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Effects of Blade Discharge Angle, Blade Number and Splitter Blade Length on Deep Well Pump Performance

E. Korkmaz[1†], M. Gölcü[2] and C. Kurbanoğlu[3]

[1] *Suleyman Demirel University, Energy Systems Engineering Department, 32260 Cunur, Isparta, Turkey.*
[2] *Pamukkale University, Automotive Engineering Department, 20017 Kinikli, Denizli, Turkey.*
[3] *Istanbul Medeniyet University, Mechanical Engineering Department, 34730, Goztepe, Istanbul, Turkey.*

†Corresponding Author Email: ergunkorkmaz@sdu.edu.tr

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ABSTRACT

Impellers with splitter blades are used for pumps and compressors in the design of turbomachines. Design parameters such as the number of blades, blade discharge angle and impeller discharge diameter impact affect pump performance and energy consumption. In this study, the effect of the number of blades (z=5, 6, and 7), blade discharge angles (β_{2b}=25°, and β_{2b}=35°) and splitter blade lengths (40, 55, 70, and 85% of the main blade length) on Deep Well Pump (DWP) performance has been studied experimentally. In the experiments, pump casing, blade inlet angle, blade thickness, blade width and impeller inlet and discharge diameters have been kept fixed while other parameters such as the number of blades, blade discharge angles and splitter blade lengths have been allowed to vary. As a result of the experimental study, the highest efficiency of all the impellers for best efficiency point (b.e.p) has been obtained on the impeller with the number of blades z=6, blade discharge angle β_{2b}=25° and 85% splitter blade addition compared to impellers without splitter blades.

Keywords: Deep well pump; Blade discharge angle; Pump performance; Impeller; Splitter blade.

NOMENCLATURE

b.e.p	best efficiency point	P_e	brake horsepower
b_1	blade inlet width	Q	flow rate
b_2	blade outlet width	Wosb	without splitter blade
D_1	impeller inlet diameter	z	number of blades
D_2	impeller outlet diameter		
e	blade thickness	β_{1b}	blade inlet angle
H_m	head	β_{2b}	blade discharge angle
L	length of main blade	η_g	general efficiency
L_s	length of splitter blade		
$\overline{L}$	non-dimensional splitter blade length		

1. INTRODUCTION

Most of the electricity consumed in industrialized countries is used in electric motor applications, which are typically pump, fan and compressor drives (De Almeida *et al.* 2003; Ahonen *et al.* 2010). Pumps of the centrifugal type are broadly used in the transfer of liquids due to the simplicity of their structure, low cost and low space demands. Deep well pumps (DWPs) are one of the most widespread types of such pumps and are currently used in a number of fields such as underground water distribution, industrial water networks, thermal water networks, agricultural irrigation systems and water supply grids. Especially in countries whose economies largely depend on agriculture, underground waters have a broad field of application in agricultural irrigation. Considering technologies used in transporting underground waters to the surface, it can be seen that DWPs with low specific speed are used. One of the largest disadvantages of such pumps is low efficiency. In the studies that have been conducted for energy saving, it has been seen that one of the areas of high potential energy saving is pumping systems (Kaya *et al.* 2008).

A variety of principles have been developed to solve the hydraulic problems of pumps with low specific speed. One of those is the splitter blade principle. In order to increase efficiency, the diameter of the impeller of a pump with low specific speed needs to be decreased. Moreover, in addition to decreasing the diameter, a larger blade discharge angle (β_{2b}) and a greater number of blades need to be used. This will ensure that the desired head is achieved. Thus, losses occurring as a result of the increase of the blade discharge angle can be decreased by placing splitter blades between two main blades. The reason for using splitter blades instead of main blades is to decrease the clogging at the impeller inlet caused by a large number of blades. The use of splitter blades plays an important role in obtaining optimum design and identifying the best area among the design points (Korkmaz, 2008).

A variety of studies have been previously conducted on the use of splitter blades to solve the hydraulic problems of pumps with low specific speeds. In the study where they considered the calculation of a three-dimensional flow in a centrifugal impeller with splitter blades, Kui and Jian (1988) have used a method they named "flow-surface coordinates iteration" to calculate the three-dimensional flow in turbo machines. Empirical data obtained from centrifugal pump impellers with different splitter blades have been provided to provide the validity of the calculation method offered. It was indicated in the said study that the aerodynamic load on the radial part of the blades could be higher in order to increase impeller efficiency where the number of blades is deceased and that splitter blades could be used in the channel in order to solve this problem.

Gui *et al.* (1989) closely examined the effects of the splitter blades on the performance of a forward-curved centrifugal fan. The study consists of two parts that involve experimental work and numerical calculations. The experiments have been conducted on a special impeller with adjustable splitter blades to observe performance under difference circumstances. The results of the experiments conducted have shown that the change in the circumferential position of the splitter blades considerably affects fan performance and that splitter blades with a proper length could increase the total pressure coefficient. As a general result of the study, it has been explicitly determined that the circumferential position of splitter blades affects the performance of forward leaning fans. It has been emphasized that when the splitter blades are positioned close to the pressure surface, the total pressure coefficient increased whereas when they were positioned close to the suction surface of the main blade, the efficiency increased somewhat. It has been claimed that increasing the length of the splitter blade might increase the total pressure coefficient but that there was no certain rule relating to the effect of the splitter blade length on the efficiency curve. It has been further emphasized that the discharge angle of the splitter blade did not have a great effect on the efficiency and a suggestion was made to study the blade discharge angle in a more detailed manner in order to increase the fan efficiency.

Kaya (2003) experimentally investigated regaining the tangential velocity energy of axial flow pump by using two different axial flow pump impellers with and without splitter blades. As a result of the investigation, it is found that using splitter blades and regaining the tangential velocity energy of the pump improves the total pump efficiency about 3%.

Miyamoto *et al*. (1992) have studied the effect of splitter blades on the flow and performance by measuring speed and pressure in closed and half-open impeller channels. Flows and characteristics in the channel of impellers with splitter blades have been compared with the flows and characteristics in the channel of impellers without splitter blades. It has been claimed that in impellers with splitter blades, the blade loads tended to decrease and emphasized that absolute peripheral speeds and total pressures of such impellers was significantly higher than that of impellers without splitter blades.

When splitter blades are used in impellers, it is considerably important for the pump efficiency to determine the optimum length and position of the splitter blade. Splitter blades are generally positioned at the geometric center between two main blades and have a suitable size. Yuan (1997) wrote that the radial length of a splitter blade was generally 2/3 of the length of the main blade or 0.5~0.75 times the impeller outlet diameter while the inlet radius of the splitter blade was approximately 0.4~0.6 times the impeller outlet diameter.

Zha and Yang (1986) used super-short blades that had a very small radial length or whose inlet radius was 0.75~0.85 times the impeller outlet diameter in the improvement of pump performance. The optimum length and position of splitter blades in the improvement of pump efficiency should be supported experimentally.

Blade discharge angle has very important role in the performance of the centrifugal pump (Li, 2012). It is well-known that the Euler's head is inversely proportional to flow rate Q with discharge angle of blade $\beta_2 < 90°$ and the smaller the β_2, the more rapid the decrease of Euler's head with flow rate. Former experience indicates that with appropriate other geometrical parameters the head-flow rate curve is basically stable when $\beta_2 < 30°$. In some cases β_2 can be as high as 40° without droop (Yuan, 1997).

In this study, the effect of the number of blades (z=5, 6 and 7), blade discharge angles (β_{2b}=25° and β_{2b}=35°) and splitter blade lengths (40, 55, 70, and 85% of the main blade length) on DWP performance has been studied experimentally. In the experiments, pump casing, blade inlet angle, blade thickness, blade width and impeller inlet and discharge diameters have been kept fixed while other parameters such as the number of blades, blade discharge angles and splitter blade lengths have been allowed to vary.

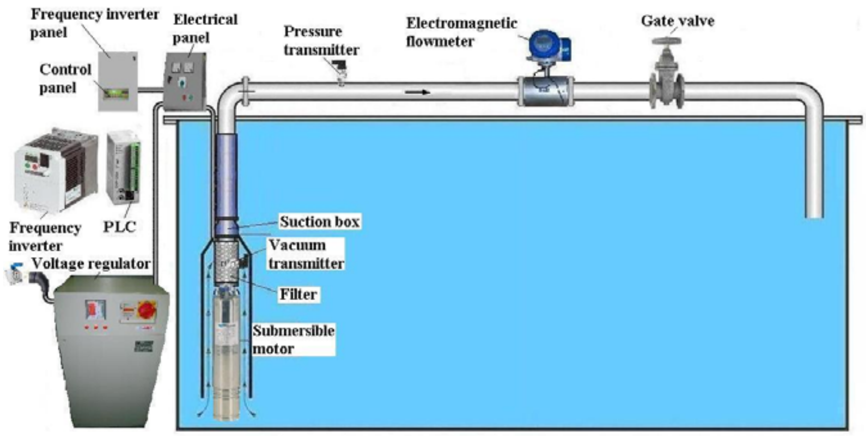


Fig. 1. Deep Well Pump test rig (not scale).

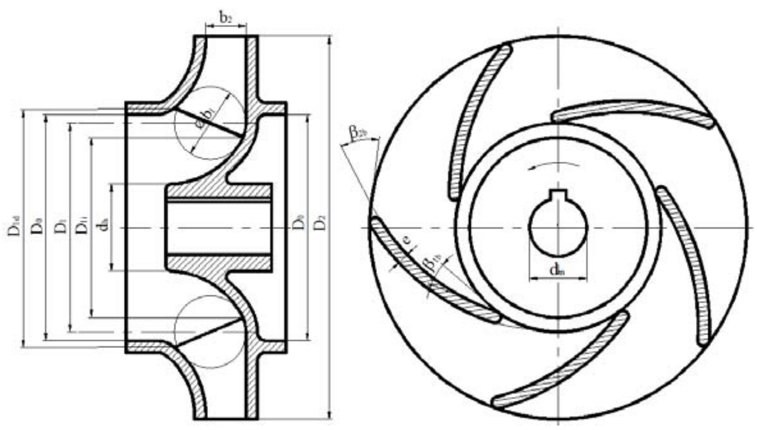


Fig. 2. An example of the pump impellers designed (β_{2b}=25°, z=5, impeller without splitter blades).

Table 1 Deep Well Pump design point values

Q	36 m^3/h	β_{1b}	18°	**e**	4 mm	D_1	72 mm
H_m	13 m	β_{2b}	25° and 35°	d_m	20 mm	D_{1i}	62 mm
n	2850 rpm	b_1	25 mm	d_h	30 mm	D_{1d}	82 mm
z	6	b_2	14 mm	D_0	78 mm	D_2	132 mm

2. TEST RIG

A DWP with low specific speed has been designed in this study. Geometric factors other than the number of blades (z=5, 6 and 7), blade discharge angle (β_{2b}=25° and β_{2b}=35°) and length of splitter blades have been kept constant and splitter blades were added to the geometric center of two main blades at the size of 40, 55, 70 and 85% of the main blade length. The purpose was to experimentally study the effect of splitter blade length on the pump performance for different numbers of blades and blade discharge angles and to obtain pump characteristics. A DWP test unit was assembled as shown in Fig. 1 for this purpose.

The DWP design values were used as H_m=13 m, Q=36 m^3/h, n=2850 rpm. Impellers for DWPs are generally designed using empirical equations in the literature (Stepanoff, 1957; Dicmas, 1987; Lobanoff and Ross, 1992; Tuzson, 2000; Karassik *et al.* 2001). Design point values of the DWP have been provided in Table 1 and an example of the designed impellers is shown in Fig. 2.

3. DETERMINATION OF PUMP CHARACTERISTICS

Unlike most volumetric pumps, rotodynamic pumps are capable of pumping fluids at flow rates that vary depending on project values and suction conditions. Here, the most important factor affecting the flow rate is the manometric head of the pump. Depending on the manometric head, a change in the flow rate will lead to changes in the brake horsepower and efficiency values. These curves that indicate the change of manometric head, brake horsepower and efficiency depending on a variety of flow rate values is called characteristic curves. Drawing the characteristic curves of a pump is particularly important in the determination of the pump's b.e.p. EN ISO 9906 (1999) has been referred to in determining the pump characteristics.

Manometers are generally used to measure pressure in classical pump test systems. However, pressure transducers or pressure transmitters have recently started to be broadly used due to measurement precision demands. High-precision pressure transmitters with 24 V DC supply, 2-cable connection and 4~20 mA output have been used in pressure measurements. The vacuum pressure transmitter has a measurement range of -1~0 bar while the positive pressure transmitter has a measurement range of 0~10 bar (Wika, 2015). A magnetic-type flow-meter has been used for flow rate measurements (Krohne, 2005) and a network analyzer has been used for brake horsepower measurements (Entes, 2006). The network voltage has an important effect on the motor rotation speed (and therefore on the characteristic values of the pump). To avoid a possible negative effect of this situation during the comparison of impeller characteristics, a three-phase servo voltage regulator has been used in the test settings with a precision of ±2% of network voltage. A submersible motor with a speed of 2850 rpm has been used in the experiments. A frequency inverter has been used to obtain the motor rotation speed of 2850 rpm and a PLC unit has been used for its control. Accuracy of the used devices and uncertainty of the calculated results have been provided in Table 2.

Table 2 Accuracies of the measurements and the uncertainties in the calculated results

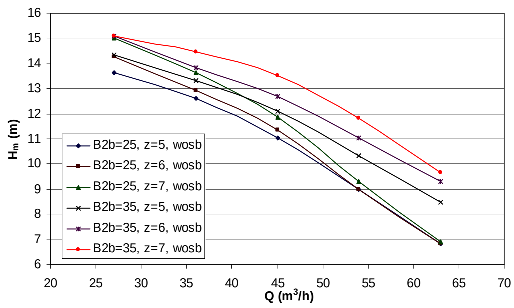
Measurements	Accuracy
Pressure transmitter	± 0.025
Magnetic flow-meter	± 0.015
Network analyzer	± 0.01
Three-phase servo voltage regulator	±%2
Calculated results	*Uncertainty*
Efficiency	±%1.2

One of the most significant problems of low specific speed pumps is their low efficiency. In order to increase efficiency, splitter blades can be positioned between two main blades with suitably selected number of blades and blade discharge angle. For this purpose, splitter blades were positioned on an impeller with no splitter blades at the geometric center of two main blades with various splitter blade lengths (at 40, 55, 70 and 85% of the main blade length). This experimentation has made it possible to study the effects of splitter blade length and changes in the blade discharge angle on DWP performance.

A total of 30 impellers have been used in the experiments, 6 of which were without splitter blades. To the number of blades z=5, 6 and 7 and an impeller with 24 splitter blades with a blade discharge angle of β_{2b}=25° and β_{2b}=35°, splitter blades were added to the geometric center of two blades that had a size of 40, 55, 70 and 85% of the main blade length. Due to instability at very low flow rates, the pump characteristics have been considered in the range of 27 – 63 m^3/h.

3.1 Characteristics of Impellers without Splitter Blades

In order to determine the effects of the number of blades and blade discharge angle on pump performance, 6 impellers were tested with numbers of blades z=5, 6 and 7 and blade discharge angles β_{2b}=25° and β_{2b}=35°. Figure 3a, Fig. 3b and Fig. 3c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics, respectively.



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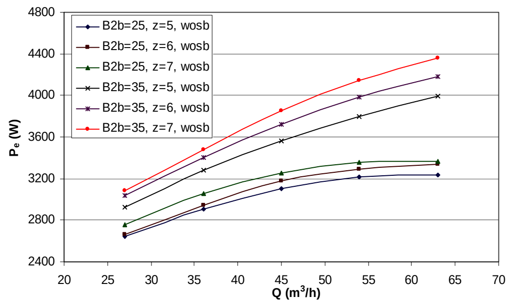

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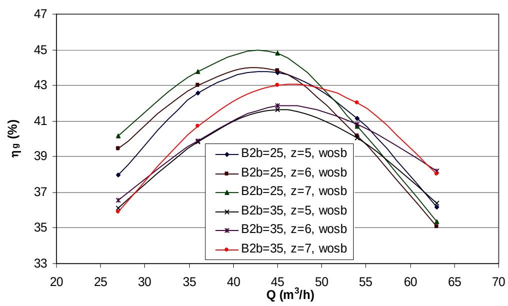


(c)

Fig. 3. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers without splitter blades for different blade discharge angles and numbers of blades.

As seen on Fig. 3a, concerning b.e.p, the head increases with an increase in the number of blades and blade discharge angle. The results obtained from the experiments are in conformity with the results provided in the literature (Yuan, 1997; Yuan *et al.* 1993; Gölcü, 2001; Gölcü, 2005; Gölcü, 2006a,b; Gölcü and Pancar, 2006; Gölcü *et al.* 2007; Li, 2012). Although an increase in the blade discharge angle leads to an increase in the head, it also increases the power consumed by the pump (Fig. 3b). Moreover, considering the b.e.p, a decrease is observed in the general efficiency as the blade discharge angle is increased (Fig. 3c).

Among the impellers without splitter blades, the highest efficiency was obtained using the impeller with the blade discharge angle β_{2b}=25° and number of blades z=7. For the same blade discharge angle, as the number of blades increased, the head also grew. Additionally, increase was also observed in the power consumed by the pump and efficiency. Increases of 7.51% in head, 4.77% in power and 2.56% in efficiency were obtained with an impeller with β_{2b}=25° and z=7 compared to an impeller with β_{2b}=25° and z=5.

If the number of blades is kept constant, as the blade discharge angle is increased, a decrease is observed in efficiency as head and power consumed by the pump increase. An increase of the blade discharge angle has increased head by 8.41% and power consumed by the pump by 14.84% in an impeller with β_{2b}=35° and z=5 compared to an impeller with β_{2b}=25° and z=5. However, a 5.60% decrease has been observed in efficiency with the increase of the blade discharge angle.

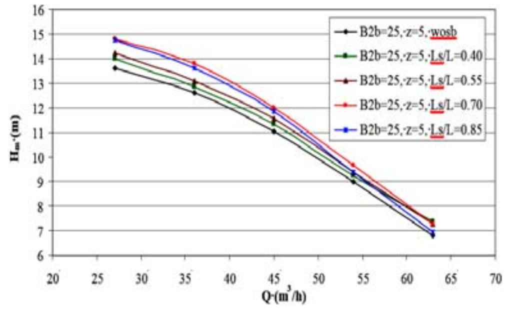
3.2 Characteristics of Impellers with Splitter Blades

In order to observe the effects of splitter blade length on the pump performance, a total of 24 impellers have been tested with the numbers of blades z=5, 6 and 7 and blade discharge angles β_{2b}=25° and β_{2b}=35°. Effects of the splitter blade length has been observed on DWP performance by adding to the geometric center of main blades splitter blades with a size of 40, 55, 70 and 85% of the main blade length on impellers with different blade numbers and blade discharge angles. On the graphs, L is used to indicate main blade length whereas the splitter blade length is indicated by L_s. For instance, L_s/L=0.70 indicates that splitter blades have been added that are 70% of the main blade length.

3.2.1 Characteristics of impellers with splitter blades and a number of blades z=5

Figures 4a, 4b and 4c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with a number of blades z=5 and blade discharge angle β_{2b}=25°, respectively. As seen from Figs. 4a and 4b, addition of splitter blades has led to an increase in head and power consumed by the pump. The greatest increase in the head and power has been obtained on the impeller with 70% splitter blade addition while the greatest efficiency increase has been obtained on the impeller with 85% splitter blade addition (Fig. 4c).

Figures 5a, 5b and 5c show H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with number of blades z=5 and blade discharge angle β_{2b}=35°, respectively. As seen on Figs. 5a and 5b, splitter blade addition has increased the head and power consumed by the pump. The highest head and efficiency increases have been obtained on the impeller with 85% splitter blade addition (Figs. 5a and 5c).



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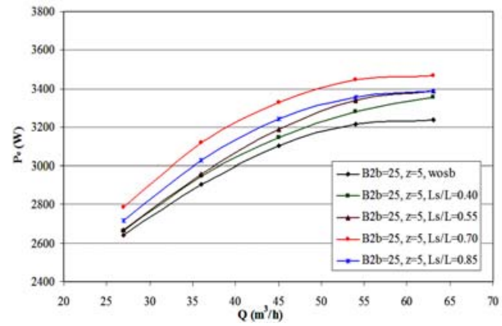

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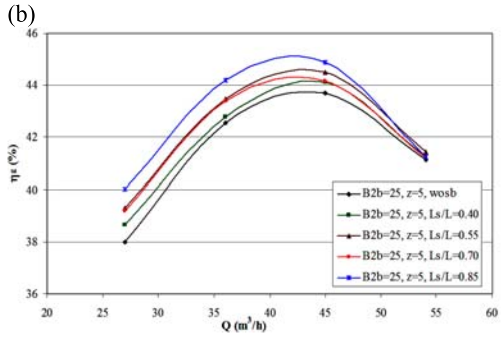
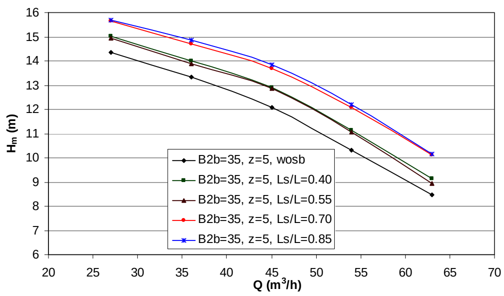


(c)

Fig. 4. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=5 and β_{2b}=25°.

Literature sources indicate that the optimum efficiency can be obtained with the number of blades range z=5-8 (Schweiger and Gregori, 1987) further stating that the critical number of blades for splitter blade applications would be z=5 (Gölcü *et al.* 2007). In the study conducted, the researchers emphasized that for the blade discharge angle β_{2b}=15°, a blade number of 5 and over would not be suitable for the use of splitter blades and that for blade numbers z=5 and higher, the blade discharge angle would need to be increased.

In a study by Gölcü *et al.* (2007), it has been detected that splitter blade addition for β_{2b}=15° and z=5, as seen on Figs. 6a, 6b and 6c, would lead to an increase in power and would decrease both head and pump efficiency at points following the best efficiency point. In our study, however, head and efficiency increases have been obtained at all operation points with the addition of splitter blades to an impeller with the number of blades z=5 after the blade discharge angle was increased (β_{2b}=25° and β_{2b}=35°) (Figs. 4a, 4c, 5a, 5c).



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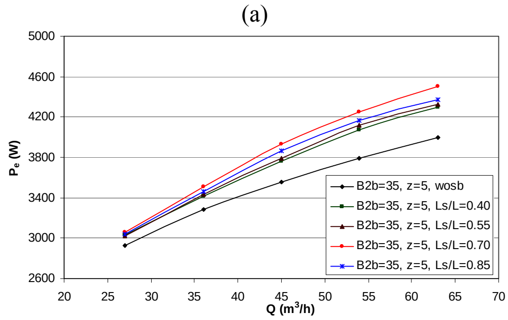

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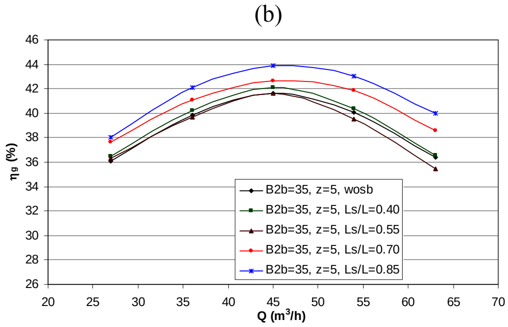


(c)

Fig. 5. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=5 and β_{2b}=35°.

3.2.2 Characteristics of Impellers with Splitter Blades and a Number of Blades z=6

Figures 7a, 7b and 7c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with a number of blades z=6 and blade discharge angle β_{2b}=25°, respectively. As seen from Fig. 7a, addition of splitter blades has led to an increase in head of the pump. The greatest head increase has been obtained on the impeller with 85% splitter blade addition. Among the impellers with splitter blades, the lowest power has been obtained on the impeller with 85% splitter blade addition. This is indicated on Fig. 7b. Among the impellers with splitter blades, the highest efficiency has been obtained on the impeller with 85% splitter blade addition (Fig. 7c).

Compared to the impeller without splitter blades, the impeller with 70% splitter blade addition had a 4.57% increase in head, 1.96% increase in brake horsepower and 2.53% in efficiency. Compared to the impeller without splitter blades, the impeller with 85% splitter blade addition had a 5.45% increase in head, 0.36% increase in brake horsepower and 5.06% in efficiency. Apparently, while 5.45% increase in head and 5.06% increase in efficiency have been observed on the impeller with 85% splitter blade addition, the increase in brake horsepower is at a negligible level.

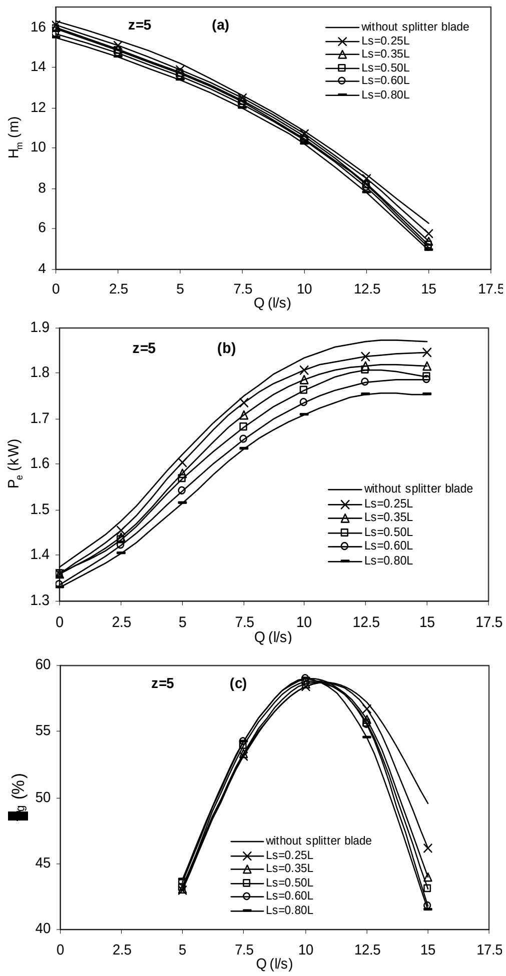
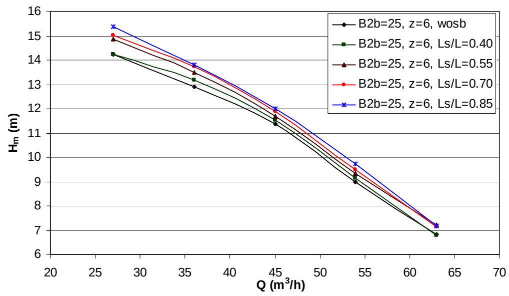


Fig. 6. (a) H_m-Q, (b) P_e-Q and (c) η_g -Q characteristics of the impeller (z=5, β_{2b}=15°) with different lengths of splitter blades (Gölcü *et al*. 2007).

Figures 8a, 8b and 8c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with a number of blades z=6 and blade discharge angle β_{2b}=35°, respectively. As seen from Figs. 8a and 8b, addition of splitter blades has led to an increase in head and power consumed by the pump. The greatest head and efficiency increase has been obtained on the impeller with 70% splitter blade addition (Figs. 8a and 8c). Compared to impellers without splitter blades, the impeller with 70% splitter blade addition yielded 10.62% head increase, 5.25% brake horsepower increase and 5.06% efficiency increase. Compared to impellers without splitter blades, the impeller with 85% splitter blade addition yielded 8.58% head increase, 7.65% brake horsepower increase and 0.83% efficiency increase.



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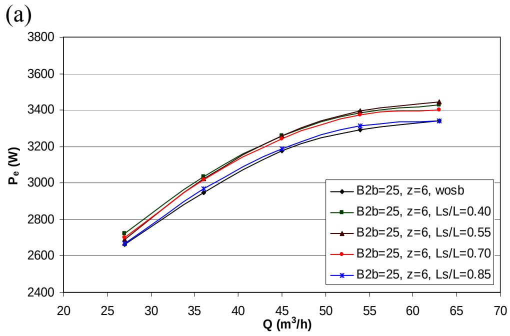

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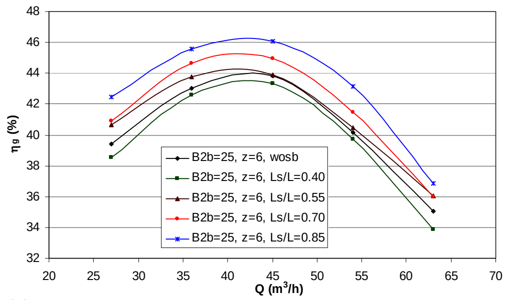

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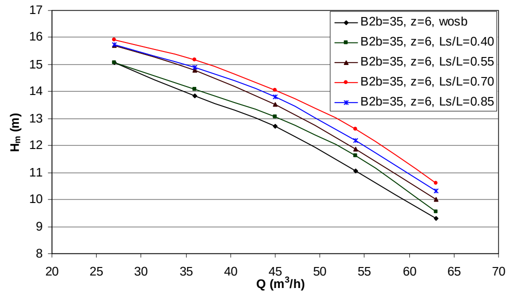
Fig. 7. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=6 and β_{2b}=25°.



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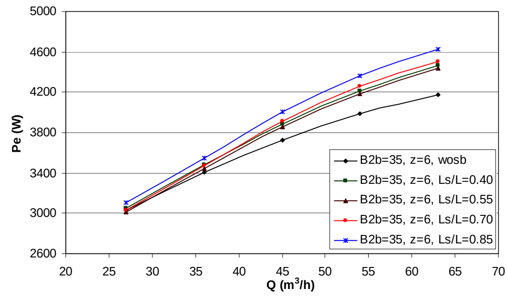

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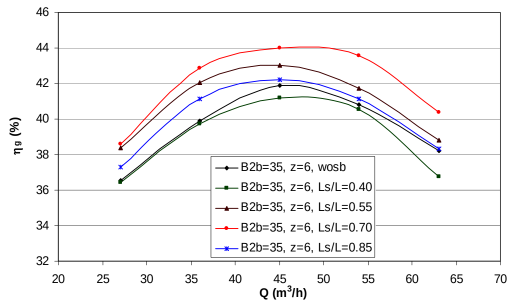


(c)

Fig. 8. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=6 and β_{2b}=35°.

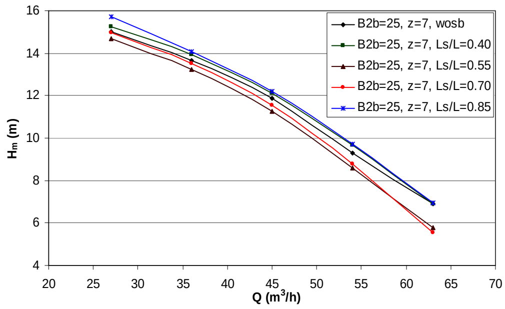
3.2.3 Characteristics of Impellers with Splitter Blades and a Number of Blades z=7

Figures 9a, 9b and 9c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with a number of blades z=7 and blade discharge angle β_{2b}=25°, respectively. As seen from Fig. 9a, the greatest head increase has been obtained on the impeller with 85% splitter blade increase.

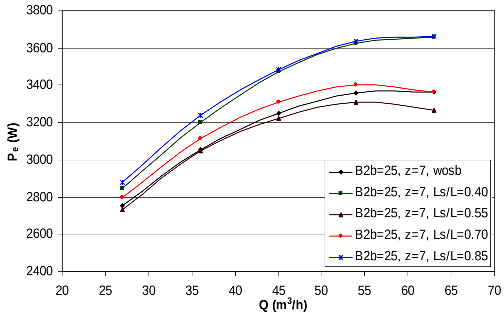
Among the impellers with splitter blades, the lowest power has been obtained on the impeller with 55% splitter blade addition (Fig. 9b). However, the head value of the impeller with 55% splitter blade addition is smaller than that of the impeller without splitter blades (Fig. 9a).

Considering efficiency values of impellers with the number of blades z=7 and blade discharge angle β_{2b}=25°, an interesting situation has been observed. While concerning b.e.p, the efficiency values of all impellers with splitter blade additions are very close to each other, they are lower than the efficiency of the value of the impeller without splitter blades (Fig. 9c). This leads us to conclude that the addition of splitter blades to impellers with the number of blades z=7 and blade discharge angle β_{2b}=25° is unsuitable.

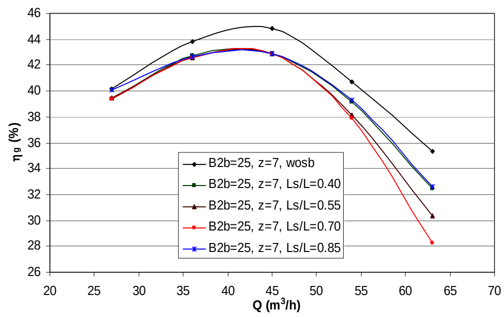
Considering the results at the best efficiency point, all additions of splitter blades decrease the efficiency, as seen on Fig. 9c. For instance, while an impeller with 70% splitter blade addition demonstrates a head decrease of 2.60% and efficiency decrease of 4.32% compared to the impeller without splitter blades, a 1.80% increase in brake horsepower has been observed. On the other hand, on an impeller with 85% splitter blade addition, a 2.60% head increase and 7.32% brake horsepower increase have been obtained compared to the impeller without splitter blades, but its efficiency decreased by 4.37%. The reason why splitter blade additions negatively affect efficiency at z=7 and β_{2b}=25° is the clogging and contraction that occur on the inlet and outlet cross sections due to the too high number of blades on the outlet (z=14 for the outlet cross section) and the too small blade discharge angle (β_{2b}=25°). As the blade discharge angle increases, the outlet contraction coefficient also grows and is inversely proportional to the number of blades. Figures 10a, 10b and 10c indicate H_m-f(Q), P_e-f(Q) and η_g-f(Q) characteristics of impellers with a number of blades z=7 and blade discharge angle β_{2b}=35°, respectively. As seen from Figs. 10a and 10b, the addition of splitter blades leads to an increase in the head and power consumption of the pump. The greatest head increase has been obtained on the impeller with 70% splitter blade increase (Fig. 10a) while the greatest efficiency increase is yielded by the impeller with 85% splitter blade increase up to the best efficiency point and by the impeller with 70% splitter blade increase after that point (Fig. 10c). Considering the best efficiency point, while efficiencies of both impellers are close to each other, the efficiency value of the impeller with 85% splitter blade addition is greater than that of the impeller with 70% splitter blade addition.



(a)



(b)

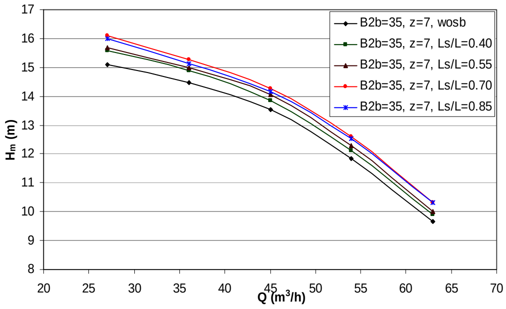


(c)

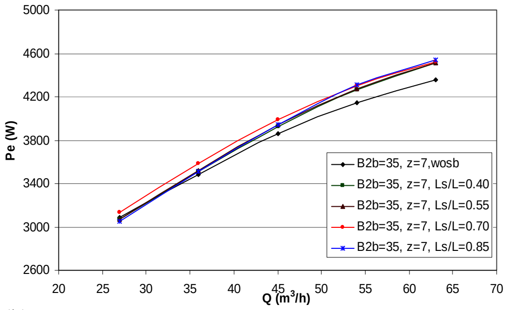
Fig. 9. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=7 and β_{2b}=25°.

Apparently, compared to the impeller without splitter blades, the impeller with 70% splitter blade increase yielded a 5.32% head increase, 3.33% brake horsepower increase and 1.92% efficiency increase. Compared to the impeller without splitter blades, the impeller with 85% splitter blade increase yielded a 4.58% head increase, 2.12% brake horsepower increase and 2.39% efficiency increase

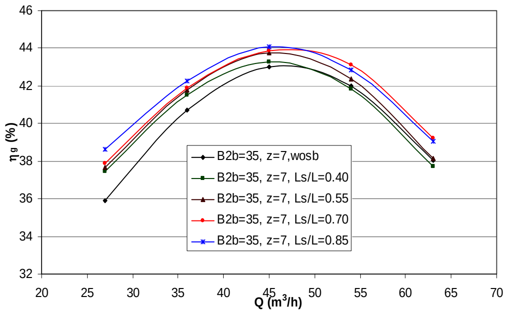
We had already concluded that splitter blade addition at z=7 and β_{2b}=25° would negatively impact efficiency. However, such addition at z=7 and β_{2b}=35° has been observed to yield an efficiency increase. Given this fact, it can be claimed that a 10° increase in the blade discharge angle in this situation eliminates the effect of contraction. While efficiency decreased at =7 and β_{2b}=25° for all splitter blade lengths, it has been observed to decrease for all splitter blade lengths at z=7 and β_{2b}=35° up to b.e.p.



(a)



(b)



(c)

Fig. 10. a) H_m-f(Q) b) P_e-f(Q) c) η_g-f(Q) characteristics of impellers with and without splitter blades for z=7 and β_{2b}=35°.

Head (H_m), break horsepower (P_e) and general efficiency (η_g) values depending on the number of blades (z), blade discharge angle (β_{2b}) and dimensionless splitter blade length ($\overline{L} = \frac{L_s}{L}$) have

Impellers with and without splitter blades used in the experiments have been assigned model numbers based on their number of blades, blade discharge angle and splitter blade lengths and H_m, P_e, and η_g values at b.e.p of the impellers according to their model numbers are presented in Figs. 11a, 11b, and 11c.

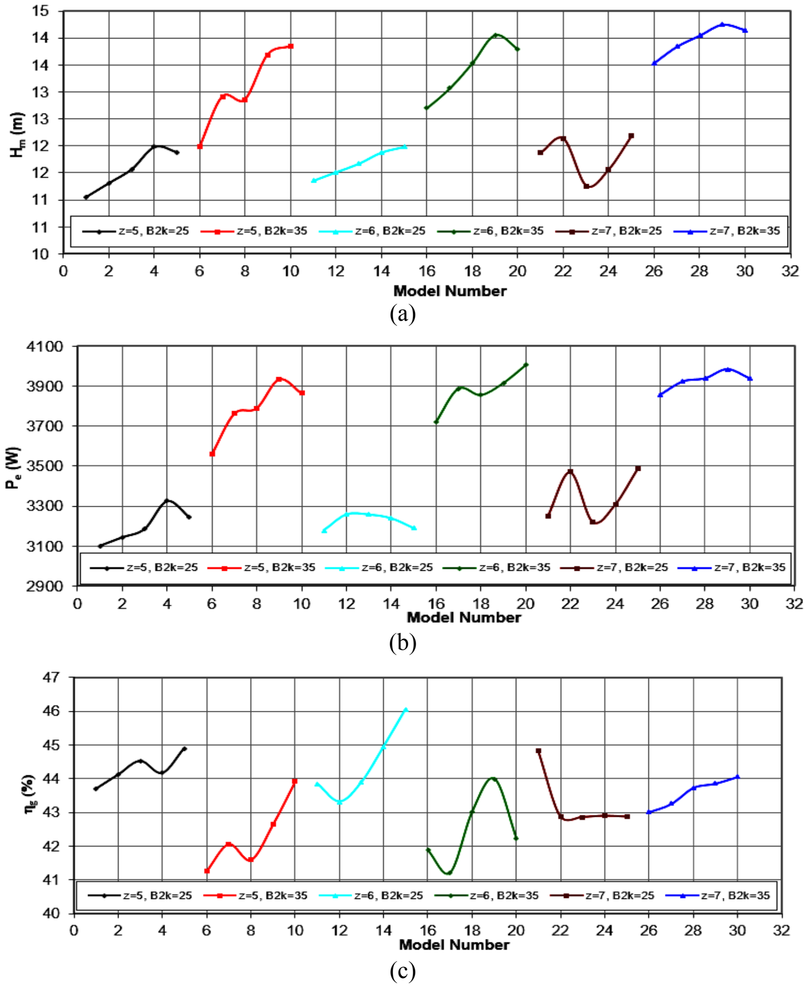


Fig. 11. a) H_m, b) P_e and c) η_g values of impellers with and without splitter blades at b. e. p.

As seen from Fig. 11a, the addition of splitter blades has ensures a head increase except for some of the impellers with splitter blades with the number of blades z=7 and blade discharge angle β_{2b} =25° (except for model numbers 23 and 24). Figure 11c shows that the splitter blade addition to impellers with number of blades z=7 and blade discharge angle β_{2b} =25° negatively affects the efficiency.

As seen from Fig. 11b, splitter blade addition has generally caused an increase in the power consumed by the pump compared to impellers without splitter blades.

As seen on Table 3 and Fig. 11c, the highest efficiency of all the impellers for b.e.p has been obtained on the impeller with the number of blades z=6, blade discharge angle β_{2b}=25° and 85% splitter blade addition compared to impellers without splitter blades. Since the increase in the brake horsepower is negligible, the 85% splitter blade addition is suitable economically. On the impeller with the number of blades z=7 and blade discharge angle β_{2b}=25°, the addition of splitter blades has negatively impacted the efficiency, yielding an efficiency value lower than that of the impeller without splitter blades for all splitter blade lengths. The best efficiency on the impeller with the number of blades z=7 and blade discharge angle β_{2b}=25° has been obtained without splitter blades.

4. CONCLUSIONS

As a results of the experimental study; an increase in head, brake horsepower and general efficiency with impellers without splitter blades is observed with increasing the blade number as similar with results of previous studies (Gölcü and Pancar, 2005; Gölcü *et al.* 2006). Moreover, it has been concluded that the blade discharge angle has greater effect on pump characteristics compared to the number of blades. As the blade discharge angle increased, head and brake horsepower have been observed to increase while a decrease has been observed in the efficiency until b.e.p. Although head value increased with higher blade discharge angle, general efficiency of the pump decreased due to a much greater increase in the brake horsepower. Due to the increase of the absolute velocity with the increase of the blade discharge angle, losses grew and pump efficiency decreased. At the same time, increase of the blade discharge angle led to an increase in separation losses. Considering the results at the best efficiency point;

Table 3 Characteristic values of impellers with and without splitter blades at b.e.p

Model Numbers	Factors			Test results (b.e.p.)		
	z	β_{2b} (º)	$\overline{L}$	H_m (m)	P_e (W)	η_g (%)
1	5	25	wosb	11.05	3101	43.71
2	5	25	0.40	11.31	3143	44.12
3	5	25	0.55	11.57	3186	44.52
4	5	25	0.70	11.98	3327	44.16
5	5	25	0.85	11.88	3245	44.89
6	5	35	wosb	11.98	3561	41.26
7	5	35	0.40	12.91	3764	42.07
8	5	35	0.55	12.86	3791	41.60
9	5	35	0.70	13.69	3935	42.65
10	5	35	0.85	13.84	3865	43.91
11	6	25	wosb	11.36	3179	43.83
12	6	25	0.40	11.52	3260	43.31
13	6	25	0.55	11.67	3260	43.90
14	6	25	0.70	11.88	3241	44.94
15	6	25	0.85	11.98	3190	46.05
16	6	35	wosb	12.70	3721	41.87
17	6	35	0.40	13.07	3888	41.21
18	6	35	0.55	13.53	3857	43.02
19	6	35	0.70	14.05	3916	43.99
20	6	35	0.85	13.79	4005	42.22
21	7	25	wosb	11.88	3249	44.83
22	7	25	0.40	12.13	3471	42.88
23	7	25	0.55	11.26	3221	42.86
24	7	25	0.70	11.57	3307	42.89
25	7	25	0.85	12.19	3487	42.87
26	7	35	wosb	13.53	3857	43.02
27	7	35	0.40	13.84	3923	43.26
28	7	35	0.55	14.05	3939	43.73
29	7	35	0.70	14.25	3986	43.85
30	7	35	0.85	14.15	3939	44.05

On impellers without splitter blades

While the highest head and brake horsepower are obtained with the blade discharge angle β_{2b}=35° and number of blades z=7, the highest efficiency has been obtained with the blade discharge angle β_{2b}=25° and number of blades z=7. Shojaeefard *et al.* (2012) have been investigated both numerically and experimentally the effects of blade outlet angle and passage width on the performance of a centrifugal pump. In their study, the optimum centrifugal pump performance has been obtained with the blade discharge angle of 30° for 21 mm blade exit width. Babayigit *et al.* (2015) conducted a numerical study to analyze the effect of blade exit angle on the performance of multistage centrifugal pump impeller for the blade exit angle values of 18°, 20°, 25°, 30° and 35°. In their study, the most convenient blade exit angle of 18° was obtained.

On impellers with splitter blades

Splitter blade addition has been observed to yield an increase in general efficiency and head on impellers with all numbers of blades and blade discharge angles other than impellers with the number of blades z=7 and blade discharge angle β_{2b}=25°.

On impellers with the number of blades z=5 and blade discharge angle β_{2b}=25°, an 85% splitter blade addition yielded 7.51% head increase, 4.64% power increase and 2.69% efficiency increase compared to impellers without splitter blades. On impellers with the number of blades z=5 and blade discharge angle β_{2b}=35°, an 85% splitter blade addition yielded 15.52% head increase, 8.53% power increase and 6.42% efficiency increase compared to impellers without splitter blades. On impellers with the number of blades z=6 and blade discharge angle β_{2b}=25°, an 85% splitter blade addition yielded 5.45% head increase, 0.34% power increase and approx. 5% efficiency increase compared to impellers without splitter blades. Since the increase in the brake horsepower is negligible, the 85% splitter blade addition is suitable economically. Unlike on other impellers, on impellers with the number of blades z=6 and blade discharge angle β_{2b}=35°, the highest efficiency has been obtained with the 70% splitter blade addition. Such addition yielded 10.63% head increase, 5.24% power increase and approx. 5% efficiency increase compared to impellers without splitter blades. On impellers with the number of blades z=7 and blade discharge angle β_{2b}=25°, none of the splitter blade addition yielded an increase in efficiency. On the contrary, the efficiency decreased by an average of

4.37% for all splitter blade sizes. Therefore, splitter blade addition is not suitable for impellers with the number of blades z=7 and blade discharge angle β_{2b}=25°. This is caused in splitter blade applications by the high number of blades (z=7) combined with the small blade discharge number (β_{2b}=25°). A decrease is also observed in the outlet contraction coefficient compared to β_{2b}=35°. On impellers with the number of blades z=7 and blade discharge angle β_{2b}=35°, an 85% splitter blade addition yielded 4.58% head increase, 2.12% power increase and 2.39% efficiency increase compared to impellers without splitter blades.

Considering the results that have been obtained, it has been concluded that for splitter blade addition, the optimal number of blades would be z=6, optimal blade discharge angle β_{2b}=25° and optimal splitter blade length in the range of $L_s=0.70 \cdot L \sim L_s=0.85 \cdot L$. Finally, it has been concluded that low efficiency, which is one of the most significant problems of pumps with low specific speed, could be solved by properly selecting the number of blades, blade discharge angle and adding splitter blades.

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