Design and Analysis of Stator, Rotor and Blades of the Axial flow Compressor

Ujjawal A. Jaiswal
Institute of Technology
Nirma University
Ahmedabad, Gujarat, India
11mmed03@nirmauni.ac.in

Prof. S. J. Joshi
Institute of Technology
Nirma University
Ahmedadabad,Gujarat,India
s.j.joshi@nirmauni.ac.in

Abstract: **Axial flow compressor is one of the most important parts of Gas turbine. In design of Axial flow compressor the work presented comprises of basic flow parameters and dimensions of parts, this makes the further design process quite simple and the results will be helpful to take further changes or improvement at the time of detailed design. The objective of work presented is to design Axial flow compressor by using mean line method for a given mass flow rate and required pressure ratio. The parameters determined also include thermodynamic properties of the working fluid, stage efficiency, number of rotor and stator blades, tip and hub diameters, blade dimensions (chord, length and space) for both rotor and stator, Mach number, flow and blade angles (blade twist) . The same parameters are also determined for all five stages. The twist of the blades can be calculated along the blade length at any required number of sections selected by the designers to obtain smooth blade twist profile. NACA 65410 profiles is used to generate coordinates of the blade. Further, in the process the first stage of axial flow compressor blade is developed using Solid works modeling. Also CFD simulation has been carried out using Ansys CFX to validate the results. Also Static structural Analysis has been performed to check whether the rotor is safe at given speed.**

***Index Terms* - Compressor, NACA, Axial flow, rotor, stator**

I. Introduction

Axial-flow compressors are used in medium to large thrust gas turbine and jet engines. The compressor rotates at very high speeds, adding energy to the airflow while at the same time compressing it into a smaller space. The design of axial-flow compressors is a great challenge, both aerodynamically and mechanically. [1]

The aerodynamic compressor design process basically consists of mean line prediction calculation, through flow calculation, and blading procedures. The mean line prediction is the first step within compressor design. It is a simple one-dimensional calculation of flow parameters along the mid-height line of the compressor where global parameters as the annulus geometry, the number of stages, and the stage pressure ratios are scaled [2].

It is necessary to design Axial flow compressor at preliminary level and require parameters can be checked at initial level so further improvement can be made at primary level before start a Detailed design.

The mean line prediction process as it is performed today is a very quick and reliable method for compressor preliminary design Typically, the results of the preliminary process are obtained by time consuming manual parameter studies based on engineering intuition or experience. The final one-dimensional solution is used as an initial guess for the subsequent design process, e.g. for through flow calculations [4].

These calculations also include thermodynamic properties of the working fluid, stage efficiency, and number of rotor and stator blades, tip and hub diameters, chord, length and space of blade for rotor and stator, Mach number, flow and blade angles. A repeated stage calculation is made to calculate the above parameters along compressor stages.

The traditional approach to axial-flow compressor aerodynamic design was to use various families of airfoils as the basis for blade design. American practice was based on various families designed by the National Advisory Committee for Aeronautics (NACA),the most popular being the 65-series family[5].NACA 65410[6] Airfoil is used here to generate blade coordinates.

In the 1954,Donald M. Sandercock, kovach and Seymour Lieblein designed a five stage axial compressor and carried out an experimental investigation with all rotor rows operating with transonic relative inlet Mach numbers designed as research unit to study the potentialities and problems arising from the compounding of transonic stages.

II. Design Process

The steps involved in design of axial flow compressor is Shown in figure 1. Suitable design point under sea level static conditions [5], given pressure ratio is 4.15 and Air mass flow rate is 20 kg/sec. Also, Axial velocity C_a= 170 m/s and blade speed is 250m/s is considered.

III. Annulus Dimensions

Stagnation and static properties can be found by simple thermodynamic equations and the Enthalpy-Entropy diagram which is shown in fig.2

Once P0, T0 and P, T at entry and exit have been found.

Consider mass flow rate, Pressure ratio, Axial velocity

⇩

Calculate Annulus dimensions using continuity equation

⇩

Select appropriate adiabatic efficiency

⇩

Calculate stagnation pressure and temperature at inlet and outlet

⇩

Estimate number of stages

⇩

Calculate inlet and outlet stagnation pressure and temperature for every stages

⇩

Calculate radius at hub ,Tip and Mean for each stage

⇩

Determine variation of Air angles from root to tip

⇩

Calculate number of blades for each stages

⇩

Calculate Blade height, Pitch, chord and Stagger angle

⇩

Generate blade coordinates for stator and rotor for all stages

⇩

Study its compressibility effects

⇩

Check weather all requirements are satisfied

Fig 1 Aerodynamic Design input and Steps

Density can be determined by $\rho = \frac{P}{RT}$. At inlet, radius at tip can be found by, $r_t^2 = \frac{m}{\pi.\rho.C_a.\left[1-\left(\frac{r_h}{r_t}\right)^2\right]}$ .From hub tip ratio and H= $r_t - r_r$ blade height and radius at hub can be found. Also, $r_m = {(r_t + r_r)}/{2}$ and Blade speed U=2.π.r.N. In mean line design methodology mean radius remain constant for all stages, In case of Exit, Area $A = \frac{m}{\rho.C_a}$.Also Blade Height, H=Area/(2.π.r_m). Tip radius at exit, $r_t = r_m + h/2$ and $r_h = r_m - h/2$.

IV. Estimation Number of Stages

The number of stages is found by dividing total temperature rise in all stages by Temperature rise per stage.

Where , $\Delta T_{0s} = \frac{\lambda.U.C_a.(C_{w2} - C_{w1})}{C_p}$.

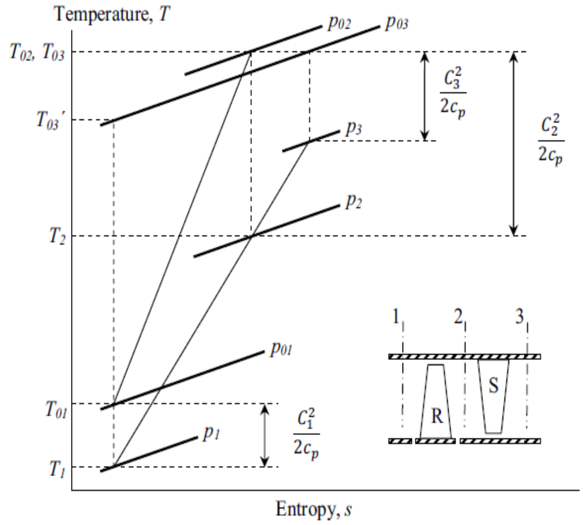


Fig 2 Enthalpy-Entropy diagram

V. Stage by Stage Design

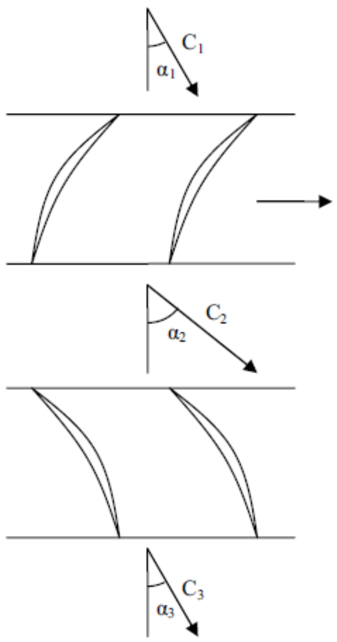
The rotor and stator of a stage are shown in figure2 Note that all angles are referred to the axial velocity vector Ca. Air exits from the previous row of stator blades at angle of $\alpha_1$with absolute velocity C_1.The rotor rows has tangential velocity, and combining the two velocity vectors gives the relative inlet velocity vector W_1 at angle β_1.At rotor row outlet the velocity triangles are similar to those draw for the axial flow pump,and absolute velocity vector C_2 moves into the stator row where the flow direction is changed to C_3 with the absolute velocity C_3The diagram have been drawn showing a large gap between the rotor and stator blades. In practice, the clearance between the rotor and stator is very small.

If the following stage is the same as the preceding one the stage is said to be normal. For a normal stage C_1=C_3 and $\alpha_1 = \alpha_3$. V_2 is less than V_1, showing that diffusion of relative velocity has taken place with some static pressure rise across the rotor blades. The air is turned towards the axial direction by the blade camber and the effective flow area is increased from inlet to outlet, thus causing diffusion to takeplace. Similar diffusion of the absolute velocity takes place in the stator where the absolute velocity vector is again turned towards the axial direction and further pressure rise occurs.

If Rotor inlet is considered as Station 1,
From the set of equations as below, Air inlet angle and Air outlet angle can be determined:
As no inlet guide vane α1 is considered as Zero.

Whirl velocity $C_{\theta 1} = C_{a1}.\tan\alpha 1$

Absolute Velocity $C_1 = \frac{C_{a1}}{\cos\alpha_1}$



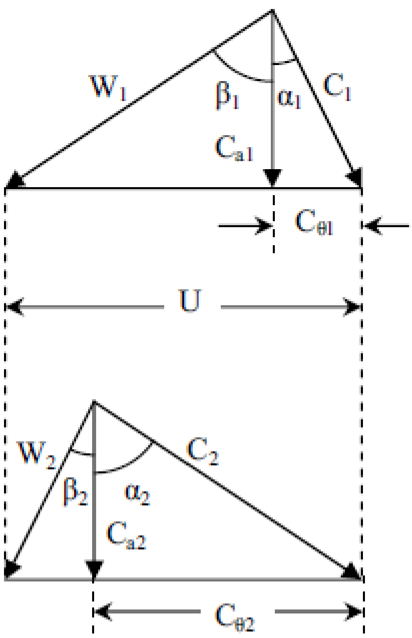


Fig 3 Velocity dia. At inlet and outlet

Exit Air angle $\beta_1 = \tan^{-1}\left[\dfrac{U_m - C_{\theta 1}}{C_{a1}}\right]$

Relative velocity $W_1 = C_{a1}.\cos\beta_1$

If Rotor outlet is considered as Station 2,

Whirl velocity,

$$\Delta C_\theta = \frac{C_p.\Delta T_0}{\lambda.U} = C_{\theta 2} - C_{\theta 1}$$

$$W_2 = \frac{U_m - C_{\theta 2}}{C_a}$$

$$\tan\alpha_2 = \frac{W_2}{C_a}$$

$$\tan\beta_2 = \frac{U_m - \Delta C_{\theta 2}}{C_a}$$

If Stator exit is considered as Station 3,

$$C_3 = \frac{C_a}{\cos\alpha_3}$$

VI. Free Vortex Condition

In Free vortex condition the assumption is made [10]:

Cw. r = Constant. So, $C_{wx} = C_w \times \dfrac{r_m}{r_x}$ and $\tan\alpha_x = \dfrac{C_{wx}}{C_a}$

Same $\tan\beta_x = \dfrac{U_x - C_{wx}}{C_{w1x}}$ .Here subscript 'x' is assumed as subscript 'h' in case of hub radius, and subscript as 't' in case of tip radius.

VII. McKenzie Method

By using McKenzie method for rotor and stator pitch and chord can be found.[9]

$$C''_p = 1 - \left(\frac{V_2}{V_1}\right)^2 \quad and \quad \frac{s}{c} = 9.(0.567 - C''_p)$$

Further,

$$\delta = \left[0.23 \times \left(\frac{2a}{C}\right)^2\right] + \left[0.1 \times \left(\frac{\alpha_2}{50}\right)\right] \text{ and,}$$

$$\theta = \alpha'_1 - \alpha_2 + \delta .$$

So Stagger angle can be found by following equation: Stagger angle= $\xi = \alpha'_1 - \dfrac{\theta}{2}$ .By following the method as above, one can get, Stagger angles of blades of rotor and stator. By assuming H/C ratio, one can get chord(C) of the blade, and also pitch of the blade can be found by assuming Pitch chord ratio.

Number of blades can be found by,

$$n = \frac{2 \times \Pi \times r_m}{S} .$$

VIII. Analysis of Design Using CFD Tools

Analysis of design involves following steps.

A. Creating a Geometry/Mesh
B. Defining the Physics of Model
C. Solving the CFD Problem.
D. Visualizing the Results in the Post – processor

A. Creating a Geometry/Mesh

For generation of geometry SOLID WORKS software is used. The blade profile coordinates are exported the software. With the help of the blade coordinates, base profile is generated. Here, we have assumed same airfoil NACA 65410 from hub to tip for rotor and stator for all stages.

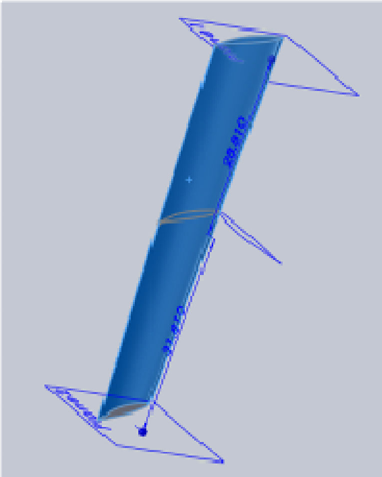


Fig 4 Blade

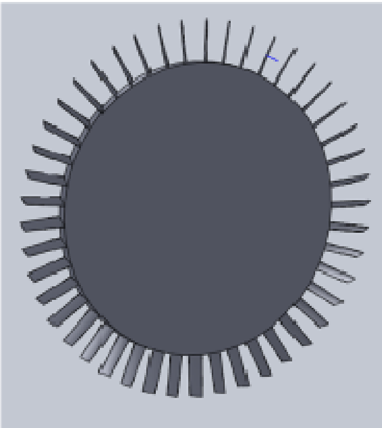


Fig 5 Rotor

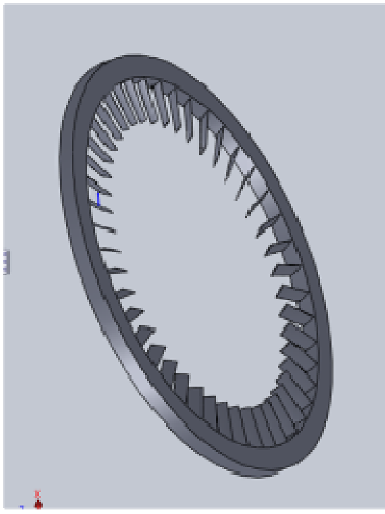


Fig 6 Stator

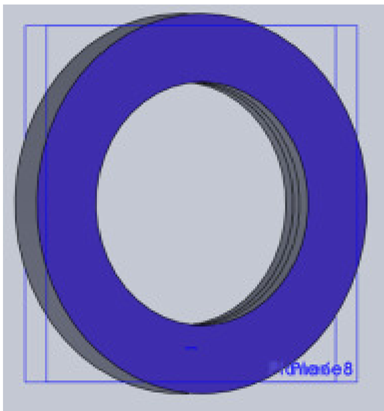


Fig 7 Rotor Cavity

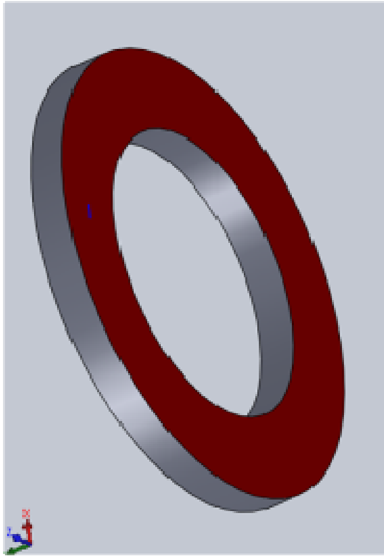


Fig 8 Stator Cavity

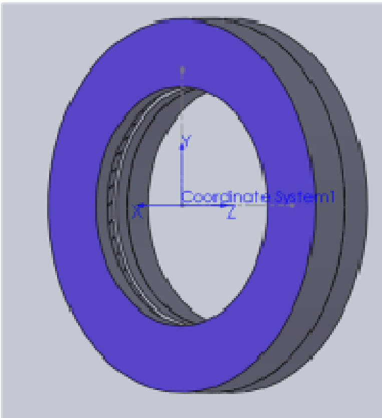


Fig 9 Assembly

Meshing in Hyper mesh 9

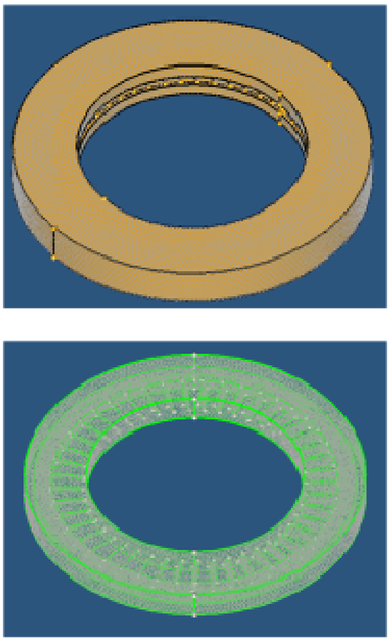


Fig 10 Meshing

Meshing has been carried out by Hyper Mesh. The type of element selected here is Tetrahedral.
In case of Rotor,
No of Nodes = 426980
No of Elements = 1874548
In case of stator,
No of Nodes = 445880
No of Elements = 1952008

B. Defining the Physics of Model

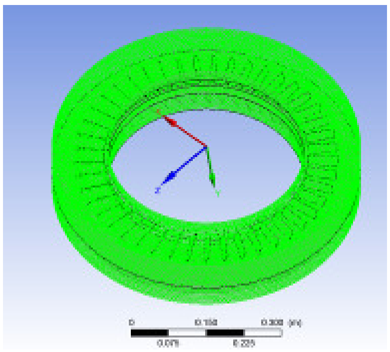
In this step we are defining the physics of Model. This includes specification of type of fluid, defining the domains, Inlet and Outlet Boundary conditions, type of analysis, turbulence model, and heat transfer model etc. The following assumptions are taken for defining physics.

1. Steady state condition
2. No leakage losses
3. Friction between walls and fluid is neglected.

C. Solving the CFD Problem.

CFX-Pre Solver parameters are set as under.

1. Air as an Ideal gas is taken as Working fluid.
2. The "K- ω" model is used for turbulence.
3. Two domain interfaces is used. Rotor Domain is Rotating and Stator Domain is Stationary.



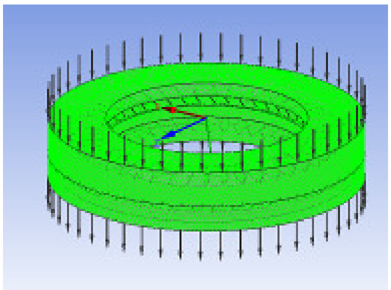


Fig 11 Ansys CFX Boundary Condition

After Setting Boundary conditions and Solver parameters, a definition file is written to be used in ANSYS CFX-Solver Manager for Solution of problem associated with the physical variables. In solver manger the number of iterations and accuracy is specified. A result file is Generated after this operation and results can be analyzed with help of CFD Post. The Procedure described above is repeated for all remaining stages and results can be obtained from the result file.

D. Result of CFD Analysis

Pressure distribution at exit of stage 1

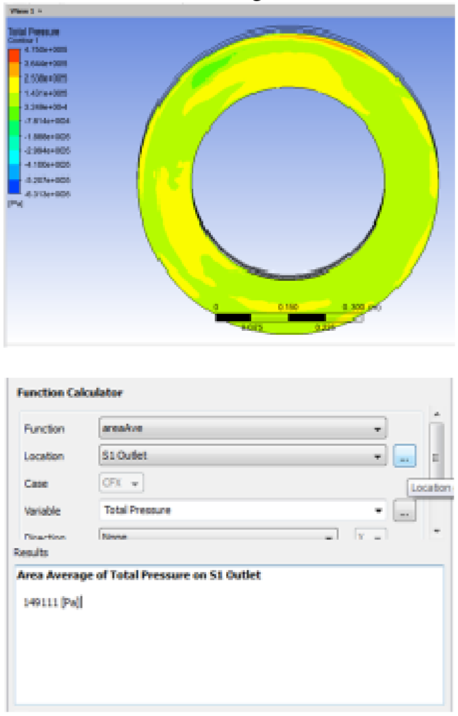


Fig 12 Contour Total Pressure

The theoretical and Analytical value of total Pressure is as below :
Theoretical result - 1.45 bar
Analytical result - 1.49 bar
Variation - 2.07%

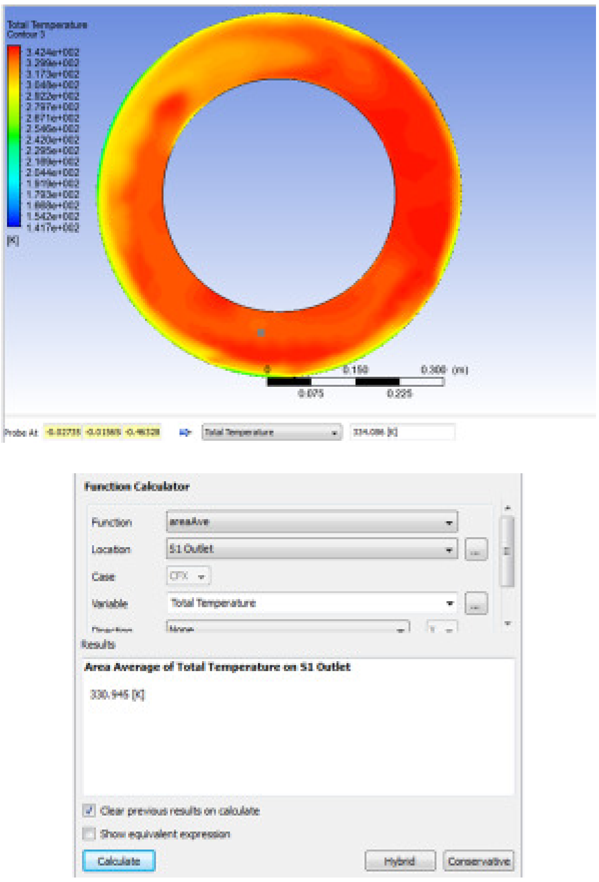


Fig 12 Contour Total Temperature

The theoretical and Analytical value of total temperature is as below:
Theoretical result - 330 K
Analytical result - 333K
Variation - 0.90%

CONCLUSION

With available data i.e. Mass flow rate, Pressure ratio and Pressure at given Altitude. The Blade profile has been generated for both Rotor and Stator analytically. The other possibility of flow separation is also checked with Mach number and Pressure co efficient.

The calculation spread sheet is made so by input the values one can get the required parameters to generate the blade coordinates.While comparing theoretical design results with analytical results, it is observed that the CFD analysis results are in agreement within acceptable range of theoretical results.

REFERENCES

[1] Saravanamutto,HIH,Rogers,GFC och Cohen,H.Gas Turbine Theory fifth Edition,Pearson Prentice Hall,2001.

[2] Nicallas Falck,'Axial Flow Compressor Mean Line Design',Master thesis.Lund University,Sweden,Feb.2008

[3] Hill,P.,& Peterson,C(1992),Mechanics and Thermo dynamics of Propulsion(2nd ed.).Addison Wesley.

[4] Akin Keskin,"Process Integration and Automated Multi-Objective Optimization Supporting Aerodynamic Compressor Design".P.hd Thesis,Brandenburg Technical University Cottbus,2007

[5] Ronald H. Aungier,"Axial Flow Compressor",2nd edition,2003.

[6] UIUC Airfoil Coordinates Database Source dat file,'Compressor'http://airfoiltools.com/airfoil/details?airfoil=naca65410-il'(accessed 10th Nov.2012)

[7] Robin Richard Rajan,Saravanan. T,Rajesh Kumar K.,'Aircraft Design Project1'B.E Thesis,Hindustan College of Engineerng,Anna University,2010.

[8] Meherwan P. Boyce,"Gas turbine Engineering handbook",2nd Addition,Gulf Professional publication,2001

[9] McKenzie, A.B. Axial Flow Fans and Compressors, Ashgate Publishing Limited,1997.

[10] Rizwan r. Ramakdawala, "Preliminary Design Code for an Axial Stage Compressor", M.Sc. Thesis, B.S.,University of Maryland ,College Park,1994.

[11] Mattingly, J. D., Heiser W. H., Daley, D. H., Aircraft Engine Design, AIAA,NewYork, 1987.

[12] Cumpsty, N.A. Compressor Aerodynamics, Krieger Publishing Company, 2

[13] S.L. Dixon, B.Eng., PH.D. Senior Fellow at the University of Liverpool, "Fluid Mechanics, Thermodynamics of Turbo machinery, FOURTH EDITION 1998

[14] V Ganesan,Gas Turbines,3rd Edition,Tata McGraw Hill,New Delhi,2010.

[15] Boyle, R. J., "Navier-Stokes Analysis of Turbine Blade Heat Transfer," ASME Paper 90-GT-42, June 1990

[16] G. Eason, B. Noble, and I. N. Sneddon, “On certain integrals of Lipschitz-Hankel type involving products of Bessel functions,” Phil. Trans. Roy. Soc. London, vol. A247, pp. 529–551, April 1955. *(references)*

[17] J. Clerk Maxwell, A Treatise on Electricity and Magnetism, 3rd ed., vol. 2. Oxford: Clarendon, 1892, pp.68–73.

[18] I. S. Jacobs and C. P. Bean, “Fine particles, thin films and exchange anisotropy,” in Magnetism, vol. III, G. T. Rado and H. Suhl, Eds. New York: Academic, 1963, pp. 271–350.

[19] K. Elissa, “Title of paper if known,” unpublished.

[20] R. Nicole, “Title of paper with only first word capitalized,” J. Name Stand. Abbrev., in press.

[21] Y. Yorozu, M. Hirano, K. Oka, and Y. Tagawa, “Electron spectroscopy studies on magneto-optical media and plastic substrate interface,” IEEE Transl. J. Magn. Japan, vol. 2, pp. 740–741, August 1987 [Digests 9th Annual Conf. Magnetics Japan, p. 301, 1982].

[22] M. Young, The Technical Writer's Handbook. Mill Valley, CA: University Science, 1989.