THE EFFECT OF FITTING SPLITTER BLADES ON THE PERFORMANCE OF CENTRIFUGAL PUMP IMPELLERS

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ABSTRACT

The present study is based on experimental results obtained on a centrifugal pump impeller with five blades, 124 mm outer diameter. The paper presents the results of the influence of fitting splitter blades at different positions, and different radii on the performance of the centrifugal pump impeller. The delivery parameters of the pump are recorded, with and without splitters. It is found that fitting of a single splitter at $r/R = 0.60$ results in the best compromises, which satisfies more flat efficiency curve and an increase in the pressure of the pump compared with the other configurations.

NOMENCLATURE

- b impeller width, mm
- c_m meridional velocity component, m/s
- D impeller diameter at outlet, mm
- g gravitational acceleration, m/s^2
- H pump total head, m
- Q pump discharge, m^3/s
- u peripheral speed, m/s
- Z number of blades
- ϕ flow coefficient $= Q / \pi (D\ b)_2 u_2$
- ψ head coefficient $= g H / u_2^2$
- η pump efficiency
- μ slip factor
- β blade angle

subscript

- 2 quantity at outlet

INTRODUCTION

Many investigations show that the flow in the centrifugal impeller suffers from separation, whirling and secondary flows. These phenomena specially occurred in the region outside the nominal operating range, and results in higher losses and thus lower hydraulic efficiency. Also as the space between adjacent blades at outlet increases, the slip factor decreases and as a result the delivery pressure will be declined. The traditional concept describes the flow within the impeller passage as a backward flow relative to the impeller. It gives a linear variation of radial velocity lowest at the pressure face and highest at the suction face. Many experimental studies of the velocity and pressure distributions in the impeller passage [1,2,3], found that secondary flow created by Coriolis acceleration developes toward the channel outlet, resulting in an accumulation of low energy fluids in the suction side of the passage. Also, for impellers with fewer number of blades, a reverse flow is generated, and the number of blades is not sufficient to constrain the flow in the impeller [4, 5]. Thus there is an increase in the acceleration of the flow in the suction side due to reducing the flow area there. So, it is rather convenient to add splitter blades between the original blades to reduce the effect of the wide space and thus decreasing hydraulic losses [6, 7]. In this paper, four configurations of fitted splitter blades are used, to detect the influence of fitting these splitters on the performance of centrifugal pump impeller.

SPLITTER BLADES CONFIGURATIONS

Splitter blades are those thin blades of 2 mm thick fitted in between each two adjacent main impeller blades, to influence the desirable direction, the losses in the impeller and the overall efficiency of the pump. In addition to the original impeller, nine impellers with different splitter blade configurations were tested to study the influence of splitter blades on the centrifugal pump performance. The dimensions and specifications of the original impeller tested are given in Figure (1). As to the impellers having splitter blades, the location of the splitter blades within the impeller channel was altered in four ways. In the first case, each splitter blade was inserted symmetrically at the middle between each two adjacent impeller blade (Figure (2-a)). In the second case, each splitter blade was inserted at one third of the periphery from the positive-pressure face of the main blade (Figure (2-b)). In the third case it was inserted at one third of the periphery from the negative-pressure face of the main blade (Figure (2-c)). In the fourth case two splitters were inserted symmetrically and equally spaced between the main impeller blades (Figure (2-d)). Further, the lengths of the splitter blades (similar shap as the main impeller blades) were changed. Thus the effect of splitter blade configurations (position and length) upon the pump performance was examined. The different configurations are arranged as follows:

- Group I: One splitter at the middle between each two impeller blades at different radii ($r/R = 0.45$, 0.60 and 0.75)
- Group II: One splitter fitted nearer to the pressure side of the following blade at one third of the flow channel periphery, at two different radii ($r/R = 0.60$ and 0.75)
- Group III: One splitter fitted nearer to the suction side of the preceding blade at one third of the flow channel periphery at two different radii ($r/R = 0.60$ and 0.75)
- Group IV: Two splitters equally spaced between each two impeller blades at different radii ($r/R = 0.60$ and 0.75)

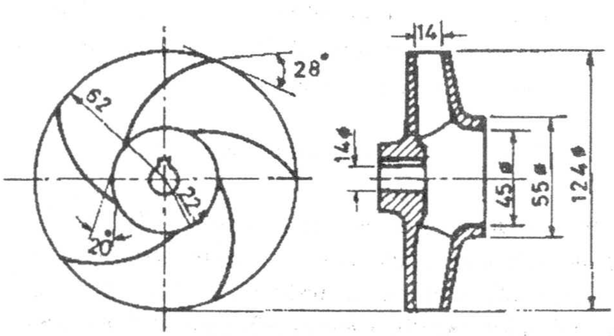
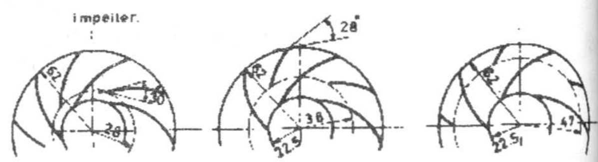
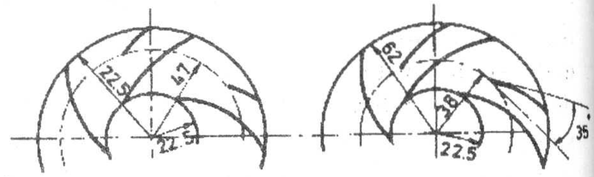


Figure 1. Dimensions and specifications of the original impeller.

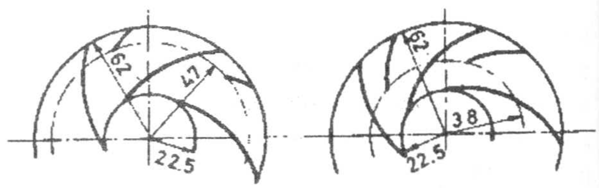
No splitters could be fitted at $r/R = 0.45$ (i.e. long splitters) for groups II, III, and IV because the flow passage at inlet will be blocked.



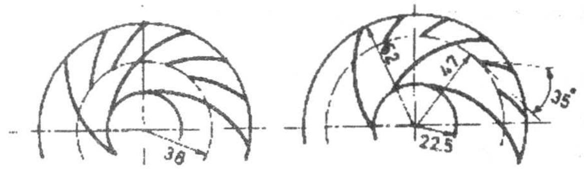
2-a. One symmetrical splitter at different radii.



2-b. One splitter near the pressure face of the main impeller blade.



2-c. One splitter near the suction face of the main impeller blade.



2-d. Two symmetrical splitter blades.

Figure 2. Splitter blades configurations.

APPARATUS AND MEASURING METHODS

Figure (3) shows the schematic diagram of the experimental equipment. Pump,"1" is connected to a 50 mm diameter suction and delivery pipes. Water from a tank,"2" was led to the pump. The pump is driven by an AC electric motor,"3" rotating at 2950 rpm. The pump head is measured by a differential pressure transducer,"4" with readout amplifier,"5". The torque and speed of the pump are measured by a torque speed transducer, "6", the measured value is directly indicated on the digital readout amplifier,"7".

The pump discharge is measured by a calibrated orifice meter "8". The pump discharge is regulated by the throttle valve "9". The experiment was carried out in a closed loop test rig type. The accuracy range of the measuring devices are ± 0.01 Nm , ± 1 rpm, ± 0.1 kPa, and ± 0.5 l/m.

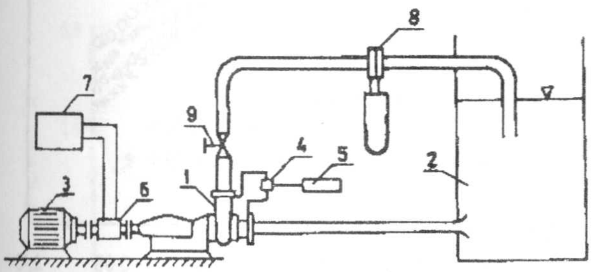


Figure 3. Schematic diagram of the experimental test rig.

TEST PROCEDURE

The pump was tested at first without splitter blades, discharge, head, and power were recorded and pump performance was plotted. Then the pump with different splitter blades configurations were tested. The performance of each group has been compared with the pump performance without splitter blades. In order to show more clearly the effects of the splitter blades on the optimum performance of the pump, the optimum performance has been abstracted from each group. The best performance of the four groups are compared with each other to abstract the best one of all configurations. The uncertainties in the performance data are : $\phi = \pm 0.00045$, $\psi = \pm 0.0008$, and $\eta = \pm 0.0075$.

RESULTS AND DISCUSSIONS

General effect of splitter blades on the flow inside the impeller passages

In general at normal deliveries, it would be expected that the head developed by the pump increases with increasing the number of blades [8]. The insertion of one splitter blade between each two impeller blades is corresponding to a double increase in the number of blades. However, the existence of two splitters between each two impeller blades leads to a triple increasing in the number of blades. Therefore, the existence of splitter blades increases the slip factor μ, and consequently increases the pump head. The addition of splitter blades, also, causes an increase in the through-flow velocity, and consequently increases the through-flow friction losses. The head increment per blade becomes smaller as the total number of blades gets larger, i.e. by increasing the number of splitters. Now, there are two opposing factors that arise when inserting splitters; the added blades lead to increasing friction losses and also reducing the flow area through the impeller, while the head developed increases due to the resulting increase in the slip factor according to the following equation, [9].

$$\mu = 1 - \frac{\pi . \sin\beta_2}{Z(1 - \frac{c_{m2}}{u_2}\cot\beta_2)} \qquad (1)$$

Figure (4) shows a schematic representation of the flow through the impeller. It is possible to conceive that the boundary layer on the main blade suction face separates and back flow occurs along the suction side blocking off the passage near the discharge face and reversing the flow direction in the passage. The fluid then accelerates towards the impeller eye. The relative eddies created at the outlet support the back flow in its direction causing the fluid to change its direction. So, the addition of the splitters will decrease the slip of the flow due to the decrease of extreme shift of flow toward the suction side of the impeller passage, and prevent the fluid at the suction blade face to come over the blade tip from the pressure side of the blade of neighboring passage.

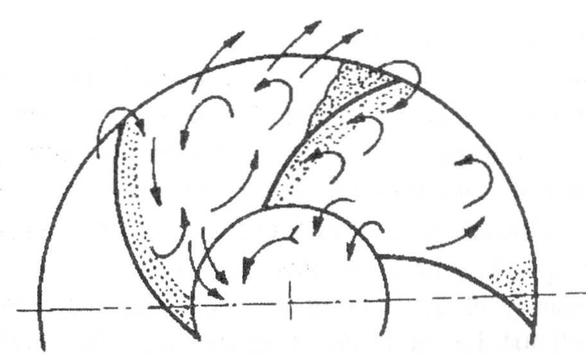


Figure 4. Boundary layer separation and reversed flow in the flow passage of the original impeller.

Effect of one symmetrical splitter

Figures (5) and (6) show the head coefficient ψ, and efficiency η against flow coefficient ϕ; for the original impeller and the impeller with one splitter at different radii (group I). The pump without splitters gives a peak efficiency of 59.1 % at ϕ = 0.0523, whereby ψ = 0.426. It is clear that the insertion of one splitter approximately has no effect on the ψ - ϕ curve up to one third of the maximum discharge. Equation 1 shows that the slip factor μ decreases with increase in pump discharge, and more slip occurs, and consequently more reduction in pump head. Thus by using one, or two splitters, according to equation 1 the slip factor increases, even at higher flow rates.

The splitter at r/R = 0.45 i.e. the longer one gives a negligible decrease in both pump head and maximum efficiency. This is apparantly due to the additional surface frictional losses of the splitters; just opposing any improvement in the pump head as a result of increasing slip factor. On the other hand the splitters decrease the flow area at inlet causing the impeller partially blocked at the impeller inlet radius, resulting in increase in the entry losses. The only benefit resulting as expected, is that the efficiency curve becomes little flat. Although the peak efficiency of the pump with long splitters is lower than that without splitters, yet the peak efficiency in this case occurs at a higher discharge. The pump in this case gives a maximum efficiency of 58.7 % at ϕ = 0.0581 and thereby ψ = 0.386.

The splitters at r/R = 0.75 (i.e. the shorter ones) have less friction loss, but due to their very short length they create many disturbance in the passage due to the wide passage area behind them. No complete guidance to the flow exists, and the splitter surface is not sufficient to constrain the flow in the impeller passage. The pump in this case gives a maximum efficiency of 58.25 % at ϕ = 0.0608, and thereby Ψ=0.404 i.e. at larger discharge than the original impeller.

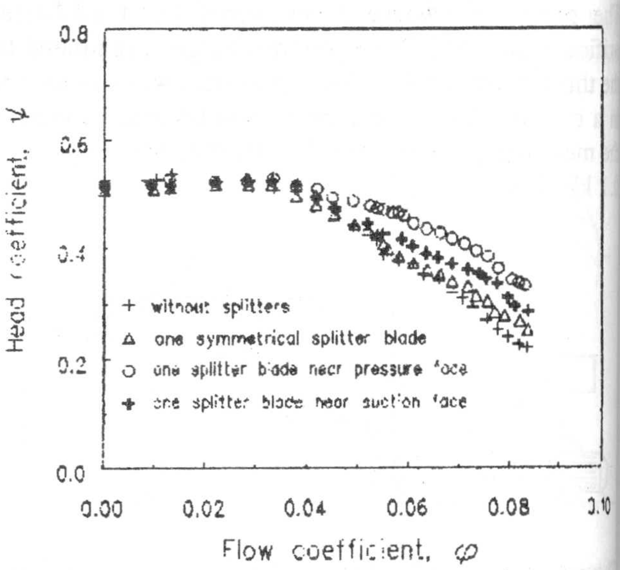


Figure 5. Head coefficient against flow coefficient group I - with one symmetrical splitter blade.

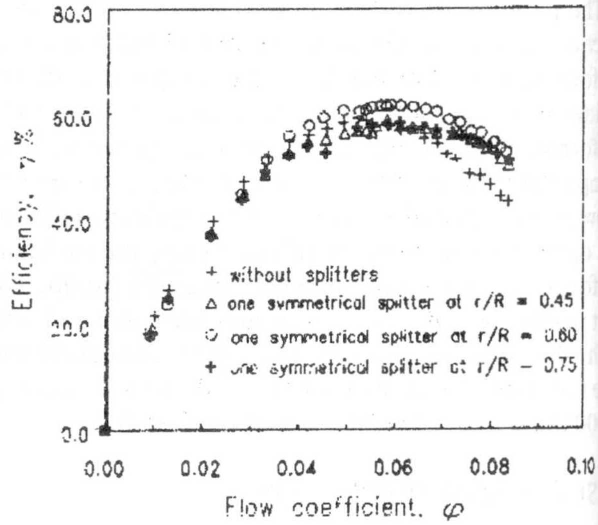


Figure 6. Pump efficiency against flow coefficient group I - with one symmetrical splitter blade.

The splitters of medium length at r/R = 0.6 give th optimum performance, due to increasing slip factor less blockage at entry and decreasing recirculator flow. In this case the friction loss is smaller than th of the longer splitters, and the pump head increase specially at high discharges. The increase in pum head is due to an enhancement in hydraulic efficiency decreasing the pressure gradient between the pressur and suction blade surfaces, and hence more uniform velocity and pressure distributions inside the passage in addition to an increase in the slip factor. A maximum efficiency of 61.8 % at flow coefficient $\phi = 0.059$, and $\psi = 0.459$ is obtained with these splitters.

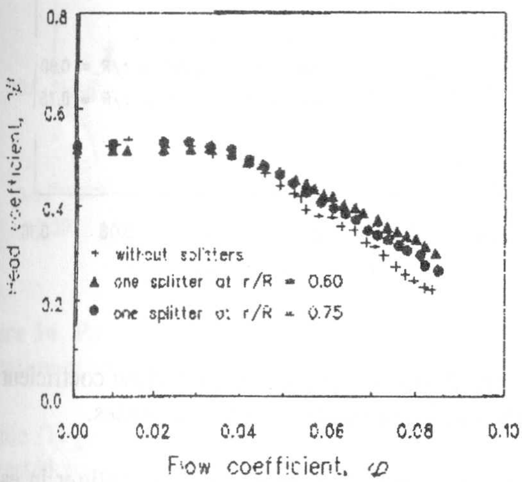


Figure 7. Head coefficient against flow coefficient group II - One splitter near blade pressure face.

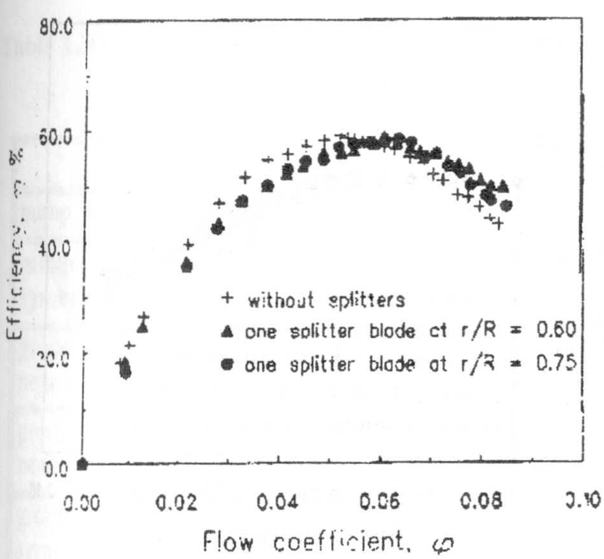


Figure 8. Pump efficiency against flow coefficient group II - one splitter near blade pressure face.

Effect of one splitter near the pressure face of the main impeller blade

Figures (7) and (8) show the pump performance with splitter blades group II near the pressure face of the main impeller. This configuration gives a considerable increase in pump head specially at large discharges due to decrease in hydraulic losses, thus pump efficiency curve seems to be somehow flatter at its middle portion. It is clearly shown in Figure (7) that the long splitters (at r/R = 0.60) gives a lesser flat efficiency curve than the short splitters (at r/R = 0.75).

Effect of one splitter near the suction face of the main impeller blade

Approximately similar results are obtained when using splitters of group III, (the splitter near the suction face of the main impeller) as shown in Figures (9) and (10). This is expected because the flow passage between the main blade pressure face and the splitter in group II is very similar to that between the main blade suction face and the splitter in group III. Thus the characteristics of the pump using splitters of group II or group III are approximately the same and no sensible difference between them, is found.

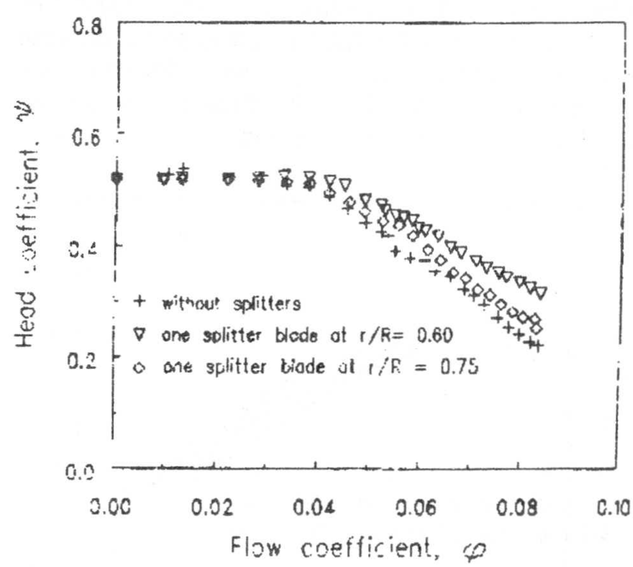


Figure 9. Head coefficient against flow coefficient group III - one splitter near blade suction face.

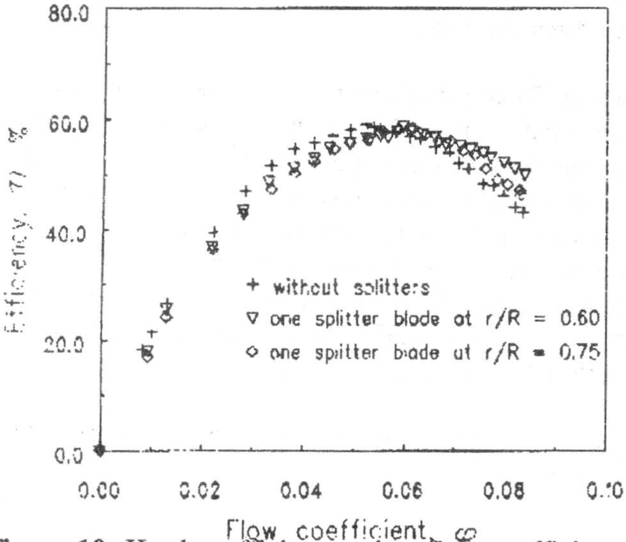


Figure 10. Head coefficient against flow coefficient group III - one splitter near blade suction face.

Effect of two symmetrical splitters between the main impeller blade

Figures (11) and (12) show results of head coefficient and pump efficiency using two sets of splitters. The general improvement in the pump head is seen to be better for the longer splitters (r/R = 0.6). Although the use of two splitters increases the slip factor, yet on the other hand the friction losses increase, and the flow area decreases. Thus the final result is some increase in pump head at higher discharges, more flat efficiency curve, and the point of peak efficiency is shifted towards the higher discharge region.

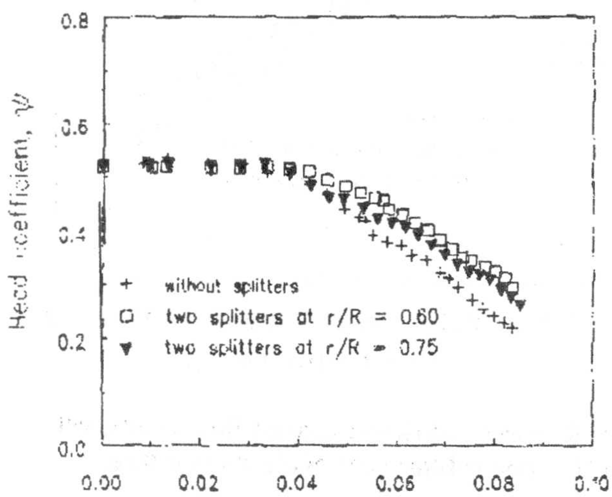


Figure 11. Head coefficient against flow coefficient group IV - two symmetrical splitter blades.

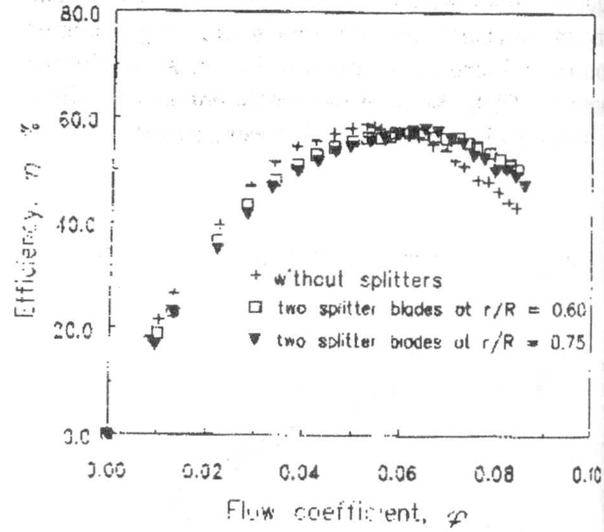


Figure 12. Pump efficiency against flow coefficient group IV - two symmetrical splitter blades.

Figures (13) and (14) show the best splitter in each group compared with the pump without splitters. It is clearly shown that the pump with a single set of splitters inserted at r/R = 0.60 gives the optimum improvement in the pump performance.

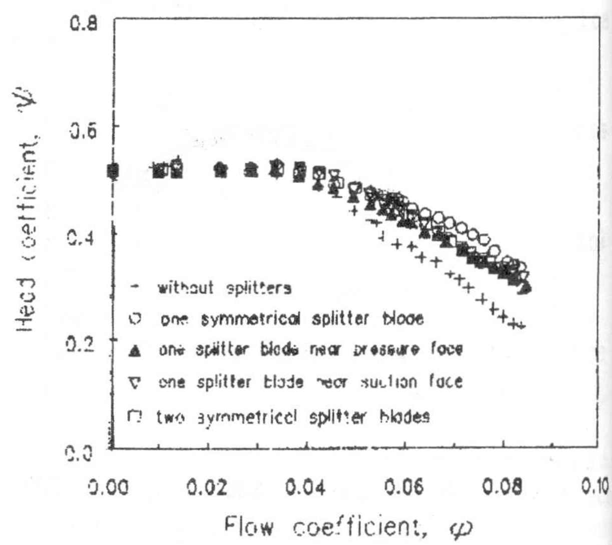


Figure 13 Head coefficient against flow coefficient for optimum splitter abstracted from each group.

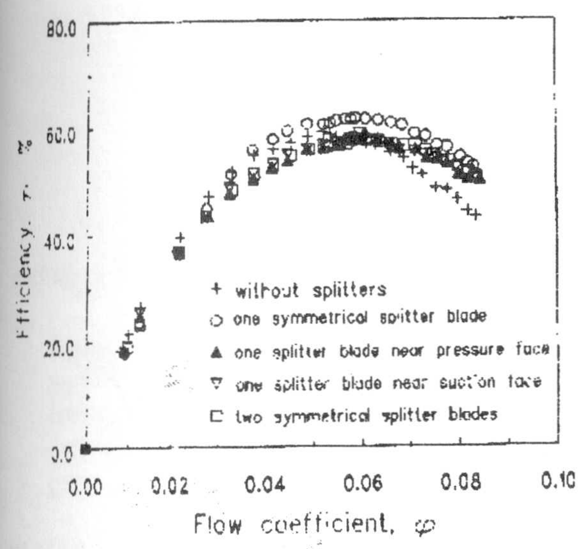


Figure 14. Pump efficiency against flow coefficient for optimum splitter abstracted from each group.

Table (1) gives a summary of the optimum results, abstracted from each splitter blade group. The table includes maximum efficiency, and the corresponding flow and head coefficients. It is clearly indicated that all splitter blades at r/R = 0.60 give the best performance in its group.

Table 1. Optimum pump performance obtained from each splitter blade group.

	ϕ	Ψ	$\eta, \%$
pump without splitters	0.0523	0.426	59.1
group I one symmetrical splitter	0.059	0.459	61.8
group II one splitter near pressure face	0.061	0.419	58.77
group III one splitter near suction face	0.0596	0.433	58.77
group IV two symmetrical splitters	0.0582	0.422	57.5

CONCLUSIONS

From the preceding study one can conclude:

1- Using splitter blades between the main impeller blades improves the flow conditions inside the flow passages, increases the slip factor, and consequently increases the pump head.

2- Whether the splitter blades are inserted near the suction side or near the pressure side the effect on the pump performance will be the same. The pump head is improved, and efficiency - flow coefficient characteristics become somehow flat.

3- The insertion of two sets of splitters, gives little improvement in the pump performance (ϕ - ψ curve)

4- The splitter configuration that affects the pump performance most, is the one symmetrical splitter at r/R = 0.6 which gives the highest pump efficiency and leads to the most significant increase in pump head.

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