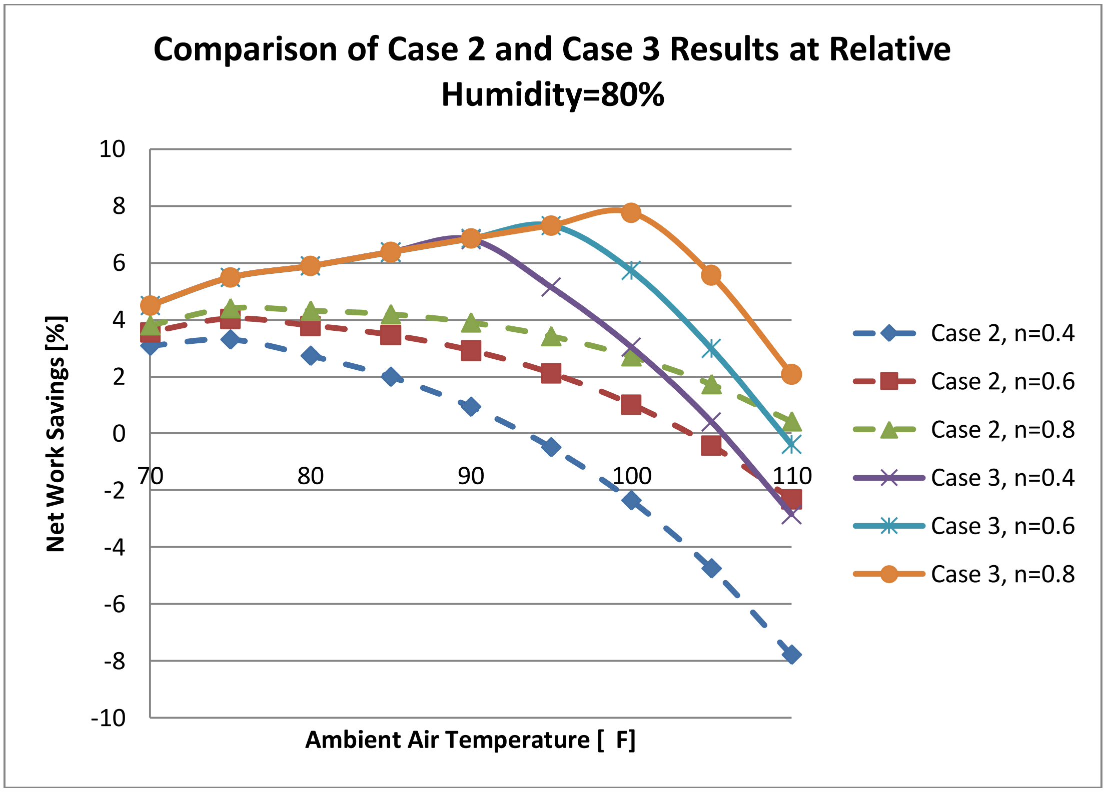

Fig. 16. Comparison of Cases 2 and 3 at RH=80%

In Figure 17, the results of Cases 2 (dashed lines) and 3 (solid lines) are compared at a constant refrigerator efficiency of 0.4 over a range of ambient RH. At lower temperatures, savings between the two systems are small (approximately 1%) and increase to a maximum of approximately 5% with higher temperatures. The difference in work savings between Case 3 and Case 2 increases with higher humidities.

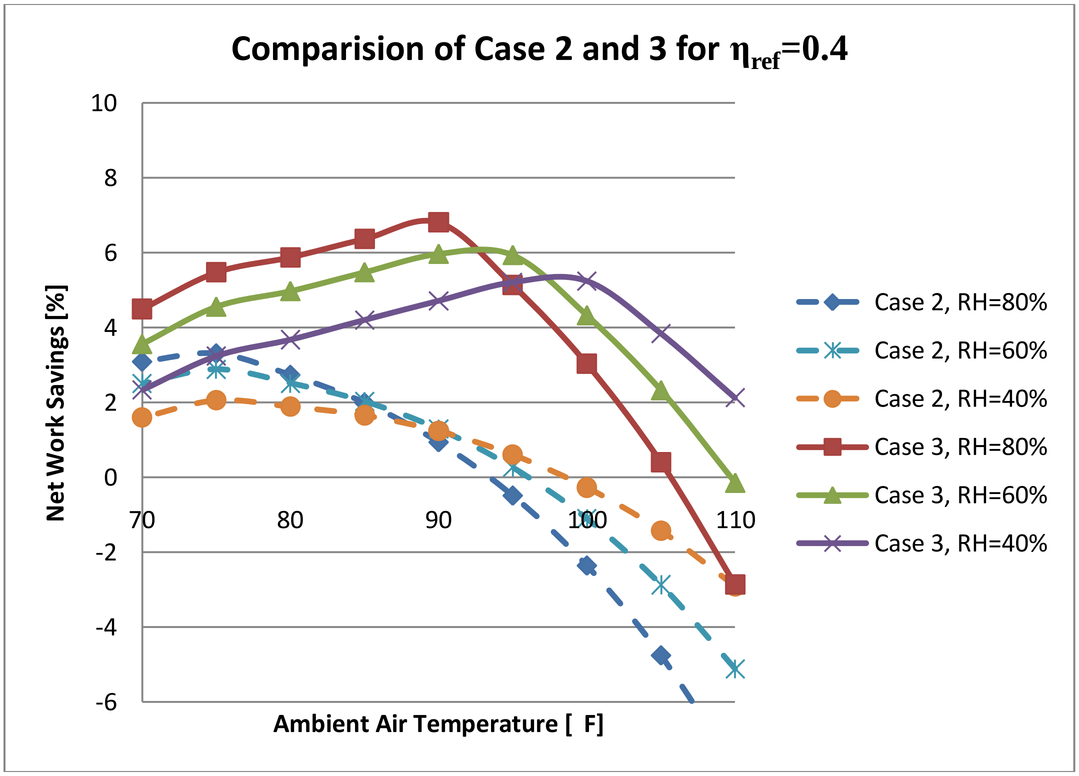

Fig. 17. Comparison of Cases 2 and 3 at η_{ref}=0.4

In Figure 18, the results of Cases 2 (dashed lines) and 3 (solid lines) are compared at a constant refrigerator efficiency of 0.6 over a range of ambient RH. At lower temperatures, savings between the two systems are small (approximately 0.8%) and increase to a maximum of approximately 5.5% with higher temperatures. The difference in work savings between Case 3 and Case 2 increases with higher humidities.

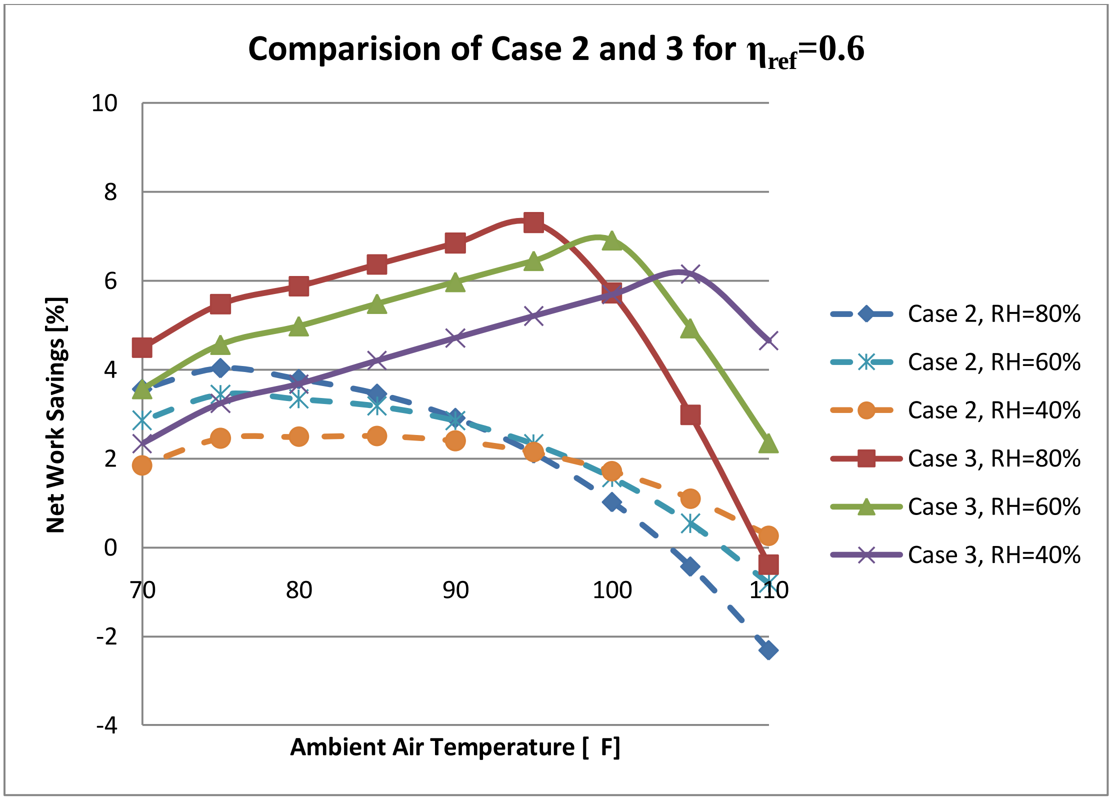

Fig. 18. Comparison of Cases 2 and 3 at η_{ref}=0.6

In Figure 19, the results of Cases 2 (dashed lines) and 3 (solid lines) are compared at a constant refrigerator efficiency of 0.6 over a range of ambient RH. At lower temperatures, savings between the two systems are small (approximately 0.5%) and increase to a maximum of approximately 5% with higher temperatures. The difference in work savings between Case 3 and Case 2 increases with higher humidities.

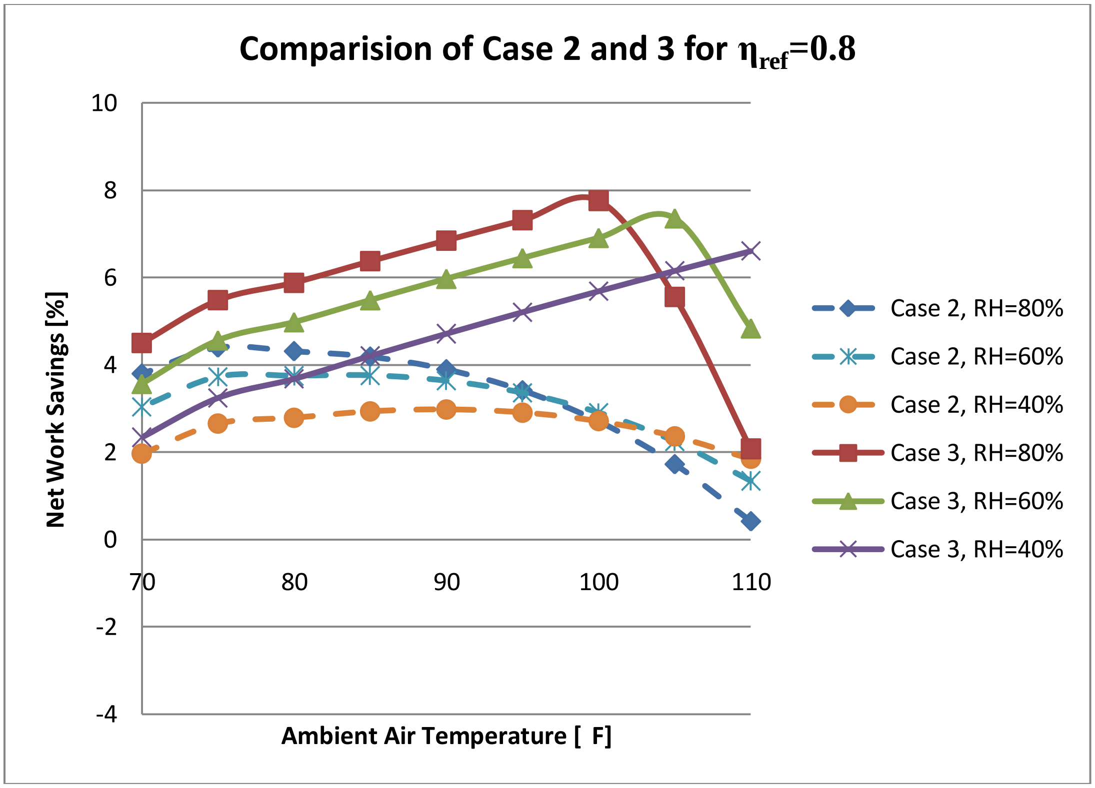

Fig. 19. Comparison of Cases 2 and 3 at η_{ref}=0.8

4. CONCLUSIONS AND CONTINUING WORK

4.1 Conclusions

The model described in this paper presents a methodology for analyzing the total isentropic work required to compress air from atmospheric pressure to 150psia when an air conditioner is used to cool the air before compression. The results show that the efficiency of an air compressor working in warmer and more humid environments can be increased by means of chilling the inlet air. These findings are promising and show a relatively easy method of improving compressor efficiencies in warmer and more humid climates. Using an inlet air precooler should be a consideration for future industrial and manufacturing facilities, as they may be able to significantly lower energy costs by making their compressors more efficient.

For higher refrigeration efficiencies (>0.6), air compressor work savings of up to 4-5% can be realized when a vapor compression air conditioning system is used to cool and dehumidify inlet air. Using a higher efficiency absorption air chiller to cool and dehumidify inlet air resulted in work savings of up to 8%. Lower refrigerator efficiency (<0.6) vapor compression refrigeration resulted in work savings of 2-3.5% and lower efficiency absorption chillers resulted in work savings of 3-7.5%. The absorption chiller case showed higher net work savings than vapor compression chilling because it was driven by the waste heat given off by the compressor system, making its required work input much lower than the vapor compression system.

4.2 Continuing Work

In order to apply the results of this study into real world systems, a more detailed engineering analysis could be performed. Chiller systems were modeled with a "black box" approach in which energy input and cooling power were related by using a coefficient of performance. Further study and research would be required to ensure the assumed COP values are valid. Also, the compressors were modeled as isentropic, which is not representative of an actual case. Future studies can determine the effects of non-isentropic compression and how behavior of real compressors will deviate from the model presented.

Also, economic investigations will be required to determine how these energy savings can be translated to cost savings. The most important factors for economic viability are the type, efficiency, and cost of the air chiller system. Actual climate data for the location in which the system will be located could be used as model input, as well as cost per kWh of electricity, daily utilization, and initial cost of the AC system could be added to determine the potential economic impact of the system.

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