Design of Multistage Centrifugal Pump Impeller for High Head Applications

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***Abstract*—This paper presents the design of multistage centrifugal pump which can be used in mountain-water distribution system. Type of pump is seven stage centrifugal pump with closed impeller and it can develop a head of 140 m and deliver 0.025 m^3/s of water. The designed impeller has 88 mm inlet diameter, 146 mm outlet diameter, 18° inlet vane angle and 23° outlet vane angle. The number of vanes is 9 and input shaft power is 48.35 kW. The inlet width and outlet width are 29 mm and 23 mm respectively. The discharge diameter is 80 mm to operate the designed head and capacity. The predicted maximum efficiency is nearly 65% and the expressed actual efficiency of designed pump is 63%. Therefore, the designed efficiency has a satisfactory value. The designed multistage centrifugal pump can fulfill the requirements of water distribution system for high mountain or far from the lake, and domestic usage in multistory building.**

***Keywords*—multi stage centrifugal pump, head, flow rate, speed, specific speed.**

I. INTRODUCTION

A multistage pump consists of two or more identical impellers mounted on the same shaft and enclosed in the same casing. All the impellers are connected in series so that liquid discharged with increased pressure from one impeller passes through the connecting passages to the inlet of the next impeller and so on, till the discharge from the last impeller passes into the delivery pipe. The impellers are surrounded by guide vanes which are generally provided within the connecting passages and are meant for the recuperation of the velocity energy of the liquid leaving the impeller into pressure energy. According to the number of impellers fitted in the casing a multistage pump is designated as two stage, three-stage, etc. Normally a pump with a single impeller can be used to deliver the required discharge against a maximum head of about 100 m. But if the liquid is required to deliver the required discharge against a still larger head then it can be done by using two or more pumps in series. The pumps in series may be used in boiler feeding system to satisfy high head demand, in multi-stations systems along pipelines such as long oil pipelines for boosting the liquid like pumping installations interposed in the water mains, etc. The higher heads may also be produced by using multistage pumps.

II. BASIC OPERATING PRINCIPLE OF CENTRIFUGAL PUMP

The two main components of centrifugal pump are impeller and casing. The centrifugal pump moves liquid by rotating one or more impellers inside a volute casing. The liquid is introduced through the casing inlet to the eye of the impeller where it is picked up by the impeller vanes. The rotation of the impeller at high speeds creates the centrifugal force that throws the liquid along the vanes, causing it to be discharged from its outside diameter at a higher velocity. This velocity energy is converted to pressure energy by the volute casing prior to discharging the liquid to the system. The kind of loss of centrifugal pumps can be differentiated in internal losses and external or mechanical losses, variations of the effective area or changes of direction losses of quantity at the seal places between the impeller and housing at the rotary shaft seals. The external or mechanical loss is sliding surface losses by bearing friction or seal friction.

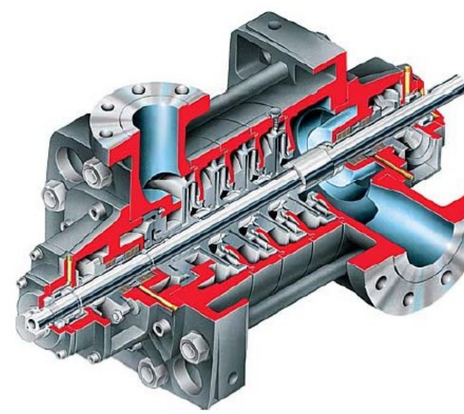

Fig.1 Multistage Centrifugal Pump

III. DESIGN OF PUMP'S IMPELLER

When the overall design of pump is considered, the shape of an impeller is the most important for optimum efficiency. Impeller design should be in such a way that, losses must be as low as possible. The design of a pump's impeller can be divided into two parts. The first is the selection of proper velocities and vane angles needed to obtain the desired performance with the best possible efficiency. The second is the layout of the impeller for the selected angles and areas.

The specifications of pump that will be designed are:

Pump head, $H = 140$ m

Discharge, $Q = 1.5\ m^3/min$

$$Q_s = (Q/60)\ \text{m}^3/\text{s} = 0.025\text{m}^3/\text{s}$$

Rotational Speed, $n = 2970$ rpm

Density of water, $\rho = 1000$ kg/m^3

A. Selection of Specific Speed

Specific speed is defined as the speed in revolutions per minute at which an impeller would operate if reduced proportionately in size so as to deliver one unit of capacity against one unit of total head. It is used to classify the type of impellers on their performance, and proportion regardless of their actual size or the speed at which they operate. It is mathematically expressed as

$$n_s = \frac{n \times \sqrt{Q}}{H^{3/4}} \qquad (1)$$

In this design, calculated value of specific speed based on required head and capacity is 385 rpm and it is within the range of low specific speed pump that is greater than 80 and less than 600.

B. Suction and Discharge Pipes Diameter

Pump efficiency, η is assumed by using Fig. 2. And also the diameter of suction pipe D_s can be estimated from this chart. The discharge pipe diameter D_d is usually selected equal to or one size smaller than that of the suction pipe. Thus, velocities in these pipes are given by

$$V_s = \frac{Q_s}{\pi \frac{D_s^2}{4}} \qquad (2)$$

$$V_d = \frac{Q_s}{\pi \frac{D_d^2}{4}} \qquad (3)$$

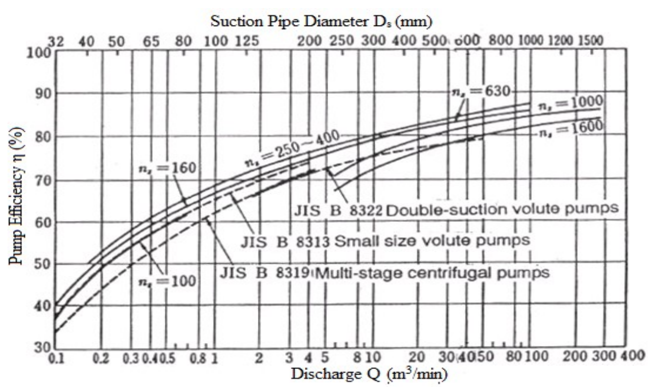

Fig. 2 Overall Efficiency Curve [2]

C. Determination of Input Power

Input power of centrifugal pump can be determined by following equation.

$$L = \frac{\rho Q_s g H}{\eta} \qquad (4)$$

For charge condition of the pump work, maximum shaft power or rated output of an electric motor L_r (kW) is decided by using Equation (5).

$$L_r = \frac{(1+F_a)\times L}{\eta_{tr}\times 1000} \qquad (5)$$

Where, F_a is the allowance factor, and 0.1~ 0.4 for an electric motor and larger than 0.2 for engines, and it is shown in Table I. And then, η_{tr} is the transmission efficiency, and 1.0 for direct coupling and 0.9 ~ 0.95 for belt drive.

TABLE I

RATED OUTPUT OF ELECTRIC MOTOR [2]

Rated output (kW)	0.4	0.75	1.5	2.2	3.7	5.5	7.5	11	15	18.25	22	30	37
Allowance factor F_a	0.4			0.4~0.25		0.25~0.15							

D. Determination of Shaft and Hub Diameters

The shaft diameter at hub section of impeller is

$$d_s = \sqrt[3]{\frac{16\,T}{\pi\,\tau}} \qquad (6)$$

Where, T is the torsional moment and it can be estimated by

$$T = \frac{60\,L_r}{2\,\pi\,n} \qquad (7)$$

Allowable shear stress of material of shaft, τ is 24.5 MPa because the main shaft is made of S30C. The estimated shaft diameter will be increased because it is difficult to predict the bending moment at this time.

The hub diameter, D_h is usually taken from 1.5 to 2.0 times of the shaft diameter and the hub length, L_h is from 1.0 times to 2.0 times of the shaft diameter.

E. Determination of Impeller Eye Diameter

The diameter of impeller eye, D_o is calculated by

$$D_o = \sqrt{\frac{4Q_s^{'}}{\pi V_{mo}} + D_h^2} \qquad (8)$$

Where, the flow rate through the impeller, $Q_s^{'}$ is Q/η_v and volumetric efficiency η_v is estimated by

$$\eta_v = \frac{1}{1+\frac{1.124}{n_s^{2/3}}} \tag{9}$$

For Equation (8), the velocity at the eye section is given by

$$V_{mo} = K_{mo}\sqrt{2gH} = (1.5\sim3.0) \leq V_{m1} \tag{10}$$

$$K_{mo} = (0.07\sim0.11) + 0.00023 n_s \tag{11}$$

F. Impeller Inlet and Outlet Diameters and Velocities

The simplified inlet and outlet velocities diagrams for the impeller are shown in Fig. 3. In this figure, the effect of circulatory flow on the outlet diagram is shown in solid lines and the virtual diagram is dotted. For a fluid flowing through the rotating impeller, u is the tangential velocity, V is the absolute velocity and v is the relative velocity of a fluid particle to impeller rotation. The angle between V and u is α and the angle between v and u is β and it is the angle made by tangent to the impeller vane and a line in the direction of motion of the vane. The tangential component and radial component of absolute velocity V are V_u and V_r respectively.

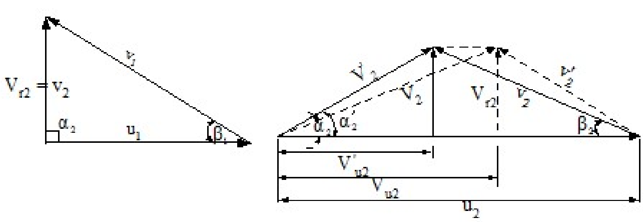

Fig. 3 Impeller Inlet and Outlet Velocity Diagrams

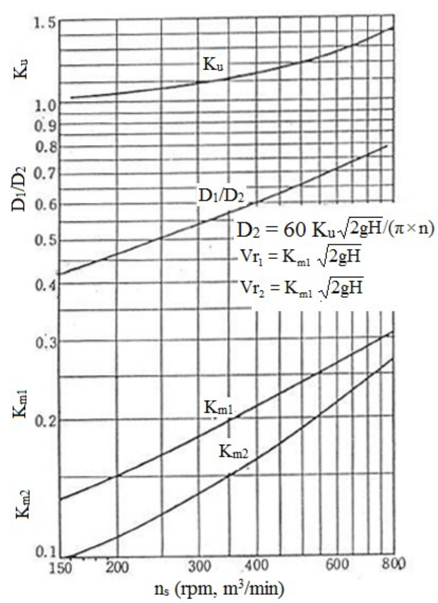

Fig. 4 Stepanoff Chart

The Stepanoff chart shown in Fig. 4 is widely used to decide the impeller geometry. The parameters K_u (speed constant), K_{m1}, K_{m2}, and D_1/D_2 are obtained on the value of specific speed in this figure. The outlet diameter D_2 is decided by considering the following relationship.

$$D_2 = \frac{u_2 \times 60}{\pi \times n} \tag{12}$$

Where, the peripheral velocity at impeller outlet is

$$u_2 = K_u\sqrt{2gH} \tag{13}$$

The peripheral velocity at the inlet is also expressed by

$$u_1 = \frac{\pi D_1 n}{60} \tag{14}$$

And then, flow velocities at the inlet and outlet are

$$V_{r1} = K_{m1}\sqrt{2gH} \tag{15}$$

$$V_{r2} = K_{m2}\sqrt{2gH} \tag{16}$$

If the incoming flow has no pre-rotation, the blade angle β_1 (deg) is given by

$$\beta_1 = \tan^{-1}\left[\frac{K_{b1}V_{r1}}{u_1}\right] \approx \tan^{-1}\left[\frac{V_{r1}}{u_1}\right] + (0\sim6) \tag{17}$$

Where, K_{b1} = 1.1~1.25

The vane outlet angle β_2 is usually made larger than the inlet angle β_1 to obtain a smooth, continuous passage. The amount of outlet angle β_2 usually has between 15˚ and 40˚. So, the vane outlet angle is assumed that β_2 = 23˚ in this study. From the velocity triangles, inlet and outlet relative velocities are

$$v_1 = \frac{u_1}{\cos\beta_1} \tag{18}$$

$$v_2 = \frac{V_{r2}}{\sin\beta_2} \tag{19}$$

The virtual tangential component V_{u2} of V_2 is

$$V_{u2} = u_2 - \frac{V_{r2}}{\tan\beta_2} \tag{20}$$

For radial-type impellers, the slip factor, η_∞ varies between 0.65 and 0.75and it is assumed that η_∞ = 0.7 average. Thus, the actual tangential component V'_{u2} of V'_2 is

$$V'_{u2} = V_{u2}\eta_\infty \tag{21}$$

Thus, the actual outlet is found by

$$\tan\alpha'_2 = \frac{V_{r2}}{V'_{u2}} \tag{22}$$

The absolute outlet velocity from outlet velocity diagram is

$$V_2' = \sqrt{V_{r2}^2 + V_{u2}'^2} \tag{23}$$

G. Number of Vanes and Impeller Widths

The number of blades, Z is decided by using the Plfeiderer formula. It is

$$Z \approx 6.5 \frac{D_2 + D_1}{D_2 - D_1} \sin[\frac{\beta_1 + \beta_2}{2}] \tag{24}$$

When the impeller is made of bronze (e.g B(6)), the minimum blades thickness is 2.0 mm and shroud thickness is 2.5 mm for an impeller having the diameter less than 200 mm. They are 2.5 and 3.0 mm respectively, if D_2 is greater than 200 mm. In this design, blade thickness and shroud thickness are taken as 2 mm and 2.5 mm respectively. The inlet passage width b_1 and outlet passage width b_2 are calculated by

$$b_1 = [\frac{Q_s'}{\pi D_1 V_{r1}}][\frac{\pi D_1}{\pi D_1 - S_1 Z}] \tag{25}$$

$$b_2 = [\frac{Q_s'}{\pi D_2 V_{r2}}][\frac{\pi D_2}{\pi D_2 - S_2 Z}] \tag{26}$$

Where, S_1 is (δ_1/sin β_1), S_2 is (δ_2/sin β_2), and δ_1 and δ_2 are blade thicknesses near the leading edge and trailing edge respectively. Moreover, S_2 can also be determined by the following relationship equation.

$$\frac{\pi D_1}{(\pi D_1 - S_1 Z)} = \frac{\pi D_2}{(\pi D_2 - S_2 Z)} \tag{27}$$

H. Required Parameters of Impeller Blade Shape

A method of drawing the impeller blade by three circular arcs is used for this present design. Each radius is given by

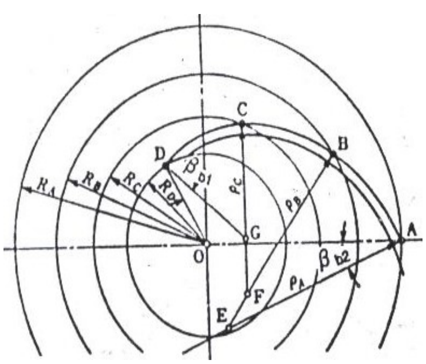

Fig. 5 Curvature of Impeller Blade

$$\rho_A = \frac{\left(R_A^2 - R_B^2\right)}{2\left(R_A \cos\beta_2 - R_B \cos\beta_B\right)} \tag{28}$$

$$\rho_B = \frac{\left(R_B^2 - R_C^2\right)}{2\left(R_B \cos\beta_B - R_C \cos\beta_C\right)} \tag{29}$$

$$\rho_C = \frac{\left(R_C^2 - R_D^2\right)}{2\left(R_C \cos\beta_C - R_D \cos\beta_1\right)} \tag{30}$$

Where, R_A, R_B, R_C and R_D are base circle radii, R_A=D_2/2 and $R_D = D_{1h}/2$.

$$R_B = R_A - \frac{R_A - R_D}{3} \tag{31}$$

$$R_C = R_B - \frac{R_A - R_D}{3} \tag{32}$$

The angles between β_1 and β_2 are divided into three angles.

IV. CALCULATED RESULTS OF CENTRIFUGAL PUMP DESIGN

The calculated results for impeller design of centrifugal pump are clearly expressed in Table II.

TABLE II

CALCULATED RESULTS OF MULTISTAGE CENTRIFUGAL IMPELLER DESIGN

No	Descriptions	Symbols	Results
1	Input Power	L	48.35 kW
2	Shaft diameter	d_s	34 mm
3	Hub diameter	D_h	51 mm
4	Hub length	L_h	68 mm
5	Impeller eye diameter	D_o	117 mm
6	Impeller inlet diameter	D_1	88 mm
7	Impeller outlet diameter	D_2	146 mm
8	Inlet angle of impeller blade	β_1	18°
9	Outlet angle of impeller blade	β_2	23°
10	Impeller passage width at inlet	b_1	29 mm
11	Impeller passage width at outlet	b_2	23mm
12	Number of impeller blades	Z	9 blades
13	Number of stage		7

Moreover, the designed impeller is drawn by AUTOCAD software.

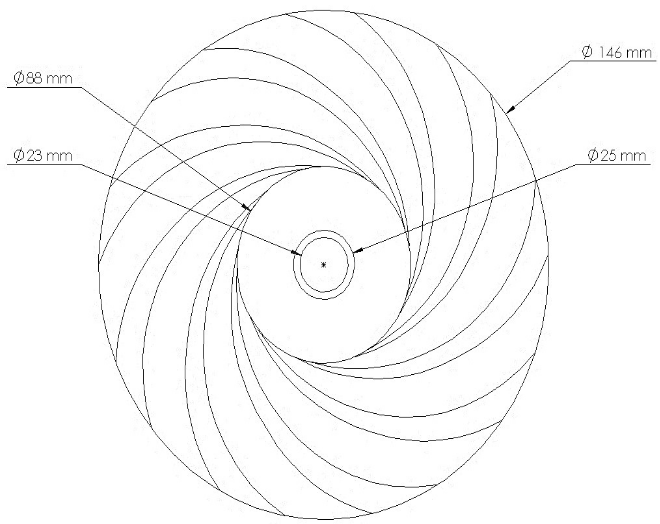

Fig.6 Detail Drawing of Designed Impeller

Before producing the designed pump, it needs to predict to the performance characteristics. Without forecasting the performance across the operating range, the loop of design cannot be closed. By seeking performance curve, operating range is determined. Performance needs to be known, not only at the rated, best efficiency point, but also off design. Pump specifications often impose special requirements, such as head at shut-off, maximum power demand, rate of head rise to assure stability, and so on. A good pump design process requires trail-and-error iteration, a check on anticipated performance with a trail geometry, and progressive approximation to the optimal design configuration [5].

V. Prediction of Performance Characteristics

A. Theoretical Head

Firstly, theoretical head is calculated by

$$H_{th} = \frac{1}{g}(u_2 V_{u2}) \tag{33}$$

Inlet and outlet areas are not available to the fluid, but these areas will be reduced by the vane thickness. Hence, a contraction factors, ε_1 and ε_2 will be assumed to compensate for the vane thickness at the inlet width, b_1 and outlet width, b_2 in determining the gross inlet area and outlet area. The factor, ε_1 is generally between 0.8 and 0.9 and the factor ε_2 is usually between 0.9 and 0.95. So, the inlet and out contraction factors, ε_1 and ε_2 are assumed 0.85 and 0.925 respectively in this study [4]. Where,

Whirl velocity, $V_{u2} = u_2 - V_{r2} \cot \beta_2$

Flow velocity, $V_{r2} = \dfrac{Q_s}{\pi D_2 b_2 \varepsilon_2}$

B. Net Theoretical Head

The circulatory flow effect reduces the theoretical head developed in a practically constant ratio. Slip value for circulatory effect is

$$\sigma = 1 - \frac{(\sin \beta_2)^{1/2}}{Z^{0.70}} \tag{34}$$

By considering this effect, the whirl velocity and the net theoretical head are

$$V_{u2} = u_2\, \sigma - V_{r2} \cot \beta_2 \tag{35}$$

$$H_{thn} = \frac{1}{g}(u_2 V_{u2}) \tag{36}$$

C. Shock Losses

Shock losses are considered as following expressing.

$$h_s = k\,(Q_s - Q_N)^2 \tag{37}$$

In the shut-off condition, $Q_s = 0$ and Q_N is design flow rate at maximum efficiency. Where, shut-off head is estimated by

$$H_{shut\text{-}off} = \frac{u_2^2 - u_1^2}{2g} \tag{38}$$

From the shut-off condition, the value of k can be calculated.

D. Friction Losses

The wall friction or skin friction losses, H_{f1} in the impeller follow the standard pipe friction model. Since the flow passage cross sections are irregular, a hydraulic radius and average flow velocities are used. The friction coefficient can be adjusted but has a default value of 0.075. The impeller friction losses are predicted by the following Equation (33).

$$H_{f1} = \frac{CF\,(D_2 - D_1)\,(v_2 + v_1)^2}{2\times(\sin \beta_2)\,H_{r1} \times 4\,g} \tag{39}$$

Where, H_{r1} refers to hydraulic radius and it is expressed by

$$H_{r1} = \frac{\pi\, b_2\, D_2 \sin \beta_2}{Z\,(b_2 + \dfrac{\pi D_2 \sin \beta_2}{Z})} \tag{40}$$

The volute friction losses can be found by

$$H_{f2} = \frac{\mathrm{CF}\,\pi\, D_3\, V_3^2}{2\,g\sqrt{\dfrac{A_{th}}{\pi}}} \tag{41}$$

Where V_3 is the volute throat velocity and A_{th} is the volute throat area. Their relationship is as follow [6].

$$V_3 = \frac{Q_s}{A_{th}} \tag{42}$$

E. Diffusion Losses

A diffusion loss H_{df} needs to be taken into account, since separation invariably appears in the impeller at some point. When the ratio of the relative velocity at the inlet v_1 and outlet v_2 exceeds a value of 1.4, it is assumed that a portion of the velocity head difference is lost. The diffusion loss is

$$H_{df} = \frac{0.25\, v_1^2}{2g} \tag{43}$$

F. Actual Head

Finally, the actual pump head is calculated by subtracting all the flow losses from the net theoretical head. Thus, the actual pump head is forecasted by the following relationship equation.

$$H = H_{thn} - (h_s + H_{f1} + H_{f2} + H_{df}) \tag{44}$$

G. Efficiency and Power

The overall efficiency can be predicted by the following relationship equation.

$$\eta = \eta_M \times \eta_{HY} \times \eta_V \tag{45}$$

The mechanical efficiency is

$$\eta_M = \frac{Output\ power}{Input\ Shaft\ Power} = \frac{\rho\, g\, (Q_s + q)\, H_{vir}}{P_{shaft}} \tag{46}$$

The hydraulic efficiency is

$$\eta_{HY} = \frac{actual\ measured\ head}{head\ imparted\ fluid\ by\ impeller} = \frac{H}{H_{thn}} \tag{47}$$

The volumetric efficiency is

$$\eta_v = \frac{delivered\ \ flow\ rate}{delivered\ flow + \text{int}\, ernal\ leak\, age\ flow} = \frac{Q_s}{Q_s + q} \tag{48}$$

By substituting these efficiencies into Equation (45), overall efficiency becomes

$$\eta = \frac{\rho g H Q_s}{P_{shaft}}$$

In this study, input shaft power is 48.35 kW.

VI. Conclusion

The designed pump is aimed to use in mountain water distribution system which has about eight working hours per day and requires high head. So, multistage centrifugal type is selected. The designed pump can develop a head of 140 m and deliver 1.5 m^3/min of water at 2970 rpm. The thickness of volute casing to withstand the discharge pressure, 6 mm is selected depending upon the suction pipe diameter. When the performance of the designed pump is predicted, the maximum efficiency has nearly 63%.Volute friction losses, H_{f2} is neglected in this study. The maximum efficiency reaches at the capacity of about 0.025 m^3/s and nearly 20 m per stage. The materials to be used should be selected depending upon the type of water. The impeller is made of bronze to protect corrosion. To reduce the leakage from discharge to suction between the casing and impeller, the clearance must be made very small. It is used only to pump water at 70˚ F and if very hot water is used this pump will be damaged. The designed multistage centrifugal pump can fulfill the requirements of high head applications and industrial application, and then can improve pump efficiency.

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References

[1] EBARA Corporation, "Application of Pumps in Agriculture" Tokyo, Japan, November, 1999.
[2] Kyushu Institute Technology, Training Course, "Fluid Mechanics of Turbomachinery", Japan, 1996.
[3] Stepanoff. A. J, "Centrifugal and Axial Flow Pumps", 1957.
[4] Austin, H. Church, "Centrifugal Pump and Blower", 1972, John Wiley and Sons, Inc, New York.
[5] Khin Maung Aye, U, "Fluid Machinery for Mechanical Engineers", December, 2000.
[6] Touzson, J, "Centrifugal Pump Design", John Wiley and Sons, Inc, 2000.