REFRIGERATION ENERGY SAVINGS WITH FLOATING HEAD PRESSURE

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ABSTRACT

Most refrigeration systems operate at a higher pressure than necessary. Several systems are available that allow head pressure and therefore condensing temperatures to float. The purpose of this study was to determine if one system that uses a liquid pump is a viable technology in reducing the energy use of a supermarket refrigeration system.

This paper reports the results of monitoring two supermarkets using this system with a 1/5 horsepower liquid pump. In one supermarket 103 horsepower of compressors were submetered for 105 days before installation and 233 days after installation. Results showed an average 23% compressor energy savings as well as a 4.2 °F reduction in case temperatures. The savings were observed from August, 1987, through March, 1988. Metered energy use for a full year will be available in August 1988.

In the other supermarket six refrigeration system parameters, including compressor suction and discharge pressures and temperatures, were measured continuously for two weeks. A 7.5 horsepower compressor serving 136 feet of produce case was monitored under a range of operating conditions. Results show savings between 14 and 42% depending on ambient temperatures.

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INTRODUCTION

Most refrigeration systems operate at a higher pressure than necessary. This higher pressure helps the system work on the hottest days of the year. However, significant savings are available by reducing the pressure during cooler weather.

There are several approaches to realize refrigeration savings, such as floating setpoint controls, variable speed compressors, electronic expansion valves, microprocessor control, and floating head (compressor discharge) pressure with a liquid subcooler or a liquid pump.

This paper will report results from monitoring one system for reducing refrigeration energy use. The system allows head pressure to float with ambient temperatures. It also includes a liquid pump to maintain refrigerant circulation and reduce the potential for flash gas. The results describe submetered compressor energy use at one supermarket and monitored refrigeration system operating conditions at another supermarket.

No endorsement by Oregon State University or the Oregon Department of Energy is intended.

BACKGROUND

Most vapor-compression refrigeration systems have a compressor, a condenser, an expansion valve, and an evaporator, as shown in Figure 1. In the evaporator the cool liquid refrigerant evaporates at a low pressure absorbing heat to provide cooling. From the evaporator the refrigerant gas flows to the compressor, where it is compressed to a higher temperature and pressure. The hot refrigerant then flows to the condenser where, as the name implies, the gas is condensed back into a liquid. Heat is given off to the outside air or water that goes to a cooling tower or down the drain. The cooled liquid refrigerant returns to the evaporator and the cycle begins again.

Of the two systems studied, one had an air-cooled condenser and the other had an evaporative condenser. The pressure typically is controlled with a pressure control valve and condenser fan switch settings. This valve is either removed or disconnected to allow the pressure to "float" or automatically follow the outside air temperature. Since the temperature of condensation varies directly with the pressure, the condensing temperature also varies. However, it stays about 15 °F warmer than the outside air so it can give off heat.

The savings depend on how much the system pressure can be reduced in both air- and water-cooled refrigeration systems. The energy savings were found to be about 1 percent for every degree the condenser temperature falls. The savings could be more than 40 percent in the winter; less in the summer.

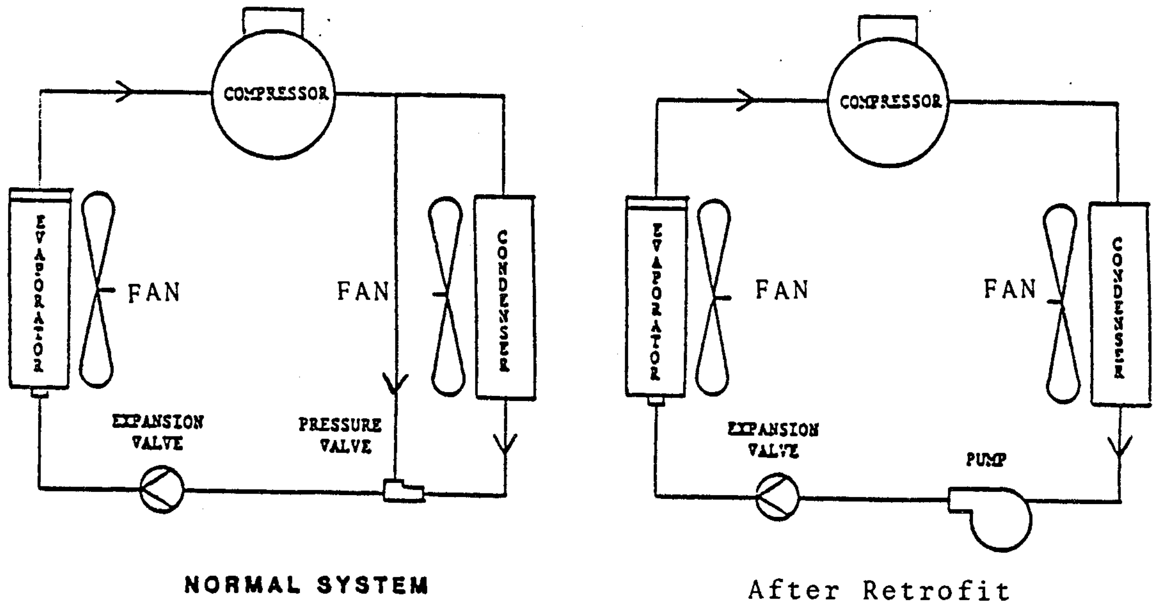

Figure 1. Refrigeration system retrofit.

However, during cold weather the pressure might fall too low. The symptoms are flash gas before the expansion valve and inadequate refrigerant circulation and capacity. The liquid pump is installed (Figure 1) between the condenser and expansion valve to reduce or eliminate these potential problems. It increases the pressure of the liquid refrigerant by 7 to 12 psi, thereby raising the flash temperature and improving circulation. With less flash gas more refrigerant per unit volume is supplied to the evaporator, thereby increasing capacity.

The small centrifugal pump operates under high pressure and has a magnetic drive so that there are no rotary or shaft seals that might leak refrigerant. Also it does not significantly restrict the refrigerant flow when it is off. The pump is rated at 1/5 horsepower, and used 246 watts.

Energy is saved in two ways:

1. **Compressor power.** It takes less power to compress the refrigerant to lower pressure (inlet conditions being equal).

2. **Efficiency.** The cooler liquid refrigerant has more cooling capacity. At lower pressure the system thermodynamic efficiency and the compressor volumetric efficiency are higher. With higher capacity and efficiency, the compressor runs less.

METERING PROJECT

The goal of this project is to determine whether floating head pressure with a liquid pump is a viable means of reducing the electrical energy consumption of a supermarket refrigeration system.

Project Description

To determine the effect on electrical energy use, 80 percent of the compressors at Supermarket One were submetered by Portland General Electric (PGE) 105 days before and 233 days after the store's refrigeration system was retrofitted with the liquid pump. The store's case temperatures were also checked before and two times after the retrofit.

The retrofit includes floating the compressor head (discharge) pressure and installing 1/5 horsepower pumps that increase the pressure of the liquid refrigerant between the condenser and the expansion valve.

The refrigeration system at Supermarket One consisted of a typical range of loads: produce, beverage, and frozen food cases: walk-in freezers and coolers; and refrigerated food preparation areas. The total rated output of compressors meeting this load was 128hp. Of this, 103hp was submetered to determine the effects of the retrofit on this store. The meters were provided by Portland General Electric.

The "Before" data was accumulated from April 9, 1987 to July 23, 1987 totaling 105 days. The "After" data was accumulated from August 10, 1987 to March 30, 1988 totaling 233 days.

To determine how effective the retrofit was in saving energy the "Before" average daily compressor energy consumption was compared to the "After" average daily energy consumption.

Results

The store's refrigeration system temperatures were checked once before the retrofit and on two occasions after. The observed temperatures are shown in Table I. The case temperatures were reduced a total of 82 °F (284-202), for an average case temperature reduction of 5 °F per case (82 °F/16 cases).

Comparison of the "Before" and "After" data in Table II shows a 23 percent reduction in compressor energy use. After the retrofit the store's total electrical load decreased by about 8 percent and case temperatures were reduced an average of 5 °F. Several case thermostats were set to lower temperatures following the retrofit. The effect of case temperature on refrigeration load is discussed in the next section.

Table I: Average temperatures (°F).			
System		Before 7/6/87	After 8/25/87
1.	Produce	44	35
2.	Produce cooler	48	45
3.	Beverage cooler	36	39
4.	Meat cooler	40	31
5.	Multi deck meat freezer	30	24
6.	Glass door freezer	-3	-6
7.	Glass door ice cream	-4	-9
8.	Wide island freezer	-7	-28
9.	Frozen food freezer	-9	-14
10.	Dairy walk in	38	39
11.	Self serve meat	27	19
12.	Multi deck deli	35	36
13.	Multi deck dairy	37	35
14.	Glass door frozen food	-8	-12
15	Glass door frozen food	-11	-9
16.	Frozen food	-9	-23
Totals		284	202

Table II: Compressor metering.				
	Before	After	Savings	%
Days	105	233		
Time (hours)	2,527	5,592		
Total use (kWh)	170,890	290,200		
Daily use (kWh/day)	1,623	1,245	378	23%
Peak demand (kW) *	90	72	18	20%
Average Demand (kW)	67.6	51.9	15.7	23%

* Peak demand was the highest demand recorded in a 30-minute interval during each period. Average demand is the average power use over each period, including equipment cycling time.

Figures 2 and 3 show how the ambient temperature affected the average rate of compressor energy use (average power in kWh/hr). The graphs include 14 time periods between meter readings from 4/9/87 to 10/15/87. Note that the average energy use in Figure 3 follows the ambient temperature more closely than in Figure 2 before the retrofit.

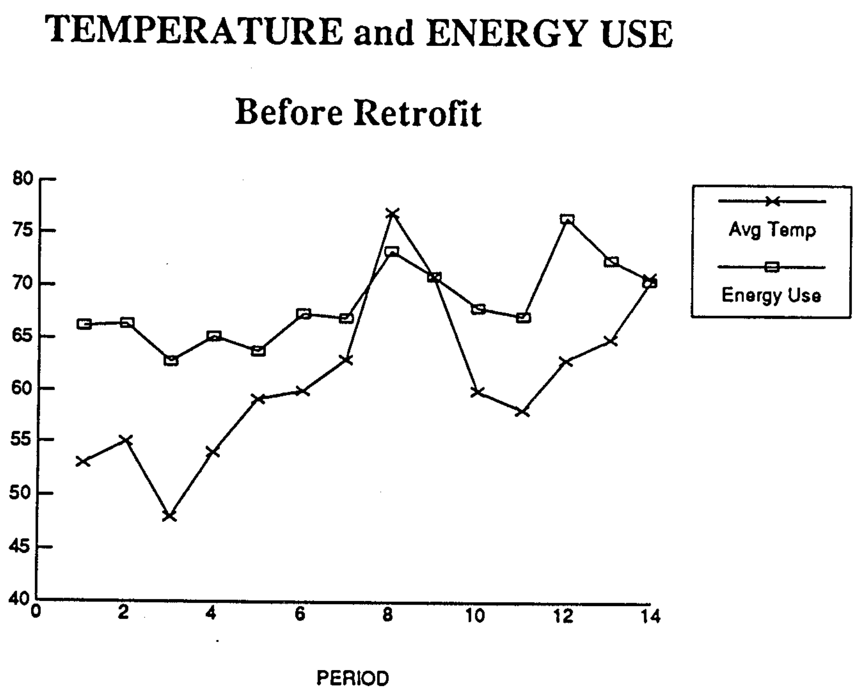

Figure 2. Ambient temperature and compressor energy use before retrofit.

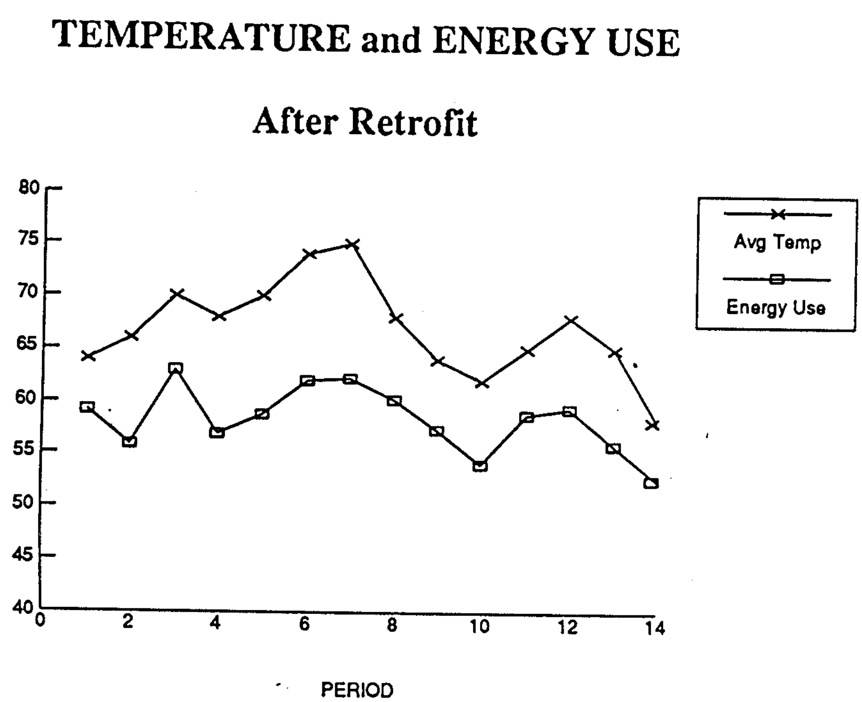

Figure 3. Ambient temperature and compressor energy use after retrofit.

Energy savings compared to the base period were observed to be greater at lower ambient temperatures as shown in Table III. The greatest savings of 28% occurred when the average ambient temperature was 45 °F. Correcting the before and after periods from Figures 2 and 3 for an average ambient temperature of 50 °F, we estimate annual compressor savings to be about 23%.

Table III. Metered compressor energy use.

Period	Energy Use (kWh/day)	Percent Savings	Average Temp(F)
4/9/87-8/10/87	1693	0*	63
8/10/87-10/15/87	1395	14	67
10/15/87-1/27/88	1192	27	45
1/27/88-3/30/88	1175	28	45

* Base period

MONITORING PROJECT

A produce case in Supermarket Two was monitored continuously for 10 days in March, 1985. The goal was to better understand how a refrigeration system responds to floating head pressure over a range of ambient temperatures.

Project Description

Supermarket Two near Portland, Oregon, has 17 compressors with a total rating of 105 horsepower. In May, 1984, the owner had the system modified to allow head pressure to float, including the installation of liquid pumps.

The 136-foot produce case was selected because it should respond faster to changes than low or medium temperature cases or walk-ins with larger masses of product. The case was served by a 7 1/2 HP reciprocating compressor with a common air-cooled condenser. The refrigerant was R-12. The system was controlled by a combination of suction pressure switches (set with cut-in at 33 psig and cut-out at 18 psig) and a thermostat (set at 40 °F). The thermostat generally reached its setpoint first.

The following data were recorded: (1) discharge pressure (psig); (2) compressor plus pump power (kW); (3) suction temperature (°F); (4) condensing (liquid) temperature (°F); (5) ambient temperature (°F); (6) suction pressure (psig); and (7) cumulative kWh.. Over 2200 cycles were recorded. One refrigeration cycle each hour was selected and entered into a microcomputer spreadsheet program for reduction and plotting.

Results

The results of the monitoring project are summarized in Table IV. Run #1 was the base run with no liquid pump and a 150 psig discharge pressure. The "condensing" temperature was the measured liquid temperature following the condenser. It is the temperature of the liquid refrigerant that is delivered to the expansion valves. (Note that the liquid is subcooled below the temperature at which it condenses.) For the base case the liquid temperature was 94 °F while the ambient (outside) temperature was an average 41 °F.

Table IV. Monitoring project results summary.

Run Number	1	2	2A
Pump Status	OFF	ON	ON
Discharge Pressure (psig)	150	95	88
Case Temperature (°F)	43	37	40
Ambient Temperature (°F)	41	43	40
Condensing Temperature (°F)	94	51	51
Run Time (%)	46	40	28
Compressor Power (kW)	6.2	5.6	5.8
Average Power (kW)	2.85	2.24	1.62
Energy Savings (%)	0	21	43

The compressor power was measured and recorded in Table IV. The actual running time is recorded as a percentage of "run" time to total elapsed time. To get the average power the compressor power is muliplied by the Run Time.

Then the pressure control valve was bypassed to allow the head pressure to float and the liquid pump was turned on. The results for this case are recorded in Table IV as Runs #2 and #2A. The results for Run #2 show that the discharge pressure dropped from 150 to 95 psig with a 21% compressor energy savings. However, the measured case temperature dropped from 43 to 37 °F. This might be caused by reduced flash gas in the liquid line, better feeding of the expansion valves, or inconsistent control.

We can estimate the effect of lower case temperature on energy use by assuming that the refrigeration load is proportional to the temperature difference between the case (40° F) and the surrounding store temperature (70 °F). Therefore a 6 °F lower case temperature might increase the refrigeration load 20% [6/(70-40)]. With a typical coefficient of performance (COP) between 2.0 and 2.5 the compressor load might be 8 to 10% higher.

To correct for the lower case temperature the thermostat was increased about 3 °F. The data for this run was recorded as Run #2A. With the case temperature at about 40 °F compressor savings of 43% were measured. A more detailed study is recommended to verify the higher savings due to the uncertainty introduced by adjusting the thermostat.

Electricity bills for the year following the installation compared to the previous year showed an energy savings of 214,000 kWh and a cost savings of $11,000. This is about a 12.5% savings for total store electricity use. The refrigeration system used approximately 50% of the total electricity consumed. Therefore, the refrigeration savings would be about 25%, which is consistent with the monitored results.

The ambient temperature, condensing temperature, and average energy use are shown in Figure 4 over a 24-hour period for the before-retrofit base case. Note that the condensing temperature is relatively unaffected by changes in the ambient temperature. However, for Run #2 with floating head pressure the condensing temperature in Figure 5 floats about 10 °F above the ambient temperature.

RUN #1

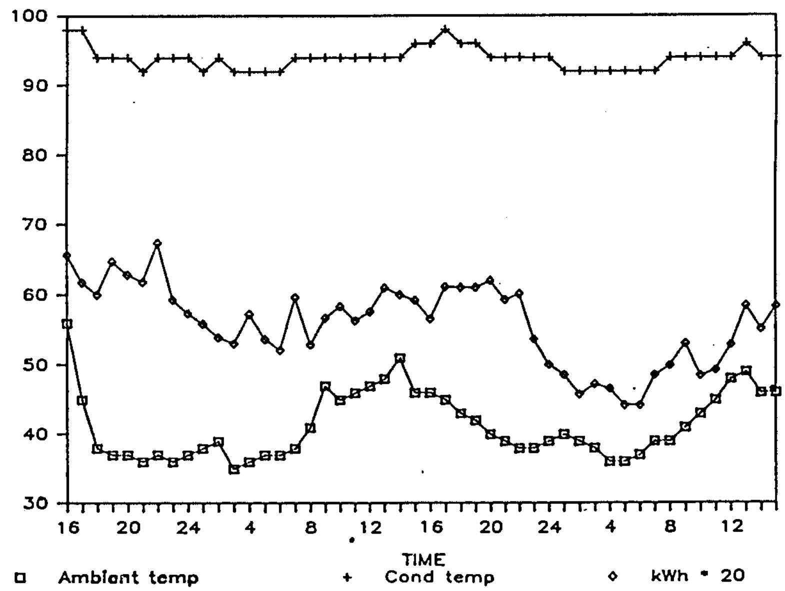

Figure 4. Ambient temperature,condensing temperature, and compressor energy use before retrofit.

RUN #2

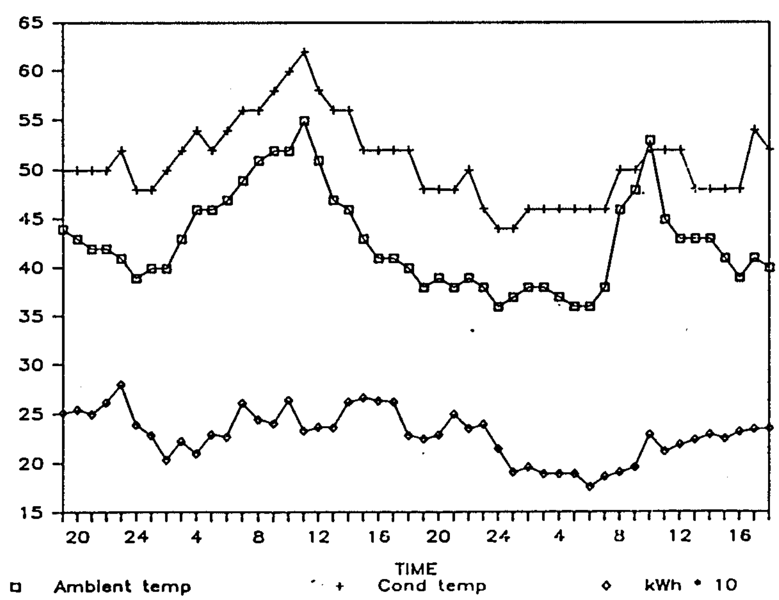

Figure 5. Ambient temperature, condensing temperature, and compressor energy use with floating head pressure.

While average energy use is lower with lower condensing temperatures, the instantaneous compressor energy use in Figure 5 does not follow changes in ambient temperature. This is confirmed in Figure 6 that shows significant scatter in a plot of energy use versus ambient temperature for one cycle each hour. Instantaneous energy use does not appear to increase directly with ambient temperature. This indicates that the refrigeration system may not be in equilibrium over a cycle time of about 6 minutes. For example, it takes several cycles after defrost before the system stabilizes.

Compressor power versus condensing temperature is plotted in Figure 7. The intermediate condensing temperatures were simulated with a variable head pressure valve. The upper line was drawn through the average of the data points for each condensing temperature. It shows 0.9% compressor savings for each degree the condensing temperature can be reduced. The lower curve is drawn through the average compressor power for Runs #1 and #2A, which represents the upper limit of 1.1% savings per degree.

Because the monitoring project only lasted 10 days we made an attempt to estimate annual savings using local bin weather data for the hours of occurrence at each 5-degree temperature range. The results are shown in Table V using the curves from Figure 7 for both the maximum savings of 1.1%/°F and the minimum of 0.9%/°F.

Table V. Annual energy estimate using bin weather data.

Temp	Bin Hours	Power and Energy Use at: Min Savings kW	Min Savings kWh	Max Savings kW	Max Savings kWh
102	1	2.80	3	2.80	3
97	5	2.80	14	2.80	14
92	22	2.80	62	2.80	62
87	54	2.80	151	2.80	151
82	132	2.80	370	2.75	363
77	210	2.80	588	2.63	552
72	363	2.75	998	2.50	908
67	571	2.63	1502	2.37	1353
62	972	2.51	2440	2.25	2187
57	1299	2.39	3105	2.12	2754
52	1300	2.27	2951	2.00	2600
47	1302	2.16	2812	1.87	2435
42	1270	2.04	2591	1.74	2210
37	735	1.92	1411	1.62	1191
32	332	1.80	598	1.49	495
27	114	1.68	192	1.36	155
22	42	1.56	66	1.24	52
17	21	1.44	30	1.11	23
12	10	1.32	13	0.98	10
7	2	1.20	2	0.85	2
Total	8757		19,899		17,520
Savings			19%		29%

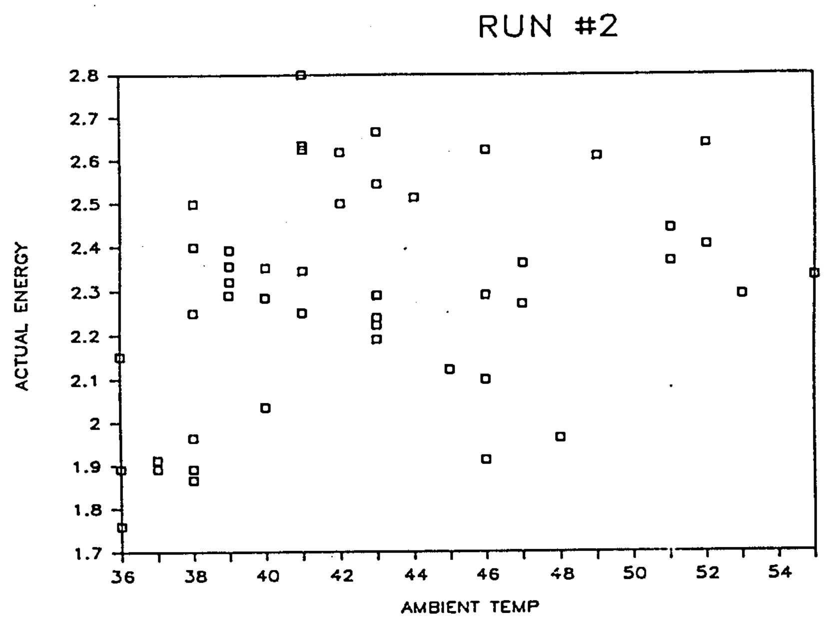

Figure 6. Actual compressor energy use versus ambient temperature with floating head pressure.

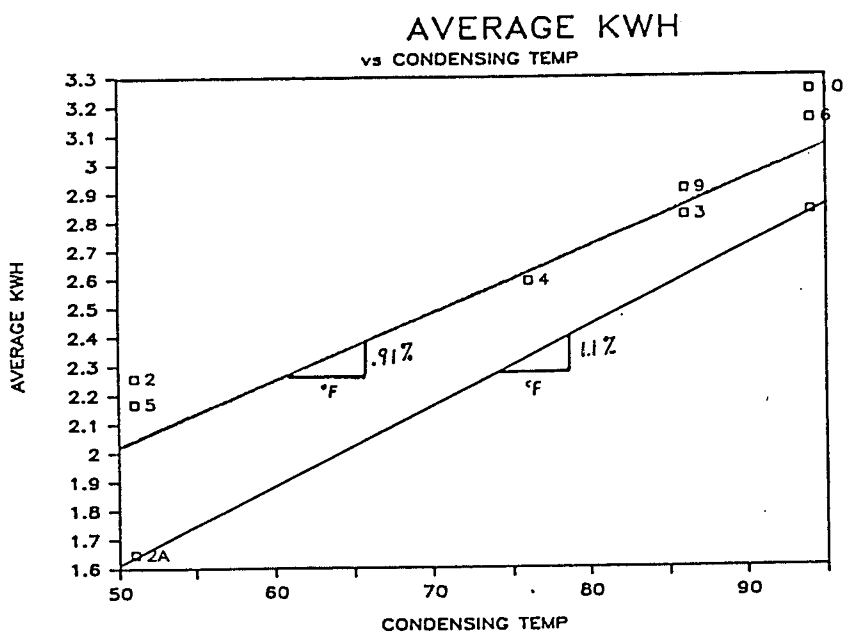

Figure 7. Average compressor power versus condensing temperature.

The annual compressor energy savings are estimated at 19% based on all data points. Annual savings might be as high as 29% if the second run with the thermostat setting increased 3 °F is justified.

Conclusion

Floating head pressures at two supermarkets in western Oregon with the installation of liquid refrigerant pumps reduced compressor energy consumption between 19 and 43 percent. Case temperatures after the retrofit averaged 3 to 5 °F lower. The retrofit system was an effective energy conservation device in this application.

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