DESIGN AND ANALYSIS OF A POSITIVE-DRIVE DOUBLE-PISTON SYNCHRONOUS GENTLE PUMP

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Abstract - The paper presents the design and analysis of a novel positive-drive double-piston synchronousgentle pump whichtargetsstress-sensitive fluids. The pump consists of two pistons, which rotate in a synchronized manner inside a circularchannel with rectangular cross section. Each piston is positively driven by an external servomotor, and the motion of the two servomotors is synchronized by a microcontroller. Analytical modeling of the flow field in the suction and in the compression channels of the pump result in a set of analytical expressions, which allows estimating the torque, power and hydraulic efficiency of the pump. Computational fluid dynamics (CFD) software was used to verify the analytical results.

Keywords - Double-Piston Synchronous Pump, Stress-Sensitive Fluids, Positive-Drive, Gentle Pumping

I. INTRODUCTION

Many applications in the biomedical and bioprocessing fields require gentle pumping for stress sensitive fluids or for fluids mixtures containing stress-sensitive particles. The influences of stress from the pumping device on such particles includes the alteration of morphology, release of intracellular or extracellular compounds and variation of viability [1-3]. Shear effects becomesignificant inthe processing of proteins in solution e.g. in the manufacture of enzymes [4] and in thefermentation, purification, formulation, fill and finish operations for protein drug products. To preserve the fragile constituents of these particles during pumping a gentle pumping action must be maintained [5].

Currently available pumps have problematic effects when used with stress-sensitive fluids. For example, the roller pump cause damage to the pumped particles due to squeezing the particles between the roller and the tube. Additionally, those pumps suffer from tube fracture and spallation [6]. Centrifugal pump also cause particle damage due to the abrupt pressure and velocity changes, and the high shear stresses that result from the high-speed rotation of the blades [7]. Lobe type pumps and double diaphragm pumps are restricted to pressures below 13.8 atmospheres (1400 kPa) and are not suitable for high viscosity fluids.

A gentle pump targeting stress-sensitive particles and utilizing the synchronized rotation of two pistons inside an annular channelwas previouslydescribed [8, 9]. The pumpemploystwo electromagnetically driven pistons, and achieves gentle valve-less and bearing-less pumping, but electromagnetic actuation causes piston slippage underhigh pressurerequirements [10, 11].This paper presents a new pump designbased on the previously described concept, butwith the two pistons driven positively by two servomotor [12]. An externalmicrocontrollercontrols the motion of the two servomotors and ensures the synchronized rotation of the pistons. The pump design achieves gentle a pumping action that avoids turbulence, stagnation, vortices, recirculation, high shear stresses, and negative pressure. The new conceptovercomes the slippage problem encountered on the electromagnetically driven pump design.

II. CONSTRUCTION AND OPERATING PRINCIPLE

The construction and the operating principle of the pump are described with reference to Fig. 1. Themain parts of the pumpincludea casing, a cover, and two piston pieces. Each piston piece is composed of a central cylindrical hub connected to a piston blade. The casing has two openings in its vertical walls, which serve as inlet and outlet ports. The cover and the casing have cylindrical openingsat their centers to allow the drive shafts of two servomotors to engage and drive the hubs of the upper and the lower piston pieces.

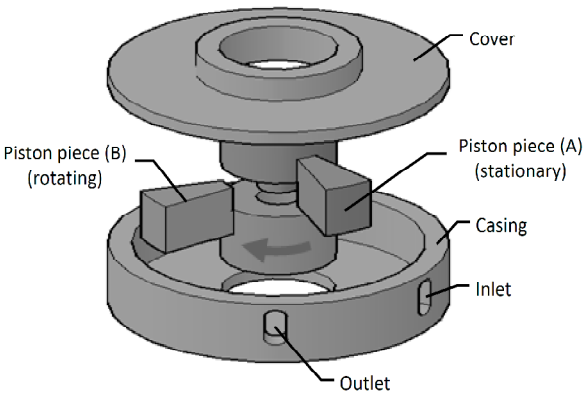


Fig. 1. Main Parts and Operating Principle

The pumping cycle consists of two phases. Fig. 1 refers to the first phase in which pistonAis stoppedbetween the inlet and outlet while piston Brotates clockwise. As it does, it creates vacuum in the vicinity behind its back face, anddraws fluid through the inlet port, while simultaneously pushes fluid at its front face through the outlet. When piston B approaches piston A from the right, piston A starts rotating clockwise, clearing that location for piston B, which stops there, and exchanges roles with piston A. This ends the first phase of the cycle, and begins the second phase, where piston A rotates clockwise. When A completes theloop, it exchanges role with B and the cycle is regenerated.

III. MECHANICAL DESIGN

Each piston piece is composed of a central hub and a piston blade. As illustrated in Fig. 1, and in Fig. 2, the central hub of the upper piston piece hasa circular protrusionthat fits into a matching hole in the hub of thelower piece. Whenthe pieces are assembled inside the casing, they effectively create a geometry of two pistons inside in a circular channel of rectangular cross section. The outer cylindrical faces of the two central hubs would produce the effect of the inner wall of the channel, and the inner face of the casing wall would produce the effect of the outer wall of the channel. Theupper and lower faces of the piston piece blades would be flush with the upper and lower face of the effectivechannel. The two pieces can rotate freely with respect to one another, and each piece has a key slot at its central hub that engages with a matching key in the servomotor drive shaft.The assembledpiston pieces produce the effect two piston placed inside an annular housing. The outer walls of the two central hubs assume the role of the inner wall of the annular channel of the pump, while the inner walls of the casing produce the outer walls of the annular channel.

Each piston piece is positively driven by an external servomotor and the motion of the two servomotors is synchronized by a microcontroller. Fig.2 illustrates the attachment of the two servomotors to the two piston pieces. The cover and the casing are provided with central collar at their centers. The collars serve to house and support the motors axially, radially and in the transverse directions. Each collar is provided with a shoulder on which the motor may rest, and a protrusion which fits into a matching groove in the motor's outer plate.. This arrangement allowsthe weight themotors carried to be carried by the shoulder on the collar, and transmitted to the pump casing, rather than the piston pieces. The piston two piecesattach to the motors by couplings, which allows the two piston pieces to float in the pumping chamber and thus rotate freely without rubbing the upper or lower faces against the lower face of the casing or the upper cover.

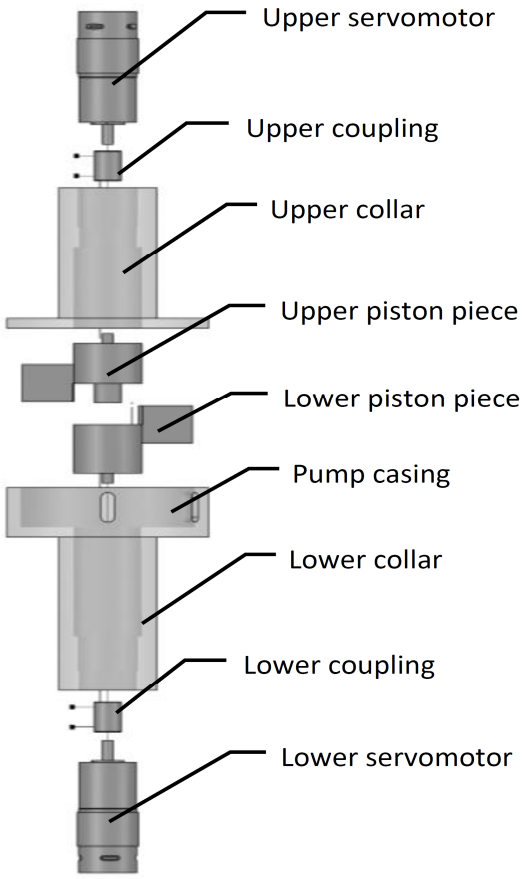


Fig. 2. Pump assembly

IV. ANALYTICAL MODELING

Consider a piston piece with a piston blade of mean radius r_m rotating inside the channel with an angular speed ω_p. The flow field in a segment of the annular channel ahead or behind the moving piston can be modeled as a plug flow in an annular channel with rectangular cross section. We will restrict this analysis to low values of Reynolds number, $Re < 2000$. The flow under this assumption is laminar and inertia effects are negligible. In this model, one wall boundary moving at the speed of the piston $v_p = \omega_p r_m$and one free boundary is maintained at the outlet pressure or inlet pressure p_{out}or p_{in}. Fig, 3 shows the flow model relevant dimensions for a segment in the channel ahead or behind the moving piston. For an incompressible Newtonian flow, the continuity and momentum equations are [13]:

$$\nabla . V = 0 \qquad (1)$$

$$\left[\frac{\partial V}{\partial t} + V.(\nabla V)\right] = \mu \nabla^2 V - \Delta p = 0 \qquad (2)$$

where$V$ is the velocity field and $p$is the pressure.
Using cylindrical coordinates andassuming that the

flow is laminar, fully developed, and unidirectional in the θ- direction ($v_r = v_z = \partial v_\theta/\partial\theta = 0$,where v_r, v_z and v_θ are the radial, vertical and transverse components of the velocity, (1) is automatically satisfied and the θ- component of (2) becomes

$$\mu\left\{\frac{\partial}{\partial r}\left[\frac{1}{r}\frac{\partial}{\partial r}(rv_\theta)\right] + \frac{1}{r^2}\frac{\partial^2 v_\theta}{\partial\theta^2} + \frac{\partial^2 v_\theta}{\partial z^2}\right\} = \frac{1}{r}\frac{\partial p}{\partial\theta} \quad (3)$$

The inertia and centrifugal forces for low Re are negligible compared to viscous force, and thus (3) becomes

$$\mu\frac{\partial^2 v_\theta}{\partial z^2} = \frac{1}{r}\frac{dp}{d\theta} \quad (4)$$

Integrating (4) while applying the boundary conditions $v_\theta(r,0) = v_\theta(r,h) = 0$, the velocity field across the channel is

$$v_\theta(r,z) = \frac{1}{2\mu r}[z^2 - hz]\frac{dp}{d\theta} \quad (5)$$

The volumetric flow rate $Q \equiv \omega_p r_m A$isobtained by integrating v_θ over the cross sectional area of the channel

$$Q = -\frac{h^3}{12\mu}\ln\left(\frac{r_o}{r_i}\right)\frac{dp}{d\theta} \quad (7)$$

where$r_o$and r_i are, respectively, the outer and inner radii of the channel. Equation (7) may be used to relate $dp/d\theta$ to ω_p by

$$\frac{dp}{d\theta} = -\frac{6\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\omega_p \quad (8)$$

An expression for $v_\theta(r,z)$ is obtained in terms of the pistons speed may be obtained using (5) and(8)

$$v_\theta(r,z) = 3\frac{r_o^2 - r_i^2}{\ln(r_o/r_i)}\frac{1}{r}\frac{z}{h}\left[1 - \frac{z}{h}\right]\omega_p \quad (9)$$

The pressure drop Δpalong a segment of the channel with an angular span $\Delta\theta$is obtained by integrating (8) between over $\Delta\theta$ to obtain

$$\Delta p = 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\Delta\theta\omega_p \quad (10)$$

Equation (10) allows for estimating the pressure distribution in the suction and in the compression chambers,$p_s$and p_c,in terms on the inlet and outlet pressures p_i and $p_o$to obtain

$$p_s = p_i - 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\theta_s\omega_p \quad (11)$$

$$p_c = p_o + 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\theta_c\omega_p \quad (12)$$

where$\theta_s$and θ_c are, respectively, the anglesfrom the inlet port or outlet ports to the radial segment of interest in the suction or compression chamber.

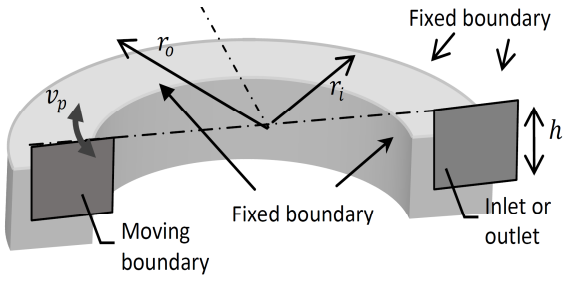


Fig. 3. Channel flow model with relevant dimensions

V. CFD MODELING

3-D CFD models for the flow field in the pump were constructed using ANSYS Fluent CFD for the purpose of verifying the analytical solution and for the general study of the pressureand speed in the pump. ANSYS Fluent provides user interfaces to input problem parameters and to examine the results graphically. It also provides mesh flexibility, and support 2D triangular and quadrilateral and 3D tetrahedral, hexahedral, pyramid, wedge, and mixed (hybrid) meshes, making it suitable for solving flow problems with complex geometries.

TheCFD model of the pump's channel was constructed with different channel heights and was meshedusing a structured grid with hexahedral elements. Pressure inlet, pressure outlet, and zero-velocity with no-slip at the walls were defined for the boundary conditions. ANSYS Fluent uses a finite volume technique to solve for the three dimensional equations of momentum and continuity. The pressure, momentum, and pressure-velocity coupling were achieved through second order, second order upwind, and simple-consistent schemes. The model is defined to be steady state, viscous, laminar, without considering the energy equation, and the solver to be segregated. A viscous fluid with $\rho = 889$ kg/m^3 and$\mu = 1.06$ Pa.s was used as the interior fluid. For the convergence criterion, the solver iterated the equations until the scaled residuals were less than 10^{-5} or until it stabilized at a constant value, which is still small enough to ensure convergence. A grid independent solution has been ensured by implementing first different meshes to determine the suitable element size and shape through observing the outlet mass fluxes through the microchannel. The scaled residual for the continuity and momentum equations was used to judge the convergence of the scaled over the unscaled equations.

VI. RESULTS AND DISCUSSION

The pressure values ahead and behind the moving piston blade, p_a and p_b, are readily obtained from (10) and (11)as

$$p_a = p_i - 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\theta_{ip}\omega_p \quad (13)$$

$$p_b = p_o + 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}\theta_{po}\omega_p \quad (14)$$

whereθ_{ip} and θ_{po} are, respectively, the angular spans between the channel inlet and the trailing edge of the piston blade, and between the leading edge of the pistonblade and the channel outlet.The difference in pressure between the leading and trailing faces of the piston is

$$\Delta p_p = p_{io} + 6\frac{\mu(r_o^2 - r_i^2)}{h^2\ln(r_o/r_i)}(\theta_{po} + \theta_{ip})\omega_p \quad (14)$$

where p_{io} is the difference in pressure between the inlet and outlet, or the imposed pressure on the pump. The expression $\theta_{po} + \theta_{ip}$ is recognized to be the angular span between the inlet and outlet ports θ_{io} minus the piston angle θ_p, which are design parameters and are independent of piston location as it moves during pumping. Equation (14) thus becomes

$$\Delta p_p = p_{io} + 6\frac{\mu(r_o^2 - r_i^2)}{h^2 ln(r_o/r_i)}(\theta_{io} - \theta_p)\omega_p \quad (15)$$

Equation (15) shows that the pressure against which the piston moves is composed of two parts; the imposedpressure on the pump, or its head, and an additional component p_v resulting from viscous losses in the suction and the compression chambers, which is given by

$$p_v = 6\frac{\mu(r_o^2 - r_i^2)}{h^2 \ln(r_o/r_i)}(\theta_{io} - \theta_p)\omega_p \quad (16)$$

Equation 15 may be used to calculate the net torque T on the piston blade due to fluid pressure by integrating the differential torque $dT = r\Delta p_p dA$ between r_i and r_o

$$T = \Delta p_p hwr_m = (p_{io} + p_v)hwr_m \quad (17)$$

where $w = r_o - r_i$ is the channel width, and $r_m = (r_o + r_i)/2$ is its mean radius.The actual mechanical power needed to rotate the piston $P_a = T\omega_p$ is

$$P_a = (p_{io} + p_v)\omega_p hwr_m = \Delta p_p Q \quad (18)$$

Equation (18) shows that the power needed to move the piston is composed of two components; an ideal power $P_i = p_{io}Q$ which would berequired in the absence of viscous losses, and a viscous power loss component $P_v = p_v Q$. The hydraulic efficiency, $\eta_H = P_i/P_a$ is found from (18) to be

$$\eta_H = \frac{\Delta p_p - p_v}{\Delta p_p} = 1 - \frac{p_v}{p_{io} + p_v} = \left(1 - \frac{p_v/p_{io}}{1 + p_v/p_{io}}\right) \times 100\% \quad (19)$$

Equation (19) highlights the significance of the pressure ratio $r_p = p_v/p_{io}$ on the value of η_H. For $r_p = 0$, $\eta_H = 100\%$, which is an expected result in the absence of viscous losses. The value of η_H drops to 50% for $r_p = 1$, and continues to drop with r_p until it approaches zero as $r_p \to \infty$.

To compare the derived analytical expressions with CFD results, consider a pump with the design and operational parameters in Table I with its moving piston located midway between the inlet and the outlet such that $\theta_{ip} = \theta_{po} = \pi/2$. Fig. 4 shows a contour plot of the CFD estimates of the pressure distribution in the compression vicinity. CFD estimates the average pressure at the leading edge in front of the piston to be $p_a = 14.050$ kPag compared with $p_a = 13.939$kPa from the analytical expression in (13). For the suction chamber, CFD estimate the average pressure at the trailing edge of piston to be $p_b = -4.010$ kPag compared to $p_b = -3939$ kPa. Based on the above calculations, Table II compares the analytical and CFD estimates of Δp_p, T, P_L and η_H for the pump with the parameters in Table I.

Table I. Parameters used in CFD

Parameter	Value
r_i(mm)	22
r_o(mm)	56
r_m(mm)	39
h(mm)	6
μ(Pa.s)	1.06
ω_p(rad/s)	5
$(\theta_{io} - \theta_p)$	π
p_i(kPag)	0
p_o(kPag)	10

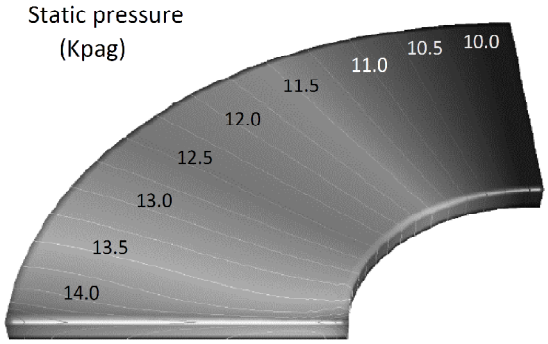


Fig. 4. CFD pressure contours

The plot in Fig. 4 shows a gradual change in the pressure along the channel, which confirms the prediction of (10) of a gradual pressure change with θ. It also confirms the assumptions employed in arriving at this equation.

CONCLUSION

Thepaper presented a new design for a double-piston synchronous pump targeting stress-sensitive fluids. The design provides positive drive for the pistons, which overcomes the problems encountered in designs employing electromagnetically driven pistons. The design of the pistonsassembly inside the pump casing effectively replicates a geometry of two pistons inside in a circular channel of rectangular cross section. The two pistons are provided by central hubs which allows them to be driven positively through two motor drive shafts which penetrate the casing.

Analytical expressionsfor the flow field in the suction and compression channels of the pumpwere developed based on laminar unidirectional flow assumptions, and

yielded expressions for the velocity and pressure fields. These expressions were used to derive simple expressions for the piston torque, power and the pump's hydraulic efficiency in terms of its design and operation parameters. CFD models were developed using ANSYS-FLUENT, and CFD results were in good agreement with those obtained from the derived analytical expressions.

REFERENCES

[1] Millward, H.R. and Bellhouse, B.J. Mammalian cell damage in a novel membrane bioreactor. Biotechnol. Bioengng. 43, pp. 899–906, June 1994.

[2] Jaouen, P., Vandanjon, L., and Quéméneur, F. "The shear stress of microalgal cell suspensions (Tetraselmissuecica) in tangential flow filtration systems: the role of pumps," Bioresource Technology, Volume 68, Issue 2, May 1999.

[3] Wisniewski C. and Grasmick A. Combined bioepuration and separation by tangential microfiltration. Proceedings of Euromembrane Vol. I, pp. 544–547. Bath, UK, May 1995.

[4] Ghadge RS, Patwardhan AW, Sawant SB, Joshi JB."Effect of flow pattern on cellulase deactivation instirred tank bioreactors". ChemEngSci 60:1067-1083, Feb 2005.

[5] Rathore N, Rajan RS "Current perspectives on stability of protein drug products during formulation, fill and finish operations"BiotechnolProg 24:504-514, May 2008.

[6] Y. Nosé, M. Yoshikawa, S. Murabayashi, and T. Takano, "Development of Rotary Blood Pump Technology: Past, Present, and Future," Artificial Organs 24: 6, June 2000.

[7] Chua L and Akamatsu T "Measurements of gap pressure and wall shear stress of a blood pump model Med." Eng. Phys. 22 175–88, April 2000.

[8] M. Kilani and A. Abbadi, "Method and Apparatus for Pumping a Fluid using Magnetic Pistons," GB2477276 British Patent, August 2011.

[9] M Kilani, H Khasawneh, A Badran, A Awidi, "Further development on a gentle electromagnetic pump for fluids with stress-sensitive microparticles" Sensors and Actuators A: Physical 247, 440-447, June 2016

[10] AT Al-Halhouli, M Kilani, S Büttgenbach, "Development of a novel electromagnetic pump for biomedical applications," Sensors and Actuators A: PhysicalVol 162, Issue 2, August 2010, Pages 172-176

[11] M Kilani, H Khasawneh, AA Abbadi "Design and testing of a gentle pump with rotating magnetic field for fluids with stress-sensitive microparticles," 2015 10th International Symposium on Mechatronics and its Applications (ISMA)

[12] M Kilani and A Khan, "A positive drive double piston synchronous gentle pump," U.S. Provisional Pat. App. No. 35001677, April 2019.

[13] White F. M. Viscous Fluid Flow, Tata Mcgraw Hill; 3rd edition 2011.

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