International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 3 Issue 3, March - 2014

Hydraulic Study of Minimizing Energy Consumption in Storm Water Pumping Station Suction Side

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***Abstract* — One of the important factors effecting the efficiency of pumped-storage power stations is the losses induced at the hydraulic waterways. One part of the hydraulic waterways is the intake-inlet structure at the wet basin of the station.**

Through this study, a physical hydraulic model was constructed with typical relative dimensions derived from common design guidelines, to simulate storm water pumping stations adopted in Iraq. Sensitive and accurate measurement devices were used in the model to measure water flow, power and pressure.

This research aims to determine the hydraulic effect of suction pipe size and inlet configuration as provided with bell-mouth end. This effect was investigated through observing the saving in energy consumed by the pump, and pressure reduction detected at the inlet section. Variables considered in the experiments curried out during this study were; suction pipe position (90º elbows), suction pipe diameters (5.08cm, 7.62cm and 10.16cm), and using bell-mouth inlet with different flow velocities.

The results, generally, indicate that there is energy saving when using bell-mouth and when increasing suction pipe diameter. The best installation was through using bell-mouth with (10.16) cm pipe diameter were give best specific energy (power /flow rate) . On the other hand, The Largest energy saving of bell-mouth effect is recorded when using bell-mouth with (5.08)cm pipe diameter .

KEYWORDS: inlet suction pipe, bell-mouth, pump sump, vortices

I. INTRODUCTION

Wastewater lift stations shall be designed to satisfy the hydraulic conditions of the planned facility. The pump head shall be determined as accurately as possible taking into account all majar and minor head losses.

Pump suction pipe design and installation shall not permit the accumulation of air in the suction piping or induce excessive turbulence in the pump suction area. Long radius suction piping bends shall be used whenever possible and eccentric reducers are to be used with flat side up to prevent formation of air pockets.; "**City of Reno, 2007**".

Pumping stations represent one of hydraulic appliances using electric motors, the biggest consumer of electricity in industry. Pumping systems account for nearly 20% of the world's electrical energy demand and range from 25-50% of the energy usage in certain industrial plant operations **"Europump and Hydraulic Institute, 2001; Xenergy, 1998"**.

Wet well and suction pipe intake design of the pump station should be such as to avoid turbulence near the intake and to prevent vortex formation. The most important purpose of this study is to find the energy saving potential in storm pumping systems with focusing up on minimizing local head loss and present suitable hydraulic conditions in suction side. This has been done through choosing suitable inlet pipe diameter, location, shape and using configuration as bell-mouth. It is important to take into consideration in this study the complete pumping system, which consists of pump, pipework, motor and control devices.

The following variables were investigated in the physical model study:
* Different suction pipe diameters ;
* bell-mouth configurations;
* Different suction pipe flow velocities and
* Different suction pipe shape and position.

The experimantal platform (physical model) did not consider some aspects, such as the widening or narrowing of the side wall of pump station, or velocity distributions at the impeller eye. Wet well dimensions of the mpdel was derived from previous researches. Variable speed drive (varial speed pump) instead of the on-off control has been adopted in controlling flow rate through the model. Energy saving has been detected by means of sensitve power meter device connected to the pump.

II. EXPERIMENTAL WORK

This part of study includes comprehensive description for the experimental platform, equipments and devices used in this study. Also, the procedure adopted to carry out the experimental work is described, too.

As mentioned before, this work highlights on wet well suction pipe side in order to determine and evaluate factors that covering flow hydarulics in this side, and their effect on minor head loss, hence power saving potential.

Geometric dimensions of wet well pump basin has bean designed and built on the basis of available researches and design guidelines that ensure a regular flow to the pump without cavitation or palpitations.

- **Modeling experimental platform tank dimensions**

As usual, when facing difficulties in providing hydraulic conditions in a model because of prototype to model scaling ratio, smaller scale ratio should be used to minimize model dimensions, and hence achieving and controlling hydraulic conditions needed, and taking into account the cost factor. On this basis, the physical models are constructed with minimum dimensions that accomplish desired flow conditions, such as turbulent flow and high Weber number with a consideration of geometric scale ratio within 1:4–1:25.;"**Prosser, 1977**". Accordingly, the dimensions used in constructing the physical model were within this range as compared with typical storm water pumping stations, and satisfy the minimum recommended Reynolds and Weber numbers of 4×10^3 and 11, respectively.

The experimental platform tank dimension expressed as follows:

❖ For experimental Platform basin depth, the pump basin depth must be ensured to meet all water level requirements in the test, the minimum water depth in the basin related with inlet velocity then inlet diameter (D). Throguh reviewing available literature for appropriate pipe diameter, largest pipe diameter used in this study was (10.16cm). On the other hand, to avoid vorticies formation, the depth must be more than critical submergence. **Figure (1)** shows critical submergence for this purpose.

From previous research, it has been found that critical submergence $S_c \geq 1.5D$.; " **Prosser, 1977**; **Knauss, 1987**". And also, it has found that critical submergence, $S_c=D(1+2.3Fr)$.; "**Hydraulic Institute, 1998**; **Karassik et al., 2001".**

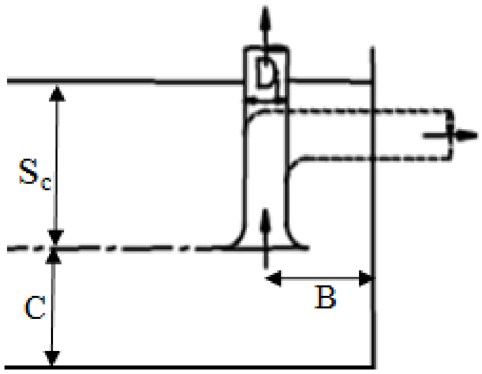


Figure (1). Inlet position (Knauss, 1987)

Some other literature stated that critical submergence is $S_c=1.7Fr$, which means that critical submergence greater than 1.75D.; **"Flygt, 2002**". However submergence of S=2.5D has been used in this study to preventing surface vortices formation or at least to be reduced significantly.

Regarding the distance between the inlet pipe and basin floor (C) as in **Figure (1)**, the referable distance mentioned in the literature is 0.5D. The value of intak open (D) in this study equal 1.9d for bellmouth design, see **Figure (8)**. As the largest pipe deameter (d) used in this study is 10.16cm, the largest inlet open D is18.288cm.

As the submergence used in this study is 2.5D, so the max value of submergence operated in this study that corresponds to largest suction pipe diameter used (10.16cm) is 54.8cm. Adding a free board of (25.2cm) to facilitate handling with water inside the tank and fulfill all the requirements. The total depth adopted for the tank was 100 cm. This is to provide all requirements needed, like installing an over flow discharge pipe, and water circulation pipe.

❖ For experimental Platform basin width, the minimum requirements for pump basin width is 2D for one pump "**Prosser, 1977**". There are other width requirements related to the amount of storage required. In addition, the model requires the provision of space to install baffles and to eliminate the hydraulic (interfere) effect of the adjacent walls of pump station. Basin width has been duplicated around three times to be as 1.2m. This width provide also suitable space to handle the fittings inside model tank.

❖ For experimental Platform basin length, the distance from centerline of pump inlet bell/volute to screen ≥4D, "**Federal Highway Administration, 2001; American National Standard ANSI/HI 9.8-1998; Karassik, 2008**". Accordingly,the experimental Platform basin length needed is (4D+B) that's equal to (86.8cm). Adding 13cm for tranch width, leading to experimental Platform basin length as (99.9cm). The length adopted in this study was $L_{(model)}$ =100cm

- **Pump capacity:**

Flow velocity in pipeline has to be kept less than or equl to 4 m/s. "**Sulzer Pump Ltd, 2010**". Pump capacity used in the physical model satisfys this condition, and provide turbulent discharge in all used suction pipes, as well as stability of the measurement devices used. To investigate effect of different suction pipe flow velocities of the model (up to 4 m/s), with smallest diameter used in the tests (0.0508m), the following steps were adopted to determine pump power :

The power of the driver can be calculated from the well known equation:

$$P_d = \frac{\rho\, g\, Q\, H}{\eta_p \times \eta_m}$$; "**Europump and Hydraulic Institute, 2001**".

The flow rate Q is:-

Q=A*v= π(0.0508)2/4*4=0.008107 m3/s.

The total head H can be claculated using the equation:-

$$H = H_{geod} + \left(f\frac{L}{D} + \sum \xi\right)\frac{v^2}{2g}$$.;"**Groundfos westwater,2002** "

For a pipe diameter of (0.0508m) carrying the discharge (0.008107 m^3/s) above, Reynolds number is 203200. Using Moody diagram, friction factor f= 0.027. local resistance (ξ) is 0.3 for 90ºelbow of 2 inch PVC pipe, with six 90º elbows, with flow velocity v_m = 4m/s , H_{geod}=2.4m, and 10m long PVC discharge pipe (with the same diameter of 0.0508m), the total head H is:-

$$H = 2.4 + \left(0.027\times\frac{10}{0.0508} + (6\times0.3)\right)\frac{4^2}{2\times9.81} = 8.202 \text{ m}$$

Using pump effeciency as 0.75 and pump motor effeciency as 0.8, the pump actual (overall) power is ,

$$P_d = \frac{9810\times0.0081\times8.202}{0.8\times0.75} = 1086.23 \text{ Watt}$$

Using variable speed driver efficiency as 0.97, so,
Pd =1086.23 /0.97 = 1119.82 Watt
Pd = 1119.82 /746 = 1.5 hp

The driver should not operate continuously at its maximum capacity, but at an 85 to 90% load so:
Pd= 1.5/0.85= 1.77 hp

According to commercial availablity, a centrifugal pump with three phase of 2.2 hp , (1.65 KW) power was used in the model.

- **Description of experimental platform (model)**

Experimental platform has been designed and built in such a way to allow testing all variables and experiments needed in this study **Figures (2), (3), (4), (5),(6),** and **(7)** . The model has been built in the Labs of Faculty of Engineering in Kufa University.

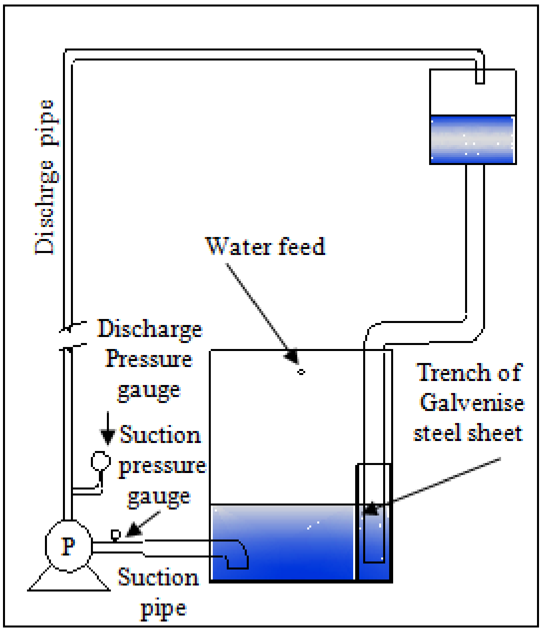


Figure (2). schematic diagram of the experimental model of the study.

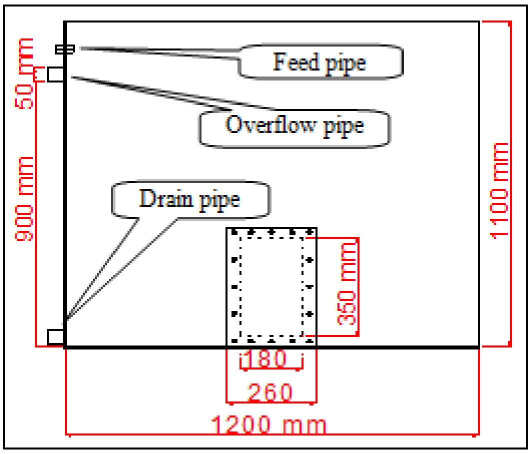
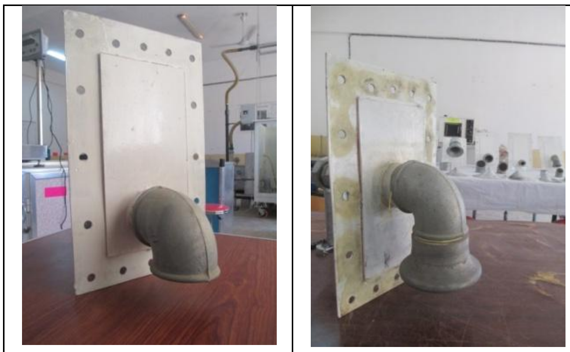


Figure (3). Front view details for Experimental model of the study

Figure (4). Side view for the experimental platform of the study

	
d=0.0508m.	d=0.0508m, with bell-mouth

Figure (5). The manufactured flange and the adopted configurations for 5.08cm diameter suction pipe.



d=0.0762m	d=0.762m, with bell-mouth

Figure (6).The manufactured flange and the adopted configurations for 7.62cm diameter suction pipe.

d=0.1016 m	d=0.1016m, with bell-mouth

Figure (7). The manufactured flange and the adopted configurations for 10.16cm diameter suction pipe.

- **Pressure gauges**

The model was provided with a mechanical pressure gauge on discharge pipe and digital pressure gauge on suction. This digital gauge is operatedly 9V-DC battery power , with zero offset, and accuracy of +/-0.5% F.S (full scale), LCD digital display with 4 digits with response time less than 1ms, and total error (typical) of +/-1.00%.

- **Bell-mouth dimension**

To facilitate the flow smoothly into the inlet pipe and reduce (minimize) inlet losses and disturbed flow into the pump, the inlet should be provided with a bell-mouth. Most pump manufacturers use the following ratio of bell-mouth diameter (D) to pump suction opening (d), as D/d = 1.5 to 1.8. In general, most design guidelines (Hydraulic Institute Standards,1975; Prosser, 1977; Knauss, 1987) suggest that the pump bell diameter be between 1.5 to 1.8 times the diameter of the discharge pipe.; "**Frizell K. W., 1994**". Other design guidelines state that bell diameter approximately 1.5–2.0 times the inside pip diameter., "**American National Standard for Pump Intake Design(ANSI/HI) ,1998; Karassik, 2008**".

Bell-mouths designed and fabricated to be fit with inlet pipe sizes, hence, it designed with inlet open diameter equal to 1.9 times the inside diameter of suction pipe. It is designed and casted as in **Figure (8)** and **Figure(9)**. **Figure(10)** shows three sizes of bell-mouth at same scale ratio with pipe diameter used in the test. **Table (2)** Illustrates scaled dimensions of bell-mouths for each pipe diameter used in the test .

Table (1).Bell-mouth dimensions adopted

d mm	D (mm)	H	R
	D= d ×1.9	H=d	D×0.267
50.8	96.52	50.8	25.77
76.2	144.78	76.2	38.65
101.6	193.04	101.6	51.54
Where, d: suction pipe diameter, H: Bell-mouth hight , R: Bell-moth sid wall carve , and D: Bell-moth inlet open diameter .			

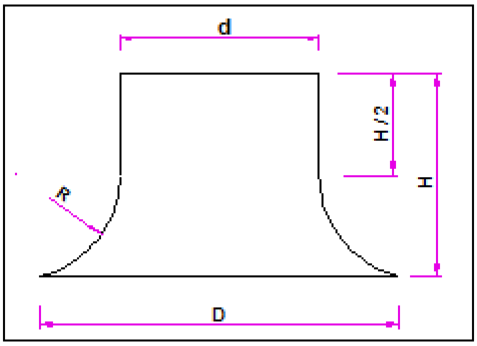


Figure (8). Bell-mouth details adopted in the study

Figure (9).Bell-mouth fabrication through the study

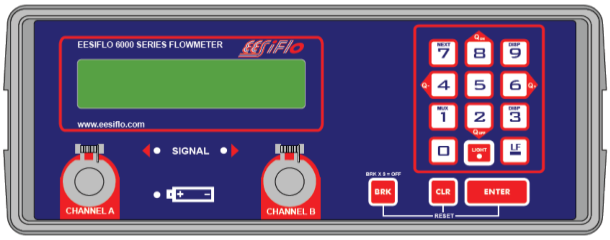
Figure (10).Three sizes of bell-mouth at same scale ratio with pipe diameter

- **Flow meter**

Digital flowmeter type (**EESIFLO 6000**) was used in the experimental work. It was ultrasonic signals to measure the flow in pipes or conduits, **Figure(11)** .

With a special probe, can also measure the thickness of pipe walls.

The transducers of the flowmeter can be used at temperatures between -30°C and 130°C. Measurement can be made on all commonly used pipe materials such as steel, synthetic material, glass or copper. Pipe diameters may range from 10 up to 6500 millimeters depending on the transducer type. The two clamp-on transducers allow for non-invasive measurement that do not affect the pipework or the liquid to be measured.



Figure(11).front view of digital flow meter

- **Variable speed driver system**

Variabe speed driver system (VSDs) type **(SV-iG5A)** frequency inverter was used in the experimantal wark. It is utilizing three phase with 2hp power **figure (12)**. Its suitable as variable flow driver as it is needed in this study.

Figure (12).Variable speed driver(VSD)

- **Balancing storage tank**

Additional water storage tank with storage cappacity of $1m^3$ was used to feed testing platform and to controll water temperature prior to be supplied into the model. To transport water to and from the Balancing storage tank, two pumps have been used. One of them was to feed the model and the other was to transfer water from the model to balancing storage tank .

- **Constants used in the tests**

There were some constant dimensions adopted in the tests, those constants were considered from previous researches as explained in literature review, the constants are shown in **Table (2).**

Table(2). Values of constants considered in this study

Sample	Description	Dimension
C	Distance between the inlet of suction pipe level and floor level	0.5 D
B	Distance from the back face wall to the inlet centerline of suction pipe.	0.75D
H_{geod}	geodetic head	2.5 m
S	Submergence	2.5 D

Where, D is inlet diameter of suction pipe

- **Description of tested scenarios**

Throughout the experimental work, different test runs were accomplished. Suction head wase measured for different cases with different variables. In each case, total power consumption of the overal system (of the pumpe) was measured by mean of power meter, while suction head measured using the digital gauge, and flow were measured by flow meter. For all cases, measures were carried out after 10 minutes of conducted operation to reach stability conditions and ,hence, give trusted observations. Temperature wase maintained within (25.3ºC-25.9ºC) through all tests, tests senioresas are present in **Table (3)**.

Pumped water power wase calculated using the common equation:

$$Power = \gamma\ Q\ h$$

But, power saving due to using of inlet configration wase calculated as:

$$powerSaving = \gamma Q \Delta h / \gamma Q h_1$$

where:

γ : Water Density (Kg m/s)

Q: Flow-rate (m^3/s)

Δh: difference in head (m) = h_1-h_2

h_1= Suction head in meters without using of bellmouth, (m), **h_1=p_{s1}*0.703**

h_2= Suction head in meters with use bellmouth, (m), **h_2=p_{s2}*0.703**

Table (3). Seniores of variables tested

No.	Suction pipe diameter (cm	Water Velocity m/s	With bell-mouth or no
1		0.15	yes
2		0.15	no
3		0.233	yes
4		0.233	no
5	5.08	0.367	yes
6		0.367	no
7		0.4	yes
8		0.4	no
9		1	yes
10		1	no

No.	Suction pipe diameter (cm	Water Velocity m/s	With bell-mouth or no
11		2	yes
12		2	no
13		3	yes
14		3	no
12	7.62	Steps (1-10) to be repeated with change suction pipe diameter to 7.62 cm	
13	10.16	Steps (1-8) to be repeated with change suction pipe diameter to 10.16 cm	

III. ANALYSIS AND DISCUSION OF THE RESULTS

Results of scenarios tested using the implemented physical model and their representing graphs are presented in this section. The results of different cases were analysed to investigate the effects of different variables that governing the present problem.

The energy consumption under effect of many configurations that consists of three types of pipe diameter and bell mouth, were studied with the variation of Reynolds number. Then, comprehensively, the effects of the main controlling parameters that maximize the hydraulic power saving were displayed.

Bell-mouth effect through the percentage of power saving with variation of Reynolds number according to three different pipe diameters of 0.0508,0.076 and 0.1016 m are shown in **Figures** (**13**),(**14**) and (**15**), respetively. The bellmouth effect for 0.0508,0.076 and 0.1016 m inlet pipe diamteres with 90º elbow, all togeather, is presented in **Figure** (**16**). From **Figure (13)**,the percentage of power saving for inlet suction pipe of 0.0508 m diameter has highest values of approximately 25% when Reynolds number is greater than 170000.

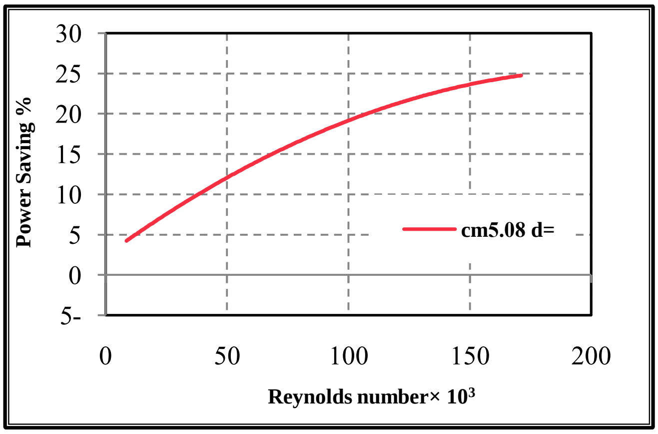


Figure (13). Percentage of power saving of bell-mouth effect as a function of Reynolds number for (5.08cm) inlet suction pipe diameter.

From **Figure (14)**, the percentage of power saving for inlet pipe of 0.0762 m diameter, gives the highest values of approximately 26% when Reynolds number is around 40000, and be decrease with increasing Reynolds number more.

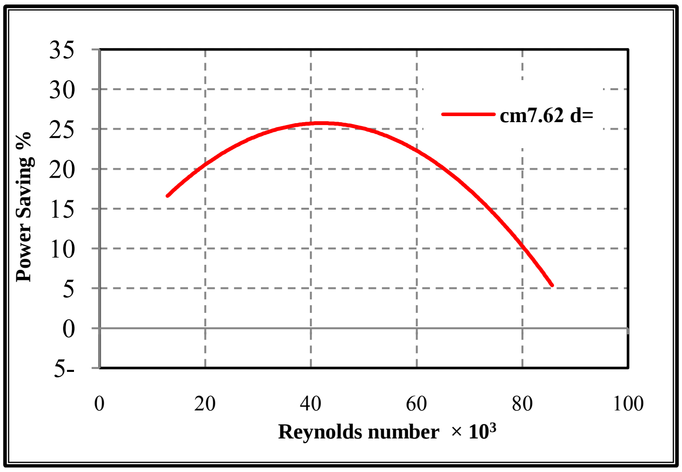


Figure (14). Percentage of power saving of bell-mouth effect as a function of Reynolds number for (7.62cm) inlet suction pipe diameter.

Figure (15) reveal percentage of power saving for inlet bell mouth with 0.1016m pipe diameter.In this pairing, it is found that the highest value of percentage of power saving which is approximately 40% occures when Reynolds number is about 17000, It is observed that power saving decreases with increasing Reynolds number till Re=36000 were the saving is rest on about 2.5%.

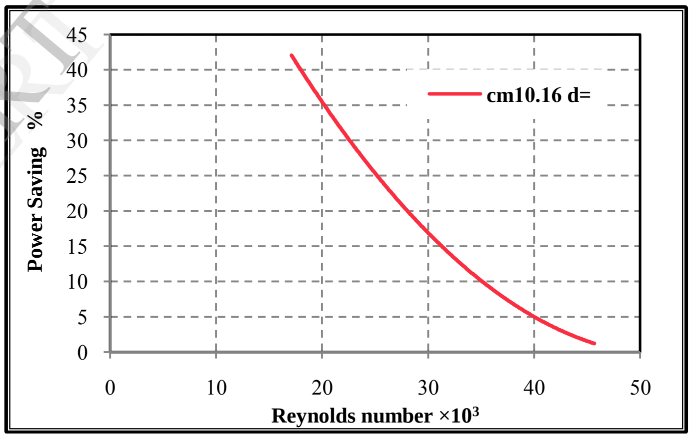


Figure (15). Percentage of power saving of bellmouth effect as a function of Reynolds number for (10.16cm) inlet suction pipe diameter.

Figure (16) shows the comparison between percentage of power saving with bell-mouth in three adopted pipe diameters and 90º elbow. The percentage of power saving of 0.0508m diameter inlet pipe is approximately 13% when Reynolds number is about 20000, and percentage of power saving will increase to the highest value of about 25% with the increasing of Reynolds number to 160000. On the other hand, when Reynolds number is 40000 and more there percentage of power saving decreases in 0.0762m and 0.1016m pipe diameters is generally decreas with increas of reynolds number.

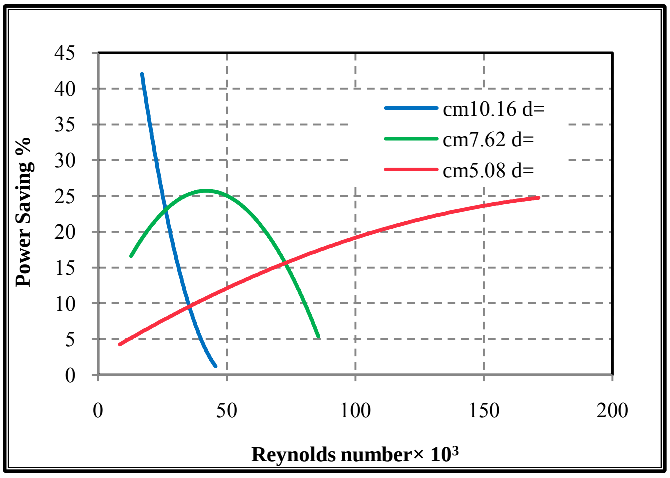


Figure (16). Percentage of power saving as a function of Reynolds number for different suction inlet pipe diameter with bell-mouth.

IV. CONCLUSIONS

The following main conclusions were obtained:

1- Increasing the suction pipe diameter lead to decrease specific energy of flow (power per discharge).

2- The use of bell-mouth has a clear impact in saving energy for all pipe diameters used. For the same Reynolds number tested, highest effect of bell- mouth appeared in the smaller pipe diameters (5.08cm). This explain that bell-mouth is more effective to save power for relatively high flow velocities.

Although installing bell-mouth at the suction pipe inlet minimizes the head loss, hence, the power consumed, but this effect becomes very clear for relatively high Reynolds number when considered on the basis of the specific power consumed.

V. LIST OF SYMBOLS

Symbol	Definition	Units
Δh	difference in head	m
η_m	Motor efficiency	-
η_p	pump efficiency	-
d	Inside pipe diameter	m
D	Inlet suction pipe diameter	m
f	Friction factor	-
g	Acceleration of gravity	m/s^2
H	Total head	m
h_1	Suction head in meters without using of configuration	m
h_2	Suction head in meters with use configuration	m
H_d	Pump dynamic head	m
H_f	Pipe friction losses	m
H_{geod}	pump geodetic head	m
H_J	pipe losses in the system	m
H_L	Local loss	m
H_{st}	Static head	m
L	pipe length	m
p_d	Required power supply	W
P_{s1}	Suction pressure without using any configuration	psi
P_{s2}	Suction pressure with using certain configuration	psi
Q	Flow rate	m^3/s
Sc	Critical submergence	m
v	Fluid average velocity	m/s
ν	kinematic viscosity.	m^2/s
ρ	Density of the liquid	kg/m^3
ξ	Local resistance coefficient	-

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