

DESIGNING AN ULTRA-LOW SPECIFIC SPEED CENTRIFUGAL PUMP

by

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Kazuo Uchida is a Consultant in Shizuoka, Japan. He started with DMW Corporation in 1957 and retired in 1999. He is currently a part-time supervisor of pump manufacturers including the DMW family. Mr. Uchida's professional career began as a design engineer of valves, and he is experienced in almost all hydraulic machines. He especially devoted himself to the estimation method of pump performance in his later career in the company, and

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Yinchun Cao is Manager of the R&D Division of Sanso Electric Co., Ltd., in Himeji, Japan. His work has been mainly in the research and development of small-sized pumps. Dr. Cao has established a new method of designing a small-sized pump aimed at high efficiency and reliability, and this was awarded a prize by the Turbomachinery Society of Japan. He has contributed several papers and has a number of inventions related to turbomachines.

Dr. Cao received his Engineering Master's degree from Jiangshu University (China, 1996), and Doctoral degree from Kyushu University (Japan, 2001). He is a member of JSME and the Turbomachinery Society of Japan.

ABSTRACT

Traditional commercial centrifugal pump designs have specific speeds (N_s) greater than 500 because low values of N_s are usually associated with low efficiency, pulsation, and a drooping head characteristic. In this instance, acceptable hydraulic performance is achieved with a unique impeller design coupled with direct current (DC) motor characteristics.

When the specific speed of a centrifugal pump falls below 500, the ratio of friction loss to entire loss grows extremely large and satisfactory performance can no longer be expected. Consequently, positive displacement pumps dominate this specific speed regime.

In this development, a combination of trial calculations, using a method of performance estimation for an impeller, and the idea of machining the flow passage and ditching a lot of shallow spiral grooves into the impeller face to maintain a small vane tip clearance (the axial gap between the impeller and liner) was adopted. Together with the adoption of a DC-canned-motor, miniaturization of the pump unit was achieved. The test results of two similar impeller designs show peak efficiencies of 28 percent and 25 percent are attained for the specific speeds of 370 and 340,

respectively, even though it is a palm-sized pump. Moreover, the noise level of both impellers is 52 decibels (A).

INTRODUCTION

It is commonly accepted that the lower practical limit of specific speed of a centrifugal pump is 500, and pumps with specific speeds lower than this value suffer reduced efficiency and are no longer practical. Even so, probably for the sake of seeking lower levels of noise or pulsation, studies on “very low specific-speed impellers” have been continued by Kurokawa, et al. (1990, 1997), Matsumoto, et al. (1998a), Choi, et al. (2003), Yinchun, et al. (2003), etc. The points made in these studies—that the reasons for the reduced efficiency of low specific speed centrifugal pumps are that “impeller exit width narrows with the lowering of specific speed, the friction loss increases and disk friction becomes larger comparatively”—suggest some ideas that must be taken into account.

A customer inquired about making a pump that can operate at the very low flow rate of 0.8 gpm (3 l/min) while producing a head of 100 ft (30 m) using a centrifugal impeller to minimize noise. To respond to this, a thoroughly new impeller design was needed and a design method that has not been used for conventional impellers was developed. Fortunately the expectation was achieved.

HYDRAULIC DESIGN OF IMPELLER

Design Point

Figure 1 depicts a widely published relationship between turbopumps efficiency versus specific speeds. It is considered that the range of specific speeds under 500 is impractical and not many commercial pumps are made in this range. In Figure 1 it can be seen that large pumps that discharge a large quantity of water attain high efficiencies in contrast to small pumps that are apt to present lower efficiency. The range of higher efficiency exists at specific speeds of 1650 to 3000. The efficiency of large pumps, with nozzle diameters exceeding one foot, attains upward of 80 percent, and it should be satisfactory if small pumps of domestic use attain 50 percent. Also, small pumps producing high heads at low flow rates will inevitably invite lower efficiency.

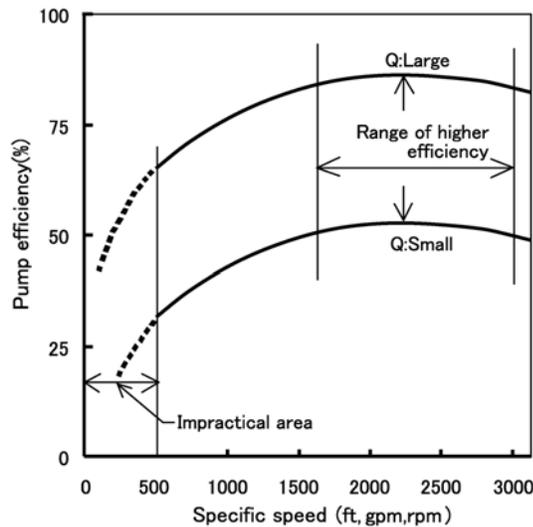


Figure 1. Turbo Pumps Efficiency Versus Specific-Speed.

Considering the practical use, first the authors aimed at 4.2 gpm (16 l/min) and 82 ft (25 m) as the design point, and decided on $3/8$ inch and $1/4$ inch as the suction inlet and discharge outlet nozzle diameters, respectively. Next, they selected 4950 rpm as the rotational speed after taking into account the friction loss. These design parameters result in a specific speed of 372. The authors used this “higher specific speed” because they could not find any advantage

in the selection of a lower specific speed than this through the preliminary calculation concerning the efficiency of a specification point of 100 ft at 0.8 gpm according to Uchida (1994). In the design, the authors tried to get a stable head curve accompanied by gradual descending characteristics.

Impeller Type

The impeller was designed as a “radial” type, in which the flow passage is entirely perpendicular to the rotating axis to maximize the centrifugal force and a semi-open construction (having only a back shroud) to facilitate fabrication and minimize disk friction. Generally, a closed type impeller, which has both a back and front shroud, is used for low specific speed centrifugal pumps. This is because it is difficult to maintain the close axial clearance between the impeller face and liner, even though a semi-open impeller has a distinct advantage in reducing the disk friction and ease of manufacturing.

Impeller Discharge Angle

As seen in Figure 2, which shows the theoretical head of an idealized pump for different discharge angles, a comparatively high head for a wide range is easy to get by adopting a large discharge angle. However, if the head/flow rate curve is unstable (namely, the curve has the characteristic that developed head increases as the flow rate increases), it is apt to invite instability like surging in high static pressure systems or parallel operation. Then, taking into account the amount of the friction loss accompanied by the increment of flow rate, a large β_2 (a steep impeller vane exit angle) was adopted, as long as the curve does not show the head increasing with an increase in flow rate. That is, 13 degrees and 110 degrees as the entrance angle and discharge angle, respectively, implying the given suction performance, and the passages between entrance and discharge are connected smoothly.

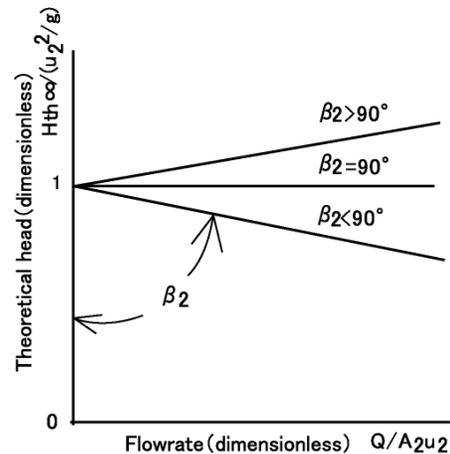


Figure 2. Effects of Impeller Discharge Angle on Theoretical Heads.

The ratios of friction and impingement loss versus Euler’s head, and hydraulic and mechanical efficiency that result from disk friction of the impeller, when 110 degrees is adopted as the discharge angle, are shown in Figure 3. In the same figure, the total head that is derived by removing the hydraulic loss consisting of friction and impingement loss from the theoretical head are also displayed according to Uchida (1994). It is expected that the descending curve of head will be attained, even though adopting 110 degrees as the discharge angle.

Width of Impeller Flow Passage

In designing a centrifugal impeller, usually the width of flow passages is made smaller with decrease in specific speed. However,

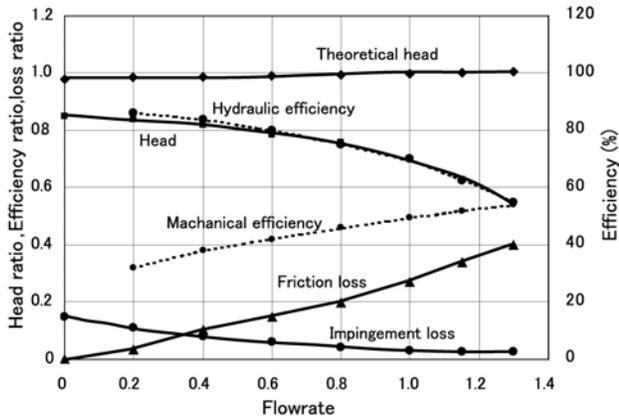


Figure 3. Various Heads, Efficiencies, and Losses Versus Flow Rate.

if the impeller were designed according to this traditional way, it is known from the preliminary design that the width of the impeller would become extraordinarily small, and the friction loss at the flow passage would become large. Therefore, to relieve the increase in friction loss, the method of using ditches rather than blades, rectangular near-radial flow passage on the plane of the impeller was adopted. The flow passages, though those of normal impellers diverge toward the periphery, were kept almost uniform, so the average width of the flow passage and the depth (the same with blade width of an ordinary impeller) is 4.5 mm and 2.0 mm, respectively, as shown in Figure 4, and four passages are channeled into the plane of flat impeller shroud. As a result, the front view of the impeller becomes a disk with four ditches channeled into its surface.

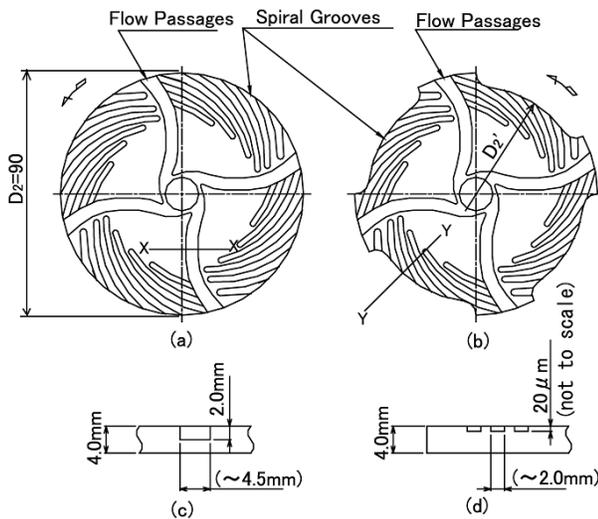


Figure 4. Dimensions of Impellers. (a) is circular disk type, name A-type, (b) is partially cut down type of A-type at periphery, named B-type, (c) is section of flow passage (X-X section), and (d) is spiral-grooves of impeller tip surface (Y-Y section).

Impeller Diameter

Impeller diameter is one of the principal determinants of the power and head of a pump. Regarding this, some trial calculations were performed for the preliminary design, with the idea of applying the lowest power possible. At that time, two kinds of impellers were designed to investigate the effect of disk friction on total power, which is shown in Figure 4. The disk friction increases proportionally to the fifth power of disk diameter, and though the relation with head must also be taken into account, as small a

diameter as possible is desired. As shown in Figure 4 (a) and (b), the diameters of both impellers near the periphery of the flow passages are the same ($D_2 = 90$ mm). The periphery of impeller disk (a) is circular at a constant diameter, D_2 , while impeller (b) is cut down to a diameter of $D_2' = 80$ mm aiming at the reduction of disk friction, and is made a “partial open impeller.” Hereafter, the impeller (a) is called A-type and the impeller (b) is called B-type.

Impeller-Axial Tip Clearance

As mentioned in the explanation of impeller type, the most serious problem for adopting a semi-open impeller is the control of the impeller-axial tip clearance. A large clearance would invite leakage loss, while small clearance would run the risk of galling or seizure by contact. Moreover, it is very difficult to regulate the “end-play” of the rotor if using a canned motor (in fact, a canned motor is used in this pump). The solution was to employ the principle of hydrodynamic bearings, where the axial clearance is limited to the order of film thickness. This will be explained in more detail when the spiral grooves on impeller tip surface are explained in the next section.

The estimated performance curve of A-type is shown in Figure 5, based on the above-mentioned points. Although the estimated performance curve of B-type is not shown, the authors expected an increased rotational speed and consequently increased head, owing to the decrease in disk friction.

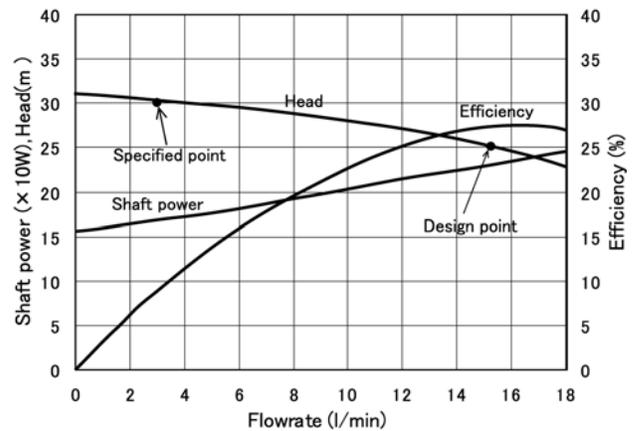


Figure 5. Estimated Performance Curves at 4950 RPM.

The impeller is made of 304-stainless steel with hard facing applied to the channeled face.

Spiral Grooves on Impeller-Tip Surface

After setting the impeller in the pump casing, adequate axial clearance between the impeller-tip and impeller-liner must be maintained. So that the existence of end-play of the rotor can be ignored, the back surface of the impeller is kept free from any kind of regulator and static pressure acts on it pushing the impeller toward the liner. Here, in spite of forcing the impeller against the liner—this is actually necessary to raise the head—galling or seizure between the impeller-tip and -liner must be considered. One countermeasure used is a lot of curved grooves (called “spiral grooves”) etched into the plane surface of the impeller-tip as shown in Figure 4 (d). The spiral grooves are very shallow (approximately 20 μ m) and sharp-edged. It is said that the introduced fluid from the impeller periphery creates hydrodynamic peak pressure at the edges and maintains an axial clearance on the order of several 10 μ m between the impeller tip surface and liner (Yamane, et al., 2002). The second countermeasure is the treatment of hard facing of the liner and the impeller to protect them from galling or seizure at low speeds during startup and shutdown. The impeller tip surface rotates close to the impeller liner, with the minimum clearance, without galling or seizure.

PUMP CONFIGURATION

Volute Casing

Impellers of small pumps are frequently designed to rotate in the center of concentric circular casings; however, a single-volute casing was adopted in this design. As illustrated in Figure 6, 1 mm of gap is set at the tongue (identified as zero degrees) between the periphery of the impeller and the casing wall, and the gap is gradually enlarged to 5.5 mm at the 350 degree revolved position. The width (= depth) of this volute casing is machined constant all around the circumference with numerical control, and the thickness of the tongue is 1 mm. Water is extracted through the diffuser that is equivalent to 7.2 degrees in sectional change from tongue to discharge port.

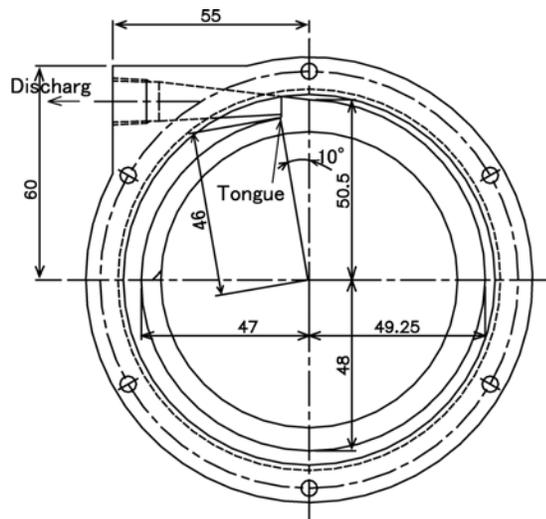


Figure 6. Dimensions of Volute Casing.

The aforementioned impeller-liner, the surface of which is a ground plane, is set on the inside of the casing cover.

Canned DC Motor

The overall configuration of the pump is shown in Figure 7 together with the adoption of a DC-motor. Although the diametrical dimensions are almost the same, longitudinal dimensions are decreased to nearly $\frac{2}{3}$ those of a conventional vane pump with an AC-motor. Another feature worth mentioning is that the concept of a canned motor was adopted for the simplicity of a nonseal pump, despite the additional friction loss of the motor rotor.

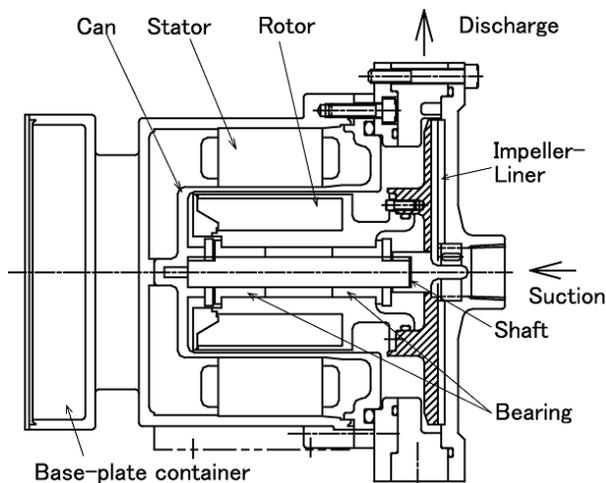


Figure 7. Overall Pump Configuration Using A-Type Impeller.

The motor-can is made of engineering plastic (polyphenylene sulphide, PPS) to avoid the generation of electric current by magnetic flux, which is inevitable when using conductive materials.

The rotor of this motor, which is designed as brushless, uses a ferrite-magnet integrally molded with plastic. The motor power output is 200 W—at normal temperature. At the nondrive-end of the motor, a hole-diode is placed to control the rotation, and a baseplate containing the driving circuit is built into the far end of the unit to avoid the heat of stator-coil and/or pumped liquid.

The motor is cooled with pumpage, which enters the volume behind the impeller and progresses to the annular clearance between the can and rotor absorbing the heat generated by the stator-coil, then flows into clearance of two bearings having very small dimensions, and flows out of them after lubricating the sliding parts of the rotor-shaft and bearings. Finally, as shown in Figure 8, the pumpage is sucked by the impeller eye through the center hall of the impeller, as so-called “leak-flow.” The flow path is tortuous and it is reasonable to estimate that the flow rate of this leak-flow would be very small compared to that leaked through the wearing-rings of a conventional closed type impeller.

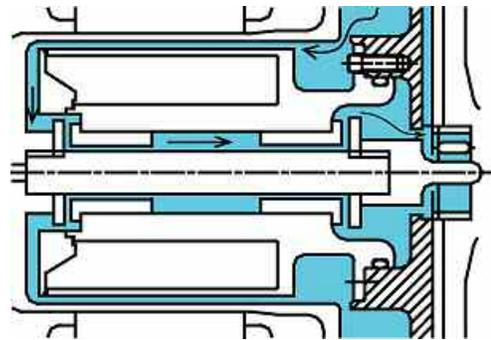


Figure 8. Passage of “Leak-Flow” (Clearances Are Exaggerated).

A rotor-shaft, made of alumina-ceramic, is fixedly inserted into the can-end on one end and a shaft-holder extended from spider set into the suction-flange on the other end. The two bearings are molded with carbon-impregnated plastic.

The driving force of the motor is transmitted through two pin-and-hole couplings that use 180 degree oppositely located pins and holes on the end of the rotor (halls) and impeller boss (pins).

By this coupling, only the torque is transmitted without being influenced by the end-play of the rotor.

PERFORMANCE TESTING

Test Method

Performance testing was carried out based on JIS B 8301 (2000). In spite of the fact that the rotational speed of a DC-motor varies considerably with the load, the tests for both A- and B-type were performed without regard to rotational speed.

The DC-motor performance (power output and rotational speed versus input voltage and current) was measured during separate tests. During the pump performance tests the motor power input parameters, voltage and current, and rotational speed were measured then subsequently converted to power used to calculate hydraulic and unit efficiency.

In the field of small pumps, cavitation performance is apt to be ignored; however, it is rather important to these pumps, because they will be utilized over a wide range. For the net positive suction head required (NPSHr) tests, the authors set the test pump above an open sump. To vary the net positive suction head available (NPSHa), the water level in the sump was changed. During the NPSHr tests they maintained constant flow rate and reduced the NPSHa until the developed head was reduced 3 percent.

At the test, because this pump is greatly affected by the fluid friction on the impeller disk and motor rotor, the authors maintained a constant viscosity of tap water by controlling the water temperature at $68 \pm 4^\circ\text{F}$ ($20 \pm 2^\circ\text{C}$).

In the measurement of noise level, which was carried out according to JIS B 8310 (1985), a microphone was set in a horizontal position perpendicular to the pump axis at a distance of 1 m (3.3 ft), and pressure fluctuation was also measured simultaneously using a strain gauge type of pressure sensor set on the piping close to the discharge port. Both data were recorded and analyzed offline.

Results of Performance Test

Results of the test using A- and B-type are shown in Figure 9. In the figure, besides the head, power, and pump efficiency, rotational speed is plotted against flow rate. It is found that the rotational speeds are considerably decreased by the increase of loads. Now, the estimated performance in Figure 5 is made for a constant speed of 4950 rpm, but the rotational speed of a DC-motor varies according to load as mentioned above. So, even though the head at the specification point cleared the estimated head, it falls below estimates at the range of over flow rate including design point where the power exceeds the target. However, as shown in Figure 10, when the rotational speed is normalized to 4950 rpm, all of the design criteria of A-type are satisfied, though the efficiency of B-type is inferior to A-type in spite of generating higher head approximately by 5 m.

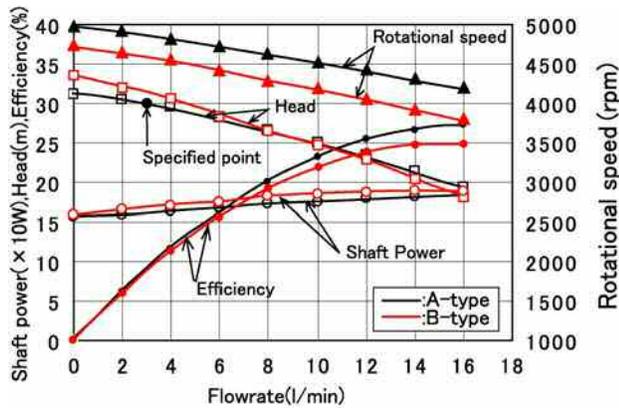


Figure 9. Results of Performance Tests.

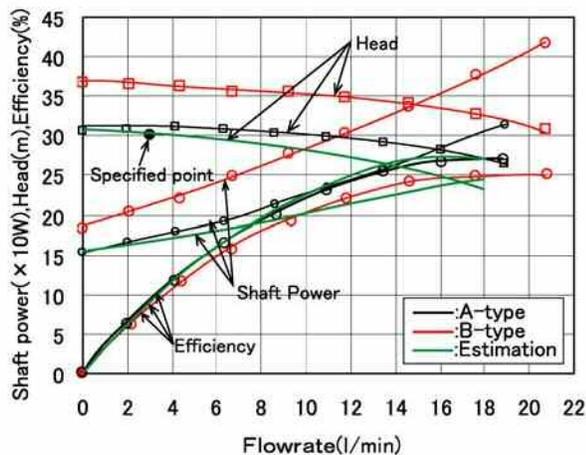


Figure 10. Results of Performance Tests after Normalized to 4950 RPM of Estimated Performance.

Best efficiency is 28 percent for A-type and 25 percent for B-type impellers. The specific speeds at the best efficiency points (BEP) are calculated as 371 and 340 for A- and B-type, respec-

tively. Here, B-type shows a smaller value of specific speed than A-type, in spite of a lower rotational speed. The reason for this difference seems to be as follows. As illustrated in Figure 4, the impeller disk of B-type is cut back to a diameter of 80 from 90 mm, and has four projected portions of 5 mm. These projections function as “partial open impeller vanes” and the head and power would rise by the hybrid effect in spite of inviting increase of resistance resulting in decrease of rotational speed.

As for cavitation performance, result of 3 percent head-drop using A-type that is superior to B-type in pump efficiency is shown in Figure 11. Ordinary, centrifugal pumps require less NPSH at flow rates below the BEP, conversely, this pump requires more NPSH as the flow rate is reduced.

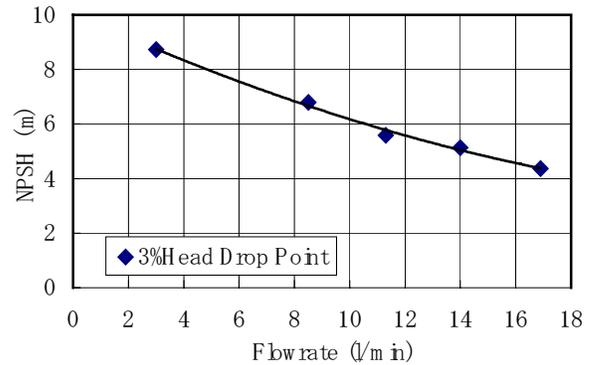


Figure 11. Cavitation Performance of A-Type.

Noise Level and Pulsation

Noise levels and pressure fluctuations of A- and B-type at the specified points keeping the output constant are presented in Figure 12 (a) and (b), and (c) and (d). Initially, B-type was considered disadvantageous to A-type in the point of noise due to its uneven periphery. However, contrary to expectations, there was not a significant difference. Both are approximately 52 dB (A), and very silent operation is attained. But the pressure fluctuation indicates a difference, though the levels are low, and B-type is four-times higher than A-type. The spectrum of the pressure fluctuation corresponds with the so-called NZ component of an impeller, and this seems to be exaggerated by the projections of a partial open impeller.

FIELD APPLICATION

This pump now finds its application in the fields of pumping cooling-water for laser-head and pure water for fuel cell, concentrating enzyme, etc.

CONCLUSION

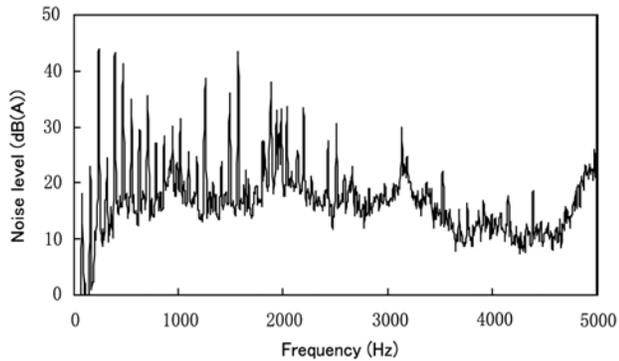
For an ultra-low specific speed centrifugal pump like this, a major issue is how to raise the mechanical efficiency apart from the hydraulic efficiency. Among the various methods for decreasing friction loss, some strategies are proposed for disk friction. One of which is to make the disk surface coarse (Matsumoto, et al., 1998b), and the other is to ditch streak lines on the disk surface (Watanabe, et al., 2002). Unfortunately, the authors have not yet had a chance to try these. They wish to adopt them in the near future and have decided to develop a 150 ft (46 m) class pump.

NOMENCLATURE

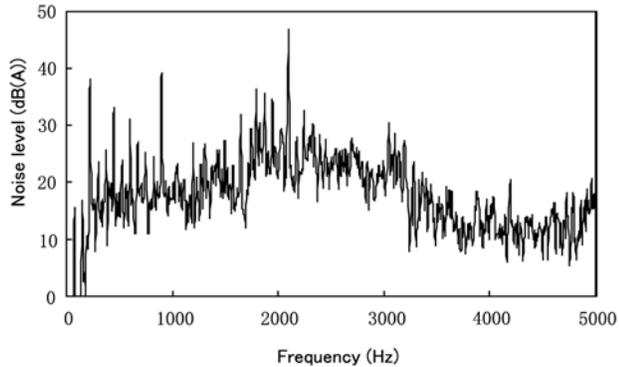
$$N_s = \frac{\text{rpm} * \sqrt{\text{gpm}}}{\text{ft}} = \text{Equation for } N_s, \text{ specific speed in Imperial units}$$

$$D_2 = \text{Major impeller diameter}$$

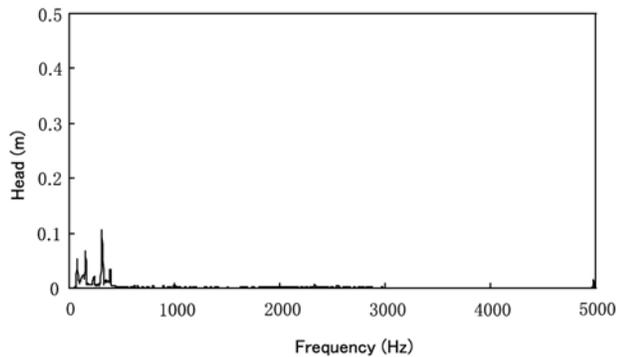
$$\omega = \frac{\text{rpm}}{2\pi} = \text{Rotational speed}$$



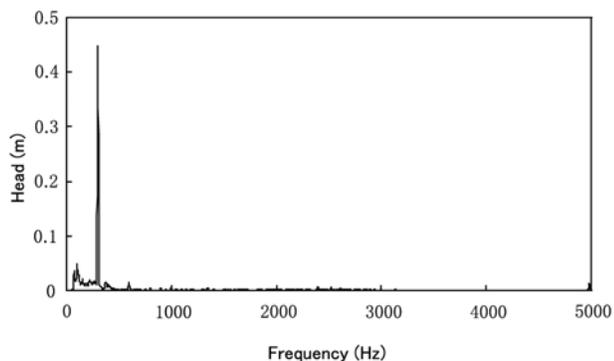
(a) A-type impeller's noise level.



(b) B-type impeller's noise level.



(c) A-type impeller's pressure fluctuation.



(d) B-type impeller's pressure fluctuation.

$$U_2 = \frac{D_2}{2} * \omega \quad = \text{Peripheral velocity}$$

$$H_e = \frac{U_2^2}{g} \quad = \text{Euler's head}$$

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Figure 12. Results of Noise and Pulsation Tests.