A STUDY TO DETERMINE THE ENERGY IMPACT OF ADDING POLARSHIELD TO AIR CONDITIONING SYSTEMS

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ABSTRACT

PolarShield is a polarized refrigerant compressor oil additive containing the α-olefin molecule which is a commonly used oil additive to reduce high pressure viscosity breakdown. The manufacturers of this air conditioner compressor oil additive (COA) claim significant energy savings as a result of using their product. The objective of this study was the evaluation of the potential kWh savings that would result from the addition of an α-olefin molecule COA such as PolarShield to an air conditioner unit operating under typical hot outdoor conditions (95 degrees F). The test was operated in a "before" - "after" manner with each before-after segment operated for twelve days and three tests were conducted - one on a new 2.5 ton system where 1.25 oz. COA was added, one on a 5 ton older unit where 2.5 oz. COA was added, and a third on the same 5 ton unit where an additional 2.5 oz. COA was added to total 5.0 oz added to the system. The heat and humidity loads were carefully held constant for the before and after time periods, and the air conditioner equipment was allowed to cycle on its thermostat to meet the loads and maintain a stable indoor condition. The results of this series of tests showed no energy savings when the PolarShield COA was used.

INTRODUCTION

PolarShield is a polarized compressor oil additive (COA) that contains the α-olefin molecule. This molecule is a chlorinated paraffin which remains in a liquid state and readily mixes with refrigeration oil. In refrigeration equipment, oil is used to lubricate the moving parts of the refrigeration compressor. The COA is added to that oil. PolarShield sales literature indicates that a product also using the α-olefin molecule was marketed under the registered brand name Frigaid. The company distributes the Frigaid literature as examples of previous successful applications of the PolarShield product. Though the formulations are proprietary and unrevealed, other than that they use the α-olefin molecule which has been patented for this use, it is assumed that the formulations of the COA products PolarShield and Frigaid are similar. Quoting from the company literature, *"Activated polar molecules bond to all metal surfaces displacing oil, carbon and dirt resulting in reduced electric demand at startup due to better heat transfer and lubricity. PolarShield dislodges stagnant, insulative oil and other deposits promoting heat transfer."* [1] The manufacturer further claims in their published literature that they have monitored before-after sites that have shown up to 36% savings in energy use as a result of the installation of their product. [2]

PREVIOUS STUDIES

The very nature of this product, that it is a compressor oil additive and is claimed to attach to the internal metal surfaces of the system and this claimed action is not reversible, requires that the product be tested for energy savings in a before - after type of test. Thirteen summary packets were provided to the Florida Solar Energy Center (FSEC) by the manufacturer of the COA product with each describing a before-after test of the oil additive where a measurement was made on before-after energy use or demand. These reports describe energy monitoring and claim energy savings ranging from 12 to 36% after the COA was added. These tests measured "before" and "after" energy use, however none of these tests measured the load on the air conditioning or refrigeration equipment, that is, none of the tests made sufficient measurement whereby the conditions under which the systems were operating for the before and after periods could be adequately determined. For these reports, no statistical tests were used on the data to determine the validity of any conclusions, nor were any statistical methods used to bring before-after operational conditions to similarity for "apples to apples" comparison. Because of the lack of scientific rigor under which these tests were conducted, it was necessary to consider these thirteen tests inconclusive.

Two test reports were found in the literature where the α-olefin molecule additive was studied for potential energy savings and the load on the equipment, both inside load and outside load, were monitored, and other attributes of the equipment were held constant for the before - and after time periods. These two studies were conducted by nationally recognized laboratories. One study was conducted at the Oak Ridge National Laboratory where three tests were run [3] and the other study was conducted at the Energy Conversion Laboratory of the University of Florida [7].

Oak Ridge Tests

The Oak Ridge test was conducted on a 15 year old, nominal 3 ton single package heat pump. All tests were done at steady state, that is, the unit was not cycled, but run continuously and at the American Refrigeration Institute (ARI) 95 degree outdoor EER rating point, indoor 80 degree F and 51%RH [8]. This test was done with the COA brand named Frigaid [3]. First the unit was run to obtain baseline data, then three ounces of additive were put in. It was run for one day, and then data taken periodically over the next six days using a 15 minute data acquisition period. At day seven, one and one half additional ounces of additive were put in the unit and the unit monitored for two more days. Then for part of day nine, an additional one and one half ounces of additive were put in (for a total of 5.5 ounces added) and data recorded after 6 hours of operation.. A trend in a slight reduction in the Watts used by the unit was measured from 5485 to 5385 Watts from the start to the end of the test. However, the researchers concluded that this 1.8% reduction in energy use was within the plus or minus 2% accuracy of the test and could not be separated from the random experimental error at the 95% confidence level [4]. A paragraph from their paper states:

"Table 2 indicates that no improvement in heat pump performance was measured in our laboratory tests as a result of adding this product to our test unit. The small changes (±2%) in steady state compressor power consumption and cooling capacity shown in Table 2 are most likely attributable to random experimental errors, although a small 2.5% improvement in EER is indicated for 3 ounces of the additive. It is also worth noting that the evaporator air entering/leaving temperatures and the compressor pressure ratio showed no significant change as a result of additive addition. Both of these observations are consistent with no improvement in heat exchanger performance. There was, however, a noticeable, but unquantified, decrease of compressor noise resulting from additive addition."[5]

In a further paragraph they write, "*We also measured no significant decrease in compressor power due to reduced mechanical friction with the additive, even though the compressor appeared to operate with less noise. The compressor pressure ratio increased only slightly as a result of the presence of the additive. Therefore, we conclude that mechanical friction is not reduced by the additive in our system."*[6]

University of Florida Tests

The University of Florida (U of F) conducted a before and after test on a 2 ½ ton air conditioning system. This test was done with the α-olefin oil additive, Frigaid and one ounce per ton was added for the test "after" period, that is, 2.5 oz of the additive was added. These tests were conducted in the University of Florida's (UF) Department of Mechanical Engineering Solar Energy and Energy Conversion Laboratory's environmental control chambers. They were conducted in accordance with the measurement and temperature conditions for the standard ARI/ASHRAE test of air-conditioner/heat pump performance, that is, the outdoor chamber was held at 95 degrees F, and the indoor chamber held at 80 degrees and 51% RH [7]. In addition to the required ARI air- enthalpy measurements for the determination of energy use and COP, the U of F team also measured the pressure of the refrigerant at the compressor discharge (psig) and the pressure of the refrigerant at the compressor suction (psig).

These tests were run at steady state, that is, the compressor was run continuously and the operation of the equipment was stabilized. The ARI tests allow that when the equipment is run for 40 minutes under measured stabilized operation within the specified limits, then one hour of test data is retrieved [8]. This one hour of test data is considered sufficient to characterize the equipment. The UF test went well beyond this requirement. Instead of one hour of test data after stabilization, they took data on the equipment for at least three continuous test hours. In addition, they repeated the "before" test five more times for six separate days of testing. After the oil additive was inserted into the equipment, they ran the test ten times on ten subsequent days. For their before tests, the equipment averaged an energy use of 2.32 kWh across any test hour. For their after tests, the equipment averaged an energy use of exactly the same, 2.32 kWh across any test hour. The data showed less than a half of a percent improvement (0.47%) in the performance of the system (COP) after the additive was put in. Because the accuracy of the testing was determined by UF to be ± 1.5%, the measured change, less than ½ %, was not considered significant.

In their conclusions they write, "*Based on the performance data obtained during this investigation, it can be concluded that the addition of FRIGAID to the refrigerant has no effect on the performance of an air conditioning system. Therefore an electrical utility will not derive any benefit in kW reduction and the customer will not derive any benefit in kWh reduction. The cost of charging FRIGAID is expected to be between $95 and $125. However, payback period is not applicable since there are no savings."*[7]

Marketing Objections

The sales personnel in marketing this COA product, have responded to the results of the Oak Ridge and U of F tests by claiming that their product was tested improperly. They indicated that thier COA product provides its benefit during start-up of the equipment, when it is under high friction load, and the improved heat exchange allows for faster evaporator cool down, and condenser heat up. That is why the steady state tests did not show any improvement - but equipment cycling on a thermostat to meet a load as it is in the field, would demonstrate their stated energy savings. As a result, the FSEC test was set up to test energy performance while the test systems cycled on a thermostat as they would in any field installation.

FSEC TESTING METHODOLOGY

Two air-conditioning units were instrumented and monitored for energy use in the FSEC Appliance Laboratory (ALT) environmental control chambers. The two units selected for the test were: 1) a new 2.5 ton split system air conditioner unit by Carrier (this unit had only been in operation for two months), and 2) a 10.5 year old, 5 ton split system air conditioner by Bryant. The tests on these units will be hereafter referred to as the "Unit #1 test" and the "Unit #2 test" respectively. A "before" and "after" test was conducted. The "before" test ran the equipment without the COA, the "after" test ran the equipment with the COA added to the crankcase. These systems were tested using the ASHRAE/ARI tolerances on the test equipment [9]. Prior to testing, the details of the tests were sent to the COA manufacturer who responded with comments and requests for additional measurements. All recommendations as to how the test should be conducted by the COA manufacturer were incorporated into the test procedure. The test was conducted with a 95 degree F outdoor condition on the condenser unit. However, unlike the ARI test specification which calls for a stabilized 80 degree indoor condition, it was requested that the test be operated with a 75-76 degree F indoor set-point condition. Thus, all tests were run with this heavier load on the air conditioning equipment, that is, the indoor thermostat setpoint was placed at 75 deg. F.

Where the ARI test calls for a stabilized indoor condition, the manufacturer of the COA product requested that a thermostat be used to cycle the air conditioning equipment against a load, as it would operate in the field. Therefore, the two test units were set up to each operate on a thermostat that was installed next to the return air intake for each unit. The stable indoor load was established by the addition of a constant amount of heat to the space by electric space heaters. Heaters were added to the space such that the "before" test units operated at an 80 to 90% duty cycle. That is, when they normally cycled on and off from their thermostat, they would, over a period of cycles, be running between 80 to 90% of the time. This fixed amount of heat load for the ac units was thus maintained the same for the before and after tests. Moisture was also added to the space by a computer controlled humidifier. The amount of moisture (ml/min) that was needed under baseline operation was determined such that the indoor space RH stayed about 50% RH. Like the heat load, this moisture load was maintained constant for the before-after tests. The thermostat (the same thermostat was used for both machines) was quite typical as it demonstrated about two degrees swing. The thermostat was set on 75 degrees. When the space was cooled to 75 degrees, or slightly below, the air conditioner would shut off. It would slowly warm up for 8 to 15 minutes until the temperature reached approximately 76.5 degrees or slightly warmer. The thermostat would then cycle the air conditioner on for 75 to 90 minutes until the space was cooled to about 75 degrees again.

For the baseline test, Unit #1 was operated on its thermostat for twelve days while data was taken on the unit. Then ½ oz per ton (1.25 oz) of the COA was added to the machine and it was operated in the same way for an additional 12 days as the "after" test period. Unit #2 was also set up to cycle on the signals of the thermostat and also to run at an 80 to 90% duty cycle prior to starting the testing.

Unit #2 was run for 12 days to establish its baseline performance. Then 2 ½ oz. of the COA was added (1/2 oz/ton). The unit was then run and monitored for 12 additional days. Finally, at the suggestion of the manufacturer, an additional 2 ½ oz of COA was added for a total of 1 oz./ton, and the unit monitored for an additional 12 days.

The following parameters were monitored: Run time, ambient temperature, output temperature, indoor humidity, kWh, kW, KVA, amp draw, noise level, and humidity removal (condensate). In addition, indoor temperature, and output humidity were measured to determine the operational capacity and efficiency of the equipment. To determine if the addition of the COA produced a reduction in noise level of the compressor, the noise level was measured during the first day of each test sequence and on the last day of each test sequence. Octave band sound pressure data was retrieved from 31 to 8000 Hz and loudness was calculated by averaging the measured loudness in decibels at nine audible frequencies. All sound measurements were made from the same location, outside each condenser coil, 18 inches from the compressors' side.

Testing Method Background

The experimental system included the instrumentation required to monitor the cooling provided by the evaporator within the air handler, and the energy use of each system when installed. A Campbell Scientific, CR-7 data acquisition system was employed to acquire and store the test data for each test sequence. The data tables were downloaded daily from the CR-7 to a laboratory computer for archival to hard drive and floppy disk for later analysis.

The ARI Standard 210/240 [8] references the ASHRAE Method of Testing for Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps, ANSI/ASHRAE 116-1983 [9]. For measurement of cooling contribution provided by the evaporator in the before and after alternatives, the test set-up was configured to use the Tunnel Air Enthalpy Test Method Arrangement of Standard 116 (section 6.1.1). Further control was provided by a pair of air handlers fed by a 5 h.p. R-502 condensing unit, which conditioned the outdoor chamber to the 95 degrees F. and maintained it to the tolerances prescribed in Standard 116 to conduct the tests.

Temperature measurements used for the air enthalpy measurement were provided by two type T thermocouple grids located before and after the indoor coil. These grids were constructed in accordance with specifications described in section 7.4.3.1 of Standard 116. Furthermore, in order to comply with this section, turbulating vanes were fabricated as per ASHRAE Standard 41.1-74 and placed after the indoor coil. These vanes provided the proper mixing of air flow so as to ensure a uniform temperature distribution across the output thermocouple grid. After initial calibration, these temperature probe arrays were calibrated against each other to reduce any variation as a result of line lengths. Wet bulb temperature measurements were obtained from a pair of Omega HX93C Hygrometers placed before and after the evaporator coils of each unit. These transducers were initially calibrated using the NIST salt solution method and then calibrated against each other as pairs. These values were recorded as relative humidities.

The air flow measurements for the Air Enthalpy Method were provided by means of a cup air flow anemometer Model 3101, R. M. Young Company. Calibrations were accomplished after installation using a Solomat MPM 500 hot wire anemometer.

The use of electrical energy by the system was measured by a General Electric KWH meter, Type 1-70-S, Model 720X0G1 (240 V, 30 Amp). The meter was calibrated by a Magtrol Model 4612 Power Analyzer. The recorded electrical energy included the outdoor unit with fan and compressor and the indoor unit with its fan, summed to a single Kwh value. The Magtrol Power Analyzer was also used to provide a measure of the peak KW demand pulled by the air conditioner units as they cycled on and off.

Thermocouples were calibrated according to procedures established by the National Institute for Science and Technology (NIST) [10]. Calibration on the thermocouples and RH sensors were accomplished after line lengths had been measured and cut, but prior to installation so all transducers could be brought to a single location for calibration. The condensate was collected in a covered plastic container located below the suction trap of the air handler condensate line. It was measured after each daily run with a 1000 ml graduated beaker.

TEST RESULTS

Test of Unit #1 (New)

No problems were encountered during the calibration and set up process of this 2.5 ton AC unit. Prior to any testing, the unit was evacuated and recharged with R-22 such that the evaporator was operating fully flooded at an 80 degree indoor temperature, (approximately 10 degrees subcooling). The unit was carefully checked for leaks and none found. The computer control programs were written and implemented into the Appliance Laboratory control chamber computers to control the temperature of the outdoor chamber at 95 degrees F and to monitor the equipment. Data was scanned every second and averaged and archived every minute. This minute by minute data was then averaged for each day's run to provide "daily" averages for the before and after data.

Twelve days of baseline operation were obtained and twelve days of operation of the AC equipment after the COA was added were obtained. It was found that a continuous heat load of approximately 8 kW to the indoor chamber was required to cause Unit #1 to cycle on its thermostat at an 80 - 90% duty cycle (four 2000 watt heaters). All tests, both before and after, were run with a constant heat and humidity load. Humidity was provided by two high pressure humidifier misting nozzles which were computer cycled to maintain 51% RH within the indoor chamber during the shakedown tests. This load was maintained throughout all tests. These nozzles replaced the moisture into the air which was removed by the air conditioner equipment. In addition, the environmental control chambers of the Appliance Laboratory utilize chambers that are constructed within a large controlled chamber. With the surrounding chamber set to maintain 78 degrees, 50% RH, with the heat and moisture load to the indoor space held constant, and with the outdoor space held at a constant 95 degrees by the laboratory control system, very stable test conditions were maintained and the same heat and moisture load on the systems for the before and after tests were obtained.

Test conditions - Unit #1 test.

A plot of the daily Unit #1 test conditions are provided below in Figure 1. It can be seen that the before and after space condition maintained by test Unit #1 stayed the same for the before - after time periods as it cycled on its thermostat. Also, the outdoor condition was maintained at 95 ± 0.6 degrees F for the before and after test days.

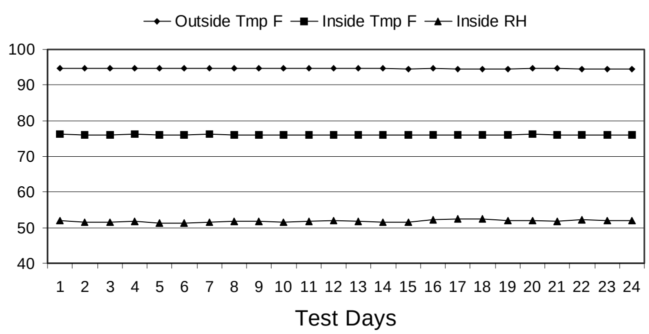

Figure 1. Unit #1 Test - Daily Conditions (The COA Was Added Between Day 12 and 13).

Run times on Unit #1.

The plot in Figure 2 shows the run time data on the Unit #1 for the before and after test days. The COA was installed between day 12 and day 13. The total run minutes for each day are plotted as divided by 24, that is, the plotted number represents the average run minutes for each hour. The variability in run times is due to the fact that the cycling of the unit was controlled entirely by its thermostat. If the recording day happened to end just at the end of a run cycle, then a higher duty cycle was recorded for that day, such as on day 15 and 17. Day 16 recorded a lower duty cycle, because it had more "off " periods. There appears to be a trend of higher duty cycle and run times after the COA was added between day 12 and 13, however, this trend is not considered to be significant due to the variability in the data.

Unit #1 test performance data.

The temperature of the air off of the coil, the measured cooling delivered by the unit in Btu/hour, and the measured EER of the unit (the average Btu/hour divided by the Kwh used in each hour) are provided in Figure 3 for the before and after test days of the Unit #1 test. No significant change is noted between the before data and the after data.

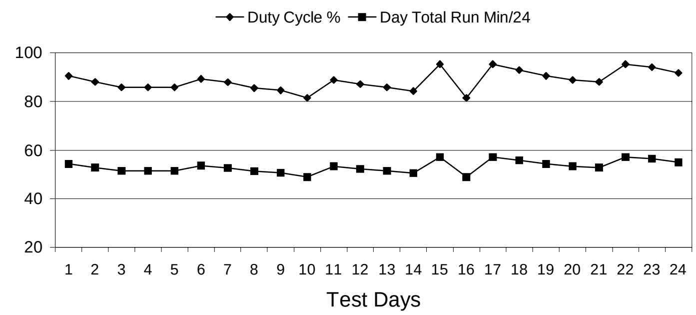

Figure 2. Plot of Daily Equipment Run Times for the Unit #1 Test.

The fact that there was no change (reduction) in air temperature measured after the coil in the after data, indicates that the COA did not enhance heat exchange as claimed.

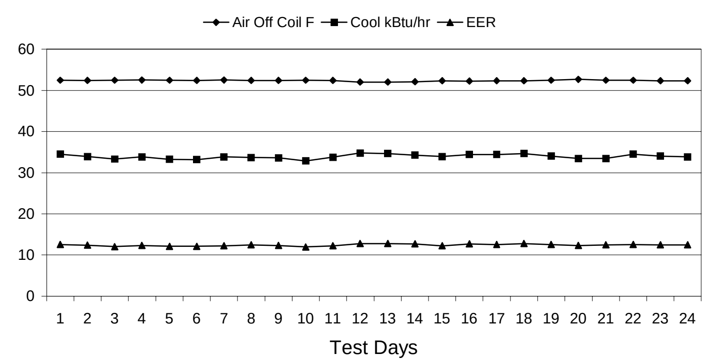

Figure 3. Equipment Performance for Unit #1 Test

Unit #1 Test Energy Use

Figure 4 provides the measured energy use for the Unit #1 test of the COA.

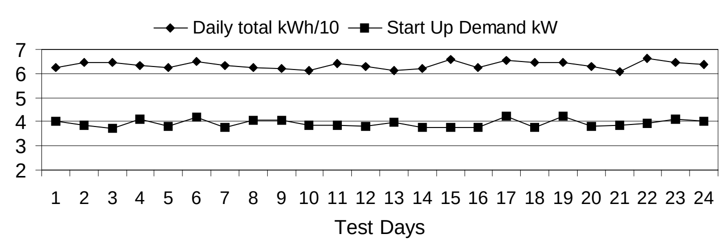

Figure 4. Before – After Unit #1 KWh Daily Use and Start Up Demand.

The total kWh daily energy use as recorded by the kWh meter is shown divided by 10 on the graphic presentation. For example, on day one, the unit used a total of 62.25 kWh as recorded by the standard rotating kWh meter over the 24 hour period. This is plotted as 6.225. No significant change in energy use was recorded for the equipment after the COA was added.

The maximum demand as recorded by the Magtrol digital power analyzer during the day is shown as Start Up Demand in kW because such demand always occurred during the amperage surge associated with a start up of the air conditioner unit. It should be noted that this demand is NOT EQUIVALENT to the demand that would be recorded on a commercial demand meter. The peak demand recorded by the digital meter can be picked up with a duration of less than half a second. When a commercial demand meter sees a step change in demand, it takes from 10 to 15 minutes for the meter to record this full change in demand. Short duration demands such as the transient start up demand recorded here are not picked up by typical commercial demand meters. The fact that there was no change in this start up demand measurement after the COA was added indicates that there is no measurable reduction in start up energy on this unit due to the addition of the COA.

Unit #1 sound level tests

An octave band analyzer by Quest Technologies, Inc, Model 08-50 was used to measure the sound level of the compressor unit. Measurements were made at 31, 63, 125, 250, 500, 1K, 2K, 4K, and 8K Hz. Each test included three sets of measurements which were averaged for each frequency and then all frequencies were averaged to provide the average distributed sound level across audible frequencies for each test. The results of the test readings are plotted as Figure 5.

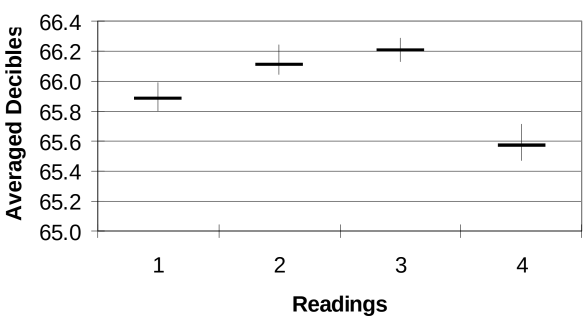

Figure 5. Measured Averaged Sound Levels Across 31 to 8K Hz on Unit #1.

All measurements were made from the same location, 18 inches from the side of the compressor and outside the condenser coils. The unit was tested for reading 1 at the start of the baseline test (test day 1) and for reading 2 at the end of the baseline test (day 12). Reading 3 was obtained about 4 hours into the first test day after the COA was added (test day 13), and reading 4 was obtained near the end of the last day of the "after" test sequence (test day 24). The change recorded, less than 1 db, is not perceptible to the human ear at this sound level, and is well within the ± 3 db accuracy of the test. Thus, no significant change in sound level was found to occur in the Unit #1 equipment in the before operation and after the COA was added.

Test of Old Unit #2

No problems were encountered during the calibration and set up process of the older five ton AC unit. The system was evacuated and recharged with R-22 such that the evaporator operated fully flooded at an 80 degree indoor temperature. The system was checked carefully for leaks. A new duct sensing thermocouple array was constructed, calibrated and installed in the Unit #2 ducting, rather than removing the array installed in Unit #1. The same computer control programs were used that were written and implemented into the Appliance Laboratory control chamber computers to control the temperature in the outdoor chamber at 95 degrees F. It was found that approximately 18 kW of heat from electric heaters was needed to cause Unit #2 to cycle on its thermostat at an 80-90% duty cycle.

There were no failures, problems or faults during the operation of the testing. Highly stable and consistent operational data was obtained on the AC system as it cycled on its thermostat to meet the stable load. The testing of Unit #2 with one half ounce per ton (2.5 oz.) added to the system was completed, then an additional 2.5 oz. was added to the system. The Unit #2 test has twelve days of the "before" baseline test, then twelve days of "after" test with 2.5 oz. added, and then an additional twelve days operation with 5.0 oz. COA added to the unit. The total test sequence for the older Unit #2 thus contains 36 test days, 2.5 oz. added between test days 12 and 13, and an additional 2.5 oz. added between test days 24 and 25.

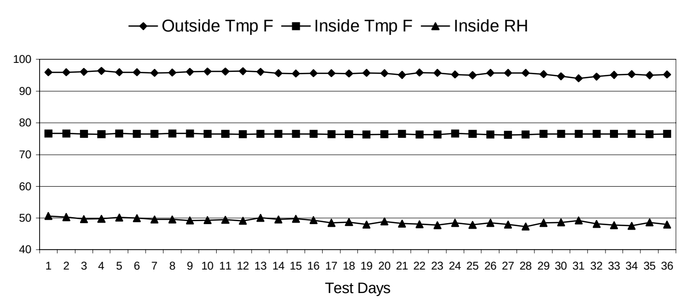

Figure 6. Unit #2 Test Daily Conditions.

Note that during the second half of the test days, there is a slight decrease in the temperature of the outdoor unit, and a slight decrease in the humidity of the indoor space. This occurred because these later testing days were in the heart of the winter, and temperatures outside dropped below freezing with RH below 30%. These conditions are extreme for the Central Florida location of the Appliance Laboratory, and put a strain on the containment chamber control system. Never the less, the temperatures of the outdoor chamber did not drop more than 1 degree F, well within the ± 1.8 degree F range of the ARI test standards, and the RH of the containment chamber did not drop more than 1.4 % RH, well within the ± 2.5 % RH range of the ARI standard. These trends are in the direction of REDUCING the load on the AC system for the periods "after" the COA was added, and thus there would be the expectation of a slight improvement in the operational energy use of the equipment in this "after" period because the induced load on the equipment was slightly lower and the equipment cycled to maintain the same before and after thermostat set point.

Run times on Unit #2.

The plot in Figure 7 below shows the run time data on Unit #2 for the before and after test days. Two and one half ounces of the COA were installed between day 12 and day 13, and an additional two and one half ounces installed between day 24 and day 25. The total run minutes for each day are plotted as divided by 24, giving the average run minutes per hour.

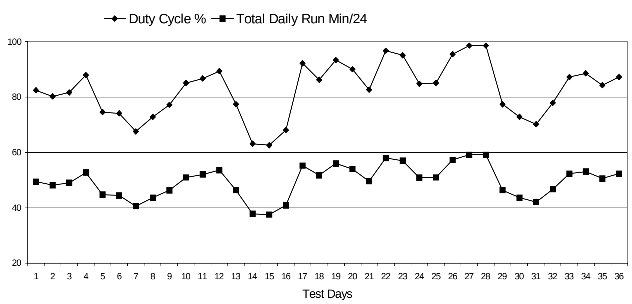

Figure 7. Plot of Daily Equipment Run Times for the Unit #2 Test.

As with the Unit #1 test, the variability in the run times from day to day is associated with the fact that the cycling of the unit was controlled entirely by its thermostat and any one day may pick up more "off" periods or more "on" time, depending where the unit was in its cycle at the start and end of the day. It is the mean run times over the test period of days that are of interest in evaluation of the potential improvement of the COA.

There appears to be an improvement in run time initially after the COA was added, but the longer trend looks worse after the COA was added. During the first 12 days of test without the COA (a total of 17280 minutes), the unit ran 13,814 minutes to maintain the indoor thermostat condition. The next 12 days with 2.5 oz additive, the unit ran 14,284 minutes, and the last 12 days with 5.0 oz. of COA, the unit ran 14,904 minutes. Though these increases in run time with the COA added might seem significant, they are not significant due to the variability in the day to day run time data. However, the trend toward slightly poorer performance of Unit #2 after the COA was added shows up as statistically significant in some of the other monitored data.

Unit #2 test performance data.

The temperature of the air off of the coil, the measured cooling delivered by the unit in Btu/hour, and the measured EER of the unit (the average Btu/hour divided by the kWh used in each hour) are provided below in Figure 8 for the before and after test days of the Unit #2 test. The previous graph indicated a trend of increased run times with the COA, consistent with that monitored data, there is apparently a trend toward a reduction in cooling provided by the equipment after the COA was added, a very slight increase in temperature after the coil, and a slight reduction in the EER.

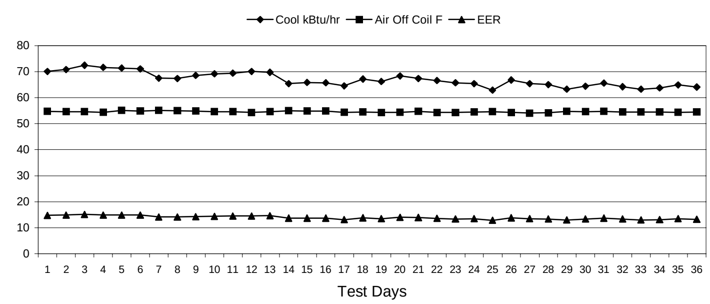

Figure 8. Equipment Performance for the Unit #2 Test.

Figure 9 provides the measured energy use for the Unit #2 test of the COA additive. The total kWh daily energy use as recorded by the kWh meter is shown divided by 10 on the graphic presentation. As with the Unit #1 test, the start up demand is the maximum measured by the Magtrol Power Analyzer, and is not indicative of the reading that would be provided by a kilowatt-hour/demand meter. As with some of the previous data, there is a slight trend of higher power use after the COA was added. The start

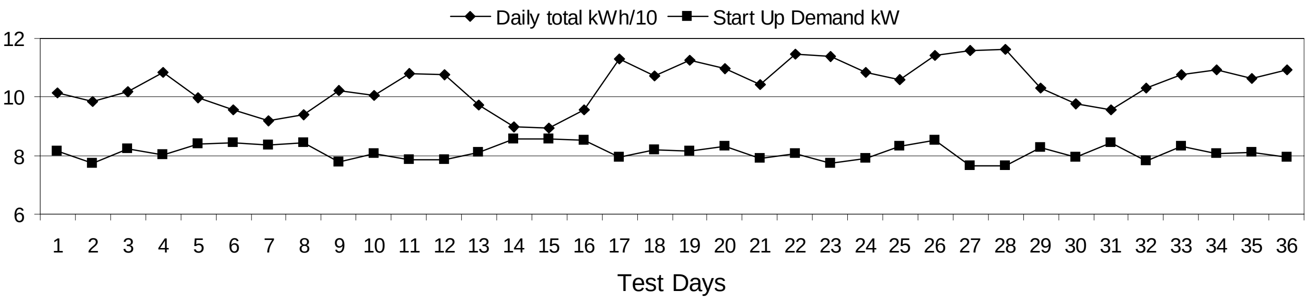

Figure 9. Before - After Unit #2 KWh Daily Use and Start Up Demand.

up demand data shows no apparent before - after change.

Unit #2 sound level tests.

An octave band analyzer by Quest Technologies, Inc, Model 08-50 was used to measure the sound level of the Unit #2 compressor. The measurement technique was the same as that used for the Unit #1 compressor and is detailed above. The results of the tests are plotted as Figure 10.

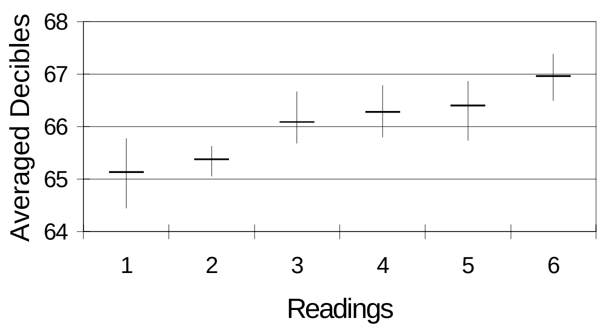

Figure 10. Measured Average Sound Levels Across 31 to 8K Hz on Unit #2 Test.

Reading 1 was made at the start of the Baseline test, reading 2 at the end of the 12 day baseline test, reading 3 at the start of 2.5 oz of COA added, reading 4 at the end of 12 days testing at 2.5 oz, reading 5 was at the start of the 5.0 oz COA added and reading 6 was at the end of the 12 days testing with 5.0 oz COA added. The readings show a trend of the compressor being slightly louder with the COA added, but the 2 Db increase in sound is not perceptible to the ear at this sound level.

ANALYSIS OF TEST RESULTS

The objective of this study was the evaluation of the potential kWh savings that would result from the addition of an α-olefin molecule compressor oil additive (COA) such as PolarShield to a typical air conditioner. In addition, a number of other parameters were measured before and after to get a measure of the claims of the sales literature, that is, that this COA increased lubricity and enhanced heat exchange. Because the heat and moisture load on the equipment was carefully held constant for the before and after time periods, for such a test, any change due to the treatment would be the difference in the before and after average test parameters. A paired t-test of means was conducted on each before-after test parameter to determine if any statistical difference was present. All tests were conducted at the 95% confidence level. The test hypothesis for each parameter was defined as:

H_o = There is no difference in the "before" average parameter and the "after" average parameter when the COA was added.

H_a = The average "before" parameter is greater (or less) than the average "after" parameter when the COA was added and thus the COA provided a benefit.

Tables of Test Results

In addition to the monitoring of the load conditions to assure that the loads on the equipment were the same for the before and after time periods, a total of eight test parameters were monitored to determine if there was an improvement to the air conditioner equipment caused by the addition of the COA. These parameters and the implication of the measured parameters are listed below.

1) Daily energy use (kWh). This is most direct measure of energy savings, these data represent the savings that would be seen by the utility read meter. A lower "after" reading would indicate an energy savings by a customer with the addition of the COA.
2) Daily run time. A surrogate measure for kWh use, this is the number of minutes that a system was running during each day as it cycled on its thermostat. The implication is that given a constant load, if the daily run time is less in the after period, then the daily energy use would also be less, and thus an energy savings to the customer.
3) Duty cycle. Also a surrogate measure for kWh use, this is the ratio of the daily run time divided by the total minutes in a day (1440), expressed as a percentage. The implication is that if the duty cycle were lower after addition of the COA, then the run time for each day would be lower and thus an energy savings for the customer.
4) Cooling delivered. This is a direct measure of the cooling delivered by the unit in each hour and measured by kBtu/hr. It is also a direct measure of the combined effects of the claims of the PolarShield literature - that the addition of this COA increases heat exchange in the evaporator and condenser, and improves lubrication in the compressor. If these claims were correct, then with a constant heat load, the cooling from the unit should increase with the addition of the COA. The implication is that if the COA caused increased cooling, this increased cooling leads to shorter run times which leads to energy savings for the customer.
5) EER. This is the Energy Efficiency Ratio and represents a measure of the efficiency of the equipment. It is used by manufacturers and others as a surrogate measure for potential energy use of their equipment and is calculated by dividing the cooling delivered in kBtu/hr by the energy use, kWh used to produce that cooling. The implication is that if the equipment shows an improvement in efficiency due to the addition of the COA, then it will use less energy and provide an energy savings for the customer.
6) Coil Air Temperature. This is the temperature of the air as it exits the cold coil of the air conditioner. The implication is that if the COA improved heat exchange on the refrigerant side as claimed, then the better heat exchange from the cold refrigerant would cause the air to be colder coming off of the coil. Other changes that are known to improve heat exchange in the coil, such as cleaning the coil fins, or adding rows to the coil, have the effect of lowering the air temperature as it exits the coil after the change is made. This is a direct measure of the COA manufacturer's claim that it improves heat exchange in the coils.
7) Start Up Demand. Measured in kW, this is the maximum demand recorded on any start up during each test day as recorded by the digital power analyzer. This is a surrogate measure of the lubrication of the compressor. The implication is that if the COA makes a significant improvement to lubrication as claimed, then the reduction in friction in the compressor would cause a reduced power surge when the compressor turned on.
8) Sound Level. This is the average sound level in decibels of the stable running compressor across the sound frequencies of 31 Hz to 8K Hz. This is also a surrogate measure of the lubrication of the compressor. Poorly lubricated compressors are noisy. The implication is that if the COA additive improves lubrication as claimed, then the reduction in friction in the compressor would cause a reduced sound level when the compressor was running.

Tables 1, 2 and 3 below provide the average values for each parameter tested in the before-after test periods. Also provided is the result of the t-test conducted on the data - whether the change is "no difference" caused by the COA treatment or "yes there was a difference" with the tests conducted at the 95% confidence level.

Table 1. Results of COA Treatment Before - After Tests on Unit #1 (12 Day Averages)

Parameter	Before	After	% Change	T-Test	Result
Daily Energy (kWh)	63.17	63.70	+ 0.85 %	No difference	No savings
Daily Run Time (Min)	1248	1300	+ 4.12 %	No difference	No savings
Duty Cycle (%)	86.69	90.26	+ 4.12 %	No difference	No savings
Cooling (Btu/hr)	33.70	34.14	+ 1.31 %	No difference	No savings
EER	12.28	12.54	+ 2.12 %	No difference	No savings
Air Off Coil (F)	52.43	52.35	- 0.15 %	No difference	Not colder
Start Up Demand (kW)	3.92	3.93	+ 0.26 %	No difference	No reduction
Sound (Db)	66.00	65.89	- 0.17 %	No difference	Not quieter

Table 2. Results of COA Treatment Before - After Tests on Unit #2 (12 Day Averages - With 2.5 oz)

Parameter	Before	After	% Change	T-Test	Result
Daily Energy (kWh)	99.53	104.56	+ 5.05 %	No difference	No savings
Daily Run Time (Min)	1151	1190	+ 3.37 %	No difference	No savings
Duty Cycle (%)	79.9	82.7	+ 3.50 %	No difference	No savings
Cooling (Btu/hr)	69.93	66.48	- 4.93 %	Yes difference	Less cooling
EER	14.61	13.71	- 6.16 %	Yes difference	Lower EER
Air Off Coil (F)	54.74	54.56	- 0.33 %	No difference	Not colder
Start Up Demand (kW)	8.12	8.16	+ 0.49 %	No difference	No reduction
Sound (Db)	65.26	66.19	+ 1.43 %	No difference	Not quieter

Table 3. Results of COA Treatment Before - After Tests on Unit #2 (12 Day Averages - With 5.0 oz)

Parameter	Before	After	% Change	T-Test	Result
Daily Energy (kWh)	99.53	107.01	+ 7.25 %	Yes difference	Uses more
Daily Run Time (Min)	1151	1228	+ 6.69 %	No difference	No savings
Duty Cycle (%)	79.9	85.3	+ 6.76 %	No difference	No savings
Cooling (Btu/hr)	69.93	64.44	- 7.85 %	Yes difference	Less cooling
EER	14.16	13.29	- 9.03 %	Yes difference	Lower EER
Air Off Coil (F)	54.74	54.49	- 0.46 %	No difference	Not colder
Start Up Demand (kW)	8.12	8.08	- 0.49 %	No difference	No reduction
Sound (Db)	65.26	66.69	+ 2.19 %	No difference	Not quieter

DISCUSSION OF RESULTS

The Unit #1 test results show no improvement in the operation of the equipment and no energy savings that can be attributed to the addition of the COA. But, Unit #1 did not operate worse after the COA addition. This system just operated the same within the accuracy of the test. However, there is a trend for Unit #2 that shows the unit operated a little worse, less cooling and lower EER when the first addition, 2.5 oz of COA was added. This trend is proved at 95% confidence with the second 2.5 oz addition, totaling 5.0 oz of COA added. The system used more energy, provided less cooling, and showed a lower EER. This is consistent with work from ASHRAE that indicates that too much oil in a system can degrade its performance [11].

In addition to the parameters specified and investigated above, the condensate was collected at the end of each day Because a fixed moisture load was added to the space by a computer controlled humidifier cycling on a pre-established pattern, and the space humidity drifted down slightly during the Unit #2 test due to the extremely low outdoor humidities, it was of interest to determine if this had an effect on the moisture removal of the equipment. Thus, the Latent Ratio of the equipment was also investigated. Latent Ratio is the percentage portion of the cooling that is due to moisture removal from the space. As would be expected, the liters of condensate collected each day varied in direct proportion to the run time of the equipment and similar to the run time, showed a high variability.

There appeared to be less condensate per hour of run time in the later runs of Unit #2 with the COA added, but neither total condensate per day, or condensate per hour showed any significant change at the 95% confidence level. Also, the Latent Ratio showed a trend toward a slight reduction in Unit #2 with 5.0 oz of COA added, but none of the Latent Ratio differences were significant for the before-after tests. These results are consistent, because with unchanged air flow, latent ratio and condensate per hour are functions of the evaporator coil temperature which did not show a change between the before - after test periods.

CONCLUSIONS

The compressor oil additive (COA) PolarShield sales literature provided to FSEC indicates that this is a “proven” technology from field testing claiming energy savings from 12 to 36%. However, because before - after loads on the equipment were not sufficiently monitored for these tests, nor was statistical hypothesis testing conducted to evaluate “apples to apples” for the before - after segments of these tests, they must be considered inconclusive. The results of the three tests of this paper, where before - after load conditions on the equipment were carefully monitored and held the same, and the equipment was allowed to cycle on its thermostat, showed absolutely no energy savings associated with the addition of the COA. A slight reduction in performance and a higher energy use was demonstrated on Unit #2 at the 95% confidence level when an additional 2.5 oz of this COA was added to total 5.0 oz added.

Seven tests on actual equipment have now been completed on the α-olefin molecule compressor oil additive and conducted by nationally recognized laboratories - three tests at FSEC's Appliance Laboratory, one at the University of Florida, and three at Oak Ridge. All seven tests found the same result - no energy savings caused by the addition of this compressor oil additive.

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