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AN INDUSTRIAL VAPOR ABSORPTION AIR CONDITIONING APPLICATION

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Abstract: In this study, the application of VAR system to an industrial company is investigated. For this purpose, the company which requires an air-conditioning system to its office building was chosen. After determining this company had an industrial furnace, this research aimed to design the VAR system utilizing waste heat from this furnace flue gases to air-condition the office building. Firstly, the physical properties and heat quantity of the flue gases were determined and the cooling load of the office building was calculated. It was found out that the flue gases had enough heat capacity to drive the VAR system. Next, single effect VAR system was introduced and the thermodynamic, energy and exergy analysis were made. The COP of the VAR system is calculated to be 0.64 and the highest exergy destruction was obtained at the generator which is 37.19kW. Finally, the application of the VAR system to this industrial company was analyzed in detail. The capital, maintaining and operating costs of the VAR system were analyzed and compared with alternative systems.
Keywords: Absorption, Water-Lithium bromide solution, Vapor Absorption Refrigeration.

ABSORPSİYONLU SOĞUTMA SİSTEMİNİN SANAYİYE UYGULANMASI

Özet: Bu çalışmada, Absorpsiyonlu Soğutma (ABS) sistemin bir sanayi firmasına uygulanması incelenmiştir. Bu amaçla, ofis binasında bir klima sistemi ihtiyacı olan bir firma seçilmiştir. Firmada bir endüstriyel fırın bulunduğu belirlendikten sonra, ofis binasının klimalandırılması amacıyla bu endüstriyel fırının baca gazındaki atık ısı ile tahrik olan ABS sisteminin tasarlanması araştırılmıştır. Başlangıç olarak, baca gazının fiziksel özellikleri ve içerdiği atık ısı miktarı tespit edilmiş ve aynı zamanda ofis binasının soğutma yükü hesaplanmıştır. Baca gazının ABS sistemini tahrik edebilecek miktarda atık ısıya sahip olduğu bulunmuştur. Daha sonra, tek kademeli ABS sistemi izah edilmiş ve termodinamik enerji ve ekserji analizleri yapılmıştır. ABS sisteminin Soğutma Tesir Katsayısı (STK) 0.64 olarak hesaplanmış ve en büyük ekserji yıkımının 37.19 kW ile kaynatıcıda meydana geldiği saptanmıştır. Son olarak, ABS sisteminin bu sanayi firmasına uygulanması detaylı bir şekilde analiz edilmiştir. ABS sisteminin yatırım, işletme ve bakım maliyetleri araştırılmış, alternatif sistemler ile karşılaştırılmıştır.
Anahtar Kelimler: Absorpsiyon, Su-Lityum bromür eriyiği, Absorpsiyonlu Soğutma Sistemi.

NOMENCLATURE

a	Characteristic dimension associated with time of spreading [m]
A	Surface area [m^2]
c_p	Specific heat [kJ/kg ºC]
COP	Coefficient of performance
d	Diameter [m]
E	Energy [kW]
EA	Equivalent annual value
EAC	Equivalent annual cost
Ex	Exergy [kW]
F	Dry area coefficient
ex	Specific exergy [kJ/kg]
f	Flow ratio
g	Gravitational acceleration [m/s^2]
h	Enthalpy[kJ/kg], heat transfer coefficient [W/m^{2}K]
h^*	Modified latent heat of vaporization [kJ/kg]
i	Interest rate
k	Thermal conductivity of tube material, solution thermal conductivity [W/m K]
l	Estimated distance that a drop of solution spreads on the tube [m]
L	The unwrapped length of the tube length [= $\pi d_o/2$] [m]
m	Mass flow rate [kg/s]
n	The number of sites at which droplets form; time period; efficiency
Nu	Nusselt number [$=h \cdot d_h/k$]
P	Pressure [kPa], present cost
Pr	Prandtl number [$\mu \cdot c_p/k$]
q	Thermal load [kJ/kg]
Q	Heat transfer [kW]
r	Radius of the tube [m]
Re	Reynolds number
s	Fin pitch, entropy [kJ/kg K]

t	Thickness [m]
T	Temperature [ºC]
ΔT	Logarithmic temperature difference [ºC]
U	Overall heat transfer coefficient [W/m^2 K]
W	Power [kW]
X	Lithium bromide concentration

Greek symbols

ε	Effectiveness
η	Exergy efficiency
θ	Contact angle at the solution tube interface
μ	Dynamic viscosity [Pa s]
ρ	Density [kg/m^3]
σ	Solution surface tension [N/m]
ϑ	Specific volume of solution [m^3/kg]

Subscripts

AB	Absorber
b	Nucleate boiling
c	Initial cost, convective
ch	Chemical
ci	Cold fluid inlet
co	Cold fluid outlet
conv	Convection
cond	Conduction
CO	Condenser
d	Developing region
dest	Destruction
e	Equivalent
eff	Effective
est	Estimation
EV	Evaporator
f	Fouling, fin
fg	The difference between saturated vapor and liquid
GE	Generator
hi	Hot fluid inlet
ho	Hot fluid outlet
horiz	Horizontal
k	Component
l	Liquid
m	Mean
o	Outer
p	Pump
r	Refrigerant
ref	Reference environment
rtv	Refrigerant throttle valve
sat	Saturation
s	Solution
SHE	Solution heat exchanger
tot	Total
VAR	Vapor Absorption Refrigeration
VCR	Vapor compression system
VRF	Variable refrigerant flow
v	Vapor
w	Wall, wet area
1,2…	State points
0	Reference environment state

INTRODUCTION

Because of the limited fossil fuel sources, ever-increasing energy costs and global warming, people, especially engineers, have to concentrate on both the improvements of the efficiency of the existing systems and the utilization of the waste heat, as well as searching for new energy sources. However, energy consumption of developing counties where the economy and industry grows rapidly increases rapidly. It is claimed that the crude oil and natural gas sources will depleted in next 50 years. As far as the air conditioning and refrigeration are concerned, buildings use quite a lot of energy. The main reason for this is the air conditioning and refrigeration systems mainly use Vapor Compression Refrigeration (VCR) Systems and compressing vapor consumes really a big amount of energy. In cooling season, the consuming electrical energy can reach peak and the brownout situations can be encountered. It is reported that %15 of the generated electricity in whole world is consumed by mostly these systems. For whole buildings, it is estimated that %45 of the consumption is for air-conditioning purposes. The electrical energy consuming by VCR systems is mostly produced by using fossil fuels leads to depletion the fossil fuel sources. Another effect is that the releasing gas during production the electrical energy causes to increase the amount of greenhouse gas emissions. Furthermore, some of the refrigerants using by VCR system as working fluid such as CFCs, HCFC and HFCs which Montreal and Kyoto Protocol limits to use contributes to ozone layer depletion (Choudhury et al.,2010; Kalkan et al., 2012). Utilizing waste heat from industrial process is an effective way in respect of energy efficiency, economy and environment in order to reach sustainable development. As far as the utilization of the waste heat is concerned, Vapor Absorption Refrigeration (VAR) Systems which are devices with the unique capability of producing cold water/air by using the heat sources comes front such as tri-generation applications (Law et al.,2013; Chen et al.,2014). Economical and energy savings by applying VAR systems driven by heat in exhaust gas from industrial process was investigated also. Garimella (2012) investigated the low-grade waste heat recovery for chilled and hot water generation. Heat from waste gas from an industrial process which temperature is 120ºC is supplied to drive the system. They found that annual savings can be achieved up to $1.2 million in such an application. Balaji and Ramkumar (2012) studied the waste heat recovery from steam turbine exhaust for vapor absorption system in sugar industry.

The VAR system performance has been investigated by many researchers. The effect of operating parameters on the system performance for single effect VAR system was investigated. It was found that the solution heat exchanger (SHE) had more effects on the investigated parameters than the refrigerant heat exchanger (RHE)

(Kaynakli and Kilic, 2007). Kilic and Kaynakli (2007) investigated the second law-based thermodynamic analysis of water-lithium bromide VAR system. They analyzed the effect of the temperatures of main system components on the system performance, the irreversibilities in the thermal process and the exergy loss of each component. They found that the increasing heat source temperature is dominant on the exergetic efficiency. The exergy analysis of a single effect VAR system using water-lithium bromide solution is made (Şencan and Yakut, 2005). According to the researcher, the condenser and evaporator has less exergy loss than the generator and absorber because of the heat of mixing in the solution. Kaynakli et al. (2015) investigated the energy and exergy analysis of a double effect VAR system based on different heat sources. Their system used water/lithium bromide as working fluid pair and the refrigeration system run on various heat sources such as hot water, hot air and steam. They concluded that the exergy destruction was maximized when hot air heat source was used and minimized by utilizing hot water heat source.

As can be seen from the literature, the related papers just focused on energy savings by applying VAR systems, but not tried to find the effective way to meet the demands and use energy efficiently. On the other hand, this paper aims to investigate the utilization of the waste heat from an industrial company by using the VAR system for air conditioning purposes and comparing VAR systems with alternative air conditioning systems. Additionally, almost all of the researchers use hot water or steam to drive VAR system while making exergy analysis. Unlike the literature, the exergy analysis for the applied VAR system is applied to show exergy destructions parameters for flue gas fired VAR system in detail.

VAPOR ABSORPTION REFRIGERATION (VAR) SYSTEMS

The VAR system which is quite similar to the vapor compression refrigeration system includes a thermal compressor which consists of an absorber and a generator, instead of a conventional compressor which requires a lot of mechanical energy input. There is no need for the compressor and its compression and maintenance costs in VAR systems. Heat input is enough to operate VAR system except for the small amount of mechanical energy input to the liquid pump. This allows VAR system to be used in utilizing the waste heat and also the solar and geothermal energy (Horuz, 1998).VAR system, shown schematically in Fig. 1, operates in a cycle. The VAR system basically consists of an evaporator, a condenser, a generator, an absorber and a solution heat exchanger. The generator is driven by a heat source, the evaporator takes heat to provide refrigeration and the condenser and absorber release heat to the medium (generally to the ambient air).

The VAR cycle uses a refrigerant-absorbent solution rather than pure refrigerant as the working fluid. The absorbent acts as a secondary fluid to absorb the primary fluid which is refrigerant. This study will concentrate on the VAR system using water-lithium bromide (LiBr) solutions where water is refrigerant and water-lithium bromide is absorbent. The refrigerant-absorbent solution passing through the solution pump is referred to as a weak solution, being relatively weak in LiBr. The solution returning from the generator to the absorber contains only a little more LiBr compared to the solution being pumped from the absorber to the generator and is therefore referred to as rich solution (see Fig.1).

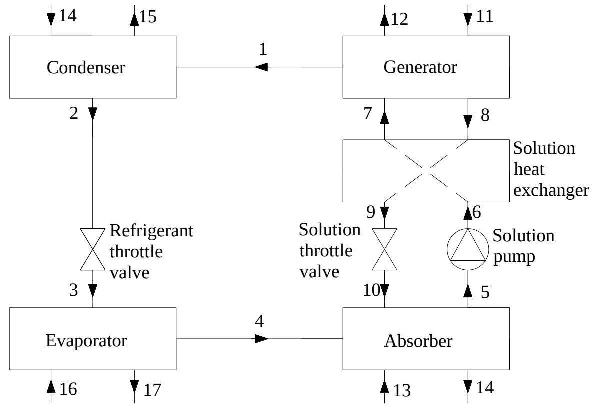

Figure 1. The schematics of the VAR system

The generator, absorber, evaporator, condenser and solution heat exchanger are designed by using heat and mass transfer calculations. The heat transfer capacity of the main components of VAR system is calculated by as follows (Genceli, 1999);

$$\dot{Q} = U \cdot A \cdot \Delta T_m \tag{1}$$

The overall heat transfer coefficient can be determined as shown below;

$$\frac{1}{U \cdot A} = \sum R_{conv} + \sum R_f + \sum R_{cond} \tag{2}$$

where thermal resistance is represented by R.

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} \tag{3}$$

where;

$$\Delta T_1 = T_{ho} - T_{ci} \tag{4}$$

$$\Delta T_2 = T_{hi} - T_{co} \tag{5}$$

The following assumptions have been made in order to develop the mathematical models for the VAR system analysis (Kilic and Kaynakli, 2007; Şencan and Yakut, 2005; Kaynakli et al., 2015; Dinçer and Rozen, 2007):

1. The temperatures of the VAR system components are constant.
2. Refrigerant leaving the condenser is saturated water at condenser pressure and refrigerant leaving the evaporator is saturated vapor at evaporator pressure.

3. The generator and condenser pressure are equal to the pressure corresponding the refrigerant saturation pressure at the condenser temperature and the evaporator and absorber pressure are equal to the pressure corresponding the refrigerant saturation pressure at the evaporator.
4. The temperature and pressure of superheated vapor leaving the generator is equal to the temperature and pressure of the generator.
5. All the system components are at steady state conditions.
6. Pressure drop in the heat exchangers and piping systems and heat loses and gains in various components and piping system are negligible.
7. Reference environment temperature and pressure are taken as 25ºC and 101.325kPa, respectively. The reference environment enthalpy and entropy used for calculating the exergy of the working fluid are the values for water at an environment temperature and pressure of 25ºC and 101.325kPa, respectively.
8. The kinetic, chemical and potential exergy of all streams at the VAR system are neglected. Chemical exergy of flue gases is considered.
9. The solution pump efficiency is taken as 0.9.

The thermodynamic analysis of an absorption system involves the application of principles of mass conservation, energy and exergy analysis to individual components of the system. The exergy analysis method provides us exergy destruction and exergy efficiency which is an important thermodynamic property. The exergy is represented by "ex" which is defined as the maximum work potential of a matter or a form of energy with respect to its reference environment (Kilic and Kaynakli, 2007).

Mass balance equations are as follows:

$$\dot{m}_7 = \dot{m}_1 + \dot{m}_8 \tag{6}$$

$$\dot{m}_7 \cdot X_7 = \dot{m}_8 \cdot X_8 \tag{7}$$

General energy and exergy balance equations are as follows (Dinçer and Rozen, 2007):

$$E_{in} = E_{out} \tag{8}$$

$$\dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_{dest} \tag{9}$$

$$\dot{E}x = \dot{m} \cdot (ex + ex_{ch}) \tag{10}$$

$$ex = h - h_0 - T_0(s - s_0) \tag{11}$$

$$f = \frac{\dot{m}_s}{\dot{m}_r} = \frac{X_7}{X_8 - X_7} \tag{12}$$

Heat capacity and the exergy destructions obtained from exergy balances of each components of VAR system illustrated in Fig. 1 can be expressed as follows:

Condenser:

$$q_{CO} = \frac{\dot{Q}_{CO}}{\dot{m}_1} = h_2 - h_1 \tag{13}$$

$$\dot{E}x_{dest} = \dot{m}_1 \cdot (ex_1 - ex_2) + \dot{m}_{14} \cdot (ex_{14} - ex_{15}) \tag{14}$$

Evaporator:

$$q_{EV} = \frac{\dot{Q}_{EV}}{\dot{m}_1} = h_4 - h_3 \tag{15}$$

$$\dot{E}x_{dest} = \dot{m}_3 \cdot (ex_3 - ex_4) + \dot{m}_{16} \cdot (ex_{16} - ex_{17}) \tag{16}$$

Generator:

$$q_{GE} = \frac{\dot{Q}_{GE}}{\dot{m}_1} = h_1 + f \cdot h_8 - (f + 1) \cdot h_7 \tag{17}$$

$$\dot{E}x_{dest} = \dot{m}_{11} \cdot (ex_{11} - ex_{12}) + \dot{m}_7 \cdot ex_7 - \dot{m}_8 \cdot ex_8 - \dot{m}_1 \cdot ex_1 \tag{18}$$

Absorber:

$$q_{AB} = \frac{\dot{Q}_{AB}}{\dot{m}_1} = (f + 1) \cdot h_5 - h_4 - f \cdot h_{10} \tag{19}$$

$$\dot{E}x_{dest} = \dot{m}_{14} \cdot (ex_{13} - ex_{14}) + \dot{m}_4 \cdot ex_4 + \dot{m}_{10} \cdot ex_{10} - \dot{m}_5 \cdot ex_5 \tag{20}$$

Solution heat exchanger:

$$q_{she} = \frac{\dot{Q}_{she}}{\dot{m}_1} = f \cdot (h_8 - h_9) = (f + 1) \cdot (h_7 - h_6) \tag{21}$$

$$\dot{E}x_{dest} = \dot{m}_8 \cdot (ex_8 - ex_9) + \dot{m}_5 \cdot (ex_6 - ex_7) \tag{22}$$

Throttle valve between evaporator and condenser:

$$\dot{E}x_{dest} = \dot{m}_1 \cdot T_0 \cdot (s_3 - s_2) \tag{23}$$

Solution pump:

$$\dot{W}_p = \dot{m}_5 \cdot (h_6 - h_5) = \frac{\dot{m}_5 \cdot \nu_s (P_{CO} - P_{EV})}{\eta_p} \tag{24}$$

$$\dot{E}x_{dest} = \dot{m}_5 \cdot (ex_5 - ex_6) + \dot{W}_p \tag{25}$$

Total exergy destruction can be expressed follows (Dinçer and Rozen, 2007):

$$\dot{E}x_{dest,tot} = \sum_{j=1}^{N} Ex_{dest,j} \tag{26}$$

where; N is the number of VAR system components and j is jth component.

The exergy efficiency can be expressed as (Dinçer and Rozen, 2007):

$$\eta = \frac{Ex_{16} - Ex_{17}}{Ex_{11} - Ex_{12} + W_p} \tag{27}$$

The coefficient of performance (COP) is a measure of a cycle's ability to transfer heat between various temperature levels (Horuz, 1998)

$$COP_{VAR} = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE} + W_p} \tag{28}$$

Table 1. Assumptions and calculated values of the VAR system

States	Substance	T(ºC)	X(%)	$\dot{m}$ (kg/s)	h(kJ/kg)	s(kJ/kgK)
1	Superheated vapor	90	-	0.0384	2668.27	8.536
2	Saturated water	40	-	0.0384	167.50	0.572
3	Water-vapor	4	-	0.0384	167.50	0.604
4	Saturated vapor	4	-	0.0384	2507.87	9.05
5	Water-LiBr	40	58.43	0.6517	107.80	0.237
6	Water-LiBr	40	58.43	0.6517	107.80	0.237
7	Water-LiBr	61.8	58.43	0.6517	150.86	0.367
8	Water-LiBr	90	62.09	0.6132	221.21	0.494
9	Water-LiBr	65.54	62.09	0.6232	175.45	0.365
10	Water-LiBr	-	62.09	0.6232	-	-
11	Flue gases	270	-	3.1887	-622.02	7.686
12	Flue gases	230.7	-	3.1887	-665.91	7.602
13	Water	29.63	-	7.4661	124.21	0.431
14	Water	33.91	-	7.4661	142.14	0.490
15	Water	37	-	7.4661	155.04	0.531
16	Water	12	-	4.2971	50.45	0.180
17	Water	7	-	4.2971	29.51	0.106

AN INDUSTRIAL APPLICATION

The aim of this section is to apply VAR system to produce chilled water that will be sent to the office building fan coil system by utilizing flue gases from an industrial furnace of an industrial company. Flue gases produced by the natural gas fired industrial furnace is normally not being utilized in this company. The flue gases volume flow rate was measured to be 5.029 m^3/s at 270ºC. To determine the substances in the flue gas, measurements were carried out. It was determined that the flue gas contains 6% H_2O, 2.74% CO_2, and 16.2% O_2. The remaining amount of flue gas content was assumed to be N_2. Normally, the office building does not have any air conditioning system and the company needs their office building to be air conditioned. The office building's comfort cooling load is calculated to be 85 kW. This section aims to investigate an application of the air conditioning system which includes the VAR and fan coil systems. So the necessity of comfort cooling will be met by utilizing the currently available waste heat. In order to do this, the application of the system is analyzed and designed.

The Energy And Exergy Analysis of The Industrial Application

In order to utilize the waste heat at the industrial company, the VAR system's parameters to be applied are needed to be determined. Firstly, the decision is made about which solution will be used. When the manufacturer catalogs of absorption chillers are analyzed, it can be seen that almost all the manufacturers use water-LiBr solution but only a few of them use ammonia-water solution. Additionally it is known that the COP performance of water-LiBr solution is better than the ammonia-water solution. Both have disadvantages and advantages. The VAR system with water-lithium bromide solutions has higher performance than the VAR system using ammonia-water system. While the VAR system with the water-lithium bromide solution is suitable for air-conditioning applications, the VAR system with the ammonia-water solution is suitable for industrial applications requiring low evaporating temperatures in evaporator. It should be considered that the VAR system using water-LiBr has crystallization risk and limitations on operating in very low temperatures because of water as being the refrigerant (Horuz, 1998). For the reasons mentioned above the VAR system using water-LiBr solution is used to air condition the office building.

The assumptions and calculated values for the VAR system to be applied were summarized in Table 1. Entropy values of the solution are obtained from Kaita (2001).

Table 2. The calculated data of the VAR system

Descriptions	Unit	System
Q_{AB}	kW	133.78
Q_{CO}	kW	96.16
Q_{GE}	kW	139.95
Q_{EV}	kW	90
$Q_{she.}$	kW	28.06
W_p	kW	0.0028
COP_{VAR}		0.64
$Ex_{dest,AB}$	kW	6.35
$Ex_{dest,CO}$	kW	1.59
$Ex_{dest,GE}$	kW	37.19
$Ex_{dest,EV}$	kW	1.89
$Ex_{dest,she}$	kW	1.63
$Ex_{dest,p}$	kW	0.0025
$Ex_{dest,rtv}$	kW	0.37
$Ex_{dest,tot}$	kW	49.04
η_{VAR}		0.082

As it can be seen from Table 2, the COP of the VAR system is 0.64 and exergy efficiency is 8.2%. Because of the large heat capacity and the temperature differences between streams, the generator has the highest exergy destruction. The next biggest exergy destruction occurred at the absorber. The pump has the lowest exergy destruction. Other components exergy destruction values are quite close. Because the exergy destruction value of the generator has the highest value in the total exergy destruction, the special attention is given to decrease it. Fig 2-7 are prepared by the values shown at Table 1.

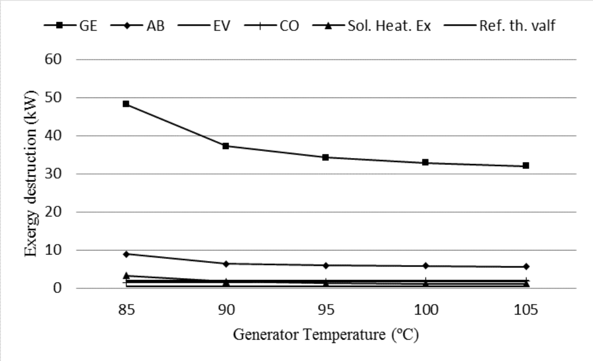

Figure 2. The exergy destruction at the components of VAR system with increasing generator temperature

Fig. 2 shows the variations of exergy destruction of VAR system components against increasing generator temperature. The generator which has the highest exergy destruction and the absorber which has the second highest exergy destruction values decreases with increasing generator temperature. The slope of decrease of the exergy destruction of the generator sharply decreases at 90°C. The highest exergy destruction value at the generator is 48.16 kW at 85°C, the lowest exergy destruction value at the generator is 32.06 kW at 105°C. The same trend is also valid for the absorber. The highest exergy destruction value at the absorber is 8.92 kW at 85°C, the lowest exergy destruction value at the absorber is 5.65 kW at 105°C. The other components of VAR system doesn't effects the exergy efficiency and exergy destruction compared to the generator and absorber because of the lower exergy destruction.

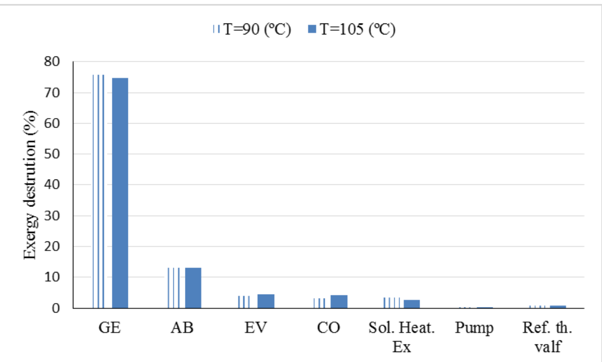

Figure 3. The relative exergy destructions of the components of the VAR system

As seen at Fig. 3 the relative exergy destruction at the generator decreases from 75.83% at T_{GE}=90°C to 74.65% at T_{GE}=105°C. The relative exergy destruction at absorber increases from 12.95% at T_{GE}=90°C to 13.95% at T_{GE}=105°C. When relative exergy destruction at the condenser and evaporator having close values to each other is approximately 3.5% at T_{GE}=90°C, it reaches approximately 4.3% at T_{GE}=105°C. When the solution heat exchanger has 3.33% relative exergy destruction at T_{GE}=90°C, it has 2.67% relative exergy destruction at T_{GE}=105°C. Refrigerant throttle valve has second lowest value like 0.86% at T_{GE}=105°C. And the pump has relatively very low exergy destruction and it can be neglected. The bigger temperature difference between the source and the generator causes the higher relative exergy destruction at the generator. When the generator temperature increases, the relative exergy destruction at the generator decreases because the streams temperature differences and the heat capacity decreases so relative exergy destruction decreases. The mixing of the rich solution coming from the heat exchanger and refrigerant from the evaporator at different temperature at the absorber causes the second highest relative exergy destruction. As increasing the generator temperature, the temperature of rich solution coming from the heat exchanger increases so the temperature difference at the mixing process increases. This causes an increase on relative exergy destruction at the absorber.

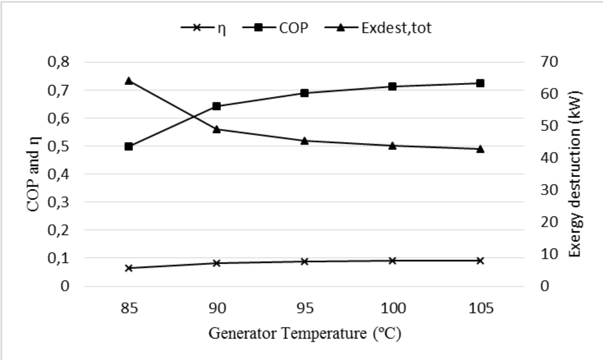

Figure 4. The COP, η and exergy destruction changes with increasing generator temperature

Fig.4 shows the changes of COP, exergy efficiency and exergy destruction versus the generator temperature. It can be observed that COP increases with increasing generator temperature. While the value of COP is 0.49 at T_{GE}=85°C, it reaches to 0.72 at T_{GE}=105°C. The exergy efficiency of VAR system increases with increasing generator temperature. While the value of exergy efficiency is 0.064 at T_{GE}=85°C, it reaches to 0.091 at T_{GE}=105°C. And also it is observed that the total exergy destruction decreases with increasing generator temperature.

Table 3. The VAR system components design parameters

Descriptions	Condenser	Evaporator	Absorber	Generator
Inside of the tube diameter(mm)	17.5	9.55	17.55	17.08
Outside of the tube diameter(mm)	19.05	12.5	19.05	21.3
Horizontal tube number	11	16	10	
Vertical tube number	-	20	19	
Tubes array	-	2x8x20	10x19	3x23
Total number of tubes	104	320	190	69
Tubes length(mm)	2200	1510	1504	1000
Number of pass	2	4	2	1
Pressure(kPa)	7.38	0.81	0.81	7.38
Temperature (ºC)	40	4	40	90
Number of fins	-	-	-	251
Fin height(mm)	-	-	-	10
Fin thickness(mm)	-	-	-	0.5
Fins area(m^2)	-	-	-	46.74
U(W/m^2K)	1622.09	952.44	604.5	17.12

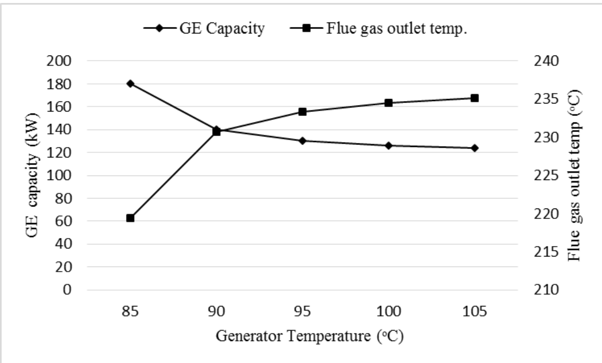

Figure 5. The generator capacity and flue gas outlet temperature variations with increasing generator temperature

Fig. 5 shows us the changes of the flue gas outlet temperature and the generator temperature with increasing the generator temperature. Initially, the generator capacity of the VAR system tends to decrease with increasing generator temperature. It varies between 124 and 180 kW. While the highest capacity of the generator of the VAR system is 180 kW at 85ºC, the lowest capacity of the generator of the VAR system is 124 kW at 105ºC. Also the slope of decrease of the capacity of the generator sharply decreases at 90ºC. The flue gas outlet temperature tends to increase with increasing generator temperature. It varies between 219.36 and 235.18ºC. While the highest flue gas outlet temperature is 235.18ºC for generator temperature at 105ºC, the lowest flue gas outlet temperature is 219.36ºC for generator temperature at 85ºC. The reason of both decrease the generator capacity and increase of the flue gas outlet temperature is the increasing COP of VAR system. This reason shows that the potential energy utilization of waste heat increases with increasing generator temperature.

Fig. 6 shows the changes of total exergy destruction and exergy efficiency with the difference between flue gas inlet temperature and generator temperature which is an important parameter. Total exergy destruction tends to increase with an increase in difference between flue gas inlet temperature and generator temperature. While the highest exergy destruction of the VAR system is 64.07 kW at 185ºC, the lowest exergy destruction of the VAR system is 42.95 kW at 165ºC. The slope of the exergy destruction curve sharply increases when the difference between flue gas inlet temperature and generator temperature reaches at 180ºC. Exergy efficiency of the VAR system tends to decrease with an increase in the difference between flue gas inlet temperature and generator temperature. While the highest exergy efficiency of the VAR system is 0.092 at 165ºC, the lowest exergy efficiency of the VAR system is 0.064 at 185ºC. These changes of total exergy destruction and exergy efficiency show how important parameter the difference between flue gas inlet temperature and generator is.

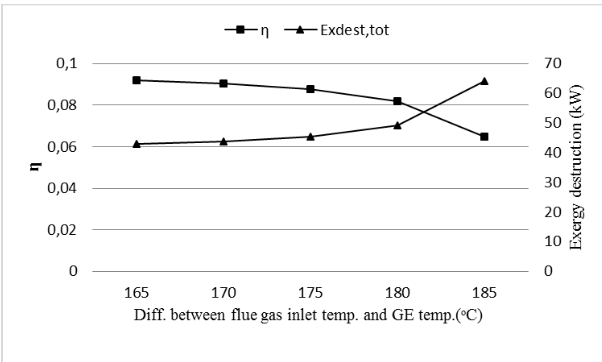

Figure 6. The effect of the difference between flue gas inlet temperature and generator temperature

Fig. 7 shows the exergy efficiency changes with increasing reference environmental temperature. It can be seen that the exergy efficiency increases with increasing reference environment temperature. For T_{GE}=90ºC, the exergy efficiency increases from 0.082 at T_{ref}=25 ºC to 0.122 at T_{ref}=32ºC. And for T_{GE}=105 ºC, the exergy efficiency increases from 0.091 at T_{ref}=25 ºC to 0.137 at T_{ref}=32 ºC. The results at Fig. 7 shows that potential energy to utilize decreases with increasing reference environment temperature and shows that the potential energy is used better at T_{GE}=105ºC comparing to T_{GE}=90ºC.

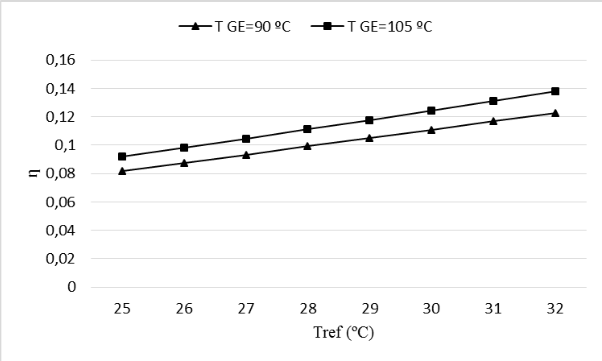

Figure 7. The variation of exergy efficiency with increasing Reference environmental temperature

The VAR System Components Design and Construction

This section aims to show how the VAR system components are designed and constructed. It is expected that the equations at the Appendix and detailed knowledge presented at this section could be a guide for researchers and engineers. To design the components of the VAR system, the equations used to calculate heat and mass transfer coefficient can be found at Appendix (Genceli, 1999; Florides et al.,2003; Seewald and Blanco,1994; Davies and Rideal,1961; Thome, 2009; Cosenza and Vliet, 1990; Lorenz and Yung, 1979; Li et al., 2011; Ribatski and Thome, 2007; Çengel,2002; Wang et al., 2012). All of the VAR system components design parameters are shown at Table 3.

Absorber is chosen to be a shell and tube heat exchanger. In the absorber, the LiBr solution is dripped over the horizontal tubes cooled by water flowing inside by using dripping tray. It drags and absorbs the water vapor coming from the evaporator and flows in a thin film around tubes. And then it is collected at the bottom of the lower shell. The absorber also has feedback system. When the water vapor isn't absorbed by rich solution, a feedback pump start to work and pumps the solution to the dripping tray. So, the more solution drips the more water vapor is dragged and absorbed. When the water vapor is absorbed completely, the solution pump pumps the solution to the solution heat exchanger. These two pumps located at the bottom of the shell are controlled by the control systems. The tubes in the absorber must be resistible to corrosion since water-LiBr solution has corrosion effect. So, CuNi10 is preferred as the tubes material in the absorber. And also the parts of shell which water-LiBr solution contacts with is preferred 316 L stainless steel as the shell material. To design the absorber, the equations used to calculate heat and mass transfer coefficient can be found in Appendix. The thermophysical properties water-LiBr solution was obtained from Florides et al. (2003). Mass transfer must also be considered. A practical model for absorption of vapors into a laminar film of water and LiBr falling along a constant temperature vertical plate was described. The details of mass transfer coefficient calculation can be found from Florides et al. (2003).

Evaporator is also chosen to be shell and tube heat exchanger. Absorber tubes and evaporator tubes positioned to the same shell. In the evaporator, the saturated water coming from the condenser is dripped over the horizontal tubes. When it flows in a thin film around tubes, it evaporates by rejecting heat from the water flowing inside. Falling film evaporators have dry area problem. When the fluid flows in a thin film around the tubes, the tubes area may not be get wet by the fluid completely. Because of that the heat transfer coefficient may decreases sharply. The refrigerant that couldn't evaporate is collected in the bottom of the lower shell because of the dry area problem. The evaporator also has a feedback system. When the refrigerant accumulates at the bottom of lower shell enough, a feedback pump start to work and pumps the refrigerant to the dripping tray. So, more refrigerant drips and less dry area may occur. When the water evaporates completely, the feedback pump stops. The feedback pump located at the bottom of the shell is controlled by the control systems. Copper is used as tubes material.

Condenser is a water cooled shell and tube heat exchanger. In the condenser, the water vapor coming from generator is condensing over the horizontal tubes cooled by water flowing inside. The cooling water will enter the condenser after it exits from the absorber. Copper is used as tubes material.

In the generator, when the water-LiBr solution flows inside the vertical tubes, the flue gas flows outside the vertical tubes. The tubes are mounted between bottom reservoir and top reservoir. The bottom reservoir is used for collecting the water-LiBr solution coming from the solution heat exchanger and for providing the distribution to tubes properly. The top reservoir is used to send the removed water vapor from the water-LiBr solution to the condenser and to send the water-LiBr solution to the solution heat exchanger. And by mounting the fins to the tubes heat transfer area is increased, so more compact construction is obtained. To protect the corrosion effect of the water-LiBr solution, 316 L stainless steel is used as the generator's tubes material. And the fin's material is selected to be aluminum. The inside heat transfer coefficient is so high relative to the outside heat transfer coefficient because of the boiling, so it is not necessary to calculate it.

When designing the solution heat exchanger, the double-pipe heat exchanger is used. To overcome the corrosion effect of the water-LiBr solution, 316 L stainless steel is used as material. When the solution heat exchanger is designed ε-NTU method was used (Çengel, 2002). The overall heat transfer coefficient is found to be 490 W/m^{2}K.

Table 4. The systems cost parameters

	P_c (Euro)	EA_c (Euro)	Annual operation cost (Euro)	EAC (Euro)
The fan coil system and the VAR system to be manufactured (1)	49300	5790	233	6023
The fan coil system and the VAR system to be bought (2)	80200	9420	361	9781
The fan coil system and the VCR system to be bought (3)	36300	4269	3108	7377
VRF air conditioning system to be bought (4)	22507	2643	1245	3888

COST ANALYSIS

This section aims to investigate and compare the cost analysis of the system which includes (1) the VAR system to be manufactured and the fan-coil system, (2) the VAR system to be bought and fan-coil system, (3) the vapor compression chiller to be bought and fan-coil system and (4) the Variable Refrigerant Flow (VRF) air conditioning system. During the cost analysis, life cycle cost technique which includes all cost factors as initial cost, operating and maintenance costs. To apply this technique, equivalent annual method is chosen. At this method all the costs taking place over a period are converted to an equivalent uniform yearly amount. The equations below are being used (Elsafty and Al-Daini, 2002). The cost analysis parameters are shown at Table 4. In the cost analysis, it is assumed that the air conditioning systems work 10 hours a day for 120 days which refers to cooling season under full load.

$$EA_c = P_c \frac{(1+i)^n \cdot i}{(1+i)^n - 1} \tag{29}$$

$$EAC = EA_c + \text{annual operating cost} \tag{30}$$

Assumptions are i: 10%, n: 20 year, Electric price: 0.0576 Euro/kWh.

To make the cost analysis, the VAR system to be bought has been chosen to be the single effect hot water driven VAR system because the commercial flue gas fired VAR system which provides the desired cooling capacity is not available in the market on this size. So, the hot water will be obtained by using the heat exchanger mounted to the chimney and run the hot water driven VAR system. The VAR system model HWAR-L 30HH is chosen from World Energy Absorption Chillers Europe Ltd. The VCR system to be bought has been chosen to be air cooled chiller. The VCR system is domestic production by Frigotek and the model is FMC-36. VRF air-conditioning system to be bought is chosen from Samsung. The model of outer unit of VRF air conditioning system is AM260FXVAGH and the inner units of VRF air conditioning system is chosen appropriately from Neo Forte type. The fan coil system has been chosen from domestic productions called Untes. Additionally, the VAR system is planned to be manufactured in Turkey.

As can be seen from the cost analysis, the initial cost of the fan coil system and the VAR system to be manufactured (1) is lower than the initial cost of the fan coil system and the VAR system to be bought (2) because it (2) is not manufactured in Turkey. The system (1) has the lowest operation cost and is third as far as the initial cost is concerned. The system (2) which has the highest initial cost compared with the other systems has quite low operation costs compared to the systems (3) and (4) but not as low as the system (1). The system (3) which has the highest operation cost compared to the other systems has lower initial cost than the systems including the VAR systems. The system (4) which has lowest initial cost compared with the other systems has quite high operation and maintenance costs compared to the systems (1) and (2). Because of the fan coil units and piping system costs, the system (3) has higher initial costs than the system (4). Although the system (1) does not seem to be the most advantageous system, it comes front by utilizing the waste heat and by decreasing the operation costs.

CONCLUSIONS

This study aims to present the investigation of utilization of the waste heat from the chosen industrial company by using VAR systems including energy and exergy analysis and comparing VAR systems with alternative systems.

It is obtained that the flue gas fired VAR system's COP is 64%, and exergy efficiency is 8.2%. The parameters that effect the VAR system are investigated. The COP of the VAR system increases with increasing generator temperature. It increases about 23% with an increase at generator temperature around 20ºC. The highest exergy destruction is found to be at the generator about 75% of the total exergy destruction because of the temperature difference between the inlet and outlet stream properties. And it decreases with increasing generator temperature around 33.4% with the difference of 20ºC at generator temperature. The exergy efficiency increases and total exergy destruction decreases with a decrease at the difference between the source and the generator temperature. Total increase at exergy efficiency is around 2.73% and total decreases at exergy destruction is about 21.13kW with a decrease at the difference of 20ºC. The effect of reference environmental temperature is also investigated. The exergy efficiency of VAR system increases by increasing reference environmental temperature. When at generator temperature 90ºC, exergy efficiency increases 4.1% with an increase at the difference of 7ºC, at generator temperature 105ºC, exergy efficiency increases 4.6% with an increase at the difference of 7ºC.

It has to be noted that from the cost analysis, the fan coil system and the VAR system to be bought has the highest initial cost like approximately three times higher than the VRF air conditioning system. The fan coil system and the VAR system to be manufactured has the lowest operation cost like approximately thirteen times lower than the fan coil system and the VCR system to be bought. As expected, VAR systems has lower operation cost than VCR and VRF air conditioning systems and VAR systems has higher initial cost than VCR and VRF air conditioning systems. The most advantageous systems are the VRF air conditioning system and the fan coil system and VAR system to be manufactured from the total annual cost. Although the VRF air conditioning system has lower total annual cost, the fan coil system and the VAR system to be manufactured can be more advantageous if the VAR system is manufactured domestically as mass production, also come front by utilizing the waste heat and by decreasing the operation and maintenance costs. Additionally, the other systems use fossil fuels to drive. As known, when the VRF air conditioning system is preferred, the carbon emissions will be released to the atmosphere and it will cause the green house affect that causes global climate change. And also the fluids as CFCs that is being used in the VRF air conditioning system may cause the ozone depletion. In addition to these disadvantages, utilizing the waste energy is so important because the countries like Turkey that do not have enough fossil fuel sources to produce their own energy. But, by preferring VAR systems utilizing waste heat, the total efficiency of industrial furnace integrated to the VAR system will be increased, thermal pollution can be decreased and also total cost of comfort cooling of the office buildings will be minimized.

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APPENDIX

The inside heat transfer coefficient for tubes flowing water can be determined as follows [12]

$$Nu = 0.023 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.4} \tag{A1}$$

At the absorber the total area of one tube can be estimated as [16, 17]

$$A_{eff} = 2 \cdot l \cdot n \cdot \pi \cdot r_0 \tag{A2}$$

$$l = \left(\frac{3 \cdot \rho_s}{32 \cdot g}\right)^{\frac{1}{4}} \cdot \left(\frac{n}{\dot{m}_s \cdot \mu_s}\right)^{\frac{1}{2}} \cdot (\pi \cdot r_0 \cdot a \cdot \sigma \cdot \cos\Theta)^{\frac{3}{4}} \tag{A3}$$

For absorber, the wavelength can be determined as follows [18]:

$$\lambda = 2 \cdot \pi \cdot \sqrt{\frac{n \cdot \sigma}{\rho \cdot g}} \tag{A4}$$

At the absorber the outside convective heat transfer coefficient can be determined as follows [19];

$$Nu = \mathrm{Re}^{0.46} \tag{A5}$$

When the outside heat transfer coefficient is determined, a model for falling film evaporation on horizontal smooth tubes can be used [20].

$$h_o = h_b + h_d \cdot \left(\frac{L_d}{L}\right) + h_c \cdot \left(1 - \frac{L_d}{L}\right) \tag{A6}$$

Falling film evaporation in a vacuum, the nucleate boiling term can be eliminated [21].
And for wet areas falling film evaporation Nu number can be determined as follows;

$$Nu = 182.1 \cdot \mathrm{Re}^{-1.56} \tag{A7}$$

At falling film evaporation, for dry areas a coefficient is determined as follows [22];

$$F = 0.0024 \cdot \mathrm{Re}^{0.91} \tag{A8}$$

For falling film evaporation, the outside convective heat transfer coefficient for developed region is determined by multiplying convective heat transfer coefficient and dry area coefficient as follows;

$$h_c = F \cdot h_w \qquad \text{(A9)}$$

The heat transfer for condensation on the outside surface of a horizontal tube gives the outside heat transfer coefficient [23].

$$h_{horiz} = 0.729 \cdot \left[\frac{g \cdot \rho_l \cdot (\rho_l - \rho_v) \cdot h^*_{fg} \cdot k_l{}^3}{\mu_l \cdot (T_{sat} - T_l) \cdot d_o} \right]^{\frac{1}{4}} \qquad \text{(A10)}$$

$$h_{horiz,Ntubes} = \frac{1}{N^{\frac{1}{4}}} \cdot h_{horiz} \qquad \text{(A11)}$$

At the generator, to calculate the outside heat transfer coefficient, the correlation below is used [24]:

$$h_o = 0.134 \cdot \left(\frac{k}{d_e} \right) \cdot \left(\mathrm{Re}^{0.681} \right) \cdot \left(\mathrm{Pr}^{\frac{1}{3}} \right) \cdot \left(\frac{s}{14.85} \right)^{0.2} \cdot \left(\frac{s}{t_f} \right)^{0.1134} \qquad \text{(A12)}$$