Measured Energy and Demand Impacts of Efficiency Tune-ups for Small Commercial Cooling Systems

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This project sought to quantify the energy and demand savings that could be achieved through efficiency tune-ups of commercial unitary cooling equipment in the service territory of a major New England utility. Cooling accounts for about 22% of electricity use and 63% of peak demand in the commercial sector in New England. Approximately half the tonnage of commercial cooling equipment consists of single package and split system unitary equipment.

Eighteen typical systems, ranging in size from 4 to 15 refrigeration tons (RT) (11 to 70 kW output), were tuned. The protocol used was derived from a residential protocol developed in North Carolina and field tested in two pilot projects in California. It emphasized correcting refrigerant charge, cleaning and adjusting the system to correct air flow and improve heat transfer, and repairing major duct leaks. Air flow was improved significantly in only three systems. Of 25 compressor circuits, 10 were overcharged as found, 8 were undercharged, and 7 had approximately the correct charge as found. Final charge was optimized in nearly all cases. None of the 13 thermostatic expansion valves (TXVs) were providing proper superheat as found. Three TXVs were successfully adjusted to correct this problem; nine of the remaining ten were not adjustable. Significant duct problems were found and repaired in seven cases.

The average absolute energy savings for the sample was 4191 kWh/y per site, and the average absolute reduction in diversified demand at 95°F was 1.89 kW. However, these absolute averages are heavily influenced by high savings at the largest site and are probably optimistic for the population as a whole. Applying the average percentage savings to the average pre-tune-up energy use and demand produces estimated average savings of 1936 kWh/y per site and 0.43 kW; however, these estimates are conservative, as there is a statistically significant relationship between percentage savings and pre-tune-up use (and demand), causing the product of their means to be a biased low estimator of mean savings. The average cost of the work was $1158 per site or $574 per system. The median payback to the customer is over six years. Adding 20% for program administration, the conservative savings estimates yield benefit/cost ratios to society which are less than one, while the optimistic savings estimates yield a ratio that just barely exceeds one if environmental externalities are included. Taking into account the increase in both savings and costs with system size, a tune-up could be expected to be cost-effective with environmental externalities if the pre-tune-up cooling use exceeds about 9,750 kWh/y, and without externalities if the pre-tune-up cooling use exceeds 18,500 kWh/y.

Background and Goals

Space cooling accounts for about 22% of commercial sector energy use and 63% of summer peak demand in New England (EPRI 1986), and therefore is a prime target for demand side management programs by capacity-constrained utilities. The utility company sponsoring the research currently provides financial incentives to customers for installing new high efficiency cooling equipment. The incentives do not adequately address the efficiency of existing equipment, however, which may be low due to poor quality maintenance or improper initial installation. In addition, for various reasons, the penetration of these measures in the small commercial market is low relative to that among large commercial customers. Recent studies show that only a minority of small businesses undertake regular maintenance of unitary cooling equipment (e.g., Hewett and Dunsworth 1989), and that even equipment on service contract can perform well below design efficiency. Price competition, inadequate technician training, lack of consumer awareness and a focus on comfort rather than efficient comfort limit the ability of service work to result in efficiency improvement (NCAEC 1988).

The goal of this project was to determine whether and how the utility should incorporate efficiency tune-ups for small C&I cooling systems into its demand-side management strategy, by assessing the potential for cost-effective reductions in energy consumption and peak demand through tune-ups. A number of recent studies have found that packaged air conditioners and heat pumps frequently operate below design efficiency, and that moderate to substantial energy and demand savings are possible through carefully designed tuning (Proctor 1991; Proctor et al. 1989; Smilie et al. 1983, 1984; Smilie 1985; Neal and Conlin 1988; NEES, 1988). A number of authors have also reported substantial energy penalties from duct leakage in residential buildings (e.g., Cummings et al. 1990; Modera 1989). Since previous tune-up research projects gave inconsistent estimates of energy and demand impacts, and since most had worked with residential systems, the utility needed to quantify the energy and demand savings and costs for tuning small commercial cooling systems, based on measured results of a carefully implemented pilot study.

Methodology

Sample Selection

The utility determined that the small commercial customers comprising the primary target group for a cooling tune-up program are those with average monthly peak demand less than 50 kW or annual consumption less than 150,000 kWh. Past participants in the utility's C&I lighting program served as the primary source of project participants. The project budget constrained the sample to a total of 18 cooling systems at nine sites. Due to the diversity of building and cooling system types, the small sample size, and self-selection bias, the sample could not be statistically representative of the customer base as a whole. The sample does, however, include the most common system types and features of the target population.

The screening criterion for size was 3 to 20 refrigeration tons (RT) cooling capacity (11 to 70 kW), which captures most of the small commercial market of interest, based on both industry data on shipments and stock and previous experience. Screening on summer cooling energy use as well was considered, since the utility does target high users in some of its current programs. Estimated cooling energy use (excess of summer over shoulder use) from billing data showed tremendous scatter as a function of floor area and cooling capacity, however, and was not thought to be sufficiently reliable for screening (though later analysis for the test sites showed a very strong relationship between cooling energy use estimated from bills and cooling energy use measured by the data acquisition systems, suggesting that such a screening criterion could indeed be used in a full scale program). Based on a rule of thumb of 400 to 500 ft^2 of cooled area per RT (10.6 to 13.2 m^2/kW), candidates with less than 1200 ft^2 (110 m^2) of floor area were eliminated.

A lower limit on age was set at four years, since for new units, if initially properly adjusted, a tune-up cannot be expected to produce savings. Though manufacturers estimate design life at about 13 years, the upper limit on age was set at a reported age of 20 years, since many of the potential candidates were tenants who had little real knowledge of equipment age.

Since the sample was too small to cover the spectrum of possible features, the strategy employed was to select equipment within each size category that was typical of the most common features for that size category, in terms of single package versus split system design, thermostatic expansion valve (TXV) versus capillary tube (cap tube) metering device, presence of multiple compressors or of economizers, single versus multiple zones, and constant versus variable air volume distribution systems. Interviews were conducted with distributors for the four largest U.S. manufacturers of unitary cooling equipment to determine the prevalence of these features in light commercial products.

Other criteria were established to allow for reliable test results and to protect the project from liability, including absence of supplemental cooling or rarely used cooling units, exclusion of facilities with high internal gains or substantial free cooling from refrigeration, and elimination of systems with frequent breakdown problems or incipient compressor electrical failure.

Contractor Selection and Training

Based on research projects elsewhere, highly skilled and trained technicians were believed to be needed to conduct effective efficiency tune-ups. A careful screening process was used to select firms and technicians with strong interest and skills and a quality orientation. The refrigeration technicians received 60 hours of training, half in the classroom and half in the field. The trainer is the head trainer for a major southern heat pump skills center that was established specifically to teach quality in service work (NCAEC 1988). Training covered the theory required to understand the tune-up procedures discussed below, diagnosis of system problems, use of diagnostic equipment, and data collection necessary for the research project. Seven systems (not in the research sample) were tuned during the training.

As discussed below, this project addressed only major duct leakage. The duct technician was chosen by a similar process and was trained at the first research site by an engineer from the research team who had been involved in previous research projects involving duct sealing.

Monitoring System

Each of the 18 systems was monitored continuously during the pre- and post-tune-up periods. A single computerized data logger at each site was used to record 30 minute averages of each sensor output. All sensors were polled and computations performed at five second intervals. Return air dry bulb temperature, air handler run-time, compressor run time (each compressor for dual systems), and true electrical power were measured. In addition to average system power during the entire period, the power was separately averaged during periods of steady state operation (typically starting three minutes after the compressor began running).

Weather stations were constructed at one site in each of the two areas included in the study, Worcester, Massachusetts and Providence, Rhode Island. The weather stations recorded outdoor air dry bulb temperature, relative humidity, and horizontal solar insolation.

Analysis of Seasonal Energy Use

Cooling season air conditioner energy use was modelled assuming a linear relationship between weekday average electrical use and outside dry bulb temperature, with zero electrical use below a reference temperature and linearly increasing use for outside temperatures above the reference temperature. Outside temperature data were taken from the weather monitoring site closest to each test building. Long term weather data from the NOAA stations at the Worcester and Providence airports were used to determine the cooling season length and average outside temperature corresponding to each cooling reference temperature; these values were applied to the regression models to estimate a normalized seasonal use. Because the businesses generally had reduced weekend operating hours, nominal seasonal use values were then adjusted for the average difference in use between weekdays and Saturdays or Sundays.

Because it was expected that outside temperature might not be an adequate predictor of air conditioning use by itself, the performance of inside/outside temperature difference models as well as models including humidity and solar terms in addition to temperature were examined. Since the different weather variables could be correlated with one another in ways that would produce unreliable or unrealistic results from conventional multiple regression, the use of regression on statistically independent factor scores derived from a prior principal components analysis of the raw weather variables was also evaluated.

Analysis of Demand Reduction

Analysis of demand reduction from the customers' perspective was based on statistically estimated changes in 15-minute demand at the site. Of greater concern to the utility is the diversified demand impact of thousands of air conditioners; when air conditioners are cycling, this impact is more accurately represented by some average demand (i.e., energy use divided by time). In this analysis, two-hour average demand was used to represent the utility demand impact at a given temperature.

Air conditioning demand reduction for each site was determined using a double change-point model that had been successfully used for residential air conditioning systems (Proctor 1991). In moderate weather, the air conditioners cycle and average demand is assumed to increase linearly with outside temperature along the cycling operation line (see Figure 1). As the outside temperature increases, the system eventually reaches a point, called the onset of continuous operation, where it runs continuously to keep up with the cooling load. Above this outside temperature, the air conditioner runs continuously and follows the steady-state operation line. At many of the sites, there were time periods when the air conditioner did not operate at all and the average demand was well below the model. The linear model for average demand was adjusted by multiplying the demand by the fraction of time the demand was at or near the modelled value, rather than being near zero. Since none of the sites with multiple air conditioning units had a distinct separation between the zones served by the different units, the demand from the units was added and modelled as if one air conditioner served the entire building.

The cycling demand line for each site and data period (pre- or post-tune-up regression) was determined by performing a least-squares linear regression of two-hour average data from between 10 AM and 6 PM on weekdays. In order to minimize the influence of inconsistent thermostat operation on the regression line, data points were excluded from the regression analysis according to the following criteria: (1) demand was less than 5 percent of the steady-state demand at 95°F; or (2) the point deviated from the regression line by more than 2.1 times

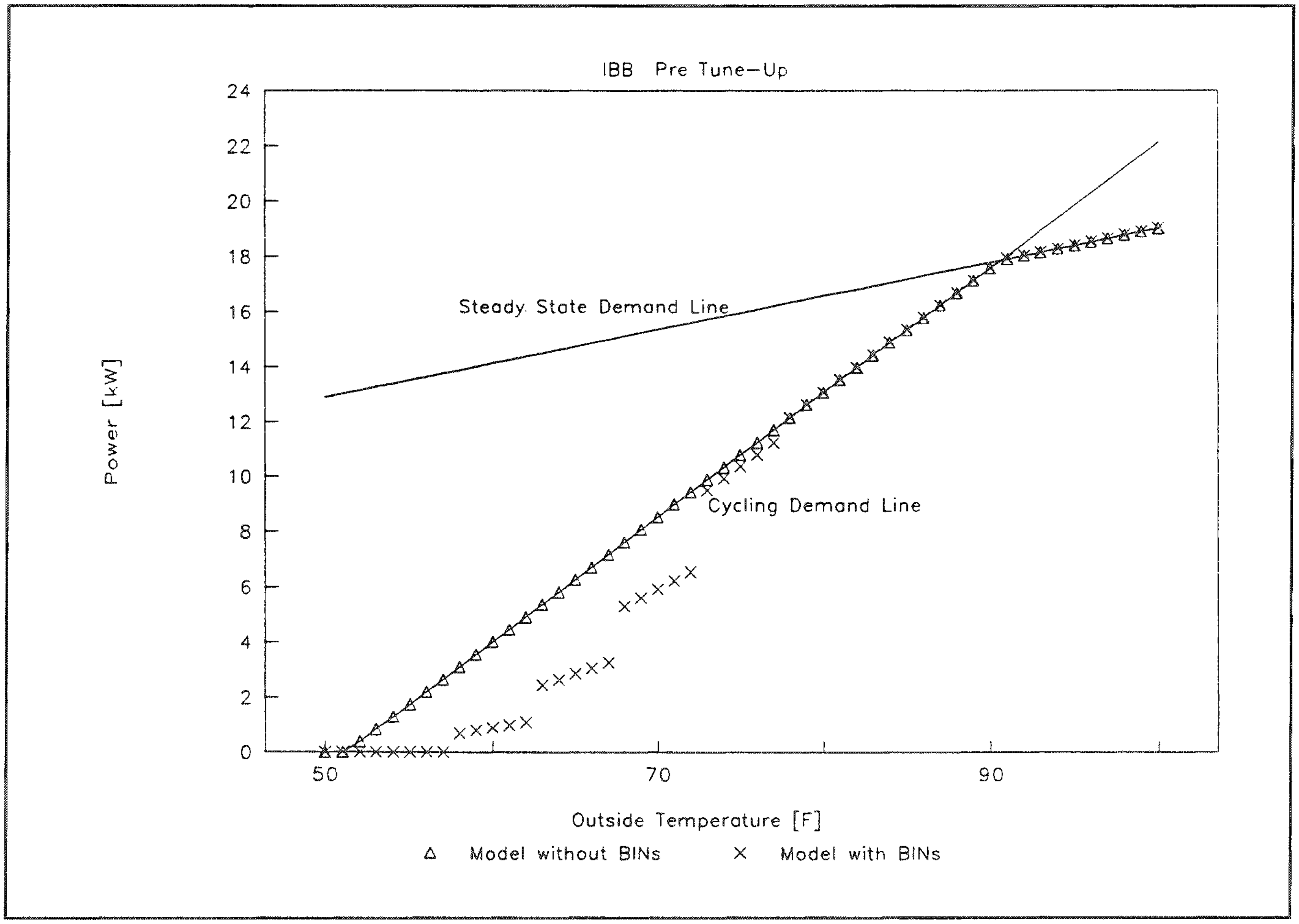

Figure 1. Two-Hour Average Demand Model

the residual standard error. The regression analysis and rejection of outlier points was performed iteratively until all points used in determining the cycling demand line fell within the specified bounds of the resulting regression line.

The steady-state regression line was found for each site and data period by performing a least squares regression of steady-state power using two hour "averages" of steady-state demand. The method of collecting steady-state power data produced data over a wide range of outside temperatures so that the regression lines are well defined. However, the effect of rising indoor temperature at outdoor temperatures above the onset of continuous operation is probably underestimated (higher indoor temperatures should lead to higher steady-state demand), because in most cases, few of the points used in determining the steady-state regression line fell beyond the onset of continuous operation, and therefore, most of the regression data had roughly the same indoor temperature.

Using 5°F intervals, a bin analysis of the fraction of demand values near the regression line for a given outdoor temperature was performed to accurately account for inconsistent thermostat operation. The pre- and post-tune-up data sets for each site were combined in the bin analysis because any changes in behavior were expected to be related more to seasonal or other factors than to the tune-ups and because there were not enough data to produce well-defined bin values for both the pre- and post-tune-up periods. The two-hour average demand for any outside temperature was then determined by multiplying the value of the appropriate regression line (either cycling operation or steady-state operation) by the fraction of time near the regression line for that outside temperature bin. The resulting model of average demand as a function of outside temperature was combined with ten years of data on the daily maximum temperatures at the time of the utility system peak for each month to determine normalized monthly utility demand impacts for use in the cost-effectiveness analysis.

Analysis of Diagnostic Data

The diagnostic data (i.e., instantaneous measurements made during the tune-up) were analyzed in two ways to estimate the impacts of the tune-up work. First, instantaneous manual pre- and post-tune-up input and output measurements were analyzed using a diagnostic spreadsheet program developed for this purpose by the City of Austin and revised by the research team to accommodate three phase commercial systems and impacts on cycling as well as steady state demand, and to automate some of the previously manually calculated inputs. The spreadsheet corrects the capacity, power and efficiency measurements and calculations at off-design conditions to Air Conditioning and Refrigeration Institute (ARI) design conditions using relationships taken from the DOE-2 simulation model (Los Alamos Scientific Laboratory 1980), allowing pre- and post-tune-up measurements taken under different conditions to be compared so that tune-up impacts can be estimated.

The second method of analyzing the diagnostic data evaluated the descriptions and data on the work done, and assigned approximate quantitative impacts based on previous laboratory or field results relating capacity, efficiency, and power to changes in air flow and charge. Rough quantitative estimates of duct sealing energy impacts were also included, based on the static pressure data, approximate leakage reduction estimates, and information on the location of the leaks relative to the conditioned space.

Sample Characteristics

Based on our interviews with major manufacturers, the characteristics of the final sample (Table 1) agree very well with those of the small commercial unitary equipment stock as a whole. On average, there were two systems per site. While this had been found typical in a previous study of small commercial buildings (Hewett and Dunsworth 1989), in this case it is an artifact of the research design for minimizing monitoring equipment needs, since the lighting program population as a whole had a preponderance of sites with single systems.

Tune-up Procedures

One point on which a number of the previous studies agree is that in order to be effective, tune-ups must be conducted according to a protocol carefully designed to optimize efficiency and with close quality control. Simply allowing contractors to tune systems according to their standard operating procedures will not optimize efficiency.

Air flow and charge are prime determinants of system efficiency, capacity and power requirements. The main objective of the tune-ups was to adjust these parameters as optimally as possible. In addition, the tune-up sought to improve heat transfer by cleaning coils, and to identify and correct miscellaneous problems as needed. The tune-up also checked for system problems such as phase imbalance and incipient compressor electrical or valve failure. Duct sealing was a secondary goal of the project, and sought only to eliminate major leaks. Finally, the protocol sought to record enough details of the work done to allow semi-quantitative estimates of impacts to be made, and to measure and document the system performance parameters necessary to calculate the change in performance using the diagnostic spreadsheet.

Air flow influences both system capacity and energy use. Design air flow (with a wet coil) is normally 400 cfm (ft^3/min) per RT (0.054 m^3/s per kW output). Low air flow is a more common problem than high air flow. Low air flow decreases capacity by decreasing heat transfer to the evaporator, and while it also decreases system power, the former effect is larger, so that there is a net loss in efficiency. In addition, low air flow tends to cause low suction pressures, which may cause the evaporator coil to freeze. A remedy for this problem commonly used by technicians is to overcharge the system, which increases energy use. High air flow unnecessarily increases power, and if air flow is excessively high, may decrease capacity by re-entraining condensed water from the coil or simply by raising the coil temperature to the point where latent capacity is lost.

Accurate measurement of air flow was considered to be one of the major challenges in the tune-up protocol by researchers with previous experience (John Proctor, Proctor Engineering Group, pers. comm. 1991; Leon Neal, North Carolina Alternative Energy Center, pers. comm. 1991). Two methods were ultimately used. Wet coil air flow measurements were taken with a back-pressure compensated flow hood designed to produce accurate results at low static pressures. This flow hood was checked against a known low static pressure air flow and found to be within 3%. At sites where the flow hood could not be used, dry coil air flow was calculated from the measured gas input, efficiency and temperature rise of the furnace. This method appeared to give spurious estimates of changes in air flow, which may have been due to differences in coil wetness between the pre- and post-tune-up tests, and resulting depression of the dry bulb temperature. Since the flow hood cases showed a good relationship (R^2 = 61%) between percent change in air flow by the flow hood method and change in air flow based on fan law computations (flow ratio = cube root of

Table 1. Sample Characteristics

System	Size RT+	Type, Stages+	Meter Device+	Estimated Age
ALL A	5	P,1	C	12-15
ALL B	5	P,1	C	12-15
ANK A	7.92	P,2	T/C	12
ANK B	5	P,1	C	20
BUS A	7.92	P,2	T/C	12
BUS B	7.92	P,2	T/C	12
BUS C	4	S,1	C	12
BUS D	4	S,1	C	12
DEL A	7.5	S,1	C	10
DEL B	7.5	S,1	C	10
GRO B*	15	P,2	T/T	10
GRO A*	15	P,2	T/T	10
GRO C*	10	P,1	T	10
IBB*,**	10	P,2	T/T	10
TAY	5	S,2	T	10
TOO	7.5	P,2	T/T	7-8
TRN B	4	S,1	C	4-5
TRN A	4	S,1	C	4-5

Note: All are single zone, constant volume systems.
+ 1 RT = 12,000 Btu/h (3.52 kW)
P = single package, S = split system
1, 2 = number of compressors
C = capillary tube,
T = thermostatic expansion valve
* Maintenance contract.
** Economizer.

power ratio), the measured changes in fan power were used to estimate changes in air flow for the temperature rise cases.

Air side and heat transfer work included cleaning the evaporator and condenser coils with coil cleaner and a pressure washer, cleaning or replacing filters where possible, straightening condenser fins, and adjusting belts and sheaves on belt driven blower motors.

Proper charge is a major determinant of system performance. Overcharged systems will have higher condenser pressures and temperatures, increasing system power consumption. Overcharged capillary tube systems will have improperly high evaporator pressures and temperatures and as a result will lose latent capacity. Undercharged systems use less power, but lose capacity more rapidly as more of the evaporator is given over to superheating rather than phase change. Grossly undercharged TXV systems lose their liquid seal at the TXV and suffer a gross decrease in capacity.

Proper charge of capillary tube systems was determined using a standard charging chart that relates proper system superheat (suction line temperature minus suction saturation temperature) to outdoor dry bulb temperature and return air wet bulb temperature. Charging of TXV systems was done using a target of 10°F (6°C) of condenser subcooling (condenser saturation temperature minus liquid line temperature) for conventional systems and 15°F (8°C) for high efficiency systems, but also considering the effect on system superheat, which while theoretically approximately constant, was found to be significantly affected by charge in some of these systems. TXVs were also adjusted where possible to provide a superheat of 10°F (6°C). The amount of charge added or removed was measured with a temperature-compensated charging cylinder. Since the entire charge was removed and measured in only a few cases, the approximate percent change in charge was estimated by assuming that the final charge was equal to the nameplate charge, or, for split systems without a nameplate charge, assuming a charge of 2 lbm/RT (0.26 kg/kW).

Duct sealing was a minor emphasis of the project. The primary objective of the duct sealing was to find and repair all major leaks, disconnected ducts, and blockages, and then to repair as much of the diffuse leakage as possible within a four hour visit. These gross problems were detected by visual inspection. The priority of duct sealing repairs was (1) disconnected ducts, (2) leaks to the outside, (3) cabinet leaks, (4) other large leaks, and (5) diffuse leaks, working from the plenum to the registers. Leaks were repaired using mastic and fiberglass mesh tape, butyl backed aluminum tape, neoprene gasketing, sheet metal and screws, and nylon cable ties, as appropriate.

Diagnostic data collected to facilitate interpretation of the duct sealing work included the location of leaks in the duct system and relative to the conditioned space, a description of the sealing work completed, very rough estimates of the volume of leakage reduction achieved, and pre- and post-tune-up supply and return static pressures. More intensive measurements, such as duct pressurization tests, were omitted both because of the expense and because past experience indicated that the intrusiveness is often unacceptable in an office or retail environment.

Based both on interviews with contractors, feedback from the technicians, and past experience of the trainer in teaching many experienced technicians, many of the procedures used during the tune-up differed substantially from methods commonly used by technicians in the field. Refrigeration technicians almost never measure air flow. Coils are cleaned relatively rarely. Although the charging procedures used are widely documented and recommended in manufacturers' training materials and trade journal articles, technicians very seldom charge by these procedures, typically charging to suction pressure instead, even though, especially for TXV systems, suction pressure is very insensitive to charge.

Results

Field Findings

Pre tune-up air flows (wet coil) ranged from 196 to 481 cfm/RT, with a mean of 334 cfm/RT. Only three systems were within ± 10% of 400 cfm/RT, while about half were within ± 20%. All coils were cleaned except where inaccessible. None of the condenser coils were found to be visibly dirty, while about half of the evaporator coils that could be cleaned were found to be dirty or very dirty. Filters could not be replaced routinely because the great variety in sizes made it impossible to carry the required inventory; however, these were generally found to be in better condition than anticipated. Loose belts and improperly adjusted sheaves were found and corrected in only four cases, and these were the only systems which showed substantial changes in air flow, ranging from 7 to 16%. Nevertheless, for most systems, the air flow appears to have been optimized as much as possible given the existing duct system. Of the systems which were left with air flow under 400 cfm/RT, only two (IBB and GRO A) could have had their air flows increased (by an additional 8% and 5%, respectively) while remaining within the rated amperage of their air handlers. One system (GRO C) whose air flow was (incorrectly) increased actually started with an air flow over 400 cfm/RT.

While substantial changes were made in the charge of many of the systems, it does not appear that there was a consistent pattern of over- or undercharging in the pre-tune-up condition. Charge was added to eight circuits and removed from 10, while 7 had approximately the correct charge as found. The percentage changes in charge ranged from -41% to infinite (for one circuit found empty), and -41% to 123% with the empty circuit excluded. The median change made was zero.

Most of the 13 systems with TXVs appear to have been charged optimally during the tune-up. Most of the cases with final subcooling higher than the target could not be reduced further because of dropping suction pressure (GRO B1, TOO 1 and 2), rising superheat (IBB 1), or wide TXV hunting (BUS B1). The two systems in which subcooling was increased substantially (and to more than 10°F) were charged by weighing in charge rather than by the subcooling target (BUS A1, GRO A1). While both may have ended up somewhat overcharged, both showed signs of some gas at the expansion valve prior to charging, so their performance is expected to have been improved, even if not maximized.

The refrigerant circuits metered by TXVs were uniformly found to have superheats well above the target of 10°F. Further, eight of these were of the non-adjustable type, and another was internally corroded and did not respond to adjustments. Three of the other four were adjusted, reducing the superheat somewhat, though probably not quite optimally. Of the non-adjustable TXVs, the charging itself improved the superheat in two cases, made it worse in two others, and affected it by less than 1°F on the other five.

Eight of the twelve cap tube systems also appear to have been brought as close to the target superheat as possible without causing other problems. Of the other four, one (ALL A) was not improved as much as it could have been, two were (incorrectly) not improved at all (BUS A and ANK B), and one appears to have had a restriction that should have been diagnosed and repaired (DEL A).

Three other significant changes were made. On one system (GRO C), a disconnected condenser fan (one of two) was reconnected, lowering the split (condenser temperature minus outdoor air temperature) by 33°F. On another (DEL A), air was purged from the system, but this effect was offset by a subsequent substantial overcharging (due to a restriction that made the superheat unresponsive to charge adjustments). On a third (ANK A) the second stage was found empty, and was charged after repairing a leak at the service valve.

Of the eighteen duct systems, duct leakage was estimated to be appreciably degrading performance on seven. Five of the others were located entirely within the fully conditioned space, and six systems were generally tight. The majority of the worst problems were able to be fixed in the time available. Several problems were not repaired. An apparent supply to return bypass (IBB) was inaccessible and could not be repaired. Significant evaporator coil bypasses at one site (DEL A and B) could not be repaired because the deteriorated cabinets did not afford a suitable mounting surface for any blocking material. One duct system (ANK B) was not fully repaired because of time limitations.

A number of other problems were found that could not be repaired during the course of the tune-up. The most serious and prevalent of these were low air flow and improper TXV superheat. It is not known whether the former could be improved by a relatively low cost means such as replacing high pressure drop diffusers, or whether the overall duct design was inadequate. Five of the non-adjustable TXVs were replaced in a follow-up project, but the results are not presented here. A number of other site-specific problems were also found but were outside the scope of the tune-up service.

In summary, most systems were tuned optimally, within the scope of tune-up work. The proportion that were not is probably no higher than might be expected in a full scale program.

Energy Impacts

Energy use was initially analyzed by several methods. Simple regression on outside temperature was found to be the best predictor, but this method was not without problems. The constrained timeline of the project allowed only a nominal three weeks each of pre- and post-tune-up data, but the final data sets were often smaller (from 9 to 18 usable pre-tune-up days and from 9 to 32 usable post-tune-up days) after deletion of days with known operating anomalies or monitoring equipment problems. The small data sets in turn yielded models that were not as well defined as would have been preferred. In addition, pre-tune-up and post-tune-up data sets typically had different ranges of outside temperatures, and necessarily spanned the earlier and later portions of the cooling season. Analysis of subsets of a few of the longer post-tune-up data sets found that the models varied in response to some inadequately normalized correlate of temperature or of date within the season. Another issue that emerged was a common pattern of increasing reference temperatures after the tune-up, for which we could find no plausible explanation (except perhaps moderate non-linearity of use versus outside temperature combined with the higher average temperature during the post-tune-up period).

Because of these problems, several different methods were used to calculate seasonal use from the regression models. One approach used was to perform separate regressions of the pre- and post-tune-up data, but to calculate the seasonal energy use from the regression parameters using a single season length and average temperature for both periods (the shorter season, which was the post-tune-up model season in all but one case). This minimizes the effect of apparently spurious changes in reference temperature. The second approach estimated percent savings from the sum of the daily residuals of the pre-tune-up period (with reference to the model from the post-tune-up period) divided by the sum of the daily use values for the pre-tune-up period. To minimize unadjusted seasonal effects, the residuals analysis was limited to that subset of days having an average outside temperature closest to the average of the days used to estimate the comparison model. This approach minimizes the dependence of the savings estimates on the exact parameters of the models and the results they yield at one specific temperature value. After extensive consideration of the results and the factors that might influence them, the electrical use savings were estimated using the separate regression/same cooling season approach for the six cases where the observed average temperature in one data set was within two degrees of the seasonal average temperature estimated from the other data set. Savings for the other three sites were obtained using the temperature-matched average residual method comparing the pre-tune-up days to the post-tune-up model (since the post-tune-up model typically had more observations and higher estimating precision than the pre-tune-up model).

The energy savings ranged from 46.8% to -6.0%, with average savings of 10.9% (Table 2). This average savings is marginally non-significant ($p < 0.10$), as one might expect from such a small and variable set of results. The average absolute savings of 4191 kWh is heavily influenced by one site (GRO), and is probably an optimistic estimate of potential savings for the population as a whole. Applying the average percentage savings to the average pre-tune-up energy use for cooling produces an estimated average savings of 1936 kWh. This latter method of estimating average impact is conservative, as there is a statistically significant relationship between percentage savings and pre-tune-up energy use for the sample (even with GRO removed), causing the product of their means to be a biased, low estimator of mean savings. Thus these two estimates bracket the potential savings for the population, but without a larger sample, the expected savings range cannot be narrowed further.

Demand Impacts

The model chosen for determining demand reduction performed reasonably well. Figure 2 shows the data and regression lines for one data set. The R^2 for the cycling demand regressions was greater than 70% for ten of the sixteen regressions. Five of the sites showed effectively no change in the reference temperature of the cycling demand line between the pre- and post-tune-up data periods; this suggests that the cycling demand line is fairly reliably determined for these sites. Three other sites (ALL, BUS, and TRN) had changes greater than 8°F. (The ninth site, TAY, was analyzed using only the steady state regression lines and bin analysis, as the system was operated in an on-off fashion with little normal cycling). Eleven out of the eighteen steady-state demand regression lines had an R^2 of 80% or greater. R-squareds lower than 80% were a result of low dependance of use on outside temperature and not necessarily large scatter in the data. Both the tight grouping and low slope of the steady-state data are evident in Figure 2. The fractional bin analysis

Table 2. Electric Use Savings

	Seasonal Savings		
Site ID	kWh	SE (kWh)	Percentage
ALL*	1,584	848	17.8%
ANK*	-307	747	-6.0%
BUS**	589	2,585	3.8%
DEL**	5,151	5,409	16.6%
GRO*	26,540	4,165	46.8%
IBB*	4,891	1,269	29.1%
TAY*	-171	751	-2.7%
TOO*	-110	516	-2.7%
TRN**	-449	1,643	-5.0%
Average	4,191	---	10.9%
Std. Error	2,887	---	6.1%
Ave % Savings x Ave Pre Use = 1,936			

* Separate Pre-Post regressions, but same season length and same seasonal average temperature.

** Pre-residuals relative to Post regression model, temperature-matched subset.

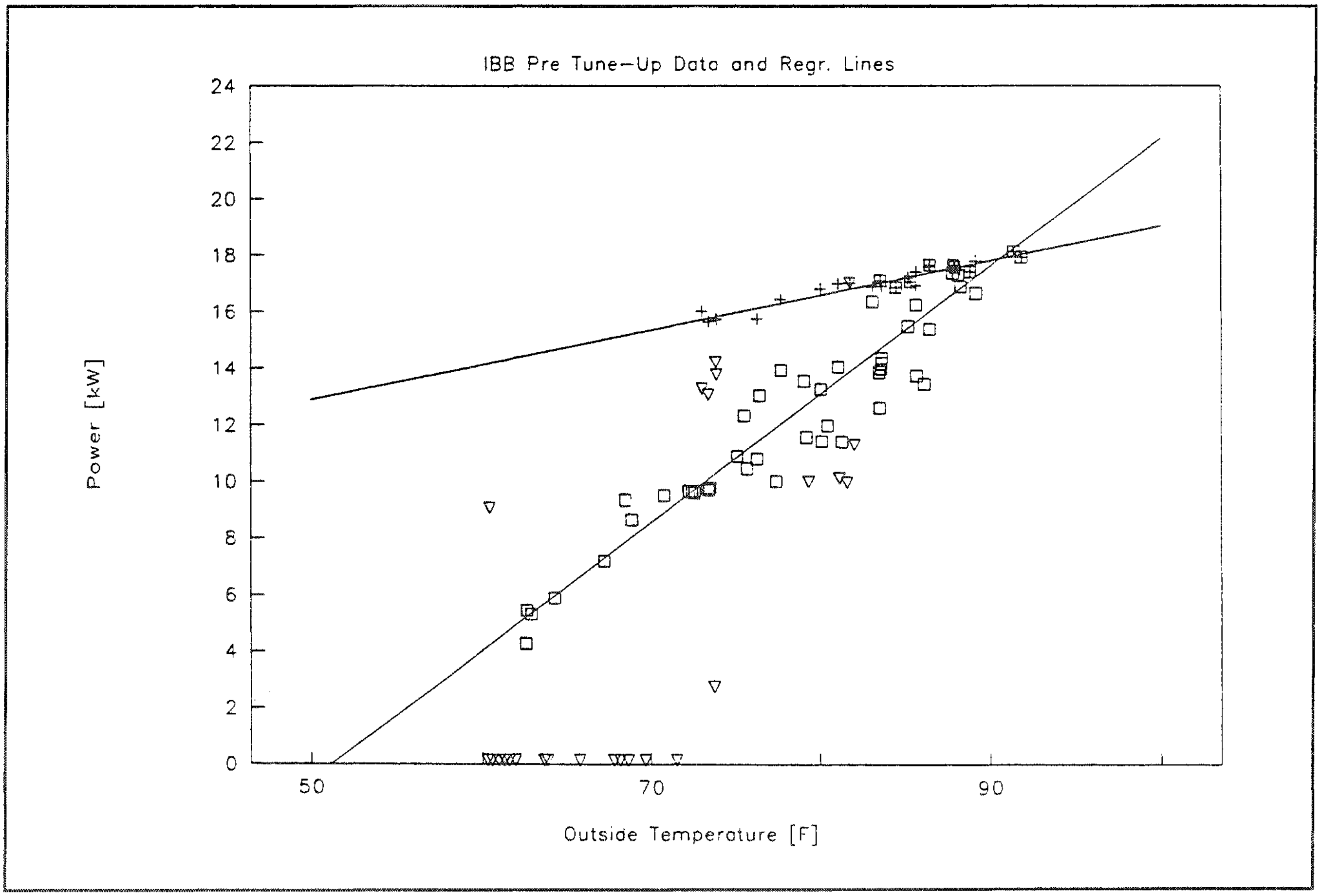

Figure 2. Two-Hour Average Demand

gave reasonable results and only affected 5 of the 9 sites. Of these sites, the effect on demand was 16% or less for temperatures above 77°F.

Table 3 shows the reductions in both two-hour average demand and steady-state power for temperatures ranging from 75°F to 95°F. The savings for each site is shown in terms of kilowatt reduction and percent reduction. The overall savings percentages were found by averaging the percent decreases for the individual sites.

Six of the nine sites showed a reduction in two hour average demand at 95°F, leading to an average absolute decrease of 1.89 kW. However, this absolute average is heavily influenced by the decrease at the largest site (GRO), and may well be an optimistic estimate of the potential for demand reduction in the population as a whole. The average percentage decrease in two hour demand was 2.1%. Applying this to the average pre-tune-up demand of 20.26 kW produces an estimated average demand reduction of 0.43 kW. This estimate is conservative, as there is a statistically significant relationship between percentage demand reduction and pre-tune-up demand. Thus, these two estimates bracket the potential demand reduction for the population, but without a larger sample, the expected demand reduction cannot be determined more precisely.

The relative savings were larger at lower outside temperatures, making the average demand reduction calculated from the average percent decrease times average pre-tune-up demand nearly constant over the range of outside temperature. Although the overall average savings did not vary greatly with temperature, many of the sites showed large variations in savings (both kilowatt and percent) with outdoor temperature. For GRO, IBB and TAY, the tune-up effects on demand for temperatures 80°F and above were highly significant ($p<0.001$). The changes for ALL, ANK, DEL, and TRN were also significant ($p<0.01$) for some of the reductions shown in the table.

The changes in steady-state power were much more modest than the changes in two-hour average demand. The dependance of steady-state power reduction on outside temperature for the individual sites also tends to be much less. At 95°F there was an average reduction of

Table 3. Demand Reduction

		Two-Hour Demand*			Steady-State Power		
Site		75F	85F	95F	75F	85F	95F
ALL	Decrease (kW)	1.68	1.19	-0.57	-0.43	-0.50	-0.57
ALL	% Decrease	29.6%	8.7%	-3.2%	-2.8%	-3.0%	-3.2%
ANK**	Decrease (kW)	-0.09	-1.07	-2.04	0.35	0.40	0.45
ANK**	% Decrease	-3.5%	-20.5%	-26.2%	2.7%	2.9%	3.1%
BUS	Decrease (kW)	-1.92	0.25	2.41	0.86	0.13	-0.61
BUS	% Decrease	-17.3%	1.3%	8.8%	2.7%	0.4%	-1.7%
DEL	Decrease (kW)	1.73	2.36	2.95	0.30	0.65	1.00
DEL	% Decrease	14.4%	14.9%	15.2%	1.6%	3.3%	4.8%
GRO	Decrease (kW)	2.96	9.42	12.86	-0.89	-0.40	0.10
GRO	% Decrease	16.3%	20.8%	19.7%	-1.6%	-0.6%	0.2%
IBB	Decrease (kW)	0.93	1.17	1.57	1.37	1.47	1.57
IBB	% Decrease	8.9%	7.6%	8.5%	8.6%	8.5%	8.5%
TAY***	Decrease (kW)	-0.09	-0.43	-0.53	-0.48	-0.50	-0.53
TAY***	% Decrease	-7.0%	-7.0%	-7.0%	-7.0%	-7.0%	-7.0%
TOO	Decrease (kW)	0.18	0.12	0.06	-0.06	0.03	0.11
TOO	% Decrease	5.0%	2.3%	0.9%	-0.5%	0.2%	1.0%
TRN	Decrease (kW)	0.72	-0.04	0.27	0.57	0.42	0.27
TRN	% Decrease	9.3%	-0.4%	2.3%	5.4%	3.8%	2.3%
Per Site	Pre (kW)	8.05	15.09	20.26	20.10	21.31	22.53
Per Site	Average Decrease (kW)	0.68	1.44	1.89	0.18	0.19	0.20
Per Site	Avg. of % Decrease	6.2%	3.1%	2.1%	1.0%	0.9%	0.9%
Per Site	Std. Error of Decrease	4.6%	4.1%	4.5%	1.5%	1.5%	1.5%
Per Site	Avg % Decrease X Avg Pre kW	0.50	0.47	0.43	0.20	0.20	0.20

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* Either cycling or steady-state operation depending on outside temperature and site regressions.

** The large increase for ANK is because the tune-up fixed a previously unused compressor (this would probably not have been done in a regular tune-up).

** For Taylor the steady-state power regression lines are used with a combined pre and post tune-up bin model.

0.9%. There were significant changes in steady-state power for some of the sites, but these changes did not necessarily correlate with changes in the cycling demand.

The results of combining data on the temperature at the time of the system peak each month with two hour average demand as a function of temperature are shown in Table 4. The absolute averages show a maximum diversified demand savings of 1.74 kW, in July, while the product of the average percent demand reduction and the average pre-tune-up demand shows a maximum diversified demand savings of 0.47 kW per site, in September. As mentioned previously, the former estimate is thought to be optimistic, while the latter is conservative.

Table 4. Monthly Utility Demand Reduction

Month	Outside Temp. (°F)	Average Decrease (kW)	Avg. Percent Decrease X Pre kW
May	74.1	0.84	0.36
June	83.5	1.37	0.45
July	89.8	1.74	0.44
August	85.6	1.45	0.46
Sept.	81.6	1.25	0.47
Peak	89.7	1.74	0.44

One utility concern about a tune-up program is that capacity increases could lead to higher demand if the air conditioners are undersized and must run continuously. According to the pre-tune-up model, the air conditioners would run continuously at five of the sites when it is 95°F outside. Three of these sites showed reduced demand and two showed increased demand after the tune-up. At the average temperature of the utility's summer peak, 90°F, none of the sites would have had continuous air conditioner operation according to the pre-tune-up models. Therefore, increases in capacity (and resulting increases in steady-state power) would not have increased the two-hour average demand, unless the efficiency was decreased. Moreover, comparison of the onset of continuous operation for all sites before and after the tune-ups did not show any strong overall tendency of either increasing or decreasing capacity.

Comparison of Monitored Results with Estimates of Savings Based on Diagnostic Model and Based on Work Done

The diagnostic spreadsheet did well at predicting the change in steady state power at 95°F (R^2 = 65.7%, highly significant), but very poorly at predicting the more important cycling demand and energy use (R^2 = 6.1% and 2.7%, respectively). Several analyses were done in an attempt to understand the poor results. Of most interest, the manual measurements of power, supply and return dry and wet bulb temperature, and outdoor dry bulb temperature, which enter heavily into the power, capacity and EER calculations and into the off-design correction factors, were compared with monitored data taken by the computerized data acquisition equipment. This included power and outdoor dry bulb temperature for all systems, and supply and return dry and wet bulb temperature for six intensively monitored systems. The manual and monitored steady state power and outdoor dry bulb temperature measurements agreed well (R^2s of 94% and 88% and slopes very close to 1). However, the data on enthalpy drop across the coil did not agree at all (R^2 = 0.9%). This large error in enthalpy measurements is thought to be the primary reason for poor agreement between the spreadsheet and monitored data on cycling demand and energy impacts. Enthalpy estimates are extremely sensitive to wet bulb temperature, and although the manual measurements were taken with reasonable care, it is possible that steady state measurements were not obtained. As yet it is uncertain whether these measurements could be improved enough to make the diagnostic spreadsheet as useful in practice as it is in concept. Assessments of possible bias or error due to differences between average pre-test and post-test weather and return air conditions or due to poor fit of the DOE 2 equations implicit in the spreadsheet to the systems modelled are currently incomplete, but work to date does not suggest that these are significant sources of error.

The qualitative estimates of the expected impacts, based on the work done and measurements taken, were made prior to the availability of the monitored results (i.e., in a "blind" fashion), based on previous laboratory and field work relating changes in power, capacity and EER to changes in air flow and charge. These qualitative estimates were then compared with the monitored results.

The relationship between the percent change in steady state power predicted by the qualitative method and the measured change in steady state power is very poor (R^2 = 5.7%). The predictions of impact on energy use and cycling demand at 95°F are substantially better than those from the diagnostic spreadsheet, though not quite reaching statistical significance (R^2s of 36.8% and 43.6% respectively). The average predicted change in energy use is within 2% of the measured change. The average predicted change in cycling demand is considerably larger than the measured change. The predicted change in cycling demand assumes that the systems could meet the cooling load at 95°F in the pre-tune-up period, and that therefore, any increases in capacity would result in decreases in cycling demand, whereas in reality, the systems at five of the nine sites could not meet the cooling load at 95°F, which may account for part of the discrepancy. At present, the qualitative estimates appear to offer better estimates of energy and demand savings than the diagnostic spreadsheet. They also require somewhat fewer field measurements, though they do require interpretation by a knowledgeable person who understands the response of vapor compression systems to various perturbations.

Costs

In order to estimate the cost of a tune-up performed as part of a program, costs for tasks specific to the research project were deleted, based on detailed timekeeping during the tune-ups. In addition, the costs of the performance tests necessary to use the diagnostic spreadsheet were eliminated. With these costs excluded, labor and material costs for the tune-ups were still quite high, averaging $824 per site for refrigeration work and $334 for duct work, with labor at $48 and $30 per hour respectively, for a total of $1158 per site (Table 5) or $574 per system. For a full scale program, some increase in productivity and decrease in costs could be expected, but administrative costs would also have to be added.

It may be possible to streamline the tune-up to reduce costs without degrading savings. As a point of comparison, Proctor (1991) estimated program costs of $50 per system to correct airflow and $100 per circuit to correct overcharge. Applied to the averages of 2 systems and 2.8 refrigerant circuits per site for this project, this implies an average of $380 per site for the refrigeration work, compared with the total of $824 for this sample (duct sealing costs between the two projects are not comparable). By limiting the tune-up protocol to reducing overcharges and increasing low airflow, Proctor has reduced the cost of refrigeration work on current residential projects to $110 per system (one refrigeration circuit) (John Proctor, Proctor Engineering Group, pers. comm., 1992).

Cost-Effectiveness

Customers' avoided demand and energy costs were estimated based on the electric rates for each specific site. Average annual savings were $390. The median payback is over 6 years, while the aggregate payback (total savings divided by total cost) is almost 3 years (Table 5).

Cost effectiveness was also evaluated from a societal perspective, for the participant sample and as a function of pre-tune-up air conditioning consumption derived from billing data. This analysis was performed using the utility's DSM cost-effectiveness screening tool, which computes benefit/cost ratios for particular measures, given annual energy savings, monthly demand impacts, measure cost, measure life, and projected costs of energy and capacity avoided by the measure over its lifetime. The analysis for the study participants used the installed cost of $1158 per site plus 20% assumed program administration costs, and savings computed in two ways: average percent savings multiplied by average pre-tune-up air conditioning use and demand (the conservative estimate)

Table 5. Tune-Up Economics

Site	Duct Cost	Refrig. Cost	Annual Savings	Payback (years)
ALL	$360	$694	$172	6.14
ANK	$388	$778	($40)	--
BUS	$650	$1,588	$55	41.02
DEL	$234	$847	$439	2.46
GRO	$328	$1,699	$2,369	0.86
IBB	$353	$453	$370	2.18
TAY	$298	$420	($13)	--
TOO	$128	$487	($8)	--
TRN	$265	$452	$171	4.20
Median	$328	$694	$171	6.14
Average	$334	$824	$390	2.97 *
Std.Dev.	$142	$490	$761	--

* Based on average costs and savings.

and average absolute savings (the optimistic estimate). A measure life of 4 years was assumed. The conservative savings estimate yielded benefit/cost ratios less than one, while the ratio based on the optimistic savings estimate just barely exceeded one if environmental externalities were included.

Both savings and costs were found through regression to increase with pre-tune-up air conditioning consumption, but savings increase more rapidly than costs, so that cost-effectiveness improves with increasing pre-tune-up energy use. This relationship was used to determine a cost-effectiveness threshold which could be used to screen potential program participants. On this basis, a tune-up could be expected to be cost-effective if the pre-tune-up cooling use exceeds 18,500 kWh/y, giving no allowance for environmental externalities. If externalities are included, the threshold is a cooling use of 9,750 kWh/y.[1] This analysis is based on regressions against pre-tune-up cooling use as estimated from customer billing data, so the screening process could be applied directly to data already available in the customer billing data base.

Conclusions and Recommendations

The air conditioning systems were found to be in somewhat better condition on average than anticipated. Though airflows were low on average, they could potentially be increased in only 5 of 18 systems through tune-up procedures. Only about half the evaporator coils were visibly dirty, while none of the condenser coils were. Though substantial changes in charge were made in many of the refrigerant circuits, the median change was zero, limiting the potential impact on steady state demand. All of the TXVs were found to be giving improper superheat, but only 4 of 13 were adjustable. Significant duct leakage was found in 7 of 18 systems.

The project demonstrated that an advanced diagnostic tune-up procedure, when properly applied to commercial unitary cooling equipment, can achieve energy and demand savings. Although monitored results of tune-ups performed varied widely, on average air conditioning energy use was reduced 11%, and two-hour average demand at 95°F was reduced 2.1%.

Both seasonal variations in energy use relationships across the cooling season and inconsistent operation of thermostats made analysis of the data problematic, but methods of dealing with these problems were developed and used successfully to quantify annual impacts using data collected in short (three-week) pre- and post-tune-up measurement periods.

A "qualitative" method for estimating annual energy savings based on on-site tune-up diagnostics was shown to agree fairly well with monitored results; such a method would be an essential component of wide scale implementation of a tune-up program, as an inexpensive means of impact evaluation. Demand savings were overestimated by the method, indicating that further development is needed.

The results indicate that it may be necessary to screen small commercial customers on cooling energy use to develop a cost-effective program. It appears that a tune-up program would be cost-effective, using a societal test and including a credit for environmental externalities, if targeted to customers whose cooling energy use exceeds 9,750 kWh/y. Without externalities, a program would be cost-effective if targeted at customers with cooling use over 18,500 kWh/y. An important caveat is that both of these cutoff criteria exclude most of the customers in the pilot study on which the savings projections are based.

Further field tests would be extremely useful to validate the results from this small sample. The protocol and quality control procedures, though well beyond standard contractor procedures, could perhaps be further refined to increase savings and reduce costs.

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Endnote

1. The avoided cost of environmental externalities used here is calculated by applying unit impacts specified by the Massachusetts Department of Public Utilities (in dollars per ton of emissions) to the avoided emissions of marginal plants.

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