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November 14, 2023

Ms. Abigail Daken
U.S. Environmental Protection Agency
1200 Pennsylvania Ave. NW
Washington, DC 20460

Re: ENERGY STAR Residential Boilers Discussion Guide June 2023

Dear Ms. Daken,
Burnham Holdings Engineering Company (BHEC) is providing comments in response to the Discussion Guide referenced above. BHEC is a subsidiary of Burnham Holdings Inc (BHI), which owns several manufacturers of boilers (including electric boilers), as well as radiation and other components used in hot water heating ("hydronic") systems.

During the public meeting held on June 21, we objected to the use of the term "heat pump boiler" to describe heat pumps used in hydronic applications. We have noted similar objections in recent DOE consumer boiler rulemakings. Our objection to the use of this term stems from the fact that the heat pumps in question cannot, and in the foreseeable future likely will not, be able to operate under the design conditions employed in most US residential hydronic systems. Most residential hot water boilers are sold for use in replacement installations that were originally designed to operate with water temperatures of 180°F (or higher) at an outdoor temperature below 10°F. To the best of our knowledge, current residential electric air-water heat pump technology available in the US is incapable of generating water temperatures in excess of 140°F and even then, only at much higher outdoor temperatures. Calling such a product a "boiler" will create marketplace confusion which is bad for consumers and bad for the HVAC industry. For the purposes of these comments, we will use our preferred term for heat pumps used in this application, which is "hydronic heat pump" (HHP).

The primary purpose of this letter is to describe a proposed rating procedure for HHPs. It is based on the premise that the seasonal COP of a HHP should be based as closely as possible on the same conditions used to rate fossil fuel boilers, so that consumers are presented with comparable efficiency metrics for all types of products used in the same application.

BHI is primarily a manufacturer of gas and oil-fired boilers. We do not claim to be experts in the testing and rating of heat pumps (our need to educate ourselves on this topic is the primary reason these comments are being submitted after the official comment due date). We recognize that HHPs are entering the US residential hydronic heating market and that there is a need to ensure such products are consistently rated in a manner that provides the public with a means of comparing HHPs to alternative products. BHI therefore supports the use of a standard test procedure that fairly and accurately represents HHP seasonal performance.

The following basic principles are applied in developing this proposal:

1) The scope of this procedure is limited to air-water electric HHPs used in residential hydronic heating systems. We have deliberately stopped short of specifying an exact capacity range. Since this is meant to cover HHPs used in residential applications, however, it would make sense to limit the scope to HHPs capable of operation on single phase power. In addition, the 65,000 BTU/hr limit specified by UL 60335-2-40 could also be applied to the scope.
2) To the extent possible, an existing testing and rating procedure should be used. Obviously, this will minimize both the time to adoption and the burden on HHP manufacturers. We believe that EN 14825 provides an excellent foundation for a US HHP test procedure. Our proposal therefore attempts to use it with as few modifications as possible.
3) Since HHPs are incapable of meeting a typical US heating load under design conditions, a source of back-up heat is assumed. Given (1) it is logical to assume that the source of back-up heat will be electric resistance (EN 14825 does provide methods for predicting performance with fossil fuel back-up and these could be adopted in the future).
4) EPA asked about measurement of HHP performance used to generate both heating and domestic hot water (DHW). The industry has been wrestling with this problem on combination fossil fuel boilers for a very long time (consider the history of ASHRAE 124) with limited progress to date. Attempting the same with “combination” HHPs will likely only bog down the development of any HHP rating procedure. It is also worth noting that, as comprehensive as it is, even EN 14825 does not attempt this. We have therefore restricted our proposal to the measurement of space heat performance.
5) We propose Annual Coefficient of Performance (ACOP) should be the primary rating and the only metric used by incentive programs. The term ACOP is used to differentiate this metric from SCOP generated by EN 14825. This proposal is not intended to preclude the publication of other ratings under alternate conditions. HHP performance is highly dependent on design water and outdoor air temperatures and system designers need alternative ratings to optimize HHP performance on new or redesigned hydronic systems. Any such ratings, however, should be published *in addition to* the ACOP metric we propose.

Identification of Appropriate Rating Conditions

Any attempt to develop a seasonal rating procedure for HHPs comparable to that for conventional boilers must start by identifying the following:

1) The standard curve for supply water temperature as a function of outdoor air temperature that is assumed by ASHRAE 103.
2) A set of bins that are representative of the “national average” climate assumed by ASHRAE 103.

The AFUE generated by ASHRAE 103 is based on the following conditions (nomenclature in parentheses taken from ASHRAE 103 where applicable):

Outdoor Design Temperature (TOAT): 5°F

Average Outdoor Air Temperature (TOA): 42°F

Indoor temperature (TIA): 70°F

Temperature at which Boiler Stops Operating (TSH): 65°F

Nominal Inlet Water Temperature During Test: 120°F

Nominal Water Temperature Rise During Test: 20°F

Average Annual Heating Degree-Days (HDD): 5200°F-day

While not stated in ASHRAE-103, the above water temperature conditions are based on a design mean water temperature of 190°F (i.e., 190°F water required on a 5°F day) and the following relationship between the hydronic radiation output and mean water temperature[1]:

$$Q = K(Tw - TIA)^{1.5} \tag{1}$$

Where:

Q = Rate of heat transfer from radiation

Tw = Mean water temperature in radiation

TIA = Indoor air temperature

K = Constant for a given hot water distribution system

The following relationship therefore exists between mean water temperature and part load ratio:

$$pl = \frac{Q}{Qd} = \frac{K(Tw - TIA)^{1.5}}{K(Twd - TIA)^{1.5}} = \left[\frac{Tw - TIA}{Twd - TIA}\right]^{1.5} \tag{2}$$

Where:

pl = Part load ratio

Qd = Design heating load

Twd = Design mean water temperature

The above equation can be rearranged to show mean water temperature as a function of part load ratio as follows:

$$Tw = pl^{0.67}(Twd - TIA) + TIA \tag{3}$$

Heating load can be written as a linear function of the difference between outdoor temperature and the "no load" outdoor temperature (i.e., the lowest outdoor temperature at which the heating load is zero). From this, the Part Load Ratio can be expressed as follows:

$$pl = \frac{Q}{Qd} = \frac{C(TSH - To)}{C(TSH - TOAT)} = \frac{(TSH - To)}{(TSH - TOAT)} \tag{4}$$

Where:

C = Constant for a given building

[1] NBSIR 81-2110 (*Recommended Testing and Calculation Procedures for Estimating the Seasonal Performance of Residential Condensing Furnaces and Boilers*) funded by DOE and Published by the National Bureau of Standards in April of 1981. It can be shown that the 130F mean test water temperature specified by ASHRAE 103 corresponds to TOA = 42F.

To is outdoor temperature at pl

TOAT = Design outdoor air temperature

TSH = Outdoor temperature at which heating load is assumed zero

Substituting equation 4 into equation 3 yields mean water temperature as a function of outdoor temperature:

$$Tw = \left[\frac{TSH-To}{TSH-TOAT}\right]^{0.67}(Twd - TIA) + TIA \quad (5)$$

This function is plotted using the inputs identified from ASHRAE 103, along with the value for Twd (190°F) in Figure 1 as "Tw". Note that the mean water temperature at an outdoor temperature of 42°F (i.e., TOA) on this curve is 133°F, which is close to the mean nominal test water temperature prescribed by ASHRAE 103 (130°F). The 3°F difference results of the fact that TIA is 5°F higher than TSH (i.e., the boiler shuts down when the outdoor air temperature is 65°F, even though the indoor temperature is assumed to be 70°F).

In the interest of simplicity, and to better align the new water test conditions with those in EN 14825, we elected to define the conditions in terms of water supply temperature rather than mean water temperature. ASHRAE 103 assumes a 20°F rise at all part load ratios. This is appropriate for a non-modulating boiler that must cycle to match the heating load, and which will obtain a 20°F rise at the design heating load. Adopting this fixed rise for HHPs however, would introduce significant error, since most modulate, and all (to date) are incapable of operation at our assumed design conditions; this erroneously high supply temperature will tend to understate HHP performance. In light of this we elected to scale the 20°F nominal temperature rise by the part load ratio so that the following relationship exists between the mean and supply water temperature:

$$Ts = Tw + \frac{20pl}{2} \quad (6)$$

Finally, combining equations 5 and 6 yields:

$$Ts = \left[\frac{TSH-To}{TSH-TOAT}\right]^{0.67}(Twd - TIA) + TIA + 10\left[\frac{TSH-To}{TSH-TOAT}\right] \quad (7)$$

This is the relationship between outdoor and required supply temperature that is used for this proposal. It is plotted in Figure 1 as "Ts", along with return temperature (Tr) calculated per equation 8.

$$Tr = Tw - \frac{20pl}{2} \quad (8)$$

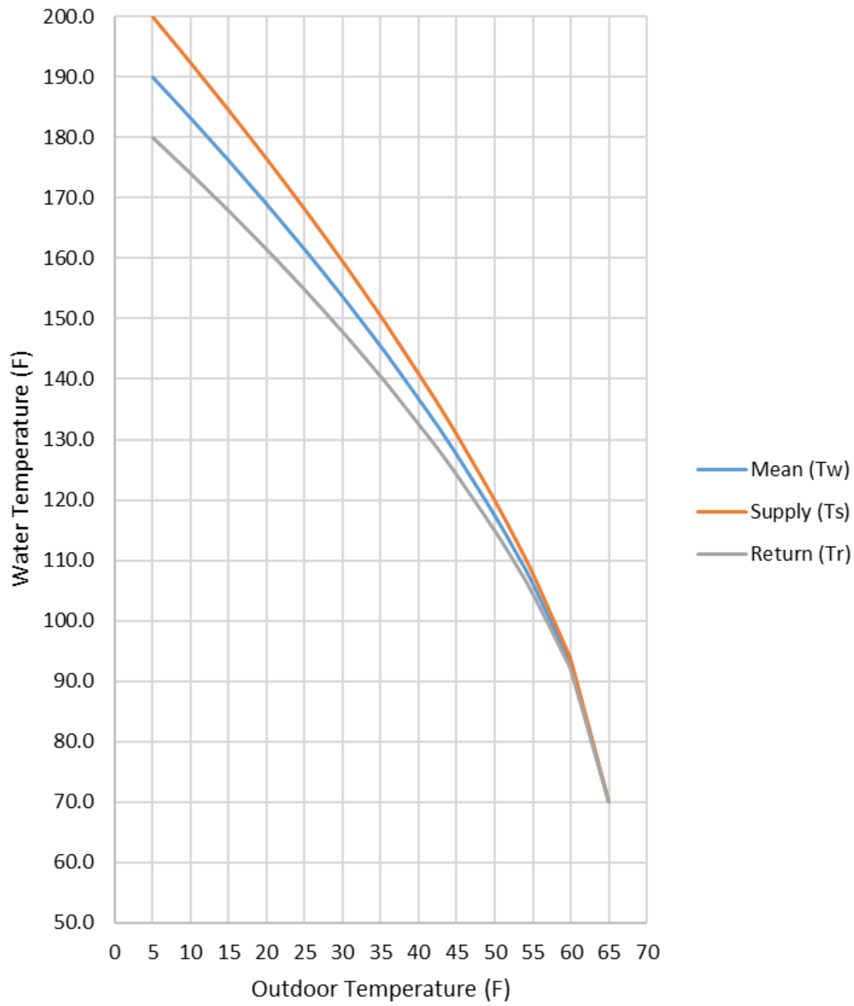

Figure 1: System Temperature Curves Derived from ASHRAE 103

Identification of an appropriate set of bins is less straight forward as ASHRAE 103 does not use a bin technique for AFUE. During very early development of what became ASHRAE 103 a bin technique was tried but it was found that there was not a significant difference in the results thus obtained and simply using the average outdoor air temperature (TOA = 42°F)[2]. There is anecdotal evidence to suggest that the climate assumed in the ASHRAE 103 calculations is based on that in Pittsburgh, PA and the bins for Pittsburgh, sourced from the National Climatology Data Center, are shown in Table 1.

The mean outdoor air temperature in Table 1, weighted by hours, is 41.6°F, which matches the 42°F value for TOA specified by ASHRAE 103. On the other hand, heating degree days (HDD) is approximately 16% higher than the 5200°F-day value assumed by ASHRAE 103. To address this, we elected to scale the hours in each bin by a factor of 0.86. The resulting bins, shown as Table 2, retain the 41.6°F TOA, while closely matching the ASHRAE 103 HDD value. These bins are used in our proposal.

[2] NBSIR 78-1543 (*Recommended Testing and Calculation Procedures for Determining the Seasonal Performance of Residential Furnaces and Boilers*) National Bureau of Standards, September 1978, Page A-9.

Table 1: Pittsburgh PA Climate Bins

Bin #	Outdoor Temperature BIN Bottom °F	Top °F	Mean °F	hr/year	Part Load Ratio (pl)	Degree-Days (°F-day)
1	60	64	62	807	0.05	101
2	55	59	57	729	0.13	243
3	50	54	52	671	0.22	363
4	45	49	47	654	0.30	491
5	40	44	42	643	0.38	616
6	35	39	37	690	0.47	805
7	30	34	32	750	0.55	1031
8	25	29	27	494	0.63	782
9	20	24	22	318	0.72	570
10	15	19	17	209	0.80	418
11	10	14	12	129	0.88	285
12	5	9	7	72	0.97	174
13	0	4	2	35	1.05	92
14	-5	-1	-3	15	1.13	43
15	-10	-6	-8	8	1.22	24
16	-15	-11	-13	3	1.30	10
17	-20	-16	-18	1	1.38	3
18	-25	-21	-23	0	1.47	0
					Total	6051

Table 2: Proposed Climate Bins

Bin #	Outdoor Temperature BIN Bottom °F	Top °F	Mean °F	hr/year	Part Load Ratio (pl)	Degree-Days (°F-day)
1	60	64	62	694	0.05	87
2	55	59	57	627	0.13	209
3	50	54	52	577	0.22	312
4	45	49	47	562	0.30	422
5	40	44	42	553	0.38	530
6	35	39	37	593	0.47	692
7	30	34	32	645	0.55	886
8	25	29	27	425	0.63	672
9	20	24	22	273	0.72	490
10	15	19	17	180	0.80	359
11	10	14	12	111	0.88	245
12	5	9	7	62	0.97	150
13	0	4	2	30	1.05	79
14	-5	-1	-3	13	1.13	37
15	-10	-6	-8	7	1.22	21
16	-15	-11	-13	3	1.30	8
17	-20	-16	-18	1	1.38	3
18	-25	-21	-23	0	1.47	0
					Total	5201

One can certainly debate the merits of the conditions assumed above, but the fact remains that they are close to those that have been used to rate US residential boilers for a very long time. Again, the ability to directly compare boiler and HHP performance would dictate that the heat pump procedure be based upon the same conditions.

Overview of EN 14825 as applied to HHPs

As previously noted, a standard for measurement of heat pump seasonal performance already exists in the form of EN 14825 *Air Conditioners, Liquid Chilling Packages, and Heat Pumps, with Electrically Driven Compressors, for Space Heating and Cooling, Commercial and Process Cooling – Testing and Rating at Part Load conditions and Calculation of Seasonal Performance*. This standard covers the seasonal COP for many other types of heat pumps, including water-water. It also includes coverage for the inclusion of both electric and fossil fuel back-up energy consumption in seasonal performance calculations.

EN 14825 refers extensively to EN 14511 *Air Conditioners, Liquid Chilling Packages, and Heat Pumps for space heating and cooling and process chillers, with Electrically Driven Compressors.* That standard prescribes the method of determining COP under a given set of standard conditions. EN 14825, in turn, defines part load conditions, as well as the means by which the COPs generated using EN 14511 are used to calculate "seasonal COP" or "SCOP".

EN 14825 design conditions for rating HHPs are located in Part 6, with most information summarized in Tables 8-11. Four water temperature curves are specified:

- "Low Temperature" specifies a 35°C (95°F) supply water temperature at design load.
- "Intermediate Temperature" specifies a 45°C (113°F) supply water temperature at design load.
- "Medium Temperature" specifies a 55°C (131°F) supply water temperature at design load.
- "High Temperature" specifies a 65°C (141°F) supply water temperature at design load.

EN 14825 also defines three sets of climatic conditions:

- "Colder" having an outdoor design temperature of -22°C (-7.6°F)
- "Average" having an outdoor design temperature of -10°C (14°F)
- "Warmer" having an outdoor design temperature of 2°C (35.6°F)

Plots of these four water temperature curves as a function of outdoor temperature are shown for the "Average" climate conditions in Figure 2.

Bins for these three climate conditions are shown in Annex B of EN 14825. These are also plotted, along with our proposed bins from Table 2, in Figure 3.

Using EN 14825, it is therefore possible to generate a SCOP for any combination of the above water temperature curves and climates. Doing so requires measuring the output and input at approximately 5 points, depending on the combination of climate and water temperature curves used. Table 3 shows these points for a variable output HHP installed in a "high temperature" application and an "average" climate[3].

[3] Adapted from EN 14825, Table 11

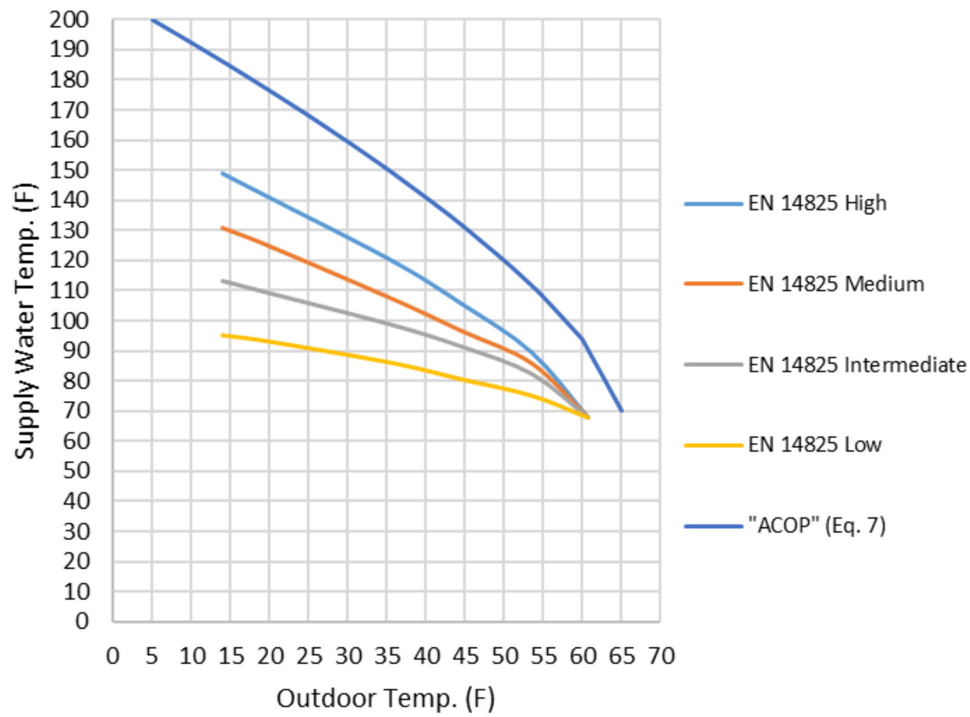

Figure 2: Supply Water Temperature Curves

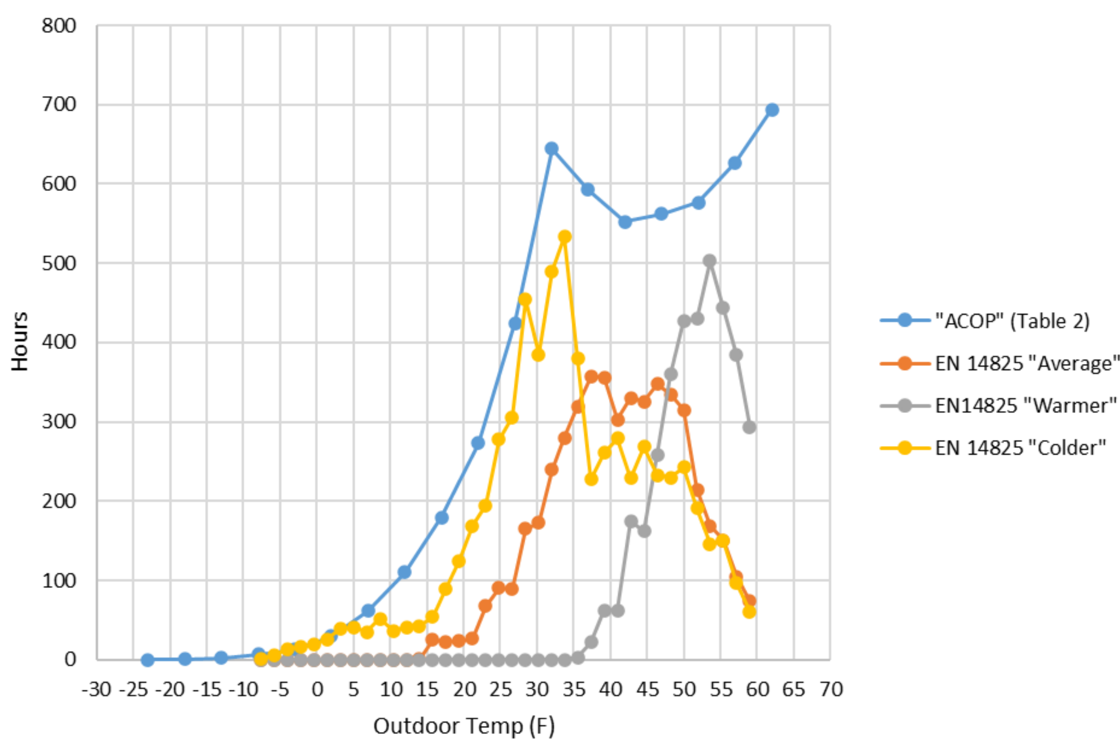

Figure 3: Comparative Climate Bins

Table 3: EN 14825 Variable Outlet Test Points for High Temp Application and Average Climate

Test Point	Outdoor Temp				System Supply Temp		Part Load Ratio
	Dry Bulb		Wet Bulb				
	°C	°F	°C	°F	°C	°F	%
A	-7	19.4	-8	17.6	61	141.8	88.46
B	2	35.6	1	33.8	49	120.2	53.85
C	7	44.6	6	42.8	41	105.8	34.62
D	12	53.6	11	51.8	32	89.6	15.38
E	Test run at higher of outdoor design temperature (-10°C) or lowest outdoor temperature at which HHP will operate (“TOL”). Corresponding test water temperature is taken from curve. If HHP can operate at outdoor design temperature, 65°C supply water is used.						
F	Test run at outdoor temperature of “Tbiv” (“bivalent temperature”), which is the lowest temperature at which the HHP can match 100% of the load without back-up heat. Corresponding water temperature taken from curve.						

The outputs, inputs, and resulting COPs generated at each of the above points consider any cycling due to defrost and/or a load below the HHP’s turndown ratio. The exact method by which this is done is described in Part 11 of the standard and includes a “degradation coefficient” (Cd) to account for energy losses associated with cycling.

In running the above tests, only the performance of the refrigeration cycle is measured, even if the HHP contains an internal back-up source. For points at which the HHP refrigeration cycle is unable to provide an output equal to the load, it is assumed the shortfall is made up by internal or external electric resistance heaters having a COP=1.0. The standard also includes procedures for the measurement of standby, off cycle, and crankcase heater power and the incorporation of these measurements into the SCOP.

The design heating load at a given combination of climate and water temperature application must be declared in order to generate a corresponding SCOP. It appears that there are two ways to do this:

a) Direct declaration by the manufacturer.
b) By declaring the bivalent temperature (Tbiv), which is the lowest outdoor temperature at which the HHP can match 100% of the load. For a given climate there is a direct relationship between outdoor temperature and part load ratio. One can therefore define the design load by dividing the output at Tbiv by the corresponding part load ratio.

Part 6 of EN 14825 also specifies a maximum difference between Tbiv and the outdoor design temperature and therefore implicitly limits the design load a manufacturer can claim based on the performance of the HHP.

Having obtained data at the above points, outputs, COPs, and inputs (including back-up power consumption) are calculated for each bin, using linear interpolation for bins between test points. The on-cycle seasonal coefficient of performance (SCOPon) is then the total energy output (heating load) for all bins divided by the corresponding total seasonal energy input. SCOP differs from SCOPon in that the SCOP also includes annual standby and off mode energy consumption in the denominator.

Proposed Adaptation of EN 14825 for the US HHP Market

None of the bins or the water temperature curves in EN 14825 are comparable to the heating load which forms the basis for ASHRAE 103. That said, EN 14825 is otherwise very close to what is needed, seems to be well thought out, and is already in widespread use. We propose adopting it for the determination of "ACOP" with three significant changes:

1) The Table 2 bins are used in place of the those in EN 14825.
2) The test points shown in Table 4 are used in place of those in EN 14825. These were selected using the temperature curves generated using Equations 7 and 8.
3) Because standby and off mode power consumption are not included in boiler AFUE's, and because it is insignificant relative to the overall amount of energy consumed, we think it reasonable to omit these from ACOP. Crankcase heater consumption, however, is unique to HHPs, is more likely to be significant, and should therefore be included.

Ideally it would be possible to calculate the ACOP from EN 14511/14825 data, without performing additional tests. We have selected points A_{US} - D_{US} based on the fact that they correspond to water temperatures that are already used in these standards and for which there is a possibility that the manufacturer will already have data. For example, Point "A_{US}" is very close to Point "B" in EN 14825, Table 11 (fixed outlet temp). Although some further research might be appropriate to confirm, we suspect that one could obtain accurate data at our proposed points through interpolation of data obtained at the same water temperature, but different outdoor temperatures, provided that there is no cycling at either point used in the interpolation.

We have deliberately stopped short of proposing wet bulb temperatures in Table 4. One approach to this (not necessarily the best one) would be to select wet bulb temperatures that correspond to the same relative humidity as that implicit to the wet bulb temperatures in EN 14825.

Table 4: Proposed ACOP Test Points

Test Point	Outdoor Temp				System Supply Temp		Water Temp Rise		Part Load Ratio
	Dry Bulb		Wet Bulb						
	°C	°F	°C	°F	°C	°F	°C	°F	%
A_{US}	2.1	35.8	TBD		65.0	148.9	5.4	9.7	48.7
B_{US}	7.2	44.9			55.0	131.0	3.7	6.7	33.5
C_{US}	11.6	52.9			45.0	113.1	2.2	4.0	20.2
D_{US}	15.3	59.6			35.0	95.0	1.0	1.8	9.1
E_{US}	Test run at higher of outdoor design temperature (5°F) or lowest outdoor temperature at which HHP will operate while generating water temperatures dictated by Figure 1 curve (i.e., "TOL").								
F_{US}	Test run at outdoor temperature of "Tbiv" ("bivalent temperature"), which is the lowest temperature at which the HHP can match 100% of the load without back-up heat.								

Table 5: Nomenclature

Nom.	Units	Description	EN 14825 Ref.
ACOP	-	Annual COP. Like SCOP, but with proposed conditions/exceptions	Eq. 15
ACOPon	-	Active mode annual COP. Like SCOPon with proposed conditions/exceptions	Eq. 16
Pdesign	kW	Design heating load	3.1.31
Tdesign	°F	Outdoor design temperature = 5°F	3.1.74
j	-	Bin number	7.6
Tj	°F	Bin outdoor air temperature (mean from Table 2)	7.6
hj	hr	Bin hours per year	7.6
pl(Tj)	-	Part load ratio. Note that this proposal uses 65°F (60.8°C) in lieu of 16°C and Tdesign = 5°F.	Eq. 20
Ph(Tj)	kW	Heating load calculated as: $Ph(Tj) = Pdesign\ x\ pl(Tj)$	7.6
Pdh(Tj)	kW	Output produced by HHP at Tj interpolated (or in the case of bin 1 extrapolated) from adjacent test points A_{US} – F_{US}. In following example, the HHP is able to vary its output to match the heating load at points C_{US}, B_{US}, and F_{US}. At D_{US}, the minimum HHP output exceeds the load, and the unit must cycle. At E_{US} the HHP is too small to match the load.	7.6
elbu(Tj)	kW	Portion of the heating load provided by back-up resistance heat. Equal to greater of: • Ph(Tj) - Pdh(Tj) • 0.0	7.6
COPd	-	COP "declared" (i.e., measured max capacity) at test points A_{US} - F_{US}	
CR	-	Capacity ratio equal to lesser of: • Ph(Tj) divided by Pdh(Tj). • 1.0 If CR is less than 1.0, unit is theoretically cycling.	Eq. 21
Cd	-	Degradation coefficient used to correct COPd for the effects of cycling. It can be determined experimentally, or a default may be used.	11.5.3
COPbin(Tj)	-	COP corrected for the effects of cycling. Where Pdh(Tj) is less than 110% of Ph(Tj) it is assumed that no cycling occurs and COPbin is set equal to COPd. Otherwise COPbin is calculated as shown in Equation 23 in EN 14825: $COPbin = COPd\ x\ \frac{CR}{Cd\ x\ CR + (1 - Cd)}$	Eq. 23
Ehout(Tj)	kWh	Annual heating load for bin = Ph(Tj) x hj	N/A (Unsummed Numerator of Eq. 16)
Ehin(Tj)	kWh	Energy consumed for bin calculated per equation: Ehin$(Tj) = hj\ x \left[\frac{Ph(Tj) - elbu(Tj)}{COPbin(Tj)} + elbu(Tj)\right]$	N/A (Unsummed Denominator of Eq. 16)
Tbiv	°F	"Bivalent temperature" – Lowest outdoor temperature at which HHP can generate 100% of required load (i.e., without back-up heat) at corresponding water temperatures in Fig 1.	3.1.13
TOL	°F	Operation limit temperature = outdoor temperature below which HHP output is zero.	3.1.58

Example

The following example details how "ACOP" would be derived using data that is typical for existing products.

- Design heating load (Pdesign): 25 kW
- Crankcase heater power consumption (Pck): 0.06 kW
- HHP has a variable output, but turndown is limited 5 kW
- On-cycle data for this example is shown in Table 6.
- Degradation Coefficient (Cd) due to cycling is 0.9

Table 6: On-Cycle Input Data for Example

Point	Air	Ts	Heating Capy (Pdh)	HHP Input	COPd	Part Load	CR	COPbin
	°F	°F	(kW)	(kW)		(kW)		
A_{US}	35.8	148.9	10.50	4.47	2.35	12.17	1.00	2.35
B_{US}	44.9	131.0	8.33	2.47	3.37	8.38	1.00	3.37
C_{US}	52.9	113.1	5.00	1.18	4.24	5.04	1.00	4.24
D_{US}	59.6	94.1	3.00	0.60	5.00	2.27	0.76	4.84
E_{US} (TOL)	33.0	154.1	10.00	5.00	2.00	13.33	1.00	2.00
F_{US} (T_{biv})	40.0	140.9	10.38	3.83	2.71	10.42	1.00	2.71

The results of Table 6 are interspersed with the Table 2 bins as shown in Table 7.

Seasonal on-cycle COP (ACOPon) is defined as:

$$ACOPon = \frac{\sum_j Ehout(Tj)}{\sum_j Ehin(Tj)} \tag{9}$$

In our example ACOPon = 52009/37119 = 1.40.

As previously noted, the only off cycle power consumption we propose accounting for is that of the crank case heater (Pck). In our example this is measured as 0.06kW. From EN 14825, Table B.3, annual on time for the crankcase heater (Hck) is assumed to be 3850hr for an "Average" climate (this value merits further examination for its applicability to the US).

ACOP is calculated by modifying Equation 9 as follows:

$$ACOP = \left[\frac{\sum_j Ehout(Tj)}{(Hck\ x\ Pck)+\sum_j Ehin(Tj)}\right] \tag{10}$$

From Equation 10, ACOP in our example is 1.39.

Figure 4 is a plot of heating load, and HHP output, as a function of outdoor temperature using the data in this example. As can be seen:

- Below 40.0°F (Tbiv) the HHP cannot meet the load alone.
- Above 52.9°F (C_{US}) the HHP minimum output exceeds the load, and the unit will cycle
- Between the above points the HHP is able to modulate to match the load.
- Below 33.0°F (TOL) the HHP provides no output and electric resistance is assumed to meet the entire load.

Table 7: Worksheet for Example

j	Tj °F	hj hr	pl(Tj)	Ph(Tj) kW	Pdh(Tj) kW	elbu(Tj) kW	COPbin(Tj)	Ehout(Tj) kWh	Ehin(Tj) kWh
1	62	694	0.050	1.25	2.26	0.00	5.07	867	171
D_{US}	**59.6**		**0.091**	**2.27**	**3.00**		**4.84**		
2	57.0	627	0.133	3.33	3.77	0.00	4.61	2089	453
C_{US}	**52.9**		**0.202**	**5.04**	**5.00**		**4.24**		
3	52.0	577	0.217	5.42	5.37	0.04	4.14	3124	773
4	47.0	562	0.300	7.50	7.46	0.04	3.60	4216	1189
B_{US}	**44.9**		**0.335**	**8.38**	**8.33**		**3.37**		
5	42.0	553	0.383	9.58	9.54	0.04	2.98	5296	1792
F_{US} (Tbiv)	**40.0**		**0.417**	**10.42**	**10.38**		**2.71**		
6	37.0	593	0.467	11.67	10.47	1.20	2.45	6919	3243
A_{US}	**35.8**		**0.487**	**12.17**	**10.50**		**2.35**		
E_{US} (TOL)	**33.0**		**0.533**	**13.33**	**10.00**		**2.00**		
7	32.0	645	0.550	13.75	0.00	13.75	1.00	8864	8864
8	27.0	425	0.633	15.83	0.00	15.83	1.00	6723	6723
9	22.0	273	0.717	17.92	0.00	17.92	1.00	4897	4897
10	17.0	180	0.800	20.00	0.00	20.00	1.00	3593	3593
11	12.0	111	0.883	22.08	0.00	22.08	1.00	2449	2449
12	7.0	62	0.967	24.17	0.00	24.17	1.00	1496	1496
13	2.0	30	1.050	26.25	0.00	26.25	1.00	790	790
14	-3.0	13	1.133	28.33	0.00	28.33	1.00	365	365
15	-8.0	7	1.217	30.42	0.00	30.42	1.00	209	209
16	-13.0	3	1.300	32.50	0.00	32.50	1.00	84	84
17	-18.0	1	1.383	34.58	0.00	34.58	1.00	30	30
18	-23.0	0	1.467	36.67	0.00	36.67	1.00	0	0
							Total	**52009**	**37119**

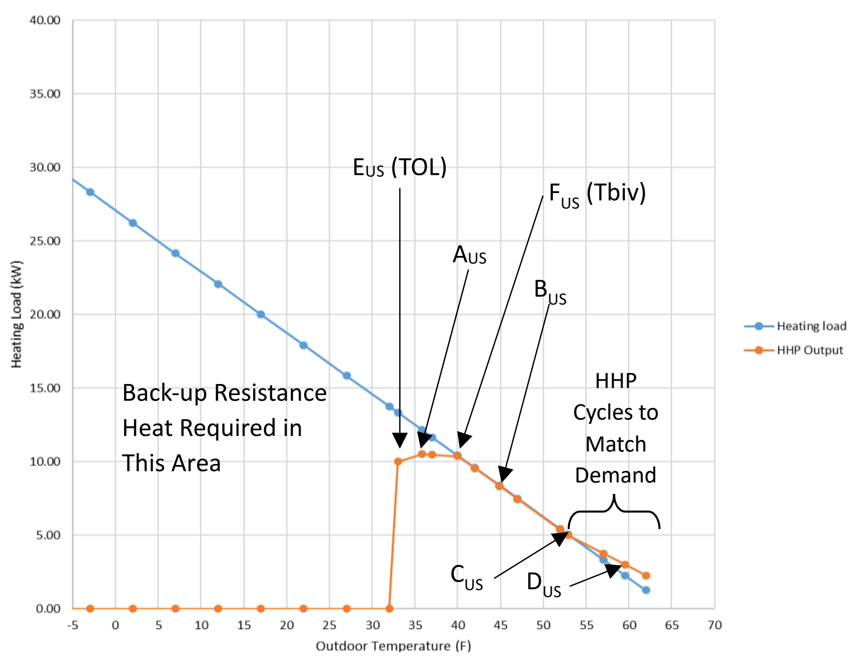

Figure 4: Plot of HHP Output and Heating Load from Example

Conclusion

The fact that the ACOP in this example is significantly lower than what might typically be reported using EN 14825 is a simple reflection of the fact that the US climate and water temperatures will result in far more reliance on back-up resistance heating. As HHP technology improves to allow operation at greater "lifts", however, this procedure will allow those improvements to be measured.

There are, of course, many open issues that would need to be addressed to use this procedure. Some have already been mentioned. Others include, but are certainly not limited to, a review of the bin profile (e.g., is finer granularity required?), management of significant figures, etc.

We appreciate the opportunity to provide these comments and hope they are of use. Please do not hesitate to contact me if you have questions.

Sincerely,

Duane L. Breneman
President
Burnham Holdings Engineering Company