Implementation of low pressure, LOW GWP refrigeration system for medium and low temperature applications

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Abstract

HCFC and HFC based refrigerants are actively being phased out globally by government regulatory bodies due to known issues with their ozone depletion potential (ODP) and global warming potential (GWP). A new refrigerant fluid class, Hydrofluoro olefins (HFOs), has recently shown significant promise as a potential class of low GWP and zero ODP fluid.

As owners and operators of refrigeration systems which utilize HCFC/HFC refrigerants are faced with the requirement to replace the refrigerant being utilized in their systems due to these new regulations, there is also an opportunity to transition systems to advanced design and control platforms that allow for significant energy and maintenance cost savings. However, owners and operators of refrigeration systems are often slower to adopt new approaches due to their inherent conservative nature.

This paper will report on the energy reduction and technical benefits from a commercial implementation of an advanced controls and design architecture from Oxford Energy Solutions Inc utilizing the low-pressure implementation of an HFO R-513a low GWP, zero ODP refrigeration system. Specifically, the paper will review the total system input electricity consumption before and after the installation of an HFO R-513a system with Oxford's system architecture implementation at a customer site in Ontario, Canada.

Introduction

Beginning January 1st, 2015, the United States EPA issued the final requirements for a 100% phase-out of R22 refrigerants in the United States. The plan issued by the EPA requires a linear year over year reduction of R-22 that can be manufactured or imported to the United States with the result that by year end of 2019 no new or imported R-22 will be allowed in the United States (Powell, 2014). In addition, accelerated phase-out of HFC based refrigerants is also well underway in F-Gas Regulation in Europe, Canada (ECCC https://tinyurl.com/y2rc9btb), California Air Resource Board (CARB) and US Climate Alliance. This has driven the need for a better chemical compound for a refrigerant, one that will exceed existing efficiencies and operating envelopes while maintaining the status of a safe working refrigerant having a low GWP with an Ashrae A1 safety classification.

As a result of extensive research and development by the global refrigeration industry for replacements to HFC refrigerants that exhibit low ODP and GWP, Hydrofluoro olefins (HFOs) have emerged as a commercially viable candidate. The refrigerant utilized in this paper, Opteon™ XP10 by Chemours, HFO-513a, is an HFO class refrigerant developed as a replacement for R-134a in new systems and for retrofitting existing systems. HFO-513a is a blend of 56 wt% HFO-1234yf and 44 wt% HFC-134a. It has been assigned a GWP value of 573 as determined by IPCC's Fifth Assessment Report (AR5) (IPCC, 2013).

The Oxford Energy Solutions platform architecture has been developed to take advantage of this low-pressure refrigerant utilizing modern advances in controls and equipment that allows for an extremely wide range of operation.

ASHRAE Number	R-513A
Composition	HFO-1234yf/R-134a
Weight %	56.0/44.0
Molecular Weight g/mole	108.4
Boiling Point at 1 atm (101.3 kPa) °C	-29.2
Critical Pressure kPa [abs]	3766
Critical Temperature °C	96.5*
Liquid Density at 21.1 °C (70 °F) kg/m3	1185.7
Ozone Depletion Potential (CFC-11 = 1.0)	0
AR5 Global Warming Potential	573
ASHRAE Safety Classification	A1
Temperature Glide °R	0

Table 1: Thermodynamic Properties (Chemours, 2019)

*Note: The relatively high critical temperature of R-513a is very advantageous when operating in warmer climates.

Base Case

The customer site, Vanessa Meats, is a mid-sized butcher and deli operation in Vanessa Ontario. The owners required modifications to the refrigeration system at the site to accommodate business expansion and also to begin the phase-out of R22 refrigerant based systems.

The base case prior to the project implementation was composed of 18 separate refrigeration systems totalling 50.5 kW of total input power capacity. The project entailed replacing systems 5 through 8 in Table 2 below, representing 14 of the 18 individual units and a total of 30.2 kW of installed load, approximately 60% of the total system capacity.

s #	***Refrigerant Type***	***Saturated Suction Temperatures (°F)***	***Saturated Condensing Temperature (°F)***	***End Use***	***Qty.***	***kW***	***kW total***
1	404	-10	120	freezer	1	4.4	4.4

2	404	-10	120	freezer	1	4.4	4.4
3	22	28	120	cooler	1	6.0	6.0
4	22	28	120	cooler	1	5.5	5.5
5	22	45	120	a/c	1	5.8	5.8
6	134	20	120	gravity deli meats	2	1.4	2.8
7	407c	25	120	production/sausage rooms	1	5.6	5.6
8	290	-10	120	glass door upright freezers	10	1.6	16.1
							50.5

Table 2: Base Case System Configuration

Energy Efficient System Architecture

The design architecture presented here is based on fundamental refrigeration system design principals which include a) lowering the required system head pressure, b) lowering the required compressor ratio leading to a reduction in the required internal heat of compression in the system, and c) maintaining the lowest possible pressure differentials throughout the system in order to achieve a long-term platform that targets zero refrigerant leakage.

	Replacement Equipment Identification	***Calculated Heat Load (BTUH)***	***Saturated Suction Temperatures (°F)***	***Defrost Heaters FLA, 230 VAC (Amps)***	***Room / Case Temp (◦C)***
1	Brema B- 5 Door Freezer	5,500 LT*	-7	16.5	-18
2	Brema 2 Door MT	1200	29	N/A	3
3	Boston 8ft MT	9000	20	N/A	
4	Chicago 8ft(Section 1) Gravity MT West	4000	20	N/A	3
5	Chicago 8ft(Section 2) Gravity MT East	4000	20	N/A	3
6	RTE Cooler	8800	29	6.25	3
7	Fresh Cooler	12200	29	8	3
8	Cut Room	14000	35	N/A	8
9	Sausage Process Room	21000	25	16	3
10	Rear Process	22000	35	N/A	8
11	Blast Cooler	12,000 **	25	8	3
12	Fermenting/Play	12,000 **	25	8	3

Table 3: Retrofit equipment summary
*Note: adds 9,000 to MT Load
**Note: Depending on Load

These features are partially achieved with Copeland's Scroll compressors and the Emerson EXV platform (Emerson, 2019) which enables the system to utilize a low pressure HFO refrigerant in this application (HFO-513a) and operate a very low compression ratio. The system operates at an average low-pressure range of 4 psig, an average medium pressure range of 20 psig and 85 psig discharge pressure in summer (60 psig in winter). Multiple low temperature loads are controlled from a single low temperature compressor which operates at different speeds based on load requirements. As a result, the freezers and low-temperature cases operate at extremely low compression ratios. This approach reduces secondary heat influences such as heat of compression and motor heat on the low temperature loads by as much as 80%.

One of the key design features to the platform is the removal of non-essential valves. Elimination of liquid, hot gas defrost and/or suction line solenoid valves, no EPRs or mechanical head pressure controls valves eliminate the pressure drop and the inefficiencies that result. Further advantages relate to leak reduction through the elimination of gasketed surface valve connections/fittings and reduced maintenance cost.

Superheating of the refrigerant vapor prior to the compressor is a standard requirement in refrigeration systems to ensure there is no liquid entering the compressor which can cause mechanical damage to the compressor. Traditionally suction vapor superheating is achieved utilizing a portion of the evaporator system itself, however this reduces by design the overall refrigeration system efficiency by reducing the amount of latent heat work done in the evaporator(s). Instead, in the approach presented here, evaporator superheat is kept to a minimum through proprietary control of the evaporator expansion valves to minimize superheating in the evaporators, and instead acquires the required superheating utilizing the built in heat exchanger in the suction accumulator.

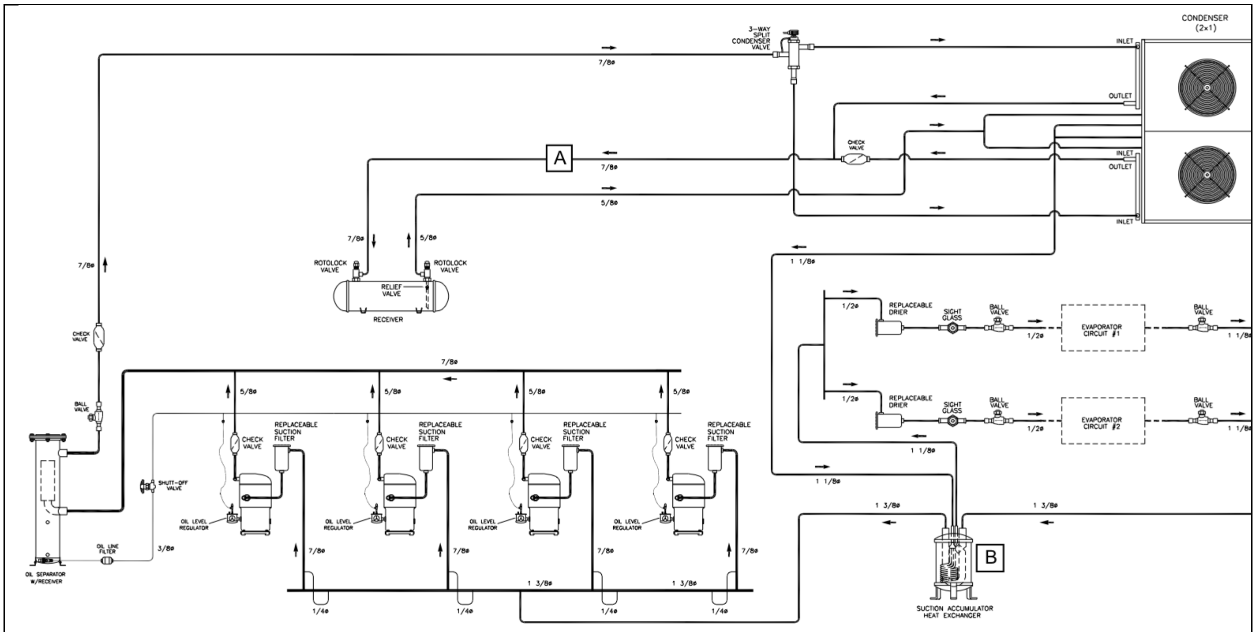
The suction accumulator heat exchanger (Refer to B in Figure 1 below) extracts heat from the liquid refrigerant providing significant liquid sub-cooling benefits, while at the same time providing the required vapor superheating. This approach essentially provides free liquid subcooling from the architecture of the system further improving the overall refrigeration system efficiency. The liquid subcooling generated in the system averages 15-30 degrees Fahrenheit without the need to expend any additional energy to achieve this subcooling (i.e. supplemental cooling units). The total sub-cooled gain in the system is achieved by reducing the amount of superheat in the evaporators, the condenser's additional sub-cooled loop and then through the heat exchange in the accumulator to further increase system subcooling – and capacity.

With the reduced super heat in the evaporators, there is now an economic and system benefit to installation of an additional piping loop through the condenser to further increase system subcooling – and capacity (Refer to A in Figure 1 below). The average subcooling gain with this architecture results in liquid temperatures around 55F with a saturated condenser temperature of 85F. As a rule of thumb, 30F of subcooling results in a 16% gain in system capacity which is established at no extra input costs. The liquid sub-cooling also guarantees a high energy liquid at the inlet of every expansion device without the negative effects of flash gas in the liquid line that occur when the refrigerant is too close to Saturation.

Utilizing a lower pressure refrigerant has many additional benefits. Foremost there is overall much less mechanical stress on key system components such as piping, fittings, gaskets and connections, which translates to substantially lower risk of potential refrigerant leaks over time.

The reduced system refrigeration charge due to the architecture, improved safety and system training all add to the benefits of this low-pressure refrigerant application.

Utilization of electric defrost as needed allows for a much simpler system architecture which helps to maintain a low condensing pressure and allows for dramatically less piping and valving as a common liquid and suction header can be run throughout the facility instead of every circuit coming back to the compressor station area. Electric defrost eliminates the need for hot gas defrost which means dramatically less valving at the rack which eliminates one of the major causes of conventional refrigeration system leakage.



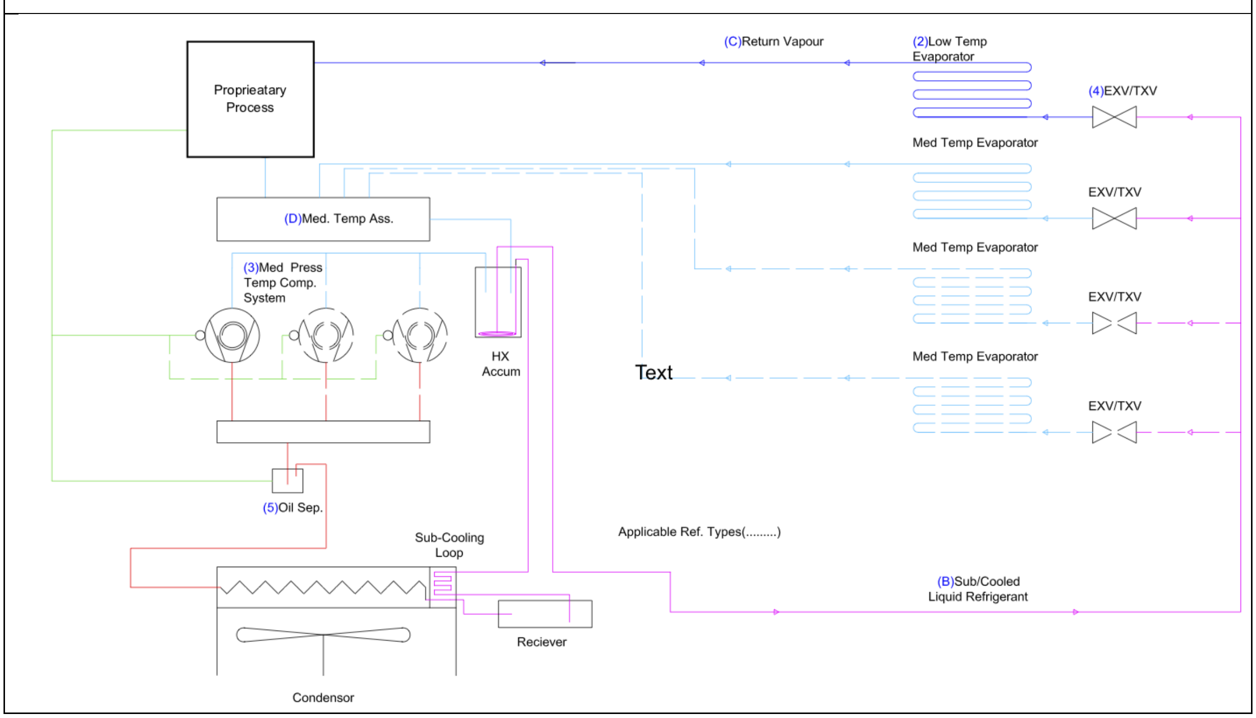


Figure 1: Typical System Architecture Layout Schematics

Analysis and Results

The measurement and verification process followed the International Performance Measurement and Verification Protocol (IPMVP) and entailed measurement of the total site incoming input power utilizing a true-RMS power logging electrical meter before and after the project was implemented. The meter utilized for both pre and post project measurements was a Candura EnergyPro. The incoming electrical supply consisted of a single-phase three-wire 240 V supply. Current and voltage on all three lines were monitored at five second intervals for 35 days starting February 21st, 2019 prior to the new refrigeration equipment installation and for 10 days post-project completion starting 13th September 2019. Temperature data was retrieved utilizing the Government of Canada historical climate data website for Brantford Airport, Climate ID #6140942.

The power data taken during the pre and post project data periods was utilized along with historical climate data from Brantford Airport to determine annualized energy consumption for pre and post project periods.

From the hourly average power values in Figure 4, which is derived from the data in Figure 2, it is clear that there are natural daily operational cycles in the facility. To differentiate between the daily operational cycles in the facility, it was necessary to derive temperature to power correlations for three separate cases of data both pre and post project:

1 – Mondays through Saturdays, hours 9 am through 4 pm.
2 – Mondays through Saturdays, hours 5 pm through 8 am.
3 – Sundays all day.

These temperature to power correlations were utilized to determine annualized electrical consumption data utilizing 2018 historical hourly average temperature values from the historical climate data. The preliminary Base Case data was then adjusted for the increase in refrigeration footprint from a pre-project value of approximately 27.9 m^2(300ft^2), plus gravity cases and freezers, Table 2, #6-8, to a post-project value of 131.0 m^2 (1,400 ft^2), plus cases and freezer, Table 4.

	Base Case			Energy Efficient Case			Project Savings		
	kW	kWh	Cost ($)	kW	kWh	Cost ($)	kW	kWh	Cost ($)
Preliminary	25.4	222,305	$ 33,346						
Adjusted	119.3	1,044,834	$ 156,725	18.8	165,093	$ 24,764	100.4	879,741	$ 131,961

Table 4 – Energy and Cost savings Summary

	Pre – Project	Post – Project
Cost/m^2	$ 1,196.44	$ 189.05
Cost/Day	$ 429.38	$ 67.85

Cost/hour	$ 17.89	$ 2.83

Table 5 – Normalized Cost Savings (Note: All electricity cost figures are estimated utilizing an all-in electricity rate of $0.15/kWh)

It is notable that the post-project data was measured during a relatively warm week in September 2019, when the average outdoor ambient temperature (OAT) was 18.5 °C, relative to the pre-project data that was measured in late February/early March 2019, when the OAT was approximately -3.1 °C.

For example, during the post – project data collection period on Thursday, March 28th, 2019, the average rate of power consumption per hour (18.9 kW) at an average temperature of 11.4 °C was almost 20% lower than the pre-project average rate of power consumption per hour (23.1 kW) on Tuesday, September 17th 2019 at 6.3 °C (50% lower temperature) – before accounting for an increase of over 4.5 times in refrigerated area.

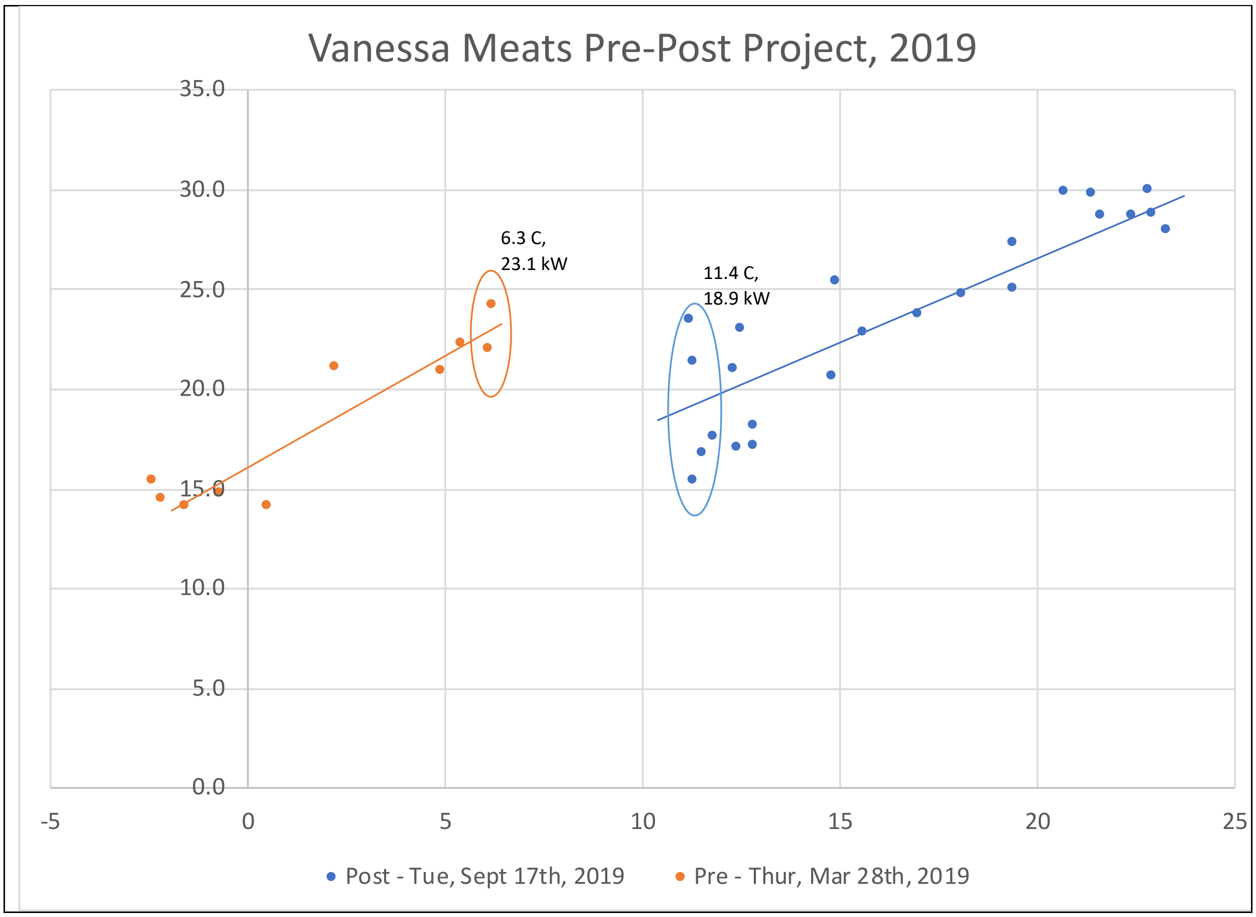


Figure 2 – Pre and Post project Temperature and Power Comparison – Single Day

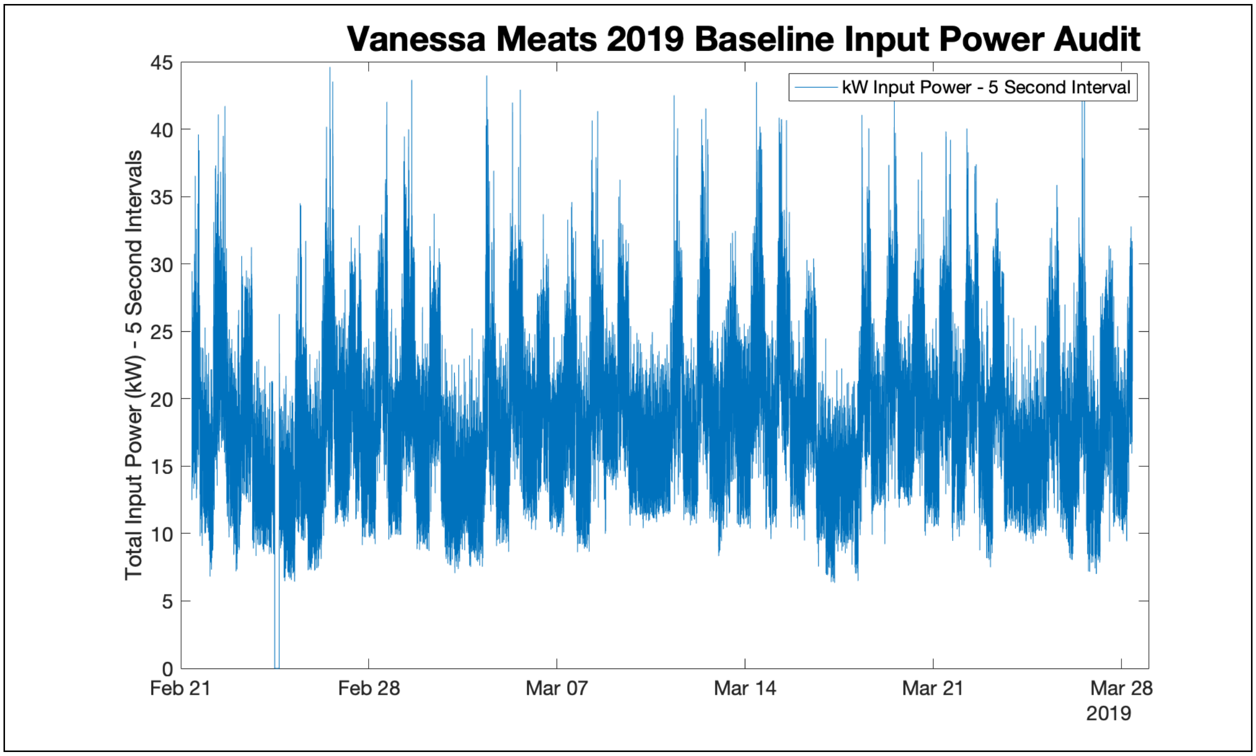


Figure 2 – Pre Project 5 Second Total Input Power

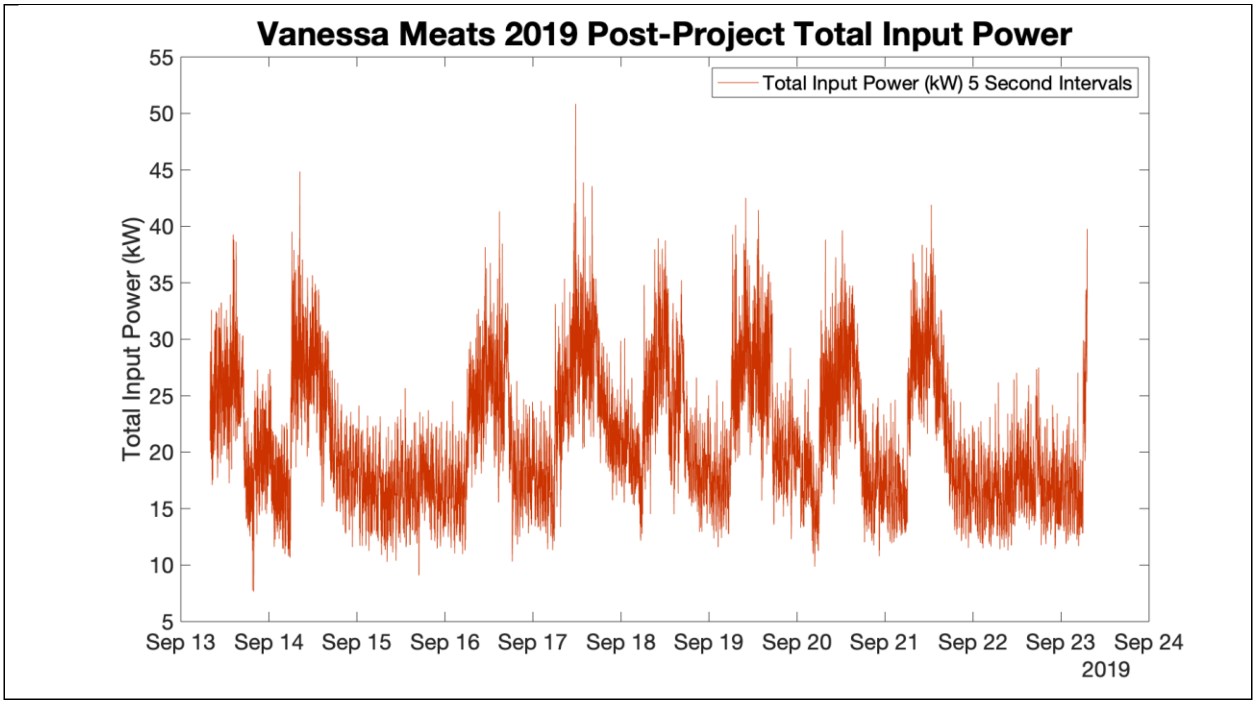


Figure 3 – Post Project 5 Second Total Input Power (kW)

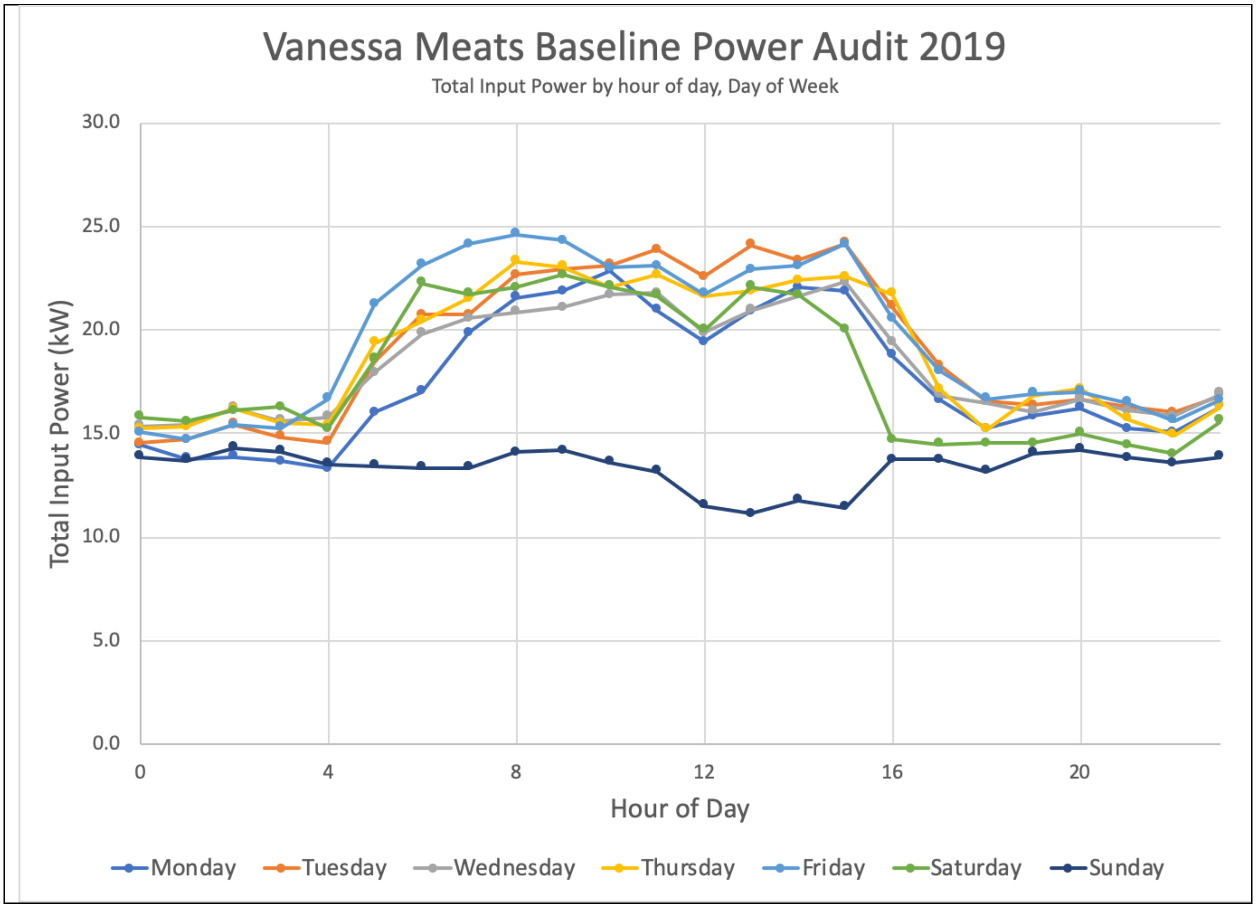


Figure 4 – Average Input Power by Hour of Day, by Day of Week – Pre Project

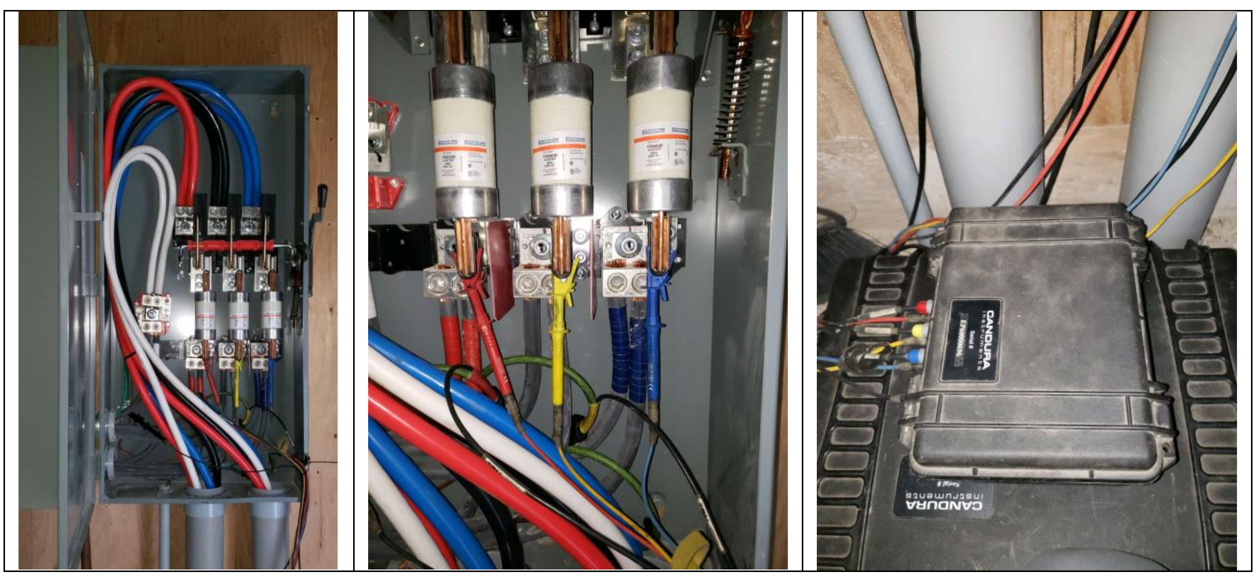


Figure 5 – Power Meter Installation – Vanessa Meats

Figure 6 – Pre Project Installation

Figure 7 – Post Project Arneg Refrigeration Case

Figure 8 – Post Project Arneg refrigeration case installation

Figure 9 – Post-Project Condensers Oxford Refrigeration System


Figure 10 – System Controls – No Mechanical/Electrical Room Required

Conclusions

Due to the combined system approach and operating characteristics of the electronic expansion valve, the low compression ratio capability of the compressors, and boosting the low temperature system into the medium temperature architecture, with proper control strategy and refrigerant control measures, the system is able to take advantage of some key benefits behind a low pressure refrigerant (R513a), and the untapped abilities of the scroll compressors that operate so efficiently at this low pressure differential.

The pre – project versus post – project measurement and verification analysis has determined a dramatic reduction over ***<u>80%</u>*** of total required input electricity power and resulting electricity operating cost savings.

The fundamental system design approach is to maintain the lowest operating refrigerant pressure differentials as technically possible at all times through advanced controls and sensor technologies while limiting internal heat generation to a minimum. The overall system architecture does not rely on one main component as a key contributor to the success of the system efficiency. The Oxford Energy platform relies on the combined net effects of all the individual benefits of these system components working together to achieve improved system efficiency with a new HFO blended low pressure refrigerant. Combined with a low pressure drop system and compound refrigeration architecture design built on a zero – leak rate approach to provide a system that is setting a new standard in reliability, low maintenance, and energy efficiency.

This platform also allows refrigerant piping architectures that operate at dramatically lower overall system charge to meet or exceed impending refrigerant regulations across North America including Quebec and California. Due to the lower operating pressures, the medium and low temperature architectures can be exempt from existing governing body regulations for field certified pressure tests and submissions such as TSSA in Ontario and other governing provinces.

In the future, areas of study will include the investigation of other low-pressure, lower GWP refrigerants and the application of this approach to refrigeration system design approach to larger installations as well as the potential to utilize different piping materials to promote a cleaner installation such as a crimped/compressed fitting system.

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Over the last 2 decades, Ben Kungl, President of the Oxford Group of Companies has been leading the refrigeration industry through innovation, ingenuity and proficiency. Over the course of his professional career, his work has focussed on developing highly efficient and integrated platforms through the careful design, installation, service, and assessment of heat and energy transfer systems. This specialized approach unifies an advanced controls mindset and base refrigeration modelling. In that capacity, his primary markets are the agricultural, commercial, pharmaceutical, mining, medical, industrial, food retail and warehousing industries. Bens aptitude and progressive approach has led to extensive success across North America

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