Consolidation of Broad Spectrum HEL Cooling Pump Designs into Discrete Product Families

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Abstract. High energy laser (HEL) systems have long depended on capable thermal management systems (TMS) to accomplish waste heat management and recovery under idle and hot fire operational modes. There are many aspects to successful TMS designs, one notable characteristic being compact three-dimensional configurations that meet restrictive space claims in HEL applications. Along with the restrictive space claim is the need for efficient, compact, and many times, high or low flow, power dense TMS cooling pump designs. Barber-Nichols has been engineering, designing, manufacturing custom turbomachinery since 1966. The company has been engineering, designing, manufacturing, and delivering diverse HEL TMS cooling pumps designs for a broad spectrum of operational requirements since 2008. As HEL platforms and capabilities have matured, BNI has noticed an uncanny consolidation of required pump designs into three product families. This paper briefly examines BNI's universe of HEL cooling pumps, applications within thermal management systems, their performance, and the resulting pump families that are now most commonly provided into high energy laser applications.

Key Words: Thermal Management Cooling Pumps

Nomenclature: Q: Flow P_1: Inlet Pressure T_1: Inlet Temperature P_2: Discharge Pressure
ΔP: (P_2 - P_1) ΔH: Head ρ: Fluid Density N: Speed (RPM) P_{SAT}: Fluid Vapor Pressure
NPSHA: Net Positive Suction Head Available Ns: Specific Speed r: Radius
g_c: Gravitational Constant Nss: Suction Specific Speed U_2: Impeller Tip Speed
ϕ: Flow Coefficient ψ: Head Coefficient v: Velocity ω: Rotational Speed - Radians/sec.
D_2: Impeller Diameter Cm_2: Impeller Exit Velocity Vector A_2: Impeller Exit Area
π: Pi b_2: Exit Blade Height t: Exit Blade Edge Thickness β_2: Exit Blade Angle
Z: Number of Impeller Blades η: Pump Efficiency

1. Introduction

High energy laser (HEL) thermal management systems (TMS) remain challenged in terms of meeting size, weight, and performance (SWaP) goals as HELs transition to military platforms in the field; especially at higher power levels. For many HEL applications, compact, power dense, high performance, low and high flow cooling pump assemblies, capable of being mounted in just about any orientation, remain a key requirement.

Since 2008 a broad spectrum of HEL TMS cooling pump designs have been provided to customers with unique cooling requirements and capabilities. Initially most pump designs provided were unique, custom tailored designs, sometimes with unique metallurgy for proprietary fluids, and have been different from one customer to another as industry has sought out the most effective and capable TMS designs for a broad range of applications, laser configurations, and power levels. Interestingly, there are signs of HEL TMS designs maturing across the customer base as pump designs have been noted to be consolidating into discrete product families. This paper looks at typical pump operating parameters provided by customers, how these parameters are applied initially to new pump designs, and how these operating parameters are used to evaluate cooling pump performance for existing pump designs; including the consolidating designs. Finally, a sampling of the universe of HEL

pump designs and capabilities are reviewed with an examination of which pump designs are consolidating into discrete product families.

2. Pump Parameters that Drive Centrifugal Pump Design

Typical requirements specified for HEL TMS cooling pumps provided by customers are as follows:

- Required Fluid
- Flow, Q: ft3/min., GPM, Liters/min.
- Pump Fluid Inlet Pressure: P_1 (PSIA, BAR A, KPa A)
- Pump Fluid Inlet Temperature: T_1 (°F, °C, K, R)
- Required Pump Boost, ΔP ($P_2 - P_1$): PSID, Bar D, KPa D (D = Differential)
- Motor Power Source (AC or DC?): AC Volts – Phase(s) – Frequency, Volts DC

- Variable or Fixed-Speed Operation?
    - If Variable Speed: Critical Operational Speeds (i.e. idle speed, full speed)
    - If Variable Speed: Required "Slew Rate" (time) between Lowest and Highest Operational Speeds

The first important parameter to determine from customer supplied information using ΔP and fluid density (ρ) (determined from the specified fluid at (P_1 & T_1)), is the enthalpy change (ΔH) or "Head" for the indicated pump operating point. For a new pump design, the head determined may be treated as a design point parameter:

$$\Delta H = \frac{\mathbf{\Delta P\ Lbf/in^2} \times 144\ in^2/ft^2}{\rho\ Lbm/ft^3} = ft{\cdot}Lbf/Lbm \qquad (1)$$

There is an over-arching parameter in turbomachinery known as Specific Speed – Ns. For one versed in the forms of turbomachinery including pumps, compressors, and turbines, one can likely determine important turbomachine configuration, parameters, and performance based on Ns alone:

$$\text{Specific Speed, Ns} = \frac{N(RPM)\sqrt{Q\,(\frac{ft^3}{sec.})}}{\sqrt[4]{\Delta H(\frac{ft{\cdot}Lbf}{Lbm})^3}} \qquad (2)$$

Turbomachinery configuration and performance, including efficiency (η) are closely tied to Ns. This includes pump performance. In the Ns equation, customers typically specify flow (Q) and Head (ΔH) by specifying ΔP. This leaves selecting speed (N) as the only available parameter for the pump designer to optimize Ns. For centrifugal pumps, Ns determines pump configuration:

- Ns of 10 to 45, Partial Emission Pump Configuration
- Ns of 50 to 150, Full Emission Pump Configuration

A mitigation to a selected pump speed can be due to net positive suction head available (NPSHA) which is determined as follows:

$$\text{NPSHA} = \frac{(P_1 - P_{SAT})\ \mathbf{lbf/in^2} \times 144\ in^2/ft^2}{\rho\ Lbm/ft^3} + \text{Fluid Height Above Pump Inlet (feet)} = \text{feet} \qquad (3)$$

P_{SAT} is the fluid "saturation" or "vapor pressure" at the indicated fluid temperature (T_1). NPSHA determines the total "equivalent" fluid column at the pump inlet. This is important to know in order to determine if damaging pump cavitation will be an issue and any mitigation steps that may be required.

$$\text{Suction Specific Speed, Nss} = \frac{N(RPM)\sqrt{Q\ (GPM)}}{\sqrt[4]{NPSHA\ (ft\frac{Lbf}{Lbm})^3}} \quad (4)$$

Nss is an indication of the pump's tendency to cavitate and an indicator of any need for remedial design action, such as the addition of an "inducer" at the pump impeller inlet to "induce" more head at the pump inlet; essentially increasing NPSHA to prevent cavitation in the flow range.

For new pump designs, there are two (2) essential, dimensionless parameters that are ***assumed*** or ***determined based on Ns*** as well as many years of pump history and experience. These are:

Head Coefficient: ψ Flow Coefficient: φ **These always exist in matched pairs.**

Head and Flow Coefficient pairs (ψ_n,φ_n) are essential in translating new pump operating conditions to existing pump designs. These allow for a full range of pump performance values to be expressed (Figure 3). This includes changes in pump operating fluids. For all new pump operating conditions provided by customers, the initial approach is to always determine if an existing pump design will meet specified operating requirements before a new or modified pump design is proposed.

Pump Head Coefficient, ψ, is **assumed** for new designs and is determined from existing test data from existing pump designs and past experience. For similar pump configurations with similar Ns, for new designs, ψ is selected or assumed in order to determine:

$$\text{Impeller Tip Speed, } U_2 = \sqrt{g_C \frac{\Delta H\, ft\cdot Lbf/Lbm}{\psi}} = \text{ft./sec.} \quad (5)$$

g_c = 32.174 Lbm·ft/(Lbf·sec.^{2})

$$\text{Impeller Diameter, } D_2 = \frac{2 \text{ x } 60 \text{ sec./min. x } 12 \text{ in./ft. x } \mathbf{U_2\ ft./sec.}}{2\pi \text{ (Radians/Rev.) x } \mathbf{N\ (Rev./min.)}} = \text{inches} \quad (6)$$

This is a derivation based on: v = r x ω, where r = $\frac{D_2}{2}$

Pump Flow Coefficient, φ, is **assumed** for new designs and is determined from existing test data from existing pump designs and past experience. For similar pump configurations with similar Ns, for new designs, φ is selected or assumed in order to determine:

$$Cm_2 = \mathbf{U_2\ ft./sec.} \text{ x } \boldsymbol{\phi} = \text{ft./sec.} \quad (7)$$

Cm_2 is a pump impeller discharge flow ***vector*** normal to the pump impeller annular exit area **A_2**.

$$\text{Impeller Exit Area, } A_2 = \frac{\mathbf{Q, GPM} \text{ x } 144 \text{ in}^2/\text{ft}^2}{60 \text{ sec./min. x } 7.481 \text{ Gal/ft}^3 \text{ x } \mathbf{Cm_2\ ft/sec.}} = \text{in.}^2 \quad (8)$$

This is an adaptation of the "Continuity Equation": A = Q/V

$$\text{Impeller Exit Blade Height, } b_2 = \frac{\mathbf{A_2\ in.}^2}{\pi \text{ x } \mathbf{D_2\ in.} - \text{t in./Sin}(\beta_2) \text{ x Z}} = \text{inches} \quad (9)$$

t = Discharge Bade Thickness, Z = Number of Impeller Blades, β_2 = Exit Blade Angle

Equation (9) is simply the annular exit area less blockage caused by exit blade geometry and shows the solution for exit blade height **b_2** based on the known net exit area required (**A_2**) from (8).

3. Determination of Centrifugal Pump Head and Flow Coefficient Pairs

Knowing exit blade height (**b_2**) is very important for determining pump off-design performance; the key basis for determining if an existing pump design is suitable for a new or different application with different operating parameters. Exit blade height (**b_2**) is a key parameter iterated on with an existing pump's flow and head coefficient pairs, (ψ_n,φ_n); which are determined the first time the pump is performance tested. The iteration is performed by varying speed and flow coefficient with matching head coefficient until the calculated blade height is matched to the pump's physical blade height; providing a convergence to a particular pair of Head and Flow Coefficients which provide the specified ΔP (Head) and specified flow (GPM, ft^3/sec. etc.) at the correct blade height **b_2**.

ΔP (Head) drives pump speed for an existing design. Flow, via. blade height (b_2), drives the determination of the appropriate flow and head coefficient pairs (ψ_n,φ_n), and ultimately *where* the application lands on an exiting pump's performance curve.

An existing pump performance curve including data measured at discrete operating points determining the performance curve shape and range are shown in **Figure 1** (Head vs. Flow):

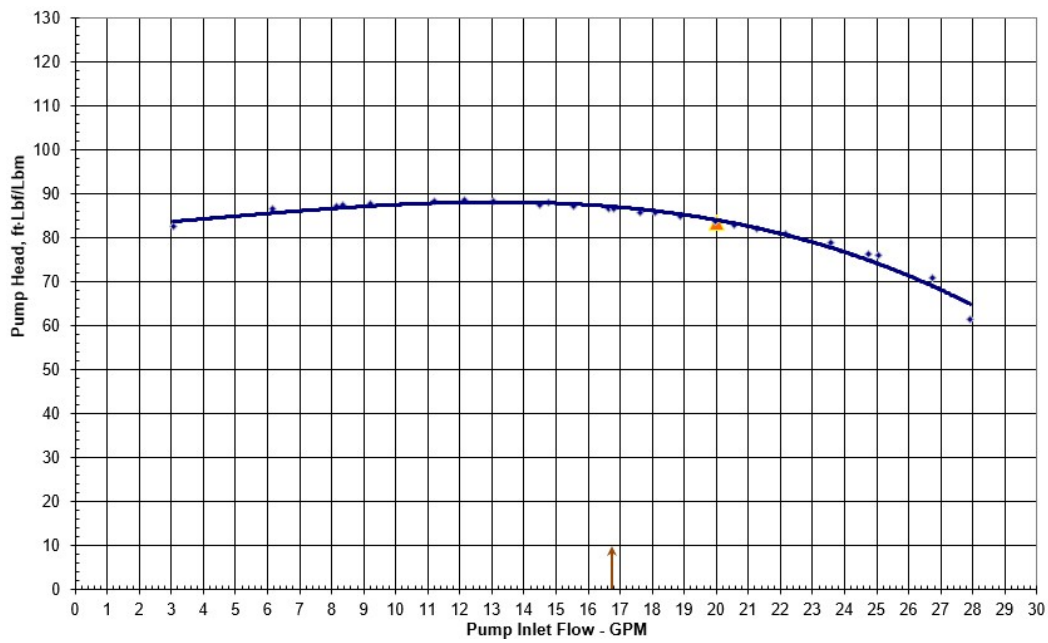

Figure 1. Typical Pump Performance Curves with Test Data Points (Flow, Head)

Since all of the pump geometry for an existing or newly assembled pump design are known, pump performance Head and Flow Coefficient pairs (ψ_n,φ_n) are calculated based on test data points for Delta-P vs. flow and then calculated for Head vs. Flow (these being the discrete test data points shown in **Figure 1** for ($Flow_n$, $Head_n$)). The pump design point in **Figure 1** is at 16.79 GPM. An off-design operating point in this instance is at 20 GPM with acceptable performance.

Known pump geometry, fluid properties, and raw test data:

Speed (N) Impeller Diameter: D_2 Exit Blade Height: b_2 Measured Flow: GPM

Fluid Density: ρ Number of Impeller Blades: Z Impeller Blade Thickness: t Measured Delta-P: ΔP
Measured Fluid Temperature: T_1 Measured Fluid Pressure: P_1

Head and Flow Coefficient Pairs are calculated as follows from known impeller geometry and measured ($Flow_n$, ΔP_n) test data pairs where:

$$\Delta H = \frac{\mathbf{\Delta P\ Lbf/in^2} \times 144\ in^2/ft^2}{\rho\ Lbm/ft^3} = ft{\cdot}Lbf/Lbm \qquad (1)$$

$$A_2 = \pi \times \mathbf{D_2\ in.} \times \mathbf{b_2\ in.} - \mathbf{t\ in.}/Sin(\boldsymbol{\beta_2}) \times \mathbf{b_2}\ in. \times \mathbf{Z} = in.^2 \qquad (10)$$

(Exit Annular Area) (Less Flow Blockage from Exit Blade Edge Areas)

b2= Blade Height, t = Discharge Bade Thickness, Z = Number of Blades, β_2 = Exit Blade Angle

$$Cm_2 = \frac{\mathbf{Q,\ GPM} \times 144\ in^2/ft^2}{60\ sec./min. \times 7.481\ Gal/ft^3 \times \mathbf{A_2\ in^2}} = ft./sec. \qquad (11)$$

$$\text{Impeller Tip Speed, } U_2 = \frac{\mathbf{D_2\ in.}}{2} \times \frac{2\pi\ (Radians/Rev.) \times \mathbf{N\ (Rev./min.)}}{60\ sec./min. \times 12\ in./ft.} = ft./sec. \qquad (12)$$

This is a derivation based on: $U_2 = v = r \times \omega$, where $r = \frac{D_2}{2}$

$$\boldsymbol{\psi} = \frac{g_c\ (Lbm{\cdot}ft/(Lbf{\cdot}sec.^2)) \times \mathbf{\Delta H\ ft{\cdot}Lbf/Lbm}}{(\mathbf{U_2\ ft./sec.})^2} \qquad (13)$$

$$\boldsymbol{\varphi} = \frac{\mathbf{Cm_2\ ft./sec.}}{\mathbf{U_2\ ft./sec.}} \qquad (14)$$

All of the test data points ($Flow_n$, ΔP_n) and thus ($Flow_n$, $Head_n$) can be reduced to Flow and Head Coefficient pairs (φ_n,ψ_n).

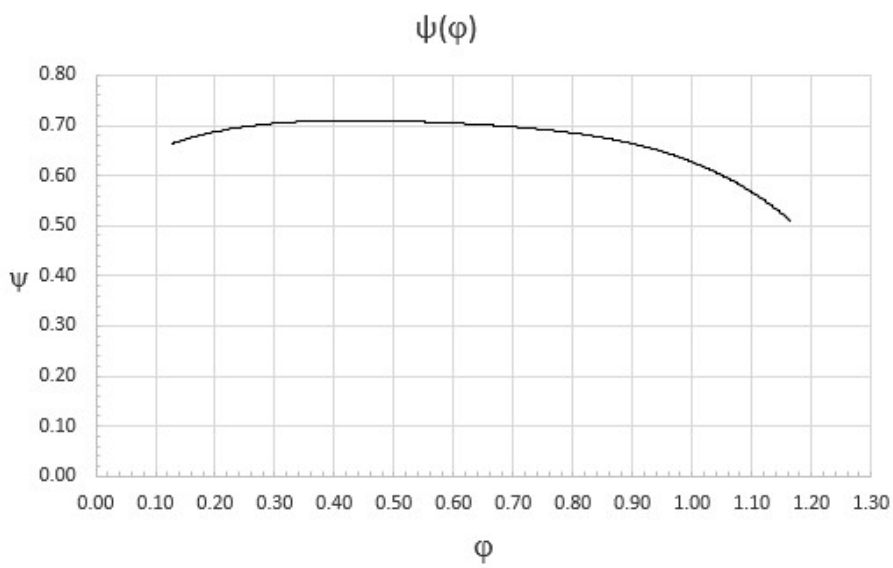

Figure 2. Performance Curve Head Coefficient (ψ) vs. Flow Coefficient (φ) from Test Data

4. Application of Existing Pump Head and Coefficient Pairs to New Operating Conditions

The beauty of dimensionless Flow and Head Coefficient pairs are that they easily translate existing pump performance from one fluid and operating condition to any other fluid and operating condition, thus providing pump performance curve parameters as shown in **Figure 3**:

411000C-6		Calcuated Values for the Performance Curve											
Actual Test Data 11 Nov. 2016		Mass Flow	Head	Mass Flow	Flow	ΔP	ΔP	Flow	Head	Flow	Flow	Head	Pump Efficiency
FC- φ	HC -ψ	Lbm/sec	ft-Lbf/Lbm	grams/sec	ft^3/min.	psid	Bar D	GPM	ft-Lbf/Lbm	m^3/hr.	Lit/min.	meter	η
0.1272	0.6664	0.450	179.7	204.1	0.408	82.6	5.7	3.052	179.7	0.693	11.6	54.8	29.1%
0.2574	0.6978	0.911	188.2	413.1	0.826	86.5	6.0	6.176	188.2	1.403	23.4	57.4	39.9%
0.3399	0.7026	1.203	189.5	545.5	1.090	87.1	6.0	8.155	189.5	1.852	30.9	57.7	45.2%
0.3485	0.7044	1.233	190.0	559.2	1.118	87.3	6.0	8.360	190.0	1.899	31.6	57.9	45.7%
0.3840	0.7074	1.359	190.7	616.2	1.231	87.7	6.0	9.212	190.7	2.092	34.9	58.1	47.7%
0.4669	0.7113	1.652	191.8	749.3	1.497	88.2	6.1	11.202	191.8	2.544	42.4	58.5	51.5%
0.5064	0.7140	1.792	192.6	812.7	1.624	88.5	6.1	12.149	192.6	2.759	46.0	58.7	52.9%
0.5430	0.7113	1.921	191.8	871.5	1.742	88.2	6.1	13.028	191.8	2.959	49.3	58.5	54.1%
0.6048	0.7064	2.140	190.5	970.5	1.939	87.6	6.0	14.508	190.5	3.295	54.9	58.1	55.5%
0.6157	0.7089	2.178	191.2	988.1	1.975	87.9	6.1	14.771	191.2	3.355	55.9	58.3	55.7%
0.6479	0.7042	2.292	189.9	1039.7	2.078	87.3	6.0	15.542	189.9	3.530	58.8	57.9	56.2%
0.6945	0.6996	2.457	188.7	1114.5	2.227	86.7	6.0	16.661	188.7	3.784	63.1	57.5	56.7%
⇨ 0.7000	0.6974	2.477	188.1	1123.4	2.245	86.4	6.0	16.793	188.1	3.814	63.6	57.3	56.7%
0.7342	0.6926	2.598	186.8	1178.2	2.355	85.8	5.9	17.614	186.8	4.001	66.7	56.9	56.8%
0.7539	0.6914	2.667	186.5	1209.9	2.418	85.7	5.9	18.087	186.5	4.108	68.5	56.8	56.8%
0.7863	0.6835	2.782	184.3	1261.8	2.522	84.7	5.8	18.864	184.3	4.284	71.4	56.2	56.7%
0.8316	0.6744	2.942	181.9	1334.6	2.667	83.6	5.8	19.951	181.9	4.531	75.5	55.4	56.3%
0.8576	0.6685	3.034	180.3	1376.2	2.750	82.9	5.7	20.574	180.3	4.673	77.9	54.9	55.9%
0.8862	0.6611	3.135	178.3	1422.1	2.842	81.9	5.6	21.260	178.3	4.829	80.5	54.3	55.4%
0.9232	0.6530	3.266	176.1	1481.6	2.961	80.9	5.6	22.149	176.1	5.031	83.8	53.7	54.6%
0.9817	0.6367	3.473	171.7	1575.4	3.148	78.9	5.4	23.551	171.7	5.349	89.1	52.3	53.0%
1.0319	0.6146	3.651	165.7	1656.1	3.310	76.2	5.3	24.757	165.7	5.623	93.7	50.5	51.2%
1.0442	0.6126	3.694	165.2	1675.7	3.349	75.9	5.2	25.050	165.2	5.690	94.8	50.4	50.8%
1.1144	0.5712	3.943	154.0	1788.4	3.574	70.8	4.9	26.735	154.0	6.072	101.2	47.0	47.7%
1.1639	0.4964	4.118	133.9	1867.8	3.733	61.5	4.2	27.922	133.9	6.342	105.7	40.8	45.2%
Design Point ⇨ 0.7000	0.6974	2.477	188.1	1123.4	2.245	86.4	6.0	16.793	188.1	3.814	63.6	57.3	56.7% ⇦Design Point
Operating Point ⇨ 0.8337	0.6739	2.950	181.7	1338.0	2.674	83.5	5.8	20.000	181.7	4.542	75.7	55.4	56.3% ⇦Operating Point

Figure 3. Application of Flow & Head Coefficients, Existing Pump, for a New Operating Condition

As can be seen in **Figure 3**, the Flow and Head Coefficient pairs provide notable pump performance curve information for an application where the operating fluid, flows, and required ΔP and resulting pump speed are different from design. This method of projecting and estimating pump performance from application to application is useful and accurate. It is by this methodology, that particular groups of pumps have been noted to be consolidating into discrete product families with the same or similar pumps being found suitable on a repeat basis for customer's HEL TMS applications.

5. Broad Spectrum HEL TMS Cooling Pump Configurations

Figure 4 shows a notional HEL TMS system and where cooling pumps have been applied in the past. There are systems with one (1) or two (2) pumps. Others have three (3) pumps as shown:

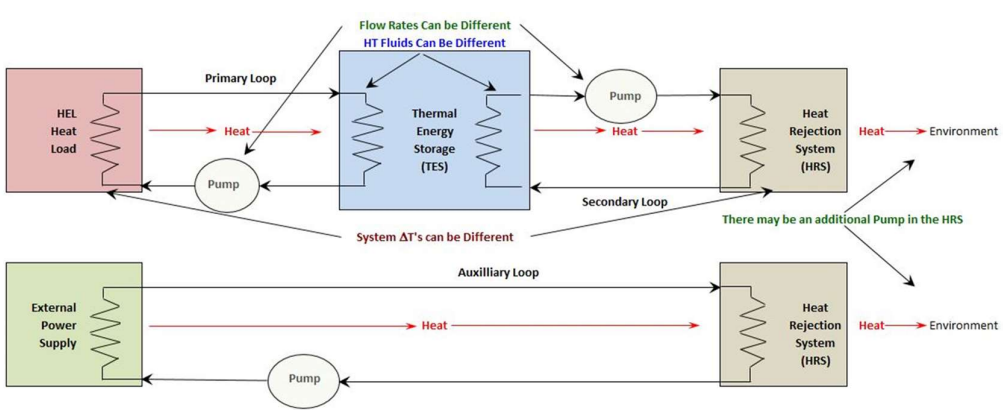

Figure 4. Notional HEL TMS Cooling Pump Applications

Based on eleven (11) years of experience in addressing HEL TMS pump needs and requirements, the following is a representative sub-set of Barber-Nichols' HEL TMS cooling pump product universe:

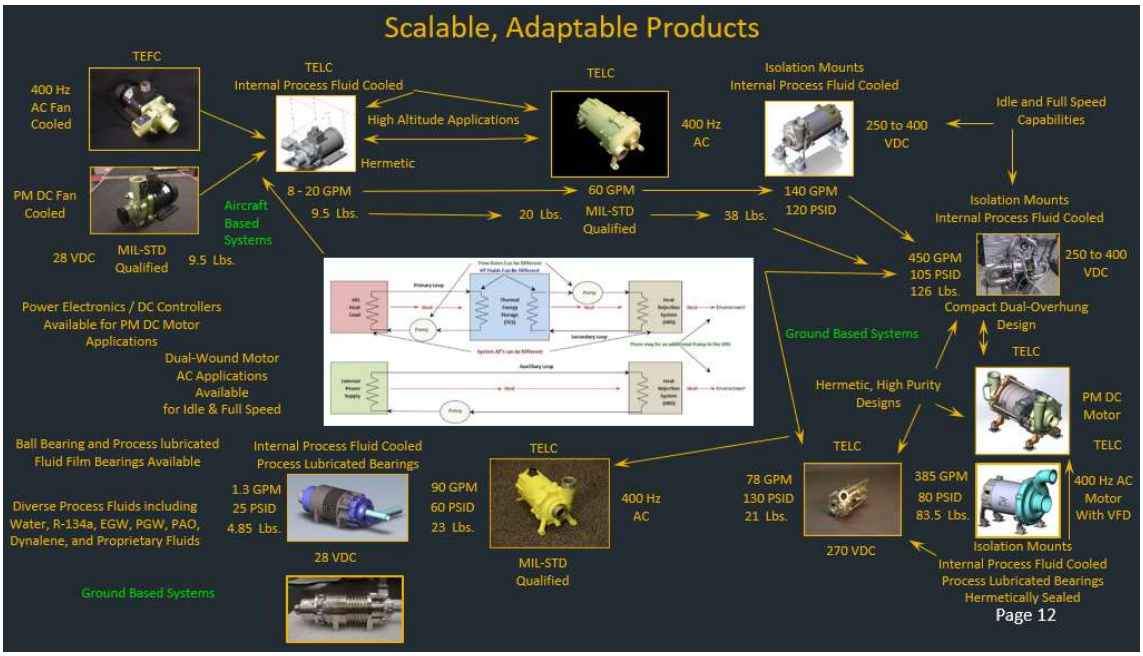

Figure 5. HEL TMS Cooling Pump Sizes and Capabilities – Stationary, Ground, and Air

6. Consolidation Broad Spectrum HEL Cooling Pump Designs into Discrete Product Families

As previously mentioned, as HEL platforms and capabilities have matured, BNI has noticed an uncanny consolidation of required pump designs into three product families as shown in **Figure 6**:

Figure 6. HEL TMS Cooling Pump Sizes and Capabilities – Stationary, Ground, and Air

Performance curves representing the complete pump universe sub-set are shown in **Figures 7** and **8**. The curves are divided into low flow applications and into high flow applications. The shaded areas of the curves in these two figures provide the emerging three (3) discrete product families based on Head and Flow:

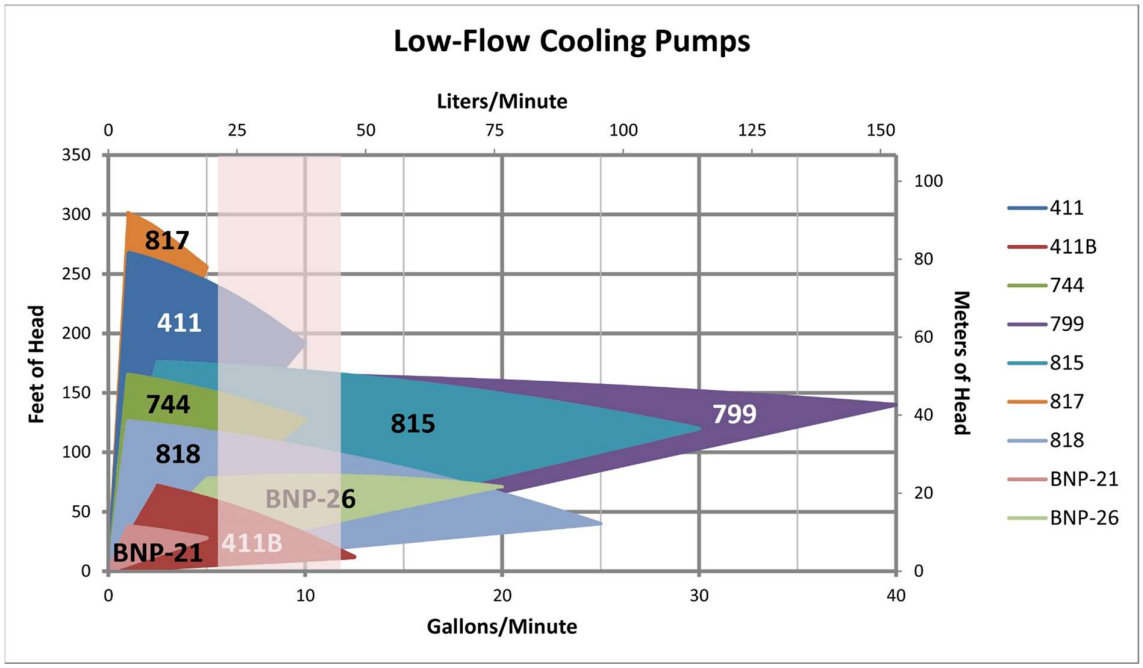

Figure 7. Low Flow HEL TMS Cooling Pumps Showing Area of Pump Family Consolidation

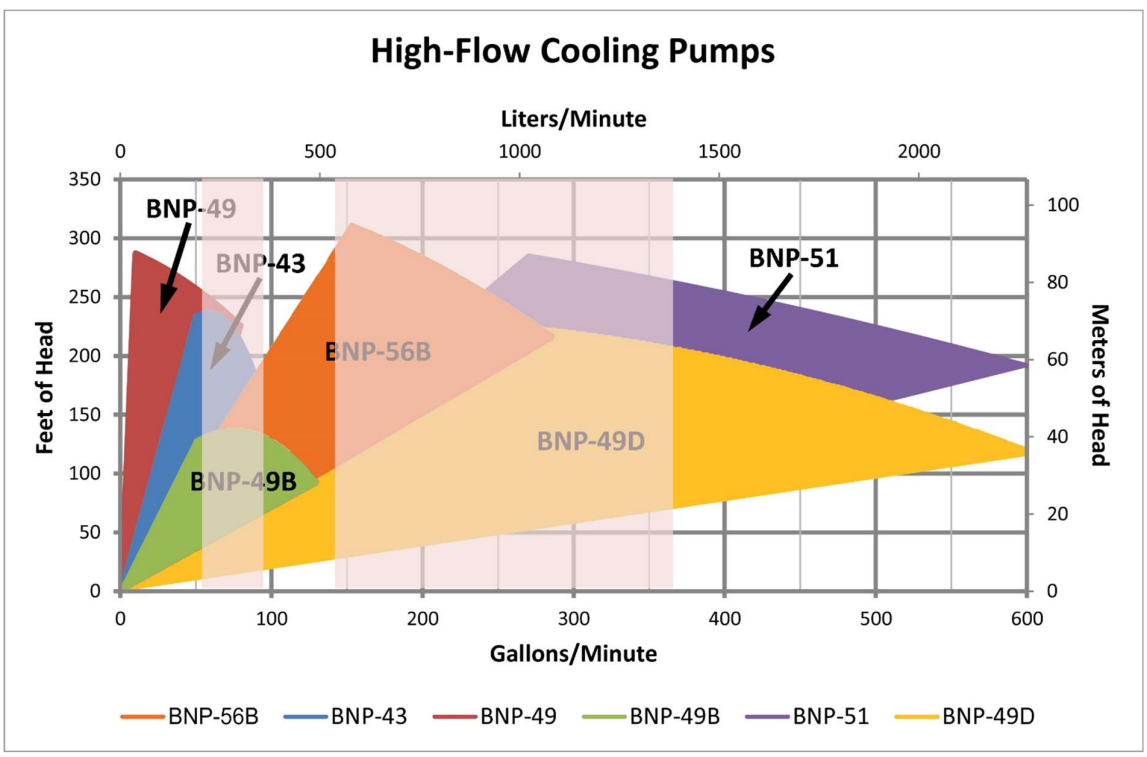

Figure 8. High Flow HEL TMS Cooling Pumps Showing Areas of Pump Family Consolidation

The highest flow consolidation is interesting in that the lower area of consolidation at about 140 GPM is in notable demand while at the same time there is a noticeable expansion of high flow requirements; most likely in tandem with higher HEL power requirements.

7. Typical Pump Fluids Encountered for HEL TMS

In closing, typical pump fluids encountered for HEL TMS systems include:

Fluids:

- Water
  - Plain
  - Deionized (DI)
- Ethylene Glycol - Water Mixtures
- Propylene Glycol – Water Mixtures
- Polyalphaolefin – PAO
- Refrigerants
  - R32
  - R134a
  - R151
  - R245fa
- Proprietary Fluids

References

[] None.