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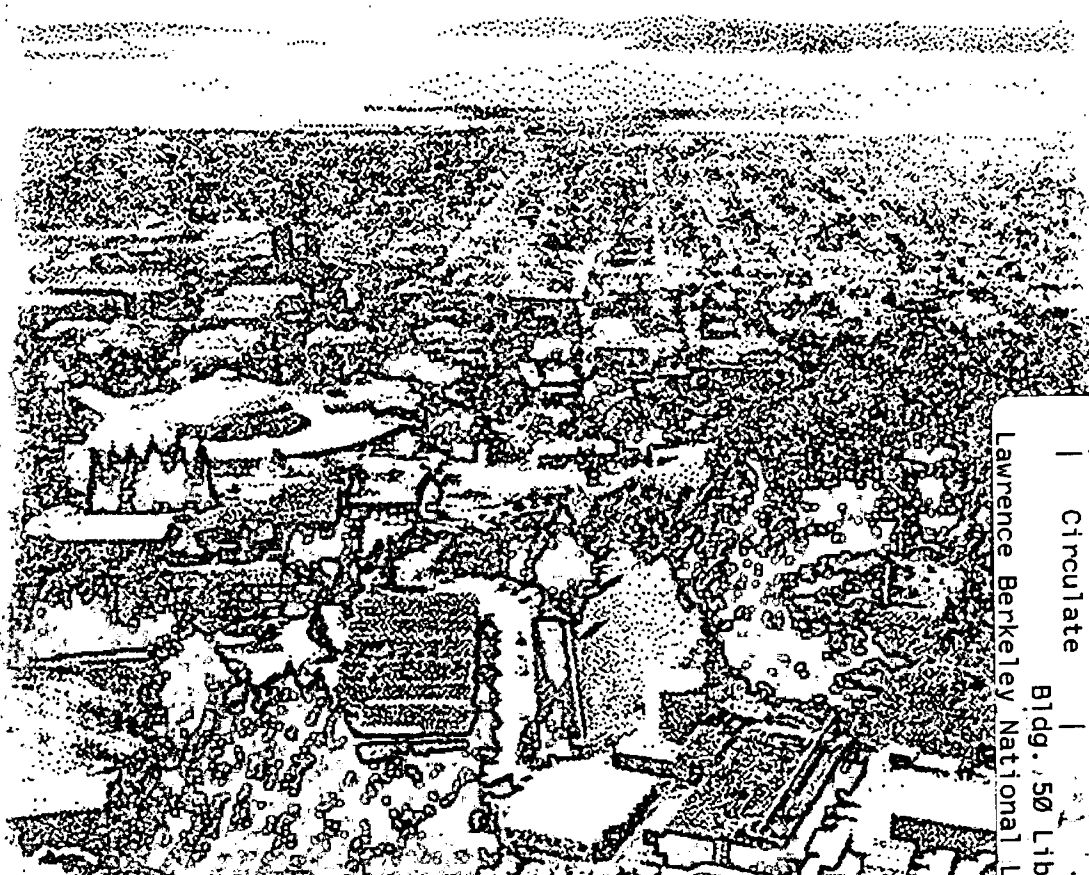

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RESIDENTIAL EQUIPMENT PART LOAD CURVES FOR USE IN DOE-2

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Residential Equipment Part Load Curves for Use in DOE-2

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Overview

DOE-2 (DOE2 90) includes several correlation curves that predict the energy use of systems under part load conditions. DOE-2 simulates systems on an hour-by-hour basis, so the correlations are intended to predict part load energy use (and efficiency) as a function of the part load ratio (PLR) for each hour, where

$$PLR = \frac{HourlyLoad}{AvailableCapacity} \qquad (1)$$

Generally residential and small commercial HVAC equipment meets the load at off-design conditions by cycling on and off. Therefore, the part load correlations must predict the degradation due to this on and off operation over an hourly interval.

Default DOE-2 Curves

The DOE-2 default curves (DOE2 93) predict the normalized change in the Energy Input Ratio (EIR), which is the inverse of COP, as a function of PLR. It is often more convenient to express part load effects in terms of degradation of efficiency under part load. NIST (PARK 77) typically referred to the normalized efficiency degradation as the part load factor, or PLF

$$PLF = \frac{PartLoadEfficiency}{SteadyStateEfficiency} \qquad (2)$$

The DOE-2 curves for EIR were rearranged to find PLF

$$PLF = \frac{PLR}{EIR(PLR)} \qquad (3)$$

Figure 1 shows the DOE-2 default curves for part load performance of various types of equipment. Table 1 lists the SYSTEMS and PLANT equipment that use these curves. Due to the form of the equation, all five basic curves show following behavior:

- the part load efficiency goes to zero at as PLR approaches zero,
- the slope of the curves is a strongly a function of loading (i.e., PLR).

The residential cooling curve shows the most part load degradation, followed by the boiler and heat pump heating curves. The furnace and PSZ cooling curves show the least amount of part load degradation.

Table 1: Default Part Load Curves in DOE-2

Description	Curve Name	Curve No	DOE-2 Systems
Residential Cooling	COOL-EIR-FPLR	16,17,20	RESYS,PTAC,HP
Commercial Cooling	COOL-EIR-FPLR	18 128	PSZ,PMZS,PVAVS PVVT
HP Heating	HEAT-EIR-FPLR	61,62,65, 75,116	RESYS, PSZ, PTAC, PVAVS, HP, WTR-CC, PVVT
Furnace	FURNACE-HIR-FPLR	111	any fuel-fired furnace
Boiler	BOILER-HIR-FPLR	BLRHIR2	HP (WLHP system) HW and Steam Boiler Plants

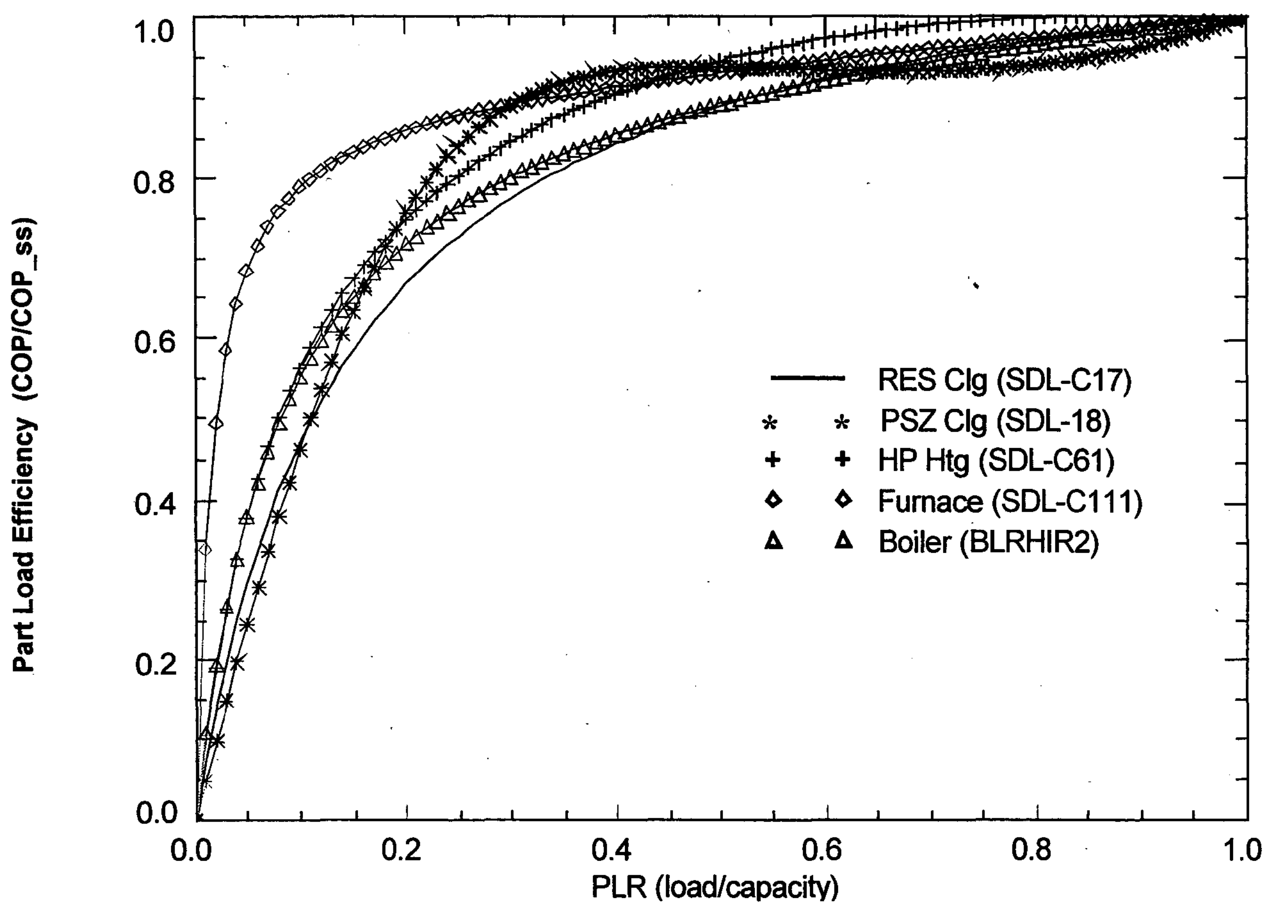

Figure 1: Default Curves from DOE-2

Part Load Degradation for Cycling Equipment

The part load performance of cycling HVAC systems depends on:

1. the response of the system at startup (usually defined by a time constant or dead time),
2. the cycling rate of the equipment (usually defined by thermostat characteristics and building thermal mass).

Parken and his co-workers at NIST (PARK 85) were the first recognize that these two factors could be combined to form a part load correlation. They used this concept to develop the degradation coefficient (C_d) used in the SEER rating procedure to predict part load effects. The concept was verified with both laboratory and field data.

Henderson and Rengarajan (HEND 96) summarize the equations necessary to calculate the theoretical part load efficiency curve. Figure 2 shows that theoretical function closely matches by the linear C_d method proposed by Parken and used in the SEER rating procedure (DOE 79). The default value of C_d in the SEER rating procedure is 0.25. This value corresponds to time constant (tau) of 76 seconds for the air conditioner at startup, and a maximum cycling rate (N_{max}) of 3.125 cycles per hour.

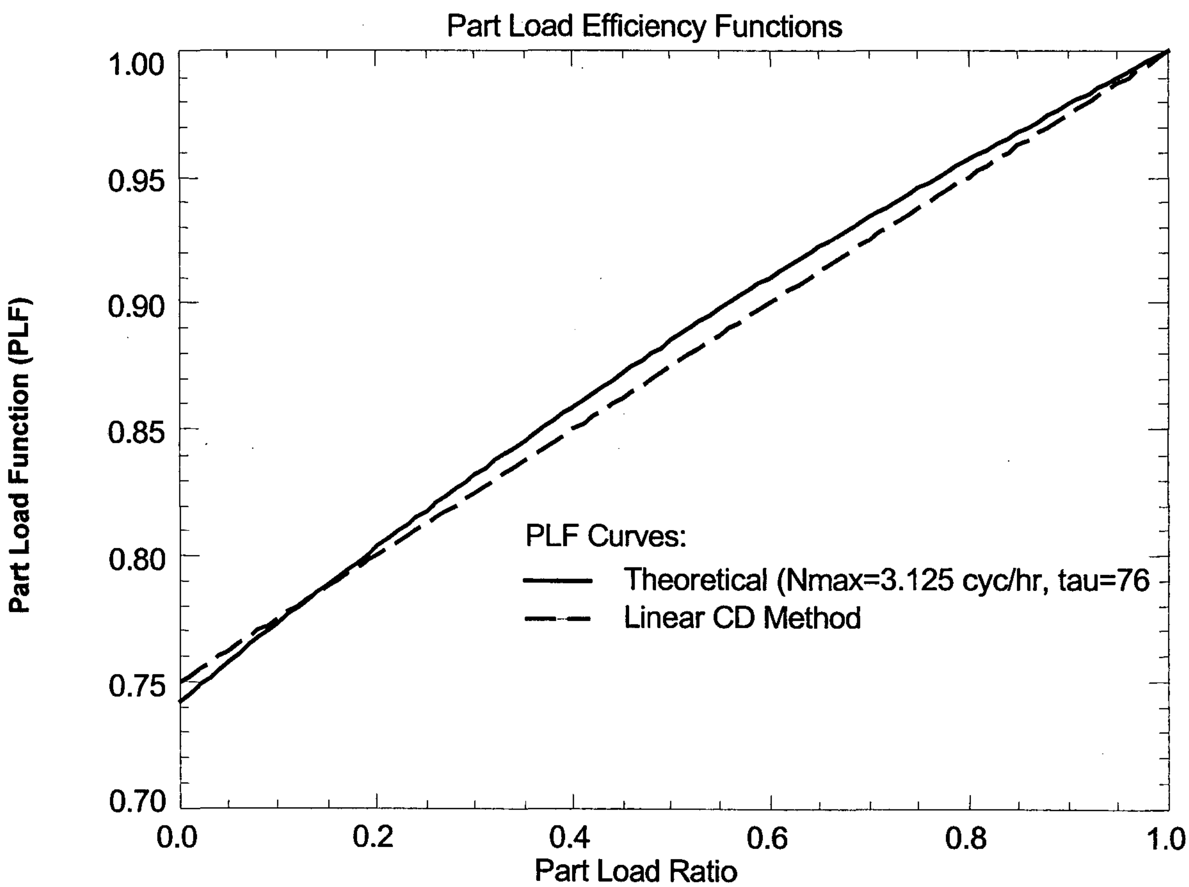

Figure 2: Theoretical and Linear Part Load Functions Based on Parken et al (PARK 1997)

Modern air conditioner and heat pump systems typically have time constant of 40 to 60 seconds. As a result, values of C_d measured for typical systems tested according to the SEER procedure (DOE 79, ASH 96) are in the range of 0.10 to 0.2. Thermostat cycling rates are also generally lower than the default value of 3.125 assumed in the rating procedure. An average cycling rate of 30 Florida homes was found to be 2.5 cycles per hour (HEND 1991). This further reduces the effective value of C_d

Converting the C_d Method into DOE-2/EIR Approach

The linear C_d method assumes that PLF is given by

$$PLF = 1 - C_d(1 - PLR) \quad . \qquad (4)$$

By equating equations (3) and (4), we can see that the function for EIR must be of the form

$$EIR = \frac{PLR}{1 - C_d(1 - PLR)} \quad . \qquad (5)$$

In DOE-2, EIR must be a polynomial. If we fit equation (5) to a cubic polynomial we get a very good fit down to very low values of PLR (as low as 0.05). Figure 3 compares the polynominal fit of EIR to the linear part load function with C_d. Table 2 lists the coefficients corresponding to each curve. These user-defined polynominal coefficients can be used to closely mimic the C_d method.

Table 2: Coefficients for EIR to Match The Linear C_d Model

Degradation Coefficient (C_d)	Coefficients for EIR-FPLR = a + b·PLR + c·PLR² + d·PLR³			
	a	b	c	d
0.05	2.73404e-006	1.05259	-0.0552087	0.00262236
0.10	1.48147e-005	1.11079	-0.121905	0.0111199
0.15	6.25583e-005	1.17517	-0.201513	0.0263344
0.20	0.000164298	1.24656	-0.296070	0.0494917
0.25	0.000362523	1.32573	-0.407603	0.0818194
0.30	0.000698967	1.41373	-0.538881	0.125027

Should the Part Load Efficiency (PLF) Approach Zero?

The linear C_d method, as well as its theoretical form, both predict that part load efficiency *does not* approach zero at as PLR approaches zero. The part load efficiency is always greater than zero because the equipment is assumed to have a minimum run time that is a function of the maximum cycle rate[1].

[1] Henderson et al (1991), Miller and Jaster (1985) and others showed that the minimum on time can be calculated from the maximum cycle rate (N_{max}) by the following equation:

$$t_{ON,\min} = \frac{60}{4N_{\max}}$$

So if N_{max} equals 3.125 cycles/hr, then the minimum run time is 4.8 minutes.

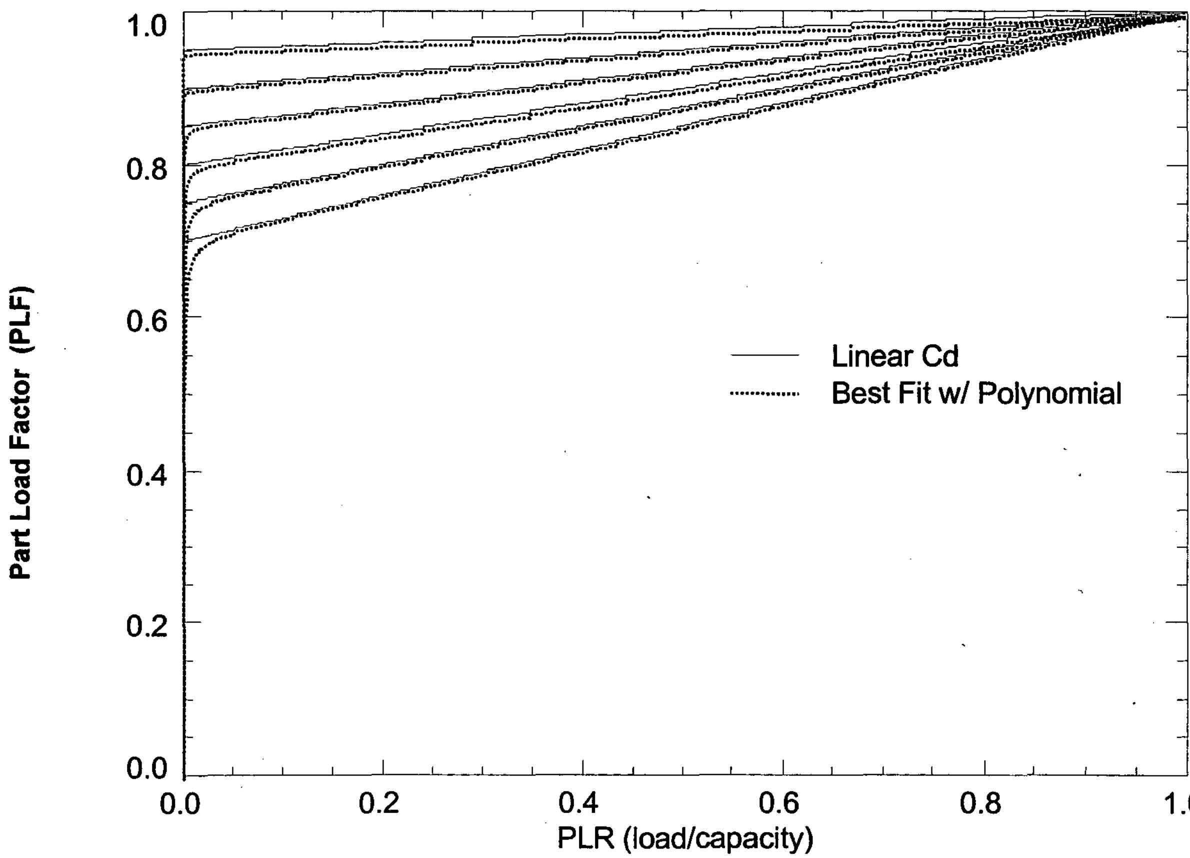

Figure 3: Polynomial fit of EIR Function to Linear C_d Method

A part load method was developed (BONN 80) by assuming that system startup is described by an equivalent delay time (Z_D) instead of a single time constant. Under these assumptions, the part load efficiency still did not approach zero at zero load.

It was shown (BONN 80, MILL 85) that, when off-cycle power consumption is considered, the part load efficiency curve does approach zero. The off-cycle power can be due to crankcase heaters, controls, fans or other factors. If the off-cycle power use is expressed as a fraction of on-cycle power use (pr), then Bonne showed that the adjusted PLF can be calculated as shown below:

$$PLF' = \frac{PLR}{\left(\frac{PLR}{PLF} + (1 - \frac{PLR}{PLF})pr\right)} \quad . \qquad (6)$$

Figure 4 shows that the consideration of even small amounts of off-cycle power has a dramatic impact on efficiency at low load conditions. A value of 0.01 for pr corresponds to about 40 Watts of off-cycle power for a typical 3 ton AC system, while 0.03 corresponds to 120 Watts.

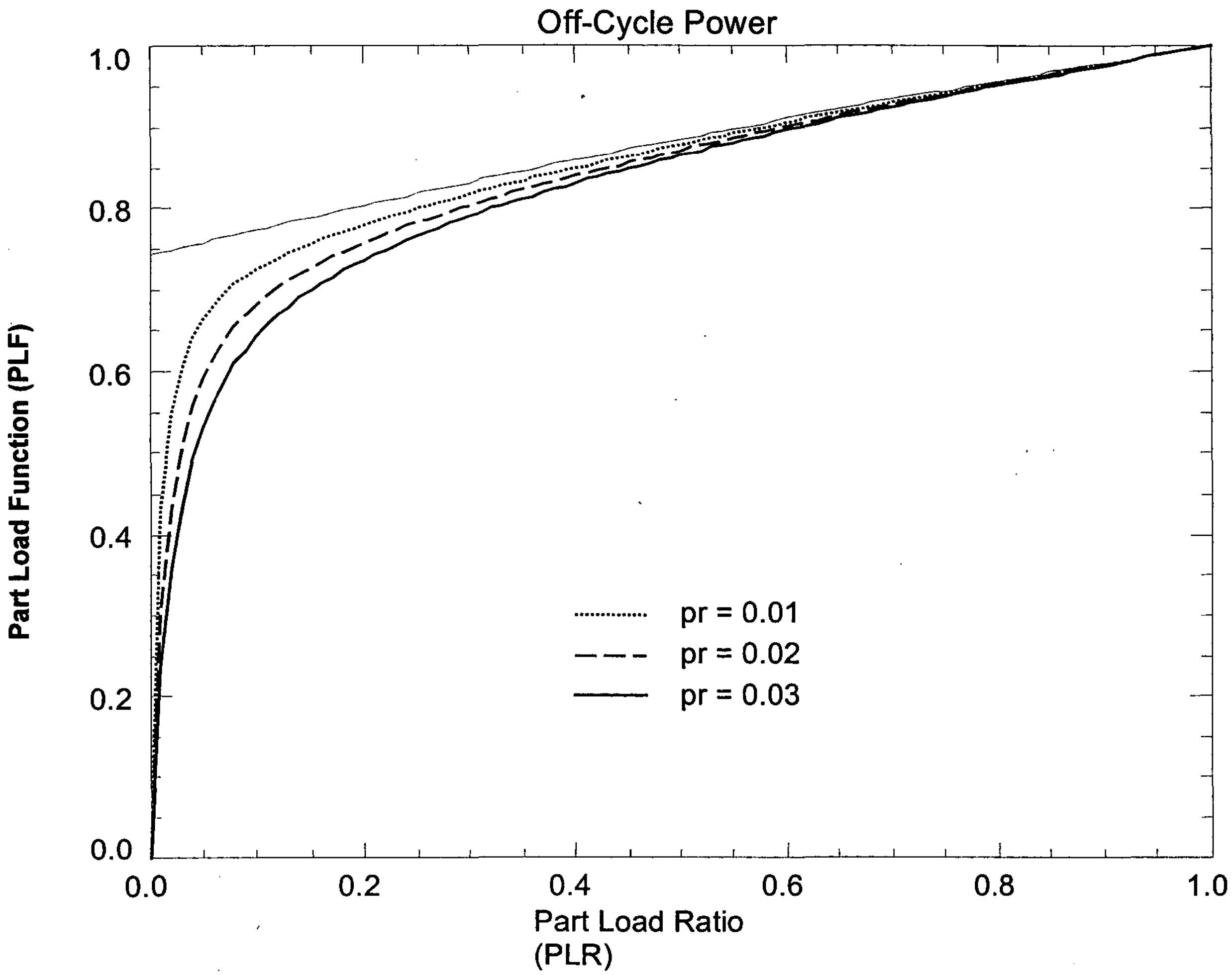

Figure 4: The Impact of Off-Cycle Power on Part Load Efficiency

Recommended Curves - Residential AC/HP Cooling

Figure 5 shows the range of part load performance that might be expected for a residential cooling system. Table 3 lists the corresponding EIR coefficients. The default RESYS curve is also shown on the plot for reference.

Table 3: Coefficients for EIR to For "Typical AC"

	Coefficients for EIR-FPLR = a + b·PLR + c·PLR^2 + d·PLR^3			
	a	**b**	**c**	**d**
"Typical AC" N_{max}=2.5, tau=60, pr=0.01	0.0101858	1.18131	-0.246748	0.0555745
"Good AC" N_{max}=2.5, tau=60, pr=0.01	0.00988125	1.08033	-0.105267	0.0151403
"Poor AC" N_{max}=3, tau=60, pr=0.03	0.0300924	1.20211	-0.311465	0.0798283

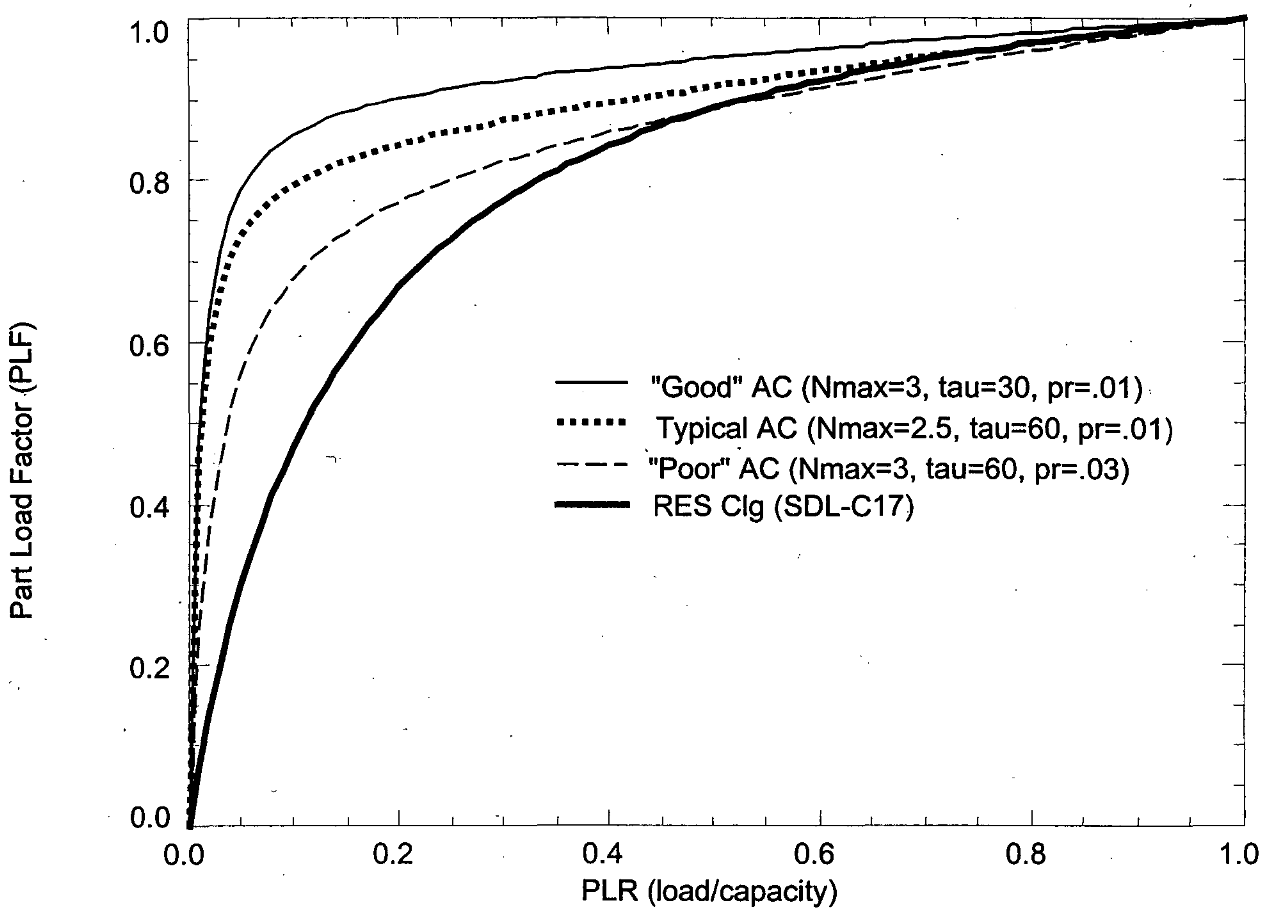

Figure 5: Recommended Parameters for a Typical AC System Compared to Default DOE-2 Curves

The "Typical AC" is assumed to have a time constant of 60 seconds at startup, which is typical of values reported in the literature and summarized by Henderson (HEND 92). These values ranged from 30 to 80 seconds. The "Good AC" might be representative of a system with a liquid line solenoid or other means of off-cycle refrigerant control and is assumed to have a shorter time constant of 30 seconds.

The maximum thermostat cycling rate is assumed to be 2.5 cycles per hour for the "Typical AC" – the average measured at 30 Florida homes (HEND 91). By comparison measured values of 1.5 to 3 cycles per hour and 3 was recommended as the "worst case" (MILL 85). Parken (PARK 85) measured values of 1.6, 2.0, and 2.3 cycles/hr in the cooling mode at three test homes. Therefore, the "Poor AC" is expected to have a cycling rate of 3 cycle per hour.

The off-cycle power use is expected to be 1% (0.01), or about 40 Watts with a 3 ton unit for the "Typical AC." This is close to the assumed value of 1.5% (BONN 80). The "Poor AC" is assumed to be 3% (0.03), or 120 Watts for a 3 ton unit.

There is not expected to be much difference between new and existing systems. Most research into part load issues was conducted in the 1970's and 1980's, though work at FSEC and other institutions seems to confirm these earlier findings. While the steady state performance of residential AC and HP systems has improved substantially over the last 10 to 15 years, there is little evidence that part load issues have changed for cycling equipment. The transient response at startup is still expected to be similar, with the exception of systems with liquid line solenoid valves or totally closeable electronic expansion valves which are expected to respond faster since refrigerant is trapped in the condenser. Also, thermostat manufacturers still design for maximum cycling rates of 2 to 3 cycles per hour.

Recommended Curves - Residential HP Heating

The part load, cyclic performance described above for cooling should be fundamentally the same for a residential heat pump in heating[2]. (MILL 85) and (BONN 80) both showed that the transient response is similar for heating and cooling. (PARK 77) showed almost identical part load curves for heating and cooling. Field data (MILL 85) also showed similar thermostat cycling rates for heating and cooling.

Therefore, we recommend that the cooling curves in Table 2 and Figure 5 also be used for an air-source heat pump in the heating mode.

Recommended Curves – Residential Furnaces and Boilers

The part load performance of residential furnaces and boilers is primarily driven by flue and stack losses. These losses vary with loading because the heat escapes from the furnace flue when the burner is off. Infiltration losses from the house are also caused by the stack. Work at NBS (CHI 78, KELL 78) formed the basis for the AFUE rating procedures developed by The U.S. Department of Energy (DOE 90). U. Bonne at Honeywell performed part of the testing and analysis. The AFUE procedure is also described in ASHRAE Standard 103-1993.

The work associated with the AFUE test and rating procedure does not explicitly give a part load curve. Their initial work showed that they could "skip" that step and directly calculate a seasonal average value. It appears that they did not identify any transient combustion effects -- all part load issues are related to infiltration and stack losses during the off-cycle.

The only part load curve we found in the literature (BONN 85) was a paper to assess the efficacy of cycling controls that were proposed as a means of efficiency improvement in the mid-1980s. A detailed model, HFLAME, was used to assess impact of increased cycle rates on efficiency. Bonne's work is interesting because it shows part load efficiency of furnaces/boilers can increase slightly at part load conditions. This effect appears to be due to the relatively oversized HX at part load (much as in a variable-speed compressor system). This is possible because there are apparently no transient losses in combustion process.

Figure 6 shows the part load curves (BONN 85) for both atmospheric combustion systems (which are no longer widely sold) as well as for induced draft (i.e., power burner) systems. These curves assume a standing plot. If an intermittent ignition system is used then the part load curves would not approach zero (for the same reasons discussed above for AC systems. Table 4 lists the HIR coefficients that fit the part load curves.

[2] However, the need for defrost, which depends on the ambient conditions, is a much more complex phenomenon and is not addressed in this report.

Table 4: Coefficients for HIR for "Furnace/Boiler"

Furnace/BoilerType	Coefficients for HIR-FPLR = a + b·PLR + c·PLR² + d·PLR³ a	b	c	d
Atmospheric	0.011771251	0.98061775	0.11783017	-0.11032275
N_{max}=6, standing pilot, no flue damper				
Induced Draft	0.0080472574	0.87564457	0.29249943	-0.17624156
N_{max}=6, standing pilot, 50% flow at off-cycle				

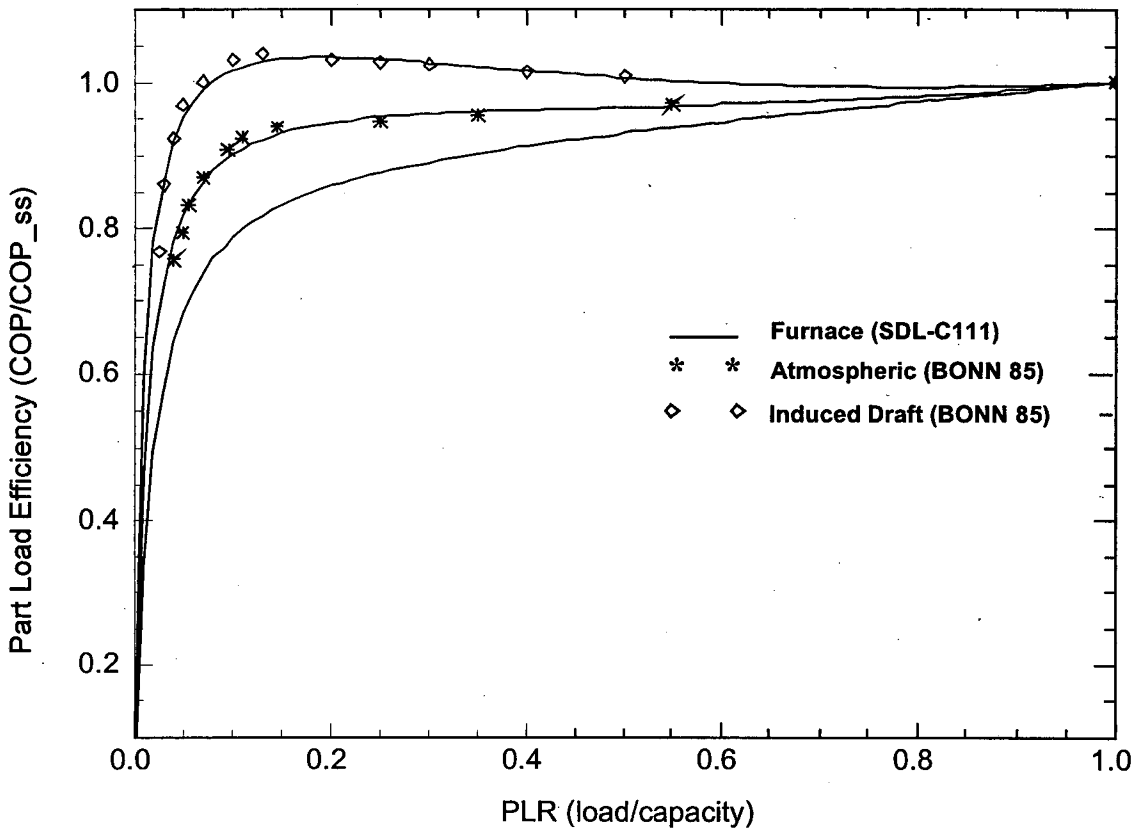

Figure 6: Recommended Curves for Furnaces and Boilers

All this implies that sealed combustion systems that are isolated from the building should have little to no part load degradation, since they eliminate the stack and flue losses. There may even be a slight increase in efficiency as part load, as shown in Figure 6.

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