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PERFORMANCE OF VARIABLE SPEED CENTRIFUGAL CHILLERS

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ABSTRACT

Centrifugal water chiller performance is rated for full-load conditions at peak condenser water temperatures—85° F (29.4° C)—and for specified partial load conditions (Integrated Part-Load Value or IPLV). Most centrifugal chillers are designed to operate at fixed speed, with load control provided by compressor inlet guide vanes and/or hot gas bypass.

The cooling water temperature available for a centrifugal chiller will vary with the seasons, the time of day, and changes in the weather. As the cooling water temperature drops, it becomes feasible to reduce the speed of the centrifugal compressor. This paper describes the performance improvements that follow from reducing compressor speed in relation to cooling water temperature and chiller load. The interaction of speed control and inlet guide vane control is discussed. Both single stage and multistage compressor applications of variable speed drive are considered.

INTRODUCTION

Both single and multi-stage centrifugal compressors are widely used for liquid cooled chiller applications ranging from about 100 ton (351.7 kW) to 10,000 ton (35,169 kW). In most cases, the compressors are designed to run at constant speed. With such a wide range of application, the system requirements can be equally diverse. They range from installations with constant condenser water temperature regardless of the load, to installations that operate continuously at 100% load with varying condenser water temperature. Most systems operate between these extremes. The HVAC industry uses a unique operating line, known as the ARI load line, to model compressor operation for comfort cooling applications. In addition, an averaged performance number, known as the IPLV, is used as an index representative of chiller performance at full and part-load conditions.

The ARI load line defines condenser water temperature as a function of chiller load, and has remained unchanged for many years. The IPLV calculation has undergone changes since its original introduction, and is now based on many assumptions of building type, operating hours, and Atlanta's annual dry-bulb temperature profile. The condenser water temperature used for the different load points is not based on weather data; rather, it is a linear distribution between 60° F (15.6°C) at 0% load and 85° F (29.4°C) at 100% load.

Many in the industry believe the ARI load line and IPLV calculation are too conservative, not showing the actual water temperature available to a chiller at different operating conditions. Review of the Atlanta case will demonstrate the origin of these concerns. While the mean wet-bulb temperature coincident with the full-load design condition is actually 74° F (23.3°C), a wet-bulb temperature of 77° F (25°C) [2]—a value observed less than 1% of the time in the summer—is used to establish a design condenser water temperature of 85° F (29.4°C). Opponents argue that, in the need to be simple and conservative, (i) it neglects night time usage of chillers distorting the true load profile; and (ii) it uses arbitrary condenser water temperature distribution as a function of load. They argue, while it is correct to use the higher wet-bulb temperature to design controls to protect the chiller from surging even during that 1% of the time it is observed, it is not correct to use it for the purpose of comparing performance of different machines. Instead, the mean coincident wet-bulb temperature should be used. According to the standard, at 10% load, the cooling water temperature is assumed to be 62.5° F (16.9°C); the lowest it will ever be as manufacturers promise operation only down to 10% load. If one generates an ARI-type load line using the true mean wet-bulb temperature distribution and reasonable tower performance, the water temperature at 10% load would be even less than 55° F (12.8°C). In fact, many customers demand operation down to 55° F (12.8°C). This means that their real operating line is far steeper than the ARI load line. Furthermore, the new operating line would weigh the lower temperature end of the part-load application heavier than the current one does.
Historically, chiller performance has been evaluated only on full-load operation, expressed in units of kW/ton. The introduction of the IPLV has more customers evaluating both full-load and part-load performance. As energy costs continue to rise, many owners and consultants are now requesting performance based on actual weather data and load profiles, rather than the generic ARI load line and IPLV formula. This includes full-load performance with low condenser water temperature, especially in multiple chiller installations. Very few applications, mainly process, are interested in constant condenser water operation.

The ARI standard establishes the bases of competition for the various manufacturers: and most importantly, includes an estimate of the proportion of time these machines operate at part-load. The weighting functions used for the IPLV calculation indicate that these compressors would have to operate the majority of the time with their inlet guide vanes substantially closed; therefore, far away from their peak efficiency points. On the other hand, a variable speed compressor can be operated in such a way to always run the compressor near its peak efficiency location. Since the locus of peak efficiency of a variable speed compressor is parallel to a typical system operating line, considerable energy savings can be realized with a strategy that includes variable speed to control capacity.

Multi-stage compressors—especially designs with inlet guide vanes only on the first stage—have disproportionately larger pressure rise in the second stage than in the first, requiring substantial closure of the inlet guide vanes on the first stage and/or a limit on how low the condenser cooling water temperature may be allowed to go. Artificial requirements to maintain high condenser water temperature, as well as use of hot gas bypass for stable operation at low load, are both highly inefficient exercises.

SINGLE STAGE CHILLERS

In what follows, we will make three sets of comparisons: what we call the 0th, 1st and 2nd order approximations. Each comparison involves a constant speed machine and the same machine run with a variable speed drive. For our study, we have considered an inverter driven variable speed machine. Therefore, typical inverter associated efficiencies, including that of an induction motor and a harmonic filter, are included. However, the observations made are equally valid for other drive means such as gas engine drive or turbine after the proper bookkeeping is done to reflect the efficiencies of the particular drive system.

IPLV Comparison

As we all know, the performance of a compressor is dependent upon where on its performance map it is selected. Therefore, for the 0th order study, we have picked two chillers representing different regions of the map. We have also used the current IPLV calculation methods to evaluate their performance. Comparison of the two variable speed chillers itself reveals a rather interesting phenomena.

The system operating lines for these selections are shown in Figure 1, labeled as “A” and “B”. First, we can see that these operating lines track the peak efficiency points of the variable speed map with slight convergence towards the low load. Second, if one draws horizontal lines successively between the operating lines—connecting points with the same condenser water temperature—and compare the efficiencies of the end points, it can be observed that “B” starts out as the poorer performer machine by a few points and surpasses “A” mid way of the operating line. Since 50% and 75% load points are weighted heavier than either 25% or 100% points, the variable speed machine that was poorer at full-load turns out to be far superior than the one that was good at full-load. For the operating line “A”, the IPLV of the variable speed machine was 5% better than the equivalent constant speed machine. However, for the operating line “B”, the variable speed was 15% better in IPLV than its constant speed counterpart.

Chiller manufacturers usually have a series of compressors from which they choose the appropriate compressor for a particular duty. These compressors have a capacity range of application that overlaps with that of the neighboring sizes. A computer program then iterates and selects the compressor requiring the least power. Since the peak efficiency points of these compressors favor the low end of its capacity range, chillers like “A” would be selected; and chillers like “B” usually would be passed over in favor of the next size compressor, which can probably do the full-load job at low power. The same programs when selecting variable speed compressors for low IPLV value criteria, will select chiller “B” over the counterpart of “A” in the larger size. These compressors are the better choices for the customer not only for the obvious operating cost savings but also because they will most likely operate outside the region of increased diffuser noise through most of their lives.
Non - ARI Comparison

The Atlanta weather profile data [3] indicates that in summer seasons the wet-bulb temperature equals or surpasses 77°F(25°C), 1% of the time. A conservative approach was taken using an 8°F(4.4°C) difference between the wet-bulb and the condenser water temperature, since a typical tower can deliver water to within 6-8°F(3.3-4.4 C°). Use of 85°F(29.4°C) is therefore, reasonable to prevent any surging of the compressor. However, Table 1 shows that the mean coincident wet-bulb temperature plus the 8°F(4.4°C) approach produces a different condenser water temperature distribution compared to—the 2 1/2 degree temperature reduction for each 10% load increment—rule of the ARI line. Because it is constructed using the true mean coincident wet-bulb temperature data, it represents the correct condenser water temperature distribution. Therefore, in the 1st order approximation we used this distribution and the weighting functions of the ARI load line. Indeed, chiller “A” showed that the variable speed version was 12% better compared to the constant speed machine; while the variable speed version of chiller “B” showed an impressive 23% improvement over the constant speed machine.

Finally, we looked at the way the weighting functions of the ARI standard are constructed. By using only the 09 hr to 16 hr weather data, the weighting functions are skewed in favor of the higher load applications because the daytime temperatures are typically higher than night time. When all 24 hr observations are considered, the weighting functions change from {.17, .39, .33, .11} to {.097, .36, .403, .141} corresponding to {100%, 75%, 50%, 25%} load. In the 2nd order approximation, we incorporated these weighting functions and the condenser water temperature distribution mentioned above. The result was that the variable speed version of chiller “A” improved its integrated part-load value by 15% over the same chiller at constant speed. At the same time the variable speed version of chiller “B” improved the same by 28% over the constant speed chiller “B”. These operating cost savings are sizable enough to provide an attractive recovery of the added capital cost for variable speed. In the case of the inverter drive, part of its cost is offset against the starter cost, as the latter will no longer be necessary.

ATLANTA OFFICE BUILDING SUMMARY‡

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<th>Outside Temp (°F)</th>
<th>mwb* (°F)</th>
<th>Total Temp Hours (hr)</th>
<th>Day T Cool Temp Hours (hr)</th>
<th>Cool Load (%)</th>
<th>Water Temp (°F)</th>
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% Time

| Outside | 8760 | 2125 |

Table 1 *(mwb: mean coincident wet-bulb) †(d: day time hours; t: total hours)

Acoustic Comparison

Robert Stabeley of York International, made studies of the impact of variable speed on chiller noise level. He found that the noise level is dependent on the compressor speed and on whether or where the system operating line crosses the boundaries demarking the onset of rotating stall. Figure 2 shows the relationships between chiller selection point, rotating stall line, system operating lines and associated noise levels. In general chillers tend to reduce their

‡ Atlanta's total day-time mechanical cooling hours is 2125 hr; and the total mechanical cooling hours is 5758 hr.
noise level as they proceed from full-load towards part-load operation although at different rates. However, as soon as any of the system operating lines cross the rotating stall demarkation, the noise level will begin to increase; and eventually surpass even the full-load level.

For this study, the sound levels of the following three cases are compared. They are: (i) a constant speed machine operating with constant condenser water temperature down to 10% of full-load capacity, (ii) a constant speed machine operating along the ARI load line to 10% load, and (iii) a variable speed machine operating along the ARI load line also to 10% load. For the constant condenser water temperature application, the noise level seemed to decrease somewhat down to 75% load; at which point it began to increase. At 50% load, the noise level was 3 db higher than the full-load level. When comparing the constant speed chiller and the variable speed chiller operating along the ARI load line, the constant speed chiller continued reducing the noise level down to 50% load; at which point it too began to rise and attained its maximum noise level at 30% load. The variable speed chiller on the other hand, continued to reduce its sound level down to 30% load. At this point, the variable speed chiller was 7 db quieter than the constant speed chillers. The added benefit of reduced noise levels is available when applying variable speed drive.

MULTI-STAGE CHILLERS

Chillers with multi-stage compressors are in principle the same as those with single-stage chillers. There is the added complexity of maintaining the appropriate interstage pressure, usually through operation of one or more sets of inlet guide vanes. As we have shown in the single stage application, variable speed chillers show more benefit when the condenser water temperature fall off fast as the chiller unloads. Therefore, any scheme that involves maintaining higher condenser water temperature will negate the benefit of variable speed.

CONCLUSION

Operating a centrifugal chiller with a variable speed drive is a good option for most cooling applications. Even with the relatively warm Atlanta climate, energy savings of 28% was possible when compared to a fixed speed drive. This type of savings offers an excellent pay back and return on investment, especially as variable speed drives become more readily available. The widespread availability will revolutionize the way buildings are cooled. Facilities that must operate 24 hours a day, such as hospitals, and hotels will see the greatest benefit due to the large amount of operating hours. Evaluation of specific building load profiles and local weather data will yield the most accurate estimate of the energy savings associated with a variable speed drive. Equally impressive is the 7 db reduction in the noise level.

REFERENCES

TYPICAL VARIABLE SPEED PERFORMANCE MAP

head
proportional to condenser water temperature

flow

Fig 1

OPERATING LINES, STALL LINE & NOISE LEVELS

Fig 2