Enhancing pump performance

Cavitation occurs in centrifugal pumps when the net positive suction head available (NPSHa) is lower than the net positive suction head required (NPSHr), causing the formation and accumulation of bubbles at pump inlet that collapse and result in a series of mini implosions. Cavitation can occur in different locations within pump hydraulics and can cause significant damage to the pump's internal components. The cavitation behaviour of cryogenic centrifugal pumps is an important performance criterion for safe operation in the event of a lack of suction pressure. Inducer technology is often used in pump applications to delay cavitation or to improve suction performance by reducing the NPSHr.

One of the most common applications for cryogenic centrifugal pumps is inside LNG storage tanks, where NPSHr determines the non-useable height of the liquid being stored. In-tank pumps are vertically suspended within a discharge column, mainly used to transfer the fluid within the tank. Due to their size and construction, storage tanks cannot be pressurised, which limits available suction pressure to the pump. A helical style inducer is installed at the lowest section of the pump to enhance suction performance (Figure 1).

This case study compares the cavitation performance of an in-tank LNG cryogenic pump with a constant pitch inducer and a pump configuration with a variable pitch inducer. Computational fluid dynamics (CFD) simulations and experimental performance tests were conducted to determine the cavitating (two-phase) and non-cavitating (single-phase) performance of the two inducers under operating conditions. NPSH with a 3% differential head drop (NPSH3) was used as a criterion to identify the true cavitation performance of each inducer configuration. Replacing the constant pitch inducer with a variable pitch inducer resulted in a 25% improvement in NPSH3 performance with minimal impact to single-phase pump performance.

Inducer design for in-tank cryogenic pumps

Due to cost and manufacturing constraints, storage tanks for any hydrocarbon application (LNG, methane, propane, butane, etc.) are constructed to handle a maximum tank pressure of 300 mbar (4.35 psi) or less. Consequently, suction pressure of the pump is often dictated by the static height of the liquid. Inducer design is crucial for these applications as it ultimately determines the minimum liquid level in the tank required for safe and stable operation. The main objective is to reduce the NPSHa as much as possible so that the height of useable liquid inside the tank can be increased. It should be noted that the overall footprint of a storage tank can be as large as 200 m (656 ft) dia. Therefore, any small improvement in the magnitude of a few centimetres can result in substantial improvement in the useable amount of liquid.

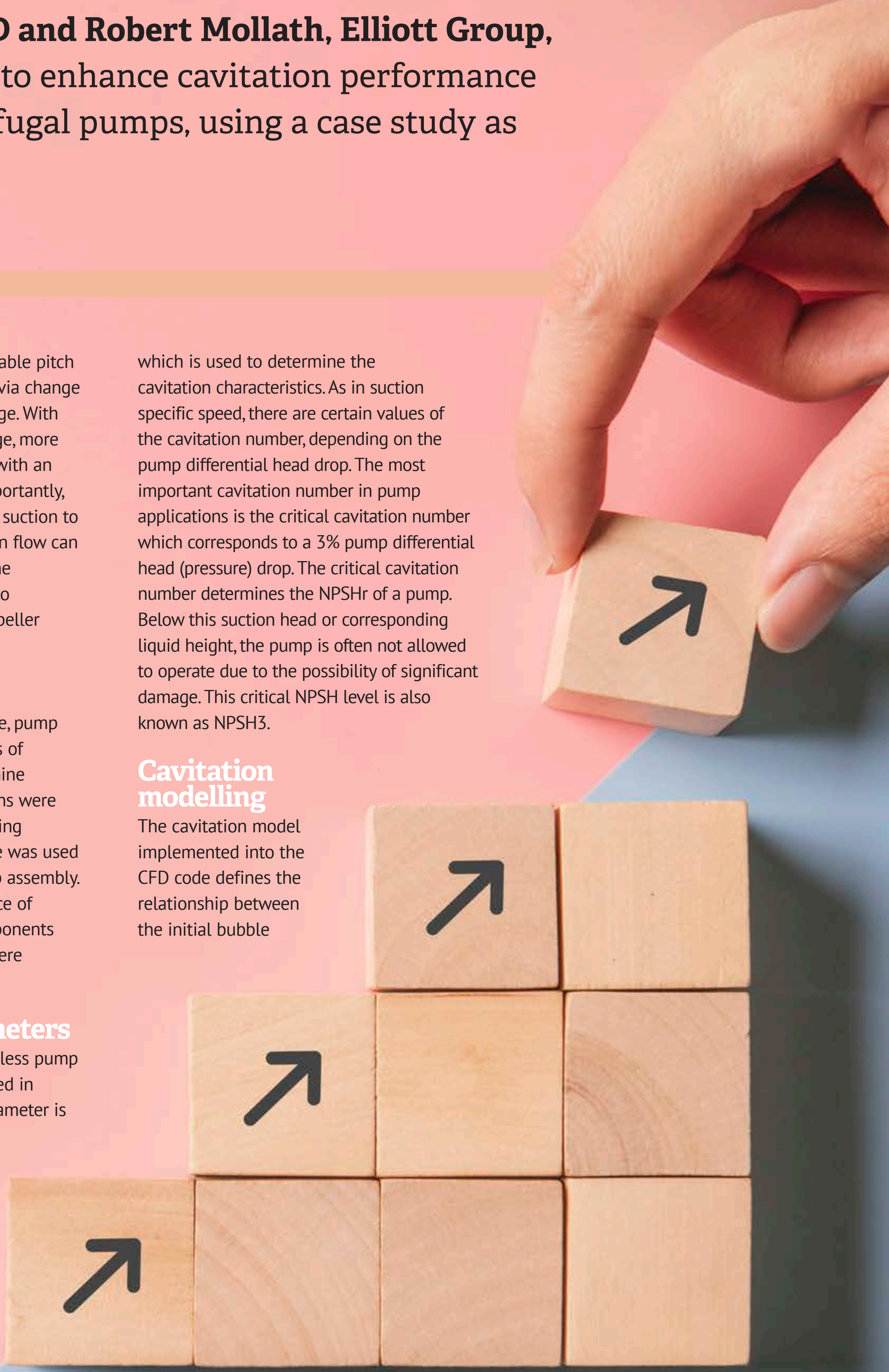
The subject retractable in-tank pump was designed to operate in propane and installed inside a storage tank. It is a vertically suspended two-stage cryogenic submerged motor pump with a helical inducer downstream of the suction impeller to enhance suction performance. All in-tank pump applications are furnished with a helical inducer to delay cavitation inception so that it can be safely operated at very low liquid levels with reduced suction pressure.

Under normal operating conditions, in-tank pumps often have enough suction head and there is no concern of cavitation during continuous operation. However, during the process of emptying the storage tank, cavitation performance dictates the liquid height. Although in-tank pumps are not continuously operated under very low suction head, in order to attain an acceptable low liquid level after the tank is emptied by the pump, NPSH is quite important for these particular applications.

The original pump design was constructed using a straight pitch constant hub diameter (constant pitch) inducer. Constant pitch inducers follow the flat plate design at which the inlet and outlet blade angles are the same. Therefore, overall head (pressure) increase mainly depends on the rotational speed, tip, and hub diameters. In this case, the blade angle is 8.9˚ at the tip diameter (Table 1). Hub and tip diameters are constant along the axial direction.

For the new (variable pitch) inducer design, the blade angle was increased from suction side to discharge side with constant change along the axial length. The inlet blade angle was determined based on the meridional flow at the suction section to ensure shockless entry to the inducer. In order to reduce the meridional flow at the suction side, the hub diameter was reduced and gradually increased towards the mid axial point. Blade count was increased from two to three to prevent any alternate blade cavitation. Blades were leaned forward to minimise the back leakage vortex that occurs between the tip of the inducer and the inlet casing bore. Table 1 shows the specifications of both the original and the new inducer design.

Enver Karakas PhD and Robert Mollath, Elliott Group, USA, describe how to enhance cavitation performance in cryogenic centrifugal pumps, using a case study as an example.



The main characteristic of the variable pitch inducer is the ability to increase head via change in blade angle from suction to discharge. With the increase in blade angle at discharge, more head can be produced by the inducer with an increase in absolute velocity. More importantly, with the variation in blade angle from suction to discharge, shockless entry at the design flow can be obtained. It should be noted that the variation in blade angle from suction to discharge is adjusted based on the impeller eye geometry.

CFD simulations

To establish a pump performance curve, pump performance was investigated in terms of non-cavitation performance. To determine cavitation performance, CFD simulations were conducted at rated flow rate with varying suction pressures. ANSYS CFX software was used to simulate the first stage of the pump assembly. In order to predict the true performance of the pump assembly, all hydraulic components (inducer, impeller, and diffuser vane) were included in the CFD model.

Pump cavitation parameters

There are two fundamental dimensionless pump cavitation performance parameters used in performance comparison. The first parameter is the suction specific speed, which is commonly used to identify the cavitation performance of a pump. It represents a non-dimensional version of the suction pressure. It should be noted that there are certain values of the suction specific speed depending on NPSH, and consequently, the amount of differential head loss.

The second parameter is the cavitation number, which is used to determine the cavitation characteristics. As in suction specific speed, there are certain values of the cavitation number, depending on the pump differential head drop. The most important cavitation number in pump applications is the critical cavitation number which corresponds to a 3% pump differential head (pressure) drop. The critical cavitation number determines the NPSHr of a pump. Below this suction head or corresponding liquid height, the pump is often not allowed to operate due to the possibility of significant damage. This critical NPSH level is also known as NPSH3.

Cavitation modelling

The cavitation model implemented into the CFD code defines the relationship between the initial bubble radius and the pressure around the bubble. The interphase mass transfer rate for condensation and vaporisation was defined using empirical constants, a proven method of predicting cavitation performance of inducers.

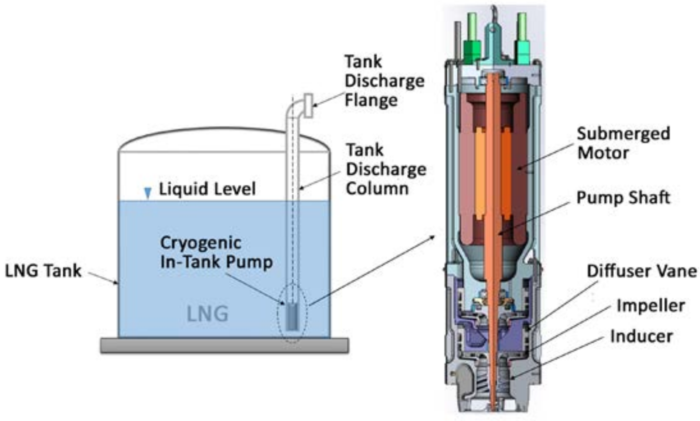


Figure 1. Typical configuration of an in-tank LNG cryogenic pump.

Table 1. Specifications of the original and new inducer design

	Constant pitch inducer	Variable pitch inducer
Rotational speed	3000 RPM	3000 RPM
Rated volumetric flow (Q)	238.5 m³/hr (1050 US GPM)	238.5 m³/hr (1050 US GPM)
Blade count (n)	2	3
Tip diameter (D_t)	196.8 mm (7.75 in.)	196.8 mm (7.75 in.)
Hub diameter (d)	70 mm (2.76 in.)	38 mm (1.5 in.) at suction; 70 mm (2.76 in.) at discharge
Blade angle at tip diameter (β)	8.9˚	7˚ at suction; 11˚ at discharge
Pitch (p)	100 mm (3.94 in.)/360˚ rotation	Variable pitch
Blade chord length (c)	611 mm (24.06 in.)	523 mm (20.54 in.)

Table 2. Boundary conditions and CFD simulation details

Inlet boundary condition	Total pressure in stationary frame: varied for cavitation simulations
Outlet boundary condition	Mass flow rate: Varied for non-cavitation simulations
Fluid	Propane at -45˚C (-49˚F) with P_{vap} = 89051 Pa (12.92 psi)
Rational speed	3000 RPM (applicable to inducer and impeller)
Interface model between rotating and stationary components	Multi frame of reference
Analysis type	RANS steady state with 'false' time scale
Turbulence model	k-ε turbulence model
Multiphase fluid model	Homogeneous model (equal temperature, equal velocity)
Heat transfer	Isothermal at -45˚C (-49˚F)
Fluid pair mass transfer option	Cavitation model according to described mass transfer equations

Boundary conditions

Table 2 outlines the boundary conditions and simulation details. Thermodynamic and transport properties of vapour and liquid are defined as a function of temperature and pressure in accordance with saturation and sub-cool tables published by the National Institute of Standards and Technology (NIST).

Test setup and procedure

Tests were conducted at Elliott Group's Cryogenic Test Facility to determine the cavitating and non-cavitating performance of each pump assembly. The tests were carried out at a constant temperature level with propane at cryogenic condition.

For non-cavitation performance testing, each pump was operated at several different flow rates according to API 610 guidelines [16], and pump differential head input power was recorded while pump rotational speed, vibration levels, liquid temperature, and other critical test parameters were closely monitored. During non-cavitation performance testing, system pressure was adjusted to maintain at least 5˚ of sub-cooling at pump suction to prevent any vapour formation. Pump efficiency was calculated based on the recorded hydraulic output power based on flow and head, and measured electrical input power to the pump assembly considering the motor efficiency.

Each pump was tested to determine the NPSH3 level according to API 610 guidelines. Pumps were adjusted to operate at the desired flow rate with sufficient suction head at saturation temperature, and the suction head was reduced by adjusting the test tank liquid level.

During cavitation performance testing, liquid temperature and pump pressure was recorded and monitored at pump suction and discharge, along with the liquid level of the test tank. While the liquid level (suction head) began to decrease, a computer controlled high-speed data recording system continued to save test parameters until the pump differential head dropped by 45% (55% of head without presence of cavitation). Since the test liquid is a hydrocarbon with a considerable amount of compressibility, and with relatively less density with respect to water, operating the pump with vapour formation was not a concern during testing. Liquid height from the inducer centreline at 3% head drop was reported as the NPSH3 of the full pump assembly. NPSH3 is also known as the NPSHr. A head drop curve for each pump assembly was plotted to determine the exact liquid height from the inducer centreline that corresponds to a 3% head drop.

Results

CFD simulation results for the constant pitch pump under normal operating conditions, and without the presence of cavitation, were compared to the non-cavitation performance test results to identify if the assumptions and boundary conditions had any impact on the accuracy of the calculations. For this purpose, CFD simulation results in terms of pump differential head and hydraulic efficiency versus volumetric flow rate were overlaid on the pump performance curve of the constant pitch pump assembly.

CFD predictions of the two-stage pump were calculated based on the differential head of each hydraulic component. Two-stage performance is essentially the total differential head produced by the suction stage plus one additional impeller and diffuser vane differential head.

CFD simulations and pump performance of the pump with variable pitch inducer were compared to actual pump performance under non-cavitation conditions. The purpose of this comparison was not only to validate the CFD predictions, but to also compare the actual test results of the pump assembly with constant pitch inducer to the pump assembly with variable pitch inducer under non-cavitation conditions. Results indicate that the head change between the constant pitch inducer and the variable pitch inducer is insignificant with respect to two-stage total head of the impeller and diffuser vane (Figure 2).

Cavitation performance of the pump assembly with constant pitch inducer and variable pitch inducer are reported in terms of NPSH3 of the full pump assembly. During NPSH testing, pumps are tested at various flow points to determine corresponding NPSH3. Computer simulations are run at rated flow (238.5 m^3/hr, 1050 GPM) to determine the NPSH3 at rated flow only. This is mainly because API 610 requires that NPSH3 cavitation performance is met at rated flow, and cavitation behaviour of the pump assembly at other flow points are for reference only.

Head drop curves with the constant pitch inducer and the variable pitch inducer are plotted to identify any cavitation performance improvements according to CFD results. Figure 3 is the CFD head drop curve of each configuration of the pump assembly. NPSH3 improvement from 1.3 m to 0.96 m (4.27 ft to 3.15 ft) is predicted by CFD simulations at rated flow. This corresponds to an NPSH3 improvement of 26% at rated flow (Figure 3).

In addition to the CFD NPSH3 prediction, vapour formation in terms of vapour volume fraction at the inducer and impeller blades was investigated. Figure 4 shows the vapour volume fraction at the inducer and impeller under suction head of 1.12 m (3.67 ft). This point is identified as 'A' in Figure 3, CFD head drop curve. As shown in Figure 4, the pump assembly with constant pitch inducer had a considerable amount of vapour formation at the inducer, and the vapour further propagated to the impeller suction side, causing a breakdown in pump discharge. Under this suction head, the pump differential pressure dropped to 66% of its original value. Under the same suction head, the pump assembly with the variable pitch inducer had some degree of vapour formation at the inducer; however, there is no evidence of vapour at the impeller eye. With the variable pitch inducer, the constant pitch pump differential head is maintained (100%) regardless of vapour formation at the inducer section (cavitation inception).

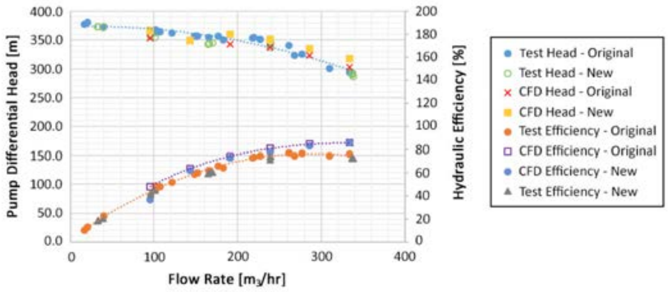


Figure 2. Performance curve of pump with constant pitch inducer as tested vs variable pitch inducer. CFD predictions.

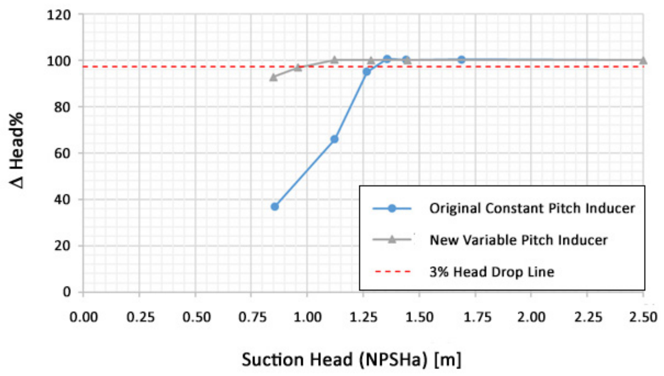


Figure 3. CFD head drop curves of both pump assemblies at rated flow.

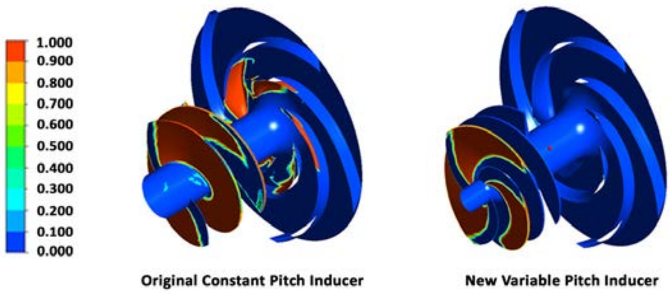


Figure 4. Vapour volume fraction at inducer and impeller blades at rated flow, NPSHa = 1.12 m, Point A.

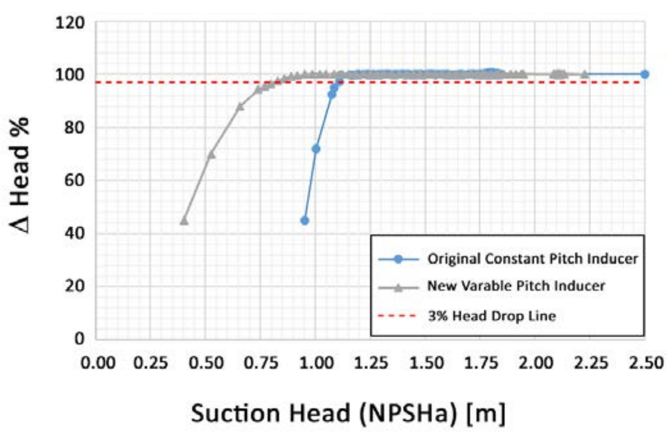


Figure 5. Head drop curves of pump assemblies as tested at rated flow.

Cavitation performance test results are shown in Figure 5. The NPSH3 improvement in terms of cavitation performance between the two inducers is 25%, which is similar to the CFD simulation results. As tested, suction specific speeds are reported to be 46 280 and 36 875, respectively. There is a discrepancy of 15 – 17% between the CFD simulation results and the performance test results, which can be attributed to the complexity and accuracy of both simulations and testing.

Conclusions

According to the results under non-cavitation conditions, the CFD simulations closely predict the actual pump performance for both configurations of the pump assembly. Due to the insignificant relative head increase between each inducer with respect to total head produced by impellers and diffuser vanes, there is minimal change to pump performance under non-cavitation conditions between each pump configuration.

Cavitation performance in terms of NPSH3 is improved by approximately 25% at rated flow by the variable pitch inducer which is designed with consideration of pump operating conditions and impeller eye geometry to attain good NPSH3 performance.

The complete pump assembly (inducer, impeller, and diffuser vane) is included in the simulations since NPSH3 cavitation performance is defined as the NPSHa value when pump's differential head is dropped by 3%. It is observed that cavitation volumes at inducer locations do not necessary result in the pump head drop to NPSH3 level unless the vapour formation propagates to the impeller eye section. Therefore, it is highly recommended to model at least the first stage (suction stage) to determine the true cavitation performance.

Finally, a secondary finding of the analysis is that vapour formation in the impeller eye section negatively influences NPSH3 performance. This suggests that future research and analysis to optimise impeller vane geometry with the variable pitch inducer could further improve NPSH3 performance. **LNG**

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