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مؤلفين آخرين:	Kiwan, Suhil M.، Aldoss, Taha khalil(super)
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By: Mahmoud Ersheid AL-Bawaneh
Supervisor: Prof. Taha K. Aldoss
Co-supervisor: Dr. Suhil M. Kiwan

ABSTRACT

The finite element program (**ANSYS**) is used to analyze laminar flow of an incompressible Newtonian fluid between rotating multi–finite discs simulating vaneless impeller pump. The flow characteristics (velocity and pressure distribution) are calculated; in addition the performance of such rotor as a machine impeller is investigated. Different parameters affecting the performance such as Re (rotational Reynolds number), N_Q (dimensionless flow rate), N_P (rotor spacing ratio), N_C (clearance spacing ratio), and the number of rotating discs at certain pump discpac ratio are investigated.

Two basic flow structures are observed: Batcholer-type flow, with a separate boundary layers on each disc with a rotating core of fluid in between, and Stewartson-type flow with virtually no core rotation.

N_P, N_C, Re and number of rotating discs at certain pump discpac ratio are found to strongly affect both the flow and the performance characteristics of the pump.

سريان مائع غير قابل للأنضغاط بين أقراص محددة الأبعاد لتمثيل مضخة بدون زعانف

إعداد: محمود ارشيد البواعنة
إشراف: أ.د طه الدوس. رئيساً
د. سهيل كيوان. مشاركاً

ملخص

لقد تم دراسة سريان مائع غير قابل للأنضغاط بين أقراص محددة الأبعاد لتمثيل مضخة بدون زعانف باستخدام معادلات الحركة. لقد حولت هذه المعادلات إلى مجموعة معادلات جبرية ، وتم حلها باستخدام برنامج يسمى انسيس. مما أمكن الحصول على نتائج لتوزيع سرعة و طبيعة المائع وخصائص هذه المضخة من حيث الاستطاعة والطاقة المبذولة والفعالية إعتماداً على عدة متغيرات، منها: رقم رينولدز، تدفق المائع، المسافة بين الأقراص الدوارة، وعدد الأقراص الدوارة. لقد أظهرت الدراسة أن رقم رينولدز، تدفق المائع،المسافة بين الأقراص الدوارة، وعدد الأقراص الدوارة تؤثر على توزيع سرعة و طبيعة المائع وخصائص هذه المضخة من حيث الاستطاعة والطاقة المبذولة والفعالية.

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CHAPTER ONE

CHAPTER TWO
MATHEMATICAL FORMULATION

CHAPTER THREE
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CHAPTER FOUR
RESLUTS AND DISCUSSION

NOMENCLATURE

D = diameter of the impeller [m].
g = acceleration due to gravity [m/s^2].
H = pump discharge head [m].
n = number of rotating discs.
N_C = clearance spacing ratio, S_C/D
N_H = dimensionless pump head, $gH/(\omega D)^2$.
N_P = rotor spacing ratio, S_P/ D.
N_{PAC} = pump discpac ratio, S_{PAC}/D.
N_{PS} = shaft input power, $P_S/ \rho D^5 \omega^3$.
N_Q = dimensionless flow rate, $Q_{IN}/ (\pi\ \omega R^2{}_1 D)$.
P = pressure [N/m^2].
P_S = shaft input power, $T_S\ \omega$, [N.m/s].
R_1 = inner radius of the impeller [m]
R_2 = outer radius of the impeller [m].
Re = rotational Reynolds number, $\omega\ D^2/\upsilon$.
r, θ, z = dimensionless cylindrical coordinates axes [m, rad, m]
T_S = shaft torque on the impeller [N.m].
S_C = clearance spacing [m].
S_P = rotor spacing [m].
S_{PAC} = pump disc pack spacing, $(S_C + [n-1]\ S_P)$, [m].
V_r = radial velocity component [m/s].
V_θ = swirl velocity component [m/s].
V_z = axial velocity component [m/s].
Q = Flow rate [m^3/s].

GREEK SYMBOLS

η = overall pump efficiency, $(\rho gHQ/P_S) = (N_H\ N_Q/N_{PS})$.
μ = dynamic viscosity [N.s/m^2].
τ_w = wall shear stress [N/m^2].
υ = kinematic viscosity [m^2/s].
ρ = Density [kg/m^3].
ω = impeller angular velocity [rad/s].

SUBSCRIPTS

a = average
IN = Inlet.
r, θ, z = Radial, swirl and axial.

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JORDAN UNIVERSITY OF SCIENCE AND TECHNOLOGY

August 2001

Incompressible Viscous Flow Between Finite Discs Simulating Vaneless Impeller Pump

By

Mahmoud Ersheid Ahmad Al-Bawaneh

Thesis Submitted in Partial Fulfillment of the Requirements of the Degree of M.Sc. in Mechanical Engineering

At
Faculty of Graduate Studies

JORDAN UNIVERSITY OF SCIENCE AND TECHNOLOGY

August 2001

Signature of Author………… Date of Signature ٢٠٠١/٨/٢٦

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Dr. O. M. Haddad	O.M. HADDAD Aug. 19, 200
Dr. N. Jubeh (Cognate. Albalqa Applied University)	N. Jubeh

DEDICATION
TO

My Parents.......................................
My Brothers : Hakim, Hakimat, Ahmad,
Mohammad &Ebrahim.

My Sister.......................................
My Friend (Hashem)...............................
My Best Friends.................................
All Strongly I Love..............................

ACNOWLEDGEMENTS

I'am glad to express my deep appreciation for the assistance of many others in the preparation of this thesis. My Supervisor Prof. Taha K. Aldoss and my Co-supervisor Dr. Suhil M. Kiwan who encouraged me strongly in this work and provided a smooth and effective help and support throughout the thesis.

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CHAPTER ONE

1.1. INTRODUCTION

The problem of disc flows has occupied a central position in the field of fluid mechanics in recent years. Disc flows have immediate technical applications (rotating machinary, lubrication, viscometry, heat and mass exchangers, biomechanics, oceanography), but quite apart from that they have intrinsic interest.

The disc pump uses a unique non-impingement pumping principle, which is neither centrifugal nor positive displacement. The pumping mechanism is called the Discpac; a series of parallel, equally spaced discs which move product using the forces of boundary layer and viscous drag.

When a fluid enters the pump, its molecules adhere to the surfaces of these discs, providing a boundary layer. As the discs rotate, energy is transferred to successive layers of molecules in the fluid between the discs, generating velocity and pressure gradients across the width of the Discpac. This combination of boundary layer and viscous drag effectively creates a powerful dynamic force field that "pulls" the product through the pump in a smooth, pulsation-free flow. The fluid moves parallel to the discs, with the boundary layer creating a molecular buffer between the disc surfaces and the fluid. The key point is that there is no "impingement" of the fluid on the pump's moving parts. This non-impingement design is where the disc pump differs from other pumps on the market, all of which impinge on the product, in effect "pushing" it through the system.

The disc pump lack of impingement and laminar flow leads to numerous benefits in handling difficult fluids, such as viscous, abrasive, high solids and air-entrained fluids, and delicate and shear sensitive products. Disc pumps resemble conventional centrifugal pumps, except that the impeller consists of a set of closely spaced parallel smooth discs. When impeller rotates it rotates the fluid inside it, this create the centrifugal forces needed to transfer the angular momentum (and hence energy) to the fluid by means of shear forces, so some times the disc pumps are called shear pumps.

Experiments show the success of the idea of using multiple discs for pumps.

The advantages for such pumps relative to conventional vaned impeller pumps include greater stability, low sensitivity to cavitation, and the ability to operate with unusual fluids for which conventional pumps are unsuitable such as highly viscous fluids, two phase gas-liquid mixtures, highly loaded slurries and suspensions, non–Newtonian fluids (such as multiple discs centrifugal blood pump), etc, so it is easier to manufacture and cheaper than the conventional pumps and this would reduce the overall cost of the machine. Although the efficiency of these pumps with the ordinary fluids is less than that of conventional vanes rotor pumps, their ability to operate with unusual fluids (such as blood in biomechanics applications) for which conventional pumps fail makes them attractive for variety of special purpose applications.

1.2. LITERATURE REVIEW

Relevant previous research concerned itself almost entirely with infinite–disc flow. The sole reason for this, one suspect, is that the similarity transformation available when the discs are infinite, reduces the number of spatial dimension of the problem to one.

Starting with the basic work by Karman [1], where a self-similar solution of the complete Navier–Stokes equations for a flow above a rotating disc was examined. The analyses of references [2, 3, 4, 5] are for infinite stationary or rotating discs.

References [6, 7, 8, 9] considered analytical and experimental investigation of flow between finite rotating discs.

Several other experimental investigations have been reported [10, 11, 12, 13] in which the flow between finite rotating discs were constructed and tested. These have shown the feasibility of multiple discs turbomachinary and rotating discs and have presented the uniqueness of certain performance characteristics.

Szeri and Adams [8] studied finite discs who used an approximation where the radial variation of shear stress is neglected. Szeri et al. [10] measured the velocity of water between finite–rotating discs with and without through flow by using Laser–Doppler device. The equilibrium flow is unique, and at mid-radius flow shows a high degree of independency from boundary layer in radial co-ordinate. Gan et al. [11] described a combined experimental and computational study of laminar and turbulent flow between contra-rotating discs. Laminar computations produce Batchelor–type flow, radial outflow occurs in boundary layers on the discs and inflow is confined to a thin shear layer in the mid-plane: between the boundary layers and the shear layers, two contra-rotating cores of fluid are formed.

Wilson et al. [12] described the flow in the internal cooling–air systems of gas turbines as simple rotating–disc systems.

Wilson et al. [13] described the computation of flow and heat transfer in a rotating cavity with a stationary outer casing, using a steady state, axisymmetric finite volume solver.

Moffatt [14] revealed that air viscosity is responsible for the observed abruptness of the settling process in rotating discs. Ng [15] investigated the flow structure and the temperature distribution in a ventilated computer hard disc passage using the finite element method. The problem is analyzed as corotating and counter rotating discs. Kitamura and Akihiro [16] analyzed the unsteady liquid film flow of nonuniform thickness on a rotating disc. Karabay et al. [17] performed theoretical, computational and experimental studies into the fluid mechanics, thermodynamics and heat transfer characteristics of the cover-plate pre-swirl systems.

Henderson and Alan [18] studied an implantable centrifugal blood pump with a recirculating purge system (Cool-Seal System). Rice [19] performed both theoretical and experimental studies of laminar flow disc pump using the assumption of a constant friction factor over the radius of the disc. Breiter and Pohlhausen [20] analyzed the laminar flow between two rotating discs using the Navier-Stokes equations. Byrne [21] analytically derived performance curves for laminar flow disc pumps for use in liquid fueled rockets, by expanding the equations of Breiter and Pohlhausen. Hasinger and Kehr [22] analyzed the performance of a laminar flow disc pump assuming a parabolic flow profile between discs. Their predictions agreed to within 5 percent with the results of Breiter and Pohlhausen. Balje [23] applied the Hasinger and Kehr method to predict the performance of laminar flow pumps. He found that the pumps having a low sensitivity to cavitation.

In the present work, the problem will be analyzed in new format and style dividing the model problem to three main major parts:

1. Studying the flow structures and characteristics where the swirl and radial

velocities will be calculated.

2. Studying the performance characteristics (head, shaft input power and overall efficiency) of the rotor pump as a vaneless rotor.

3. Studying the effect of flow structures between parallel spaced discs and the performance characteristics of the pump and determining how really important the relation between them in the vaneless impeller pump performance.

Also, the effect of clearance spacing ratio and rotor spacing ratio on the steady laminar flow analysis between parallel spaced discs and the effect on the performance of the rotor will be investigated. The resulting predicted performance of the vaneless impeller is presented in dimensionless parameters, which influence the pump performance. The results show, in a generalized form, how pump performance depends upon such dimensionless parameters as Re (rotational Reynolds number), N_Q (dimensionless flow rate), N_P (rotor spacing ratio), Nc (clearance spacing ratio) and the number of rotating discs at certain pump discpac ratio, etc.

CHAPTER TWO
MATHEMATICAL FORMULATION

2.1. INTRODUCTION

Consider the physical situation of multi-finite rotating discs simulating a vaneless impeller pump. The cylindrical polar co-ordinates (r, θ, z) are used. The lower stationary disc, in the plane $z=0$ has a zero angular velocity. The upper two rotating discs, at $z=S_C$ and (S_C+S_P) respectively, has the same angular velocity ω. The considered flow field is laminar and steady, and the fluid is incompressible and Newtonian with constant viscosity.

The flow field is considered laminar based on the reference [24] where Schlichting has indicated that turbulence occurred above $(Re=3\times10^5)$. Figure 1b shows a schematic diagram of the problm setup of flow between rotating discs simulating a vaneless impeller pump.

2.2. GOVERNING EQUATIONS

- **Assumptions :**

1. The fluid is incompressible and Newtonian with constant viscosity.
2. The considered flow field is laminar and steady.

For axisymmetric flow of an incompressible fluid, the continuity equation and the Navier-Stokes equations written in cylindrical co-ordinates are:

Continuity

$$\frac{\partial}{\partial \hat{r}}(\hat{r}\hat{V}_r) + \frac{\partial}{\partial \hat{z}}(\hat{r}\hat{V}_z) = 0 \qquad (1)$$

Momemtum : r -componrent

$$\hat{V}_r \frac{\partial \hat{V}_r}{\partial \hat{r}} - \frac{\hat{V}_\theta^2}{\hat{r}} + \hat{V}_Z \frac{\partial \hat{V}_r}{\partial \hat{z}} = \nu\left(\nabla^2 \hat{V}_r - \frac{\hat{V}_r}{\hat{r}^2}\right) - \frac{1}{\rho}\frac{\partial \hat{P}}{\partial \hat{r}} \qquad (2)$$

Momemtum : θ -componrent

$$\frac{\hat{V}_r}{\hat{r}}\frac{\partial}{\partial \hat{r}}(\hat{r}\hat{V}_\theta) + \hat{V}_z \frac{\partial \hat{V}_\theta}{\partial \hat{z}} = \nu\left(\nabla^2 \hat{V}_\theta - \frac{\hat{V}_\theta}{\hat{r}^2}\right) \qquad (3)$$

Momemtum : z -componrent

$$\hat{V}_r \frac{\partial \hat{V}_Z}{\partial \hat{r}} + \hat{V}_z \frac{\partial \hat{V}_z}{\partial \hat{z}} = \nu \nabla^2 \hat{V}_z - \frac{1}{\rho}\frac{\partial \hat{P}}{\partial \hat{z}} \qquad (4)$$

THE BOUNDARY CONDITIONS ARE:

No slip boundary condition:

$$\hat{V}_r(\hat{z}=0)=0 \tag{5}$$

$$\hat{V}_r(\hat{z}=S_C)=0 \tag{6}$$

$$\hat{V}_r(\hat{z}=S_{PAC})=0 \tag{7}$$

Impermeable boundary condition:

$$\hat{V}_z(\hat{z}=0)=0 \tag{8}$$

$$\hat{V}_z(\hat{z}=S_C)=0 \tag{9}$$

$$\hat{V}_z(\hat{z}=S_{PAC})=0 \tag{10}$$

Swirl boundary condition: (boundary conditions of rotating discs)

$$\hat{V}_\theta(\hat{z}=0)=0 \tag{11}$$

$$\hat{V}_\theta(\hat{z}=S_C)=\hat{r}\omega \tag{12}$$

$$\hat{V}_\theta(\hat{z}=S_{PAC})=\hat{r}\omega \tag{13}$$

Inlet boundary condition: (inlet axial velocity boundary condition)

$$\hat{V}_z(\hat{z}=0;0\prec\hat{r}\prec R_1)=\left(\frac{Q_{IN}}{\pi R_1^2}\right) \tag{14}$$

Outlet boundary condition:

$\hat{p}(\hat{r}=R_2\ =\frac{D}{2})=0$ (It is recommended based on the ***ANSYS manual***) (15)

The reference length is the outer diameter of the impeller and the reference velocity is the angular velocity multiplies by the outer diameter both designated by D and (ω D) respectively. Thus, the variables in the dimensionless form are:

$$r = \frac{\hat{r}}{D} \qquad z = \frac{\hat{z}}{D} \qquad p = \frac{\hat{P}}{\rho(\omega D)^2} \tag{16}$$

$$V_r = \frac{\hat{V}_r}{\omega D} \qquad V_\theta = \frac{\hat{V}_\theta}{\omega D} \qquad V_Z = \frac{\hat{V}_z}{\omega D} \tag{17}$$

Using the above dimensionless quantities, the dimensionless form of equations 1-4 are:

Continuity

$$\frac{\partial}{\partial r}(rV_r) + \frac{\partial}{\partial z}(rV_z) = 0 \tag{18}$$

Momemtum : r -componrent

$$V_r \frac{\partial V_r}{\partial r} - \frac{V_\theta^2}{r} + V_Z \frac{\partial V_r}{\partial z} = \frac{1}{\mathrm{Re}}\left(\nabla^2 V_r - \frac{V_r}{r^2}\right) - \frac{\partial P}{\partial r} \tag{19}$$

Momemtum : θ-componrent

$$\frac{V_r}{r}\frac{\partial}{\partial r}(rV_\theta) + V_z \frac{\partial V_\theta}{\partial z} = \frac{1}{\mathrm{Re}}\left(\nabla^2 V_\theta - \frac{V_\theta}{r^2}\right) \tag{20}$$

Momemtum : z -componrent

$$V_r \frac{\partial V_Z}{\partial r} + V_z \frac{\partial V_z}{\partial z} = \frac{1}{\mathrm{Re}}\nabla^2 V_z - \frac{\partial P}{\partial z} \tag{21}$$

THE DIMENSIONLESS BOUNDARY CONDITIONS ARE:

No slip boundary condition:

$$V_r(z=0)=0 \tag{22}$$

$$V_r\left(z=\frac{S_C}{D}\right)=0 \tag{23}$$

$$V_r\left(z=\frac{S_{PAC}}{D}\right)=0 \tag{24}$$

Impermeable boundary condition:

$$V_z(z=0)=0 \tag{25}$$

$$V_z\left(z=\frac{S_C}{D}\right)=0 \tag{26}$$

$$V_z\left(z=\frac{S_{PAC}}{D}\right)=0 \tag{27}$$

Swirl boundary condition: (boundary conditions of rotating discs)

$$V_\theta(z=0)=0 \tag{28}$$

$$V_\theta\left(z=\frac{S_C}{D}\right)=r \tag{29}$$

$$V_\theta\left(z=\frac{S_{PAC}}{D}\right)=r \tag{30}$$

Inlet boundary condition: (inlet axial velocity boundary condition)

$$V_z\left(z=0;0\prec r\prec\frac{R_1}{D}\right)=\left(\frac{Q_{IN}}{\pi\omega R_1^2 D}\right) \tag{31}$$

Outlet boundary condition:

$$P(r=1/2)=0 \tag{32}$$

Thus, from the above dimensionless governing equations and applied boundary conditions, the dimensionless parameters affecting the problem will be as follows:

The rotational Reynolds number Re:

$$\mathrm{Re} = \frac{\omega \times D^2}{\nu} \tag{33}$$

Dimensionless flow rate N_Q:

$$N_Q = \left(\frac{Q_{IN}}{\pi \omega R_1^2 D} \right) \tag{34}$$

Clearance spacing ratio N_C:

$$N_C = \frac{S_C}{D} \tag{35}$$

Rotor spacing ratio N_P:

$$N_P = \frac{S_P}{D} \tag{36}$$

Pump discpac ratio N_{PAC}:

$$N_{PAC} = \frac{S_{PAC}}{D} \tag{37}$$

The effect of the above dimensionless parameters will be investigated and their effects on the flow and performance characteristics of the pump will be presented.

CHAPTER THREE

METHOD OF SOLUTION

3.1. INTRODUCTION

FLOTRAN is a finite-element based, general-purpose algorithm, which solves the Navier-Stokes, and energy equations using a segregated or sequential solution method. The velocity-pressure formulation uses an equal-order approximation for velocity and pressure, and solves for each variable field in an iterative manner.

FLOTRAN uses a monotone streamline upwind technique to discretize the advection terms and has demonstrated improved accuracy of conventional upwind methods for finite elements.

The effects of periodic boundary conditions, porous media flow, distributed resistances, moving walls, conjugate heat transfer, thermal buoyancy, turbulent flow, incompressible and compressible flow, and rotating reference frames may be simulated.

3.2. BOUNDARY CONDITIONS OF THE ROTATING DISCS

Solving the problem of the flow between two discs, one is stationary and the other is rotating cannot be handled directly by the ANSYS code. A seperate FORTRAN program was written to generate the needed boundary conditions on the rotating disc. The results of the Fortran program were added to ANSYS program code.

In order to verify the modifications, it is applied on a sample problem of reference [10]. The sample problem is flow between two finite discs: one is rotating with certain angular velocity and the other is stationary. The flow is introduced through opening at the center of both discs as shown in figure 1a.

3.3. OVERVIEW OF A FLOTRAN ANALYSIS

Almost all commercial CFD codes contain three main elements:

(a) A pre-processor.

(b) A solver.

(c) A post-processor.

The function of each of these elements can be briefly described as follows:

(a) A pre-processor:

- Definition of the geometry of the region of interest: the computational domain.
- Grid generation: the sub-division of the domain into a number of smaller, non-overlapping sub-domains or grid (or mesh).
- Selection of the physical phenomena that need to be modeled.

- Definitions of fluid properties.

- Specification of appropriate boundary conditions.

In general, the larger the number of grids, the better the solution accuracy. A grid-independency solution must be obtained.

(b) A solver:

Flotran is a finite element code; the following steps form the basis of the solver:

- Approximation of the unknown flow variables by means of simple functions. Finite element method uses simple piecewise functions valid on elements to describe the local variations of the unknown flow variables.

- Discretisation by substitution of the approximations into the generalized equations. As a result a set of algebraic equations for the unknown coefficients of the approximating functions are obtained. Thus, the system of the PDE is converted into a system of algebraic equations.

- Solution of the algebraic equations.

(c) A post-processor:

The post-processor includes:

- Domain geometry and grid display.
- Vector plots, particle tracking and derived quantities.
- Line and shaded contour plots, etc.

A typical FLOTRAN analysis consists of seven main steps:

1. Determining the problem domain.
2. Determining the flow regime.
3. Creating the finite element mesh.
4. Applying the boundary conditions.
5. Setting the FLOTRAN analysis parameters.
6. Solving the problem.
7. Examining the results.

3.3.1. DETERMINING THE PROBLEM DOMAIN

It is needed to determine the proper domain of this analyzed problem: definitions of the geometry of the region of interest (the computational domain). The geometry of the problem is the region (or domain) between an axisymmetric plane, one disc is stationary, and two discs are rotating with certain angular velocity as in figure 1b.

3.3.2. DETERMINING THE FLOW REGIME

The character of the flow is specified. The fluid properties are needed to be estimated such as fluid density and viscosity. The laminar option is activated based on the estimated rotational Reynolds number Re as indicated in Schlichting [24]. The incompressible option is also activated.

3.3.3. CREATING THE FINITE ELEMENT MESH

Mapped meshing is used, it is more effective, maintains a consistent mesh pattern along the boundary and it gives accurate results. The size of the mesh is specified on the boundary of the computational domain. The size of the grid is 120 (radial) x 120 (axial) where the grid-independency is obtained.

3.3.4. APPLYING THE BOUNDARY CONDITIONS

The boundary conditions can be applied before or after meshing the domain. Consider every model boundary, If a condition is not specified for a dependent variable, a zero gradient of that value normal to the surface is assumed. The boundary conditions can be changed between restarts, if it is needed. The boundary conditions of the problem are specified as shown in equations 22 up to 32.

3.3.5. SETTING THE FLOTRAN ANALYSIS PARAMETE

The FLUID141 2-D is activated, solve for two-dimensional single-phase viscous fluid. The dimensions shape (or characteristics) of FLUID141 is Quadrilateral, four nodes or triangle, three nodes.

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