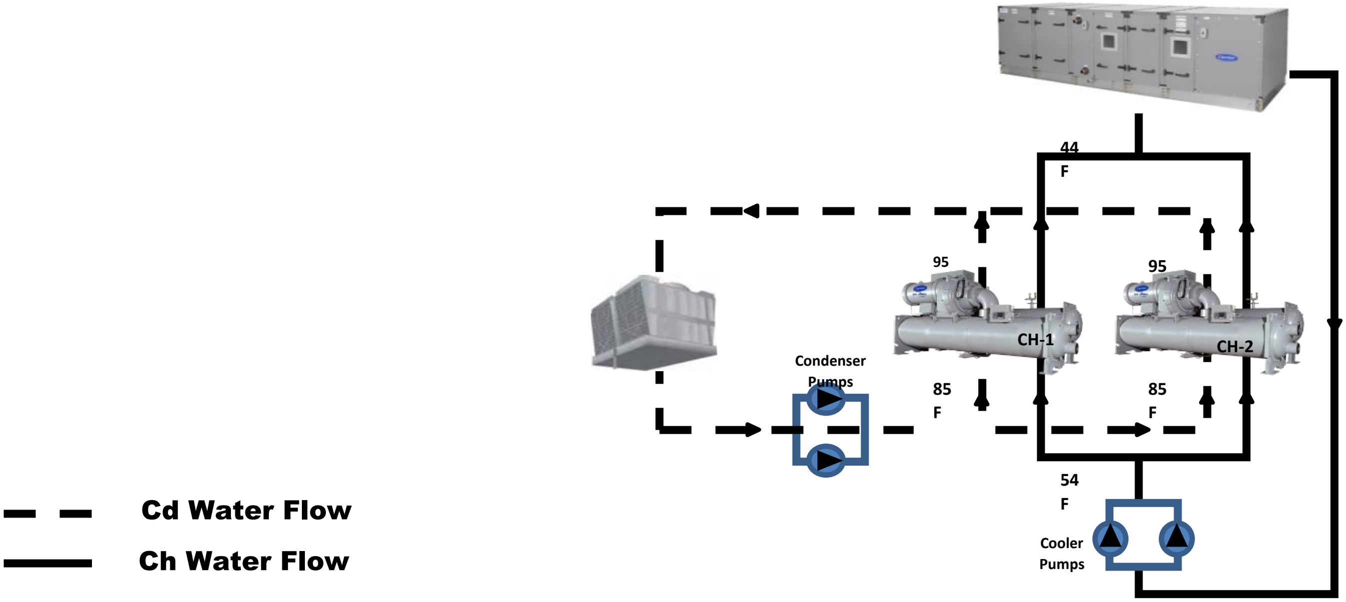
The Benefits of Utilizing Series Counterflow Variable Speed Screw Chillers

The purpose of this document is to detail the benefits of utilizing variable speed screw chillers in a series counterflow plant configuration.

Parallel Chillers Overview

Lift is defined as the temperature difference between the saturated condensing temperature and the saturated suction temperature and can be approximated from the difference between the leaving condenser water and the leaving chilled water.

When two (2) or more chillers are connected in parallel, both chillers experience the same lift. In Figure 1 below, the lift each chiller experiences is approximately 51F.



Chiller	Leaving Chilled Water Temperature	Leaving Condenser Water Temperature	Lift$_{approx.}$
CH-1	44F	95F	**51F**
CH-2	44F	95F	**51F**

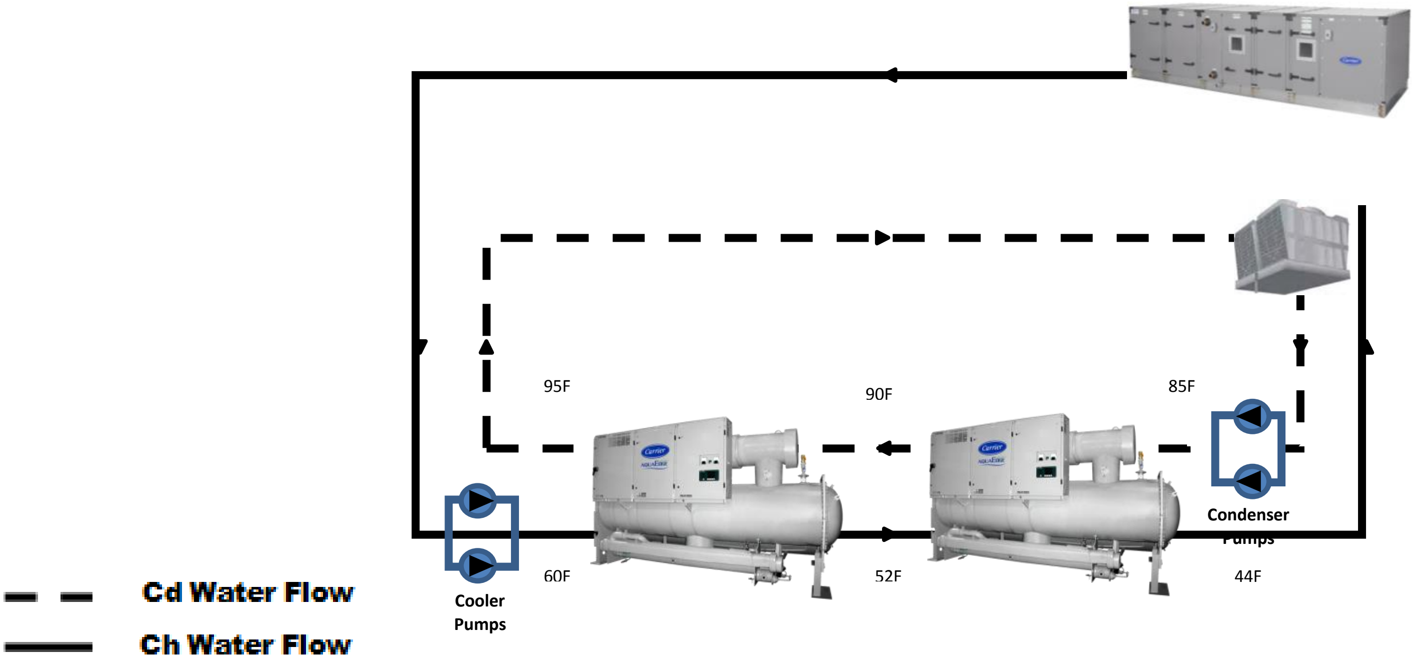
Figure 1

Series Counterflow (SCF) Chillers Overview

Two or more chillers are piped in series counterflow (SCF) when the following conditions are met (Figure 2):

a. Water flows through an evaporator in series A→B→C (Figure 2)
b. Water flows through the condenser in series D→E→F (Figure 2)
c. Water flow in the evaporator is opposite of the water flow in the condenser, a counterflow arrangement (opposite to each other)

In a SCF arrangement, the work done (Lift) by each compressor is reduced, which significantly improves the efficiency of the chillers at full and part load conditions (Figure 2).



Chiller	**Leaving Chilled Water Temperature**	**Leaving Condenser Water Temperature**	**Lift$_{approx.}$**
CH-1	52F	95F	**43F**
CH-2	44F	90F	**46F**

Figure 2

The table below provides the resultant lift of this SCF system operating at AHRI conditions.

The total lift produced by the SCF system can be up to ***<u>20% lower than a conventional parallel system</u>***. Lower lift means less work, and therefore a reduction in power consumption for a given cooling capacity.

Application Considerations

When a system is in a SCF arrangement, appropriate field water piping design and proper controls algorithms are required to achieve the desired operational results. In Figure 2, notice that CH-2 has to work harder than CH-1 due to higher operating lift conditions. To ensure that the chiller plant meets set point when one chiller is down and to accommodate for possible system transients, both of the chillers should be independently capable of performance at higher lift conditions.

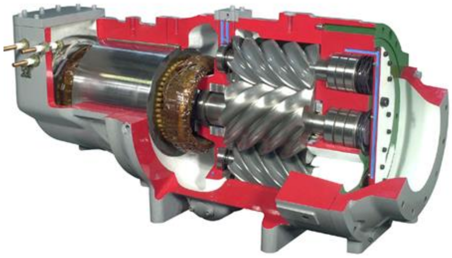
The following eight (8) performance characteristics demonstrate how the 23XRV variable speed screw chiller will perform differently than a centrifugal chiller when operating both independently and in SCF.

1. The 23XRV Variable Speed Screw Ensures Surge Free Operation

Inherent to its nature, centrifugal chiller performance results from a dynamic compression process and as such, is subject to the phenomenon of surge. This occurs when condensing pressure rises to levels beyond the design parameters of the machine causing the refrigerant gas to flow in the reverse direction through the compressor. The recurrence of surging over a period of time can have a detrimental effect on the maintenance and sustainability of the machine. An example of conditions that could cause a chiller to surge, is when the chiller experiences high lift and low load. Such an occurrence could be observed on a day when the wet bulb temperature is high and the building load demands are low. For instance, a school in the summer time when building occupancy is low and wet bulb temperatures are high could provide for surging conditions.

The optimal performance of the 23XRV results from a positive displacement compression process which inherently cannot surge, providing for robust, reliable, sustainable operation, including conditions that exceed design parameters, even for extended periods of time.*

- **Positive Displacement Screw Compressor**
- **Surge Free**
- **Minimum Moving Parts (2 or 3 Screw Rotors)**
- **Fast Response to System Changes (70%/minute)**



Q/R Compressor show for reference

23XRV Compressor has few moving parts and is completely field serviceable.

Figure 3

2. The 23XRV Meets ASHRAE 90.1 2010 Efficiency Requirements at Non-standard AHRI Conditions

Standard chiller operating conditions have been established by AHRI to be per the following:

Evaporator— 44F 2.4GPM/Ton
Condenser— 85F 3.0GPM/Ton

Have you ever wondered why ASHRAE 90.1- 2010 provides minimum efficiency requirements at non-AHRI conditions for *centrifugal chillers* but not for *screw chillers*? The standard represented in Figure 4 accounts for the fact that a centrifugal chiller, configured for an application that calls for 46F of design lift (as in Figure 2), may not be capable of operation at AHRI conditions of 51F of lift. Therefore it may not be capable of being tested at AHRI conditions to prove minimum efficiency compliance. Consequently, ASHRAE 90.1 establishes centrifugal efficiency requirements from a set of equations that are tied to lift. The relationship between lift and efficiency for a centrifugal chiller can be seen from Figure 5. Inherent to centrifugal chillers, ***an increase in lift results in a reduction in chiller efficiency.***

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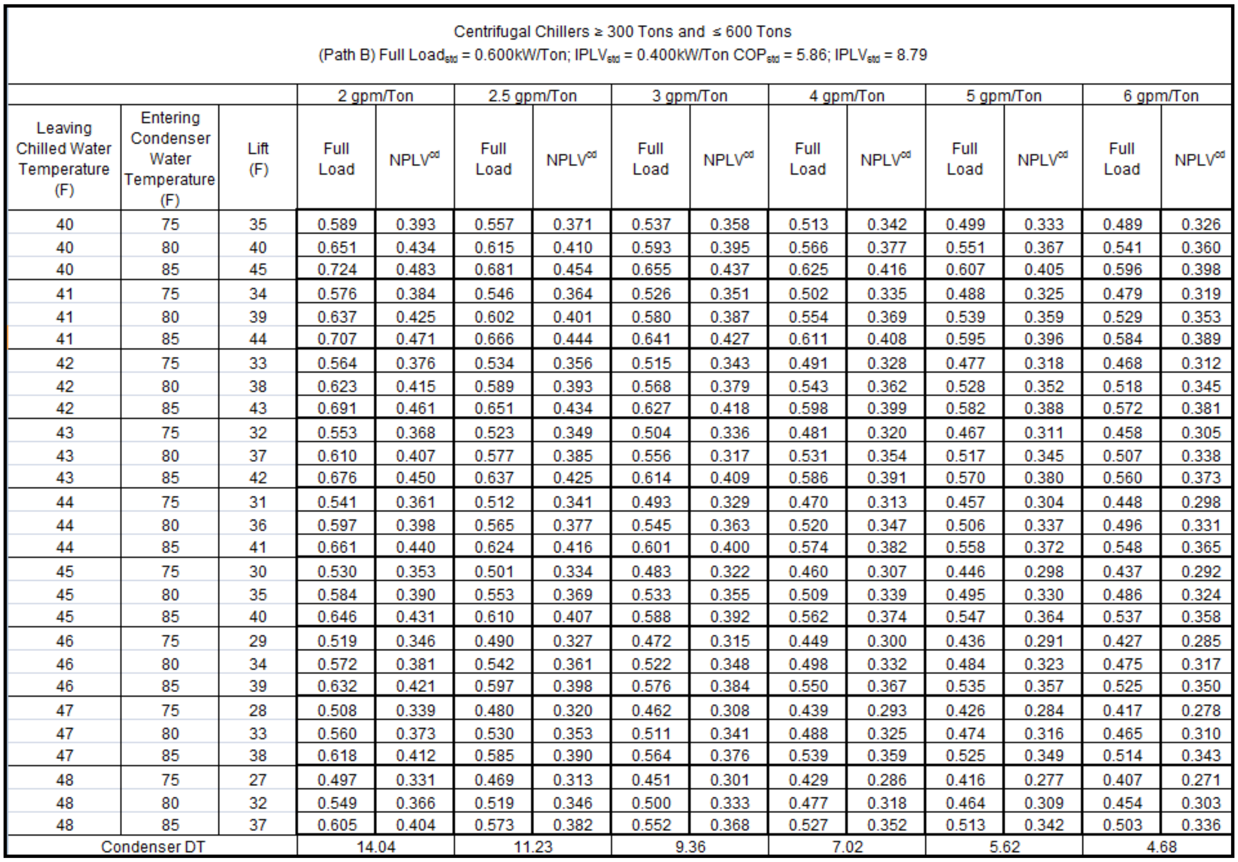
* This quality of the 23XRV enables it to be an excellent selection even for projects wherein there are plans for future plant expansion requiring more capacity.



Centrifugal Chillers ≥ 300 Tons and ≤ 600 Tons

(Path B) Full Load$_{std}$ = 0.600kW/Ton; IPLV$_{std}$ = 0.400kW/Ton COP$_{std}$ = 5.86; IPLV$_{std}$ = 8.79

			2 gpm/Ton		2.5 gpm/Ton		3 gpm/Ton		4 gpm/Ton		5 gpm/Ton		6 gpm/Ton	
Leaving Chilled Water Temperature (F)	Entering Condenser Water Temperature (F)	Lift (F)	Full Load	$NPLV^{od}$	Full Load	$NPLV^{od}$	Full Load	$NPLV^{od}$	Full Load	$NPLV^{od}$	Full Load	$NPLV^{od}$	Full Load	$NPLV^{od}$
40	75	35	0.589	0.393	0.557	0.371	0.537	0.358	0.513	0.342	0.499	0.333	0.489	0.326
40	80	40	0.651	0.434	0.615	0.410	0.593	0.395	0.566	0.377	0.551	0.367	0.541	0.360
40	85	45	0.724	0.483	0.681	0.454	0.655	0.437	0.625	0.416	0.607	0.405	0.596	0.398
41	75	34	0.576	0.384	0.546	0.364	0.526	0.351	0.502	0.335	0.488	0.325	0.479	0.319
41	80	39	0.637	0.425	0.602	0.401	0.580	0.387	0.554	0.369	0.539	0.359	0.529	0.353
41	85	44	0.707	0.471	0.666	0.444	0.641	0.427	0.611	0.408	0.595	0.396	0.584	0.389
42	75	33	0.564	0.376	0.534	0.356	0.515	0.343	0.491	0.328	0.477	0.318	0.468	0.312
42	80	38	0.623	0.415	0.589	0.393	0.568	0.379	0.543	0.362	0.528	0.352	0.518	0.345
42	85	43	0.691	0.461	0.651	0.434	0.627	0.418	0.598	0.399	0.582	0.388	0.572	0.381
43	75	32	0.553	0.368	0.523	0.349	0.504	0.336	0.481	0.320	0.467	0.311	0.458	0.305
43	80	37	0.610	0.407	0.577	0.385	0.556	0.317	0.531	0.354	0.517	0.345	0.507	0.338
43	85	42	0.676	0.450	0.637	0.425	0.614	0.409	0.586	0.391	0.570	0.380	0.560	0.373
44	75	31	0.541	0.361	0.512	0.341	0.493	0.329	0.470	0.313	0.457	0.304	0.448	0.298
44	80	36	0.597	0.398	0.565	0.377	0.545	0.363	0.520	0.347	0.506	0.337	0.496	0.331
44	85	41	0.661	0.440	0.624	0.416	0.601	0.400	0.574	0.382	0.558	0.372	0.548	0.365
45	75	30	0.530	0.353	0.501	0.334	0.483	0.322	0.460	0.307	0.446	0.298	0.437	0.292
45	80	35	0.584	0.390	0.553	0.369	0.533	0.355	0.509	0.339	0.495	0.330	0.486	0.324
45	85	40	0.646	0.431	0.610	0.407	0.588	0.392	0.562	0.374	0.547	0.364	0.537	0.358
46	75	29	0.519	0.346	0.490	0.327	0.472	0.315	0.449	0.300	0.436	0.291	0.427	0.285
46	80	34	0.572	0.381	0.542	0.361	0.522	0.348	0.498	0.332	0.484	0.323	0.475	0.317
46	85	39	0.632	0.421	0.597	0.398	0.576	0.384	0.550	0.367	0.535	0.357	0.525	0.350
47	75	28	0.508	0.339	0.480	0.320	0.462	0.308	0.439	0.293	0.426	0.284	0.417	0.278
47	80	33	0.560	0.373	0.530	0.353	0.511	0.341	0.488	0.325	0.474	0.316	0.465	0.310
47	85	38	0.618	0.412	0.585	0.390	0.564	0.376	0.539	0.359	0.525	0.349	0.514	0.343
48	75	27	0.497	0.331	0.469	0.313	0.451	0.301	0.429	0.286	0.416	0.277	0.407	0.271
48	80	32	0.549	0.366	0.519	0.346	0.500	0.333	0.477	0.318	0.464	0.309	0.454	0.303
48	85	37	0.605	0.404	0.573	0.382	0.552	0.368	0.527	0.352	0.513	0.342	0.503	0.336
Condenser DT			14.04		11.23		9.36		7.02		5.62		4.68	

Figure 4. Representation of ASHRAE 90.1 2010 Path B minimum efficiency requirements (Non-AHRI conditions)

The graph in Figure 5 shows the inverse relationship between lift and efficiency. Notice as the lift increases, the efficiency decreases as is seen by the increase in power consumed (kW/Ton).

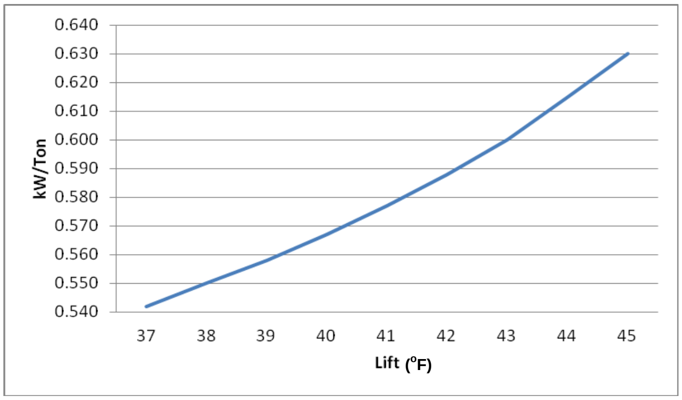


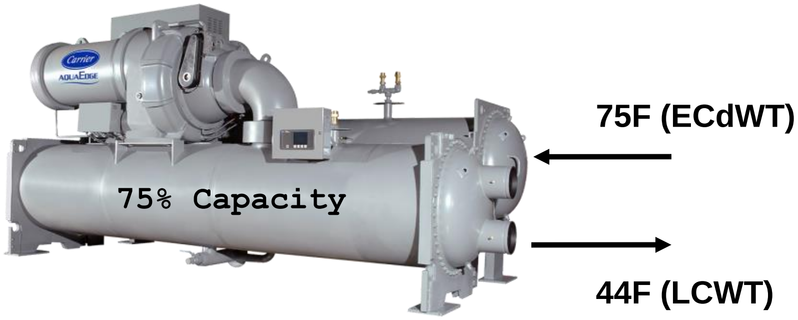
Figure 5. Lift/Efficiency relationship for centrifugal chillers

Unlike centrifugal chillers and as a result of its positive displacement design, a screw chiller applied for non-standard lift conditions can be tested at AHRI conditions of 51F of lift. Consequently, the ASHRAE 90.1 standard does not need a separate table and set of equations to obtain efficiency requirements for a screw chiller as it does with a centrifugal chiller.

3. The 23XRV Screw Chiller Ensures Optimal Performance by Matching Speed with Capacity

The following example illustrates how centrifugal chiller performance is leveraged by the ideal fan laws with respect to the relationship between speed and capacity.

Given:
AHRI conditions (**Evap: 44F 2.4GPM/Ton Cond: 85F 3.0GPM/Ton**)
Constant speed pumps (cooler and condenser)
Chiller plant at 75% capacity
75F ECdWT
10F Condenser delta T at full load



Find:
Chiller speed at given capacity

Basic Equations:

$$\text{Flow} \propto \text{Speed}$$
$$\text{Lift} \propto \text{Speed}^2 \quad \text{\{Ideal Fan Laws\}}$$
$$\text{Power} \propto \text{Speed}^3$$

$$\text{Tons} = \frac{\text{GPM}(\Delta \text{T})}{24} \quad \text{\{Chiller Capacity\}}$$

Since flow is constant, 75% Capacity occurs at 75% of condenser ΔT or 7.5F

Since the pumps are constant speed, the only parameter that affects lift is the condenser delta T. The leaving chilled water temperature is 44F and the leaving condenser water temperature is the entering condenser water temperature plus the delta T across the condenser. Since the chiller is at 75% capacity, the delta T is 7.5F. Compressor lift can be approximated as the difference between the leaving chilled water and the leaving condenser water. The lift requirement at 75% capacity will be 75%.

$$\%\,\text{Lift} = \frac{\text{Operating Conditions}}{\text{Design Lift}} = \frac{(75+7.5)-44}{(85+10)-44} = \frac{38.5}{51} \approx 75\%$$

For optimal chiller efficiency, the compressor speed should vary directly with chiller capacity. However from the ideal fan laws for centrifugal chillers, to meet lift requirements, *the actual compressor speed is greater than the speed required by the chiller capacity.*

1. Speed required to match the given capacity condition is 75%
2. Speed required to match the given lift condition is 86%

$$\text{Speed}^2 = \text{Lift}$$
$$\text{Speed} = \sqrt{\text{Lift}} = \sqrt{.75} \approx 86\%$$

3. Operating speed is the greater of 1. and 2. which is 86%
4. To unload to 75% load at 86% rpm, the IGV must close.

The ideal fan laws establish that the compressor's minimum speed will likely be dictated by lift (square relationship) rather than capacity (linear relationship). From the ideal fan laws, two (2) vital conclusions can be established at this point.

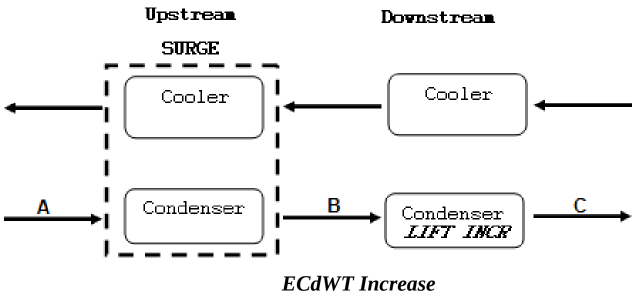
1. Operating ***below*** this speed could cause the chiller to surge.
2. Operating ***above*** this speed results in exponential costs in power consumed by the plant ($\text{Power} \propto \text{Speed}^3$) .

In addition to speed, inlet guide vanes are also built into centrifugal chiller design to respond to changes in capacity. It's important to note that generally the first line of defense against surge is to increase the compressor speed, and that compressor speed compensation is significantly faster than inlet guide vane positioning, so there will be a delay before the inlet guide vanes can lower their position to compensate for the extra refrigerant flow through the compressor as a result of the speed increase.

4. The Redundancy of the 23XRV Ensures Sustainable, Robust Performance and Optimizes it for SCF Plant Configuration

Depending on where it is being used (both geographical location and within the system), the components of the centrifugal chiller will be chosen and/or sized accordingly (i.e. Impeller, shroud, motor, etc.). Recognizing that the efficiency of a centrifugal chiller is sensitive to design lift, in a series counter-flow plant configuration, each chiller will experience different lift conditions (see Fig.2). It may be tempting to select two centrifugal chillers such that each is optimized for the lift that the plant requires based on its place in the system (upstream or downstream). However, when one chiller is down, the plant must still be able to maintain set point. To accomplish this, the chillers should be independently capable of performance at higher lift conditions to accommodate for unexpected system transients (system load, ambient temperature, component malfunction/failure). In addition, when two centrifugal chillers are piped in series counterflow, if one surges, it is likely that the adjacent centrifugal chiller will surge as well. Figure 6 below illustrates this “domino effect “of surging that is characteristic of centrifugal chillers.

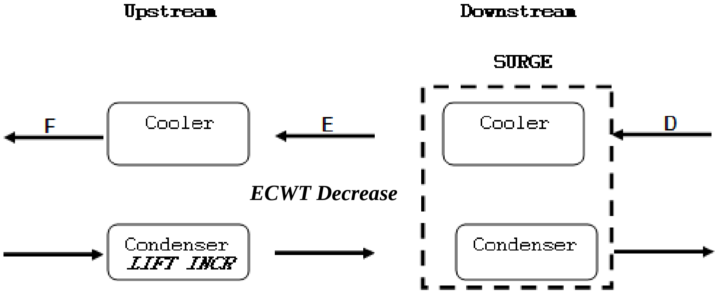
In case one, if the upstream chiller surges as a result of a rapid increase in ECdWT from 85F to 88F. The resulting increase in compressor speed will cause an increase in the leaving condenser water temperature from 90F to 93F (B). This water immediately enters the downstream chiller, which limits the time that its compressor speed has to react, and will likely bring it immediately into a surge condition.



Case 1: Downstream Surge Resulting From Increase in ECdWT		
Temperature	Before Upstream Surge	After Upstream Surge
A	85	86
B	90	91
C	95	96

Figure 6A. Surge Resulting From Centrifugal Chillers In Series Counterflow Configuration

In case two, if the downstream chiller encounters a transient that brings it into surge as a result of a rapid decrease in its ECWT from 65F to 60F (D). The resulting increase in compressor speed causes its leaving cooler water temperature to decrease from 52F to 51F (E). This sudden temperature change immediately impacts the upstream chiller which limits the time that it’s compressor speed has to react which will likely it immediately into a surge condition as well.



Case 2: Upstream Surge Resulting From Increase in ECWT		
Temperature	Before Downstream Surge	After Upstream Surge
D	60	59
E	52	51
F	44	43

Figure 6B. Surge Resulting From Centrifugal Chillers In Series Counterflow Configuration

5. The 23XRV Surge Free Operation Simplifies the Chiller Controls which can Optimize Set Up, Operation, and Maintenance Time.

Typically, a chiller is designed with surge prevention algorithms such that its minimum speed is a safe distance from the surge point. This speed differential is called the deadband (Figure 7).

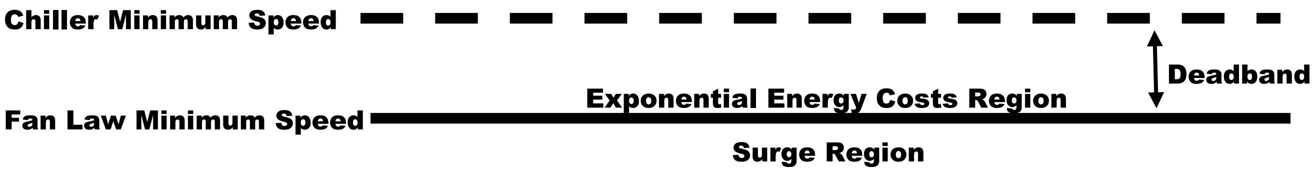


Figure 7. Deadband

Should a plant encounter issues with repeated surge events, the service office will likely be called in to address this issue. To fix it, they may dampen the response. Making some local changes to key parameters will result in the chiller not being allowed to operate as close to the surge line in order to reduce the chances of surge. Notice, however, that this means the efficiency may be compromised (increasing operating costs) since at any given condition, further speed reduction may be possible, but it may be prevented in order to respond to a transient change in load or lift condition that could cause the chiller to surge.

When all of the chillers being bid on a job are centrifugals, one can assume that the effect will be about the same for any manufacturer. However, when comparing variable speed screws (that cannot surge) with centrifugals, one quickly understands that the elimination of surge prevention algorithms associated with centrifugal chillers, works to simplify the chiller controls of the 23XRV screw chiller which minimizes it’s need for site adjustment allowing it to maintain its “as sold” efficiency.

York provides a “surge protection” offering for it’s centrifugal chillers. It functions by raising the leaving chilled water temperature (LCWT) in a effort to lower the lift. Whereas theoretically, this operation could reduce the opportunity for compressor surge, operating the chiller at a setpoint above design for an extended period of time could have a detrimental effect in applications such as hospitals and data centers where maintaining design conditions are critical. Raising the LCWT raises the coil temperatures which reduces the latent heat capacity. This results in increased humidity to the conditioned space.

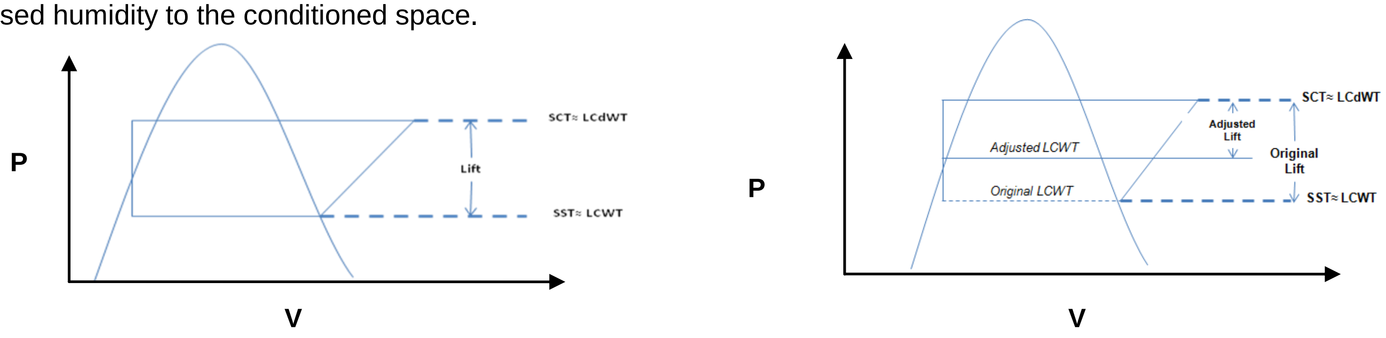
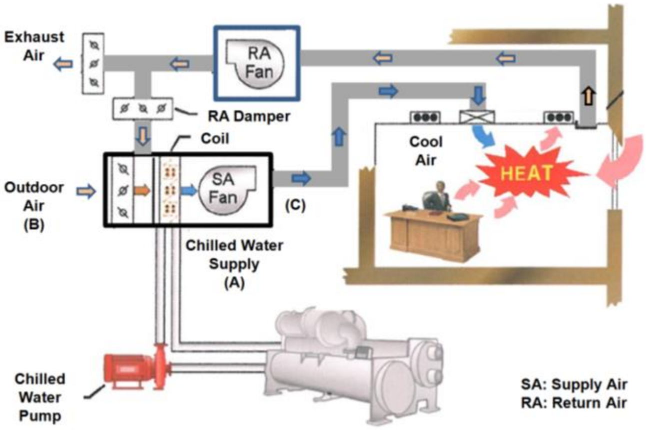


Figure 8 Raising the LCWT Lowers the Lift

When the chilled water supply temperature is above the design set point, the performance of other components of the HVAC system can also be impacted. Consider the diagram in Figure 9. The capacity of the air handler unit (AHU) is determined by the temperature differential of the outdoor entering air (B) and the chilled water supply temperature (A) across the coil. The coils are sized to maintain design space temperature (C) in accordance with a specific chilled water supply temperature. Raising the chilled water supply temperature increases the amount of work required by the coils. This reduces the capacity of the AHU while increasing its operating costs, since it now has to run longer to meet the conditioning requirements of the space.



AHU Capacity for Varying Chilled Water Supply Temperatures at 200GPM and Entering Air 80/67F (WB/DB)		
Chilled Water Supply Temperature (F)	Chilled Water Return Temperature (F)	Total Cooling Capacity (MBH)
42	50.61	860.88
45	52.66	766.66
48	54.72	672.27

Figure 9 AHU capacity reduction resulting from increased LCWT

6. The 23XRV Produces Industry Leading Part Load Efficiencies in Series Counterflow.

The table below shows the part load efficiency comparisons of four series counterflow arrangements.

- **Case 1** compares the efficiencies of two constant speed centrifugal machines with two full load optimized 23XRVs.

- **Case 2** compares the efficiencies of two constant speed centrifugal machines with two part load optimized 23XRVs.

- **Case 3** compares the efficiencies of two variable speed centrifugal machines with two full load optimized 23XRVs.

- **Case 4** compares two variable speed centrifugal machines with two part load optimized 23XRVs. Notice the part load efficiency advantages of the 23XRV in each case.

Case	Description	Chiller Systems	NPLV kW/Ton	Cooler GPM	Condenser GPM
1	2 **constant** speed SCF centrifugals vs. 2 SCF **Full Load Optimized 23XRVs**	2 x 450 SCF CS Centrifugals	0.512	1350	2700
		2 x 450 SCF FLO 23XRVs	0.311		
		23XRV Advantage	***40%***		
2	2 **constant** speed SCF centrifugals vs. 2 SCF **Part Load Optimized 23XRVs**	2 x 450 SCF CS Centrifugals	0.512	1350	2700
		2 x 450 SCF PLO 23XRVs	0.288		
		23XRV Advantage	***43%***		
3	2 **variable** speed SCF centrifugals vs. 2 SCF **Full Load Optimized 23XRVs**	2 x 450 SCF VS Centrifugals	0.328	1350	2700
		2 x 450 SCF FLO 23XRVs	0.311		
		23XRV Advantage	***5%***		
4	2 **variable** speed SCF centrifugals vs. 2 SCF **Part Load Optimized 23XRVs**	2 x 450 SCF VS Centrifugals	0.328	1350	2700
		2 x 450 SCF PLO 23XRVs	0.288		
		23XRV Advantage	***12%***		

Figure 10 23XRV Optimization of Full and Part Load Plant Efficiency

7. The 23XRV Has Been Documented To Maintain Sustainable Operation Through a Tower Fan Failure

Optimal performance of a centrifugal chiller is dependent upon a number of variables: ambient temperature, flow rate, tube fouling, etc. An unexpected rise in lift could result in a surging occurrence dependent upon how fast the chiller is able to respond. Figure 11 below demonstrates an actual event that occurred and illustrates the robustness of the 23XRV variable speed screw chiller in its response to this occurrence. The chart shows the resulting temperature and capacity outputs of the 23XRV when it encountered a tower fan failure resulting in excessively high condenser temperatures (≈105F) and pressures for over 13 hours. Such an occurrence would likely cause a centrifugal chiller to shut down.

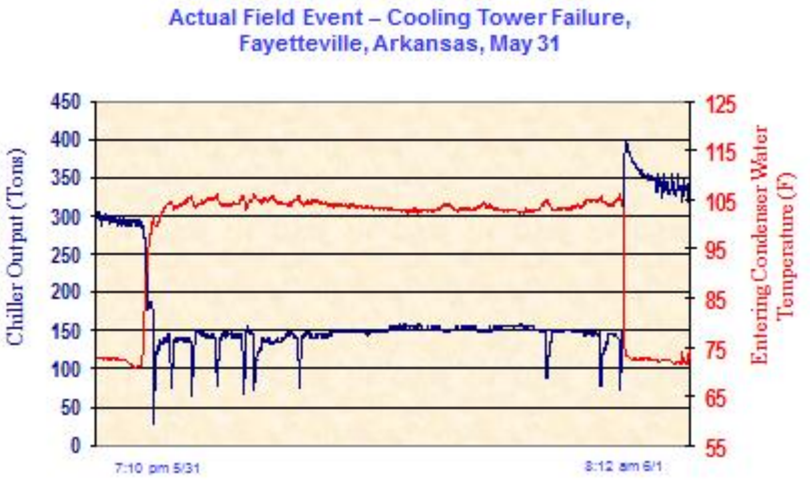


Figure 11 23XRV Response To Cooling Tower Fan Failure

8. The Robust Design of the 23XRV Enables its Sustainable Operation when Encountering System Transients

Figure 12A below illustrates a typical SCF chiller plant arrangement.

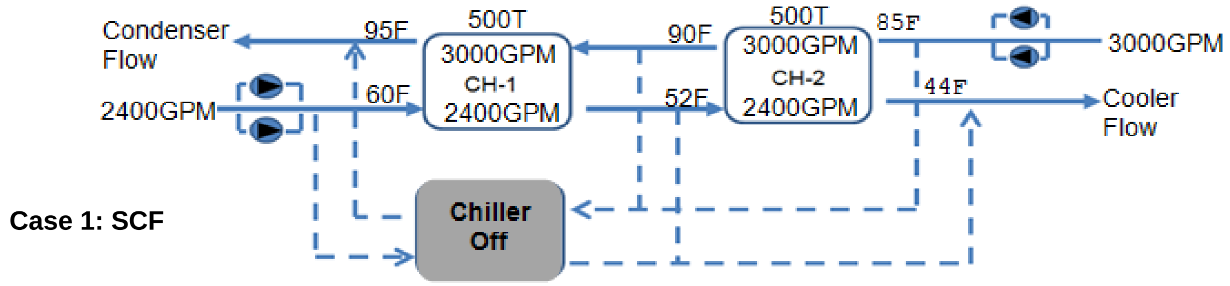


Figure 12A Lift variations for different chiller configurations

What would happen in the event that CH-2 (downstream) failed (Fig. 12B)? It is for such an event that optimal series counterflow design typically includes an additional parallel chiller. The isolation valves of CH-2 (downstream) would close, eliminating it from the system. In addition, the isolation valves of CH-3 would open, allowing CH-1 and CH-3 to provide cooling for the given application. Unlike the series configuration where CH-1 and CH-2 were both provided with full pump flow, the parallel configuration causes CH-1 and CH-3 to each receive half of the available pump flow.

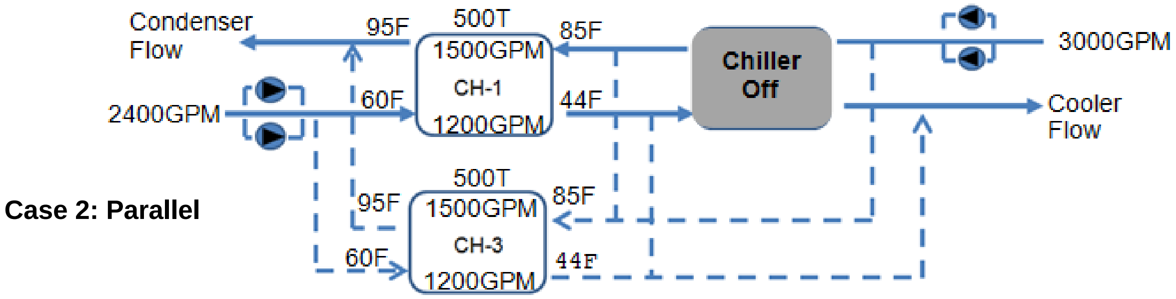


Figure 12B Lift variations for different chiller configurations

In case 3, the parallel chillers are provided with half of the condenser flow as a result of a pump malfunction.

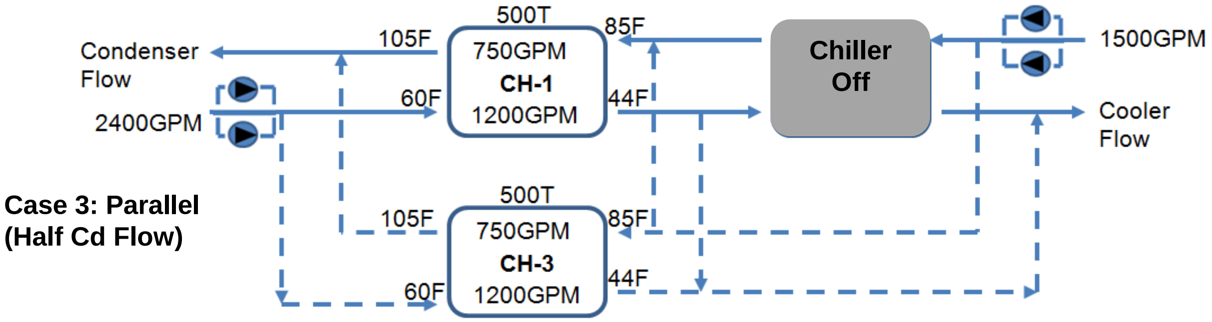
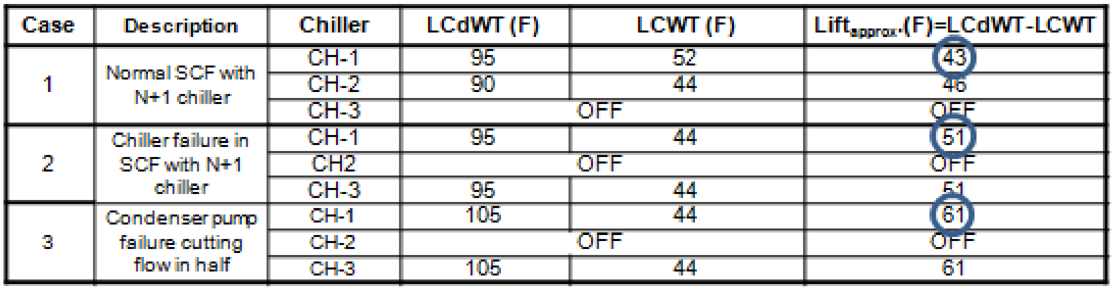


Figure 12C Lift variations for different chiller configurations

Consider the lift variations of the three cases illustrated above, in the table shown below. *<u>Efficiency optimization of a centrifugal chiller is leveraged by the applied nature of the machine</u>*. As a result, it is likely that a centrifugal chiller would not be able to provide sustained operation should it encounter the system transients depicted above. Conversely, the 23XRV variable speed screw chiller will provide robust, reliable, sustainable performance at all of these conditions.



Case	Description	Chiller	LCdWT (F)	LCWT (F)	$\text{Lift}_{\text{approx.}}$(F)=LCdWT-LCWT
1	Normal SCF with N+1 chiller	CH-1	95	52	43
		CH-2	90	44	46
		CH-3	OFF		OFF
2	Chiller failure in SCF with N+1 chiller	CH-1	95	44	51
		CH2	OFF		OFF
		CH-3	95	44	51
3	Condenser pump failure cutting flow in half	CH-1	105	44	61
		CH-2	OFF		OFF
		CH-3	105	44	61

Figure 12D Lift variations for different chiller configurations

Based on the reasons outlined in this document, VFD driven centrifugal chillers have limitations in offering a balanced performance at all load conditions while introducing operational complications in the system. In conclusion the 23XRV provides:

- ✓ *More consistent, reliable, robust and efficient performance compared to any centrifugal chiller in SCF configuration.*
- ✓ *Up to 20% reduction in lift when piped in series counter-flow arrangement*
- ✓ *Surge free operation*
- ✓ *Optimal performance by matching speed with capacity*
- ✓ *No mechanical unloaders*
- ✓ *Optimal redundancy in series counterflow as opposed to centrifugal chillers*
- ✓ *Optimal efficiency in varying lift conditions*
- ✓ *Simplified chiller controls resulting from the absence of surge algorithms*
- ✓ *Compressor speed variation from 100%-25% resulting in exponential savings*
- ✓ *Optimal full and part load efficiencies when in series counter-flow*

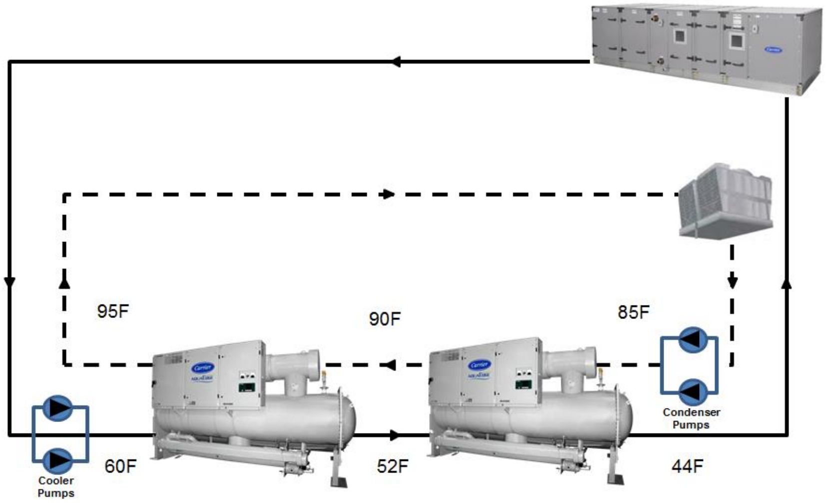


Figure 13 23XRV Series Counterflow