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Using Magnetic Bearing Orbit Information to Maximize Centrifugal Compressor Efficiency at Off-Design Conditions

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ABSTRACT

Active magnetic bearings used on oil-free centrifugal refrigeration compressors have lower stiffness than conventional oil-lubricated journal or rolling element bearings. The lower stiffness of these bearings makes them sensitive to internal flow instabilities that are precursors of rotating stall or compressor surge. At operating conditions far away from surge the internal flow is very stable and the magnetic bearings keep the shaft centered, resulting in a minimal bearing orbit. The internal flow instabilities that arise when the compressor approaches the surge limit result in some radially fluctuating forces on the shaft. The active magnetic bearings correct for these fluctuating radial forces on the shaft. The bearing orbit increases with the size of these radial forces.

At high flow conditions capacity is controlled by compressor speed for the imposed head. At low flow conditions a combination of variable speed and a means of range extension (e.g. inlet guide vanes, variable-geometry diffuser, or flow recirculation) are required to control capacity at the imposed pressure ratio and guarantee stable compressor operation. When operating with flow recirculation optimum compressor efficiency occurs close to surge at incipient stall conditions when the smallest amount of flow is recirculated with the compressor remaining safe from surge. The positional feedback system of the active magnetic bearing control loop system indicates the bearing orbit which relates to compressor efficiency.

The bearing orbit signal can be used to determine which speed/geometry combination gives the highest compressor efficiency. If the bearing orbit is below a minimum value the compressor runs too close to choke and a reduction in speed combined with an opening of the diffuser throat area would increase compressor efficiency. If, on the other hand, the bearing orbit is above a maximum value the compressor runs too close to surge and an increase in speed combined with a closing of the diffuser throat area is required to increase compressor efficiency and guarantee surge-free compressor. Experience with this control scheme will be illustrated for a newly developed 350 ton two-stage centrifugal compressor where the variable geometry hardware is replaced with a controlled internal flow recirculation.

1. INTRODUCTION

The capacity of variable-speed centrifugal compressors used on water-cooled chillers is primarily controlled by speed variation. Higher compressor speed increases the compressor flow rate when more cooling capacity is required and vice versa. Speed variation is also used to adjust flow for a change in head resulting from changes in ambient temperature.

For each compressor speed there is a minimum flow rate below which the compressor becomes unstable. Compressor flow instability starts as a localized rotating stall phenomenon in one of the diffusing elements of the compressor. The effect of rotating stall on overall compressor behavior is limited to an increase in sound and vibration. Compressor capacity and pressure ratio are not affected by rotating stall.

At lower flow conditions rotating stall transitions into a complete compressor stall also called surge, which manifests itself through large temporary drops in overall head and substantial compressor flow variations, even complete compressor flow reversal. Motor power to the compressor also fluctuates strongly during surge. Compressor surge results in unstable chiller operation forcing a system shut down.

Water-cooled chillers demand stable compressor operation at part-load. Figure 1 shows the required compressor operating area for a water-cooled chiller versus the stable operating zone of the variable-speed centrifugal compressor.

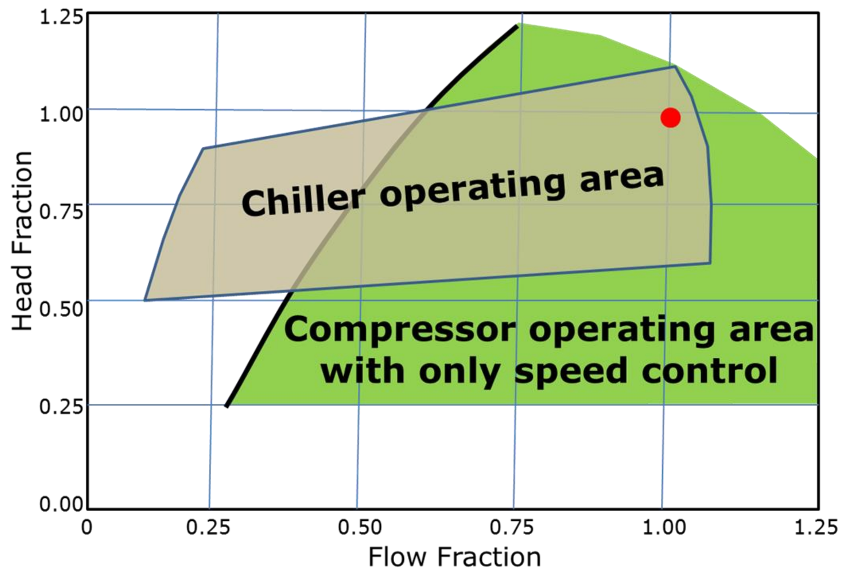

Figure 1. Compressor map area that can be covered with variable speed control.

To understand the compressor control it is important to realize that the compressor controller is a slave to the overall chiller controller. It gets a signal from the chiller master controller that is based on required leaving chilled water temperature asks the compressor to increase or reduce its capacity. Compressor control in the speed only control area of the map is straightforward. It is a single-input-single-output (SISO) control system. Each head-flow combination in the speed-only control area of the map has a unique speed and a unique efficiency that is represented by speed lines and efficiency islands. Speed is increased or decreased if more or less capacity is required. A new head-flow point on the map is reached with its own speed resulting in its own efficiency. With proper PI controller settings the compressor control loop will respond to required changes in capacity without too much overshoot or too slow of a response. The speed control loop also, although indirectly, adjusts for changes in head. For example, if the compressor has to deliver higher head because of an increase in condenser pressure as a result of higher ambient temperature, the compressor flow rate will initially decrease. The resulting chiller capacity reduction will be noticed by the chiller master controller which will then ask the compressor to increase its capacity.

A variable-speed compressor needs a mechanical or aerodynamic variation in order to cover the total chiller operating area. Figure 2 shows how the stable operating area of a 350 ton two-stage centrifugal refrigeration compressor is increased with internal flow recirculation controlled by an external valve as described by Brasz (2014).

Each head/flow combination in the extended range area of the map has two control inputs (speed and valve opening) and can be realized by an infinite number of speed/valve opening combinations. This allows selection of that speed/valve opening combination that for the given point on the map produces the highest efficiency. The single-input-single-output (SISO) control system in the speed only control area has now been replaced with a patent pending, Thornton (2014) multi-input-multi-output (MIMO) control system with speed and valve opening as inputs and capacity and optimum efficiency as outputs. In other the MIMO controller finds that combination of speed and valve opening that results in the best efficiency for a given head/flow combination.

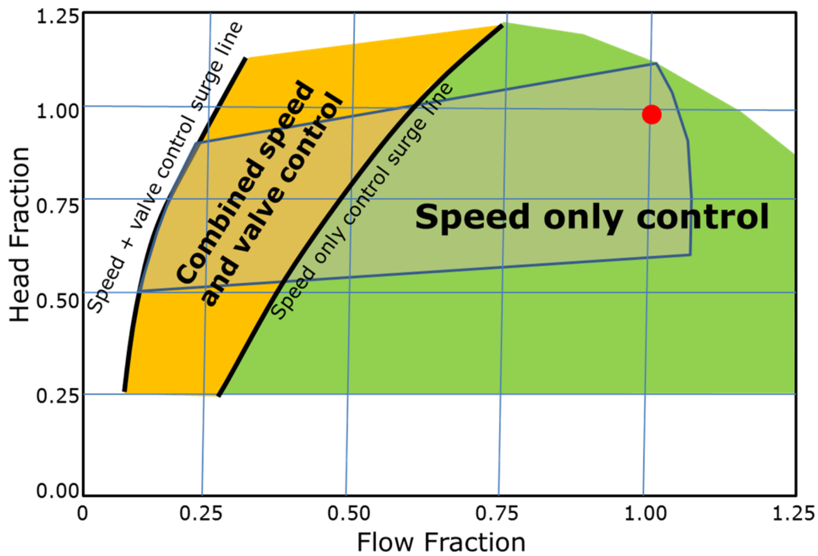

Figure 2. Compressor map area with speed-only control and extended map area with combined speed and valve control.

2. DETERMINATION OF OPTIMUM EFFICIENCY

When a centrifugal compressor is operating with flow recirculation for range extension the peak efficiency is reached when the least amount of flow is recirculated while keeping the compressor safe from surging. Rotating stall disturbances begin as a compressor nears surge. These rotating stall cells increase in intensity at higher head eventually transitioning into compressor surge. Peak efficiency for a given valve opening is reached between weak and medium rotating stall. This peak efficiency is achieved because the compressor is recirculating the least amount of flow possible without surging. Figure 3 shows lines with different percentages recirculating flow valve openings for one compressor speed. Opening the recirculating flow valve results in compressor curves to the left of the speed line with the valve closed, showing the range extension that is obtained this way.

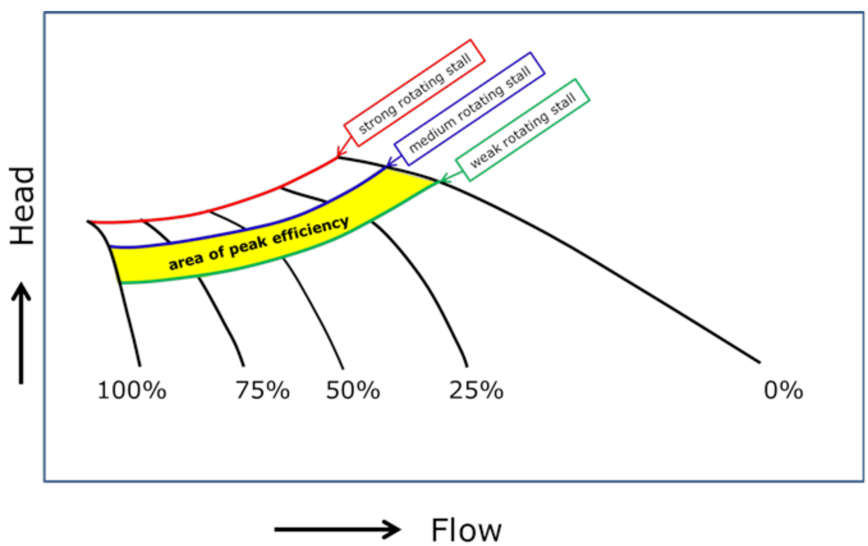

Figure 3. Peak efficiency for different recirculation valve openings at a given compressor speed

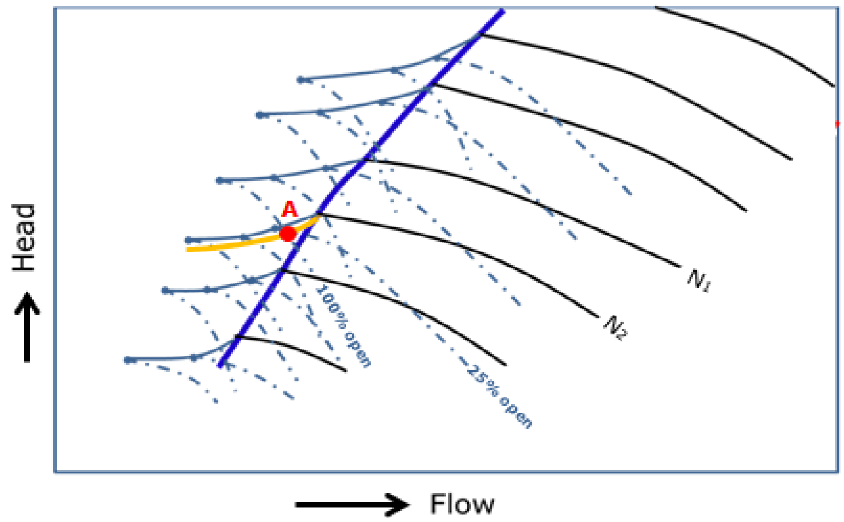

Figure 4. Gives an illustration of multiple speed/valve opening combinations to realize point A. The efficiency at point A is larger with speed N2 and 25% valve opening than with speed N1 and 100% valve opening.

The area of combined speed and valve control allows multiple combinations of speed and valve opening to reach a certain flow at a required head. The controller has to determine the combination of speed and valve opening which results in the highest efficiency. Knowing that peak efficiency occurs close to surge as shown in Figure 3 allows the selection of the appropriate speed and valve opening combination. Figure 4 illustrates that the efficiency of point A is highest when run at lower speed N2 with less (25%) valve opening instead of running at higher speed N1 with 100% valve opening.

3. MAGNETIC BEARING ORBITS

A model-based feed-forward MIMO control scheme could be implemented to run the compressor at peak efficiency in the combined speed and flow valve control area of the compressor map. This would require accurate knowledge of the location of the speed and valve opening lines (the solid and dashed lines in Figure 4) of the compressor. Some safety margin would have to be built into such a model-based control scheme to protect the compressor from going into surge as a result of inevitable model inaccuracies. This will result in some loss of efficiency.

Active magnetic bearings measure and control continuously the radial and axial position of the magnetically levitated shaft. The bearing orbits are continuously monitored. At compressor operating conditions far away from surge the orbit is small. It increases in size when approaching mild rotating stall close to compressor peak efficiency. The rotating stall flow instability exerts a fluctuating radial force on the shaft which is measured by the radial bearing positioning sensors and acted upon by the magnetic bearing controller. This disturbance gets larger when the rotating stall instabilities get stronger. The size of the bearing orbit increases with increased rotating stall compressor instability. Using radial bearing orbit size information that is already measured to control the active magnetic bearings we can determine whether the compressor is running close to its peak efficiency and far enough away from surge. This allows a feedback control loop for compressor efficiency optimization. This control system is implemented in the following way:

Ideally, the shaft is perfectly centered as shown in Figure 5. That situation occurs when the compressor is running far away from surge beyond its peak efficiency. It is interesting to realize that the distance between the outer diameter of the centered shaft and the physical limit where the shaft hits one of the touch-down rolling element bearings is on the magnitude of about 0.1 mm.

When the compressor reaches peak efficiency there are some small shaft vibrations that position the shaft slightly off-center. The shaft will cross Limit 1 but will stay within Limit 2. Limit 1 corresponds to weak rotating stall, Limit 2 to medium rotating stall and compressor peak efficiency occurs when the shaft orbit exceeds Limit 1 but stays within Limit 2. Figure 6 relates bearing orbit position to the location of the operating point relative to compressor surge.

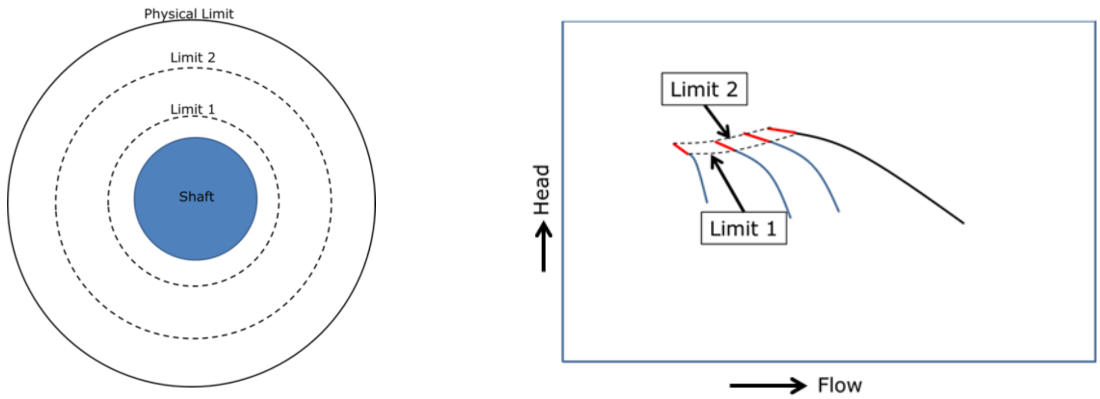

Figure 5. Graphical representation of bearing orbit limits 1 and 2 and their location on the compressor map.

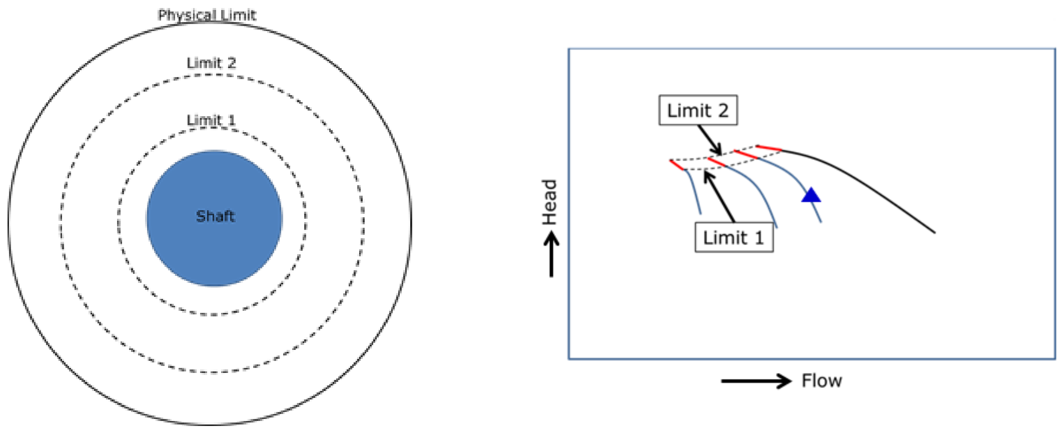

Figure 6. Relationship between bearing orbit position and operating point location on the compressor map with orbit smaller than limit 1

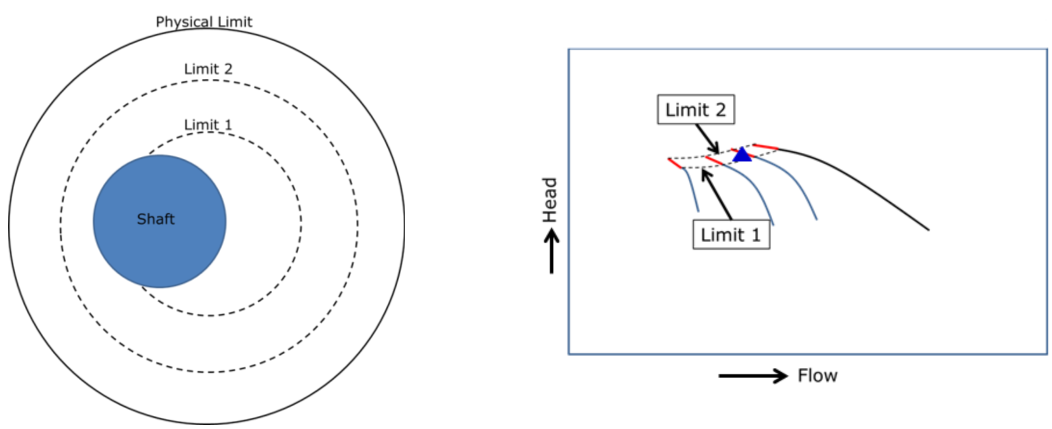

Figure 7. Relationship between bearing orbit position and operating point location on the compressor map for orbit between limit 1 and limit 2

4. CONTROL ACTIONS

When controlling the compressor in order to maintain peak efficiency the most important action is to keep the compressor safe from surge. This takes precedence even over maintaining capacity. When operating in the region of the compressor map where the recirculation valve is required, any time the orbit increases above limit 2 the speed will increase and push the compressor back to a safe operating region. This action is shown in Figure 7, where the head rises from point 1 to point 2. At point 2 the orbit is greater than limit 2, and the compressor must increase speed from N1 to N2 going from point 2 to point 3.

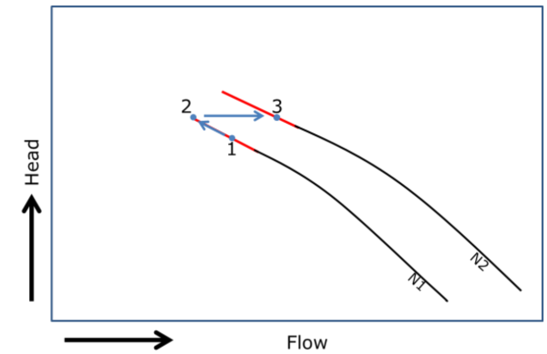

Figure 7. Compressor increases speed when orbit is greater than limit 2.

This increase in speed whenever orbit is greater than limit 2 is required regardless of the compressor demand, and it occurs whether the compressor is loading, unloading, or meeting the demand.

When the compressor is operating with the recirculation valve, whether loading or unloading there are three possible orbit conditions that can occur. These conditions correspond to points 1, 2, and 3 in Figure 8, respectively.

1) Orbit > Limit 2
2) Limit 1< orbit < Limit 2
3) Orbit < Limit 1

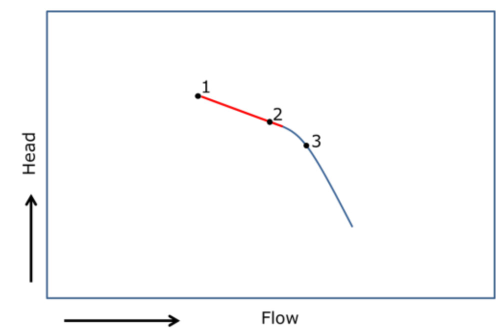

Figure 8. Three conditions for compressor loading and unloading.

The first case (point 1 in Figure 8) when orbit is greater than limit 2 the compressor will always increase speed, this is done regardless of if the compressor is loading or unloading. The second case (point 2) represents when the orbit is between limit 1 and limit 2, indicating that the compressor is operating with a weak rotating stall and near the peak efficiency for the given condition. When this is the case the compressor adjusts capacity by either opening the flow recirculation valve to reduce capacity or closing the recirculation valve to increase capacity. In this case the speed remains constant.

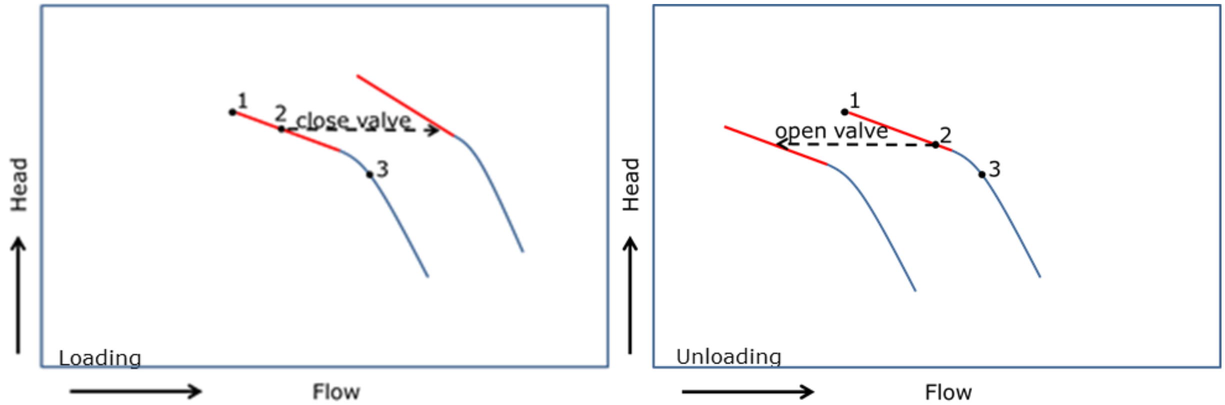

Figure 9. Compressor increases capacity (left) and decreases capacity (right) by adjusting only recirculation valve.

The third condition (point 3) is when the compressor needs to adjust capacity, when operating far from surge. This represents a low efficiency point, with orbit less than limit 1. When the compressor reduces the capacity from this point 3 the reaction is to simply reduce the speed. This can be seen on the right side of figure 10. The compressor slows down in this scenario in order to reduce capacity but also move back to peak efficiency, with mild rotating stall and a larger orbit.

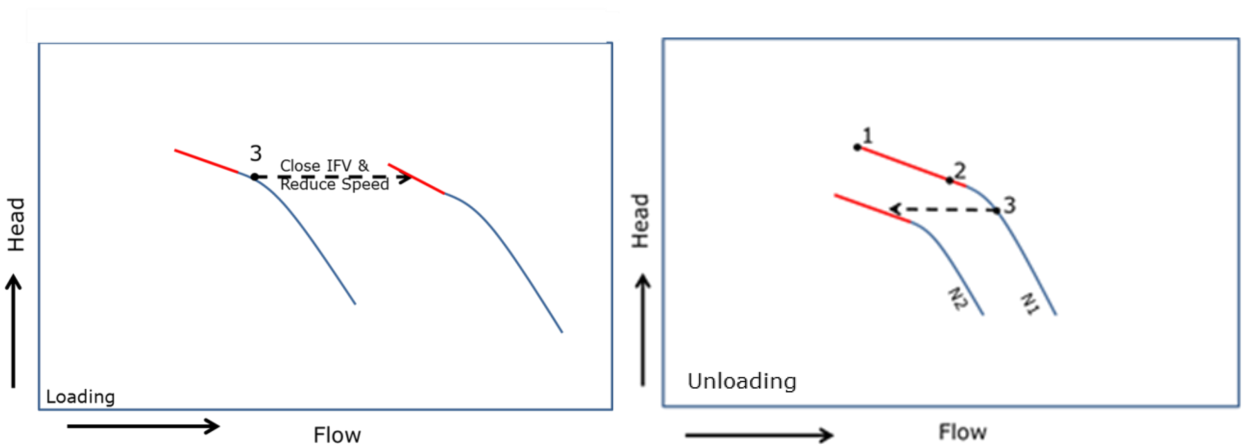

Figure 10. Compressor increasing capacity (left) and decreasing capacity (right) when the orbit is less than limit 1

When the compressor is loading with orbit less than limit 1 the reaction is more complicated. In order to maintain peak efficiency and also increase capacity the compressor will combine a reduction in speed and closing the recirculation valve. This dual action is required because either of these two actions individually will not give the increase in capacity and maintain peak efficiency. If the only action taken were to close the recirculation valve the compressor would then operate even further away from the peak efficiency. Similarly, if the compressor were to only reduce speed the flow would actually decrease resulting in less capacity. Therefore it is only through a combination of speed reduction and closing the recirculation valve that the compressor can increase capacity and maintain peak efficiency.

5. SUMMARY AND CONCLUSIONS

- Internal flow recirculation extends the stable operating range of two-stage variable speed centrifugal compressors.
- The addition of flow recirculation to compressor speed complicates the compressor control system by requiring control of two independent variables.
- Each desired head/flow combination can now be realized by an infinite number of speed/recirculation combinations, resulting in different compressor efficiencies for the same operating point.

- The control system should find that combination of compressor speed and flow recirculation that results in the highest compressor efficiency
- A feedback control scheme has been developed that optimizes compressor efficiency while delivering the required capacity.
- The orbit of the magnetic bearing has been identified as the feedback signal used to optimize compressor efficiency.
- The operation of this control system has been illustrated for the various possible changes in cooling demand and head a centrifugal compressor might encounter.
- This control system has been successfully implemented on a centrifugal compressor. Testing is currently underway to guarantee stable controller operation at all possible compressor operating conditions. However, due to the nature of the control it was not possible to present quantitative test data to validate the control scheme.

REFERENCES

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