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THE EFFECT OF CENTRIFUGAL PUMP IMPELLER TRIMMING ON PUMP P ERFORMANCE AND GENERATED NOISE (AN EXPERIMENTAL INVESTIGATION)

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ABSTRACT

In the present study, centrifugal pump noise is examined with increasing the redial gap between the impeller and pump casing. Increasing the radial gap will be obtained by impeller trimming. Impeller trimming is the process of decreasing the diameter of an impeller by machining to decrease the energy added to the fluid system. The impeller trimming leads to a reduction both of pump flow rate and head. A closed loop with circulation pump is designed and fabricated to study the effect of impeller trimming on centrifugal pump performance and pump noise. Pump performance is obtained for the three impellers at constant drive speeds. Also, sound pressure level and frequency spectrum of noise is measured at different circumferential positions around impeller for three impeller diameters in the same casing. Results of pump performance at constant speed are found to be in good agreement with affinity laws. Finally, the sound pressure level is found to increase with impeller trimming at some locations around the impeller and decreases at others due local changes in flow behavior.

KEYWORDS: Centrifugal Pump, Impeller Trimming, Radial gap, Pump Performance, Sound Pressure Level, Noise.

تأثير قص حافة دفاع المضخة الطاردة المركزية على أداء المضخة و الضوضاء المتولدة

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الملخص

فى هذه الدراسة؛ سيتم البحث فى ز يادة المسافة القطرية بين دفاع المضخة الطاردة المركزية وغلافها و تأثير ذلك على كفاءة المضخة وكذلك الضوضاء الصادرة منها. تتم عملية القطع فى حافة دفاع المضخة الطاردة المركزية عن طريق اجراء عملية خراطة عليه بغرض تقليل الطاقة التى يعطيها للسائل وعندها فيقل كلا من كمية السائل الخارج من المضخة وضغطه. تأتى هذه الطريقة والتى الغرض منها تقليل كمية السائل الخارج من المضخة كبديل عن غلق بلف الطرد تجنباً لما قد يحدث من فقد فى الطاقة. لهذا تعتبر هذه العملية أكثر توفيرا و كفاءةً مقارنة بشراء مضخة جديدة تعطى كمية السائل المطلوبة. وذلك مع الآخذ فى الاعتبار تأثير تقليل قطر الدفاع على الضوضاء المتولدة من المضخة لما فيه من تأثير على بعض المستخدمين مثل المستشفيات والبنوك الخ.

وضع تصميم لدائرة مياه مغلقة بها مضخة طاردة مركزية وتم تصنيعها لدراسة تأثير القطع فى حافة دفاع المضخة الطاردة المركزية على أداء المضخة والضوضاء الصادرة منها. وتم رسم كلا من منحنى الأداء والقدرة للمضخة

عند الثلاثة أقطار المشار إليها عند ثبوت سرعة الدوران وتم التأكد من صحة النتائج وفقا للـ (قوانين التقارب). وكذا تم قياس مستوى ضغط الصوت والتحليل الطيفى عند أكثر من موضع لدفاع المضخة وذلك لثلاثة أقطار مختلفة للدفاع عند نفس غطاء المضخة.
وجدت النتائج الخاصة بمنحنى الأداء والقدرة موافقة للنتائج النظرية المستنتجة من الـ (قوانين التقارب). وقد وجدت زيادة في مستوى ضغط الصوت مع تقليل كمية السائل التى تخرج من المضخة. وكذلك وجدت زيادة فى مستوى ضغط الصوت مع تقليل قطر الدفاع عند بعض المواضع ووجد نقصان فى مستوى ضغط الصوت عند مواضع أخرى. يختلف مستوى ضغط الصوت بين موضع لأخر على دفاع المضخة لنفس كمية السائل وقطر الدفاع.

الكلمات المفتاحية : مضخة طاردة مركزية ، قص حافة الدفاع ، المسافة القطرية، كفاءة المضخة، مستوى ضغط الصوت، الضوضاء

1. INTRODUCTION

A centrifugal pump is a rotating machine in which flow and pressure are generated dynamically. Centrifugal pumps are used to transport liquids/fluids by the conversion of the rotational kinetic energy to the hydro dynamics energy of the liquid flow.
In the typical simple case, the fluid enters the pump impeller eye along or near to the rotating axis and is accelerated by the impeller, flowing radially outward into a diffuser or volute chamber (casing or discharge nozzle), from where it exits.
Impeller trimming is the process of decreasing the diameter of an impeller by machining to decrease the energy added to the system fluid. Impeller trimming provides a useful correction to pumps that are oversized for their application. Trimming an impeller is an alternative to purchasing a smaller impeller from the pump manufacturer. Trimming a pump impeller is often neither difficult nor expensive.
Impeller trimming reduces tip speed, which in turn directly lowers the amount of energy imparted to the system fluid and lowers the flow and pressure generated by the pump consequently, the pump noise. A primary benefit of reducing impeller size is decreasing operating and maintenance costs, in addition to decreasing pump noise. Less fluid energy is wasted in the bypass lines and across throttle valves, or dissipated as noise and vibration through the system in addition to reducing the wear on system piping, valves, and piping supports. However, trimming an impeller changes its operating efficiency, and impeller diameters are rarely reduced below 70 percent of their original size. Excessive trimming can even result in a mismatched impeller and casing, resulting in internal fluid recirculation and reduced efficiency. Pump generated noise is considered for some applications and pump users like hospitals, hotels, banks, etc that the noise can affect on customer satisfaction and comfort. So, the Impeller trimming effect on the sound pressure level shall be investigated. [2]
Sound pressure level (SPL) or sound level is a logarithmic measure of the effective sound pressure of a sound relative to a reference value. It is measured in decibels (dB) above a standard reference level. [1]. The standard reference sound pressure in air or other gases is 20 µPa, which is usually considered the threshold of human hearing (at 1 kHz). Mechanical sources are vibrating components or surfaces which produce acoustic pressure fluctuations in an adjacent medium. In centrifugal pumps, noise is generally associated with improper installation of coupling or other assembled parts or if the pump speed is near or passes through the lateral critical speed. The liquid noise sources can be produced by the pressure fluctuations by liquid motion. Potential fluid dynamic sources include unsteady flow fluctuations, vortices, recirculation fluctuation, cavitation, and impeller interaction with the pump cutwater. [6]. Environmental noise usually does not emanate directly from the energy source; rather, it is transmitted along mechanical or liquid paths before it finally radiates from some vibrating surface into the surrounding environment. The approaches to treating pump noise generally include source modification or transmission interruption. The pump noise can be reduced by the modification of the pump configuration, and casing vane clearances, casing cutwater (tongue). [7]
Also, checking the suction pressure, operating condition, and bearing lubricant are important to reduce the pump noise. Interruption of transmission is to prevent sources

from generating airborne (or over side) noise by interrupting the path between the energy source and the listener. This approach may range from isolation mounts at the source to physically removing the listener.

2. TEST RIG

The objective of the test rig is to investigate experimentally the effect of impeller trimming on the centrifugal pump noise at variable flow rates in addition to the effect of the trimming on the pump performance. The test rig is a simple water closed loop, figure 1, use the pump to draw the water from the tank through the suction pipe back in to the tank through the discharge pipe. The test rig components are water tank with 19litres volume, suction ball valve, discharge globe valve, and centrifugal pump.

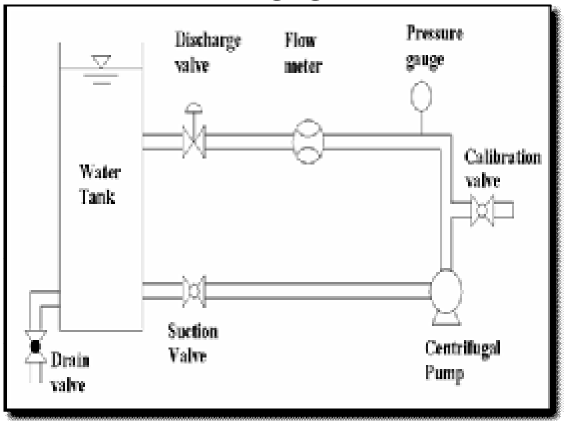

Fig.1 Schematic diagram for centrifugal pump test rig

The centrifugal pump used in our test is a close coupled centrifugal pump with the following specifications:

- Pump is driven by 2-pole induction motor 0.5Hp at max load, and 2850 rpm.
- Threaded ports for suction and discharge flanges.
- Single impeller φ112 mm with 6 blades.
- Blade pass frequency (BPF) is 285 Hz.
- $NPSH_{req}$ is 0.3m at B.E.P.
- Suction specific speed is 3960.

The used measuring devices to draw the pump performance curves are the rotameter, pressure gauges, and AVO-meter. The sound pressure level produced from the pump is measured using on-line MATLAB programming with vocal microphones fixed on the pump casing as shown in figure 2. Sound level meter uses a Discrete Fourier Transform (DFT) to determine the frequency spectrum as shown in figure 3.

Fig. 2 Vocal microphones on pump casing

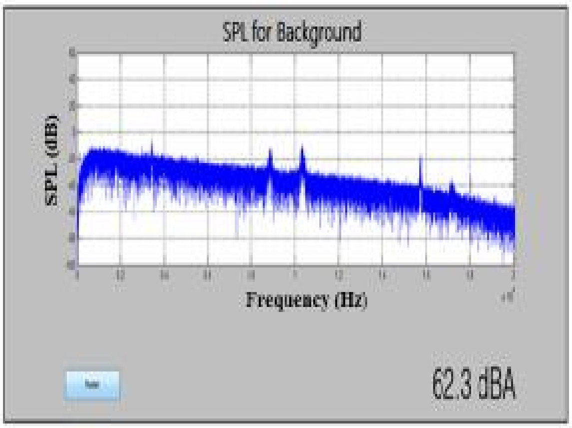

Fig.3 Background sound pressure level spectrum

All measuring devices are calibrated. The pressure gauges calibrated using a test bench with calibrated master gauge. The value indicated by the pressure gauge is compared to the corresponding value indicated by the master gauge at each pressure increment. The percent error between the two values is calculated and the gauge reading is adjusted as necessary. The rotameter is calibrated by collecting a specific volume of liquid in a certain time. This test was repeated more than one time and checks the average results with the flow meter reading. Calibration for the sound level meter is very important to ensure that the results are accurate and representative for the centrifugal pump noise. Sound pressure level meter calibration was performed by comparing the results captured by the sound level meter (MATLAB programming) and another calibrated sound level meter with a built in calibrator. From the results of comparison the sound level meter coefficient (C) in MATLAB program was corrected.

The performance curves (H - Q, Ps – Q, and η – Q) of the tested pump at constant operating speed (2850RPM) are obtained experimentally.

(H-Q) curve is obtained by measuring the pressure and the flow. The water head was calculated from equation (1) where the water density is (1000 Kg/m3) and the gravity is (9.81 m/s2). Figure 4 shows the (H-Q) curve.

$$\mathbf{H} = \frac{P}{\rho g} \qquad \textbf{(1)}$$

Where:
H: pump head (m)
P: Pressure (Pa)
ρ: Water density (kg/m3)
g: gravity (m/s2)

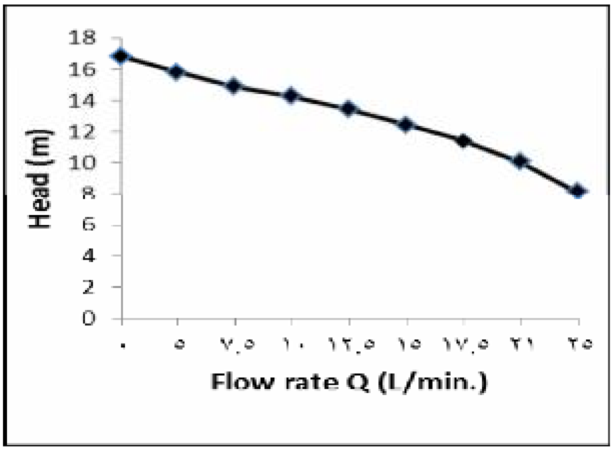

Fig.4 Pump Performance H-Q Curve at "2850 RPM"

(P_s-Q) curve is obtained by measuring the motor current and the flow. The electric motor input current was measured using "AVO-meter" at different flow rates and the consumed power was calculated from the equation (2). Constant power factor and motor efficiency were taken from figure (5) and (6) for the used single phase induction motor. (P_s – Q) performance curve is shown in figure 7.

$$\mathbf{Ps = V.\ I.\ Cos\ \varphi.\ \eta_m} \qquad \mathbf{(2)}$$

Where:
Ps: Consumed motor power
V: Motor voltage
I: Motor current
Cos φ: the motor power factor (~64%)
ηm: Motor efficiency (~80%)

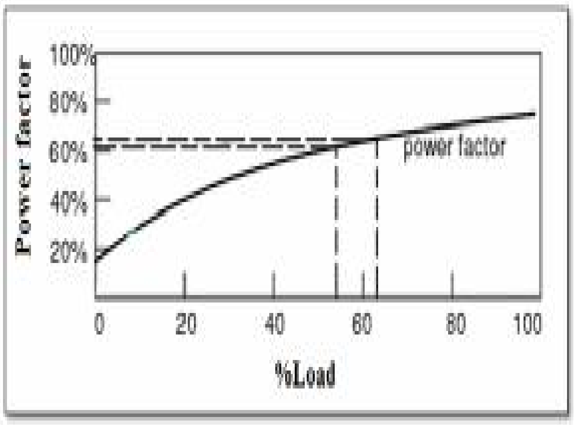

Fig. 5 Motor power factor variation with load

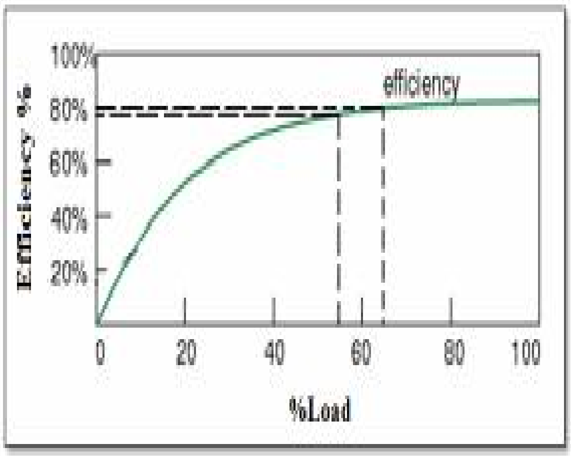

Fig. 6 Motor efficiency variation with load

From the performance (H – Q) and (Ps – Q) curves, the pump efficiency is calculated from equation (3). (η - Q) performance curve is shown in figure (8).

$$\mathbf{ή\ (overall\ efficiency)} = \frac{\rho\, g\, Q\, \Delta H}{P_s} \qquad \mathbf{(3)}$$

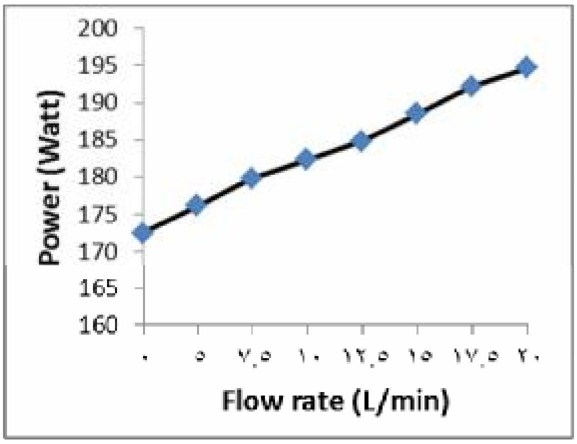

Fig. 7Pump Performance (Ps - Q) Curve at "2850 RPM"

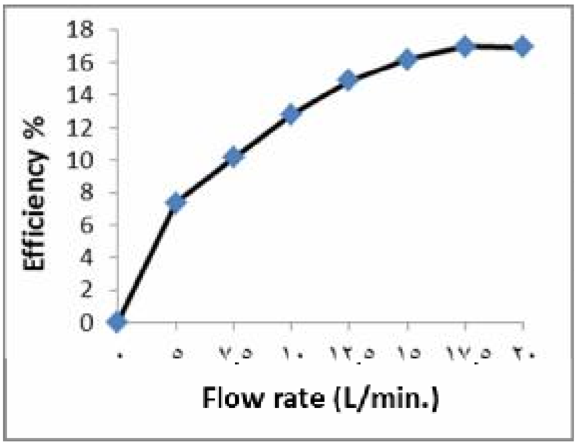

Fig. 8 Pump Performance (η - Q) Curve at "2850 RPM"

Igor et al. mentioned in [4] that the fiber glass is one of the most effective materials in sound proofing. A wooden cover was fabricated from fiber glass and installed to cover the pump and also mounted between the pump and motor to prevent noise transmitted to the pump axially. The sound pressure level slightly decreases after sound insulation installing.

3. TEST MEASUREMENTS PROCEDURES

Pump performance and measured noise will be captured for three different impeller diameters:

- Original impeller of diameter (D=112mm).
- 1st trimmed impeller of diameter (D=108 mm).
- 2nd trimmed impeller of diameter (D=104 mm).

Noise measuring will be captured at certain flow and impeller size where measurements will be for 4 points rounded on the pump diffuser, referred to a reference angle (φ), as shown in figure 9where the datum of the reference angle is considered the beginning of the flow diffusion area. Along with the noise measurements, the performance curves of the pump are detected for the above three cases.

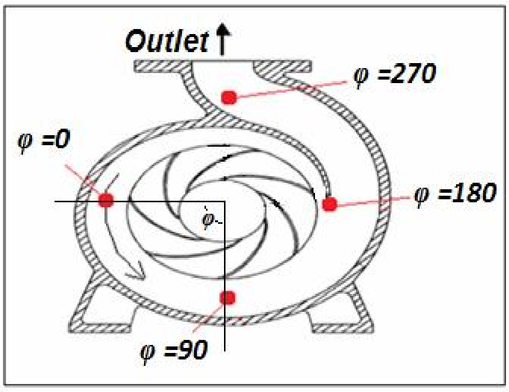

Fig. 9 Reference angle on the pump

4. RESULTS AND DISCUSSIONS

Pump performance curves changes with impeller trimming according to affinity laws. Also, pump sound pressure level changes with impeller trimming according to pressure fluctuation changing.

Results will be showed in three sections. Section (a) will shows the pump performance with impeller trimming, section (b)will shows the results of the pump noise with impeller trimming, and section (c) will show the sound spectrum peaks.

Section (a): Pump performance results

According to affinity laws, it is expected that both of pump flow, head, and shaft power will be decreased as a result of impeller trimming, as shown in equation (4), (5), and (6). However, the power index shown in the affinity laws is not accurate. True affinity requires geometric similarity, which in turn yields the scaling and similitude relationships. [5]

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1} \quad \textbf{(4)}$$

$$\frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2 \quad \textbf{(5)}$$

$$\frac{Ps_2}{Ps_1} = \left(\frac{D_2}{D_1}\right)^3 \quad \textbf{(6)}$$

Where
D_1: impeller diameter before trimming.
D_2: impeller diameter after trimming.
Q_1: Pump flow before trimming.
Q_2: Pump flow after trimming.
H_1: Pump head before trimming.
H_2: Pump head after trimming.
Ps_1: Pump power before trimming.
Ps_2: Pump power after trimming.

Figure 10, 11 and 12 show the difference between the experimental H-Q curve, power curve, and efficiency curve for the original impeller diameter, 1st, and 2nd impeller diameter of 104mm.

The results shown in figures 10 and 11 are matching with affinity laws. Also from figure 12, it is shown that the efficiency decreases with impeller trimming to impeller

diameter of 104mm due to more recirculated flow consequently pump performance. This is matching with Tsang [9].
According to affinity laws, it is shown the difference between (H - Q) and (Ps – Q) curves resulting from the theoretical calculations and experiment. From figure 13, it is found a slightly difference between H-Q curve drawn from the theoretical calculations and H-Q curve drawn from the experiment after 1st trimming. But, a large difference is found after 2nd trimming as shown in figure 14.

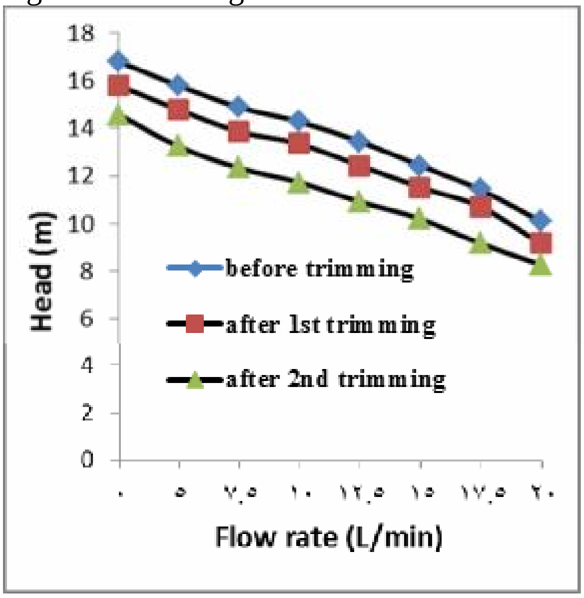

Fig. 10 (H-Q) curve of the pump at three impeller diameters

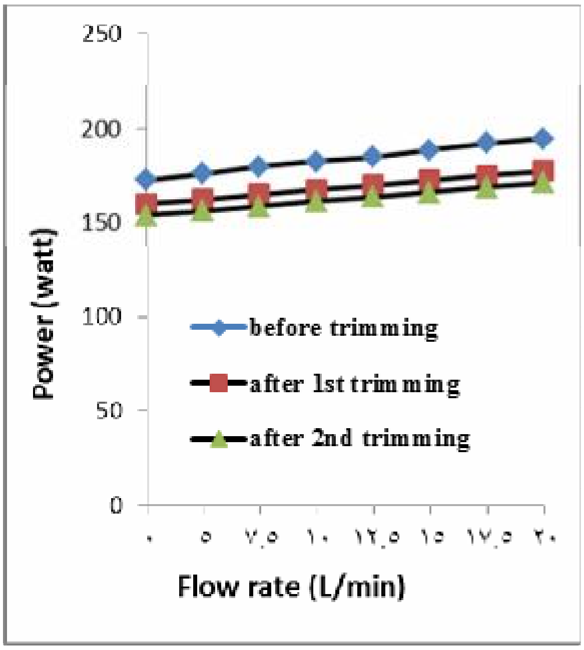

Fig. 11 (Ps-Q) curve of the pump at three impeller diameters

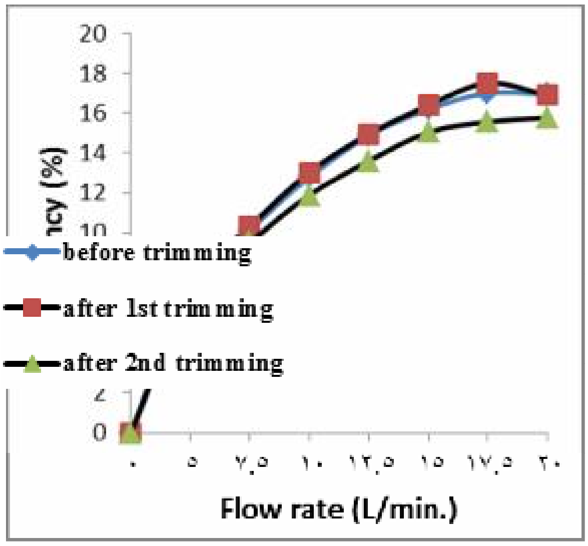

Fig. 12 (η-Q) curve of the pump at three impeller diameters

The large difference is due to increasing the radial gap consequently the recirculated flow. The recirculated flow consumes power but not discharged to the pump discharge.

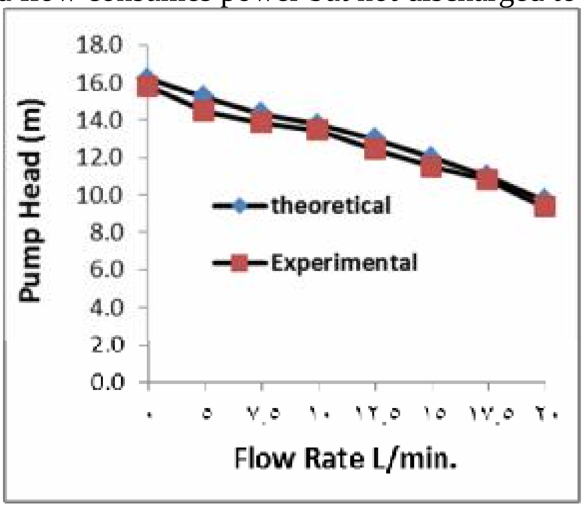

Fig. 13(H - Q) curve for theoretical and experimental results after 1sttrimming at "RPM=2850"

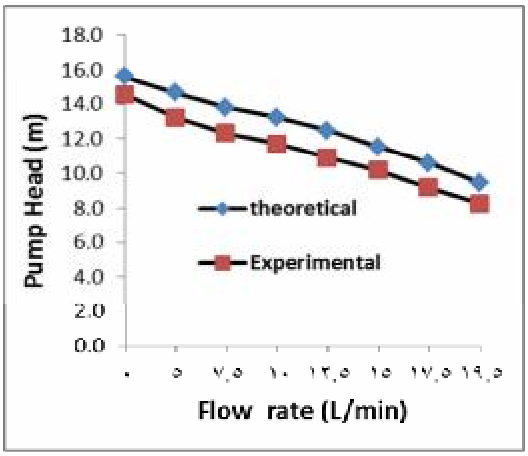

Fig. 14 (H – Q) curve for theoretical and experimental results after 2nd trimming at "RPM=2850"

From figures 15 and 16, it is found a large difference after 1st and 2nd trimming. The large difference is due to the recirculated flow in addition to inaccurate motor power factor and motor efficiency for two phase induction motor.

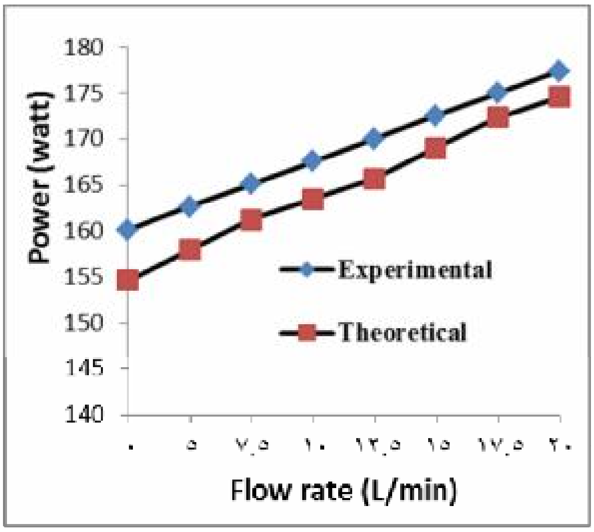

Fig. 15(Ps – Q) curve for theoretical and experimental results after trimming at "RPM=2850"

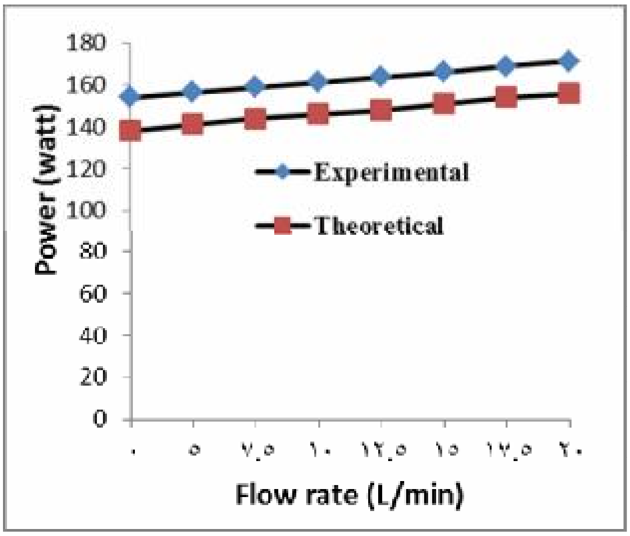

Fig. 16(Ps – Q) curve for theoretical and experimental results after 2ndtrimming at "RPM=2850"

Section (b): Pump noise

The measured values from the sound level meter in "dB" are the total sound pressure level "Lpt" including the back ground noise "Lpg".

The pump sound pressure level "Lpm" in "dB" is calculated from equation (7):

$$L_{pm} = 10 \, Log_{10} \left(10^{L_{pt}/10} - 10^{L_{pg}/10}\right) \qquad (7)$$

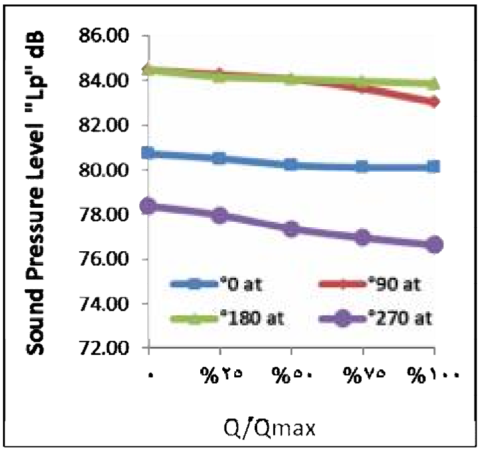

Fig. 17Sound pressure level at different flow rates and reference angles for untrimmed impeller

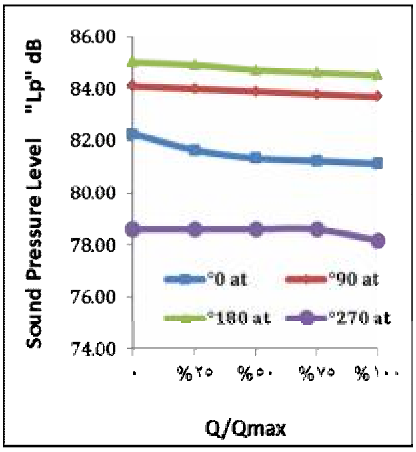

Fig. 18 Sound pressure level at different flow rates and reference angles for trimmed impeller down to “108mm” diameter

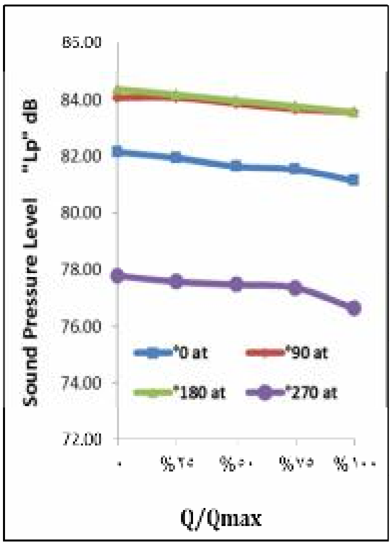

Fig.19 Sound pressure level at different flow rates and reference angles for trimmed impeller down to (104mm)

From figures (17), (18), and (19) it is shown that the highest sound pressure level is found to occur close to the tongue. One may attribute this results to what follows:
When sound from the positive direction reaches the tongue, most of the sound energy will be directed towards the pump outlet and some fraction will continue for a new circulation along the positive direction. On the contrary, part of the sound energy that had been directed towards the pump outlet (c) could be reflected due to a change in the direction of the duct or the presence of a singularity and re-emitted back to the pump (b) as shown in figure20.

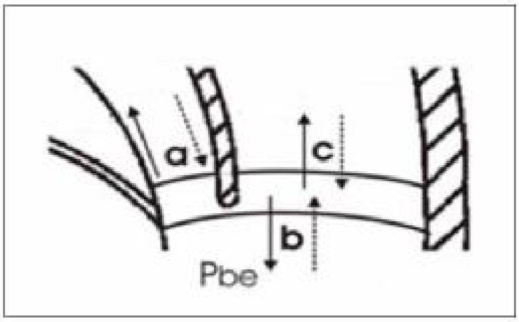

Fig. 20 Flow recirculation, discharged, and backed to pump [10]

All these fluctuations will increase the sound pressure level at reference angle φ=180º. The results match with Perotti et al [10] which reported that the higher sound pressure level is near to the impeller tongue.

The maximum flow rate discharged from the impeller channels is near to the impeller tongue which cause highly sound pressure level near to the impeller tongue as Haung shows in [11].The flow rate distribution in all channels of the impeller as shown in figure 21. It can be seen that the values of flow rates in two channels (Channel 1 and 6) are higher than the average value, but the values in the rest channels are lower than the average one. Among these flow rates, the maximum value is 4 times of minimum one. The reason of this phenomenon is that the flows in the channels far from volute exit or close to the volute wall are blocked, resulting in decreasing of flow rate. Contrarily, the flow rate and speed are higher in the channels far from volute wall. Therefore, it is imagined that the flow rate and speed vary in all channels with the rotation of the impeller, causing instability of flow in the pump which cause highly noise near to impeller channels 1, and 6.

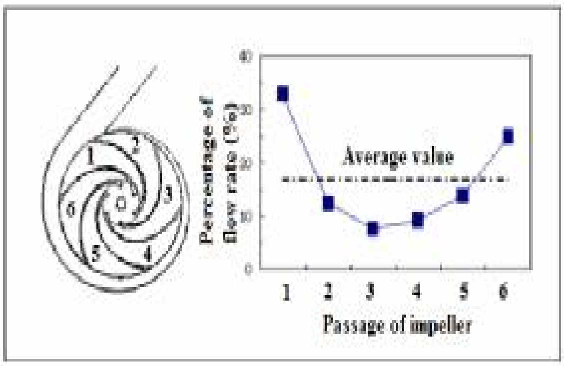

Fig. 21Flow rate distribution in all channels of the impeller [15]

It is seen that the lowest sound pressure level SPL was emitted from point (4) at reference angle φ=270. This is logical result also, because SPL emitted at the pump discharge “at point 4” is due to friction with the pump discharge only without impeller interaction or turbulence at pump exit.

It is showed that as increasing the flow rate, the sound pressure level decreased. The reason is that at lower flow rate there occur many other disturbances like eddies, back flow turbulence, rotator stall and large vortex in impeller. At low flow rate, the blade angle of inlet flow is small and the blade angle of attack is large, thereby increasing the loss at the impeller approach.

At the same time the rotating stall and vortex is intensified, which may make resonance with the pipe system. But as the flow rate increases, the blade angle of inlet flow becomes larger and the blade angle of attack decreases, thereby decreasing the pressure pulsation. Amit [3], Yuang et al. [8] mentioned the same results.
Figures 22, 23, 24, and 25shows a comparison between sound pressure level at original impeller diameter, trimmed impeller down to 108mm, and trimmed impeller down to 104mm referenced to flow rates at the same reference angle.

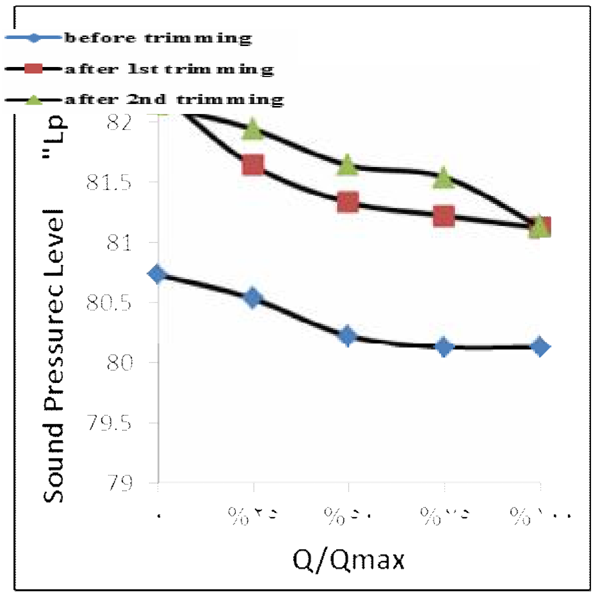

Fig.22 Comparison between sound pressure level at original impeller diameter, 1st, and 2nd trimmed impeller at φ=0.0

At φ=0.0, it was noticed that the sound pressure level increased with the impeller trimming, as shown in figure 22. The increasing in sound pressure level is as a result of increasing the radial gap between the impeller and pump casing which cause increasing the circulated flow and thus increasing the sound pressure level.

At φ=90, it was noticed that the sound pressure level decreased after impeller trimming, as shown in figure 23. The decrease in sound pressure level is as result of increasing the radial gap between the impeller and pump casing which causes more damping in fluctuations from the pumped flow.

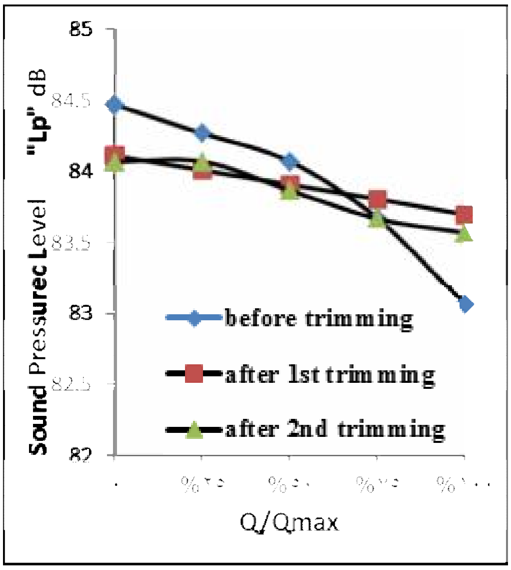

Fig. 23Comparison between sound pressure level at original impeller diameter, 1st, and 2nd trimmed impeller at φ=90

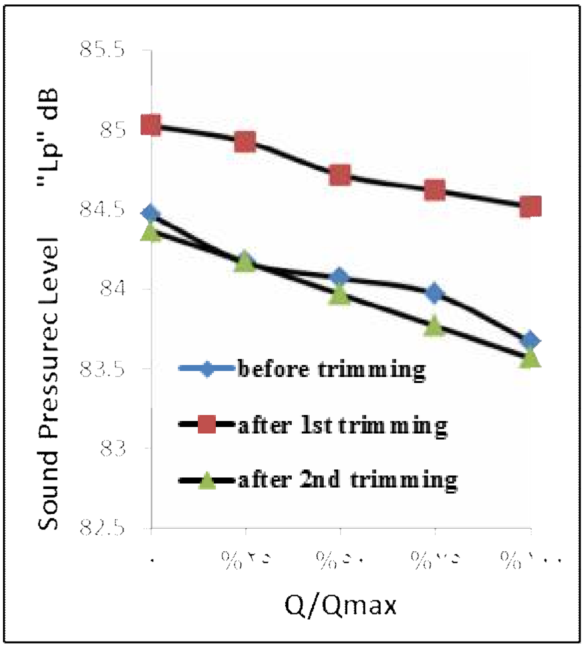

Fig. 24 Comparison between sound pressure level at original impeller diameter, 1st, and 2nd trimmed impeller at φ=180

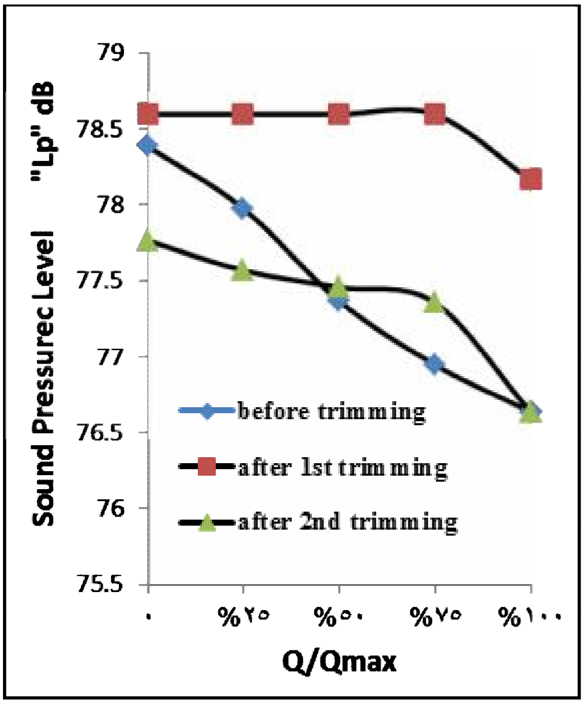

Fig. 25 Comparison between sound pressure level at original impeller diameter, 1st, and 2nd trimmed impeller at φ=270

At φ=180, it was noticed that the sound pressure level increase after impeller trimming to "108mm" impeller diameter and decreased after impeller trimming to "104mm" impeller diameter, as shown in figure 24. After 1st impeller trimming the sound pressure level increased due to more flow interaction and turbulence near to the impeller tongue which cause increasing in pump noise. After 2nd impeller trimming and as increasing the gap between the impeller and the impeller tongue, the flow fluctuation at the impeller tongue decreased which cause decreasing in sound pressure level.

At φ=270, it was noticed that the sound pressure level increased after impeller trimming specially in highly flow rates and decreased after impeller trimming to "104mm" impeller diameter "in low flow rates" as shown in figure 25.After 1st trimming and highly flow rates in 2nd trimming, the sound pressure level increased due to more flow fluctuations at the pump exit because part of the flow will discharged and the rest will circulate with the impeller which cause more eddies and turbulence at the pump exit which cause increasing in sound pressure level. After 2nd impeller trimming in low flow rates, there is no enough flow to make turbulence or fluctuations consequently decreasing the pump noise.

Section (c): Sound spectrum peaks

The centrifugal pump noise expressed in sound pressure level is mainly induced at the blade pass frequency from flow interaction with the impeller blades in addition to secondary flow induced due to flow separation. The flow stall is caused by secondary flows in the impeller due to rotation, the finite number of blades and finite blade thickness, but also by effects of turbulence. Pressure pulsations are detected at discrete frequencies that are multiples of the rotating frequency and the number of blades; these frequencies are also called blade passing frequencies (BPF) where:

$$\textbf{Blade Pass Frequency} = Z \times RPS \qquad (8)$$

The amplitude of these pressure pulsations depends on a number of design parameters of impeller and diffuser and operating parameters.

One of the most important parameters is the distance between impeller and the volute tongue. Smaller distances typically result in much higher amplitude of the BPF. However, this distance also affects the efficiency of the pump.

BPF used in test = $\frac{6\,x2850}{60}$ **= 285 Hz**

The pressure pulsations induced from the flow - blades interaction are detected at the blade pass frequency and its harmonics.
Figure 26 indicate the pressure pulsation at blade pass frequency represented in sound pressure level "dBA" at impeller diameter "D=112mm" near to the impeller tongue "reference angle φ=180".
Figure 27 indicate the pressure pulsation at blade pass frequency represented in sound pressure level "dBA" at impeller diameter "D=104mm" near to the impeller tongue "reference angle φ=180".

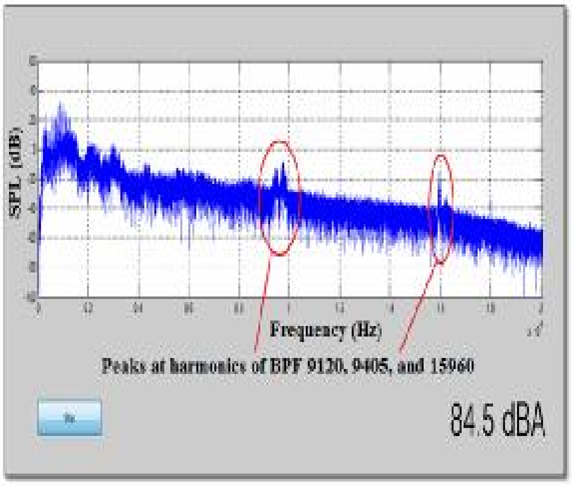

Fig.26Pressure pulsation at BPF and its harmonics at impeller diameter D=112mm

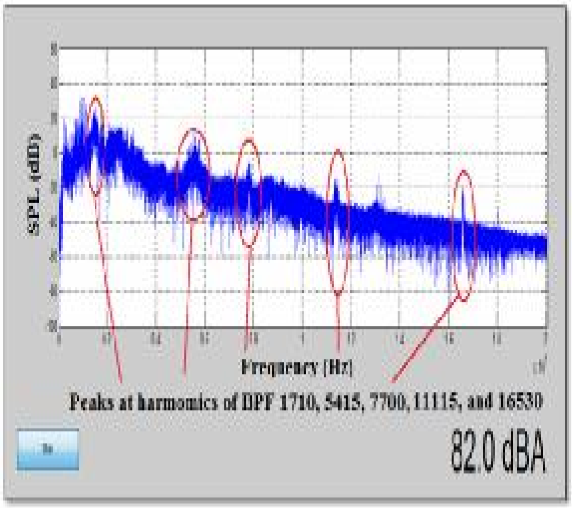

Figure 27 Pressure pulsation at BPF and its harmonics at impeller diameter D=104mm

5. CONCLUSIONS AND RECOMMENDATIONS

On decreasing the impeller diameter, the pump flow, head and power decreases according to affinity laws. The experimental data show a good agreement with affinity laws results.

Generally, the sound pressure level increases with decreasing the pump flow rate by partial to fully closing the discharge valve for the three cases of impeller diameter (112mm, 108mm, and 104mm) respectively.

Generally, the sound puressure level was found high at φ=90 and 180° compared with another locations.

At the reference angle φ=0.0, the reference angle considers the angle between the location of the measured sound pressure level and the beginning of the flow diffusion area; the sound pressure level increases with increased impeller trimming for all flow rates by amount of 1.5dBA.

At the reference angle φ=90, the sound pressure level was found almost constant with impeller trimming. At reference angle φ=180; the sound pressure level increases after the first impeller trimming and decreases after impeller trimming to "104mm" diameter". The variation was found around 0.5 dBA; the amount can be neglected. At φ=270 the sound pressure level increases after the first impeller trimming (specially at high flow rates) and decreases after impeller trimming to "104mm" diameter" at low flow rates. An increase of~ 1.7dBA is recorded with trimming to 108mm in high flow rates and a decrease of about 0.5 dBA is recorded with trimming to 104mm in low flow rates.

Accordingly, it is recommended for some pump users like hospitals and hotels that the noise can affect on customer satisfaction and comfort; when increasing the radial gap between the impeller and the casing that resulted from the trimming by 5%; the generated noise will get high at 180°. So, sound insulator shall be used. If the trimming was by 10%, the generated noise will be lowered.

On increasing the gap between the impeller and pump casing, the pressure pulsation is noticed to increase at blade pass frequency and its harmonics.

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6. NOMENCLATURE

c: Sound propagation speed (m/sec.).
D: Impeller diameter (m).
f: Frequency (Cycle/sec or Hz).
H: Pump head (m).
Is: Sound intensity (Watt/m^2).
n: Pump rotational speed (rpm).
Ps: Pump power (Watt).
Q: Pump flow rate (L/min.).
T: Periodic time (sec).
T_k: Temperature (K).
v: Flow velocity (m/sec.).
V: Motor voltage (volt).
W: Sound power (Watt).
Z: No. of impeller blades.
λ: Wavelength (m).
ρ: Density of air (Kg/m3).
η: Pump efficiency (%).
ηm: Motor efficiency (%).
φ: Reference Angle.