New Technology Field Evaluation of Gas-Engine-Driven Rooftop Air-Conditioning Equipment at the Willow Grove (PA) Naval Air Station

Peter R. Armstrong, Srinivas Katipamula, and David R. Conover, Pacific Northwest Laboratory

The first new technology demonstration in the program involves three 15-ton gas-engine-driven air conditioning installations at the Naval Air Station. Two of the installations are rooftop units serving the Base Exchange (BX), a 15,000 ft^2 retail store. The two rooftop units have been monitored since June 1992. The third gas-engine-driven machine is a 15-ton split system serving a library and office space in a 4,000 ft^2 wing of Building 3.

The performance of the gas-engine-driven equipment serving the BX has been characterized by measuring gas and electric input and air-side output. The instrumentation and analytical procedures are described. The energy and operating cost savings realized at the site and the national resource potential are also reported.

Introduction

Federal government facilities use about 1.5 x 10^{15} Btu (2.5% of national consumption) at a cost of nearly $10 billion annually. Through its office of the Federal Energy Management Program (FEMP) the U.S. Department of Energy (DOE) provides technical support to help federal agencies reduce energy consumption and energy costs in their buildings and facilities. The New Technology Demonstration Program (NTDP) is a FEMP program designed to accelerate adoption of new U.S. energy-related technologies at federal sites. Operation of the NTDP at Department of Defense (DoD) facilities is supported by the Strategic Environmental Research and Development Program (SERDP).

By involving a federal site, its serving utility, a new technology manufacturer, and an interested trade or industry organization in a collaborative effort, the U.S. Departments of Defense and Energy can evaluate a large number of new technologies and disseminate the demonstration results to all federal energy managers at modest cost.

This paper reports on the field performance evaluation of a new U.S. cooling technology that has been installed for the first time at a federal facility. The demonstration and evaluation were made possible by cooperation of the U.S. Department of Energy Federal Energy Management Program, the American Gas Cooling Center (AGCC), Philadelphia Electric Company (PECO Energy), Thermo King Corporation, and the Naval Air Station at Willow Grove (NASWG).

Technology Description

The first gas-powered cooling technology selected for the New Technology Demonstration Program was a 15-ton natural gas-engine-driven rooftop air conditioning unit. The technology is designed as a direct replacement for conventional single-zone, gas-heating, electric-cooling rooftop units. A cutaway view of the unit, showing the main mechanical components, appears in Figure 1. Each unit is factory assembled, charged with R-22 (an HCFC refrigerant), and requires single-point connections of electrical power (for the supply air fan motor) and natural gas. Maximum cooling capacity is 190,000 Btuh at ARI Standard 360 (ANSI/ARI 1986) rating conditions. Maximum heating capacity is 216,000 Btuh. Variable engine speed allows for cooling operation in three stages, and cylinder unloading provides an additional stage for light load conditions as indicated in Table 1. Other characteristics of the equipment are given below.

Condenser. A 5-row coil with a face area of 15 ft^2, tube diameter of 3/8 in., and fin spacing of 12 per inch is provided to reject heat from the condensing refrigerant.

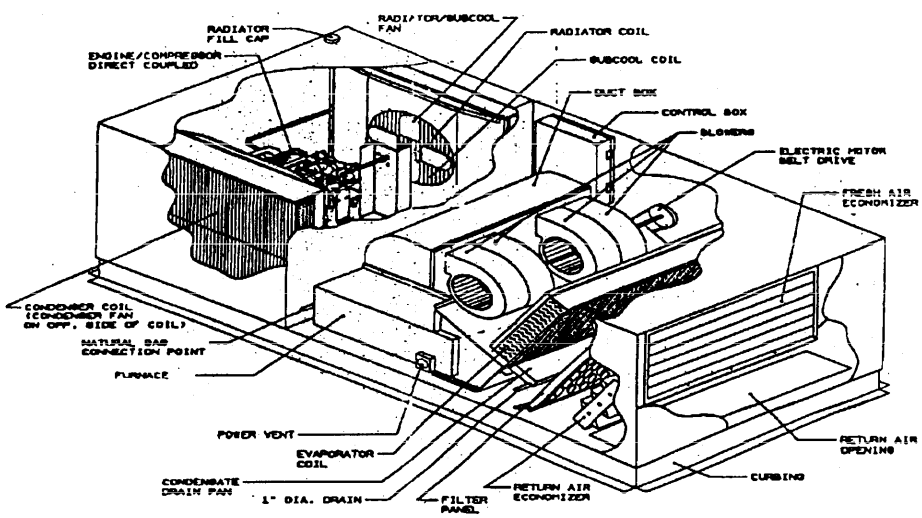

Figure 1. Cutaway View of the Gas-Engine-Driven Rooftop Unit

Table 1. Capacity and Related Operating Parameters

Stage	Mode	Capacity (kBtuh)	Engine Speed (rpm)	Supply Air Fan (speed)	Supply Air Fan (hp[b])
H1	Heating	108	n.a.	full	5
vH2	Heating	216	n.a.	full	5
C1	Cooling	28	1000[a]	half	1
C2	Cooling	84	1000	full	5
C3	Cooling	143	1500	full	5
C4	Cooling	190	2400	full	5

(a) With two of four compressor cylinders unloaded.
(b) Fan power depends on distribution system flow-pressure characteristic. Nominal values for the standard motor, as installed at NASWG are shown.

The engine-driven condenser fan is 24 inches in diameter and provides 9000 cfm of air flow with the engine running at 2400 rpm.

Evaporator. The 6-row evaporator coil has a face area of 13 ft^2, tube diameter of 3/8 in., and fin spacing of 9 per inch. A 12-ton expansion valve is provided for the upper (75%) section of the evaporator and a 3-ton expansion valve is provided for the lower (25%) section of the evaporator. Both are external equalizer type with a superheat setting of 20(±5)°F.

Compressor. The 4-cylinder V-configuration compressor displaces 30 in^3, and cylinder unloading is provided on two cylinders. The refrigerant loop holds a charge of approximately 30 lbm of R-22. Cooling capacity is varied to meet thermal load by changing engine/compressor shaft speed and (for the lowest stage) by unloading two of four compressor cylinders. The rooftop unit can, therefore, operate in one of four cooling stages designated C1 through C4. The nominal capacities are 28,000[1] Btuh in cooling stage 1 (Cl), 84,000 Btuh in stage 2 (C2), 143,000 Btuh in stage 3 (C3), and 190,000 Btuh in stage (C4).

Engine. A 4-cylinder in-line engine displacing 163 in^3 produces 40 hp at 2400 rpm. An 18-inch-diameter fan provides 4000 cfm through the engine radiator and refrigerant subcooler at 2400 rpm engine speed. The units serving the BX were equipped with mechanical distributors but later models employ distributorless electronic ignition systems.

Supply Fan. Two centrifugal blowers on a common, belt-driven shaft provide nominal supply airflow rates of 6000 cfm at full speed and 3000 cfm at half speed. A 208-volt, 3-phase, 5-hp two-speed (1720/860 rpm) motor with an adjustable drive sheave powers the blowers. Table 1 gives the supply fan motor speed for each heating and cooling stage. The air-side design is typical of small to medium sized rooftop units in not incorporating a return fan or an exhaust air path.

Economizer. A single actuator motor operates a pair of opposed blade dampers in the return air and outside air streams; this subsystem allows modulation of the outside air fraction from 0 to 100%. Economizer operation is controlled by the thermostat based on outdoor and return air temperatures. (Later models use differential enthalpy control.)

Maintenance. The rooftop unit requires biannual preventive maintenance which includes coil cleaning and supply fan and damper maintenance identical to the maintenance required for the conventional technology. Maintenance peculiar to the gas-engine-driven technology includes engine winterization at the end of each cooling season and an oil change every spring. Replacement or refacing of valves, seats, and guides is recommended by the manufacturer at 10,000-operating-hour intervals, and engine rebuilding is recommended at 20,000-operating-hour intervals. These rebuild intervals appear to be realistic based on natural-gas-fueled engine operating experiences of others (Mathews, 1992).

Thermostat. A programmable thermostat is used to control the two heating and four cooling stages, the economizer, and night setback and temperature recovery operation of each rooftop unit. The thermostat uses proportional-plus-integral control to determine the required heating or cooling stage and the cycling rate that will minimize deviations from the setpoint. The thermostat also provides intelligent outside air control and ramped setpoint recovery from night setback. The heating and cooling stages are enumerated in Table 1, where the heating or cooling capacity and related operating parameters are given for each stage.

Field Evaluation

The retail sales area of the Base Exchange (BX) covers 12,500 ft^2 and was previously served by two conventional rooftop units that used No. 2 fuel oil for heating and electric-motor-driven vapor compression systems for cooling. One of the new rooftop units is shown in Figure 2; high resolution gas metering instruments are visible in the photo. The arrangement of the units relative to the air distribution system and thermostats is shown in the plan view, Figure 3.

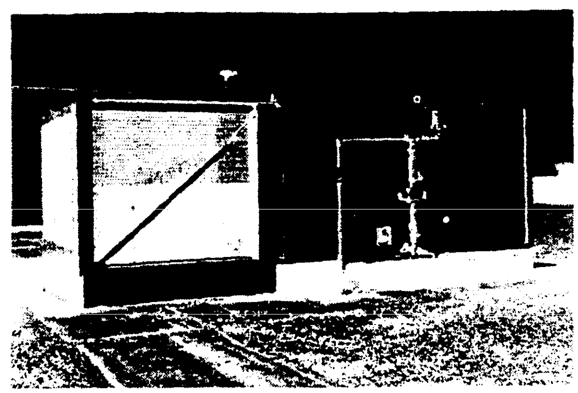

Figure 2. Unit 1: View from the South

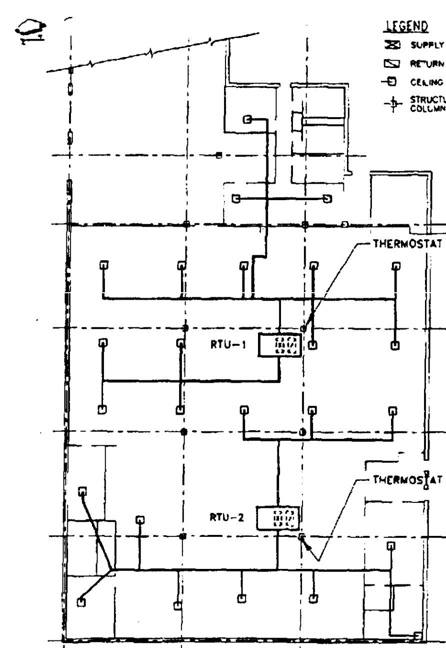

Figure 3. Location of Thermostats in Relation to Rooftop Units and Air Distribution Systems

To evaluate the new technology, the building and the new gas-engine-driven units were instrumented and monitored. Data collected during the demonstration included outdoor and indoor temperature and humidity, gas and electric consumption of the equipment, refrigerant pressures and temperatures, and operating time associated with different stages of the units. The measurements, generally averaged and recorded at the end of each 15-minute scan interval, were used to evaluate the equipment performance and energy use and to diagnose operational problems. Additional measurements were taken to provide redundant measures of performance and for monitoring system diagnostics. The monitoring points are shown in the refrigerant-side schematic, Figure 4, and air-side schematic, Figure 5. The sensor specifications are reported by Armstrong and Conover (1992). The sensors used at the monitoring points shown in Figures 4 and 5 are described in Table 2.

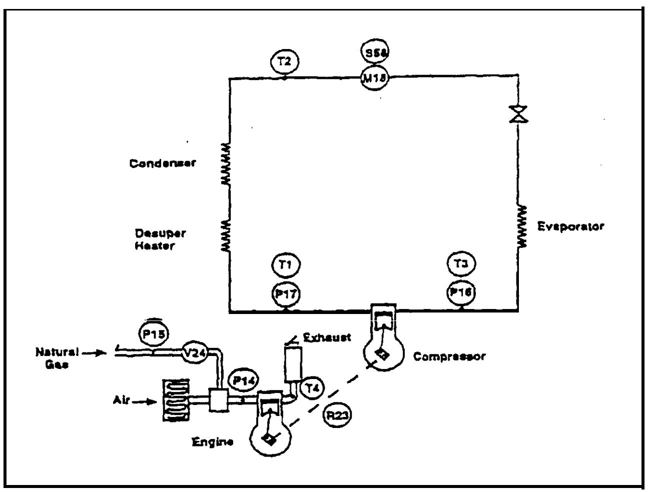

Figure 4. Sensor Locations in Refrigerant Loop

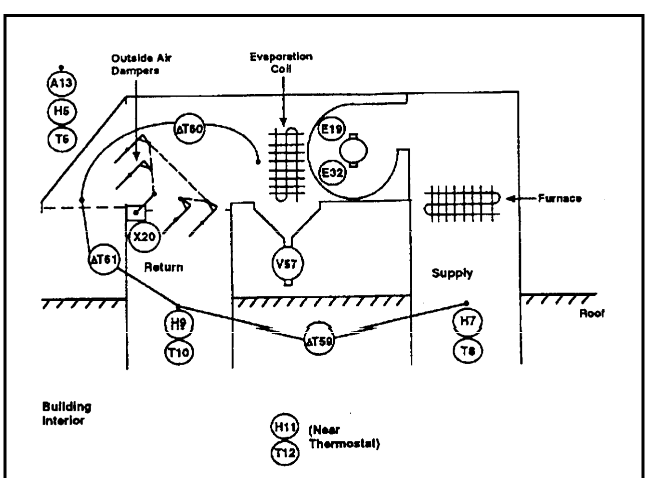

Figure 5. Sensor Locations in Zone and Air-Side Loop

Performance Analysis

The field data were analyzed to determine seasonal efficiencies in each cooling and heating stage, to determine overall seasonal efficiencies, and to determine what, if any, cost-effective operational improvements could be obtained.

Measured Cooling Performance

A model is needed to extrapolate the field evaluation data to a normal weather year. To this end, the relation between daily input energy (cooling mode) and ambient temperature was explored. Figure 6 shows the daily gas loads for Units 1 and 2 combined as a function of ambient temperature. Data for Mondays are not shown because the BX is closed Mondays and the cooling loads are therefore lower (as expected) on Mondays. The regression line ($r^2 = 0.7$) indicates a combined load of 165 kBtuh per Fahrenheit degree above a mean daily temperature of 60°F on business days. However, one unit used almost twice as much fuel as the other unit. A small difference in the room temperature setpoints of the two units may have caused the load imbalance.

The coefficient of performance (COP) for unit 1 was computed from air-side sensible and latent loads and gas input energy rate. The COP was relatively constant in each cooling stage, ranging between 0.5 and 0.6 in C1, between 0.8 and 1.0 in C2, between 0.65 and 0.75 in C3 and between 0.5 and 0.6 in C4. The overall daily COP was found to track the stage 1 (C1) COP on most days as shown in Figure 7. However, a second-order effect, due to the influence of C2 and C3 COPS, gave somewhat higher COPS on higher load days.

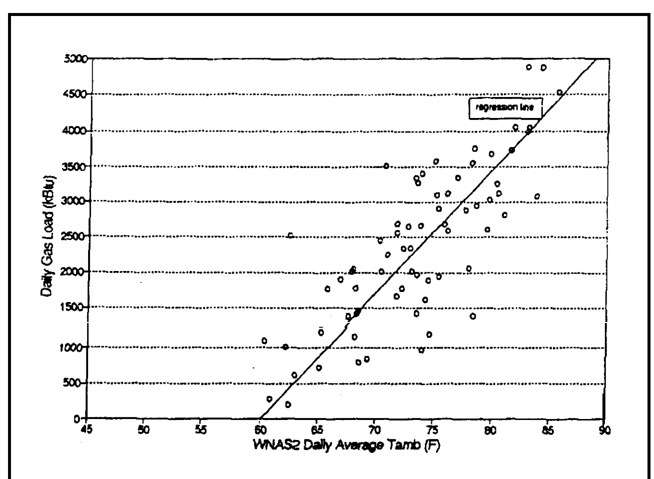

Figure 6. Relation Between Daily Gas Used for Cooling and Mean Daily Temperature

Table 2. Sensor Specifications

Point Code [a]	Point Description	Input Type [b]	Sensor Range	Normal Range	Unit[c]	Sensor Type	Sensor Manufacturer	Sensor Model
T1	Refrigerant discharge	DE	-340..1112	0..250	°F	Pt RTD	Hy-Cal	RTS-64-T-100-3-12-X1
T2	Refrigerant subcool	DE	-340..1112	0..150	°F	Pt RTD	Hy-Cal	RTS-64-T-100-3-12-X1
T3	Refrigerant suction	DE	-340..1112	0..50	°F	Pt RTD	Hy-Cal	RTS-64-T-100-3-12-X1
T4	engine exhaust	DE	-20..1000	0..600	°F	TC, type J	Omega Engrg	TJ36-CASS-116-12
T51	T.C. reference	SE	-13..122	-13.122	°F	Thermistor	Campbell Scientif	10TCRT
H5	Outside air	SE	0..100	0..100	%RH	Polymer	General Eastern	RHT-2-l-OA
T6	Outside air	SE	-20..140	-20..100	°F	Pt RTD	General Eastern	RHT-2-l-OA
H	Supply air	SE	0..100	0..100	%RH	Polymer	General Eastern	RHT-2-l-R
T8	Supply air	SE	-20..140	50..140	°F	Pt RTD	General Eastern	RHT-2-l-R
H9	Return air	SE	0..100	0.100	%RH	Polymer	General Eastern	RHT-2-l-R
T10	Return air	SE	-20..140	60..80	°F	Pt RTD	General Eastern	RHT-2-l-R
H11	Room air	SE	0..100	0..100	%RH	Polymer	General Eastern	RHT-2-l-R
T12	Room air	SE	-20..140	60..80	°F	Pt RTD	General Eastern	RHT-2-l-R
A13	Barometer	SE	0..25	14.5..15	psia	strain gage	Setra	C280E
P14	Intake manifold vac	SE	0..10	0..5	psig	strain gage	Setra	C239
A15	Gas supply	SE	0..25	14.5..15	psig	strain gage	Setra	C280E
P16	Compressor suction	SE	0..250	50..200	psig	strain gage	Setra	C207
P17	Compressor discharge	SE	0.500	150..350	psig	strain gage	Setra	C207
M18	Refrigerant mass flow	SE	0..60	5..45	ppm	inertial	Micromotion	D-40-119
E19	Total Electric power	SE	0..6000	0..4000	W	Hall effect	Ohio Semitronics	WOE-23ET/60K
X20	Outside damper positn	SE	0..100	0..100	%	rheostat	Honeywell	Q181A-1007
R23	Engine RPM	P	0..5000	0..2600	rpm	timing ckt	Transwire	XP160-SP153-lSO
V24	GAS ACFH	P	0..1000	0..300	cfh	PD lobe	Roots	8C175psig050932-101
S25	Cooling step 1	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S26	Cooling step 2	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S27	Cooling step 3	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S28	Cooling step 4	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S29	Heating step 1	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S30	Heating step 2	IN	1 = ON	0..900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
S31	Engine enable	IN	1 = ON	0.900	s	LED Relay	Stevens Engrg	RH2B-UL-AC24V
E32	Electric power pulse	P	0..1500	0..1000	Wh	Hall effect	Ohio Semitronics	WOE-23ET/60K
DT59	Return-outside air	DE	+/-200	+/-25	°F	TP, type T	C.S. Gordon	T24-2-505
DT60	Mixed-outside air	DE	+/-200	+/-75	°F	TP, type T	C.S. Gordon	T24-2-505
DT61	Supply-return air	DE	+/-200	+/-75	°F	TP, type T	C.S. Gordon	T24-2-505
V5	Condensate drain	P	0..10	0..1	pps	tip bucket	Texas Electronics	501

(a) T = temperature, DT = differential T, H = humidity, S = contact status, M = mass flowrate, E = electric power, A = absolute pressure, P = gauge pressure, R = rotational speed, V = volumetric flowrate

(b) DE = differential, SE = single-ended, P = pulse count, IN = integrated ON-time

(c) °F = degree Fahrenheit, ppm = lbm per minute, pps = pulse per second, psia = ibf per inch^2 absolute, psig = ibf per inch^2 gauge, %RH = percent relative humidity, mV = millivolt, rpm = revolution per minute, W = watt, Wh = watt hour, cfh = ft^3 per hour, s = second; all units apply to both the Sensor Range and Normal Range columns.

The measured COPS are compared to the COPS quoted by the manufacturer (Thermo King 1990, p. 25) in Figure 8. The C1 COP was almost double the quoted value[2], the C2 COP was very close to the quoted value, and the C3 and C4 COPs were both about 10% less than the quoted values.

The *seasonal* COP is, in simplest terms, the sum of the part-load COP and part-load operating time products divided by the total operating time. The seasonal COP for unit 1 is substantially lower then the integrated part-load value (IPLV) quoted, due mainly to longer operation in stage 1 and shorter operation in stage 2 relative to the IPLV shares of operating time assumed by the manufacturer. The measured and ARI standard part-load curves are compared in Figure 9.

The units consumed 190 million Btu (MBtu) of fuel (based on the higher heating value of natural gas) and provided 115 MBtu of cooling effect during the monitoring/analysis period.

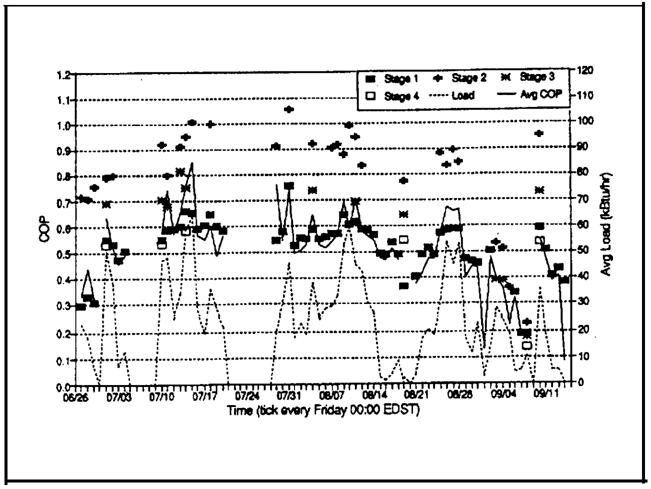

Figure 7. Average Daily COP by Cooling Stage

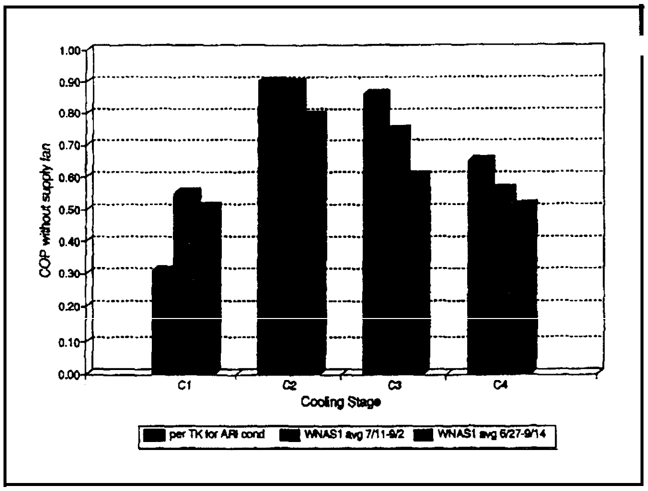

Figure 8. Measured COPS Versus Rated COPS

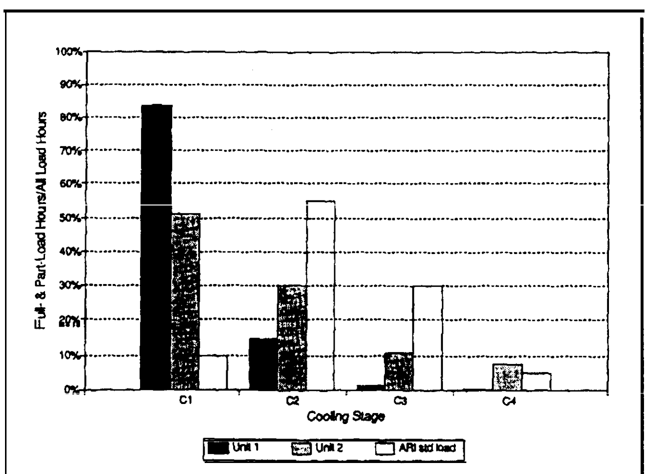

Figure 9. Unit 1 Measured Part-Load Operating Times Versus ARI Values for Seasonal COP

Cooling Operational Improvements

A number of opportunities for improved operation were identified through analysis of the field performance data.

Scheduling. The programmable thermostats provide seven-day scheduling of heating and cooling setpoints and fan operation. The BX operates from 8:00 am to 6:00 pm Tuesday through Friday and from 10:00 pm to 6:00 pm Saturday and Sunday. The BX is closed on Monday. The daily profiles in Figure 10 indicate that the thermostats are not correctly programmed for unoccupied operation from 00:00 am Monday through 07:00 am Tuesday. The cooling season savings that will be realized by correcting the schedule programs are about 40 MBtu/yr in cooling fuel and about 860 kWh/yr in fan energy.

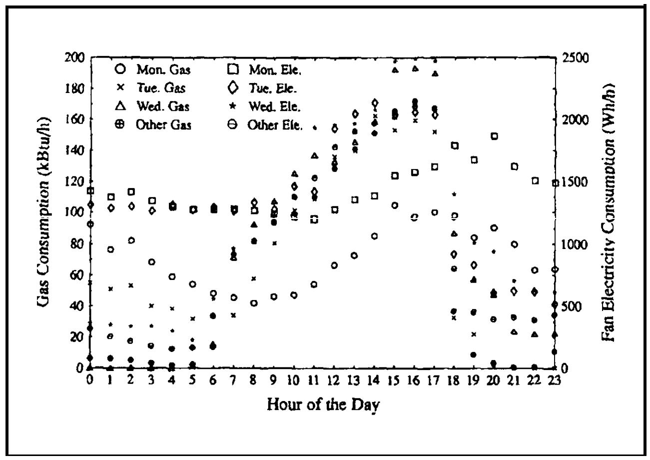

Figure 10. Average Day Profiles in the Cooling Season

Economizer operation and Control. Figure 11 shows the effect on daily cooling load of economizer operation. The large incidence of increased load is due to the latent load associated with outside air on days when outside air is cooler, but more humid, than return air. Retrofit of differential enthalpy based economizer controls in both units will result in an annual savings of about 3 MBtu/yr and eliminate periods of excess humidity that sometimes occur with the existing controls.

Coordination and Staging. Unit 2 provided more cooling than Unit 1 throughout most of the test period. There was no indication, however, that this was due to a significant difference in load between the areas served by the two units. Rather, the cooling load imbalance was most likely caused by the thermostat location or a small difference in setpoint. Connecting the two distribution systems to form a single-zone distribution system would likely not result in a noticeable degradation in temperature and humidity uniformity.

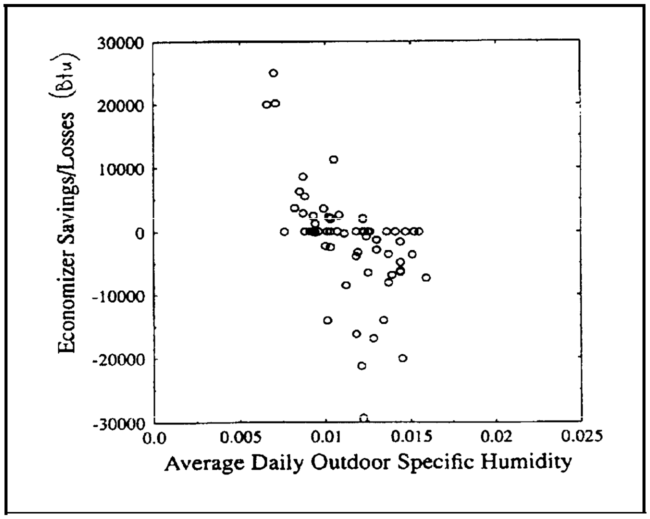

Figure 11. Reduction in Daily Cooling Load that Resulted from Economizer Operation

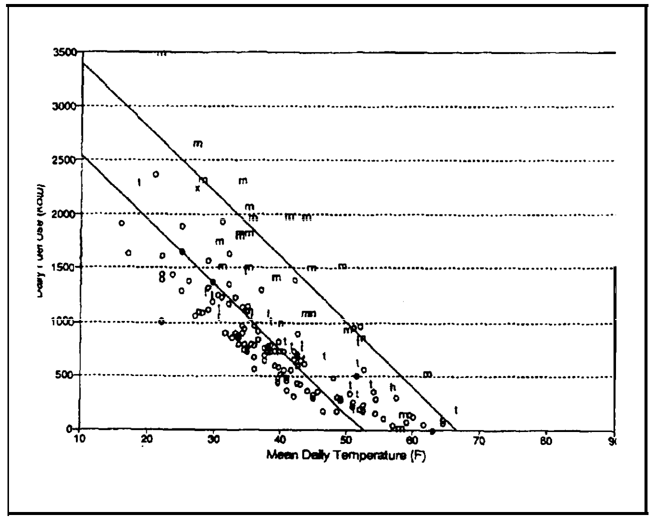

Figure 12. Relation Between Daily Gas Used for Heating and Mean Daily Temperature

A single zone system would have three benefits in this application. First, redundancy would be provided. If one of the two units failed (as happened) the loss of capacity would be spread over the entire zone with no resulting “hot spot. ” Second, equal wear of the two units would be ensured. The units could be operated in an alternating lead-lag arrangement for light loads and in tandem for heavy loads. Third, improved efficiency could be realized. Many times Unit 1 cycled between stage 1 and 2 cooling while Unit 2 cycled between stage 2 and 3 cooling. The aggregate load could have been met by operating both units at stage 2, the most efficient stage. Similarly, many times Unit 1 cycled between off and stage 1 cooling while Unit 2 cycled between stage 1 and stage 2 cooling. The aggregate load could have been met by operating one of the units at stage 2, the most efficient stage, most of the time.

Preventive Maintenance. The checklist for spring maintenance should include—in addition to engine, fan, filter, condensate drain, and coil maintenance-the following checks: economizer sensors, actuator, linkage, dampers and overall economizer control; minimum outside air setting; distribution system (obstructions and leaks) and building pressurization; thermostat setpoint and fan schedules.

Measured Heating Performance

The fuel consumption for heating was essentially equal to the fuel consumption that would have occurred using comparable gas heating/electric cooling rooftop units in place of the subject units. Daily fuel use in the 1992-93 heating season responded to mean daily outdoor temperature as shown in Figure 12. The data suggest a balance point of about 67°F on Mondays (BX closed) and 53°F on days when the BX was open for business. A specific fuel rate of about 60 kBtu per °F-day is also apparent. Total heating energy for the 1992-93 season was 159 MBtu of fuel. The measured annual heating efficiency was 80%, which is consistent with the 81% rating quoted by the manufacturer.

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Economic Analysis

The market potential for this technology hinges on its aggregate life-cycle cost with respect to existing or conventional technologies. Aggregate life-cycle cost is simply the sum, over all potential applications, of the life-cycle cost (expressed, e.g., as a net present value) in each cost-effective application. In this section we evaluate the life-cycle cost at the NASWG BX.

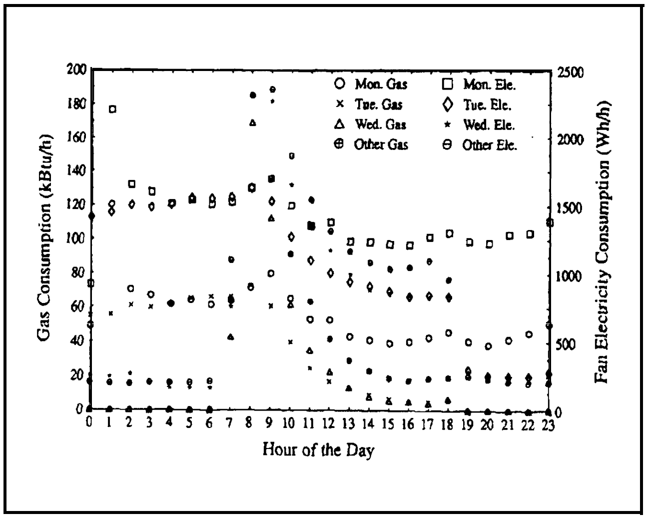

Figure 13. Average Day Profiles in the Heating Season

Normal Year Energy Use

The cooling load is predominantly a sensible cooling load, well correlated with mean daily temperature. The balance point (temperature above which a cooling load exists) is about 60°F. Base 65°F cooling degree days (CDD) are tabulated in Chapter 3 of NAVFAC P-88 (NAVFAC 1978; aka TM 5-785) for NASWG and nearby locations in Pennsylvania, New Jersey and Delaware. Long-term historical temperature distributions are tabulated for Newark IAP, McGuire AFB, Wilmington AP, and Wilkes Barre-Scranton, but not for NASWG, Philadelphia, or Camden, in Chapter 5 of NAVFAC P-88 (TM 5-785). The data indicates annual base-60 and base-65 cooling load potentials as shown in Table 3.

From these data it is clear that Wilmington's climate is more (in terms of annual cooling load) like that of Willow Grove than the climate of any other locations except, possibly, distant McGuire AFB. The Wilmington climate was deemed sufficiently similar to be used without further adjustment.

Normal-year cooling loads and energy use were extrapolated from the monitoring period by applying a cooling degree-day factor of 1.96. The units are expected to consume 370 MBtu of gas and provide 225 MBtu of cooling in a normal cooling season. The estimated normal year heating energy use is about 190 MBtu of fuel.

The distribution of cooling load intensity was assumed to be the same for the normal year as it was during the monitoring period. The relative share of light load operations may have been somewhat underestimated as a result. Monthly peak 15-minute demand was assumed to be 100% of connected load for all months with at least one hour at, or over, 80°F and to decrease linearly with peak hourly temperature to 60°F.

Energy Costs

Annual operating cost for the electric powered units consists of electric energy, demand, and ratchet penalty costs. The cost of compressor motor maintenance is assumed to be negligible and the other maintenance costs (filter, damper, controls, refrigerant loop, and supply fan maintenance) are ignored since they will be the same for the gas-powered and electric powered units. The PECO rate schedule that applies to NASWG charges $0.0299 per kWh and $23.70 per monthly peak demand kW. The load peak for the electric powered units is assumed to coincide with the peak at the PECO meter that measures all electrical energy supplied to NASWG.

Table 3. Available Climate Data

Degree-Day Base--> / Description-->	65°F CDD(a)	65°F CDD(b)	60°F CDD(b)	65°F HDD(a)	60°F HDD(c)
Willow Grove NAS	946	NA	NA	5368	NA
Philadelphia IAP	1104	NA	NA	4865	3749
Camden	1104	NA	NA	4865	NA
Newark IAP	1024	1168	1835	5034	3905
McGuire AFB	983	1076	1778	5139	NA
Wilmington AP	992	1154	1822	4940	3818
Wilkes Barre-Scranton	608	798	1336	6277	4994

(a) Based on mean of daily max/min temperatures, Chs. 1&5 of NAVFAC P-89 (TM 5-785)
(b) Based on one- or three-hour temperature readings, Ch.3 of NAVFAC P-89 (TM-5785)
(c) Balcomb et al. 1980 (NAVFAC P-89) Ch.1 base-65 HDD numbers match Balcomb's

The annual operating cost for the gas powered units is considered to consist of natural gas purchases plus annual engine service of $400.00/yr. The PECO rate schedule (Philadelphia Electric Company 1991) that applies to NASWG charges $6.50/kcf for the first 200 kcf purchased each month and $5.60 per additional kcf.

The normal weather year refrigeration machine energy requirements are shown, for the electric and natural gas powered alternatives, in Figure 14. The electric and gas numbers are in almost constant proportion because the electric motor and gas engine efficiencies are almost independent of shaft speed and load. The corresponding monthly energy costs are shown in Figure 15. The electric cost is almost double the gas cost in July and the cost difference is even larger in the other months. The distribution of monthly energy costs for the electric powered alternative are shown in Figure 16. The figure shows that demand charges account for more than half, ratchet charges account for over 30%, while energy charges account for less than 10% of the total annual energy cost.

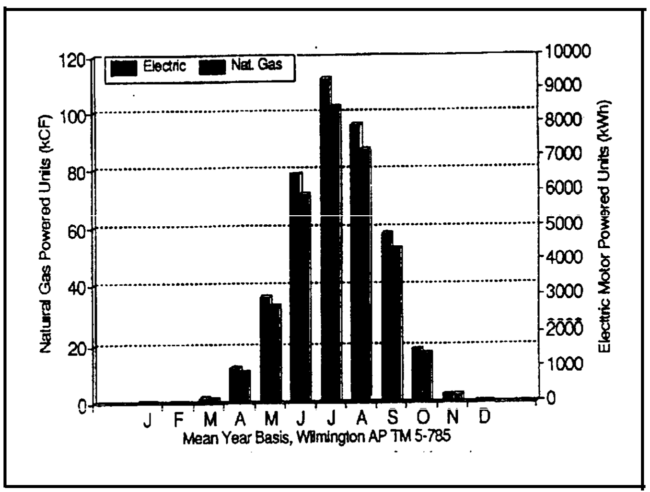

Figure 14. Energy Requirements for Electric and Natural Gas Alternatives

Life-cycle Cost

The life-cycle cost analysis was performed using the FEMP analysis program BLCC (NIST 1986; NIST 1987a; NIST 1987b; NIST 1992). The financial parameters for the cost study are as follows:

Analysis Type: Federal Analysis-Energy Conservation Projects
Study Period: 15 Years (1993 through 2007)
Discount Rate: 4.0% Real (exclusive of general inflation)

The energy related costs for electric (base case) and gas powered alternatives are shown in Table 4.

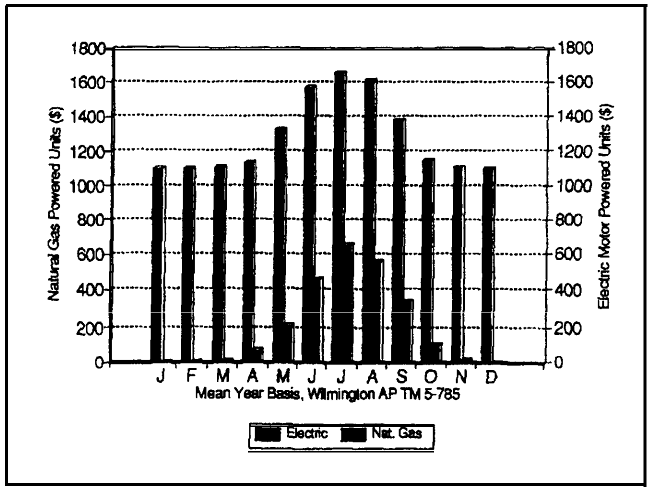

Figure 15. Monthly Energy Costs for Electric and Natural Gas Alternatives

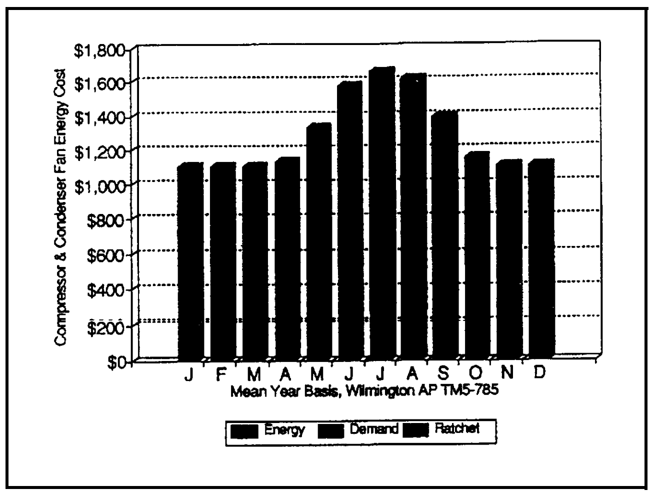

Figure 16. Distribution of Monthly Energy Costs for the Electric Alternative

Table 4. Energy-Related Costs

Energy Type	Use Units/Yr	Price $/Unit	Demand Cost	Total PV Cost
Electric	34.6MWh	0.030	$14,379	$182,696
NatGas	3,875	0.634	$0	$29,114

The present value costs for the two alternatives are shown in Table 5.

The net savings for the gas powered alternative is the difference between the present value of non-investment savings, $148,023, and the increased total investment of

Table 5. Comparison of Present-Value Costs

	Base Case: Electric ($)	Alternative: Nat Gas ($)	Savings ($)
Initial Cost	51,330	77,972	<26,642>
Subtotal	51,330	77,972	<26,642>
Future Costs			
engine O&M	0	5,559	<5,559>
energy	182,696	29,114	153,582
Subtotal	182,696	34,674	148,023
PV L-C Cost	234,026	112,646	121,381

$26,642. This gives a net savings, expressed as a present value, of $121,381 for the gas powered alternative. The net savings-to-investment ratio is 5.56 assuming the previous electric units had to be replaced anyway at the time the units were installed. In cases where the existing electrical equipment is relatively new, the net savings to investment ratio would be much less, but it is still positive (0.88[3]) even for the hypothetical case of replacing brand new electric equipment, given PECO's rate structure and the load distributions observed at the NASWG BX.

Conclusions

Application, Operation, and Maintenance

The new and unique nature of the equipment required extra care and support from experienced local service people. However the added cost, in terms of lost performance, was relatively minor. There were only two shutdowns that were directly related to the equipment. These were attributable to removal and replacement of the engine starter on one unit and a unit shutoff due to refrigerant pressure. Service for this installation will probably not be a problem in future years because the local service people are now familiar with the new technology. The growing market penetration that is currently underway nationally should ensure the availability of qualified local service in the future.

There appear to be opportunities for improving the seasonal performance of the equipment by modifying the staging control sequence. There may be further opportunities for improving seasonal performance and reducing aggregate annual engine starts in multi-unit applications by coordinated and integrated control of the units. A multi-zone configuration with the two units feeding a single air distribution system would be appropriate at the Willow Grove BX.

In the analysis of the retrofit economics for other sites, it is important to determine-by simulation or by monitoring existing equipment with simple run-hour clocks[4] and logs—the distribution of part-load hours for a typical cooling season. This is imperative where the gas powered alternative is expected to be only marginally attractive or where there is reason to believe that the existing equipment is significantly oversized or undersized. A feasibility study can then be completed using a standard analysis technique such as the ASHRAE Bin Method.

Cost Effectiveness

The annual energy costs for cooling, based on the loads and performance measured during the monitoring period and normalized to a normal weather year, are $2,450 to operate the compressors and condenser fans in the two units. The normal-year energy costs to operate the compressors and condenser fans in comparable electric powered units would be $1040 in energy charges and $14,380 in demand and ratchet charges. Maintenance of the units is expected to cost about $400/yr more than the maintenance of comparable electric powered units.

The units will use about the same amount of source energy[5] for cooling as comparable electric powered units. The operation of electric-motor- and gas-engine-driven units is very similar and the level of comfort provided by the two types of units is indistinguishable.

The net savings for the gas-engine-driven unit is the difference between the present value of savings, $148,023, and the increased total investment of $26,642. This gives a net LCC savings, expressed as a present value, of $121,381. The net savings to investment ratio is 5.56 assuming the previous electric units had to be replaced anyway at the time of the retrofit.

In cases where the existing electrical equipment is not worn out, the net savings to investment ratio would be less, but it is still positive (0.88) even for the hypothetical case of replacing brand new electric equipment, given the rate structure and load distributions at the NASWG Base Exchange.

A gas-engine-driven unit is generally cost effective in areas with high (e.g., > 0.15 $/kWh based on marginal annual cost to provide cooling) electric rates, moderate (e.g., < 7.00 $/kcf) gas rates, and annual cooling loads that result in present value electric energy/operating costs that are at least twice the comparable electric unit's replacement cost minus any utility rebate. Annual operating cost is primarily a function of annual full load equivalent operating hours (cooling) and gas and electric rate structures.

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Endnotes

1. Two values of the first stage nominal capacity are possible depending on whether one or both sections of the evaporator are selected. With one section (25% of face area) selected, the nominal capacity is 28,000 Btuh; with both, it is about 40,000 Btuh.

2. The quoted COP was based on use of 1/4 of the evaporator coil by means of a control valve that can be used to stop flow through the other 3/4 of the coil in C 1. This valve was disabled (as is standard in all units manufactured after 4/92) in unit 1. Hence, the full coil was used, resulting in higher capacity at about the same shaft power (due to higher refrigerant flow but lower suction pressure) and higher COP.

3. $\frac{121,381-51,330}{77,972} = 0.8842$ where $121,381 is the net present value for the replace-on-failure case, $121,281-$51,330 is the net present value for the replace-immediately case (assuming no recovery of the $51,330 recently invested in brand new electric powered air conditioning equipment), and $77,972 is the initial implementation cost for gas powered equipment.

4. A run-hour clock connected to each cooling stage control line will give a good indication of part load distribution. The readings should be logged at least weekly. A daily log of the readings, together with daily outdoor maximum and minimum temperature, will facilitate extrapolation of the part-load distribution to a normal weather year.

5. Source energy is defined here as the energy input to the power plant in the case of electricity, or the energy input to the regional gas transmission system in the case of natural gas, to provide a given amount of energy at the customer's meter. By this definition, source energy includes transmission and distribution losses and generation losses, but does not include energy input required for fuel extraction, processing, and transportation to get the fuel to the region.

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