ABSTRACT
Cavitation instabilities due to low inlet pressures and high rotational speeds have limited the allowable operating conditions of pumps and inducers. Researches have been focused on improving inducer designs to delay the onset of cavitation and allow operation at lower inlet pressures. Computational fluid dynamic simulations were employed to explore a new technology known as a stability control device. With the implementation of the stability control device, an inducer was able to operate stably below the design flow coefficient and the suction performance increased by a factor of two.

NOMENCLATURE

- $A_{LE}$: Cross-sectional area at the inducer leading edge
- $k$: Mass flow gain factor
- $m_{inlet}$: Inlet mass flow rate
- $m_{LE}$: Inducer leading edge mass flow rate
- $N_{ss}$: Net suction specific speed
- $P_v$: Water vapor pressure
- $P_{00}$: Inlet total pressure
- $P_{02}$: Total pressure just downstream of the inducer
- $Q$: Volume flow rate through the inducer
- $U_{tip}$: Velocity of the inducer blade tip
- $\beta$: Water flow angle at the inducer leading edge
- $\beta_{blade}$: Leading edge inducer blade angle
- $\gamma$: Incidence angle
- $\rho$: Water density
- $\sigma$: Cavitation number
- $\sigma_b$: Breakdown cavitation number
- $\phi$: Flow coefficient
- $\psi$: Head coefficient
- $\omega$: Inducer angular velocity

INTRODUCTION
Cavitation in pumps is detrimental to the machine performance. In order to mitigate the effects of cavitation, an inducer is often the first stage of high suction pumps. The inducer operates under cavitating conditions and increases the pressure of the flow sufficiently such that cavitation does not occur in the rest of the pump. Significant cavitation instabilities ultimately can lead to failure in the inducer.

At low inlet pressures, the high rotational speeds of the inducer blades accelerate the fluid to high velocities, dropping the local static pressure below the vapor pressure, resulting in cavitation. At low inlet pressures, the cavitation on the blades increases to the point that cavitation can fully block the passage between the blades [1]. The blades stall and the performance of the inducer drops rapidly. This phenomenon is referred to as machine breakdown. It can be best described as a sharp decrease in head coefficient with small changes in cavitation number. The head coefficient, equation 1, is a non-dimensional number to express the pressure rise through the machine. $P_{02}$ is the pressure just downstream of the inducer, $P_{00}$ is the inlet total pressure, $\rho$ is the water density, and $U_{tip}$ is the inducer blade tip speed.

$$\psi = \frac{P_{02} - P_{00}}{\rho U_{tip}^2}$$

The cavitation number, $\sigma$, is a non-dimensional number...
used to determine how susceptible the flow is to cavitation and is expressed in equation 2, where $P_i$ is the water vapor pressure.

$$\sigma = \frac{P_{00} - P_i}{\frac{1}{2} \rho U_{tip}^2}$$ (2)

The amount of cavitation in the inducer is inversely proportional to the cavitation number. As the inlet pressure approaches the fluid vapor pressure, the amount of cavitation increases and can lead to a significant cavitation events and pump failure.

Cavitation can also cause high radial loads, shaft vibrations, and blade flapping [2][3]. Asymmetric cavitation patterns on the blade create an imbalance of forces on the blade. The magnitude and oscillations of this force as the blade rotates can lead to structural failure at cavitation numbers greater than the breakdown cavitation number of the machine. Cavitation surge is when large cavities of cavitation grow and collapse. This acts as a source and sink in the domain and can create large oscillations in the mass flow rate through the pump. Fluctuations in the mass flow rate can create large vibrations in the entire system and again lead to pump failure. Large pressure spikes are also known to be associated with the collapse of cavitation. The pressure waves sent through the machine result in large forces that can cause structural failure [2].

Cavitation instabilities are introduced in the flow by one of two mechanisms. The first is characterized by a decrease in the cavitation number. At lower cavitation numbers, the inlet pressure becomes lower and the fluid is more susceptible to cavitate. Cavitation instabilities are also introduced into the flow by operating the pump at off-design flow coefficients. The flow coefficient, equation 3, is a the volumetric flow rate, $Q$, divided by the cross-sectional area and the leading edge of the blade, $A_{LE}$, and the blade tip speed.

$$\phi = \frac{Q}{A_{LE}U_{tip}}$$ (3)

The inlet blade angle of inducers are optimized for the specific flow field of an inducer. The difference between the inlet blade angle, $\beta_{blade}$, and the flow angle, $\beta$, is known as the incidence angle, $\gamma$. Large incidence angles lead to flow separation on the blade. Strong vortices are produced when the flow separates and often leads to cavitation inception and significant cavitation events. Low incidence angles are also problematic. Cavitation grows on the blades as the cavitation number drops. At higher incidence angles, the cavitation only grows on one side of the blade. At low incidence angles, cavitation can grow on both sides of the blade, leading to blade passage blockage at higher cavity numbers. For these reasons, the incidence angle is typically designed to be approximately 3$^\circ$ [4].

If the inducer operates at a flow coefficient other than the designed value, the flow field of the inducer, and thus the flow angle is changed. At flow coefficients below the inducer the axial velocity decreases and the flow angle also decreases leading to an increase in incidence. The increased incidence produces large flow separation at the blade tip and a strong vortices that lead to pump failure.

Suppressing cavitation instabilities is vital to improving the suction performance of inducers. It is well known that the leading edge sweep has a favorable effect on the cavitation performance of inducers [5]. Acosta showed that the incidence angle in the cross-flow plane is the main factor in enhanced cavitation stability due to a blade sweep [6]. Shimiya analyzed the effects of a shallow groove at the blade tip in the casing. Results showed that the occurrence region of major cavitation instabilities was diminished [7]. Blade tip clearance has been studied by Torre [8]. Different tip clearance configurations cause different cavitation behavior. For higher blade clearance, breakdown starts at an higher cavitation number but the performance drop is more gradual. Fuji studied the effects of the tip geometry on cavitation performance. It was determined that a sharp edge on the blade provided the best cavitation performance [9]. Kimura explored how variations in the shroud geometry can also be useful in suppressing cavitation. Increasing the diameter of the shroud before the leading edge of the blade up until the blade is known as a gutter section. It was shown to confine the backflow to the gutter and suppress cavitation surge [10]. Stangeland used a jet injection at the inducer inlet as a method of suppressing the reversed flow at the blade tip [11].

All of these techniques have been proven to delay the onset of significant cavitation instabilities and to allow an inducer to operate at lower cavitation numbers, specifically at the design flow coefficient. This increases the suction performance. Even with implementing these advances in inducer design, at off-design flow coefficients, large incidence angles produce extreme flow conditions with significant cavitation. This restricts the operation of inducers to be very near the design flow coefficient.

The theoretical limit of pump suction performance is known as the Brumfield criterion [12]. The Brumfield line, shown in Fig.1, shows the theoretical maximum net suction specific speed, $N_{ss}$, for an inducer as a function of $\phi$. Net suction specific speed, defined in equation 4, is a measure of the suction performance of an inducer. Where $\omega$ is the rotation speed of the blade and $m_{inlet}$ is the inlet mass flow rate.

$$N_{ss} = \frac{\omega \sqrt{m_{inlet}}/\rho}{(P_{00} - P_i)/\rho^{0.75}}$$ (4)

Inducer suction performance increases gradually with decreasing flow coefficient for $\phi > 0.5$. As the flow coefficient
FIGURE 1. THEORETICAL MAXIMUM NET SUCTION SPECIFIC SPEED, KNOWN AS THE BRUMFIELD CRITERION, AS A FUNCTION OF FLOW COEFFICIENT.

continues to decrease, the suction performance increases rapidly. Significant improvements in the suction performance of inducer can be made by operating inducers at lower flow coefficients.

The typical design flow coefficient for high suction inducers is near \( \phi \approx 0.07 \). The design flow coefficient is limited by structural limitations. Low flow coefficients have small flow angles and require small inlet blade angles on the inducer. Small blade angles decrease the flow area between the blades and make the inducer more susceptible to cavitation blockage at higher cavitation numbers. To compensate, inducer blades have become thinner to maximize the flow area; however, thin blades are more susceptible to structural damage. Unsteady flow conditions caused by asymmetrical cavitation can lead to blade flapping and structural failure [3].

In order to significantly increase the suction performance, inducers need to be able to operate stably at flow coefficients below \( \phi \approx 0.05 \). Structural limitations prevent high suction inducer from being designed at flow coefficients that low and cavitation instabilities prevent current inducers from operating stably at flow coefficients far below the designed value. Sloteman designed a backflow recirculator, capable of allowing an inducer to operate stably even at flow coefficients far below the design flow coefficient [13]. Japikse of Concepts NREC patented a similar device known as stability control device (SCD) [14].

In Fig. 2, the geometry of an inducer with an SCD is shown. The device operates by capturing fluid from the region of back flow near the leading edge of the inducer and reintroduces it into the flow upstream. The local mass flow rate between the re-injection location and the bleed slot (where the fluid enters the SCD near the inducer blade tip) is the sum of the inlet mass flow rate and the mass flow rate through the SCD. The mass flow gain factor, \( k \), is the ratio of the mass flow rate at the inducer leading edge to the mass flow rate at the inlet of the inducer (equation 5).

\[
k = \frac{\dot{m}_{\text{LE}} + \dot{m}_{\text{inlet}}}{\dot{m}_{\text{inlet}}} \tag{5}\]

Krise employed single-phase, steady-state numerical simulations to explore the effects of an SCD on a flat plate inducer [15]. His results showed that the onset of breakdown was delayed to a lower cavitation number when an SCD was implemented. This paper explores the stabilizing effect of an SCD on a state-of-the-art inducer (referred to as the baseline inducer).

Methods

Computational fluid dynamic (CFD) simulations were conducted on the baseline inducer with and without an SCD. Simulations were conducted at the design flow coefficient, \( \phi = 0.07 \), and at two off-design flow coefficients lower than the design value. Star-CCM+ was employed as the CFD solver for the three-dimensional, time-accurate, multiphase simulations. Turbulence was modeled using the realizable Two-Layer K-\( \varepsilon \) model. The two-layer approach employs wall functions to compute the boundary layer at high wall \( y^+ \) regions of the mesh, and it is assumed that the viscous sublayer is resolved by the mesh at low \( y^+ \) values. In the Multiphase Segregated Flow model, each distinct phase has its own set of conservation equations. Phases are assumed to coexist or be in an interpenetrating continua. The Volume of Fluid multiphase model was used to model the cavitating behavior of the regime. Cavitation is defined as the multiphase interaction between water and water vapor, using the basic Rayleigh-Plesset formulation [16].

A cross-section of part of the computational domain is shown in Fig. 2. The geometry consists of the full inducer geometry with the shroud extending 12 diameters upstream of the

FIGURE 2. SCHEMATIC ILLUSTRATION OF AN INDUCER WITH A STABILITY CONTROL DEVICE
TABLE 1. COMPARISON OF IMPORTANT MONITORS FOR CONVERGED SOLUTIONS WITH VARYING TIME STEPS FOR THE INDUCER WITH AN SCD AT THE DESIGN FLOW COEFFICIENT AND OUTLET PRESSURE OF 300 KPA. PERCENT CHANGES IN THE MONITORS WERE CALCULATED FROM THE \( \frac{1}{2} \)° PER TIME STEP SOLUTION.

<table>
<thead>
<tr>
<th>Time Step</th>
<th>( \sigma )</th>
<th>( \psi )</th>
<th>( k )</th>
</tr>
</thead>
<tbody>
<tr>
<td>4°</td>
<td>-1.93%</td>
<td>-0.26%</td>
<td>-1.8%</td>
</tr>
<tr>
<td>1°</td>
<td>-1.34%</td>
<td>-0.74%</td>
<td>-1.5%</td>
</tr>
<tr>
<td>1/2°</td>
<td>0.0223</td>
<td>0.272</td>
<td>1.11</td>
</tr>
</tbody>
</table>

inducer and four diameters downstream. Meshes were generated using the internal meshing tools inside of Star-CCM+. The meshes contained 6.4 million and 6.6 million polyhedral cells for the inducer without and with the SCD respectively. The mesh was sufficiently fine to apply the low wall \( y^+ \) treatment with an average wall \( y^+ \) value of 17. A grid independence study was performed on the inducer with an SCD increasing the number of cells to 12.2 million cells. After both simulations had converged, critical monitors such as the head coefficient only varied by 0.34%. It was determined that the mesh was sufficiently fine to accurately predict the performance of the inducer.

A time dependence study was also performed. Time steps of 4°, 1°, and 1/2° of inducer rotation per time step were all considered. In table 1, the differences in critical monitors for converged solutions are shown for the inducer with an SCD at the design flow coefficient and an outlet pressure of 300 kPa. There is little difference between the time averaged values. The noise in the monitors was greatly reduced by decreasing the time step. In order to detect time-accurate cavitation instabilities, the 1/2° per time step was selected for the study.

Convergence was determined on case by case basis by evaluating a monitors of the inlet total pressure, the inducer exit pressure, the rotor torque, the domain volume fraction of vapor, and the rotodynamic forces on the blades. When the monitors had become flat or periodic for greater than 10 revolutions the simulation was determined to be converged.

Data was obtained for each breakdown curve by first solving a non-cavitating, steady simulation. This was done for all scenarios, at each flow coefficient. Multiphase simulations were then conducted starting at a high back pressure where virtually no cavitation was present in the domain. Subsequent simulations were obtained by marching the back pressure down until breakdown occurred. On average, each data point obtained required 128 processors and three weeks for convergence. In total, there were over 70 different simulations conducted for this study.

### Results and Discussion

Inducer performance is often characterized by evaluating the machine breakdown curve, which plots the head coefficient versus the cavitation number at a fixed flow coefficient. Shown in Fig. 3 are machine breakdown curves for the baseline inducer without an SCD at 100% and 60% of the design flow coefficient. Also shown are results for the baseline inducer with an SCD at 100%, 60%, and 40% of the design flow coefficient.

At the design flow coefficient, \( \phi = 0.07 \), this inducer has very similar machine breakdown curves with and without an SCD present in the geometry. The point of breakdown is referred to as the breakdown cavitation number, \( \sigma_b \), and is defined as the point at which the head coefficient has dropped to 90% of the non-cavitating head coefficient value. Breakdown occurs at \( \sigma \approx 0.023 \) for both scenarios at the design flow coefficient and the head coefficient prior to breakdown is nominally 5% higher for the scenario with the SCD.

At 60% of the design flow coefficient, \( \phi = 0.042 \), significant differences in the machine breakdown curves for the two scenarios exist. For the baseline inducer without the SCD, breakdown starts at a relatively high cavitation number, \( \sigma \approx 0.049 \). The breakdown profile decreases in steps. After the initial decrease in head coefficient, the head coefficient remains constant at \( \psi \approx 0.32 \) until \( \sigma \approx 0.017 \), where the head coefficient begins to drop rapidly again with small changes in cavitation number. At all cavitation numbers significant cavitation instabilities were present.

Figure 4 shows the forces in the y-direction versus the forces.
in the x-direction on the inducer over time for the inducer without an SCD at $\sigma \approx 0.33$ and $\phi = 0.042$. This kind of plot is referred to as an orbit plot. Large forces can displace the inducer from its center position creating irregular shaft orbits. The rms of the forces on the rotor is nominally 5 N with periodic spikes up to as high as 14 N. This plot illustrates the time varying nature of the forces even at high cavitation numbers where no cavitation is present in the domain. At lower cavitation numbers, the rotodynamic forces increase in magnitude but the same periodic spikes are present at all cavitation numbers.

By contrast, for the SCD scenario, the head coefficient of the inducer is similar to the no SCD scenario at high cavitation numbers ($\sigma \geq 0.06$). However, the point of breakdown is delayed to a much lower cavitation number, $\sigma \approx 0.013$. Further, the inducer never experienced large pressure spikes as the scenario without an SCD did.

In operation, the inducer without the SCD would be unable to operate at 60% flow because of the extreme flow conditions that prevail. Implementing an SCD in the inducer design allows stable operation to exist. To further explore the stabilizing capabilities of an SCD, the inducer with an SCD was simulated at 40% of the design flow coefficient. For this flow coefficient, breakdown occurs at $\sigma \approx 0.011$. Table 2 summarizes the point in which breakdown occurs for both scenarios, the inducer with and without and SCD, at all flow coefficients.

Comparing the 40% breakdown curve in Fig. 3 to the 100% curve, the drop in performance below the breakdown cavitation number is more gradual for the 40% case. At the design flow coefficient, the breakdown curve is vertical at the breakdown cavitation number. This implies that small changes in the inlet pressure would result in a large drop in head rise across the inducer, resulting in complete machine failure. Operating this inducer at the design flow coefficient at or below the breakdown cavitation number would not be advisable. In contrast at 40% flow, the head coefficient is gradually decreasing in performance at cavitation numbers as low as $\sigma \approx 0.005$. The corresponding inlet total pressure at this cavitation number is the same order of magnitude as the water vapor pressure. The gradual decrease in head coefficient would allow operation of the inducer below the breakdown cavitation number.

Including an SCD in the inducer design increases the ability of the inducer to operate stably at low, off-design flow coefficients. Recall from Fig. 1 that greater suction performance is achievable at lower flow coefficients. The inducer without an SCD had a maximum suction performance of $N_{ss} \approx 12.5$. The suction performance of the same inducer with the SCD increased to $N_{ss} \approx 23$. Figure 5 shows the Brumfield Criterion plotted with the maximum suction specific speed for the inducer with and without the SCD at all of the flow coefficients explored. At $\phi = 0.07$, the suction performance is the same, just as the breakdown curves were virtually the same in Fig. 3. At $\phi = 0.042$, the scenario without an inducer does not show improvement in the suction performance as expected. This is because of the high incidence angles that are present and the extreme cavitation conditions.

The scenario with an SCD had increased in suction performance at $\phi = 0.042$ by nearly 40%. With an SCD the inducer was also able to operate stably at $\phi = 0.028$ where the maximum suction performance was achieved. The data for the inducer with an SCD follows the Brumfield Criterion. However, there is room for improvement in the SCD design and it is believed that some modifications to the SCD design could allow an inducer to operate at a suction performance beyond the theoretical limit.

**TABLE 2. A SUMMARY OF THE BREAKDOWN CAVITATION NUMBER FOR THE INDUCER WITH AND WITHOUT AN SCD AT ALL FLOW COEFFICIENT SIMULATED.**

<table>
<thead>
<tr>
<th>Inducer Geometry</th>
<th>$\phi$</th>
<th>$\sigma_b$</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO SCD</td>
<td>0.07</td>
<td>0.023</td>
</tr>
<tr>
<td>NO SCD</td>
<td>0.042</td>
<td>0.049</td>
</tr>
<tr>
<td>SCD</td>
<td>0.07</td>
<td>0.023</td>
</tr>
<tr>
<td>SCD</td>
<td>0.042</td>
<td>0.013</td>
</tr>
<tr>
<td>SCD</td>
<td>0.028</td>
<td>0.011</td>
</tr>
</tbody>
</table>
FIGURE 5. THE ACHIEVABLE SUCTION PERFORMANCE FOR THE INDUCER WITH AND WITHOUT AN SCD AT ALL OF THE FLOW COEFFICIENTS EXPLORED COMPARED TO THE BRUMFIELD CRITERION.

Conclusions
An SCD was invented to improve the suction performance of high speed pumps. The same inducer with and without an SCD was analyzed at three flow coefficients.

At the design flow coefficient, the implementation of an SCD had very little effect on the performance of the inducer. Breakdown occurred at $\sigma \approx 0.023$ for scenarios with and without the SCD. At 60% of the design flow coefficient, the stabilizing effect of an SCD is more apparent. The scenario with an SCD greatly reduced the cavitation instabilities, resulted in negligible rotordynamic forces and improved suction performance, with breakdown occurring at $\sigma \approx 0.013$. In comparison, the inducer without an SCD at 60% of the design flow coefficient had significant cavitation instabilities present at all cavitation numbers, large, temporally oscillating rotordynamic forces, and breakdown occurred at $\sigma \approx 0.049$.

The stabilizing effect of an SCD allows the inducer to operate stably at coefficients far below the design flow coefficient. The inducer design with an SCD operated stably at both 60% and 40% of the design flow coefficient. Much greater suction performance was achieved with the SCD because of the ability to operate at lower flow coefficients. In general, high speed pumps that incorporate an SCD into the design will be able to operate stably at significantly lower inlet pressures.

REFERENCES