Technical Paper #8

Study of the Energy Consumption of a CO_2/NH_3 Cascade Industrial Refrigeration System Operating in Costa Rica and Comparison with Direct Ammonia Systems on One- and Two-Stage Configurations

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Abstract

Many European and U.S. research papers claim that CO_2 only offers energy savings over an ammonia system below a certain low-suction temperature. With the objective of providing additional knowledge and real case examples on this subject, this paper presents the study of a CO_2/NH_3 cascade industrial refrigeration system with more than three years of successful operation in Costa Rica and compares direct ammonia systems in one- and two-stage configurations.

Introduction

Recent scientific findings on global warming potential point to human activities as the principal cause, and refrigeration and A/C are large contributors to this problem (Coulomb 2017). Therefore, increasing research on and development of refrigeration alternatives that are more energy efficient and environmentally friendly is critical.

Used as a natural refrigerant, CO_2 offers many advantages, such as low toxicity, reduced compressor size, nonflammability, and many energy-efficiency claims. In recent years, it has been used more frequently in industrial and supermarket refrigeration systems, either in subcritical or supercritical applications.

Because CO_2 will not condense above 87.7 °F (31°C) due to its low critical temperature, in many regions ambient conditions makes operating a supercritical system difficult (Vestergard, N.P., 2004). Hence, using CO_2 in a cascade configuration with another refrigerant has become more popular (Stoeker W.F. 2000). The basic operation is to condense the CO_2, at an intermediate pressure, on one side of a heat exchanger. This is achieved through the evaporation of a refrigerant on the other side of the heat exchanger, which will ultimately condense at ambient conditions at a reasonable pressure. For the ambient conditions that prevail in Costa Rica, a CO_2/NH_3 cascade system is the most appropriate option.

To achieve the objectives given the aforementioned considerations, this study will include the following:

1. Several days of monitoring the dry bulb (DB) temperature, dew point, and relative humidity of the outside conditions close to the condenser and also inside the holding freezers and dock;
2. Monitoring the real energy consumption of all main components of the system and comparing with the theoretical energy consumption expected; and
3. Comparing this CO_2/NH_3 cascade system energy efficiency with three configurations of ammonia system operating at the same loads and temperatures.

Description of the Facility and Its Refrigeration System

The system in this study is a 12,000 ft^2 (1115 m^2) expansion of one of the Costa Rican government's main national cold network facilities. The facility's layout is shown in figures 1 through 3 below and consists of:

- Two holding freezers, which are designed for -17.5 °F (27.5 °C) air temperature hold 640 pallets organized in a four-level rack that also serves as support structure for the roof and wall panel. Each room has two 8.5 tons of refrigeration (TR) units Evapco SSTMC2-00706-3, evaporating at -27.5 °F (-33 °C), with two 1 1/2 HP (1.12 kW) fan motors, and delivering 19,184 ft^3/min (CFM) (543 m^3/min) with 3 fins/in. The evaporators' defrost systems, are electric and activate a 23 kW heater every 8 hours of accumulated solenoid operation.
- One pre-room dock, which has five loading positions, designed at 41 °F (5 °C) air temperature. It has three 7.5 TR units Evapco SSTMC2-00547-4 evaporating at 20 °F (-6.7 °C), with two 1/2 HP (0.373 kW) fan motors, and delivering 6,746 CFM (191 m^3/min) with 4 fins/in. The evaporators' defrost systems, which are electric and activate a 12.8 heater every 8 hours of accumulated solenoid operation.
- Machinery room (see Figures 1 and 2 and Appendix E, figures 28-33).
- At low temperature, the refrigeration total load is 44 TR (154.75 kW), 32 TR (112.3 kW) in the first phase, and 12 TR (42.1 kW) for a short-term future expansion.
- At medium temperature, the load is 37.5 TR (131.9 kW), 22.5 TR (79.0 kW) in the
- first phase, and 15 TR (52.7 kW) for a short-term future expansion.

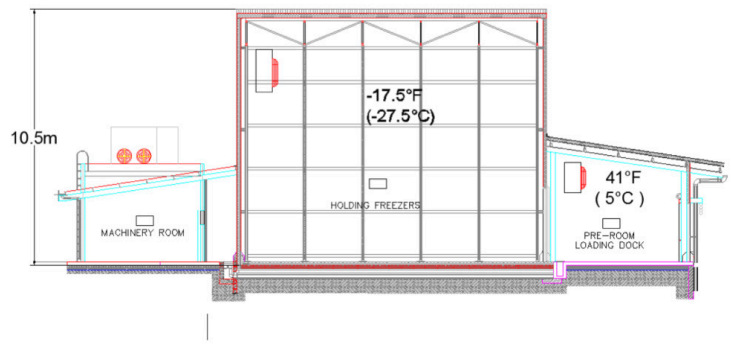

Figure 1. Side view of the refrigeration facility.

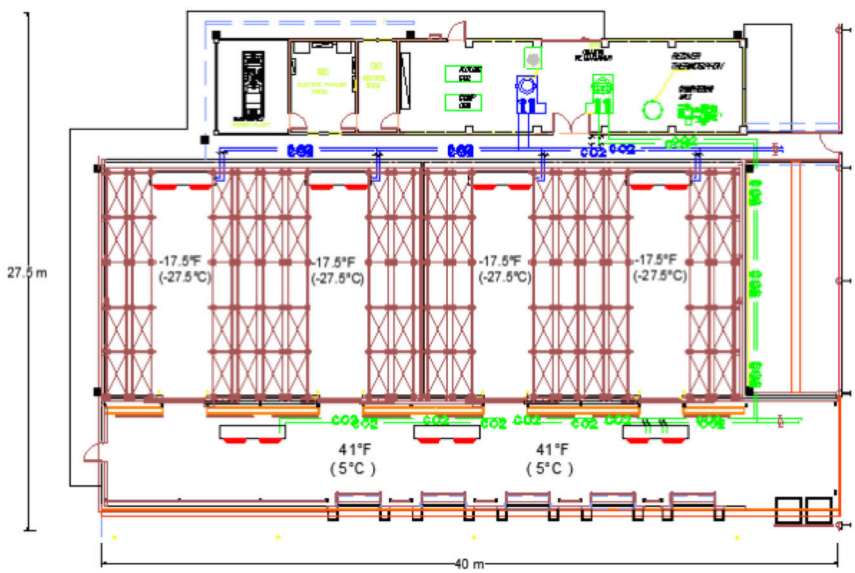

Figure 2. Layout of the refrigeration facility.

Figure 3. Refrigeration facility with CO_2/NH_3 expansion appearing on the left.
Taken by the author and permission to publish by Ing. Francisco Annet, from PIMA

The components of the main refrigeration system are shown in figures 4 and 5, and include

- One CO_2 recirculation package that operates at -27.5 °F (-33.1 °C), with two pumps.
- One four-cylinder reciprocating CO_2 compressor, with 44 TR (154.7 kW) capacity, operating at -27.5°F (-33 °C) saturated suction temperature and condensing at 20 °F (-6.7 °C), which requires 35.6 brake horsepower (BHP) and a 40 HP (30 kW) motor. The capacity control of 25%, 50%, and 75% is obtained by unloading one, two, or three cylinders.
- One shell and tube heat exchanger that evaporates ammonia at 11 °F (-11.7 °C) and condenses CO_2 at 20 °F (-6.7 °C).
- One medium-pressure receiver for the condensed CO_2 that operates at 20 °F (-6.7 °C) and serves as a recirculator for the medium temperature load with two CO_2 pumps. The amount of gas that will return from the evaporators to the top of the receiver will enter the CO_2/NH_3 condenser/evaporator where it gets condensed again. This way the CO_2 is used as brine, and the system does not require a compressor for the CO_2 medium-temperature evaporators.
- One heat exchanger that superheats the CO_2 before it enters the compressor and an oil rectifier.
- One NH_3 screw compressor for the high side of the CO_2/NH_3 cascade system, with 92TR (322.9 kW) of capacity operating at +11 °F (-11.33 °C) saturated suction temperature and condensing at 90 °F (32.2 °C), that requires 127.5 BHP with a 150 HP (110 kW) motor. The capacity control is achieved using the sliding valve.
- One NH_3 evaporative condenser with two 5 HP (4 kW) fans each with a variable frequency driver (VFD) and a 2 HP (1.5 kW) spray pump.
- One thermosyphon/high-pressure receiver.
- One emergency condensing unit to keep the CO_2 at a reasonable pressure in case the power fails.

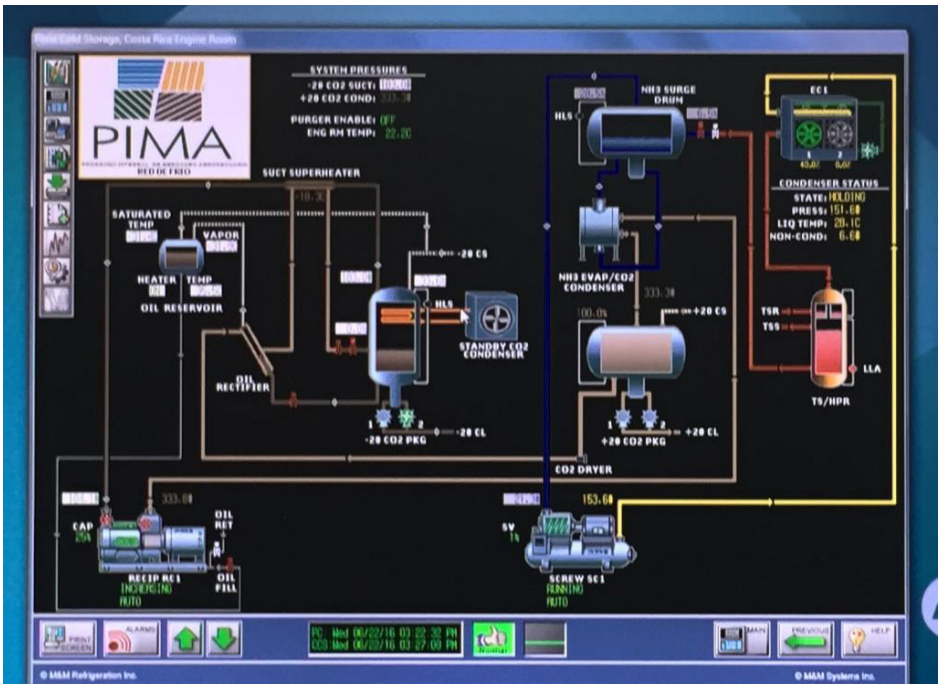

Figure 4. Flow diagram of the actual system from screen in control room.

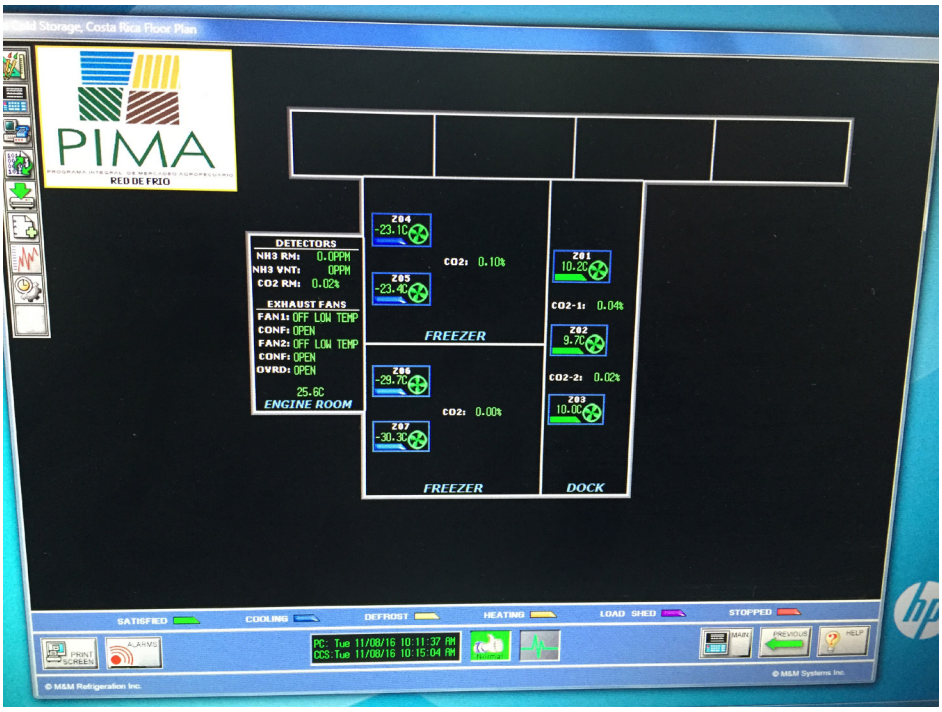

Figure 5. Evaporators and and air temperature from screen in control room.

Monitoring, Collection of Data, and Calculations

Three different systems operating with direct ammonia were selected to operate under the same loads and temperatures. The coefficients of performance (COP) expected were calculated and compared with the energy efficiency coefficient obtained for the CO_2/NH_3 cascade system. (Stoeker W.F. 1998)

Dry bulb temperature, dew point temperature, and relative humidity monitoring

Dry bulb temperature, dew point temperature, and relative humidity were monitored for seven days to determine the outside conditions and the wet-bulb temperature for the condenser operation and to verify the operation inside the holding freezers and preroom dock. See figures 6,7 and 8 that show data collection graphs.

Ambient Conditions: The maximum dry bulb temperature measured during seven days was 92 °F (33.3 °C), the maximum relative humidity was 96.7%, and the maximum combination of both resulted in a wet bulb temperature of 78 °F (25.5 °C).

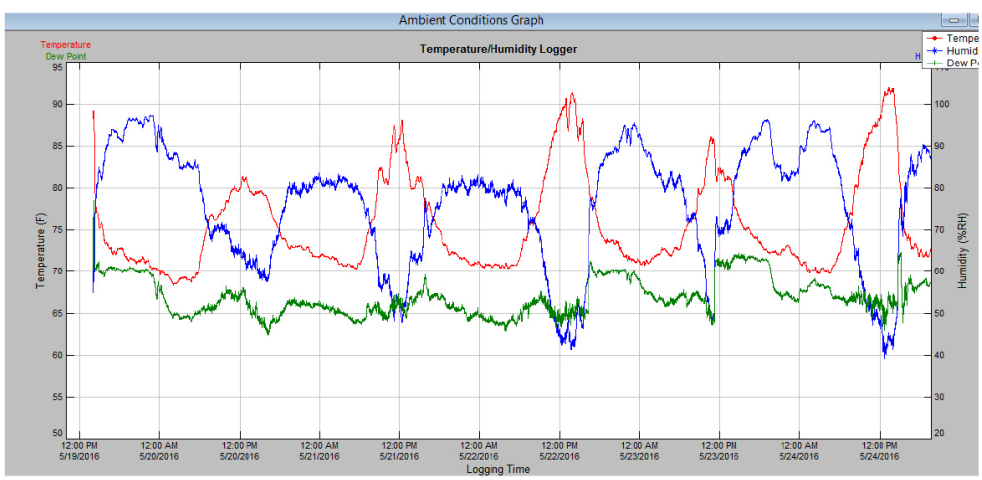

Figure 6. Ambient temperatures and relative humidity during five days. Red = dry bulb temperature; green = dew point; blue = humidity; left-side scale in °F; right side scale in %. Dock Preroom: The average air dry bulb temperature measured over seven days was 48.5 °F (9.1°C), and the average relative humidity was 85%.

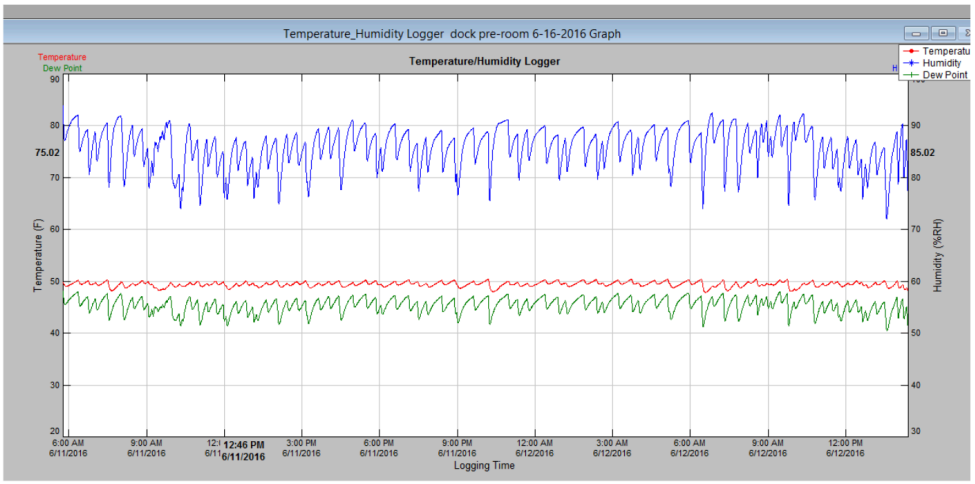

Figure 7. Dock temperatures and relative humidity over three days. Red = dry bulb temperature; green = dew point; blue = humidity; left-side scale in °F; right-side scale in %. Holding Freezers: The average dry bulb temperature measured during seven days was -15 °F (-26.1°C), and the average relative humidity was 65%.

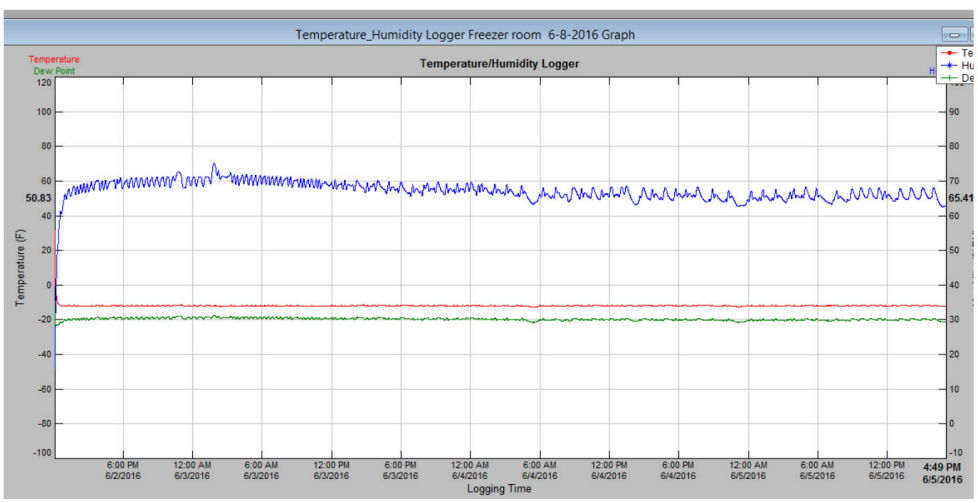

Figure 8. Holding freezer temperatures and relative humidity over six days. Red = dry bulb temperature; green = dew point; blue = humidity; left-side scale in °F; right-side scale in %.

Real energy consumption

The main electric panel has one breaker for each compressor and one breaker that feeds power to a control panel with the rest of the refrigeration equipment: condenser fans, low and medium temperature evaporators, recirculating pumps, defrost heaters, etc.

The energy consumption in kWh of all system main components was monitored for six days (See Figure 9 below), and was then compared with the calculated expected energy consumption. The average consumption in kWh/ day (24 hours) obtained is as follows:

- Breaker feeding the NH_3 screw compressor: 1,333 kWh/day;
- Breaker feeding the CO_2 reciprocating compressor: 279 kWh/day; and
- Breaker feeding the control panel with rest of the equipment: 470 kWh/day.

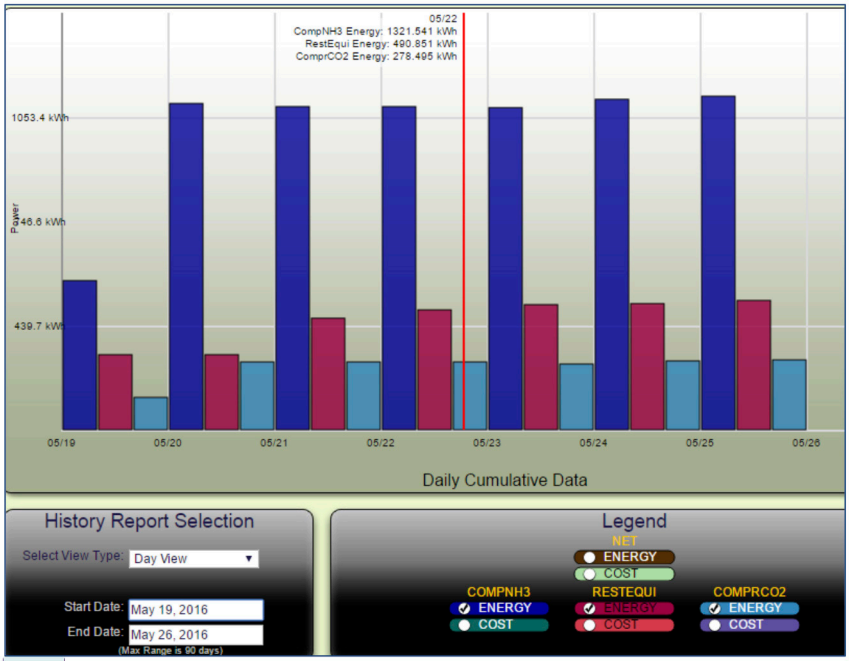

Figure 9. Energy consumption in kWh/day over six days. Blue: screw NH_3 compressor; light blue: reciprocating CO_2 compressor; purple: control panel with remaining equipment.

Appendix A includes the graphs of the energy consumed by the rest of the equipment on the control panel.

The energy consumed by the defrost heaters was measured at the seven individual breakers, adding to a total of 153 kWh/day (see Figure 10).

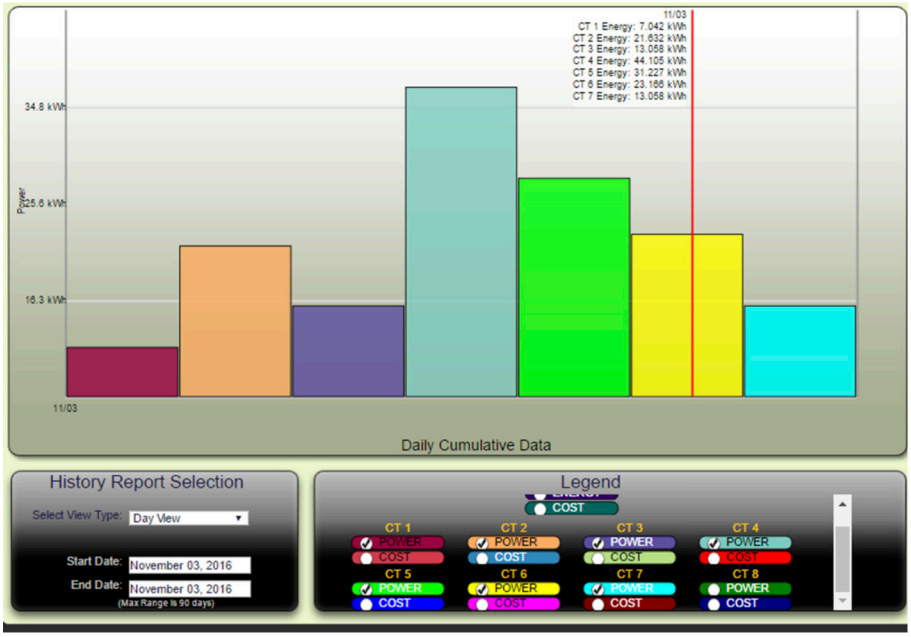

Figure 10. Energy consumption of defrost heaters in kWh/day. The bars represent kWh per day consumed by each of the seven defrost heaters.

Comparing the theoretical average energy consumption at the current partial load operation with the measured real average consumption, the equivalent real operation % of time can be obtained as shown in Table 1.

Circuit breaker on the main electric panel	Qty	BHP	kW electric HP x 0.746	Efficiency 90% (KW)	Volts	Operation hours expected	Expected kWh/ day	Measured kWh/day	Measured/ Expected
NH_3 screw compressor 150 HP motor	1	112	84	92.84	460	16	1,485	1,333	90%
CO_2 reciprocal compressor 40 HP motor	1	26	19	21.28	460	16	340	279	82%
Feed to the control panel with rest of equipment	1	n/a	n/a	n/a	n/a	n/a	659	470	71%
Total energy in kWh/day	n/a	n/a	n/a	n/a	n/a	n/a	2,485	2,082	84%

Table 1. Energy consumption in kWh/day.

The difference between measured and expected energy consumption is very small, with a total percentage of usage in all the equipment over 84%.

To confirm the heater's energy consumption, the seven circuits were measured individually. Table 2 shows the individual energy consumption obtained for the remaining equipment on the control panel (see also Figure 10).

Panel with remaining equipment	Qty	kW	Volts	Expected operation in hours	Expected kWh/ day	Expected kWh/day in groups	Measured kWh/day	
							Breakers on control panel	
Medium temp recirculation CO_2 pump	1	4.9	460	16	78.40	181		155
Low temp recirculation CO_2 pump	1	4.9	460	16	78.40			
Condenser recirculating water pump	1	1.5	460	16	24.00			
Fan #2 evap condenser	1	3.73	460	16	59.68	119		7.1
Fan #1 evap condenser	1	3.73	460	16	59.68			3.56
Evaporator 01 dock	2	0.373	460	22	16.41	49		51
Evaporator 02 dock	2	0.373	460	22	16.41			
Evaporator 03 dock	2	0.373	460	22	16.41			
Evaporator 04 holding freezer	2	1.119	460	20	44.76	179		282
Evaporator 05 holding freezer	2	1.119	460	20	44.76			
Evaporator 06 holding freezer	2	1.119	460	20	44.76			
Evaporator 07 holding freezer	2	1.119	460	20	44.76			
Defrost heater evap 01	1	12.8	460	1	12.80	130	7.042	153*
Defrost heater evap 02	1	12.8	460	1	12.80		21.632	
Defrost heater evap 03	1	12.8	460	1	12.80		13.058	
Defrost heater evap 04	1	23	460	1	23.00		44.105	
Defrost heater evap 05	1	23	460	1	23.00		31.227	
Defrost heater evap 06	1	23	460	1	23.00		23.166	
Defrost heater evap 07	1	23	460	1	23.00		13.058	
Total energy in kWh/day						659		652

Table 2. Energy consumption in kWh/day of remaining equipment and defrost heaters.

* Note that in the first set of data, the breaker feed for the defrost heaters was mismarked, giving a wrong total of 41.45 kWh instead of 153 kWh.

Table 2 shows that the equipment's real energy consumption was similar to what was expected, with the exception of the evaporative condenser, which showed lower consumption because each fan motor has a VFD installed, and the holding freezer evaporators, which had a higher real consumption.

In the latter case, the energy consumption on the breaker feeding the four holding freezer evaporators was higher because the customer had needed to set the saturated suction temperature of the rooms to -30 °F (-34.4 °C) instead of -27.5 °F (-33.1 °C) to cool rejected product that could not be frozen in a blast freezer in the old part of the plant more quickly. Figure 5 shows that one of the rooms was being maintained 7 °F lower than the other.

Energy Efficiency of the CO_2/NH_3 Cascade System

The performance coefficient (COP) was obtained for the present operation conditions as well as for the conditions when the second phase loads are added to the project.

To obtain a reasonable coefficient of performance (COP) of the system under study that can be easily compared with other configurations, establishing the following is important.

The energy efficiency factors are obtained for both conditions: (1) at the current partial refrigeration load of 32 TR (112.5 kW) low temperature + 22.5 TR (79.1 kW) medium temperature for a total of 54.5 TR (191.6 kW) and (2) when the full load is completed to 44 TR (154 kW) for low temperature and 37.5 TR (131.9 kW) for medium temperature for a total of 81.5 TR (285.9kW).

In present conditions, the total energy consumed per day is 2,082 kWh, so with 54.5 TR the rate is 38.20 kWh/day per TR. However, keeping in mind that the relationship is not linear; when the kWh/day is divided by 24 hours, it results in 1.59 kWh/TR.

Of the energy consumed per day, 78% of the total is evidently due to the compressors. It can also be assumed that the energy consumption of the remaining components of the system are very similar in other configurations. So, the comparison of the energy coefficients COP is based only on the compressor packages.

For this project, the low side of the cascade system has a reciprocating compressor Sabroe HPO 24, rated for 44 TR (154 kW) at -27.6 °F/20 °F (-33 °C/6.7 °C), using 35.3 BHP (26.3kW) with a 40 HP (30 kW) motor, and on the high side is a screw compressor Howden H33HT, with thermosyphon oil cooling and rated for 92 TR (322KW) at +11 °F/+90 °F (-11.7 °C/32.2 °C) using 127.5 BHP (95.1 kW) with a 150 HP (110kW) motor.

Because the Howden software was not available, publicly available manufacturers' software from both reciprocating and screw compressors was used to obtain the data required to compare the different configurations of refrigerant systems (see Appendices B and C).

Pressure/enthalpy (P/H) Mollier charts were plotted using the Cool Pack Refrigeration Software from the Mechanical Engineering Department of the Technical University of Denmark (see Appendix D).

The following table shows the coefficient of performance for the cascade system at full load.

NH_3/CO_2 cascade system	Load		Saturated suction temperature (SST)	Saturated condensing temperature (SCT)	Shaft power		HP/TR
	TR	kW	°F (°C)	°F (°C)	HP	kW	
HPO 24-CO_2 - Recip							
Low temp refrigeration load	44.0	154.7	-27.5 (-33)	20 (-6.7)	35.30	26.33	0.802
Low side rejected heat	51.6	181.2					
Medium temp refrigeration load	37.5	131.9	-27.5 (-33)				
RFX58-NH_3-Screw							
High side total load	90.0	316.4	11 (-11.67)	90 (32.2)	126.40	94.29	1.418
Heat reject high side	116.8	414.2					
Total refrigeration load	81.5	286.6			161.70	120.63	1.98
Note: For the current partial load conditions the COP obtained is 1.98.						COP	2.38

Table 3. COP for the NH_3/CO_2 cascade system.

Comparison of the energy efficiency obtained for the CO_2/NH_3 system with different configurations of direct ammonia systems

Three different systems operating with direct ammonia were selected to operate under the same loads and temperatures. The coefficients of performance (COP) expected were calculated and compared with the energy efficiency coefficient obtained for the CO_2/NH_3 cascade system. (Stoeker W.F. 1998)

To make the comparison with direct ammonia systems, the same loads and operating conditions were introduced into the manufacturers' software for the following three configurations:

- One-stage ammonia system using one screw compressor for the low temperature load and its side port for the medium temperature load (see Table 4).
- Direct two-stage ammonia system with one screw booster compressor and a second screw compressor for the high stage (see Table 5).

- Two-stage ammonia system with one reciprocating booster compressor and a screw compressor for the high stage. This system was chosen to have the same type of compressor on the booster side as the CO_2/NH_3 cascade system under study (see Table 6).

One-stage NH_3 with 1 screw compressor using the side port for the medium temp load	Load		SST	SCT	Shaft power		HP/TR
	TR	kW	°F (°C)	°F (°C)	HP	kW	
RFX85-NH_3-Screw							
Low temp refrigeration load	44.0	154.7	-27.5* (-33)	90 (32.2)	171.7	128.09	2.107
Medium temp refrigeration load	37.5	131.9	20 (-6.7)	90 (32.2)			
Total refrigeration load	81.5	286.6					2.107
Heat reject high side	118.5	416.6					
NOTE: For the current partial load conditions the COP obtained is 1.74.						COP	2.24

Table 4.COP for one-stage NH_3 screw compressor system using the side load.

2 two-stage NH_3 with 2 screw compressors	Load		SST	SCT	Shaft power		HP/TR
	TR	kW	°F (°C)	°F (°C)	HP	kW	
RFX68-NH_3-Screw							
Low temp refrigeration load	44.0	154.7	-27.5* (-33)	20 (-6.7)	51.30	38.27	1.167
Total low side reject heat	54.6	192.0					
Medium temp refrigeration load	37.5	131.9					
RFX50-NH_3-Screw							
High side total load	92.1	323.9	.20 (-6.7)	90 (32.2)	111.60	83.25	1.21
Heat reject high side	116.9	410.4					
Total refrigeration load	81.5	286.6			162.90	121.52	1.96
Note: For the current partial load conditions the COP obtained is 2.15.						COP	2.36

Table 5. COP for two-stage NH_3 system with two screw compressors.

Two-stage NH_3 with 1 reciprocating in the booster side and a screw in the high stage	Load		SST	SCT	Shaft power		HP/ TR
	TR	kW	°F (°C)	°F (°C)	HP	kW	
A 93B 4512 XL@930RPM							
Low temp refrigeration load	44.0	154.7	-27.5* (-33)	20 (-6.7)	53.90	40.21	1.22
Total low side reject Heat	55.4	194.9					
RFX50-NH_3 - Screw							
Medium temp refrigeration load	37.5	131.9					
High side total load	92.9	326.7	.20 (-6.7)	90 (32.2)	112.80	84.15	1.19
Heat reject high side	117.0	411.4					
Total refrigeration load	81.5	286.6			166.70	124.36	2.05
Note: For the current partial load conditions the COP obtained is 1.99.						COP	2.30

Table 6. COP for two-stage NH_3 system with one reciprocating booster and one high side screw compressor.

Table 7 summarizes the COPs calculated for the different system configurations both for the current partial load and for the full load.

System configuration	COP Partial load	Compared with base	COP Full load	Compared with base
NH_3/CO_2 cascade system	1.98	1.00	2.38	1.00
One-stage NH_3 with 1 screw compressor using the side port for the medium temp load	1.74	0.88	2.24	0.94
Two-stage NH_3 with 2 screw compressors	2.15	1.09	2.36	0.99
Two-stage NH_3 with 1 reciprocating in the booster side and a screw in the high stage	2.10	1.06	2.30	0.97

Table 7. Comparison of the COP of different NH_3 system configurations with the NH_3/CO_2 cascade system analyzed.

Conclusions

Note that because of the high pressures of the discharge gas, this CO_2/NH_3 cascade system employs electric defrost instead of hot gas defrost as in the NH_3 systems analyzed.

The analysis of this particular system shows that with an evaporative temperature of -27.5 °F (-33 °C) and an intermediate temperature of 20 °F (-6.7 °C) for the CO_2, and an evaporative temperature of +11°F (-11.7°C) and a condensing temperature of 90°F (32°C) for the NH_3, the CO_2/NH_3 cascade system with an approach of 9 °F (5 °C) has approximately a 1% advantage in efficiency over the two-stage ammonia system with screw compressors, a 6% advantage over the one-stage one screw compressor using the port side for medium temperature, and a 3% advantage over the two-stage ammonia system with a reciprocating booster and one screw compressor on the high side.

For the current operation at partial load, the two-stage ammonia system with screw compressors has an advantage in efficiency of 9% over the CO_2/NH_3 cascade system, and the two-stage ammonia system with a reciprocating booster and one screw compressor shows an advantage of 6% over the cascade system. This is due to the fact that for the cascade system a larger screw compressor, operating at partial load, is required to handle the 9 °F lower suction temperature as a result of the approach of the heat exchanger.

It is interesting to note the once the client completes the second phase of the project the relative efficiency of the system will improve, and will be closer to the efficiency of an ammonia two stage sytem with screw compressors.

To handle the low temperature load, the direct two-stage ammonia system requires a 12-cylinder compressor running at 930 RPM instead of the 4-cylinder one at 1,200 RPM required with the CO_2 system. This difference shows an advantage of the CO_2

system over NH_3 in the displacement required by the equipment at low temperatures due to the large difference in density. (Pearson, A.B. 2000)

As mentioned before, an important disadvantage of the CO_2/NH_3 cascade system is that the hot gas defrost is not a good option because a large increase in pressure is associated with an increase in temperature for the CO_2. As Table 2 and Figure 10 show, for this particular system the daily energy consumption of the heaters measured was 153 kWh, which represents 7.3% of the total 2,082 kWh per day measured.

If the defrost system is not considered in the comparison, the CO_2/NH_3 cascade system appears to be a good option for the conditions of this project operating at full load: evaporative temperature of -27.5 °F (-33 °C) and an intermediate temperature of 20 °F (-6.7 °C) with an approach of 9 °F (5 °C).

With these results it can be concluded that for the operation conditions of this plant, even at full load, any comparative energy savings achieved by the CO_2/NH_3 cascade system, would be eliminated by the energy consumption of the defrost heaters.

Because the electric defrost is rarely used in industrial NH_3 systems, increasing research into improving other energy-efficient and affordable methods for the CO_2 refrigeration systems is important.

It is also interesting to note is that the use of VFD on the evaporative condenser motors reduces the energy consumed by a considerable amount.

The energy consumption of the system measured at the partial current load shows that the system can handle an increase in operation with no problem. Because the ambient conditions in Costa Rica are very stable throughout the year, it can be assumed that the variation in the energy consumption will depend mainly on the warehouse operation hours.

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Appendix A: Energy Consumption Monitoring Diagram

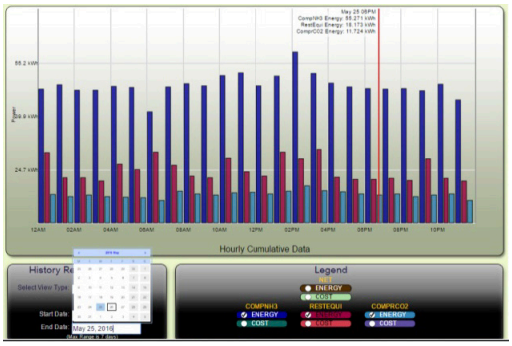


Figure 11. Refrigeration equipment kWh/hour.

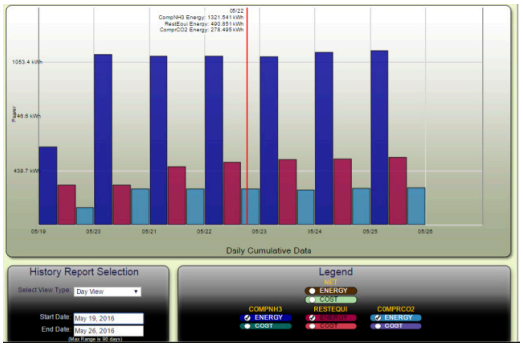


Figure 12. Refrigeration equipment kWh/day.

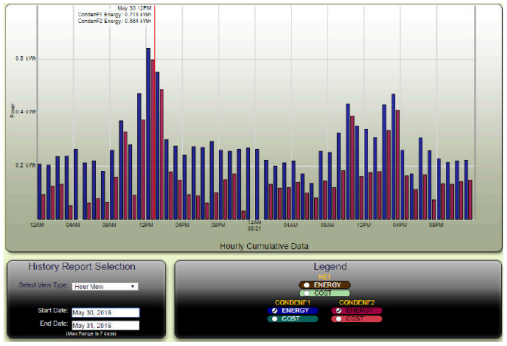


Figure 13. Fans 1, 2 of condenser kWh/hour.

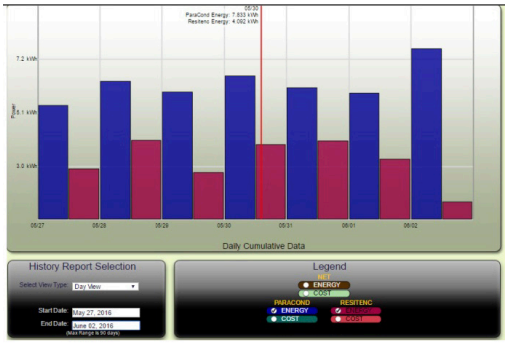


Figure 14. Fans 1,2 of condenser kWh/day.

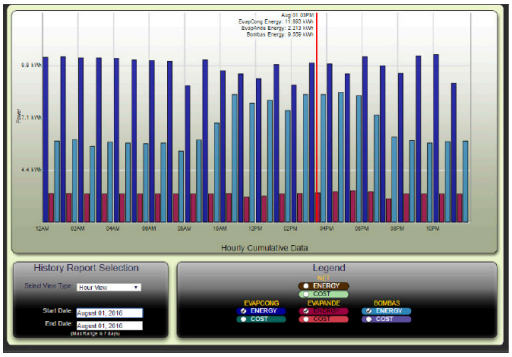


Figure 15. Low and medium temperature evaporators and pumps, kWh/hour.

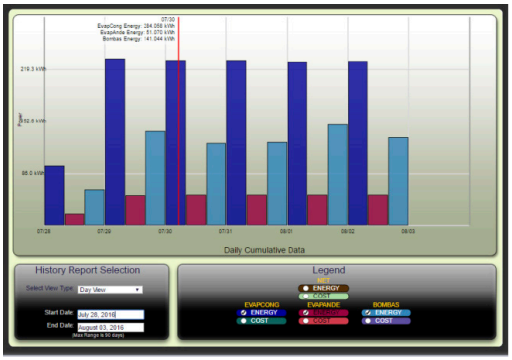


Figure 16. Low and medium temperature evaporators and pumps, kWh/day.

Appendix B: Compressor Software Runs

Future complete load	NH_3 one-stage	NH_3 two-stage		NH_3 two-stage		CO_2/NH_3	
	Using side load	Booster	High stage	Booster	High stage	Low side CO_2	High side NH_3
	RXF 85	RFX 68	RFX 50	A93 B 4512 XL	RFX 50	HPO24	RFX58
Compressor capacity (TR)	44	44	89.7	44	94.57	44	89.12
Side load capacity (TR)	37.5	0	88	N/A	N/A	N/A	N/A
Capacity (%)	88.9	93.1	88	100	92.8	100	94.6
Required power (BHP)	171.7	51.3	108.5	55.5	112.8	35.3	126.4
Rejection (MBH)	1,422	626.4	1,367	684.9	1,436	619	1,416
Speed (RPM)	3,550	3,550	3,550	930	3,550	1,200	3,550
Slide valve position (%)	96.6	96.1	93.1	N/A	95.9	N/A	94.6
Discharge temp (°F)	178.2	158	173.1	114.4	174.2	N/A	179.1
Oil heat rejection (MBH)	272	39	130	12	131	N/A	160
HP/TR	2.1067	1.167	1.21	1.224	1.1928	0.8023	1.4183

Present partial load	NH_3 one-stage	NH_3 two-stage		NH_3 two-stage		CO_2/NH_3	
	Using side load	Booster	High stage	Booster	High stage	Low side CO_2	High side NH_3
	RXF 85	RFX 68	RFX 50	A93 B 4512 XL	RFX 50	HPO24	RFX58
Compressor capacity (TR)	32	32	65.2	32	67.3	32	60.8
Side load capacity (TR)	22.5	0	0	0	0	0	0
Capacity (%)	64.6	67	64	N/A	66	67	79.7
Required power (BHP)	147.7	40.2	80.9	43.5	87.6	32.41	94.4
Rejection (MBH)	1,035	453.6	939	537.6	1,042	460	971
Speed (RPM)	3,550	3,550	3,550	930	3,550	1,200	3,550
Slide valve position (%)	87.9	72.89	61.2	N/A	71.3	N/A	71.6
Discharge temp (°F)	175.9	156.1	166.3	114.4	167.7	N/A	172.6
Oil heat rejection (MBH)	267	37	114	12	119	N/A	144
HP/TR	1.99	1.255	1.342	1.224	1.301	1.0128	1.553

Appendix C: Manufacturers' Software Runs

Software runs for the conditions at the future full load:

Booster A 93 B 4512 XL Future.pdf

RFX 50 High stage Future 89.7.pdf

RFX 50 High stage Future+ Low Recip.pdf

RFX 58 High ST-Future + Boost CO2.pdf

RFX85 with side load.pdf

RFX 68 Booster.pdf

RFX 50 92.1 TON.pdf

Software runs for the current partial load:

RFX85 Sideload.pdf

RFX 68 Booster Present.pdf

RFX 58 High stage Present+Boost CO2.p

RFX 50 High stage Present+Recip.pdf

RFX 50 High stage Present.pdf

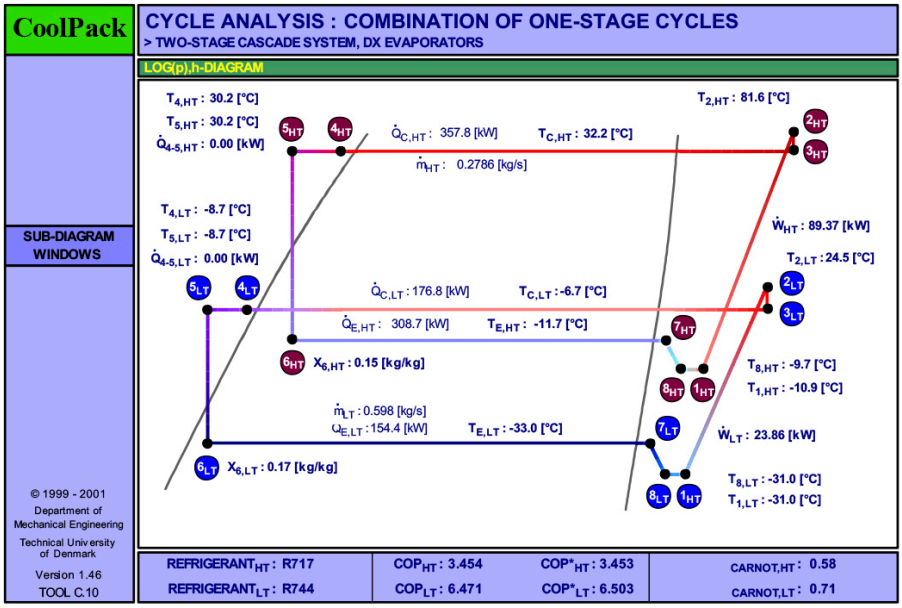
A 73B 4512 XL Present.pdf

RFX 50 High stage Present-62.8.pdf

RFX 58 High stage Present+ Boost CO2-74.1.pdf

RFX 50 High stage Present-60.3.pdf

Appendix D: Mollier Diagrams and Base of Calculations



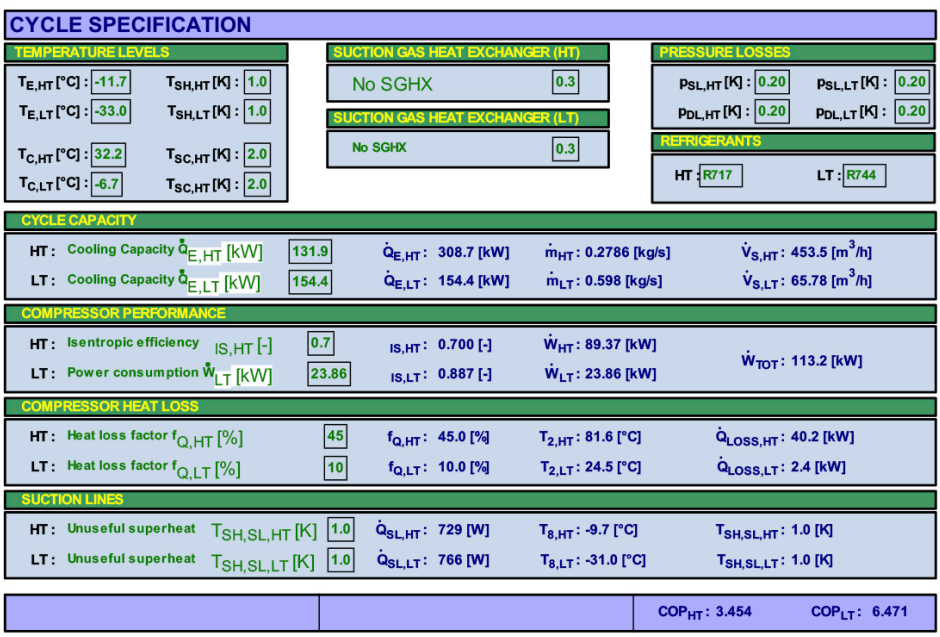
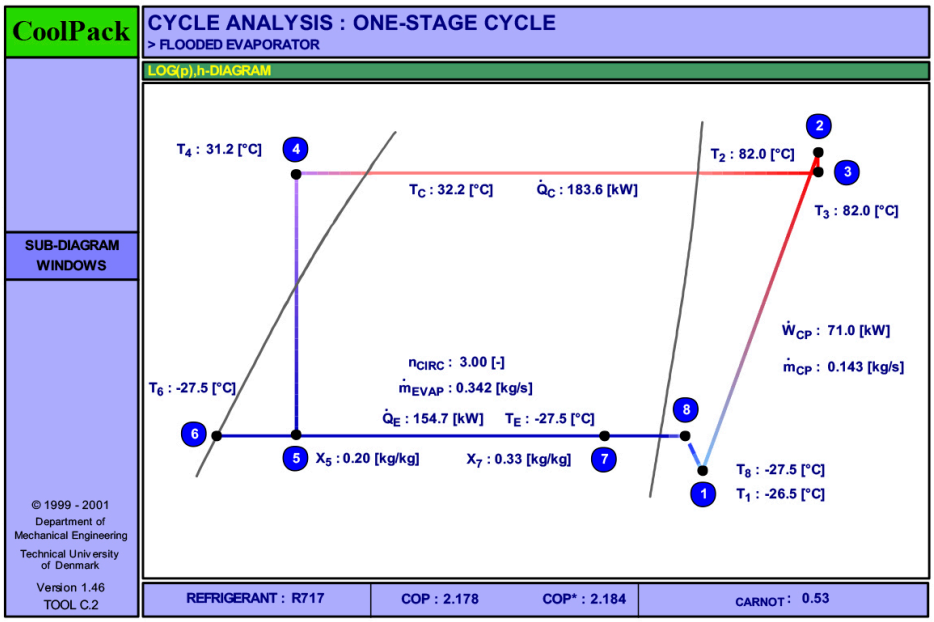


Figure 17. CO_2/NH_3 cascade system with reciprocating on the low side and screw compressor on the high side.



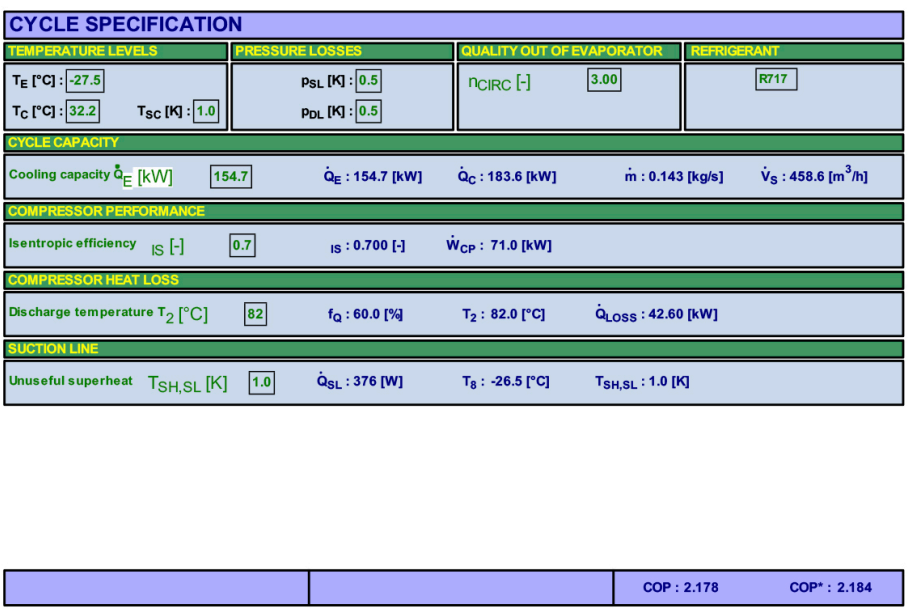
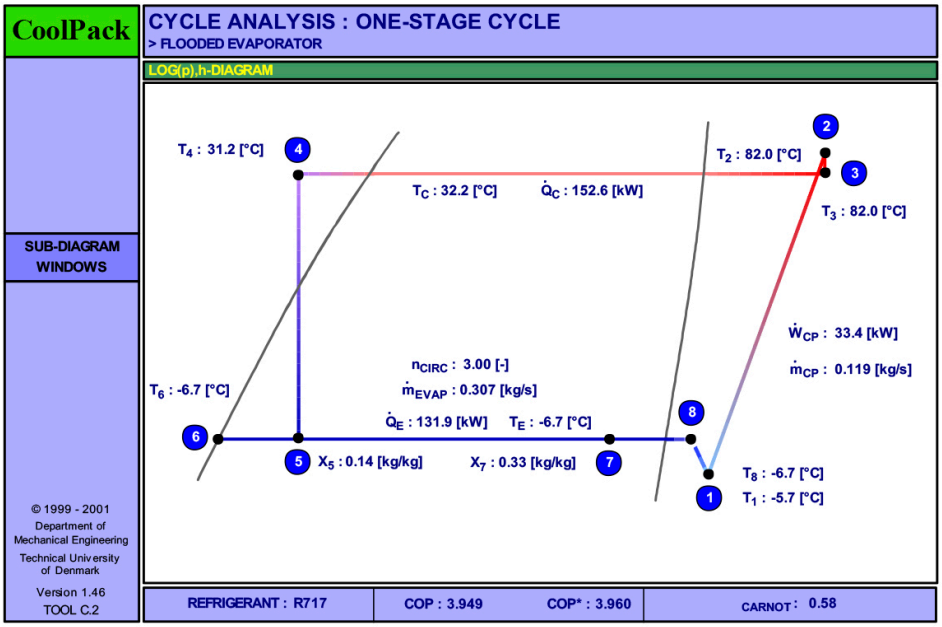


Figure 18. NH_3 one-stage one screw compressor with medium temperature loads on side port; A = low temperature load.



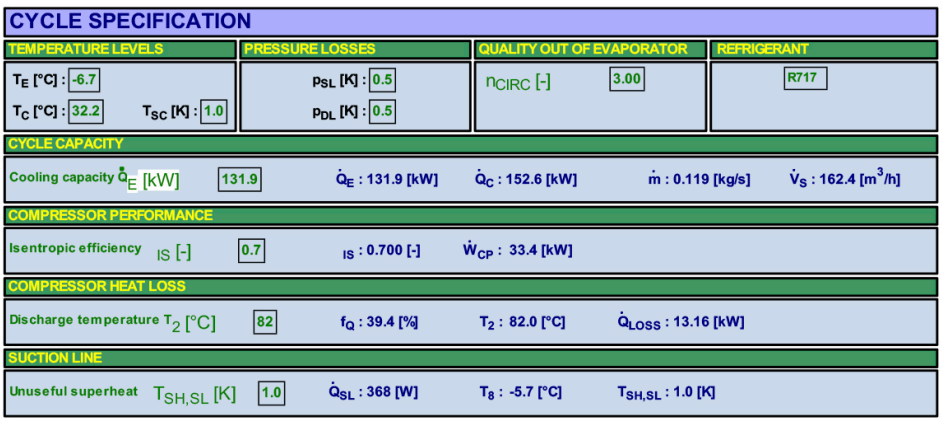


Figure 19. NH_3 one-stage one screw compressor with medium temperature loads on side port; B = medium temperature load.

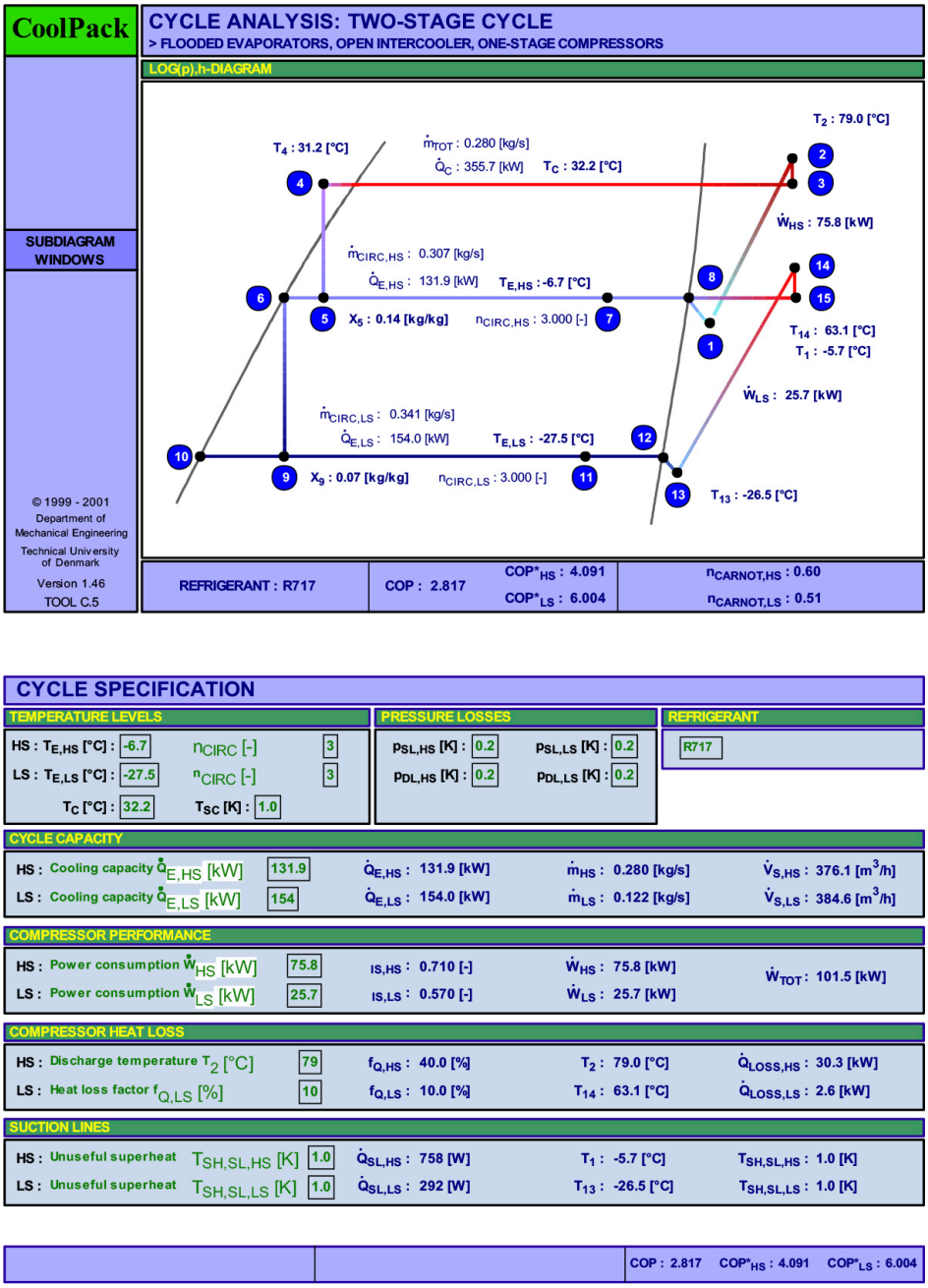


Figure 20. NH_3 two-stage with two screw compressors.

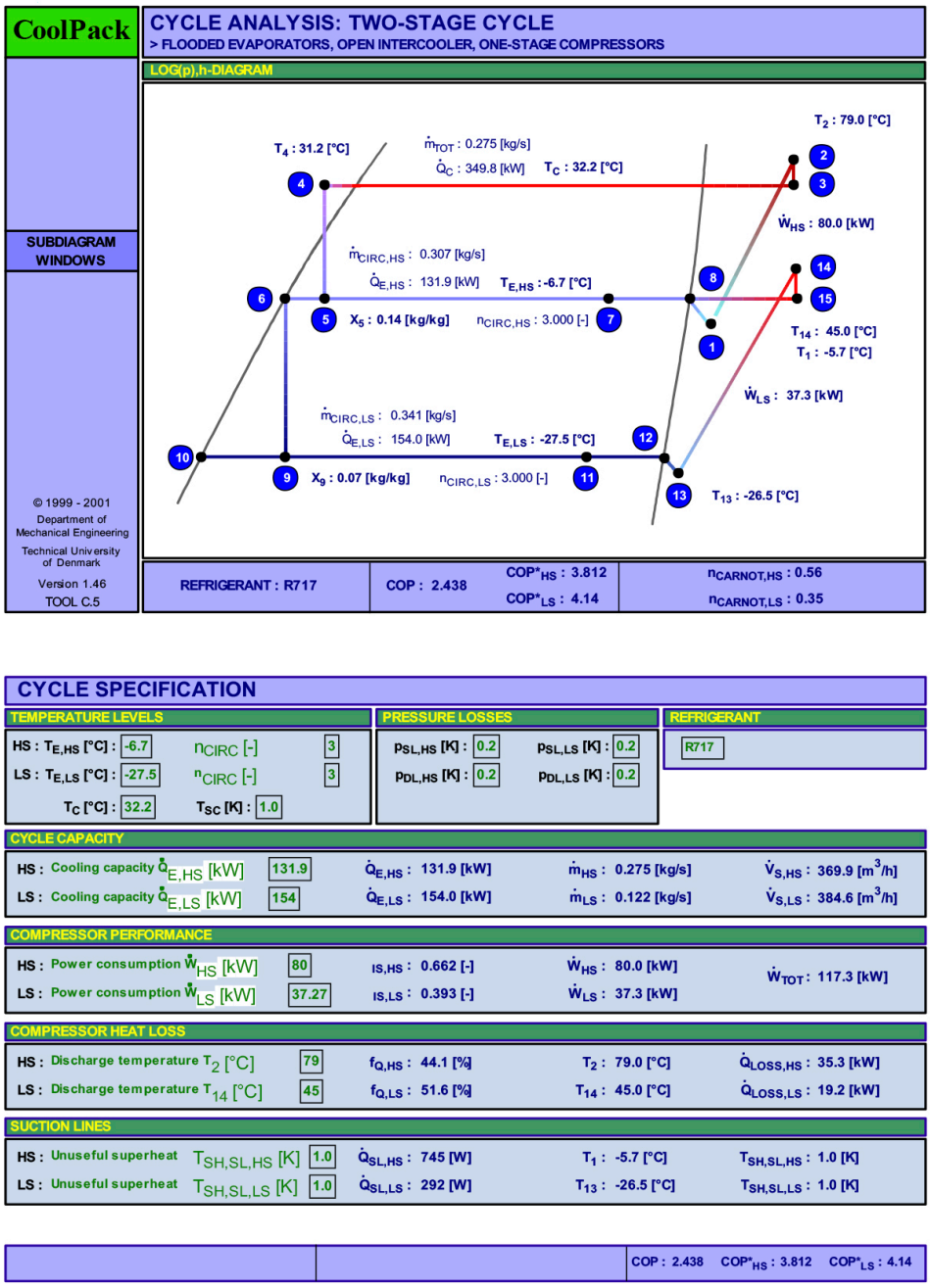


Figure 21. NH_3 two-stage with booster reciprocal and high stage screw compressor.

Appendix E: Illustrations of the Project

Figure 22. Main facility.

Figure 23. New facility low temperature warehouse.

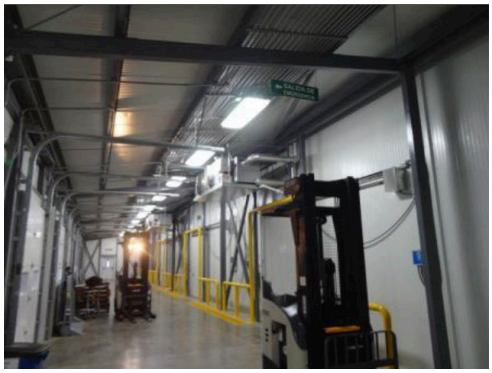


Figure 24. Dock area.

Figure 25. Low temperature area.

Figure 26. Rear view of the facility.

Figure 27. Low temperature CO_2 piping.

Figure 28. Evaporative condenser NH_3 high side.

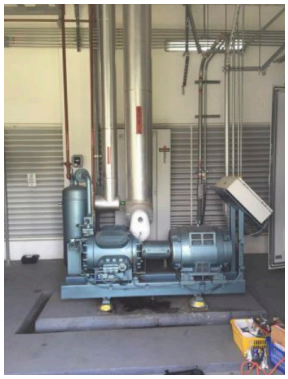


Figure 29. CO_2 HPO24 compressor low side.

Figure 30. NH_3 H33HT screw compressor high side.

Figure 31. Thermosyphon/HP receiver.

Figure 32. Low temperature CO_2 recirculation package.

Figure 33. CO_2/NH_3 heat exchanger and receiver.