

Technical Paper

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Technical Paper #3

Energy Savings in Ammonia Systems Using Low-Stage De-superheating

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Abstract

This paper demonstrates how evaporative condensers can be used as low-stage de-superheaters to improve the efficiency of two-stage ammonia refrigeration systems. The high discharge temperatures generated by ammonia compressors can help reject heat to ambient heat sinks, thereby reducing the load on the high-stage compressors, which in turn reduces the required compressor motor power and provides energy savings for the user.

Introduction

In a global environment where energy demands and costs are constantly increasing, engineers are continuously motivated to design systems with improved energy efficiency. In the past, payback periods were the main driver for the implementation of efficiency improvements, but pressure on industries to reduce their carbon footprint (legislative and social) has accelerated these discussions.

This paper demonstrates how low-stage de-superheating in two-stage ammonia systems can help reduce the peak power consumption of a plant, using established technologies. In a simple two-stage plant used for low-temperature applications, as illustrated in Figure 1, approximately 10%–13% of the induced load from the low-stage load is in the form of superheat. This is based on typical ammonia open-drive reciprocating compressor performance.



Figure 1. A simple two-stage plant used for low-temperature applications.

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The pressure-enthalpy diagram below indicates the energy contained in superheated vapor between points 2 and 3 in the plant's operating cycle. Although it is not possible to reject all this energy to ambient heat sinks because the operating temperatures are often above the saturation temperature of the intermediate condition, we considered several recovery options.



Concept Review

This study aims to evaluate the most practical alternative for reducing peak power consumption through low-stage de-superheating. The evaluation is conducted for a plant situated in a location with moderate to high ambient wet-bulb and dry-bulb temperatures. We exclude the evaluation of heat sinks, such as lakes and underground water reservoirs, because these heat sinks are not available to all plants. The evaluation is also limited to ambient air conditions (dry-bulb and wet-bulb temperatures).

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The first option includes a thermosiphon de-superheater inside the machinery room, installed in the low-stage compressor discharge line, as shown in Figure 2. The heat rejection is limited by the plant's peak condensing temperature, 35°C (95°F) in this case.



Figure 2. Two-stage refrigeration system with shell-and-tube low-stage de-superheater.

The next option is to install an air-cooled radiator outside the machinery room, as shown in Figure 3. Initial selections of the air-cooled radiator indicate that the additional fan power offsets a significant percentage of the potential savings. It is also difficult to de-superheat the outlet gas temperatures to less than 10°C above the ambient dry-bulb temperature. This option was not considered further because of the high ambient dry-bulb temperature of 36°C (96.8°F).

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Figure 3. Two-stage refrigeration system with air-cooled low-stage de-superheater.

Both options may have benefits when overall power consumption is considered for plants operating in climates with low temperatures, but they are not considered further for this analysis. The option selected for evaluation is shown in Figure 4. Specifically, the evaporative condenser coil is split into two sections: a large section for normal condensing and a smaller section for de-superheating of the low-stage discharge vapor. This option is preferred because the ambient peak wet-bulb temperature is 22°C (71.6°F), and thus the de-superheating potential is increased compared with the alternative options. Furthermore, evaporative condensers can reject large amounts of energy while consuming low input power.



Figure 4. Two-stage refrigeration system with a low-stage de-superheater coil inside the evaporative condenser.

Preliminary Calculations, Assumptions, and Equipment Selections

Plant detail and environmental conditions

The plant has a low-stage load of 400 kW (113.8 TR) and requires a low-stage suction temperature of -40° C (-40° F) at the compressor. The plant also has a small high-stage load, but this was not considered as part of this initial evaluation because we wanted to evaluate the reduction of the induced load. The saturated intermediate temperature is set to -5° C (23° F).

The ambient wet-bulb temperature is 22°C (71.6°F), and the ambient dry-bulb temperature is 36°C (96.8°F). The peak condensing temperature is 35°C (95°F).

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The plant is situated in East London, South Africa (33.0198° S, 27.9039° E). All compressors and condensers are selected to operate using variable-speed drives.

Standard equipment selections

The original plant is configured as shown in Figure 1. Two booster compressors are selected, and the specifications are shown in Figure 5. The compressor speed is adjusted to show the exact expected operating point.

	Mecha	nical	Oil Man	agement l	Motor	Cont	rols	Documentati	on Comment	ts
Compresso	r									
Model	V 14	00	*	Refrigerant	R-717				X	
Drive Type	Direc	ct Drive	~					. ~		
Frequency	50Hz	z	÷					100		511
Speed Typ	e Varia	ible	*					A COL		
mbiant Co	nditions	Condit	ion 1 🔺	1						
molent co	nuncions	Contait	1011 1							
				10						
Evap Ter	nperature	12	-40	°C (Cond Temp	erature	-5	•c		
Evap Ter Evap Pre	mperature	N	-40	C (Cond Temp Cond Press	erature ure	-5 3.5	°C	(a)	
Evap Ter Evap Pre Suction	nperature essure Line Loss		-40 0.719 0	℃ (bar(a) (bar [Cond Temp Cond Pressi Discharge L	erature ure ine Loss	-5 3.5 0	°C 55 bar bar	(a)	
Evap Ter Evap Pre Suction Superhe	nperature essure Line Loss at (Non-U	Jseful)	-40 0.719 0 1	5C (bar(a) (bar [K 5	Cond Temp Cond Pressi Discharge L Subcooling	erature ure ine Loss At Conc	-5 3.5 0	°C 55 bar bar	(a)	
Evap Ter Evap Pre Suction Superhe Superhe	mperature essure Line Loss at (Non-U at (Useful	Jseful)	-40 0.719 0 1 0	*C (bar(a) (bar [K 5 K 5	Cond Temp Cond Press Discharge L Subcooling Speed	erature ure ine Loss At Conc	-5 3.5 0 1 0	°C 555 bar bar K 43 RPM	(a) /	
Evap Ter Evap Pre Suction Superhe Superhe	mperature essure Line Loss at (Non-U at (Useful	Jseful)) ET	-40 0.719 0 1 0 CT Refr	*C (bar(a) (bar [K S K S	Cond Temp Cond Press Discharge L Subcooling Speed city Power	erature ure ine Loss At Conc Speed	-5 3.5 0 1 0 11.	°C 55 ban bar K 43 RPM Mass Flow Rate	(a) A Suct. Vol. Flow	Disch. Vol. Flow
Evap Ter Evap Pre Suction Superhe Superhe	nperature essure Line Loss at (Non-U at (Useful Condition	Jseful)) ET °C	-40 0.719 0 1 0 CT Refr	"C () bar(a) () bar [) K S K S igerating Capace kW	Cond Temp Cond Press Discharge L Subcooling Speed city Power kW	erature ure ine Loss At Conc Speed RPM	-5 3.5 0 110 111	°C 55 bar bar K 43 RPM Mass Flow Rate kg/h	(a) A Suct. Vol. Flow m³/h	Disch. Vol. Flow m³/h

Figure 5. Specifications for the two booster compressors.

The two booster compressors have a combined induced load of 499.6 kW (142 TR). A single high-stage compressor was selected, and the specifications are shown in Figure 6. Again, we adjust the speed to indicate the exact expected operating point.

	wiechan	lical		igement im	otor	Contro	s Document	auon Commen	15	
ompress	or									
Model	V 700	HS	ų	Refrigerant I	R-717			¥	Include	ed Components
Drive Typ	e Direct	: Drive	N.							
Frequenc	y 50Hz		÷				1			
Speed Ty	pe Variab	ole	Υ.							
							A AL			
nbient Co	onditions C	Conditio	in 1 +							
		12								
Evap Te	mperature	ŀ	-5	°C Co	ond Temp	perature	35 ి	с		
Evap Te Evap Pi	emperature ressure	(- 5 3.555	°C Co bar(a) Co	ond Temp ond Press	oerature sure	35 ° 13.513 b	C Þar(a)		
Evap Te Evap Pi Suction	emperature ressure 1 Line Loss	((-5 3.555 0	°C Co bar(a) Co bar Di	ond Temp ond Press scharge I	oerature sure Line Loss	35 ° 13.513 b 0 b	C bar(a) bar		
Evap Te Evap Pr Suction Superh	emperature ressure 1 Line Loss eat (Non-Us	((eful) (-5 3.555 0 1	°C Co bar(a) Co bar Di K Su	ond Temp ond Press scharge I ibcooling	oerature sure Line Loss At Cond	35 ° 13.513 b 0 b 0 K	C var(a) var		
Evap Te Evap Pr Suction Superh Superh	emperature ressure 1 Line Loss eat (Non-Us eat (Useful)	((;eful) (-5 3.555 0 1	°C Co bar(a) Co bar Di K Su K Sp	ond Temp ond Press scharge I ubcooling peed	oerature sure Line Loss At Cond	35 ° 13.513 b 0 b 0 k 1306 R	C xar(a) xar C		
Evap Te Evap P Suction Superh Superh	emperature ressure 1 Line Loss eat (Non-Us eat (Useful) Condition	((ieful) (ET C	-5 3.555 0 1 0	C Cc bar(a) Cc bar Di K Su K Sp erating Capacity	ond Temp ond Press scharge I ubcooling beed	Sure Line Loss At Cond	35 9 13.513 b 0 b 0 k 1306 R 1306 R	C var(a) var c RPM te Suct. Vol. Flow m ² /h	Disch. Vol. Flow	

Figure 6. Specifications for the single high-stage compressor.

The total heat rejected by the high-stage compressor is 499.5 kW + 107.7 kW = 607.2 kW (172.6 TR). Based on this heat rejection and the design wet-bulb temperature, the condenser is selected with a 10% minimum safety margin. The specifications of the evaporative condenser are shown in Figure 7.

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	Conde	nser Tech	nical Data	Sheet
Product Description	E	(1) ATC	187B	Hannes Steyn 1 Savio Road Brackengate II Cape Town, 7560 ZA C. 0829243752 Hannes,Steyn@gea.com
Coloria Citaria		Provide Land		Hot Saturated Discharge Air
Selection Criteria Refrigerant: Condensing Pressure: Entering Wet Bulb:	NH3 35.0 C 1249.4 kPa 22.0 C	607.00 kW NH3 522 026 kcal/hr Entering Dry Bulb Switchover:	-39.18 C	Sol Dry Air
Unit Selected One(1) EVAPCO ATC Physical Data Per Un Overall Dimensions (WxLxt Operating Weight: Shipping Weight: Heaviest Section: *weights and dimensions of	it 187B at 115% capa it 1): 2 283mm x 2 578 4 962 kg 3 760 kg 3 171 kg ould vary depending on c	acity (698.02 kW) mm x 3 404mm options selected		
Fan Motor Data Per	Unit	•	Pump Motor Data	ner Unit
Number of Fans: # of Fan Motors: Nameplate Power (380/3/: Additional Details Pe	1 1 50): 5.50 kW Per Mote	Dr	No.of Pumps: Nameplate Power (380	1)/3/50): 2.2 kW per pump motor
Air Flow: Coil Volume: Est. Refrigerant Charge: Coil Design Pressure:	16 m³/s 0.7 m³ p 83.5 kg 20.7 Bar	er unit oer unit		
Hydraulic Data				
Spray Water Flow: Evaporated Water Rate:	2	1 LPS .21 LPS		
Accessories				
(1) 1.0 Importance Factor S	Specified (1)	Nitrogen Charged Coil(s)	()	I) Fan Motor: Inverter Capable, Premium Efficient

Figure 7. Specifications for the evaporative condenser.

The power consumption of the main machinery room equipment is summarized below in Table 1.

	Capacity	Shaft Power	Commonte
	kW (TR)	kW (BHP)	Comments
Booster Compressor 1	200.1 (56.9)	49.7 (66.6)	$-40^{\circ}C/-5^{\circ}C$
Booster Compressor 2	200.1 (56.9)	49.7 (66.6)	$-40^{\circ}C/-5^{\circ}C$
High-stage Compressor	499.7 (142.1)	107.7 (143.6)	-5°C/35°C
Evaporative Condenser	607 (172.6)	7.7 (10.32)	35°C/22°C WB
Refrigerant Pump	400 (114.7)	3 (4)	4 × circulation
Total		217.6 (291.8)	

Table 1. Power consumption of the main machinery room equipment.

Initial calculations of low-stage de-superheating

The compressor performance data for the low-stage compressor is outlined as follows (per compressor):

- Compressor capacity = 200.1 kW (56.9 TR)
- Compressor discharge temperature = 97.3°C (207.1°F)
- Mass flow through the compressor = 586.2 kg/hr (1292 lb/hr)
- High-stage induced load = 249.8 kW (71 TR)
- Compressor discharge pressure = 360 kPa (A) (52.2 psi)

Operating data, performance data, and oil management data are indicated in Figure 8.



Piston Rating

GEA Grasso V 1400 R-717

	O	PERATING D	ATA - Condition 1			
Evaporator		(Condenser			
Evap Temperature	-40	°C	Cond / Inter Temper	ature	-5	°C
Evap Pressure	0.7	bar(a)	Cond / Inter Pressur	e	3.6	bar(a)
Superheat (Useful)	0	ĸ	SubCooling At Cond		0	K
Superheat (Non-Useful)	1	ĸ	Condenser HOR	\leq	249.8	ĸW
Suction			Discharge			
Suction Line Loss	0	bar	Discharge Line Los	3	0	bar
Package Suction Loss	0.001	bar	Package Discharge	Loss	0.018	bar
Saturated Suction Pressure	0.7	bar(a)	Discharge Pressure		3.6	bar(a)
Suction Temperature	-39	°C	Discharge Temperat	ture 🤇	97.3	••
Mass Flow	586.2	kg/h	Mass Flow		586.2	kg/h
Volume Flow	911.2	m³/h	Volume Flow		291.3	m³/h
Theoretical Swept Volume	1213.7	m³/h				
		PERFORM				
Compressor			Motor			
Capacity	200,1	kW	Voltage	4(00/3/50	V/PH/Hz
Power	49.7	KW	Motor Size		71	kW
Performance Factor	4.0	Qo/Pe	Frame	3	15S/M	
Speed	1143	RPM	Enclosure		IP55	
Percent Full Load	100	%	IE Classification		IE3	
Min Part Load	25	%	Full Load Amps		139	А
Drive Type	Direct Drive		Number of Poles		6	
			Frequency Regulate	d	Yes	
		OIL MANAG	EMENT DATA			
Oil Separator Type	05	5	Oil Throw Before		1.3	cc/h
Oil Separator Diameter	492.0	mm	Oil Throw Before		1.8	ppm
Oil Separator Efficiency	95.5	%	Oil Throw After		0.4	cc/h
Ambient Temperature	40	°C	Oil Throw After		0.6	ppm
Oil Type	PR-OLEO	C-MH68A	Oil Cooler Included		No	
EU Ecodesign Regulation (EU	2019/1781): 1-July	-2023, IE4 mand	atory for 75200 kW. IE	3 motors will be suppli	ed without	CE marking.
		PROJE	CT DATA			
Project Name:			Reference:			
Customer Name: New Co	ontact		Prepared By:	Hannes Steyn		
Proposal Number:			Date:	11/9/2023		

Figure 8. The operating data, performance data, and oil management data of a single compressor.

By assuming that we can reach 10 K (18°R) between the wet-bulb temperature and de-superheated gas temperature, the reduction in the induced load is calculated using EES software, approaching 10% (Figure 9). This is a positive result because high-stage motor power is reduced by an equal amount. However, the system imposes additional pressure losses on the low-stage compressors and reduces the available condensing area.

natura viedos		11 S
97.3 (intermediate evaporation temps)		Update M
22 (ambinet Wet built temperature)		Main Program
=10 (approach temp) =14+15		h2 [kJ/kg]
tes aut Americanis ToT21 Between tists increased		n3 (kJ/8g)
-p_sat/variationse v=r.s) [international proceed		h4 (kJ/kg)
enthalpy(Ammonia,T=T2,P=P2) enthalpy(Ammonia,x=1,P=P2)	Kan	P2 (KPA)
enthalpy(Ammonia,x=0,P=P2)	Unit Settings: SI C kPa kJ mass deg	Saving% [%]
enneddylwynweiwy (- 18, P-172)	K2 = 1697 [k-10a] b1 = 1457 [k-10a] b4 = 1768 [k-10a] b5 = 1547 [k-10a] 22 = 154 9 [k-24] Sama(S = 9.875 [S]) 11 = 40 [S]	T1 (C)
ng%=0;2-h6)(0;2-h4)*100	$12 - 02^{-1} (10)$ $17 - 6^{-1} (10)$ $14 - 92^{-1} (10)$ $15 - 91^{-1} (10)$ $15 - 92^{-1} (10)$ $15 - 92^{-1} (10)$	12[C]
	are not full to a full to a full to a set full	T4 [C]
	The second se	T5 (C)
	i potenta uni protecin aus paractea.	16 [C]
ine 8 Charl 47 Wiep: On Insett Caps Lock	Calculation time = 6.5 ms	

Figure 9. Calculated reduction in the induced load.

Therefore, a trade-off exists. The more condenser circuits allocated to desuperheating, the lower the impact on the low-stage compressors and the greater the impact on the condenser. As an initial starting point, we limit the additional pressure drop to 25 kPa (3.6 psi) to minimize the impact on the power consumption of the low-stage compressor. Under these conditions, the low-stage compressor power increases by approximately 5% compared with the conventional system.

The selected condenser has 80 circuits with an overall circuit length of 18.26 m (60 ft). Using the EES Pipe-Flo tool, the number of required circuits can be estimated. The calculated result is shown in Figure 10, revealing that 10 circuits are needed for

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the low-stage de-superheating section to limit the pressure drop to approximately 25 kPa (3.6 psi). The result also indicates an internal heat transfer coefficient of 750 W/ m^2 -K.



Figure 10. Initial estimation of the required number of circuits.

If 10 circuits (12.5%) are utilized for de-superheating, then 87.5% of the surface area is available for condensing. The real heat rejection capacity of the original condenser is 698 kW (198.5 TR), and if this is reduced by 12.5%, 610 kW (173.4 TR) of condensing capacity remain. Considering that the induced load is reduced by approximately 10%, the high-stage compressor requires a condensing capacity of only 545 kW (155 TR), leaving a 12% safety margin, which is within the design requirement.

The last item to consider is the performance of the de-superheating coil inside the evaporative condenser. The heat transfer through the circuit is calculated as follows:

$$Q = U \times A \times Lmtd$$
 and therefore $U = \frac{Q}{A \times LmTD}$

Q = 10% of heat rejected = $0.1 \times 499.7 \approx 50$ kW = 50,000 W (14.2 TR)

A = Circuits x Lenght x circumference = $10 \times 18.26 \times \pi \times 0.0254 = 14.57 m^2$ (143 ft²)

 $LMTD = \frac{\Delta T1 - \Delta T2}{\ln \ln \left(\frac{\Delta T1}{\Delta T2}\right)} = \frac{75 - 10}{\ln \ln \left(\frac{75}{10}\right)} = 32.25 K$

Therefore, the required U value = $\frac{50000}{14.57 \times 32.25}$ = 106 W/m²-K. With an internal heat transfer coefficient of 750 W/m²-K, this appears to be reasonable.

Substituting the above assumptions and calculations into the compressor selection software, we generate the results shown in Figure 11.

Con	dition	s Mecha	anical	Oil	Management	Mot	ог	Cont	rols	Documental	tion Commen	ts.		
Con	npres	sor												
Mo	odel	V 14	400HS		 Refrigeran 	t R-7	717		a.		¥	S Incl	uded Componer	nts
Dr	ive Ty	pe Dire	ct Drive	¢.	×					. ~	÷ _			
Fre	equen	cy 50H	z		v					1		5		
Sp	eed T	ype Varia	able		÷					2				
Amb	nient (onditions	Condit	ion 1	Condition 2	< +	1							
					1									
1	Evap 1	[emperature	-	-40	°C	Cond	d Temp	erature	-3.	2 °C				
i	Evap F	ressure		0.71	9 bar(a	a) Cond	d Press	ure	3.8	1 ba	r(a)			
3	Suctio	n Line Loss		0	bar	Discl	harge L	ine Loss	0	ba	r			
13	Super	heat (Non-l	Jseful)	1	κ	Subo	cooling	At Cond	d 0	к				
3	Super	heat (Usefu	I)	0	ĸ	Spee	ed		11	71 RP	M			
		Condition	ET °C	CT ℃	Refrigerating Ca kW	pacity	Power kW	Speed RPM	EER	Mass Flow Rate kg/h	 Suct. Vol. Flow m³/h 	Disch. Vol. Flo m³/h	w	*
	A	. 1	-40.0	-5.0	200.1		49.7	1143	4.03	586.2	911.2	291.3		
Ô	A	2	-40.0	-3.2	200.1		52.5	1171	3.81	590.1	917.2	279.1		

Figure 11. Initial high-stage compressor criteria.

The induced load becomes $2 \times (200.1 + 52.5) = 505.2 \text{ kW} (142.6 \text{ TR})$. However, the higher discharge pressure also results in a higher compressor discharge temperature of $105 \degree \text{C} (221 \degree \text{F})$, increasing the performance of the de-superheater to 55.2 kW (15.7 TR). The required high-stage compressor capacity is therefore 505.2 - 55.2 = 450 kW (128 TR), resulting in the updated high-stage compressor selection, shown in Figure 12.

Con	dition	s Mech	anica	Oil	Management	Motor	Con	trols	Documentat	ion Commer	ıts	
Con	npres	sor										
Mo	odel	V 7	00HS		 Refrigerant 	R-717		4		X	Included (Components
Dr	ive Ty	pe Dire	ect Dri	ve	1.00				. ~	- iii		
Fre	equen	cy 50H	łz		~				1		4	
Sp	eed T	ype Var	iable						1.			
Amb	pient (onditions	Conc	lition	1 Condition 2 ¥	Ì.						
		conditions	Cont		·]			-				
1	Evap 1	lemperatur	e	-5	°C	Cond Tem	perature	3	5 °C			
	Evap I	Pressure		3.5	i55 bar(a)	Cond Pres	sure	1	3.513 ba	r(a)		
	Suctio	on Line Loss	8	0	bar	Discharge	Line Los	s 0	ba	rii		
	Super	heat (Non-	Useful) 1	к	Subcooling	g At Cor	nd 0	К			
	Super	heat (Usefu	II)	0	κ	Speed		1	174 RP	M		
	62		-204			12						
	_	Condition	ET °C	CT °C	Refrigerating Capa kW	icity Power kW	Speed RPM	EER	Mass Flow Rate	Suct. Vol. Flow m ³ /h	Disch. Vol. Flow	
	A	1	-5.0	35.0	499.5	107.7	1306	4.64	1648.9	578.5	212.5	
Ô	A	2	-5.0	35.0	450.3	96.0	1174	4.69	1486.2	520.3	190.8	

Figure 12. Revised high-stage compressor criteria.

The power consumption of the updated machinery room equipment is summarized in Table 2.

	Capacity	Shaft Power	Commonto	
	kW (TR)	kW (BHP)	Comments	
Booster Compressor 1	200.1 (56.9)	52.5 (70.4)	-40°C/-3.2°C	
Booster Compressor 2	200.1 (56.9)	52.5 (70.4)	-40°C/-3.2°C	
High-stage Compressor	450 (127.9)	96 (128.7)	-5°C/35°C	
Evaporative Condenser	545 (155)	7.7 (10.3)	35°C/22°C WB	
Refrigerant Pump	400 (113.7)	3 (4)	4 × circulation	
Total		211.7 (283.9)	3% reduction	

Table 2. Power consumption of updated machinery room equipment.

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This modification results in a power saving of approximately 3%. For a low-stage freezer plant operating for 6000 hr/year at 80% average load, the reduction in electrical energy is calculated as follows: 6000 hr × 80% × 217.7 kW-211.7 kW = 28,800 kWhr.. The annual reduction in electricity results in a cost savings of ZAR 72,000.00 (approximately USD 4000) based on an average local electricity rate of ZAR 2.50 per kWhr. Although the savings appear to be relatively small, they should be compared with the minimal additional costs. The cost can be calculated as follows:

•	Cost for an additional circuit in the condenser	ZAR 4,490.00
•	Installed cost for 25 m of DN 100 (sch40) pipe	ZAR 44,897.00
•	Additional cost for two DN 100 stop valves	ZAR 7,334.00
	Total Cost	ZAR 56,721.00
	Contractor Mark-up (25%)	ZAR 14,180.25
	Total Additional investment	<u>ZAR 70,901.25</u> (USD 3,938.95)

Moreover, the system is simple and requires no additional maintenance. Therefore, the payback period is approximately one year.

Final Installation and Measurements

The on-site refrigeration system is a standard pump-circulation, two-stage ammonia refrigeration plant with reciprocating compressors, evaporative condensers, a liquid receiver, a -5° C (23°F) intercooler, and a -40° C (-40° F) low-stage vessel, as shown in Figures 13-17. The final installation is somewhat different from the original proposal because additional high-stage processes are added to the -5° C (23°F) vessel, and this changes the ultimate selection of the high-stage compressors and

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evaporative condensers. However, the low-stage compressors and low-stage desuperheating remain the same and can be tested on site.



Figure 13. Machinery room with a view of the high-stage and low-stage compressors.



Figure 14. Machinery room with a view of the separator vessels.

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Figure 15. Evaporative condensers, liquid receiver, and heat recovery condenser.



Figures 16. The additional circuit on the evaporative condenser for the de-superheating coil.



Figure 17. The pressure and temperature sensors used for taking measurements.



Site conditions

During the testing and measurement process, we experienced a period of lower ambient conditions with dry-bulb temperatures ranging between 14 and 17°C (57 and 62.6°F) and wet-bulb temperatures between 13 and 15°C (55.4 and 59°F). Furthermore, the client did not run full production and only one low-stage compressor was in operation.

Testing methodology

To verify the data, the following steps are taken:

- Calibrate all temperature sensors and pressure transducers to ensure accurate data collection.
- Stabilize the low-stage load by adding hot gas through a throttling valve into the low-stage system. This allows the low-stage compressor to operate at a stable speed. The high-stage compressors are allowed to settle and match the induced load.
- The de-superheating coil on one condenser is closed to ensure that all the lowstage discharge vapor is channeled through a single condenser.
- Condenser fan speeds are manually set to 100% because the condensers are unloaded as a result of the lower ambient temperatures and lower load conditions.
- Once the plant stabilizes, data is recorded every 5 minutes for an hour.

Measured results

When the compressor has stabilized, the data points are all similar. Thus, we can focus on a single data series, as follows:

- Suction pressure = -37 kPa (g) (-5.366 PSIG)
- Suction temperature = $-40^{\circ}C(-40^{\circ}F)$
- Compressor speed = 1236 rpm
- Discharge pressure at cylinder head = 280 kPa (g) (40.6 PSIG)
- Discharge temperature = 113.9°C (237°F)

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A software analysis of the measured suction pressure/discharge pressure and compressor speed generates the data shown in Figure 18.

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Piston Rating

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GEA Grasso V 1400HS R-717

	OF	PERATING D	ATA - Condition 3			
Evaporator			Condenser			
Evap Temperature	-42	°C	Cond / Inter Tempera	iture	-3.2	°C
Evap Pressure	0.6	bar(a)	Cond / Inter Pressure	e	3.8	bar(a)
Superheat (Useful)	0	ĸ	SubCooling At Cond		0	к
Superheat (Non-Useful)	1	ĸ	Condenser HOR		234.5	kW
Suction			Discharge			
Suction Line Loss	0	bar	Discharge Line Loss	4	0	bar
Package Suction Loss	0.001	bar	Package Discharge I	Loss	0.016	bar
Saturated Suction Pressure	0.6	bar(a)	Discharge Pressure		3.8	bar(a)
Suction Temperature	-41	°C	Discharge Temperat	ure	115.8	°C
Mass Flow	537.8	kg/h	Mass Flow		537.8	kg/h
Volume Flow	926.5	m³/h	Volume Flow		262.7	m³/h
Theoretical Swept Volume	1312.5	m³/h				
		DEDEOD				
Compressor		T ERI ORI	Motor			
Capacity	181.9	kW	Voltage		400/3/50	V/PH/Hz
Power	52.6	kW	Motor Size		66	kW
Performance Factor	3.5	Qo/Pe	Frame		250S/M	
Speed	1236	RPM	Enclosure		IP23	
Percent Full Load	100	%	IE Classification		IE3	
Min Part Load	25	%	Full Load Amps		141	А
Drive Type	Direct Drive		Number of Poles		4	
			Frequency Regulate	d	Yes	
		OIL MANA	GEMENT DATA			
Oil Separator Type	OS	5	Oil Throw Before		2.5	cc/h
Oil Separator Diameter	492.0	mm	Oil Throw Before		3.7	ppm
Oil Separator Efficiency	95.5	%	Oil Throw After		0.9	cc/h
Ambient Temperature	40	°C	Oil Throw After		1.3	ppm
Oil Type	PR-OLEO (C-MH68A	Oil Cooler Included		Yes	
Compressor power at 1500 RPI	M * safety factor	of 1.1 (74.4 kW) is greater than power (of selected motor.		
Motor selection must be confirm Selected motor torque is less th	ied by the factory an compressor to	y prior to order orque.				
EU Ecodesign Regulation (EU 20	019/1781): 1-July	-2023, IE4 mano	latory for 75200 kW. IE	3 motors will be su	pplied without	CE marking.
		PROJ	ECT DATA			
Project Name:			Reference:			
Customer Name: New Con	tact		Prepared By:	Hannes Steyn		
Proposal Number:			Date:	11/11/2023		

Figure 18. Software analysis of the measured suction pressure/discharge pressure and compressor speed.

Notably, the calculated discharge temperature is very close to the measured discharge temperature under these conditions. As it is not possible to measure the mass flow on site, the manufacturer's data sheet is used. The mass flow for the given conditions is 538 kg/hr (1186 lb/hr).

Additionally, the following data is measured:

- Ambient dry-bulb temperature = $14.3 \degree C (57.7 \degree F)$
- Ambient wet-bulb temperature = $14^{\circ}C$ (57.2°F)
- Inlet pressure to de-superheating coil = 264 kPa (g) (38.2 PSIG)
- Outlet pressure of de-superheating coil = 253 kPa (36.7 PSIG)
- Inlet temperature of de-superheating coil = $80^{\circ}C(176^{\circ}F)$
- Outlet temperature of de-superheating coil = 18.7°C (65.7°F)

Comments on the observed data

It is important to know that it was raining during the measurement. The low-stage discharge line was wet, and this may have enhanced the amount of de-superheating before the hot gas reached the de-superheating coil. In addition, the final condenser was slightly large, to cope with the additional high-stage load and two condensers, and thus 8 circuits were used for the de-superheating coil in each condenser.

This means that a circuit load of $\frac{\frac{538 \, kg}{hr}}{8 \times 3600 \, s}$ was tested (Figure 19). The circuit length for this condenser is 22.83 m. Re-entering these values into the EES Pipe-Flo function, we obtain a pressure drop of 9.98 kPa (1.45 psi), versus a measured value of 11 kPa (1.59 psi). Note that vapor outlet temperatures within 5 K of the ambient wet-bulb temperature were achieved for the entire test period. However, further performance testing is required closer to the stated design temperatures.

🖬 Equations Window			E Shinon		(_
T2=80 (inlet temp from de-superheating coil)	Update	Menu	Main		
T3=-5 (Saturated intermediate temperature) T6=18.7 (soit temp from de-superheating coil)	Main Pro	ıgram 🏠	Unit Settings: SI C kPa kJ m	ass deg	
T/=(10+12/2 (average gas temperature) maseflow=538/3600 N=8 (number of circuits}	D (m) DELTAP [kPa]		h _H = 430.8 [W/m ² -K]	$h_T = 430 \text{ B} [W/m^2 \text{K}]$	L = 22.83 [m]
m_dot=massflow/N	1		massflow = 0.1494 [kg/s]	m = 0.01868 [kg/s]	N = 8
P=364	h_H [W/m*2-K]		Nusselt _T = 396.3	P = 364 [kPa]	Re = 85557
L=22.83	L [m]		RelRough = 0.01	T2 = 80 [C]	T3 = -5 [C]
RelRough=0.01	massflow [kg/s]		T6 = 18.7 [C]	T7 = 49.35 [C]	
Call pipeflow "ammonia' T7 P m dot D.L. RelRough h T h H .DELTAP. Nusselt T. f. Re)	m_dot [kg/s]				
	Nusselt_T		No unit problems were detected		
	P [kPa]				
	Re	~	Calculation time = 62 ms		

Figure 19. Observed relationships among the circuit load, circuit length, and pressure.

Conclusions

The results exceeded expectations considering that the approach temperature is closer to the wet-bulb temperature than originally anticipated, and more energy is therefore rejected to the atmosphere. The measured pressure drop is also consistent with the calculated values. Low-stage de-superheating can effectively be achieved with the use of split circuits in evaporative condensers. This reduces the induced load from the low-stage system by approximately 10%.

Furthermore, it is relatively easy and inexpensive to install. Condenser split circuits and uninsulated scheduled pipes are low-cost items in refrigerant plants, and the payback period should be short. A savings in capital expenditures is possible if smaller high-stage compressors are selected because of the reduced induced load, but the amount of savings varies from project to project. Moreover, different configurations of the de-superheating coil should be investigated further, and shorter circuits may improve the overall impact.

Overall, the evaluated system is ideal for traditional two-stage refrigeration plants and can even be retrofitted when condenser replacements are required, without the need for modifications to the other equipment or controls.

Resources

- GEA Heating and Refrigeration Technologies RT Select Version 13.4.
- F-Chart Software Engineering Equation Software Commercial Version 10.670.
- Evapco Spectrum Version 2.2023.1031.1 C.

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