Delivering air "power" in refrigeration

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Table of Contents

Title: Convergent Refrigeration

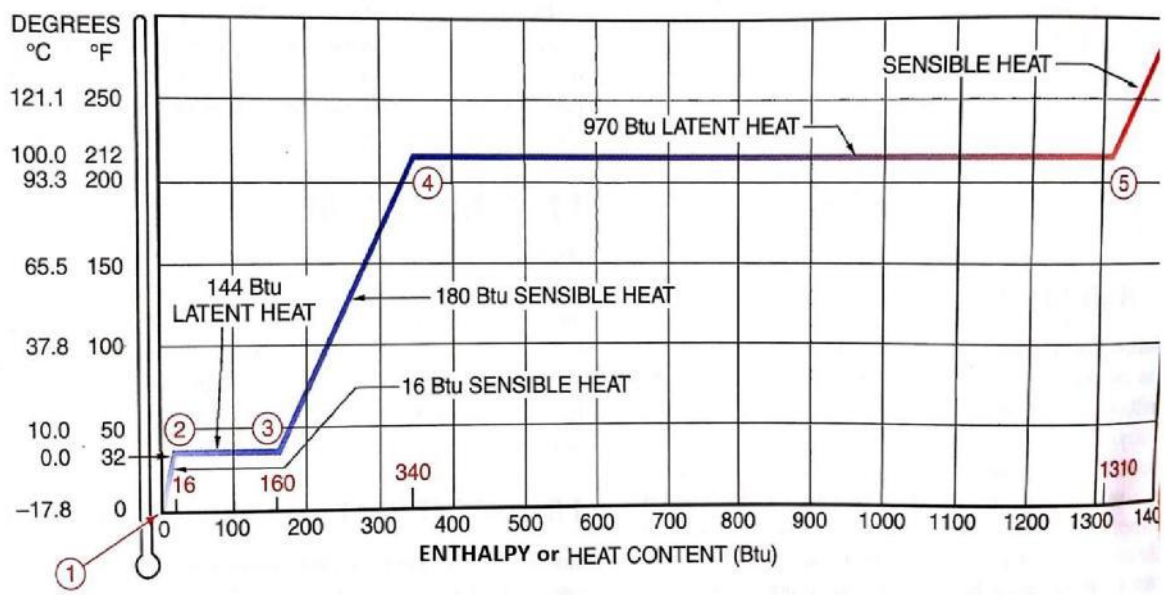
Background of the Invention

How Evaporative Cooling Works

Nature cools with heat. It's paradoxical. Nature uses the heat to be removed to provide the energy needed to remove the unwanted heat itself. Unlike common air conditioners which require significant additions of new energy (heat) to remove heat, nature puts the unwanted heat to use in moving that same heat to another place. Heat literally pulls itself out by its own bootstraps.

This paper ends with a demonstration of how refrigeration can be partially powered by the heat to be removed. This new approach is built on the foundation of natural cooling. Nature's benchmark will be used to measure old and new mechanical systems. The graph in Figure 1 shows what happens to the temperature of water as heat is added.

Figure 1: Sensible and Latent Heat of Ice, Water, and Vapor



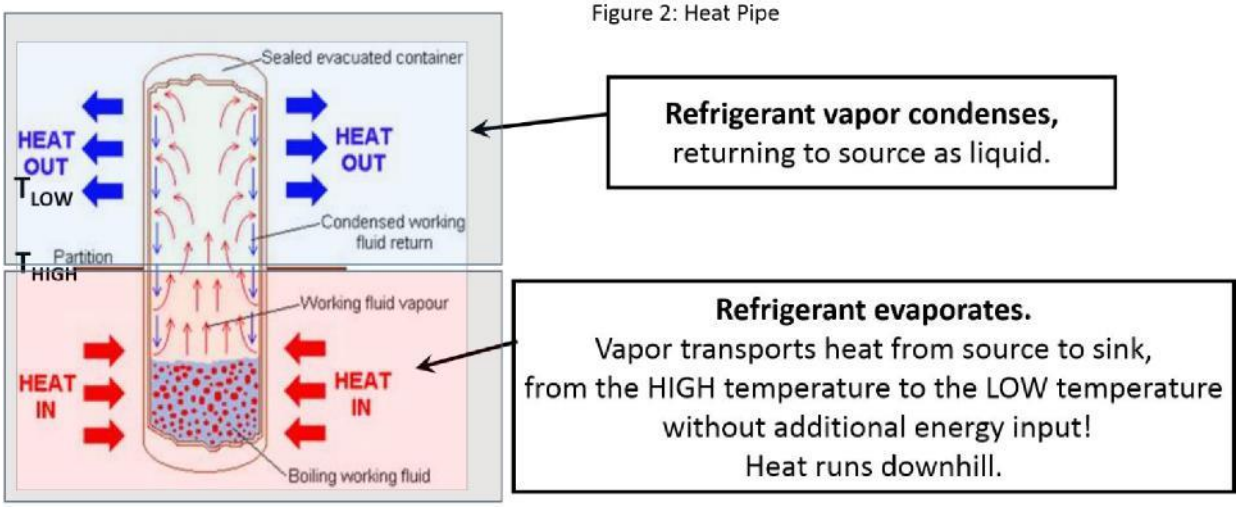
When changes in heat content cause changes in temperature, as from 1 to 2 and from 3 to 4, the heat is called sensible heat. Changing the temperature of one pound of water by 1°F defines the British thermal unit (Btu). When the addition or removal of heat does not change the measured temperature but contributes to a change of state instead, the change in heat content is called latent heat. A pound of water changes temperature from freezing to boiling from 3 to 4 with 180 Btu/lbm of sensible heat. Moving a pound of water from 4 to 5, from water to vapor, requires 970 Btu/lbm of latent heat. Without changing temperature in a contained constant pressure environment a mixture of liquid and vapor species results. The percentage of molecules remaining in the liquid state progressively decreases by 10% for every 97 Btu/lb of heat that is added. The temperature begins to change again only after 100% of the molecules have been vaporized. In other words the equivalent of almost a thousand degrees worth of sensible heat "stows away" on every molecule that evaporates. That means that a single molecule without changing temperature from 212°F will stow away 5.39 times as much heat as would be needed to move 180°F, from freezing to boiling.

The human body teaches how water removes heat every time we sweat. Bodies secrete water. Water evaporates. Cool! Fortunately fluids both evaporate (e.g. sweat) and condense (e.g. evening dew) at temperatures well below their boiling temperatures. Water's valuable capabilities will be incorporated into natural cooling and heating systems to be discussed later. For now it's enough to acknowledge the effectiveness of ***evaporative cooling below the boiling point*** in well-known systems ranging from the primitive "swamp cooler" (a fan with water sprayed into the air stream) to the massive cooling towers for nuclear power plants which depend on exactly the same technique.

Evaporative Cooling in a Heat Pipe

Figure 2 presents a simplified illustration of a heat pipe. The heat pipe refrigerant circulates from evaporation to condensation moving heat physically from one place to another. It uses only the energy from the latent heat that is being moved. The shape of the heat pipe can be a network of tubes, even flattened to work on the back of the latest cell phone. Evaporation takes place at the heat source. The vapor travels naturally to the cooler sink where the vapor rejects heat, dropping off its stow-away latent heat. With latent heat, fewer molecules are needed because each one carries so much stow-away heat.

Figure 2: Heat Pipe



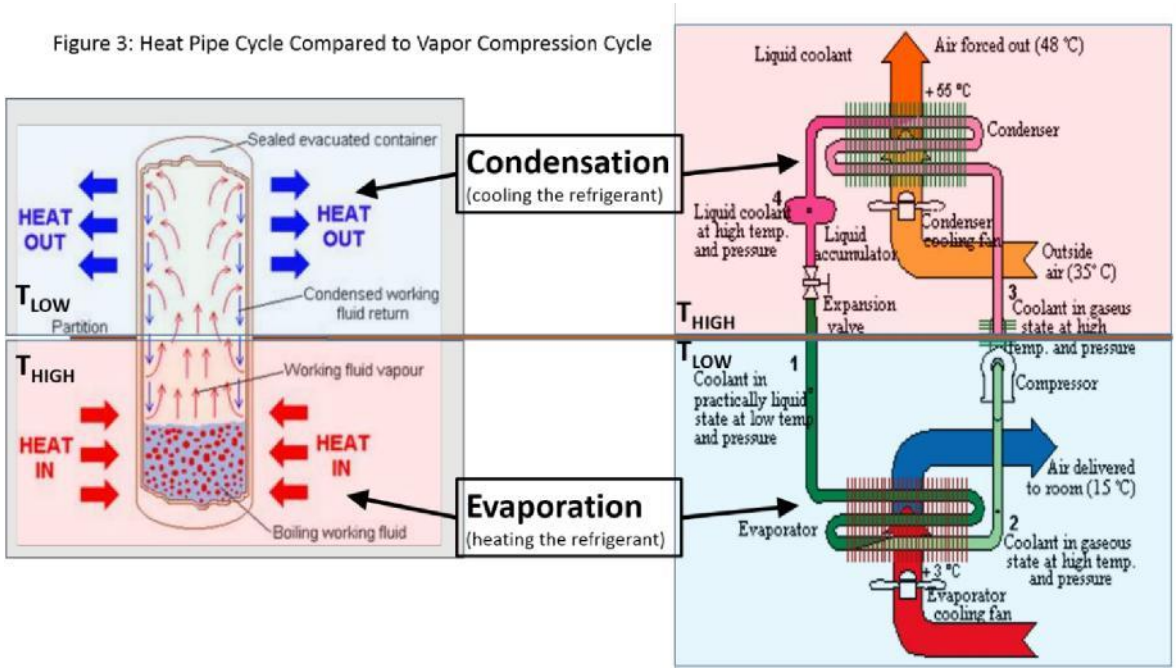
The cooled vapor will condense and return to the liquid state. The cooled liquid then flows back to the hot end for another load of heat. This natural heat conveyor runs "free", without requiring any additional input power. All the power for transporting and eliminating unwanted heat is supplied by the energy of the heat to be eliminated itself.

Air flows are separated in this illustration by a partition which prevents mixing of the heat flows or air streams. In practice the hot and cold ends may be some distance apart. The hot end may be in direct conductive contact with a heat source such as a computer or cell phone processing chip as mentioned previously. The liquid boiling point may be set to match precisely the temperature of the heat input by changing the pressure on the liquid (refrigerant) inside the heat pipe. Indeed, the liquid refrigerant mayeven be pumped for some distance and to new elevations at low cost because no change in pressure is required.

How to Turn the Heat Pipe Upside Down

Vapor compression is the method used by modern air conditioning systems to simultaneously change both the boiling point of the refrigerant for evaporation at a specific temperature and the boiling point for condensation at a different specific temperature. Figure 3 compares the heat pipe cycle to a vapor compression cycle. By raising the pressure of the condensing region above the pressure of the evaporator, heat can be removed from a lower temperature and rejected at a higher temperature. This makes it look like the heat pipe has been turned upside down. Rather than reject heat at an even lower temperature, as required by nature, work is applied to compress the vapor. As vapor compression raises refrigerant pressure, compression correspondingly also raises its condensation temperature above the target temperature to reject heat at the higher temperature.

Figure 3: Heat Pipe Cycle Compared to Vapor Compression Cycle



The vapor compression cycle engineers the temperature of each evaporation or condensation boiling point. The temperature at each boiling point is controlled by its pressure. Condenser pressure is elevated between 3 and 4 so the refrigerant temperature is also higher. Compression raises the temperature of the vapor well above its condensing temperature so most of the heat may be shed at temperatures above the condensing temperature. Lowering the evaporator pressure between 1 and 2 reduces both the refrigerant temperature and its boiling point. The evaporator will consequently accept heat when the environment presents heat at temperatures above this lower evaporation temperature.

Compressing the vapor from 2 to 3 reduces both the evaporator pressure and temperature while simultaneously increasing both the condenser pressure and temperature. Energy spent compressing the vapor does enable heat rejection at the higher temperature but the needed introduction of externally supplied energy totally negates any claim on the heat pipe's "free" heat conveyor. Vapor compression requires significant amounts of externally supplied energy. In vapor compression systems, the heat being removed no longer powers its own removal. Latent heat does reduce the number of molecules to be compressed but the net energy contribution plummets as temperatures rise.

Vapor Compression Requires "Excess Refrigerant Lift"

Work input to the vapor compression cycle is provided exclusively by compressing the vapor. This compression must be performed exclusively in the gas phase to avoid damaging the compressor. In order to measure this work and its results, various industry associations and standards bodies around the world define Rating Points. Rating Point protocols standardize the measurement of refrigerants including parameters for the mechanical systems within which they circulate. Outdoor temperatures range from 27°C-55°C while indoor temperatures range from 20°C-27°C. Only refrigerant R410A at the 35°C Rating Point will be discussed below.

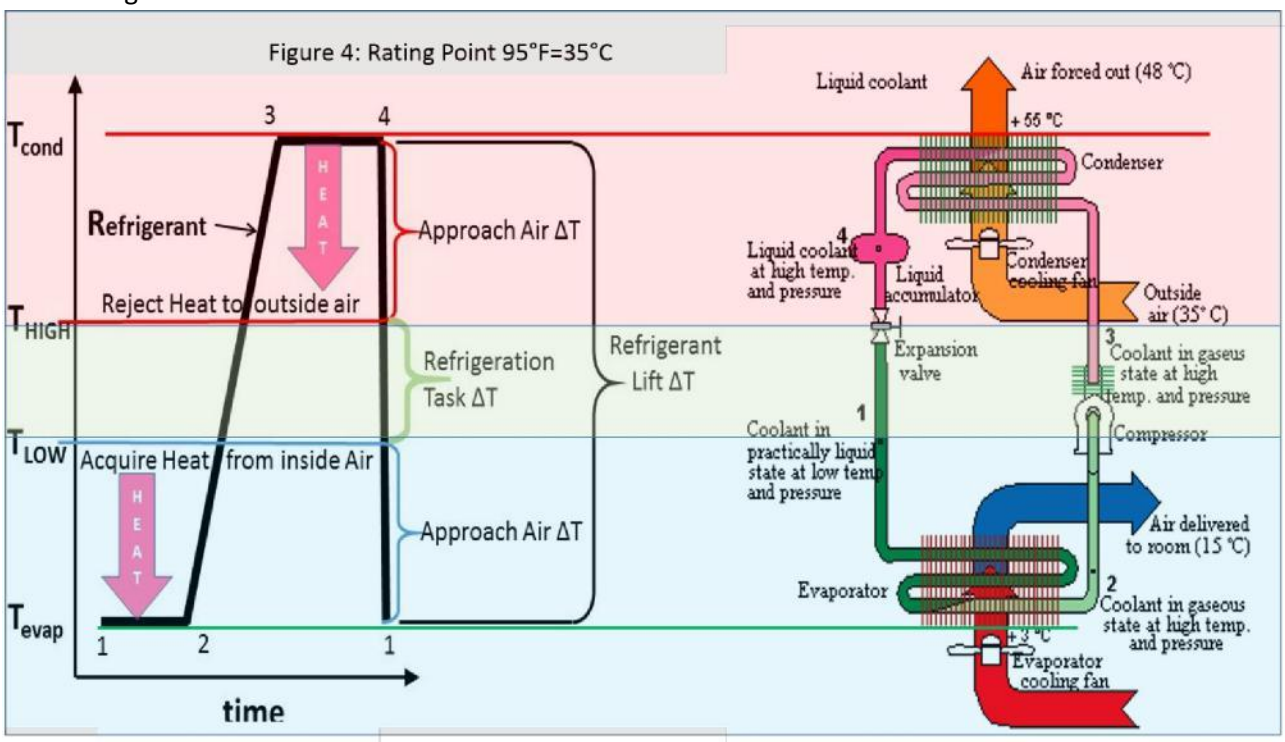


Figure 4 shows hot and cold regions, surrounding the green zone, the refrigeration task which is defined by the outside air temperature, T_{HIGH}=35°C, and the inside air temperature, T_{LOW}=23°C. In the US, this outside temperature, 35°C, defines the 95°F Rating Point. Inside air is separated from outside air by a partition such as the building wall. The refrigeration task is T_{HIGH}-T_{LOW}=35°C - 23°C =12°C. The refrigeration task itself is small compared to the temperature difference required between the evaporator and condenser, called the refrigerant lift. This refrigerant lift, T_{cond}-T_{evap}=55°C - 3°C =52°C as shown, is 4.3 times the size of the refrigeration task at the 95°F Rating Point.

Heat always flows "downhill". The amount of excess refrigerant lift needed is determined by the needed approach air temperature differential on both sides of the refrigeration task, called simply the approach temperature. Refrigerant alone creates the approach temperatures by moving evaporator and condenser temperatures outward beyond the refrigeration task itself. The size of approach temperatures is controlled by the transfer of heat from the heat exchanger to and from environmental air. The excess refrigerant lift is set to transfer heat into the air flows of the target environment at speeds near the system capacity, so approach temperatures are about 20°C for present technology.

The Inside Approach Air Temperature Differential

In practice room temperature is defined by the preference of each room's occupants. They express their choice for personal comfort by setting the thermostat, T_{LOW}, as shown in Figure 4. Hundreds of years before air conditioning, "Room Temperature" was defined by physicists. Also agreed to be the best temperature for drinking red wine, this European convention was stated in Celsius, 20°C. Changing social norms for clothing and human comfort around the world now recognize a "Room Temperature" of 23°C.

Decades ago, Great Britain led the successful global replacement of the English measurement system in all developed and commercially relevant countries, failing only against commercial opposition in the US. In the US the global standard room temperature would be stated in Fahrenheit as 73.4°F if commercial interests had not defined room temperature to suit their interests better. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) raised "Room Temperature" up to 80°F. By turning thermostats up 7°F ASHRAE could report sensible heat capacity improvement while leaving mechanical performance unchanged. This sleight of hand allows the industry to raise T_{evap}, without cutting the approach temperature. The claim of energy improvement by appearance only maintains the same inside approach temperature differential of 20°C while still cutting excess refrigerant lift and the energy needed to cool the evaporator.

This artifice creates a significant new problem. Raising the evaporator temperature also cuts the amount of humidity removed. In other words the higher T_{evap} increases relative humidity in the occupied space. Stated as the Sensible Heat Ratio (SHR), the fraction of total cooling capacity delivered as sensible heat is thereby increased without cost or technical advancement. Raising T_{evap} directly cuts the amount of condensation. Smaller amounts of total cooling capacity literally run down the drain as cold water. But higher levels of temperature and humidity have supported epidemic increases in mold, fungus, and dust mites, promoting sick building syndrome, and even Legionnaire's Disease. Yet ASHRAE continues to advertise and rate systems based on sensible heat capacity alone.

Even more boldly, ASHRAE stipulates that the energy expended in moving the inside air mass is not to be included in reports of system performance. Regardless of the fact that inside mass air flow must be reported and maintained, ASHRAE Standard 27-2009 in Table 3 stipulates that no inside energy data is to be recorded, apparently justified by the wide range of home ducting air resistance.

The Outside Approach Air Temperature Differential

As shown in Figure 4 for this Rating Point, the outside approach temperature is T_{cond}-T_{HIGH} = 55°C -35°C =20°C. This 20°C approach temperature mirrors the inside approach temperature.

In contrast to the inside operating costs which include the resistance to moving air through the unpredictable routing of building ducts, the outside or "air side" operating cost is more precisely measurable. It's the cost of moving a chosen mass flow of air through the vanes of the heat exchanger with a fan or blower. Total efficiency may be increased up to a maximum by increasing the mass flow of air, when refrigerant side mass flow is held constant.

Increasing the approach temperature will also increase the rate of heat transfer. In the best of all possible worlds, nature provides the desired cooler outside temperatures. In any real world where air conditioning is needed, the only means of increasing the approach temperature is to increase the excess

refrigerant lift. The losses of increasing excess refrigerant lift (refrigerant mass flow rate) always overwhelm the gains, but it's a necessary evil up to a point. The two mass flow rates and approach temperatures are inter-dependent and the incremental benefits related to each are not linear.

In order to optimize the design of air side operating efficiency, it would be necessary to manage the trade-offs among three separate subsystems: heat exchanger, refrigerant compressor, and external air blower. In operation it would be necessary to provide a real time controller to adapt as conditions change. Compressor and blower efficiencies appear to have plateaued over the past couple decades. The size of the heat exchanger is sometimes increased to reduce operating costs. This does raise the purchase price and justify the report of increased operating efficiency, but adding fins and tubes does not improve any technology. As was the case with raising room temperature, the industry claims to have increased efficiency in spite of the fact that this technology and its performance remain unimproved.

All three subsystems in their inside setting and in their outside setting will be considerably improved by the invention to be disclosed and the energy costs attendant to each will be shown to support adoption of the disclosed technology.

Coefficient of Performance (COP), Energy Efficiency Ratio (EER), and Seasonal EER (SEER)

These measures (dubiously) report the composite efficiency for three of the four subsystems. The COP also provides a theoretical best case standard for comparison to actual equipment.

COP is dimensionless. COP may be computed as the quotient of a relative temperature difference or as heat moved divided by work performed, heat and work being interchangeable. In addition to test conditions already defined at the 95°F Rating Point, the EER adds a standard for relative humidity. That being said, the EER is always proportional to the COP. EER=COP*3.41

The SEER applies a profile of temperature and humidity to match a range of climatological expectations. Nonetheless, it all comes back to COP which will be used to baseline comparisons between present technology and the disclosure to follow.

The best theoretical Carnot reversible COP for the refrigerant loop is defined using the absolute temperature scale. Evaporator and Condenser temperatures must be translated from Celsius into the absolute temperature Kelvin scale, Kelvin=Celsius+273. Using the condenser temperature of 55°C and the evaporator temperature of 3°C from Figure 4, the resultant COP is computed here using T_{Kelvin}.

Equation 1a: Best possible COP at Rating Point Excess Refrigerant Lift ΔT

$$\boldsymbol{COP = \frac{Tevap}{Tcond - Tevap} = \frac{276^{\circ}K}{328^{\circ}K - 276^{\circ}K} = 5.3077}$$

The National Institute of Standards and Technology (NIST) published a comparison of performance for refrigerants R410A and R22 across a range of temperatures. Compared to the best theoretical performance for lifting the refrigerant from 3°C in the evaporator to 55°C in the condenser (computed in Equation 1a) NIST observed COPs as low as 3.93 (File: c010723a.dat), dropping to 1.06 at an outside temperature of 68°C. This is the consequence of the compressor having to work harder to increase condenser pressure, hence system pressure ratios, as required to maintain the needed excess

refrigerant lift for temperatures at or near the critical point. At temperatures above the critical point a refrigerant will no longer condense. Maintaining the same approach temperature differential as outside temperatures rise is crucial because the presumed benefits of latent heat progressively disappear as temperatures approach the R410A refrigerant critical temperature.

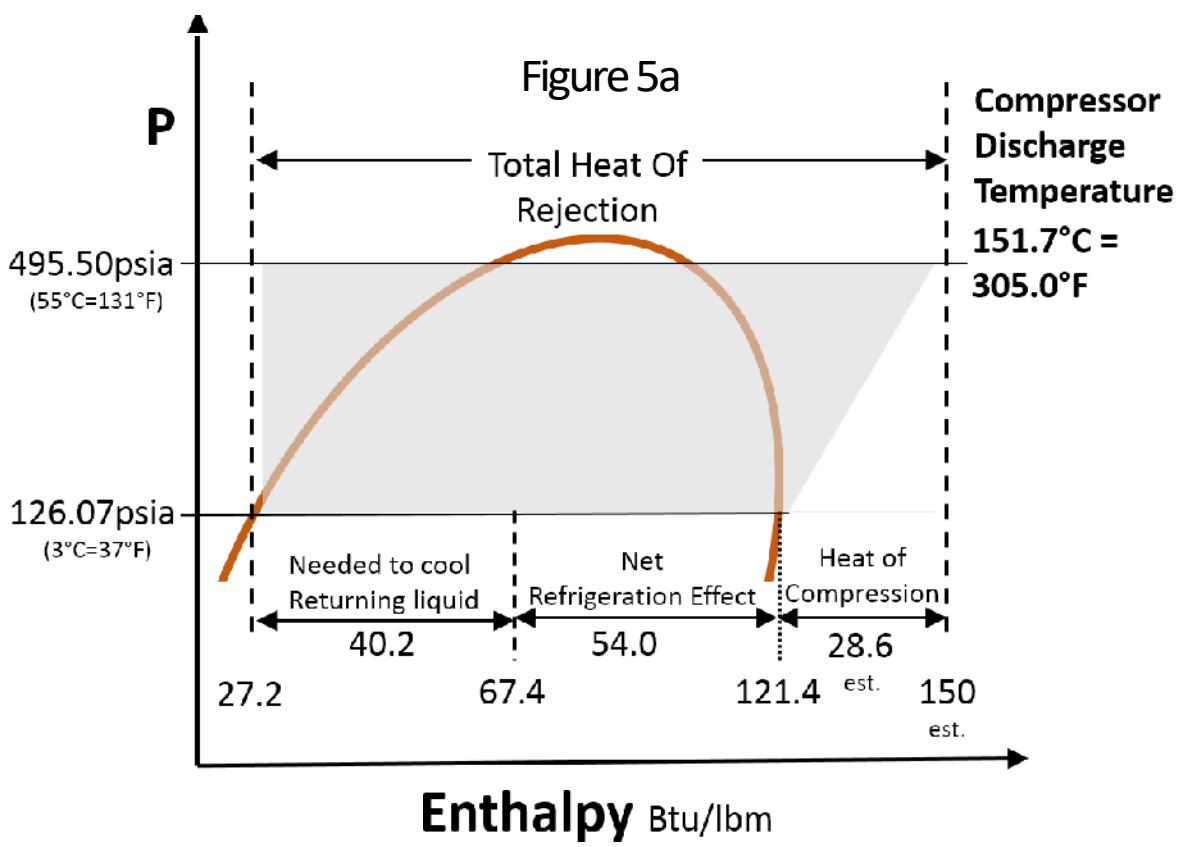
The technology to be disclosed will not be troubled by issues of high compression or the refrigerant failure near its critical temperature. It is instructive to compare the best case COP for divergent refrigeration shown in Equation 1a, COP=5.3 to the much higher COP expected for cooling the refrigeration task itself in Equation 1b, below. It should also be noted for reference. The technology to be disclosed will deliver performance much closer to the refrigeration task itself, no longer tied to excess refrigerant lift.

Equation 1b: Best Possible COP at Refrigeration Task ΔT

$$COP = \frac{Tlow}{Thigh - Tlow} = \frac{296}{308 - 296} = 24.6666$$

Myth Reliance on Latent Heat

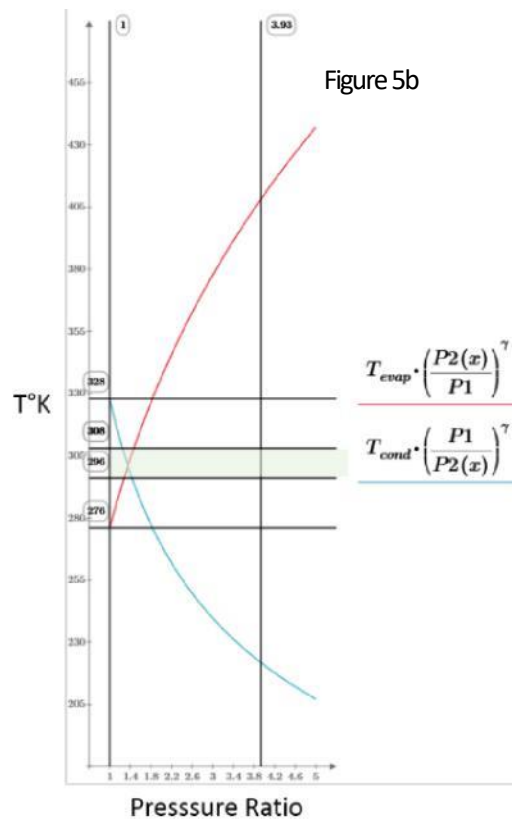
Best theoretical COP is computed in Equation 1a, above, using compressor and evaporator temperatures only. To set the boiling point for each temperature requires establishing a pressure corresponding to the boiling point of the chosen refrigerant. Because latent heat disappears altogether above the critical point (161.83°F = 72.13°C for R410A), a benchmark of latent heat contribution at the 95°F Rating Point provides an informative reference. Enthalpy numbers are provided by DuPont in R410A bulletin: T-410A-ENG, with compressor entry temperature of 57.64°F from NIST, Domanski and Payne, 2002 (b10328a).



The Net Refrigeration Effect of R401A is 54.0 Btu/lbm at the 95°F Rating Point. For reference, the latent heat of 54 Btu/lbm is 5% of the 970 Btu/lbm latent heat of water. The latent heat delivered in the condenser is only 53.6 Btu/lbm, 0.4 Btu/lbm less than the Net Refrigeration Effect in the evaporator. Consequently there is no net contribution of latent heat at the 95 F rating point. The entire refrigeration task is performed in the gas phase only.

The Pressure vs. Enthalpy graph fails to show the elevated temperatures that enable more than half of the total Heat Of Rejection (HOR) to be shed at temperatures significantly above the condenser temperature. This is shown by the red line in Figure 5b at the Pressure Ratio of 3.93. The entire refrigerant lift and all of the added work is handled exclusively as a gas, in the vapor phase. Most importantly, as the condenser temperature approaches the critical temperature, the contribution of latent heat goes to zero. Then all of the heat is rejected in the gas phase and the heat is rejected almost exclusively at temperatures above the nominal condenser temperature. Without this high temperature gas only heat rejection, vapor compression refrigeration would be useless even at common temperatures.

Note also the compressor discharge temperature shown in Figure 5a, 151.7°C=305.0°F, delivers a dramatic increase in the refrigerant lift which is neither measured nor even reported in refrigeration tables. The red line in Figure 5b shows the incease in compressor discharge temperatures as condenser pressure is increased to 495.5 psia, required at the 95°F Rating Point. The corresponding Pressure Ratio of 3.98 at that point is discussed below. Obviously both pressures and discharge temperatures continue to increase sharply as outside temperatures rise above 95°F.



The blue line in Figure 5b traces the cooling opportunity that could be recovered from an expanding gas, an opportunity foregone by the behavior of the two phase refrigerant. Not only is there no energy recovered from the expanding gas, which makes this cooling essentially “free”, the opportunity to enjoy the exceedingly beneficial refrigerant lift that mirrors high temperature discharge from the compressor is lost as well.

These measures fail to include the cost of moving the entire heat load into and out from the target environments with fans. Fans deliver the entire mass flow of air needed to move this heat twice, once on either side of the refrigerant loop. These numbers reported in Figure 5a reflect the cost within the refrigerant loop exclusively. The energy cost of operating fans and blowers to provide the mass flow of air required on both the source and sink sides of the vapor compression heat exchangers is not shown at all in cycle charts such as Figure 5a.

The disclosure below provides equivalent refrigeration at a Pressure Ratio of 1.15, working in the green zone between the temperatures of the refrigeration task itself.

Specific Heat, Pressure, Pressure Ratios, and Humidity

By restating the refrigeration problem into the context of a gas compression and expansion problem, it becomes easy to highlight several factors in ways not considered under the latent heat regime.

First problem, specific heat. Because R410A operates at or near the critical point, the contribution of latent heat is sharply reduced while contributions from sensible heat increase to take over completely as the refrigerant approaches "vapor phase only" temperatures in the condenser. The specific heat for R410A in the evaporator is less than 0.1953 Btu/lbm. The specific heat of air is 0.240 Btu/lbm. Air has a 23% higher specific heat than R410A, providing an attractive alternative to any refrigerant that fails to supply substantial contributions from latent heat.

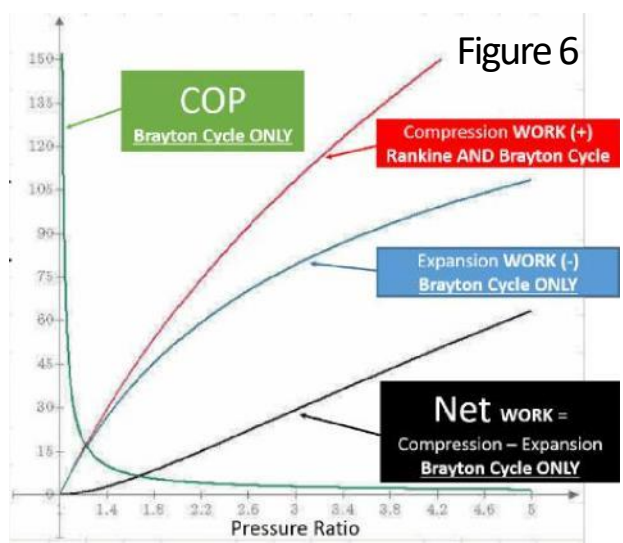
Second problem, pressure. The higher operating pressures of R410A have troubled its introduction, compelling the replacement of the R22 systems themselves rather than merely replacing their refrigerant. The R410A systems themselves cost more and are more expensive to run.

Third problem, pressure ratios. Higher pressure ratios are defined by increased compression work and necessarily higher energy costs as pressure ratios increase. The relatively high Pressure Ratio for operating R401A refrigerant loops is increasingly problematic from the energy consumption point of view. At the chosen Rating Point (95°F = 35°C) the resulting Pressure Ratio is 3.93 rising quickly above 4 with warmer outside temperatures as shown in Figure 5b.

Equation 2

$$\frac{\boldsymbol{Pcomp}}{\boldsymbol{Pevap}} = \frac{\mathbf{495.5}\ \boldsymbol{psia}}{\mathbf{126.07}\ \boldsymbol{psia}} = \mathbf{3.93}$$

To establish a reference for compression work needed in the R410A refrigerant loop, Figure 6 shows the work components and resultant net work with COP for a Brayton Cycle across a broad set of pressure ratios. As noted previously, the work input to a vapor compression process is performed exclusively on the vapor; strictly a gas phase compression which shows as the red line. Because the refrigerant returns as a liquid, there is no gas phase expansion work to offset the compression work performed on the R410A refrigerant. Consequently, the work of expansion cannot be extracted mechanically and subtracted from the work of compression. Because there is no expansion work to be subtracted from the compression work, the compression-only work necessarily increases much more rapidly as pressure ratios rise. No work is extracted as the liquid is returned to the lower pressure. And no work is extracted during the change of phase back to vapor. Instead the "suction" of the compressor is required to maintain the low



pressure of the evaporator. The mechanics of vapor compression have converted the opportunity to extract expansion work from vaporization. The “steam engine” potential is lost to free expansion.

Fourth problem, humidity. As humidity rises, performance drops precipitously due to the previously acknowledged high latent heat of water. The process of cooling air often results in cooling the air below its dew point, precipitating water which runs down the drain after consuming 20%-35% of total cooling capacity. This was discussed in some detail above in relation to the inside approach air temperature. The Rating Point model calls for raising the temperature of recirculated inside air by about 10°C, a sensible heat of 18 Btu/lbm. This strategy avoids a considerable cost for removing humidity. Condensing costs the full 970 Btu/lbm, 970/18=53.9 times more than the cost of cooling dry air by 10°C. There is no cooled air to show for this considerable expenditure of energy. Quite the opposite. The entire cooling load of condensation runs down the drain as chilled water, after having released the full 970 Btu/lbm of heat directly into the air stream that is intended to be cooled.

Recapping the state of the art

Entrenched beliefs favoring 2 phase refrigeration solutions fail to recognize the following.

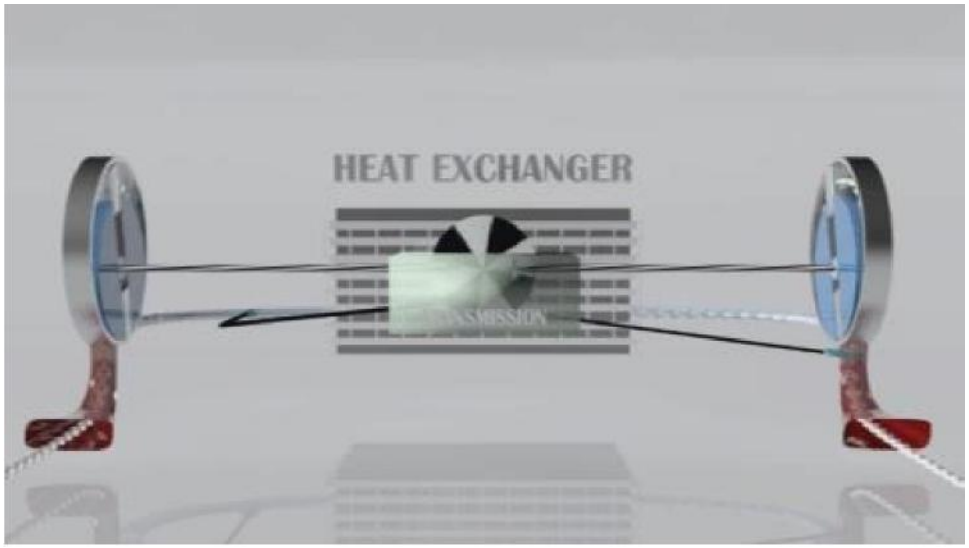
1. Latent heat makes no contribution above the 95F Rating Point.
2. ALL heat rejection is provided in the vapor phase.
3. The specific heat of air is higher than refrigerants in the vapor phase.
4. ALL heat rejection is delivered at pressure ratios at or above 4.
5. Compression of the entire gas cycle is accomplished in a primitive vane pump.
6. Compression of air as an alternative to refrigerants is dismissed “because pumping efficiencies are too low”.
7. Incredible improvements in COP are available as pressure ratios drop below 0.4.
8. Commonplace pump designs ranging from 100 to 150 years old realize adequate pumping efficiencies at pressure ratios in the needed range near 1.1.

What is the least costly way to change the temperature of the air in the room?

- It has always been known and always understood that, whether heating or cooling, we pass the needed mass of air over the heat exchanger. Air has the needed heat capacity.
- It has long been known and often understood that heat transfers into the air faster when the approaching air temperature is farther away from the temperature of the heat exchanger.
- It has recently been known and rarely understood that 40% of the heat is lost in “free expansion” when potential work of expanding and contracting gasses is never captured.
- It has not been known and never understood that the cost of changing the air temperature before it hits the heat exchanger can be much less than the cost of changing the heat exchanger temperature by the same amount.

The Better Way: Fan Replacement and Refrigerated Air Flows

1. By placing the heat exchanger between two pumps, it is possible to capture the 40% energy rebate provided by nature every time heat is transferred with air.



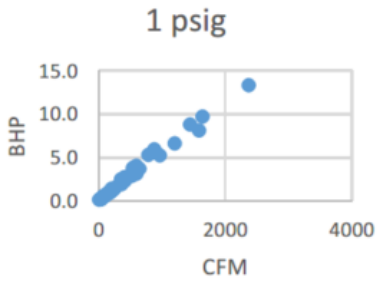
2. The fastest efficient means of changing the temperature of a mass of air is thermodynamically certifiable to be changing its pressure.
3. To secure the most favorable temperature gradient between air and any convective heat source or sink, it costs less to change the temperature of the air than the source or sink.

Proof: The Better Way is 150 years old

The following analysis separates the cost of compression mirrored by a complimentary expansion in a set of Dresser Roots Blowers. This analysis will identify the energy costs attributable to compression separating them from the cost of moving air through the positive displacement system. It will be shown that once the compression energy (offset by expansion and work capture during heat transfer) is subtracted from total work input, the cost of moving air through the dual pump system is well below the cost of moving the same mass flow of air with fans.

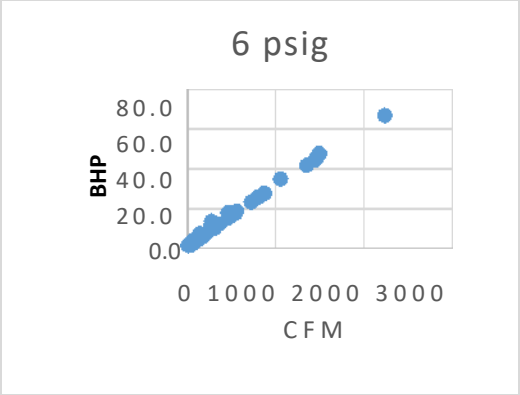
Dresser URAI blower performance is specified for the whole family of blowers on the third page of the ROOTS UNIVERSAL RAI specification (attached) and excerpted here. Mass flows are suitable as stated because air flows in refrigeration systems are normally fan driven by fans. The desired changes in pressure (temperature) maintain the same mass flow. URAI specifies inlet pressure of 14.7 psia at 68°F, specific gravity 1.0. Vacuum discharge is 30” Hg.

It can be seen in the following scatter plots at 1psig and 6psig that the energy cost to both move and compress a cubic foot of air increases roughly linearly across the range of flows and pressures regardless of the device actually chosen.



Frame Size	Speed RPM	1 PSI		6 PSI		Max. Vacuum		
		CFM	BHP	CFM	BHP	"HGV	CFM	BHP
	1160	10	0.1			46		0.2
22	3600	49	0.3	38	1.6	14	28	1.8
	5275	76	0.5	64	2.4	15	53	2.8
	1160	24	0.2			6	12	0.5
24	3600	102	0.6	83	3.1	14	69	3.5
	5275	156	0.9	137	4.6	15	119	5.5
	1160	40	0.2	21	1.4	10	18	1.1
32	2800	113	0.6	95	3.4	15	78	4.1
	3600	149	0.9	131	4.4	16	110	5.3
	1160	55	0.3	31	1.9	10	27	1.5
33	2800	156	0.9	132	4.6	14	113	5.2
	3600	205	1.2	181	6.1	15	159	7.3
	1160	95	0.5	61	3.1	10	55	2.5
36	2800	262	1.5	229	7.7	12	213	7.5
	3600	344	2.1	310	10.1	15	278	12.1
	860	38	0.2	18	1.4	8	19	0.9
42	1760	92	0.5	72	2.8	14	56	3.2
	3600	204	1.4	183	6.1	16	160	7.7
	860	79	0.5	42	2.7	8	46	1.8
45	1760	188	1.0	151	5.7	12	134	5.5
	3600	410	2.7	374	12.2	16	332	15.4
	860	105	0.6	59	3.6	8	63	2.4
47	1760	249	1.3	203	7.5	12	181	7.3
	3600	542	3.5	496	16.1	15	452	19.1
	700	72	0.4	42	2.4	10	36	2.0
53	1760	211	1.2	181	6.3	14	158	7.1
	2850	355	2.5	325	10.7	16	291	13.4
	700	123	0.7	78	4.1	10	70	3.3
56	1760	358	2.0	312	10.5	14	276	11.8
	2850	598	4.0	553	17.7	16	501	22.4
	700	187	1.0	130	5.9	8	135	3.9
59	1760	529	2.9	472	15.3	12	445	14.9
	2850	881	5.9	824	26.0	15	770	30.8
	700	140	0.8	93	4.5	12	71	4.4
65	1760	400	2.4	353	11.9	16	300	15.2
	2350	546	3.8	499	16.4	16	445	25.6
	700	224	1.2	149	7.3	10	135	5.9
68	1760	643	3.7	567	18.9	15	495	22.7
	2350	876	5.6	801	25.9	16	715	32.8
	700	420	2.3	279	13.6	8	292	8.9
615	1760	1205	6.6	1063	34.9	12	997	33.9
	2350	1641	9.7	1500	47.6	14	1389	53.4
	575	192	1.1	134	6.1	12	117	6.0
76	1400	527	3.0	468	15.4	16	413	19.7
	2050	790	5.3	731	23.4	16	674	29.5
	575	362	1.9	271	11.1	12	228	10.9
711	1400	970	5.2	880	27.7	15	793	33.5
	2050	1450	8.8	1359	41.8	16	1256	53.1
	575	600	3.1	470	18.1	10	446	14.8
718	1400	1590	8.1	1460	44.8	12	1398	43.6
	2050	2370	13.3	2240	66.9	12	2178	64.7

These plots normalize the assortment of devices and mass flows stated in the tables. Because the proposed Refrigerated Air Flow (RAF) systems will operate primarily near atmospheric pressure ±10%, rarely exceeding 20% differences, only the 1psi column contains data governing the relevant conclusions. Others provide confirming data beyond this range.



Rather than simply moving the air, the objective of RAF is to move a comparable mass flow of ambient air through a pressure differential sufficient to change its approach temperature to a desired level in relation to the heat exchanger. In conventional systems the ambient (target environmental) mass flow is passively fed across a heat exchanger whose temperature is separately engineered to provide the desired rate and direction of heat flow. Contrast this to RAF systems where the ambient (target environmental) mass flow is itself used as the refrigerant. The temperature of RAF mass flows is itself engineered to provide the desired rate and direction of heat flow now being exchanged with a "passive" heat exchanger whose source or sink is thermodynamically considered to be outside the thermodynamic system under consideration.

Correspondingly, in order to compare the energy that would otherwise be required simply to move the air it is necessary to identify the cost of compressing the air and subtract it from the reported cost of compressing the air which necessarily also includes the cost of moving the air as well.

NO HEAT TRANSFER: For the case where no heat is transferred following compression, a follow-on expansion process might recover the entire energy cost of compression directly by complimentary mechanical means. The Roots Blower offers such a mechanism. Notably this energy recovery mode during expansion is different from both the compression operation and the vacuum pump for which data is available. But a free-wheeling exit pump would not sustain the plenum pressure as needed for heat transfer under constant pressure. An electrical load would be provided to the motor/generator governing the speed of the exit pump, making it act in a manner effectively identical to the entry pump. So the cost of compression would be exactly offset by expansion accepting of course that there are losses to be recognized on both sides.

HEAT ACQUIRED: For the case where heat is acquired within the plenum, the resulting increase in volume will directly increase the energy recovered at exit, indeed this becomes a "heat engine". The introduction of heat between the two pumps is analogous to a jet aircraft engine, producing a direct energy yield due to the introduction of heat. Indeed, as defined by the coefficient of heat under constant pressure, nature provides an energy bonus equal to 40% of the heat acquired, a volume increase which can produce electricity to offset the power used in compression

HEAT REJECTED: For the case where heat is rejected within the plenum, the resulting decrease in volume will directly decrease the energy recovered at exit. In this case the departure of heat from the mass within the plenum reduces the volume (not mass) within the plenum by 40%. Strikingly, this reduction of volume affects the system and its net energy in a manner analogous to the "heat engine" behavior described above. Because the plenum pressure must be reduced by the exit pump, this energy expenditure is offset by the energy recovered at the entry, 40% more, as above, even though the volume between the two pumps is reduced.

When all is said and done, the transfer of heat makes a 40% contribution to offset the losses related to compressing and expanding air. This net contribution may substantially offset pumping losses depending on the capability of the pumps as well as on the compression ratios and the heat finally transferred. Because this exercise is limited to a pressure of 1psig, a pressure ratio of 1.068, it can be confidently

assumed that compression costs will be offset by expansion gains and vice-versa. Looking at the operating energy requirements reported by Dresser, the full value of compression energy may be subtracted from the operating energy cost, leaving all losses chargeable to air movement alone.

The compression efficiencies for each entry in the Dresser table have been identified and sorted in the order of increasing compression efficiency. This is shown in the right hand column. Any pump actually designed and developed for these low pressure ratios will easily meet or exceed the best performance of this 150-year-old design. Because the Roots Blower was intended for much higher pressure ratios, it is appropriate to benchmark compression performance at 90%, knowing that the entire cost of compression and expansion will be directly offset, i.e. zeroed out. Dresser Frame #718 delivers 1590/0.81 CFM/BHP total or 2628 CFM/Kw for air movement alone, after the cost of compression has been removed.

Compared to residential HVAC air flows (2,000 CFM/Kw inside and 4,000 CFM/Kw outside), any such 2628 CFM/Kw unit will deliver heating and cooling comfortably within the energy budget of present fan systems alone.

Frame Size	Speed RPM	1 PSI						
		CFM	BHP	ft*lbf/min	ft*lbf (ΔT)	Air Movement cost=E-F	Air Movement cost/Total	Efficiency=T /total
22	1160	10	0.1	3300	1510	1790	54.25%	45.75%
24	1160	24	0.2	6600	3624	2976	45.09%	54.91%
53	2850	355	2.5	82500	53601	28899	35.03%	64.97%
65	2350	546	3.8	125400	82441	42959	34.26%	65.74%
42	3600	204	1.4	46200	30802	15398	33.33%	66.67%
76	2050	790	5.3	174900	119282	55618	31.80%	68.20%
59	2850	881	5.9	194700	133022	61678	31.68%	68.32%
56	2850	598	4.0	132000	90292	41708	31.60%	68.40%
45	3600	410	2.7	89100	61906	27194	30.52%	69.48%
22	5275	76	0.5	16500	11475	5025	30.45%	69.55%
47	3600	542	3.5	115500	81837	33663	29.15%	70.85%
68	2350	876	5.6	184800	132267	52533	28.43%	71.57%
45	860	79	0.5	16500	11928	4572	27.71%	72.29%
22	3600	49	0.3	9900	7399	2501	25.27%	74.73%
36	3600	344	2.1	69300	51941	17359	25.05%	74.95%
711	2050	1450	8.8	290400	218936	71465	24.61%	75.39%
32	3600	149	0.9	29700	22498	7202	24.25%	75.75%
65	1760	400	2.4	79200	60396	18804	23.74%	76.26%
615	2350	1641	9.7	320100	247775	72325	22.59%	77.41%
24	3600	102	0.6	19800	15401	4399	22.22%	77.78%
33	3600	205	1.2	39600	30953	8647	21.84%	78.16%
24	5275	156	0.9	29700	23554	6146	20.69%	79.31%
33	2800	156	0.9	29700	23554	6146	20.69%	79.31%
68	1760	643	3.7	122100	97087	25013	20.49%	79.51%
76	575	192	1.1	36300	28990	7310	20.14%	79.86%
36	2800	262	1.5	49500	39559	9941	20.08%	79.92%
47	860	105	0.6	19800	15854	3946	19.93%	80.07%
65	700	140	0.8	26400	21139	5261	19.93%	80.07%
76	1400	527	3.0	99000	79572	19428	19.62%	80.38%
56	700	123	0.7	23100	18572	4528	19.60%	80.40%
53	1760	211	1.2	39600	31859	7741	19.55%	80.45%
718	2050	2370	13.3	438900	357846	81054	18.47%	81.53%
56	1760	358	2.0	66000	54054	11946	18.10%	81.90%
53	700	72	0.4	13200	10871	2329	17.64%	82.36%
59	1760	529	2.9	95700	79874	15826	16.54%	83.46%
615	1760	1205	6.6	217800	181943	35857	16.46%	83.54%
615	700	420	2.3	75900	63416	12484	16.45%	83.55%
33	1160	55	0.3	9900	8304	1596	16.12%	83.88%
42	1760	92	0.5	16500	13891	2609	15.81%	84.19%
711	1400	970	5.2	171600	146460	25140	14.65%	85.35%
68	700	224	1.2	39600	33822	5778	14.59%	85.41%
59	700	187	1.0	33000	28235	4765	14.44%	85.56%
45	1760	188	1.0	33000	28386	4614	13.98%	86.02%
32	2800	113	0.6	19800	17062	2738	13.83%	86.17%
36	1160	95	0.5	16500	14344	2156	13.07%	86.93%
42	860	38	0.2	6600	5738	862	13.07%	86.93%
711	575	362	1.9	62700	54658	8042	12.83%	87.17%
47	1760	249	1.3	42900	37597	5303	12.36%	87.64%
718	575	600	3.1	102300	90594	11706	11.44%	88.56%
718	1400	1590	8.1	267300	240074	27226	10.19%	89.81%
32	1160	40	0.2	6600	6040	560	8.49%	91.51%

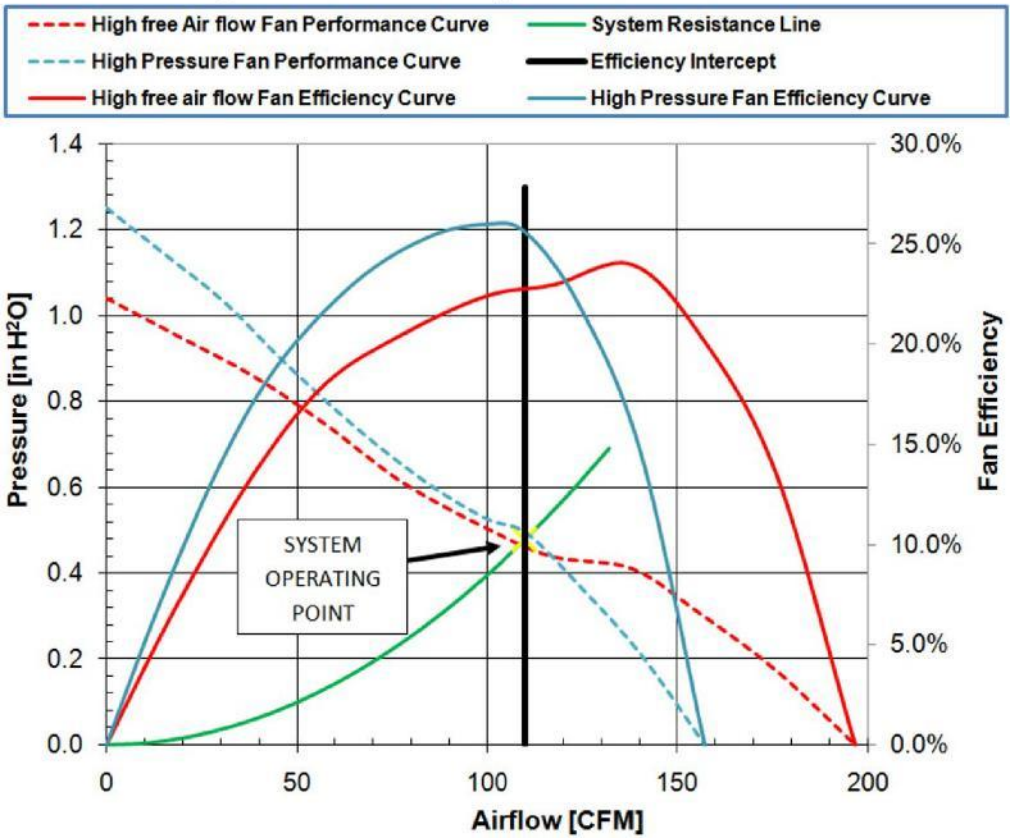
Conclusion

The analysis has identified several factors which control the energy needed to change the pressure of a mass flow of air between two pumps. Whether the temperature between the pumps is changed or not, and whether heat is transferred or not, the complimentary compression/expansion energy can be definitively identified. Subtracting this fully recovered compression/expansion energy component from the total pumping energy reveals the cost of moving air through the system, nominally through the connected system where the follow-on pressure is measured only in inches of water. The cost of moving air through the dual pump system is well below the cost of moving the same mass flow of air with fans.

Fan Efficiency vs. Operating Performance

Once the approach air differential is established, the fans on either side of the refrigerant loop become final fixed gate keepers for all heat transfer, limiting or enhancing efficiency. Yet fans and blowers generally operate well below half of their own announced efficiency. Figure 7 shows the relationship between a fan's theoretical "free air flow" operating performance and its capability once air flow resistance is encountered. Even slight resistance cuts nominal fan efficiency in half or more. Figure 7 would be typical for the outside unit of a split air conditioning system.

Figure 7: Fan Efficiency and Performance



Matching the inside fan or blower to the resistance of duct work is difficult because the length and routing of ducts is seldom known, much less understood. This explains why the inside air movement cost is generally omitted from system performance measures altogether.

Fan and blower driven systems raise pressures measured only in inches of water. The typical range of fan operating pressures is well below 1 inch of water (0.036 PSIA) which would be a gauge pressure ratio of 0.036/14.7=0.002, only two thousandths. Blowers in large building systems are powered by many horsepower, yet they seldom reach pressure ratios above 1.1. Figure 6 shows us that their efficiency should be very high if they were designed and configured for compression.

Unlike advertising claims, subject to regulation, typical energy requirements for fans and blowers are well published in repair and training manuals. These sources separate compressor data and air movement costs which are otherwise unreported. Air movement energy is reliably proportional to system heating and cooling energy so “rules of thumb” are useful and widely accepted.

- Inside mass air flow of 400 CFM is required for a ton of cooling capacity
- Energy usage is 1.1 kW/ton at the Department of Energy mandated COP of 3.2
- The outside fan uses 10%. The compressor alone draws 90%, 0.99 kW/ton. Use 1 kW/ton
- Inside air flow runs about 2.5 times the outside unit with wide variability, use 0.25 kW/ton
- Sensible Heat Ratios are 65 to 80 leaving latent heat losses of 20%-35%, use 0.30 kW/ton

These benchmarks will be referenced in the following disclosure.

Detailed Description of the Preferred Embodiments

The latent heat argument asserts that air does not provide sufficient heat capacity for refrigeration. This assertion falls before the indisputable fact that all latent heat refrigeration necessarily requires a mass flow of AIR adequate to carry all the heat that vapor compression can move. Air alone carries the entire heat load of vapor compression on both sides of every vapor compression system. There is no problem with the heat capacity of air.

Air can do the refrigeration job without vapor compression. Here's how.

1. Replace fans with pumps to capture lost energy of free expansion during heat transfers. The bonus is a direct work dividend equal to 40% of all the heat that's moved.
2. Refrigerate Air Flows (RAF) across heat exchangers to eliminate excess refrigerant lift.
3. Replace vapor compression refrigeration loops with any suitable air-to-air heat exchanger.

Quantify the Operating Principles and Cost of Operation for Refrigerating Air Flows

Heat Pipe Technology, Inc. (HPT) provides the following formula to compute power required to drive an air flow through the supplied HPT heat exchanger. A range of standard and custom heat exchangers is suggested along with a selection of air speeds to be incorporated in engineering the desired result. This exercise is strictly confined to the demonstration of feasibility in replacing vapor compression with Refrigerating Air Flows. HPT suggests motor efficiency of 0.9 and fan efficiency of 0.75.

Equation 3a

$$kW = \frac{CFM}{11674 * Motor\ Eff * Fan\ Eff}$$

The common Roots Blower provides exceptional efficiencies at the pressure ratios needed for Fan Replacement and Refrigerating Air Flows. Not only is volumetric efficiency exceptional at all but the lowest air flows, the compression efficiency is so well matched by expansion efficiency that the Roots device is often selected as a vacuum pump.

Substituting fans with Fan Replacement utilizing a pair of Roots Blowers, or alternatives, operating at pressure ratios near 1.1, the efficiency of each blower or positive displacement pump is near 0.9. Fan Replacement efficiency is correspondingly characterized as 0.9*0.9=0.81. Utilizing a typical 3 ton household air flow of 1250 CFM through the HPT heat exchanger HRM 3040 calls for the following power.

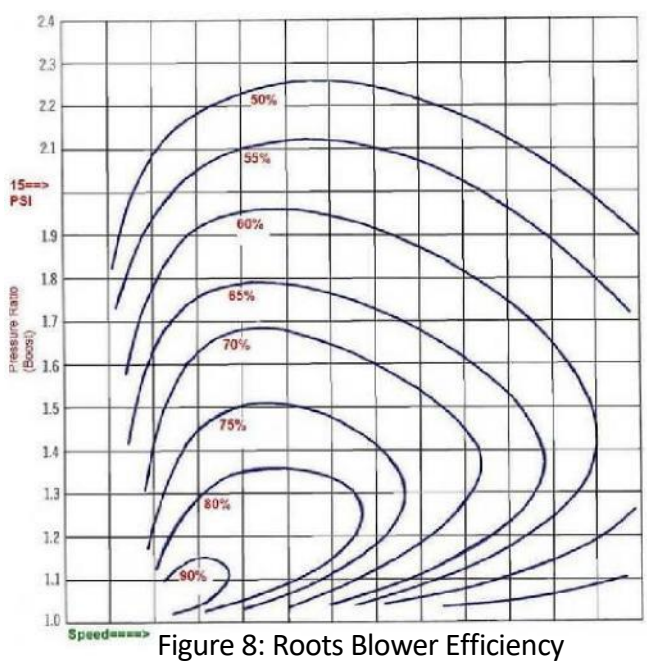


Figure 8: Roots Blower Efficiency

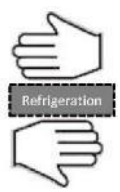
Equation 3b

$$kW = \frac{CFM}{11674 * Motor\ Eff * Fan\ Eff} = \frac{1250}{11675 * \left(0.9 * (0.9 * 0.9)\right)} = 0.147kW$$

As expected, using a pair of positive displacement pumps will move air more efficiently than the fan they replace. When heat is exchanged between the Fan Replacement pumps, the bonus harvest of 40% of the heat exchanged will be reduced by pumping losses. Nonetheless, Fan Replacement at the pressure ratio of 1.0 still yields a net gain quite close to this goal. When pressure changes are introduced to generate Refrigerating Air Flows at pressure ratios near 1.1, the pumping losses are far smaller than vapor compression systems operating at pressure ratios near 4.0. A far greater gain is the recapture of expansion energy in Convergent Refrigeration, discussed below. This formula establishes the benchmark for moving mass flows of air through HPT heat exchangers. It will be taken into the following disclosure as evidence that two pumps in sequence can move the same mass of air as a fan and with less energy. Further, by changing the pressure between the pumps, refrigerating the same mass of air that is necessarily moved across every vapor compression heat exchanger, the energy wasted on excess refrigerant lift can be eliminated. Indeed, the vapor compression loop itself can be eliminated.

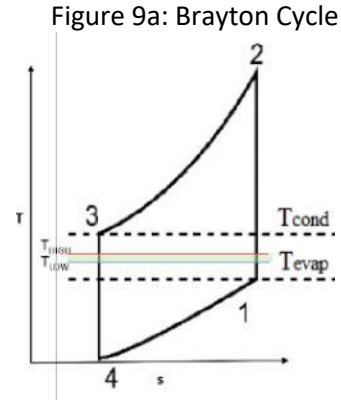
Pressure Ratios and Energy Requirements

Vapor compression refrigeration is Divergent Refrigeration because the required excess refrigerant lift diverges from the refrigeration task itself. Excess refrigerant lift is shown here by the width of the hands, extended outside the refrigeration task. The thumbs in this icon point the direction and the extent of excess refrigerant lift needed well beyond the refrigeration task. Vapor compression systems necessarily create the approach air temperature differential using excess refrigerant lift as the only available means by which to cause heat to flow to and from the external air flows. Excess refrigerant lift must be adequate to compel heat transfer through the heat exchanger from the refrigerant loop and into the external air flow. Excess refrigerant lift must be increased still further to assure the desired rate of heat flow into and out from the external air flows in balance with the capability of the refrigerant compressor.



The large temperature change characteristic for every Divergent Refrigeration system can be diagrammed as the Brayton Cycle, Figure 9. Convergent Refrigeration is performed within the green zone, between T_{HIGH} and T_{LOW}, the refrigeration task. The compression step from P_{evap} to P_{cond} is shown here as 1 to 2, followed by heat rejection at constant pressure 2 to 3. This is exactly the same path followed by the vapor in EVERY vapor compression system up to the point where condensation begins at 3. Then liquid temperatures never fall below Tevap and the latent heat of evaporation is offset by cooling the liquid and expansion losses. In vapor only systems, when there is no latent heat rejection, condensing the vapor to a liquid as in Figure 5a, the gas may be returned to its initial pressure, 3 to 4. Because much of the heat produced by compression work, 1 to 2, has been rejected along the constant pressure curve from 2 to 3, the gas expanding from 3 to 4 is cooler than it

Figure 9a: Brayton Cycle



began at 1. This cooler gas then acquires heat from its surroundings at the lower temperature at constant pressure, 4 to 1. Convergent Refrigeration engineers Brayton Cycle efficiencies by operating within the green zone.

When the expansion work can be used to directly offset the compression work, as with a turbine, the net work that must be added from an external source is reduced by the amount of energy recaptured during expansion. The resulting COP increases exponentially as pressure ratios fall. No such possibility exists for vapor compression systems because the low pressure region is constantly under suction from the compressor. As documented in the discussion of Figure 5a, above, the latest vapor compression refrigerants contribute no net latent heat in the condensation stage. It is useful to question why vapor compression systems have not been replaced with air cycle refrigeration systems.

Competitive life cycle costs for air cycle systems were not only certified in the late 1990s, the closed loop air cycle systems developed for trains at that time are still viable and continue to be re-adopted for Germany's most advanced bullet trains. Yet wider adoption of air cycle refrigeration was blocked as the 20th century drew to a close. The less effective HCFC refrigerants now mandated must accordingly fail to provide the same life cycle cost competition against proven air cycle alternatives. A fresh review of HCFC's lesser capabilities will leave them vulnerable to direct displacement in today's market by air cycle refrigeration as fashions increasingly favor environmentally friendly alternatives in the future.

Not only can the expansion work of cooling be used to directly offset the compression work of heating, the energy spent creating excess refrigerant lift as well as temperature overshoot can be essentially eliminated. Figure 9b suggests this capability of Convergent Refrigeration. Several embodiments of Convergent Refrigeration will be detailed by examples that demonstrate our claims below.

Figure 9b: Convergent Refrigeration fits inside both closed loop Vapor Compression and closed loop Air Cycle refrigeration schemes.

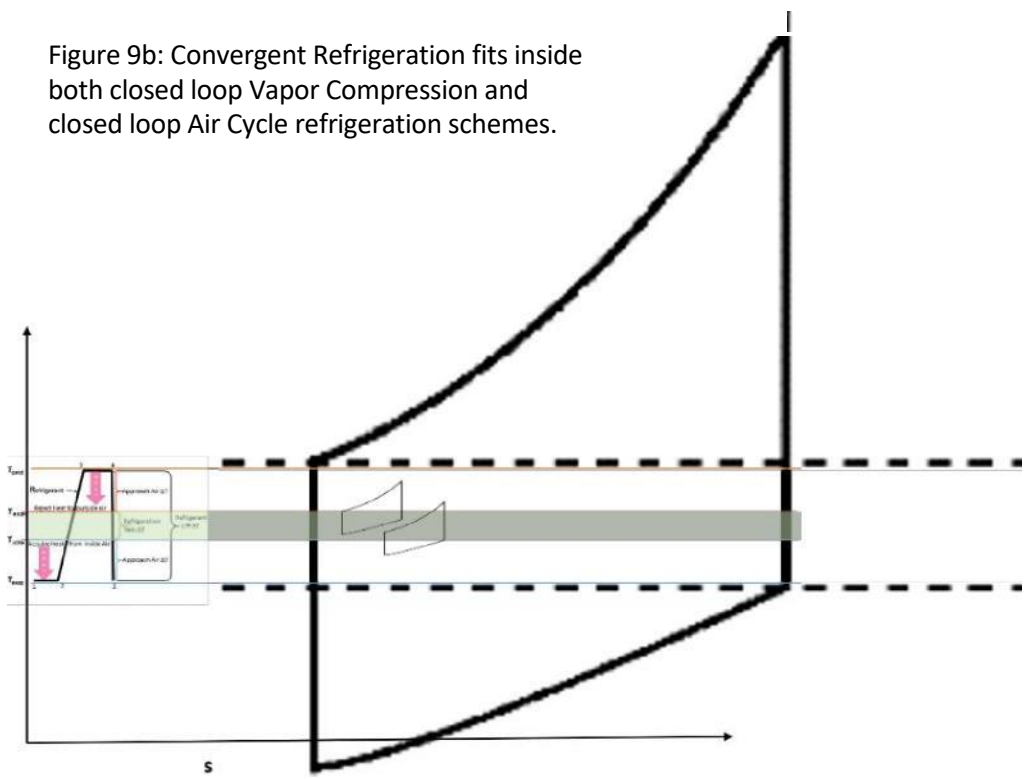
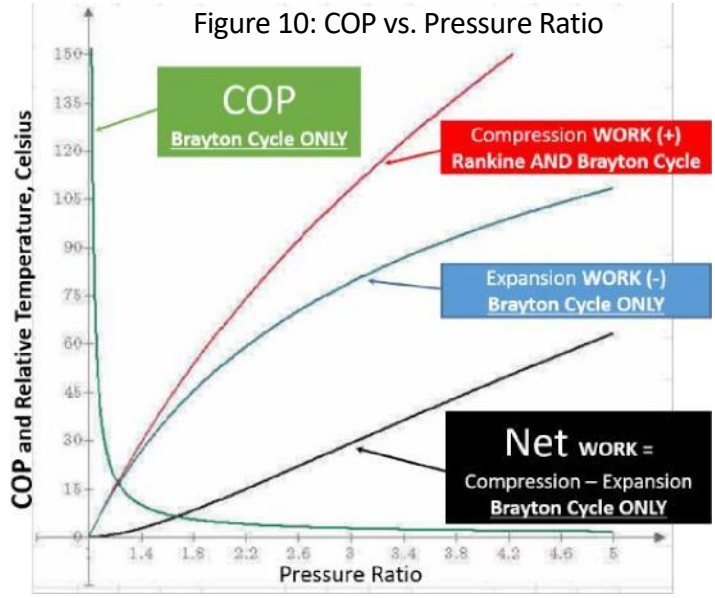


Figure 10 details the performance of the closed loop air cycle systems described above. The trace of compression work necessarily follows the path of all adiabatic compressors, even blowers and fans with losses increasing progressively for each. Note especially that the compression work shown in Figure 10 tracks necessarily with R410A in the vapor phase. Given R410A's somewhat lower specific heat when compared to air, the mass flow for R410A is correspondingly higher regardless of latent heat benefits. Without energy recovery NO single sided adiabatic compression process can compete with Convergent Refrigerating Air Flows.

Figure 10: COP vs. Pressure Ratio



The high pressure ratios required by new refrigerants are easily replaced by Convergent Refrigerating Air Flows. Likewise, the high pressure ratios of closed loop air cycle refrigeration can be replaced. Once it is recognized that the entire mass flow of air required for refrigeration is passed consistently on both sides of the vapor compression refrigeration loop, there is no need for the refrigeration loop. Fans and blowers already move the needed mass flow of air on both sides of the present refrigeration paradigm. They do it expensively, without energy recovery. Not only is the air that's already being moved sufficient to refrigerate the air that's needed, it can be moved for a lower cost and refrigerated at the same time within Refrigerating Air Flows.

When the work of expansion is subtracted from the work of compression, the COP as traced in Figure 10 shows exponentially better performance as pressure ratios are reduced, accelerating exponentially after the knee at PR 1.5. The net work is radically reduced by recovering all the pressurization work as the gas is returned to starting pressure. The relationship between heat flows and net work increases toward infinity as pressure ratios drop.

As a mnemonic device it is convenient to anchor the resulting COP defined as shown in Figure 10 with this convenient "20:20:20" relationship. A COP near 20 results from a 20% pressure ratio delivering about a 20°C temperature change.

Compared to the COP of 3.93 NIST reported (above) for cooling only an 8.3°C (15°F) refrigeration task at the 95°F Rating Point, a compelling case can be made that Convergent Refrigeration will take us into an entirely new order of energy efficiency.

The Price of Mass Air Flow

Recognizing that well over 30% of conventional air conditioning energy goes to moving the air through heat exchangers, it is attractive to consider the replacement of fans with pumps. Fans and blowers are notoriously inefficient, commonly wasting as much as 85% of applied energy. These losses result

primarily from the wasteful way that fans and blowers attempt to propel air into the resistance of a static column of air. The “Fan Replacement” patent application (US 2014/0053558 A1) identifies the opportunity to reclaim 40% of the heat energy exchanged while moving air with well-established commercial pumps proven to deliver efficiency above 95% at needed pressure ratios.

The drawings shown here as Figure 11 and Figure 12 were used to document patent US8424284 using compression to raise or lower the temperature on one side of a heat exchanger. Increasing this air flow temperature caused heat to be rejected into the target environment. Reducing the air flow temperature induces the flow of heat from the target environment and into the air flow itself. The target environment can obviously be the heat exchanger of a conventional refrigerant loop as shown in Figure 12.

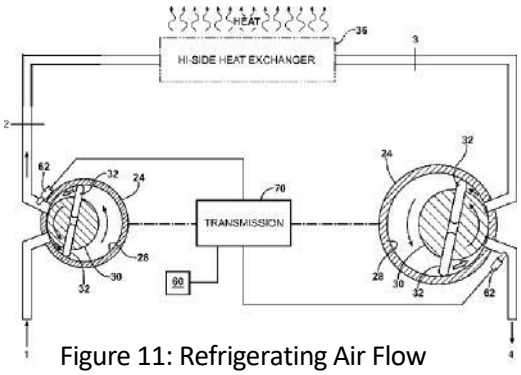


Figure 11: Refrigerating Air Flow

But the Fan Replacement pumps may be used for more than merely capturing the work otherwise wasted in heat transfers. In order to refrigerate the air flows, the pumps may be used to raise the air pressure, correspondingly increasing the approach temperature. Raising the air flow temperature in this manner enables rejection of heat into a heat exchanger. If the goal is to acquire heat rather than to reject heat, the pumps may reduce the pressure and temperature between the pumps in order to extract heat from a heat exchanger for delivery to the target environment. When these same pumps are used to deliberately change the pressure, hence temperature, of the enclosed air space between the Fan Replacement pumps, Refrigerating Air Flow technology becomes powerfully transformative for whatever technology exists on the other side of the heat exchanger.

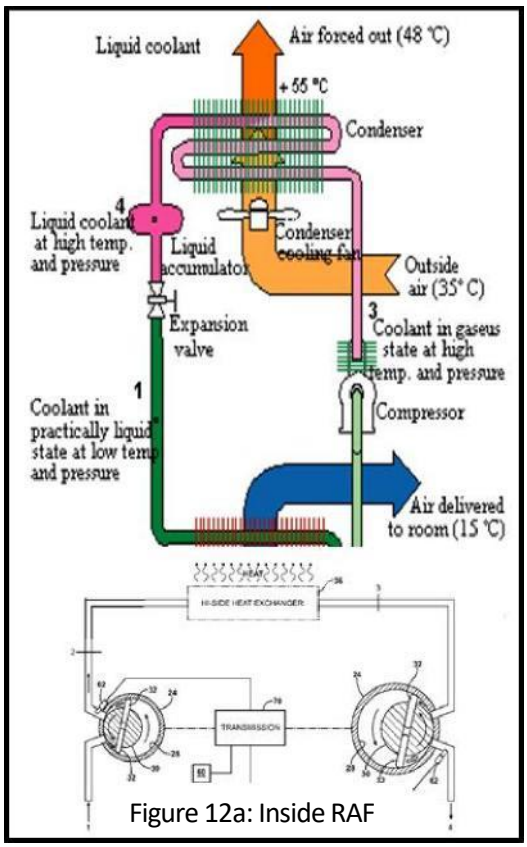
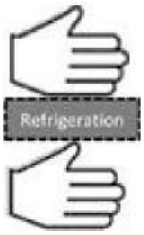


Figure 12a: Inside RAF

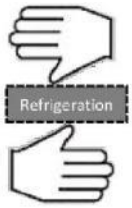
Extending this mechanism to Refrigerating Air Flows with the deliberate increase or decrease of pressure in the heat exchanger plenum between the pumps will significantly enhance the operation of vapor compression technology itself, even taking over entirely the task of increasing the approach temperature. Using Refrigerating Air Flow technology on the air side of any heat exchanger is manifestly more energy-effective for increasing approach temperatures than the cost of compressing much higher refrigerant pressure ratios as needed to deliver the corresponding excess refrigerant lift. And the Refrigerating Air Flow may well deliver the same mass flow as a fan at a lower cost.



When implemented on only one side of the vapor compression loop, 1-Sided Refrigerating Air Flow technology might be used, for example, to increase the approach temperature to

the evaporator of a vapor compression system. In this icon representing 1-sided Fan Augmentation the thumb pointing in toward the refrigeration task itself indicates that the excess refrigerant lift has been reduced or eliminated.

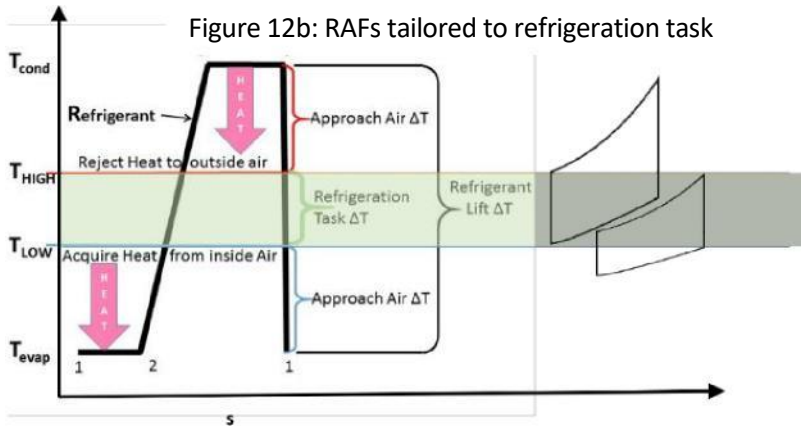
ASHRAE has raised the "room temperature" from 23°C to 27°C. This allows the increase of evaporator temperature from 3°C to 7°C while maintaining the desired approach of 20°C. This artifice increases human discomfort while allowing the manufacturers to claim substantial improvements in performance, simply reducing excess refrigerant lift. But the "1-Sided Refrigerating Air Flow" device pictured on the evaporator side of the refrigerant loop in Figure 12a can easily raise the approach air temperature by 10°C without raising room temperature and without increasing the cost of moving air. Refrigerating Air Flows cut excess refrigerant lift without cutting human comfort. Refrigerant lift can be cut directly by the same amount. Using proven positive displacement pumps such as the Roots Blower, whose efficiency at this pressure ratio exceeds 95%, will reduce the cost of moving the air while substantially reducing refrigeration costs on the other side of the heat exchanger.



Another Refrigerating Air Flow can be grafted onto the Condenser to deliver 2-Sided RAF, allowing refrigerant temperatures to stay within their effective range even as outside temperatures rise above 55°C. Not only can the costs of running the vapor compression loop itself be cut by half or more, the raw cost of moving the air alone should be substantially reduced. At the pressure ratios needed, the market already offers many proven commercial devices such as the Roots Blower capable of moving mass flows in the 1000-4000 SCFM range for a small fraction of the energy consumed by an equivalent fan.

Figure 12b depicts the overlapping temperature arrangement of two Refrigerating Air Flows centered in the green zone. Such an arrangement can replace the vapor compression loop and any analogous closed air cycle refrigeration loop. In the green zone, two temperature controlled Refrigerating Air Flows provide the offsetting temperatures needed to transfer heat in either direction using any air to air heat exchanger. In refrigeration mode the unwanted heat is simply expelled outside while the cooled air is released into the target environment. The engineering specifications of an HPT heat exchanger will be used in the following embodiments to illustrate the behavior of Refrigerating Air Flows at temperatures certified by HPT commercial parameters and advertised performance. Because every temperature change is working in the direction of the goal rather than away from it, Convergent Refrigeration inherently reduces the needed refrigerant lift. COPs well into double digits will be shown repeatedly, benefiting from the fact that a heat pipe costs nothing to run. Combined with RAFs the heat pipe eliminates vapor compression altogether, delivering a 90% reduction in air conditioning costs when compared to the acknowledged cost of operating present systems (which do not disclose the cost of moving inside air).

Figure 12b: RAFs tailored to refrigeration task

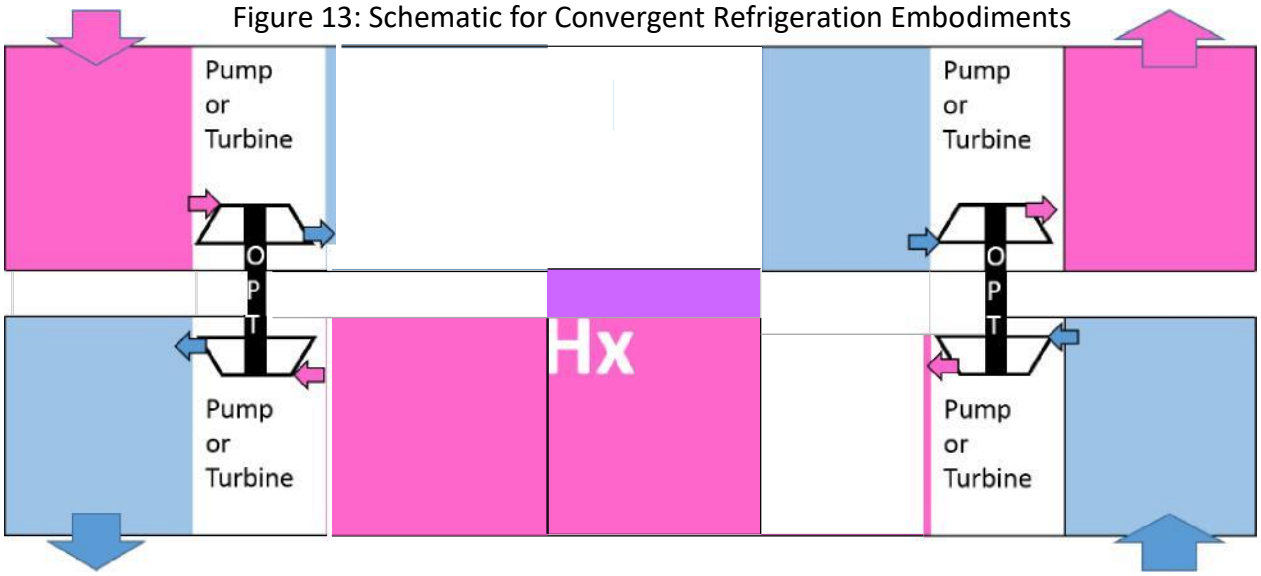


The heat pipe does use the energy of the heat to be removed to move the heat. But more relevant to its speedy adoption, the heat pipe can "plug in" exactly the present form factors and flow rates while replacing the high compression costs of present refrigerants. The heat pipe directly replaces the vapor compression loop (at zero operating cost) while Refrigerating Air Flows will deliver exactly the same mass flow of environmental air for cooling and heating at common temperatures for less than the cost of running only the fans in vapor compression alternatives. COP is increased dramatically at all temperatures. This approach invites a variety of productive innovations to be disclosed in the sequence of embodiments which follow.

Convergent Refrigeration Schematic and Discussion of General Principles

Because the physical implementation of Refrigerating Air Flows invites a wide variety of physical dimensions and engineering interpretations, the logical schematic shown in Figure 13 is presented to accommodate many canonical methods and physical possibilities.

Figure 13: Schematic for Convergent Refrigeration Embodiments



For consistency in the schematics which follow, the upper flow defines the circulation of outside air while the lower flow defines recirculation of inside air. The two flows are separated by a barrier such as a building wall. The "outside" and "inside" permanent ducting remains in place while the changeover from air conditioning to heating seasonal needs is delivered simply by changing relative pump or turbine speeds. Daytime cooling is readily complimented with heating on cold nights. For example in the summer the warmer outside air is made cooler between the pumps surrounding the heat exchanger while the cooler inside air is made warmer. Heat will naturally migrate into the outside Refrigerating Air Flow through any air to air heat exchanger, nominally a heat pipe. Depending on proximity and climate variables, the driving pumps or turbines may optionally share a common shaft. More typically, each device will be separately powered and precisely controlled using DC motor-generators.

Arrows show the direction of the Refrigerating Air Flow, counter-flow is shown here. The arrangement of any heat exchanger ducts, pipes, and fins will be engineered for best performance in counter-flow heat transfer models. There will be more than one row of pipes in the heat exchanger with optional

additions described below. Effective heat pipes can be engineered with temperature differences as small as 2°C between the source and sink. About 5°C is recognized to be optimum.

Reversing these relationships transforms the system from an air conditioner into a heat pump, moving heat from the colder outside air into the building in winter. Just as the relative pump speeds will be tuned for best efficiency as inside and outside temperature and humidity changes, the boiling point of the heat pump working fluid may be moved to the optimum temperature between Refrigerating Air Flows to follow both the size and the direction of the refrigeration task, reversing the direction of vapor and liquid flows to meet seasonal or even daily needs. Commercial air-to-air heat exchangers of this heat pipe class use typical refrigerants like R134a circulating through the same fin-and-tube heat exchangers employed by vapor compression systems. Such two phase refrigerants may even be pumped at very low cost while in the liquid phase. No longer dependent on “gravity feed” heat pipes overcome limitations of elevation and distance. The direction of flow may be reversed easily to change over from Air Conditioning to Heat Pump operation, meeting day-night and/or seasonal demands. Their boiling points may be controlled with specific pressure regulation, exactly as in Vapor Compression systems. But the crucial performance distinction for heat pipes remains that heat is acquired at a higher temperature source and rejected into a lower temperature sink. No external energy is required to compress the vapor, no more “fooling” it to condense at a higher temperature by increasing its pressure. In a heat pipe the heat always runs down hill and the vapor always does too. Latent heat always carries the load. When combined with Refrigerating Air Flows, heat pipes open up an entirely new range of refrigeration opportunities.

For illustration, the temperature values shown in examples which follow have been taken from the commercially available engineering statements of HPT. Often demonstrating greater heat flux, the heat pipe heat exchangers can deliver temperature changes often exceeding 90% of the approach compared to 60% with vapor compression. Not only will typical heat transfers be substantially higher with the same mass air flow, the total heat content will be greater because 1) the inside air flow is always “non-condensing” and 2) condensation in the outside flow will rarely occur due to consistently narrowed approach temperatures. This and other distinctions are elaborated below.

In testament to the effectiveness of heat pipes, ASHRAE concluded in its “Examination of the Role of Heat Pipes in Dedicated Outside Air Systems (DOAS)” (25 May 2012) that heat pipes provide “the most energy efficient and economical systems available, bar none!”

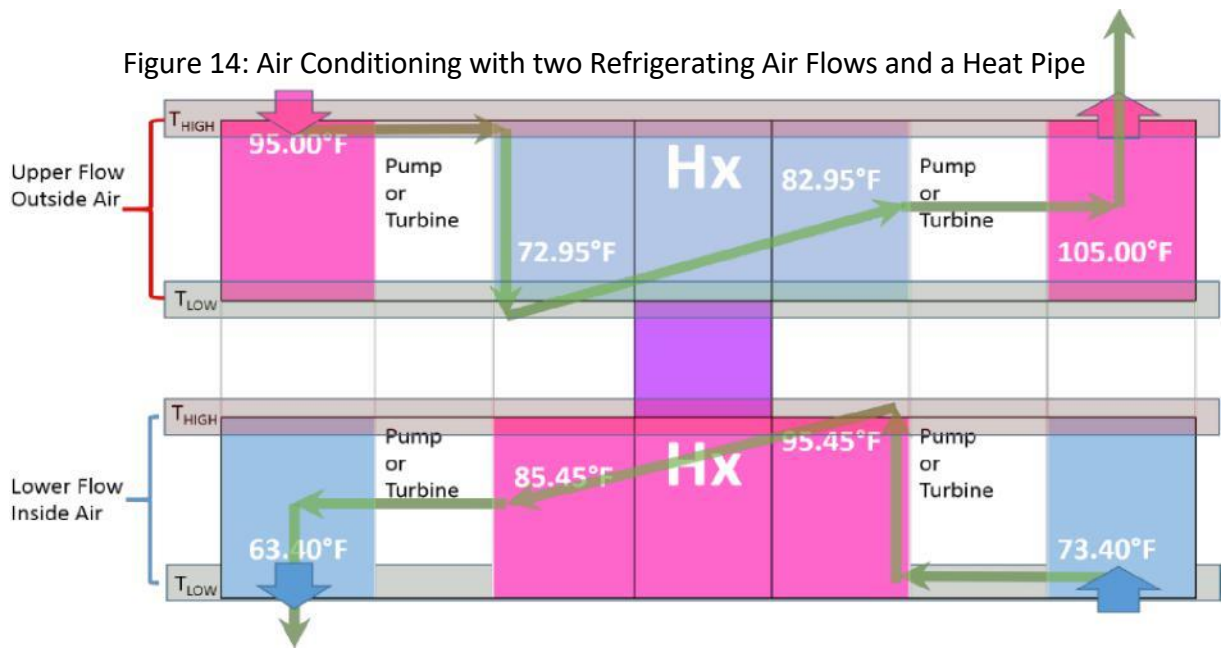
At the time of this disclosure there is no known implementation of heat pipes within Refrigerating Air Flow systems. Nor have heat pipes previously been used to displace entirely the role of vapor compression refrigeration loops between independently optimized Refrigerating Air Flows.

Simple Air Conditioning

Figure 14 provides a more complete showing of operating temperatures for air conditioning. The outside air flow is shown above the inside air flow. Temperatures have been selected to show expected relationships at the 95°F Rating Point. HPT is again the source for heat pipe performance parameters.

Figure 14 defines Refrigerating Air Flows precisely targeted to the temperatures needed to sustain heat transfer within HPT parameters while eliminating all excess refrigerant lift. All the temperature overshoot characteristic of a Brayton Cycle has been eliminated. The incoming air temperature has been selected to precisely target the exact approach air temperatures and relationships stipulated in engineering statements of HPT.

Figure 14: Air Conditioning with two Refrigerating Air Flows and a Heat Pipe



This configuration cuts 90% of the acknowledged vapor compression energy cost. Refrigerating Air Flows eliminate the vapor compression system altogether. Of course the RAF energy budget would include the previously unreported cost of moving air through the inside heat exchangers. The entire cost of refrigeration may now fall below the already incurred costs of just moving mandatory mass flows of air.

As previously shown for temperatures at and above this rating point, the only usable portion of the R410A vapor compression cycle is vapor, not latent heat. And the energy needed to raise the vapor pressure to ratios of 4 and above causes extreme temperature overshoot. Vapor compression may have benefitted from temperature overshoot by accelerating heat transfer, but temperature overshoot can be eliminated altogether by sustaining a precisely tuned approach temperature. Convergent Refrigeration may be delivered within the energy budget previously required just for moving air.

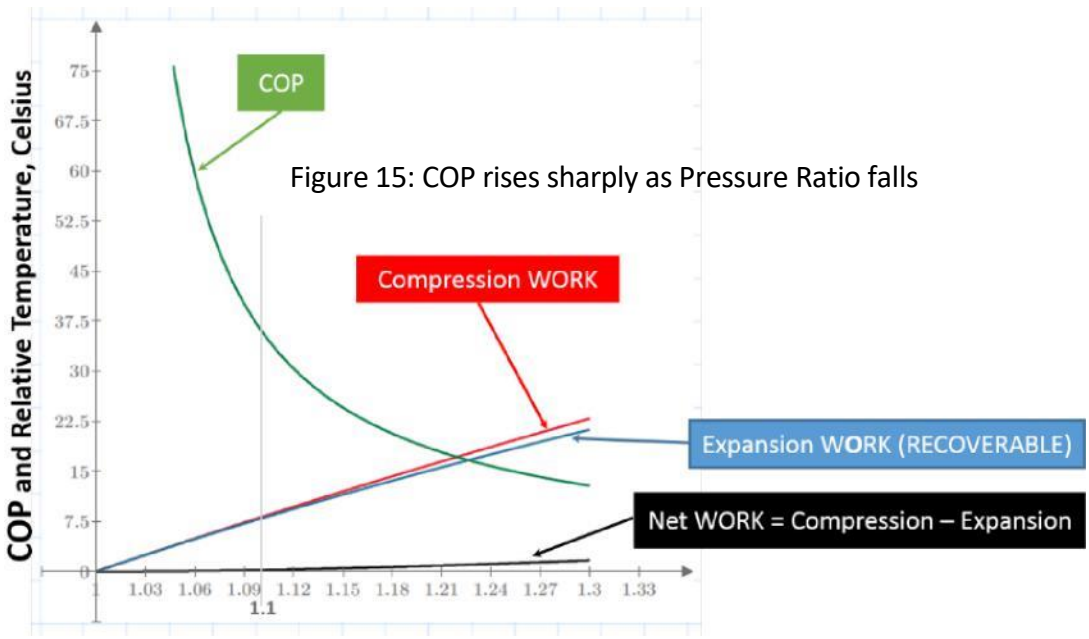


Figure 15: COP rises sharply as Pressure Ratio falls

Rather than use the 20:20:20 rule with both flows in the Simple Air Conditioning illustration and in the Simple Heat Pump below, it is easy to introduce greater precision. Both are operating at a pressure ratio of 1.15. COP is 24.17. Figure 15 shows how rapidly COP will increase at temperatures below the 95F Rating Point.

The refrigerating COP of the upper flow is mirrored by the slightly more efficient heat pump COP of the lower flow, 24.19 because of the slightly lower operating temperatures. The combined COP for moving the heat out of the lower flow and out of the building is COP =12.33.

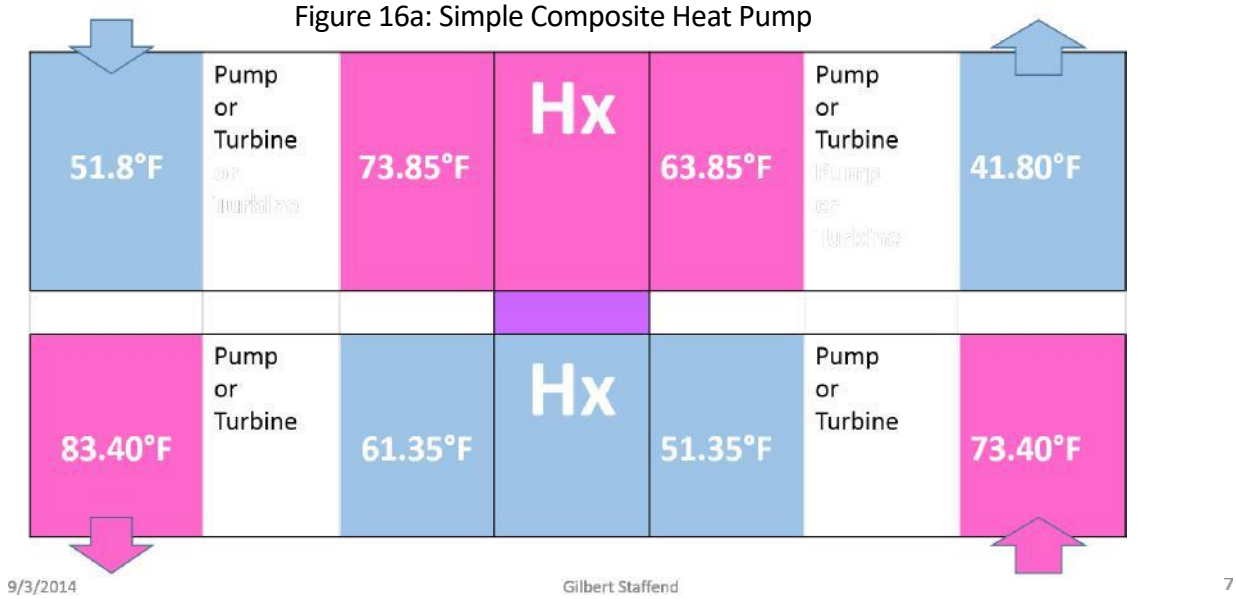
As previously stated, the temperature relationships are chosen purposefully to duplicate published data from HPT. The Refrigerating Air Flows here follow behaviors incidental to the choices made by HPT rather than the optimized values readily preferred in a working system. The stated values have also been validated computationally. These HPT numbers provide commercial certification of temperature relationships and deliverable technology unarguably capable of displacing vapor compression with Refrigerating Air Flows. Their form factors and track record provide a plug and play replacement for vapor compression heat exchangers used around the world.

Simple Heat Pump

As stated above, half of each system operates in heat pump mode while its partner operates in refrigeration mode. The first heat pump example is already shown in Figure 14 where the inside (lower) RAF is rejecting heat it moves heat into the heat pipes, operating as a heat pump rejecting heat into the lower temperature upper air flow. Its partner, the outside (upper) RAF is accepting heat from the heat pipes so it is operating in refrigeration mode. The following example is provided to show how the composite pair of Refrigerating Air Flows act together to provide the building with heat from the outside when the outside temperatures fall below the desired inside temperature. The composite operation provides a heat pump over all.

Figure 16a shows the operating temperatures under heat pump operating conditions. The outside air flow is again above the inside air flow. The temperatures selected are symmetrical with respect to Figure 14, the heat pump duplicates the same relationships but with the heat now flowing downward into the cooler lower Refrigerating Air Flow rather than upward from the lower flow. The "outside temperature" is now 21.6°F below the "inside" target temperature of 73.4°F (23°C) as it was 21.6°F above the "inside" target temperature at the 95°F Rating Point shown in Figure 14.

Figure 16a: Simple Composite Heat Pump



The same efficiencies are present here with combined COP better than 12.33 because of lower operating temperatures over all.

Heating and cooling can be delivered by Convergent Refrigeration at the same price previously charged just for blowing air. It is tempting to say "free" because this cost is unreported in industry advertising for vapor compression systems. This claim is readily deliverable with Convergent Refrigeration as long as pump efficiencies remain in the ballpark of 90%. Roots Blowers already assure commercial success.

The heat demands of very cold temperatures have been addressed and satisfied by US 8596068. Figure 16b shows the presence of an auxiliary heat source, optionally a fuel burning heat source. US8596068 establishes claims with respect to augmenting the heat pump function for effective service in extremely cold temperatures. The heat pump configuration for Convergent Refrigeration will be further outlined at Figure 16, below, now extended in this disclosure.

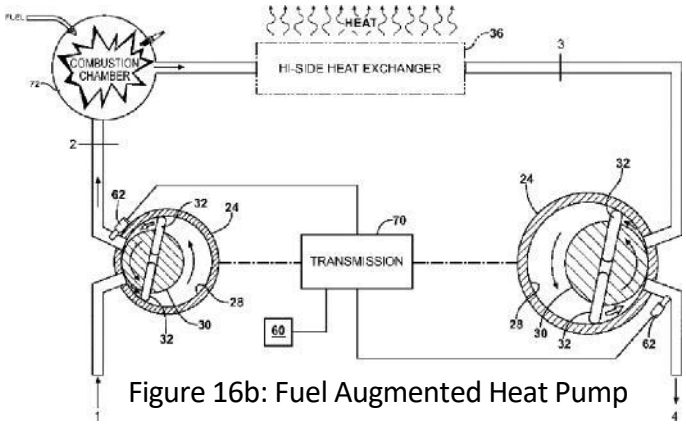
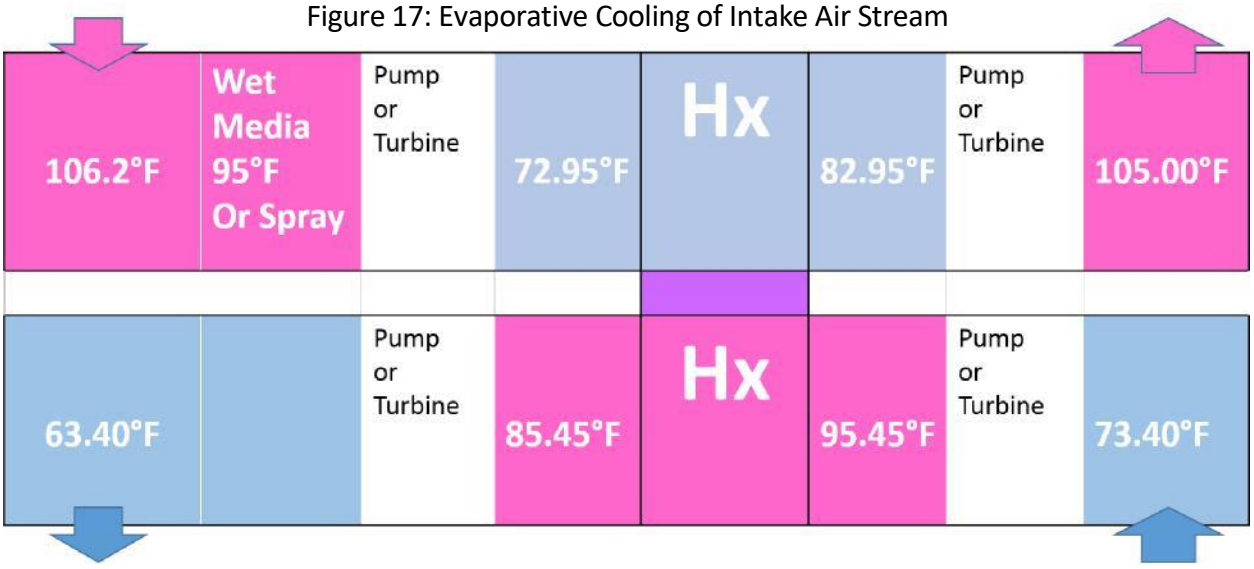


Figure 16b: Fuel Augmented Heat Pump

Add Evaporative Cooling with Water

Figure 17 shows how the addition of evaporative water cooling ahead of the first outside pump will add another 11.2°F to the capability of cooling without changing Refrigerating Air Flow energy performance so long as the mass air flow between the pumps remains non-condensing. HPT certified data is used here again for the measures of evaporative cooling as well. The increment of improvement naturally depends on relative humidity. The essential relationship is determined by the heat exchanger target temperature. As long as the incoming temperature-humidity combination maintains a dew point above the heat exchanger target temperature (72.95°F with a wet bulb temperature roughly 84.5°F), it will be non-condensing. This showing of the performance gain from including evaporative water cooling duplicates the published HPT data. HPT data is used to validate and incorporate the viability of HPT products within this disclosure of Convergent Refrigeration. Use of published HPT data is not meant to suggest any optimization within Refrigerating Air Flows. At outside temperatures below 106.2°F, the introduction of evaporative water cooling into the outside air stream can take the outside Refrigerating Air Flow well below the 10% pressure ratio where COPs well above 30 are readily apparent.

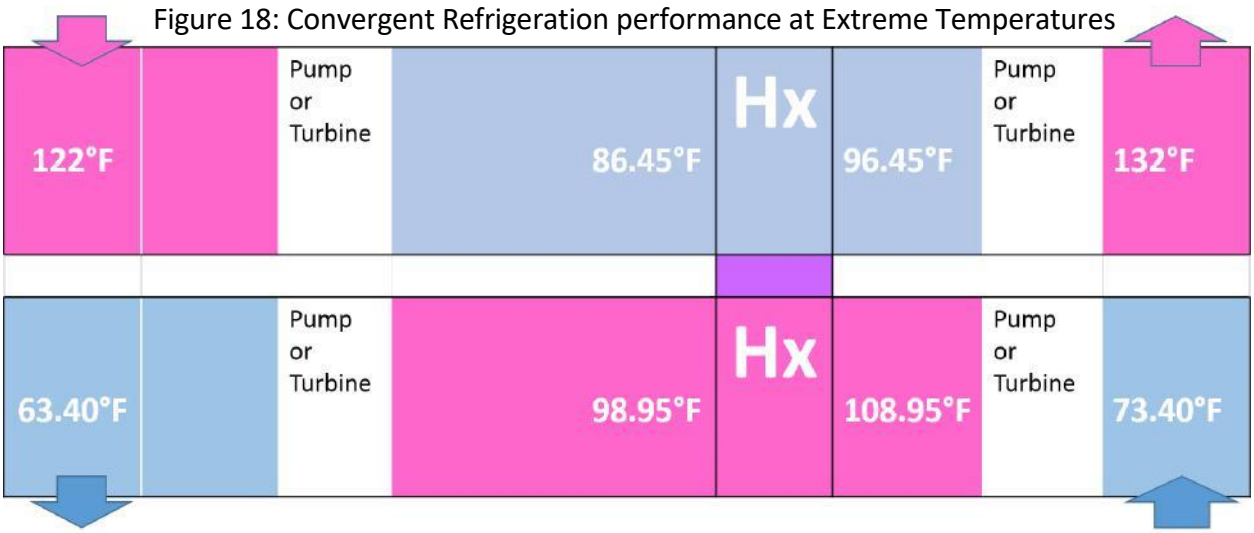
Figure 17: Evaporative Cooling of Intake Air Stream



WORST CASE SCENARIO: Desert Cooling

Figure 18 explores what it takes to cool temperatures of the Saudi Arabian desert to the older cooler room temperature of 23°C (73.4°F). This cooler temperature is reportedly preferred in those portions of the oil rich Arabian Peninsula among those wealthy enough to afford air conditioning. Recalling that this temperature was enjoyed more or less globally before ASHRAE's alteration of the testing standard to create the appearance of improved technical performance without improving the technology or mechanical capabilities even slightly, we might want to deliver the same level of comfort still sought by many who prefer the older cooler room temperature. The depiction below preserves exactly the same HPT operating temperature differences respected in all other scenarios relied on in this disclosure.

Figure 18: Convergent Refrigeration performance at Extreme Temperatures



Both air flows are refrigerated by the same temperature change, 35.55°F=19.75°C, somewhat less than needed to fit the 20:20:20 rule. It is noteworthy that refrigeration can be delivered under these extreme circumstances by increasing the pressure ratio to only 1.25 from the 1.15 needed at the 95°F Rating point described in Figure 14. In other words, Convergent Refrigeration can deliver the same comfort level under desert conditions with only a negligible increase in energy expense. Both RAFs correspondingly deliver COPs of 15 with the total system COP of 7.84 at these elevated temperatures. NIST reports a COP near 2 for both R410A and R22 at the same outside temperature while allowing the inside temperature of 80°F.

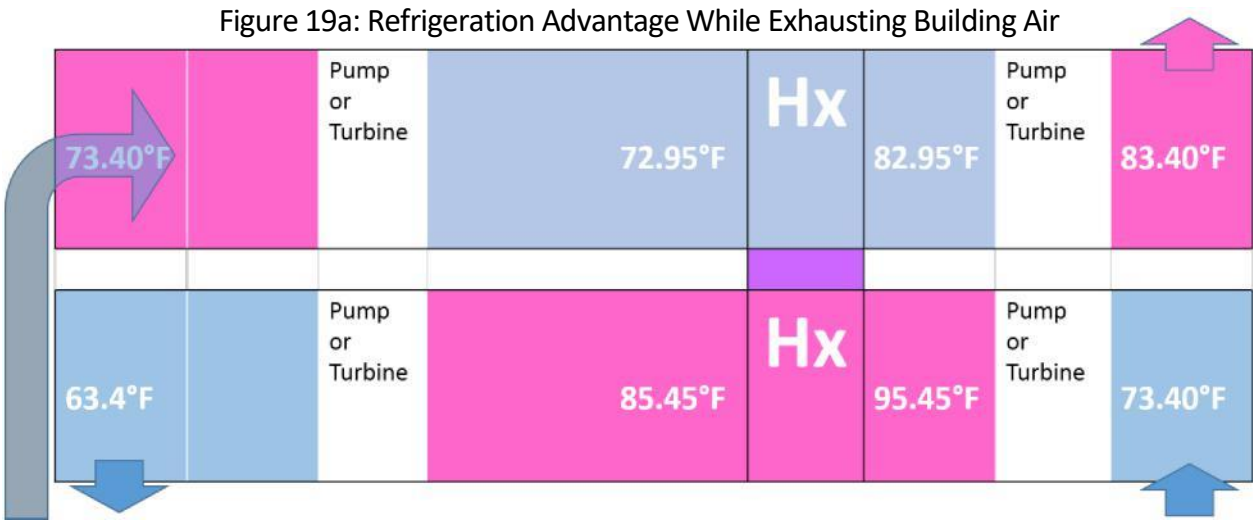
Dehumidification, Make-up Air, and Exhaust

Performance will be increased by provisions for dehumidification, make-up air, and exhaust when compared to the standard operating mode of Convergent Refrigeration. Three new capabilities detailed below far exceed the best possibilities of vapor compression alternatives.

1. Exhaust cooler inside air through outside Refrigerating Air Flow

In Figure 19a the inside air is simply exhausted. Only negligible work, if any, is needed to meet the target heat pipe temperature in the upper flow, less than half a degree Fahrenheit. COP in the lower flow will remain as it was at 25.19 indicating a total system COP at that level.

Figure 19a: Refrigeration Advantage While Exhausting Building Air



2. Dehumidify building air in the upper RAF and return through the lower flow.

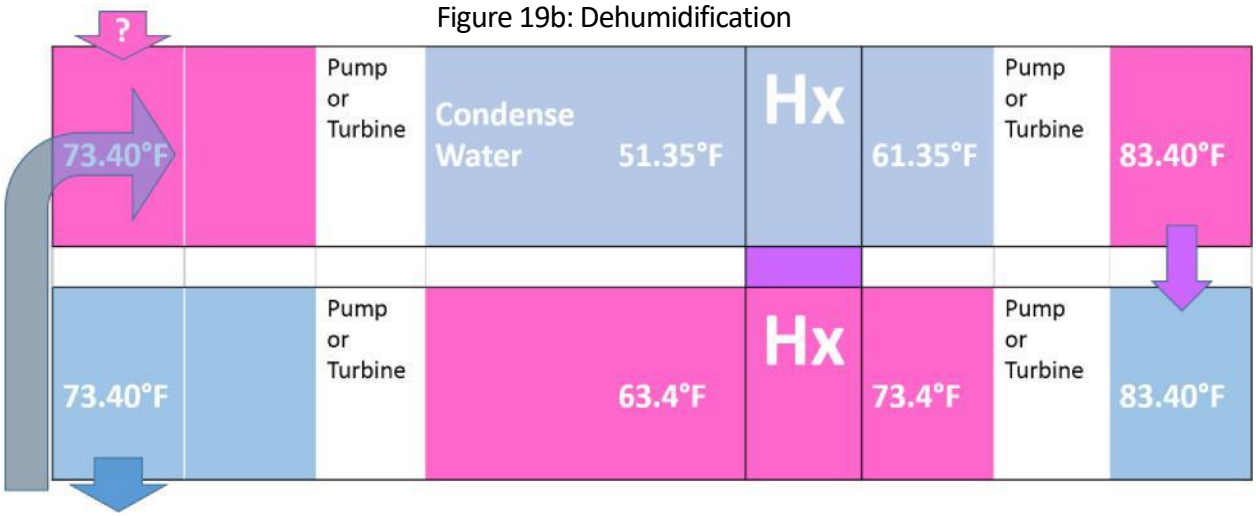
In Figure 19b the entire mass of building air is initially fed through the upper RAF. In this case the upper flow exit feeds directly into the lower flow. NOTE: choice of the upper flow path as primary for dehumidification is merely suggestive that only one path need be equipped to deal with water; evaporative cooling, and condensation. Other arrangements will be chosen depending on climate and the physical routing of ducts, their intake locations and their exhaust locations.

The process for providing and dehumidifying make up air is understood and adequately documented in the engineering of warp-around heat pipes by HPT. Although it is not detailed here, the effusive endorsement of ASHRAE was previously noted. The anticipated blending of outside makeup air to be dehumidified will increase energy use.

The heat exchanger target temperature of 51.35°F is below the best evaporator inlet temperatures recorded by NIST in the Domanski and Payne (2002) study previously mentioned. Clearly this target temperature meets the ASHRAE specifications for testing at the 95°F Rating Point.

Cooling work must be done in the upper path sufficient to assure that the target temperature chosen for the desired exit humidity level has been met. Because no external heat rejection occurs in the process as drawn, heat will accumulate from the latent heat of condensation.

Figure 19b: Dehumidification



Summary and Conclusions

Convergent Refrigeration provides an entirely new set of mechanisms and methods for minimizing heat transfer in refrigeration, delivering unprecedented high COPs with unprecedented low pressure air cycle refrigeration. In both cooling and heating (heat pumps), Convergent Refrigeration replaces the energy intensive and environmentally harmful vapor compression technology of the 20th Century.

By incorporating proven passive heat pipe technology, Convergent Refrigeration makes use of the same mass flow of air required by vapor compression technology. Of the most profound importance to certify the feasibility of Refrigerating Air Flows, vapor compression systems rely on the exactly the same mass air flow. The necessary heat capacity of circulated air has been demonstrated by vapor compression systems to be adequate mass flow to hold and move requisite heat to and from the same source and sink. Convergent Refrigeration simply uses the same mass flow of air as its refrigerant.

This disclosure has reviewed a preliminary sampling of new technologies enabled by Convergent Refrigeration, demonstrating the strength of this platform for further innovation.