TURBINE AIR SYSTEMS

Position Paper

on

Analyzing the Potential for Condensate Carry-over from an Inlet Cooling Coil

April 2003

Executive Summary

Turbine Air Systems has provided chilling systems on over 100 gas turbines. Several installations are more than 15 years old. It has been our experience that carryover of water droplets into the ductwork, and hence into the bellmouth, has not been demonstrated by long-term operational experience.

Most inlet cooling projects with which we have been associated have had filter houses and cooling coils provided by the GT OEM. Nonetheless, the designs of those systems were heavily influenced by TAS, by our predecessor organization as the Trane Industrial Division, and by our long-term coil supplier, IHT. Where we have provided new and retrofit coil installations, we have practiced the most conservative design standards of which we are aware, and in the process have become the industry leader.

In this paper, we quantify the methodology for calculating the air and water conditions for a chiller coil installation. For comparison purposes, we provide a similar calculation for a fogging system. We show that the carry-over due to fogging over-spray is several orders of magnitude greater than from a coil with drift eliminator.

We have taken a very conservative approach with this analysis, showing the worst-case assumptions, even though empirical data strongly supports otherwise. Our calculations show that a conservative coil design yields a manageable amount of condensate carry-over off the back face of the coil, which in turn is almost totally mitigated by the use of a Drift Eliminator. By applying several layers of conservatism to our calculations, we can demonstrate a high estimate of approximately 0.04 gpm of water carryover to the turbine, mostly of very small aerosol droplets.

Turnkey installation by TAS of a set of chiller coils in a GE 7FA / Donaldson filter house in 2002.

We also stand on the actual experience of 15 years of inlet chilling, including 4 years on a multiple large frame machine Thermal Energy Storage (TES) installation in which a TAS-designed retrofit has successfully operated without any discernable impact on the compressor blades.

To quote the facility manager from this plant: "From outage to outage since the installation of the chiller system four years ago, there has been no apparent wear of the gas turbine compressor blades that could be attributed to the chiller system."

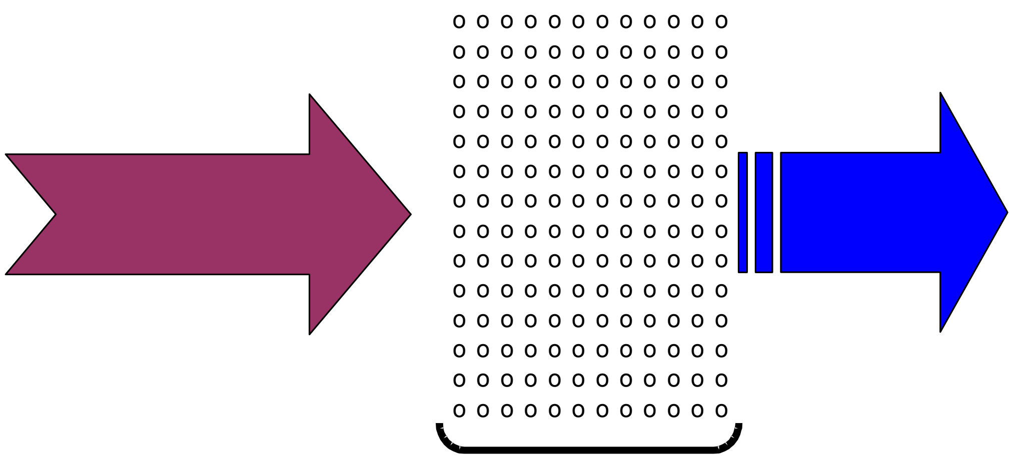
Our conclusion will show what our customers have realized for more than a decade: that the decision to add chiller coils is reasonable and prudent.

Psychrometrics

Occasionally, the question is asked concerning the possibility of water entrainment into the compressor of a gas turbine downstream of a set of chilling coils. In order to describe the concept of carry-over, it is best to start with the relationship of water and air in a coil system.

Psychrometrics is the study of the physical and thermodynamic properties of air-water vapor mixtures. The interaction of air and water with a chilling coil is well known, and relatively easy to understand.

IDEAL Chiller Coil Model



Temperature, ºF	95.0		50.0	
Wet Bulb, ºF	79.0		49.8	
RH	49.9%		98.7%	
Dew Point, ºF	73.4		49.6	
Specific Humidity, grains *	124.9		53.0	
Mass Flow, Dry Air, pph	3,553,068		3,553,068	
Mass Flow, Water Vapor, pph	63,622		27,111	-57%
Mass Flow, Condensate, pph		36,512		
Mass Flow, Total, pph	3,616,690		3,580,179	
Enthalpy, BTU / Hour	153,675,482	659,399	72,326,184	
Cooling Duty, BTU per Hour; tons			80,689,899	6,724

Figure 1, Coil Mass and Energy Balance

* *Grains of moisture per pound of dry air*

As the hot, humid inlet air is chilled below its dew point, it loses its capacity to retain water. Liquid water droplets condense on the colder surfaces of the copper tubes and aluminum fins. Most of the water will be drained into the coil drain pan, where it will be piped to a lower drain sump at the bottom of the filter house.

The above model and the data in this paper assume a T2 of 50ºF. The system is capable of lower temperatures, specifically 45ºF. At lower temperatures, the dry air mass flow will be increased, and the percentage of water removal will be even greater.

Understanding Humidity

Typically, it is assumed that the air leaving the coil has a Relative Humidity (RH) of near 100%. To the person that does not deal with psychrometrics on a frequent basis, it would be understandable to consider that the air is quite wet, unstable, and likely to cause further condensation elsewhere in the system.

In fact, while the *Relative* Humidity of the air approaches 100%, the *Specific* Humidity of the air decreases dramatically. Explained another way, the cold air has much less capability of carrying water vapor, as compared to the original hot air. Nonetheless, the amount of water vapor in the air has decreased by more than half at typical design conditions when put through a chiller coil. The chiller coil 'wrings' water out of the air.

Specific Humidity is measured in *grains* of water per pound of air. A grain is 1/7000 of a pound. Grain is an obscure measurement unit, not typically used elsewhere in gas turbine engineering calculations. Yet, in order to perform virtually any meaningful calculation in psychrometrics, it is first necessary to calculate the specific humidity of the air.

In the diagram on the previous page, a typical coil calculation is shown. The total volumetric and mass flow of the air-vapor mixture downstream of the coil is known, because it is "demanded" by the GT bellmouth. If the water vapor content of that air can be calculated, then the dry air component is easily calculated. Knowing that dry air mass flow is constant across the coil, and further knowing the specific humidity of the air upstream of the coil, it is easy to construct a mass and heat balance to determine the amount of coil condensate.

In the case shown, the air flow is of a typical PG7241FA, using the OEM values for air flow at the compressor inlet, given the air temperature (T2 = 50ºF) and the off-coil RH (~98.7%). The incoming hot air is carrying 63,622 pounds per hour of water vapor. However, the cold air (T2) coming off the coil is only carrying 27,111 pounds per hour of water vapor. The reduction in water vapor is approximately 57%. The difference in water vapor is the water that condenses on the coil surfaces. The condensate per each gas turbine is 36,512 pounds per hour, or approximately 73 gpm.

In real terms, more than half of the water vapor is removed from the air stream by the coil, at the coil. This is despite an increase in the *Relative* Humidity of the air.

In this paper, we will take pains to use the terms water *vapor* versus water *droplets*. Water vapor is a gas, and is benign. It is accounting for the size and quantity of water droplets that concerns us.

The RH of the air off the coil is typically less than 100%. This is because chilling, like most other mechanical tasks, is imperfect. In order for an air molecule to give up heat, it must collide with another molecule of less energy. So heat is transferred from energetic air molecules to less energetic air molecules, but mostly to the low energy metallic components of the coil. Nonetheless, a certain amount of air molecules can get through a coil with little or no interaction with the coil surfaces. This is called *bypass*.

While the final goal is to determine where most of the condensate goes, and how much, if any, gets all the way downstream to the bellmouth, we next need to look at the design of the filter house.

Sizing the Filter House

In order for condensate to get into the gas turbine, the cold air must carry it downstream. In practice, we know that the qualitative value of such carry over is low, due to evidence of the dry air ducts downstream of the coils in virtually all of the 100 gas turbine projects in which TAS has completed. The challenge here is to build a case for a conservative set of calculations that could describe the potential for free water carryover.

In order for water to be carried by the air, it must be "entrained". Very fine water droplets, such as those typically produced by a fogging system, 10 to 50 microns, may be carried for long distances, or even indefinitely, because of the interaction of the drag of turbulent air, as opposed to the natural force of gravity that would want to pull a water droplet in a downward direction. As compared to fog, water droplets of a coil tend to be much larger, more in the range of 87 to 131 microns (+/- two standard deviations).

In the area of the filter house, a wide cross section is allowed for air passage. In fact, the cross section for a filter house that holds coils will be significantly larger than for a standard filter house for the same turbine. In our practice, where the GT OEM has not already determined the coil size, we have historically advocated a large face area, for conservatism. Face velocity is the cubic feet of air (volumetric flow) divided by the net square footage of chiller coils. The cross sectional area is chosen such that the *face velocity* at the coil is less than 600 fpm, and where possible, around 500 fpm. *[New designs are currently being explored that would allow velocities as high as 700 to 800 fpm, with high velocity drift eliminators. However, those designs are not being contemplated for near-term projects.]*

When face velocity is kept low, the horizontal drag component of the air on the condensate is diminished, allowing gravity to drain the condensate away. The primary benefit of low face velocity is a significant decrease in differential pressure across the coil. Differential pressure can account for approximately ¼ % of engine power loss per additional inch, water gauge, of pressure drop. In some cases, the increased face area of the larger filter house, and greater number of filters, can significantly mitigate the overall pressure drop of the entire filter house assembly. It is not unusual to see lower overall pressure drop in the larger filter house system designed for coils. *[Additionally, even if the entire 1" (nominal design) of coil pressure drop is added to the filter house intake, this ¼ % loss of power could be easily gained back by cooling the air an additional 0.7º F.]*

Because the coil area is known, the way to calculate velocity is to first determine the volumetric airflow. The GT OEM does not typically quote volumetric flow, but it can be determined from the inlet flow on a mass basis, and converted through the *density* of the air.

In our case, the mass flow of air-vapor immediately at the face of the coil is shown on page 3 as 3,580,179 pounds per hour. The density of air at the T2 conditions is 0.0767 pounds per cubic foot; we can easily calculate that the volumetric flow is 777,900 CFM. Knowing that we have 12 coils with nominal dimensions of 63" tall and 306" wide, we have a net coil area of 1,607 square feet. Dividing volumetric flow by square area results in an exit velocity of only 484 fpm.

While the term *face velocity* is most often used for determining the relative impact of water droplet carry over, this may be a misnomer. We typically think of face as being the upstream side of the coil, but what we are more interested in is the backside of the coil, which is operating "wet", and where water droplets could break off and become entrained.

If we were to perform the same calculations at the upstream face of the coils, we would use a density of 0.0694 to result in a volumetric flow of 869,180 CFM. In other words, when the GT "demands" 777,900 CFM of cold air flow at T2, it will take 869,1800 CFM of hot air to meet that demand. The air seems to be "compressed" in the coil. The air velocity at front face of the coil is 541 fpm. The air actually slows down as it traverses the coil.

Installation of Chiller Coils in 7FA filter house

Entrainment

As water condenses on the copper tubes and galvanized aluminum plates of the coil, the surface tension of the condensate will cause the water to tend to sheet on the surface, and run down the vertically oriented plate, or "fin". Fin pitch is determined on a site-specific basis, but is generally 10 to 12 fins per inch of tube length. Velocities are not high enough between the fins in the typical design to cause a water film to form a wave where it can allow the drag of the airflow to break off a water droplet for entrainment. This is very similar to the way cooling coils in a large HVAC system are mounted and this is why water is not carried into the ducts and diffusers.

Two very important design considerations need to be highlighted here:

1. TAS designs use horizontally oriented tubes, with vertical fins. This means that there are internal drainage passages between each pair of adjacent fins. This causes water to drain to the bottom of the coil. There are competing designs in which the tubes run vertically, and the fins horizontally. Such designs trap the condensate, causing it all to flow to the end of the plate, where it can "sheet" downward. In such a design, the air from lower portions of the coil will break-up the sheets, and cause significant formation of water droplets. Vertically oriented tube designs can be installed more easily, with a single header on the bottom or top of the filter house. Again, TAS has never employed this design when coils were in our scope of supply. Nonetheless, Most LM6000 designs are of this type. Despite this, they still don't seem to have a carry-over problem because they employ a mist eliminator.

2. TAS designs use intermediate drip pans under each coil. When the condensate runs down the fins to the bottom of the coil, there is an intermediate drip pan waiting to collect the water. For a 6-high set of coils, this provides five intermediate locations for condensate collection. The drip pans are plumbed by PVC piping down to the lower drain sump. Again, the purpose here is to not allow the condensate from an upper coil to be dumped on top of the lower coils. Therefore we do not allow sheets of water to drain off the back of the coil, where they can be broken up and entrained by the airflow.

Entrainment can be expected to occur only on the trailing edge of a vertical fin. Here, the horizontal drag component of the air can develop enough force (when the velocities are high enough) on the water to overcome the surface tension and pull a droplet away from the film, to be carried away by the air. Testing by our coil supplier[1] shows that conservative design, with respect to air velocity, will result in the lowest amount of entrainment.

Face Velocity	Uncoated Aluminum fin	Zinc Coated fin
240 - 246 fpm	None Observed	None Observed
440 - 445 fpm	Moderate	Light
484 fpm - TAS		
590 - 610 fpm	Heavy	Moderate
800 >	Heavy	Heavy

During testing, it was estimated that "Moderate" entrainment was equal to 2% to 5% of total water weight of condensate. Our design conditions of 484 fpm, with the use of a galvanized fin, would place

[1] IHT Wind Tunnel Test Data, June 1997

us at less than the "Moderate" conditions. For conservatism sake, we will use the higher end of the range, that is, 5% of all condensate.

To place this in perspective, we can safely assume that in a 12-row coil, all of the condensate produced in the first 11 rows will be safely deposited in the drip pans under the coils. This would mean that $1/12^{th}$, or approximately 8%, of all water is available for possible entrainment. By using the 5% figure at the upper range of the wind-tunnel observations, we are conservatively assuming that more than half of the water in the 12^{th} row will become entrained. This is enormously conservative.

Backside of Coil (looking down), showing part of drip pan. The hole in the center is for intermediate drains down to the Drain Sump

Drift Elimination

When a droplet becomes entrained in the airflow, we will need to remove it. There are two methods for removal.

1. The First method would be to provide a "drop-out zone" of stable, non-turbulent airflow, where the force of gravity will cause the droplet to fall out.

2. The Second method is by use of a Drift Eliminator to capture virtually all of the water droplets.

Given enough space, the first method could provide complete water droplet removal. However, this is not typically available in Turbine Cooling applications. Moreover, the real-world imposition of air turbulence would remove the confidence that all water would drop out quickly. Therefore, we rely on a Drift Eliminator. "Drift" in this case, refers to the air-entrained water droplets.

In typical practice, a Drift Eliminator is provided immediately downstream of the coils. The pictures below show a typical Drift Eliminator, as specified by GE as the OEM solution for chiller applications (see the GE "Provision for Coils" attachment at the end of this paper.) The farther the drift eliminator is from the coils, the more likely it is that a high percentage of the water droplets will drop out before interacting with the eliminator.

Because various designs of filter houses have differing distances between the coils and the eliminators, we will conservatively assume that ALL of the water coming off the rear of the coil is carried into the drift eliminator. As an additional conservatism, no credit is taken for dropout.

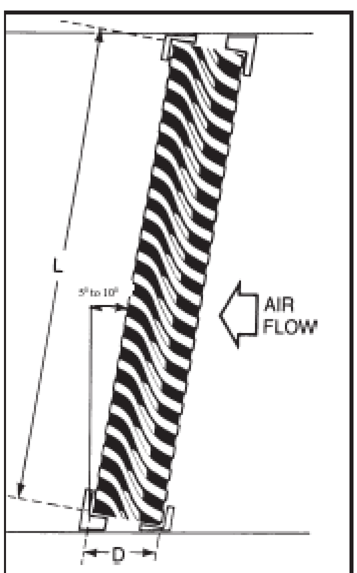


Figure 2[2]

Figure 3[3]

[2] *DRIFdek-IL, Drift Eliminators*, Munters Corporation, HumiCool Division, Product Literature MB-DE-0208, 2002
[3] Ibid

Drift Eliminator Effectiveness

The effectiveness of a drift eliminator is predicated on inertial forces that cause the water droplet to impinge on the PVC turning vanes. Therefore, the higher the velocity and the larger the droplet, the better the elimination efficiency. Munters quotes a 99% removal efficiency[4] for droplets larger than 65 microns. Calculating the carry-over that gets through the Drift Eliminator:

Carry-Over Calculation

Condensate created at the coil	36,512	pph	
Fraction of total condensate entrained in the air	5%		*
12th row condensate entrained in the air	1,826	pph	
Amount of condensate dropping out before Drift Eliminator	0%		
Condensate interacting with the Drift Eliminator	1,826	pph	
Drift eliminator effectiveness	99%		
Amount of Carry Over removed	1,807	pph	
Amount of Carry Over escaping the Drift Eliminator	18	pph	
Amount of Carry Over escaping the Drift Eliminator	0.04	gpm	

* Corresponds to the conservative 5% assumption made on page 7

The above calculations are for standard Drift Eliminators, as recommended by GE (see the GE document, "Provision for Coils" attached at the end of this paper). It may be possible to include the use of a coalescing filter, either immediately upstream or downstream of the drift eliminator. This would cause the smaller droplets to coalesce, so that they could be removed more effectively. In such a manner, perhaps more than 99% of the droplets could be removed, with particular emphasis on smaller aerosol droplets.

The potential disadvantage of a higher efficiency eliminator system is that it adds additional pressure drop to the filter house. The Munters product is expected to have a pressure drop of approximately 0.02" w.g.[5], while a higher efficiency system with coalescing filter might have a pressure drop as high as 0.2".

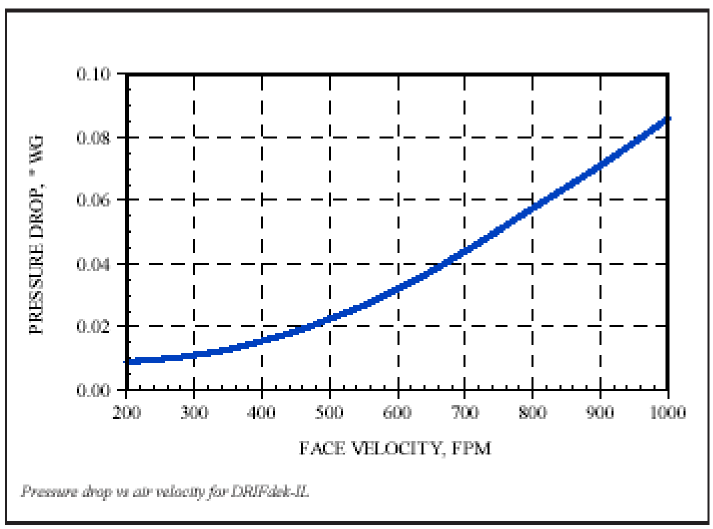


Figure 4

[4] *DRIFdek-IL, Drift Eliminators*, Munters Corporation, HumiCool Division, Product Literature MB-DE-0208, 2002
[5] Ibid

Stability of the Droplet / Vapor Equilibrium in the Ductwork

On the preceding page we demonstrated a conservative carry-over of entrained water droplets to the bellmouth of 18 pounds per hour.

On page 4 ("Understanding Humidity"), we showed that the air coming off the coil has a relative humidity of less than 100%.

If the air coming off the coils were truly saturated, then the specific humidity would be 53.6 grains per pound, not 53.0. The difference in specific humidity is 0.6. Multiplying this value times the dry air mass flow, and dividing this value by 7000 grains per pound, yields a value of 309 pounds per hour of potential water vapor that the air would hold if it were indeed saturated.

309 pounds per hour of additional vapor carrying capability compares favorably against the 18 pounds per hour of potential water entrainment.

Because the water droplets escaping the mist eliminator are likely to be on the smaller end of the size spectrum, it is plausible to postulate that a certain amount of the water droplets could re-vaporize before reaching the bellmouth.

Even if we were to ignore the potential of droplet re-vaporization, it is plausible to assume that the likelihood of additional or unaccounted-for water condensation downstream of the coil or drift eliminator is very low. Further condensation would be most likely if the air were to contact a cold surface. The ductwork sides, and the internals such as the noise baffles and trash screens, will be at a temperature slightly higher than the internal air due to conduction of outside heat through the metal structures. Therefore the air would tend to be reheated slightly which would further reduce its relative humidity and increase its ability to evaporate any moisture carryover.

Finally, there can be expected a decrease in air pressure in the converging nozzle of the bellmouth due to a rapid increase in air velocity caused by a decrease in flow area. If the air dry bulb temperature were to decrease to below the dew point of the cold air, then some additional water could be expected to condense into a cloud of vapor, most likely near the inlet guide vanes, or around the support struts.

In fact, as pressure decreases, so does the dew point of the air. This helps in negating the impact of the decreasing temperature in the bellmouth area.

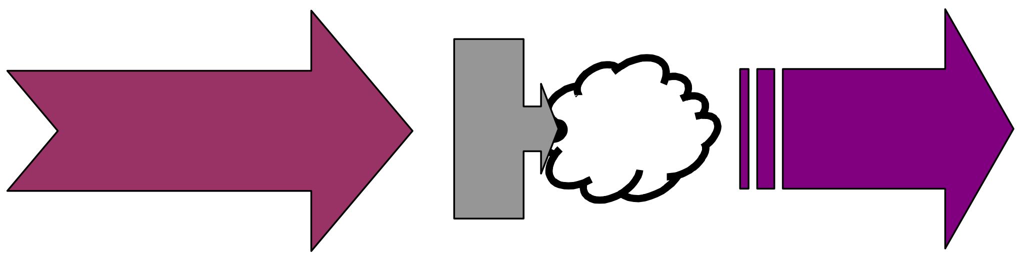
On the following page, we will discuss the over-spray of fogging droplets. In such an over-spray condition, where there is a great preponderance of droplets, it is likely that small fog droplets will contact each other several times in the turbulent air stream of the ductwork. By comparison, where the 18 pounds per hour of potential water droplets only represents 0.0005% of the total airflow, there is likely to be much less interaction between the small water droplets that can cause "agglomeration", or the spontaneous coalescing of two smaller droplets into one larger droplet.

A Fogging Comparison

In our practice, we typically model the effectiveness of a fogging system as 85%. We have felt that this value represents a reasonably fair effectiveness, to be used for a conservative economic comparison purposes against chilling systems. In fact, one of our customers has instructed us that this value might be optimistic, and that an effectiveness of 75% would be more appropriate. The psychrometric model for fogging is shown below.

This psychrometric model is predicated on a constant wet bulb (WB) temperature. The effectiveness is simply the ability to *approach* the WB from the ambient dry bulb.

IDEAL Fogging Model, 75% Effectiveness (Approach to WB)



Temperature, °F	95.0		83.0	
Wet Bulb, °F	79.0		79.0	
RH	49.9%		84.0%	
Dew Point, °F	73.4		77.7	
Specific Humidity, grains	124.9		144.8	
Mass Flow, Dry Air, pph	3,213,721		3,213,721	
Mass Flow, Water Vapor, pph	58,736		67,835	15%
Mass Flow, Fog, pph		9,098		
Mass Flow, Total, pph	3,272,457		3,281,556	
Enthalpy, BTU / Hour	139,048,792	464,568	139,813,467	
Cooling Duty, BTU / Hour			300,108	

Figure 5, Fog Mass and Energy Balance

Because of the relative inability of nearly-saturated air to absorb significantly more water vapor, the final few degrees of evaporative cooling are the most difficult to achieve. In fact, we usually limit *approach to WB* to no more than 3 degrees, no matter how effective the fog manufacturer claims their system.

In the fogger psychrometric model, instead of removing water vapor from the air (as occurs in chilling), it is necessary to spray fine water droplets into the air. In an attempt to reach higher levels of effectiveness, it has become necessary for most manufacturers to stage the fogger arrays in several locations. The current practice is to have 3 stages of nozzle arrays, only one of which is in the controlled confines of the filter house. The second and third stages of fogging, within the relatively smaller cross-sectional area of the ductwork, are ostensibly provided to cause over-spray, in the attempt to force a further approach to WB.

Over-spray is required because not all of the water droplets in fog can be expected to evaporate. Although the ideal droplet diameter is 10 microns, as much as half of the water volume (by weight) might be dispersed in a bell-curve of droplet sizes greater than 25 microns. Assuming that the likelihood of droplets greater than 25 microns to evaporate is very low, then the ideal mass flow of fog spray would need to be doubled to meet the psychrometric requirements of the specified system.

We have seen that the amount of water sprayed into a filter house / ductwork is necessarily much higher than the theoretical psychrometric model would predict. This extra amount of water aerosol will coalesce on hard surfaces, and will "agglomerate" by contacting other small droplets, forming larger droplets. This accounts for large particle carryover into the bellmouth.

We do not attempt to quantify the specific amount of over-spray required to achieve higher level of fogging effectiveness. However, we do suspect that it is quite substantial, and perhaps several orders of magnitude greater (100 to 1,000 times more) than potential carry-over from a coil-based drift eliminator. The following chart indicates the possible spectrum of over-spray conditions. The first two bars show the potential range of carry-over attributed to chilling. The next three bars show the carry-over attributed to over-spray of a fogging system, at 1%, 10%, and 100% of over-spray. The final bar at 100% over-spray indicates a system for which more than 50% of the water volume in a fogging system is in particles greater than 25 microns.

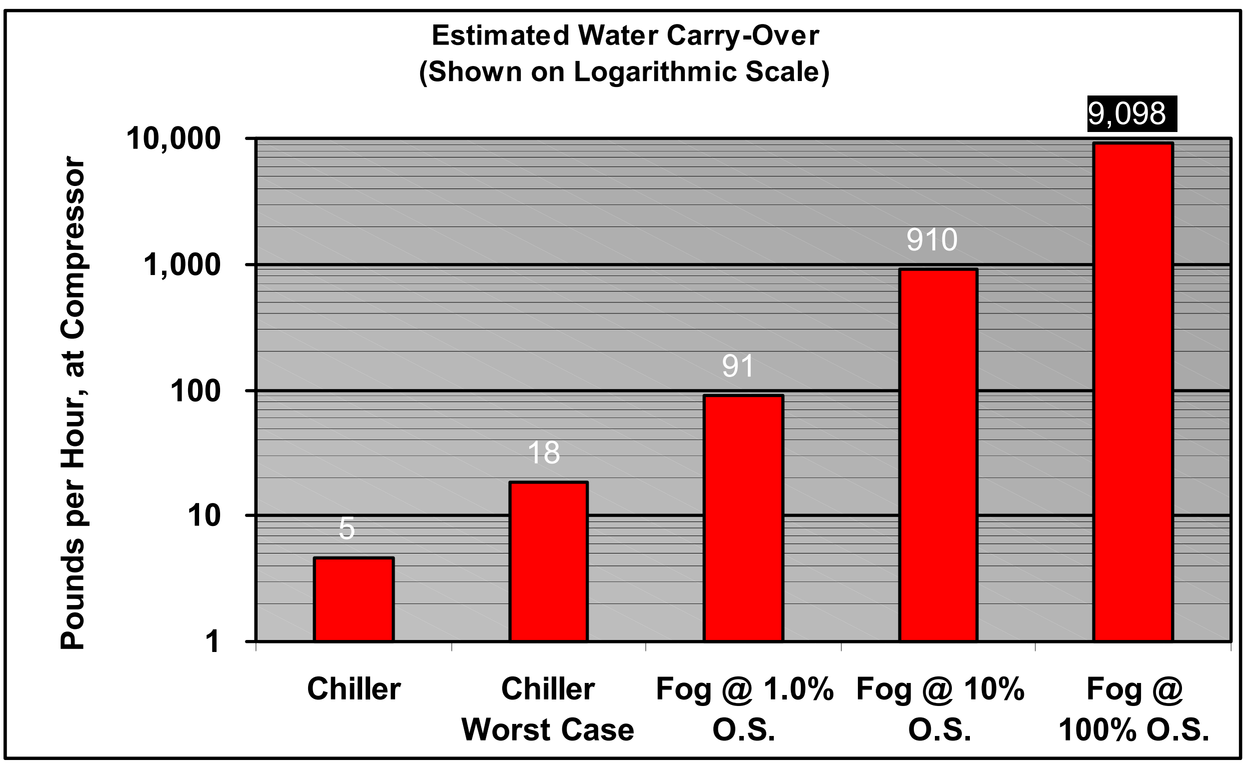


Figure 6

Additionally, we propose that any water that gets past a Drift Eliminator for a chiller coil will be on the smaller side of the size spectrum, perhaps less than 65 microns. On the other hand, the droplets that would be associated with over-spray are more likely to be on the larger end of the spectrum (because of agglomeration), approaching the size of the large stable droplet, or 110 microns.

Controlling the proper amount of water going to a fogging system can also be a problem because this flow rate is typically controlled by staging on & off pumps and nozzle stages, and is therefore a step function, not a smooth variable function. Because the ambient dry bulb temperature and RH are constantly changing, the fogger is usually limited to either lower approach (i.e. lower efficiency) or else they must risk super saturation of over-spray to maintain a tighter approach (high RH and efficiency).

Real-World Installations

The most supportive evidence is our landmark installation at the Clear Lake facility in 1999, where a complete replacement of the forward filter house was required in a turnkey retrofit with a fast-track schedule. This facility has three 120 MW W501D5A gas turbines. These turbines, while only ~75% of the power of a PG7241FA, each consume nearly as much combustion air. This facility had employed evaporative media cooling, then fogging, before finally turning to chilling as the final solution.

This plant is the largest Chill Water Thermal Energy Storage system for inlet cooling in the world.

Plant operators have described how wet the ductwork ran when the fogging system was run (the fogging system is still installed, but is very rarely operated due to the success of the chillers). When operating the chiller system, the plant operators will and have testified that the ductwork stays dry. This facility continues to set the standard for the successful and economic retrofit of chilling to large frame turbines.

> "From outage to outage since the installation of the chiller system four years ago, there has been no apparent wear of the gas turbine compressor blades that could be attributed to the chiller system."

Danny Maples, Facility Manager
Clear Lake Cogen, February 2003

6 million + gallon Thermal Energy Storage tank, at Clear Lake, TX - 1999

Conclusion

On the preceding pages, we have built a case for the development of a theoretical psychrometric model for the performance of a chiller coil. Whenever possible, we have chosen conservative assumptions for building our case. Our coil supplier has provided the results of wind tunnel observations. Each of the building blocks of this composite work is backed up by a decade-and-a-half of actual practice on more gas turbines than any other provider.

The simplistic case would be to simply describe the difference between chilling and fogging as the removal of 36,512 (63,622 incoming and 27,111 leaving) pounds per hour of water vapor for chilling, versus the addition of over 9,098 (58,736 incoming and 67, 835 leaving, theoretical with 0% over-spray) pounds per hour of vapor for fogging. However, accounting for changes in *vapor* alone does not address the real issue of *droplets*.

The numbers alone do not reflect the engineering diligence of the chiller model. Where carry-over of water droplets off the chiller coil could be plausible, despite conservative coil designs, the installation of an engineered Drift Eliminator nearly totally mitigates the threat of water carryover into the compressor. By comparison, the attempt to make fogging systems more effective has necessitated ever more aggressive designs, using significant quantities of over-spray, causing fog arrays to be installed in locations downstream of any potential droplet removal systems.

The final results of the analysis in this paper indicate that the carryover of aerosol droplets into the turbine could approach 0.04 GPM. Due to compounding conservatisms used in our calculations, it is very likely that the carryover could be less that 0.01 gpm. This is less than a gallon per hour, and less than 10 gallons total in a 16-hour operating day, assuming worst-case conditions for the entire operating period. If a deminimus level of carryover were to be proposed, we believe that we would be below that level.

We estimate that the amount of carry-over due to aggressive fogger over-spray is 100 to 1,000 times more than the potential carry-over that might be associated with chilling coils with a drift eliminator.

Both in theory and in real operational experience, we contend that the installation of a chilling coil system is an extremely low-risk operation with respect to long-term operations. The amount of water carryover from the coil and drift eliminator is almost too small to be calculated, although we have attempted to do so here. To further quantify that carryover risk is beyond even the most reasonable threshold of prudency.

To be sure, the other operational advantages of chilling, not including the obvious and immediate economic benefits, clearly advocate the decision for the installation of a chilling system.

end

The following Document is provided by GE to A/E's, EPC's, and Owners to explain the Scope of Supply for a GE filter house

GE Provisions for Chiller Coils Frame 7FA

GE Scope of Supply

GE has developed an inlet filter system that provides provisions for chiller coils supplied by the customer. The chiller coil provisions consist of and are limited to the following components.

1) Inlet Filtration System – An over sized inlet filtration system will provide a section within the clean air plenum with an average air velocity under 600 ft/min.
2) Chiller Coil Support System – A chiller coil module inside the clean air plenum will provide a stainless steel support structure. The support structure is designed to contain 14 total cooling coils. Seven coils on the right-hand side and seven coils on the left-hand side looking in the direction of airflow into the gas turbine. The coil supports are designed to mount the coils horizontally and install vertically through an opening in the roof. A removable cover is provided over the roof openings. A gasket seals the removable cover to the clean air plenum. The support system holds 4 smaller top coils, left and right, with 5 larger bottom coils, left and right. See figure 1 & 2. The different sized coils were designed to fit standard filtration systems with minimal impact on cooling capacity of each coil.
3) Loads and Pipe support points – The support steel designed for the inlet system is designed to meet a maximum load condition as specified by GE. The loads and support points for these loads are as defined in the GE customer documentation for each requisition.
4) Chiller Coil Drift Eliminators – A set of drift eliminators is supplied in a support structure downstream of the chiller coils to prevent the visible water droplets condensed out of the air by the cooling coils from entering the gas turbine. GE supplies a drain system for the condensed water collected by the drift eliminators. See figure 3.
5) Central Drain Sump – GE supplies a central water sump pan(s) for collection of condensed water off of the cooling coils.
6) Central Drains - The central drain sump pan(s) are piped to drain purchaser connections IE72-1, 2. The drain purchaser connection IE72-1 is located on the left-hand side of the unit looking in the direction of airflow. The drain purchaser connection IE72-2 is located on the right-hand side of the unit looking in the direction of airflow into the gas turbine. Two drain lines are provided for a completely redundant drain system and provide space for the generator equipment under the inlet filter house.
7) Maintenance Access – Maintenance access modules upstream and downstream of the chiller coil module are provided. The maintenance access modules each have a bolted, hinged access hatch. The access hatches are a minimum of 22 inches wide by 66 inches high. Maintenance is accomplished using temporary planking. A single maintenance platform at the bottom filter house is provided.

Customer Scope of Supply

The customer's scope of supply consists of and is not limited to the following for provisions for chiller coils provided by GE for a frame 7FA gas turbine.

1) Chiller Coils – The customer is responsible for designing and supplying the appropriate chiller coil system. The GE provisions for chiller coils supply a module capable of holding 14 total coils, 7 coils on the right-hand side of the unit and 7 coils on the left-hand side of the unit, looking in the direction of airflow into the gas turbine. Maximum size and chiller coil weight must be within the GE standard as communicated in the GE documentation for the specific requisition.
2) Chiller Coil Flashing – The customer is responsible for the flashing between coil sets to prevent by-pass of the air around the cooling coils.
3) Chiller Coil Condensed Water Collection Trays – The customer is responsible for the water collection trays on the downstream side of the coils. Condensed water exiting each coil must be collected and transported to the central sump drain pan(s) supplied by GE.
4) Chiller Coil Supply and Return Manifolds – The customer is responsible for each of the individual chiller coil supply and return manifolds usually supplied with the chiller coils from the chiller coil supplier.
5) Chiller Coil System Supply and Return Manifolds – The customer is responsible for the chiller coil supply and return manifolds to and from the banks of chiller coils on the right and left hand side of the filtration unit.
6) Piping / Manifold Insulation – The customer is responsible for the chiller coil piping insulation. This includes but is not limited to supply and return manifolds, supply and return piping and all supply and return valves, elbows, flanges, and flexible connections. NOTE: NO INSULATION SHOULD BE INSTALLED INSIDE THE CLEAN AIR PLENUM AS IT MAY BECOME DISLODGED AND BE INGESTED BY THE GAS TURBINE.
7) Piping Hangers / Hangar Support to Top of GE support steel – Any and all piping hangers are the responsibility of the customer. GE is supplying structural steel for the pipe hangers in the areas as identified within the GE customer documentation for each requisition. The maximum loads for each pipe support location shall not exceed those stated in the GE customer documentation for each requisition. GE supplies structural steel capable of supporting only those loads as stated by GE and only at those locations as defined by GE.
8) Chiller Coil Side Enclosures – The customer is responsible for the chiller coil manifold enclosures that mate with the side of the filter house. GE is supplying bolted side plates that may be used prior to the installation of the chiller coils and replaced with the customer supplied coil enclosures upon installation of the customer supplied chiller coil system. The location of the supply and return nozzles for each individual coil varies between coil manufacturers and designers and must be included as part of the customer's scope of supply.
9) Instrumentation – Any and all of the customer defined instrumentation for control of the chilling system, leak detection, performance measurement, and chiller coil pressure drop measurement is the sole responsibility of the customer. GE does not provide any mounting brackets, pressure taps, and bolting hardware outside of the standard devices as defined by GE for protection of the gas turbine.
10) Control Valves - Any and all control valves for the chiller coils and system are the sole responsibility of the customer.

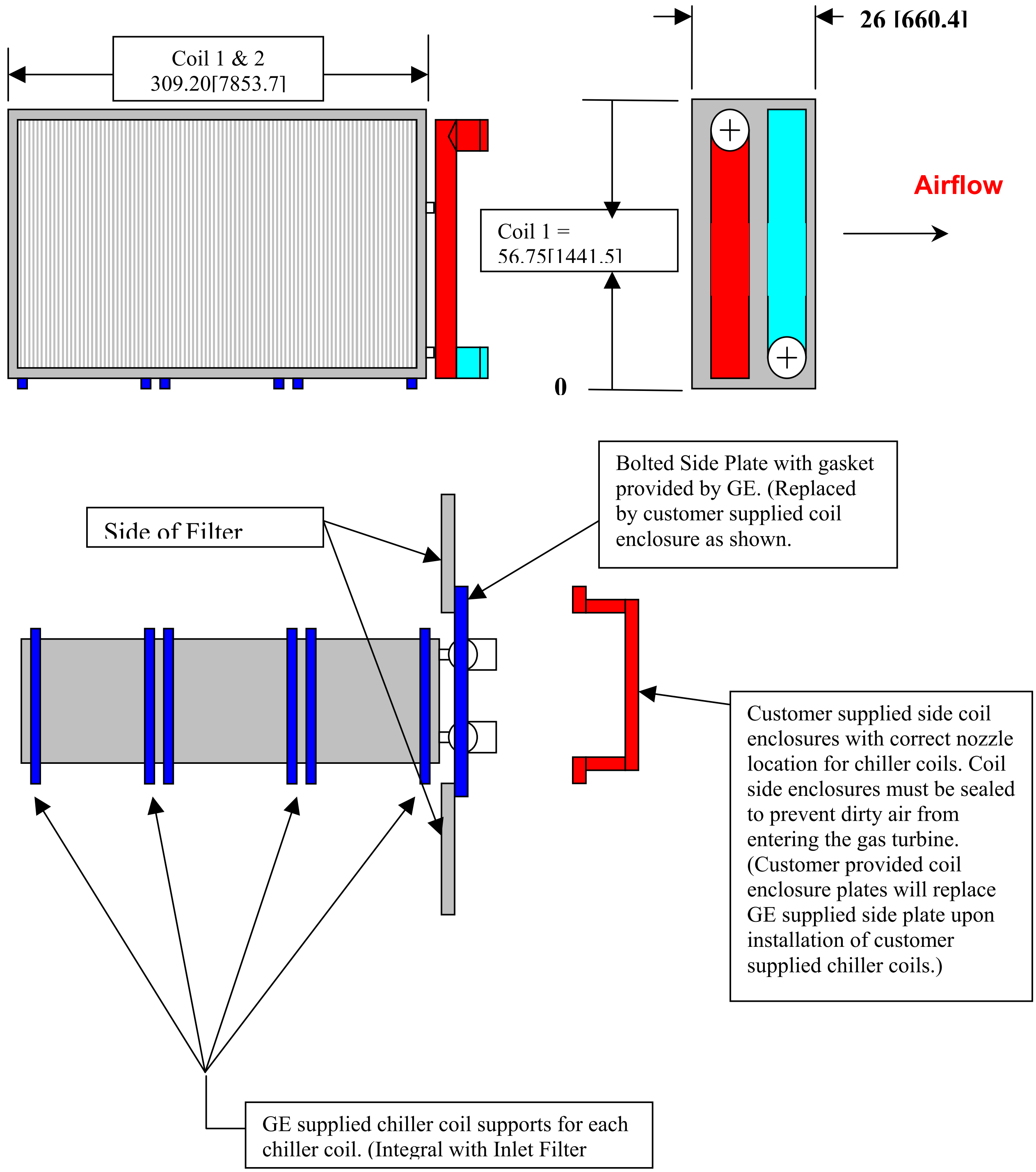


Figure 1: Maximum Chiller Coil Dimensions and Supplied Equipment.

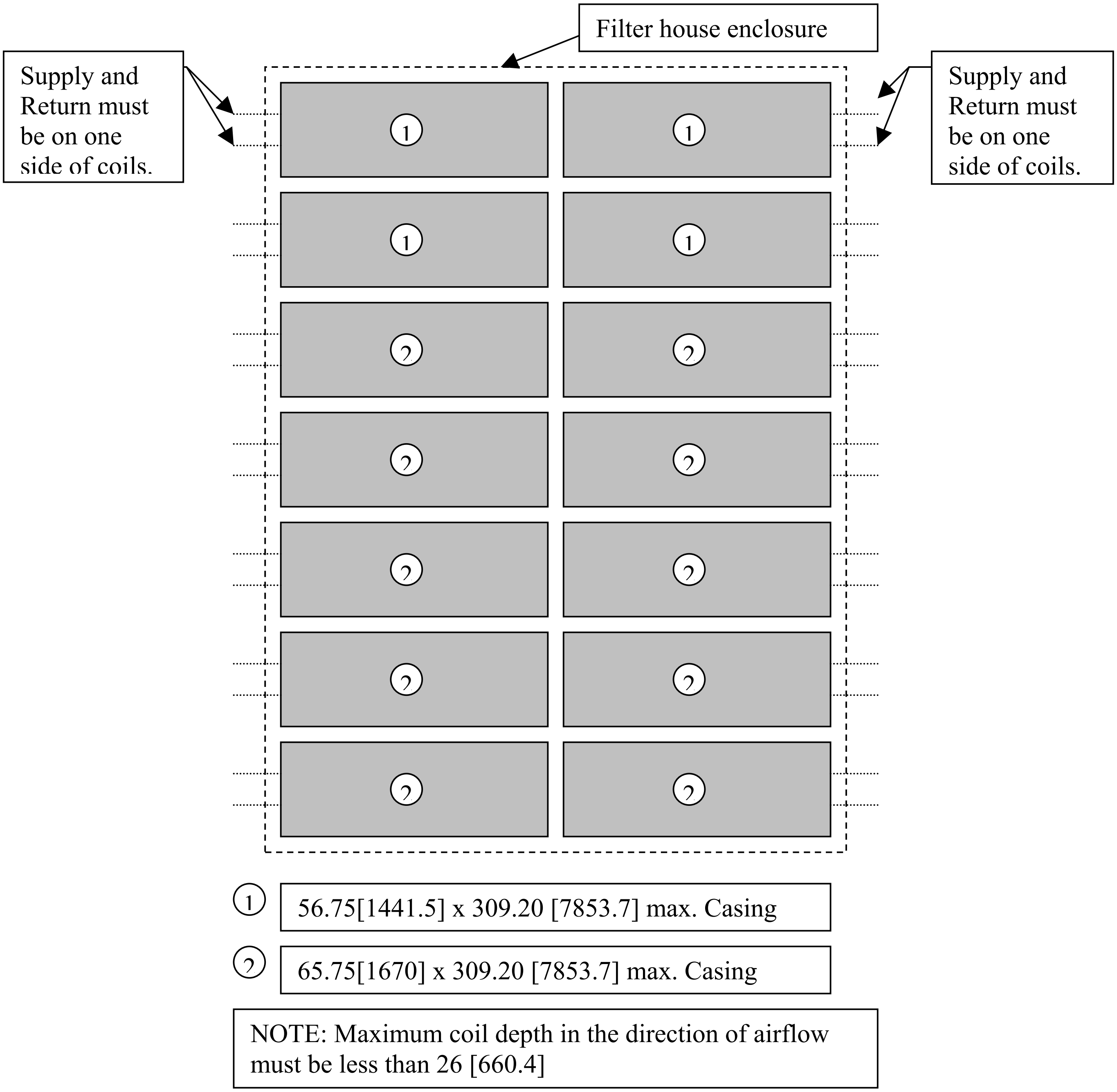


Figure 2: Chiller Coil Arrangement.

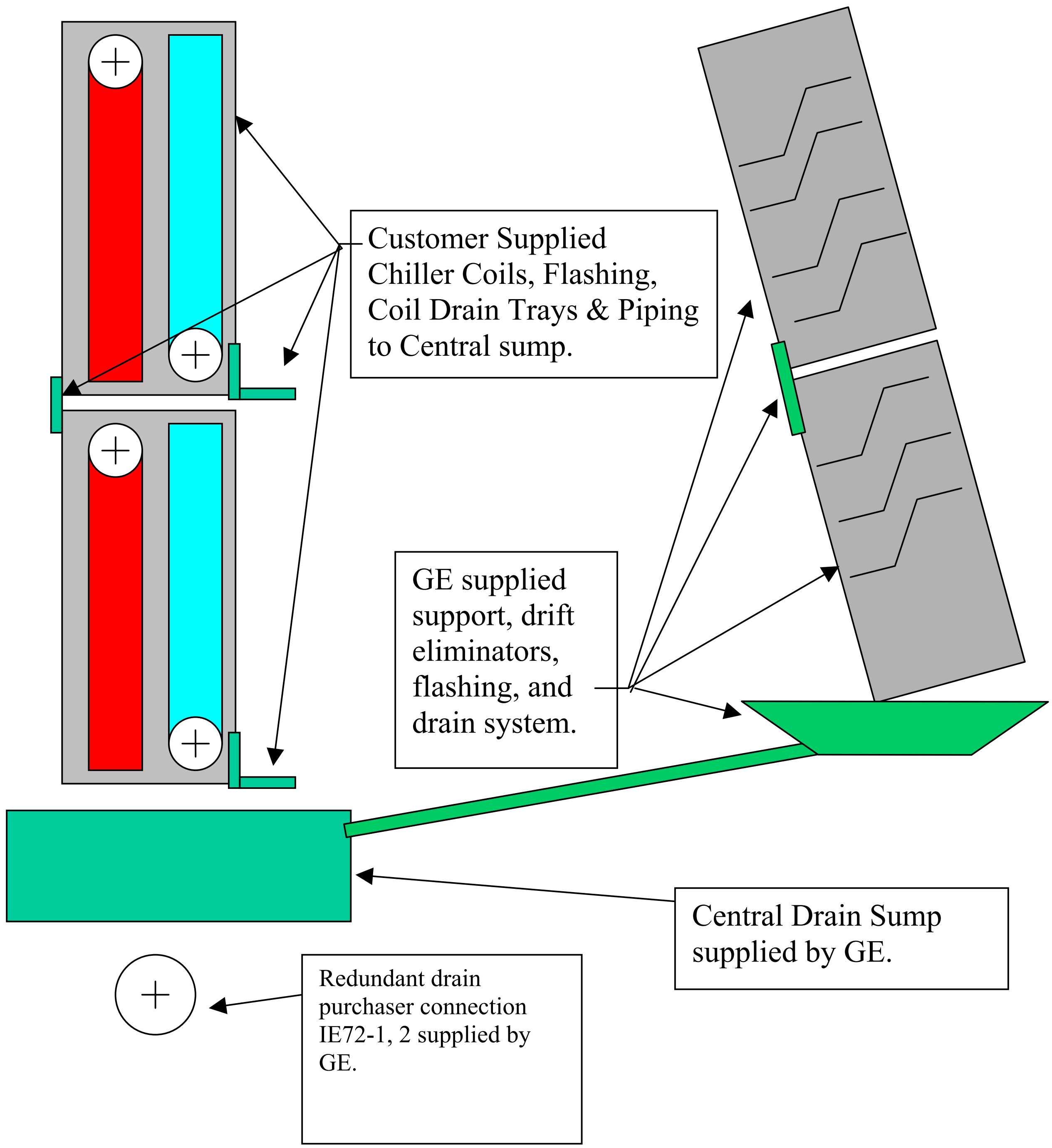


Figure 3: Chiller Coils and Drift Eliminator Scope of Supply.