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COMPUTATIONAL FLUID DYNAMICS ANALYSIS FOR DUCT LINE OF A CENTRIFUGAL COMPRESSOR TO SIMULATE THE DYNAMIC RESPONSE

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Abstract — *The working operation of the transmission station problem of compressor stability and it was a experienced that manifested itself when new compressor and non-return valves were put in to working condition . The model is used to judge the problem and suggest solution in order that the system may operate satisfactory. The first part is discusses the extension and validation of the program so that the gas transmission may be considered.*

Keywords- *Centrifugal Compressor, Duct line, CFD, Valve, Downstream, Fluid, Algorithm, Model, Element*

1.INTRODUCTION

Computational Fluid Dynamics constitutes a new “Third approach” in the philosophical study and development of the whole discipline of fluid dynamics. In the Seventeenth century, the foundations for experimental fluid dynamics were laid. The Eighteenth and Nineteenth centuries saw the gradual development of theoretical fluid dynamics. As a result, throughout most of the Twentieth century, the study and practice of fluid dynamics involved the use of pure theory on the one hand and pure experiment on the other. The learning of fluid dynamics as recently as, say, 1960, involved operating in the “Second-approach world” of theory and experiment. However, the advent of the high speed digital computer combined with the development of accurate numerical algorithms for solving physical problems on these computers has revolutionized the way we study and practice fluid dynamics today. It introduced a fundamentally important new 3rd approach in fluid dynamics – the approach of CFD. Computational Fluid Dynamics (CFD) is a latest tool of fluid analysis software predicts the interaction of a working fluid with its geometrical surroundings and operational environment. Accurately predicting these interactions is highly dependent on understanding the energy loss models embedded within the design code. These loss models dictate how severely performance diminishes due to inherent or sometimes improper geometrical and operational constraints. Such energy losses include skin friction, excessive pressure recovery, airfoil incidence, flow recirculation, and blade tip leakage to name a few.

2. Patching Procedure for the Gas Transmission Station

The section of the pipe work Fitted with the compressor station initially modeled is shown in Fig second case where further downstream ducting was considered.

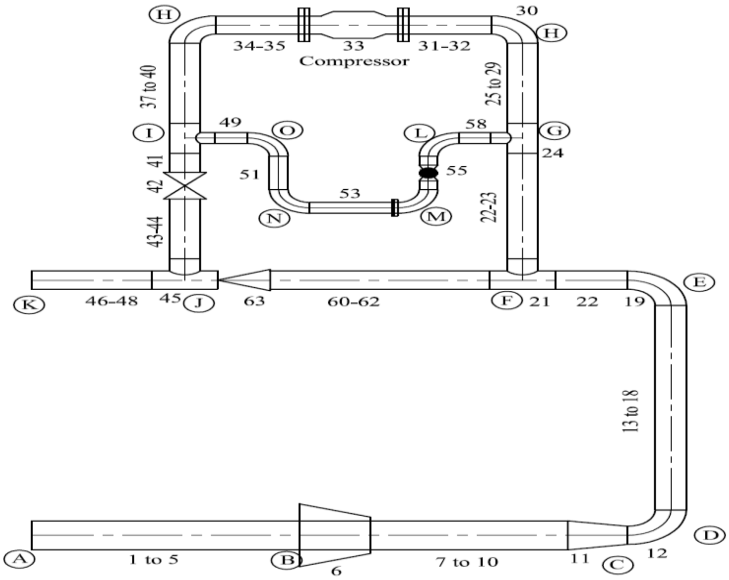


Figure 2.1 Element Distribution

The initial gas circuit (Fig. 2.1) was first Convert into shorter elements by applying the equalized volume and the frequency parameter.. The element volumes, areas and lengths derived were $0.32m^3$, $0.29m^2$ and $1.1m$ respectively. In using such a (geometrical) distribution it was apply that the model was capable of simulating a wave of frequency up to 70 Hz.

This system was then divided into a main system and three sub systems as follows.

1. Main system: System (1) from element 1 to element 48.
2. 1st sub-system System: (2) from element 49 to element (the recycle loop) 58.
3. 2nd sub-system: System (3) element 59.
4. 3rd sub-system: System (4) from element 60 to element 63.

3. System Boundary Conditions

(a) Main system

The infinite volume (nearly) boundary condition was used to define this system entry pressure and system exit mass flow. It was assumed that due to the large (near infinite) volume of the upstream pipe work the acoustic infinite volume upstream of element (1), and upstream of element (2) were identical. Similarly it was assumed the impedance downstream of element 48 and element 47 were similar. Applying the infinite volume boundary condition the system entry pressure and system exit mass flow rate are given by,

$$P_1 = P_2 - \left(\frac{P_3 - P_2}{W_3 - W_2} \right) \times \left(W_2 - W_1 \right)$$

And

$$W_{48} = W_{47} - \left(\frac{W_{46} - W_{47}}{P_{46} - P_{47}} \right) \times \left(P_{47} - P_{48} \right)$$

(b) System (2)

The entry pressure for the system is equal to the mean pressure in the branch element (in Fig. 6.3).

The exit mass flow from the system is to get by the nozzle relationship. The pressure ratio across the nozzle is determined by the ratio of the pressure in the last element of the system to the mean pressure in the branch to which the system is exhausting (i. e. pressure ratio $RP = P_{58} / \overline{P_{25}}$, where $\overline{P_{25}}$ is the mean pressure in element 25). Therefore the system exit mass flow rate is

$$W_{58} = \frac{\gamma A_{noz} P_{58}}{\sqrt{2CpT_{58}}} \left[(RP)^{\frac{2}{-\gamma}} - (RP)^{-\left(\frac{\gamma+1}{\gamma}\right)} \right]^{\frac{1}{2}}$$

(C) Systems (3) and (4)

Initially the non-return valve element 63 (Above Fig.) was closed irrespective of the pressure difference across the element. Therefore the exit mass flow rate leaving systems (3) and (4) were put to zero, The entry pressure to these two systems is determined by the mean pressure in the branch elements 21 and 45.

4. FINITE ELEMENT ANALYSIS:

By applying above boundary conditions to CAD model of duct system in FEA model, following results are obtained.

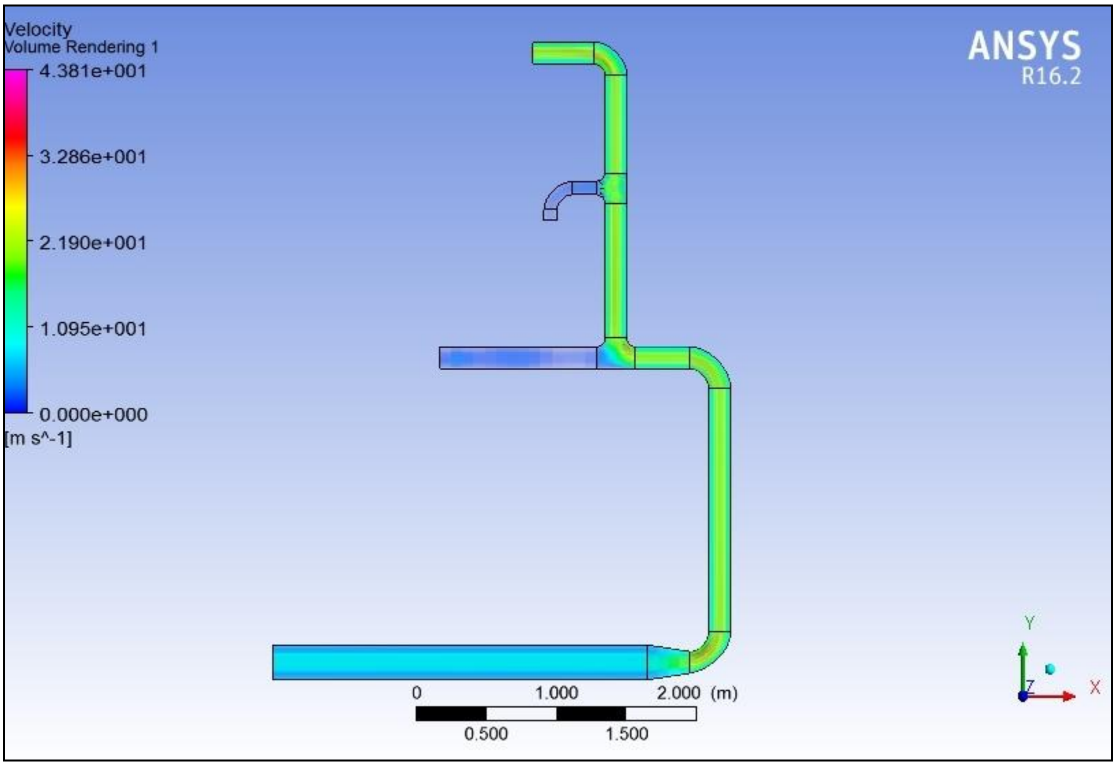


Figure 4.1 Velocity distributions in duct

Figure 4.2 Pressure distributions in duct

5. Simulation Results

The compressor characteristic provided was reproduced on the basis of polytropic head against volume flow rate (Fig. 4.1). The selected speed for the simulation was 6,500 rpm and, for the first set of tests, assumed constant.

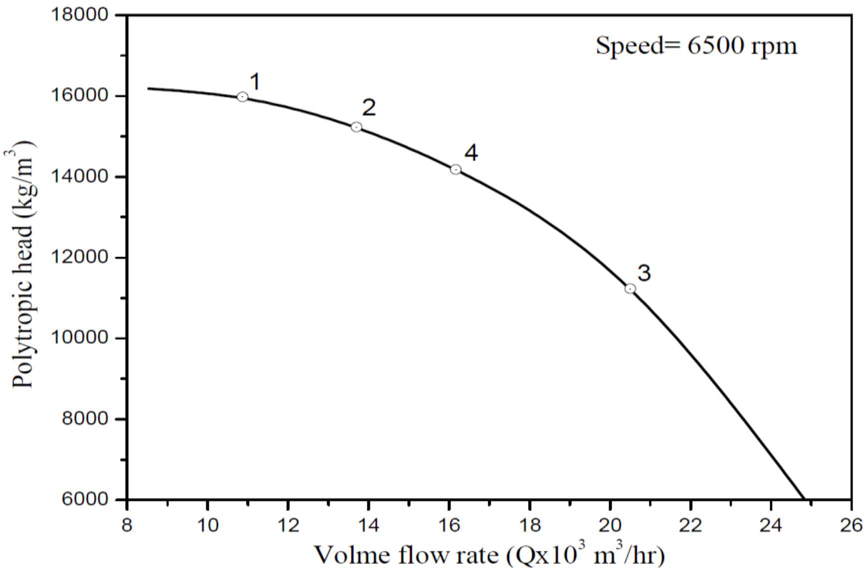


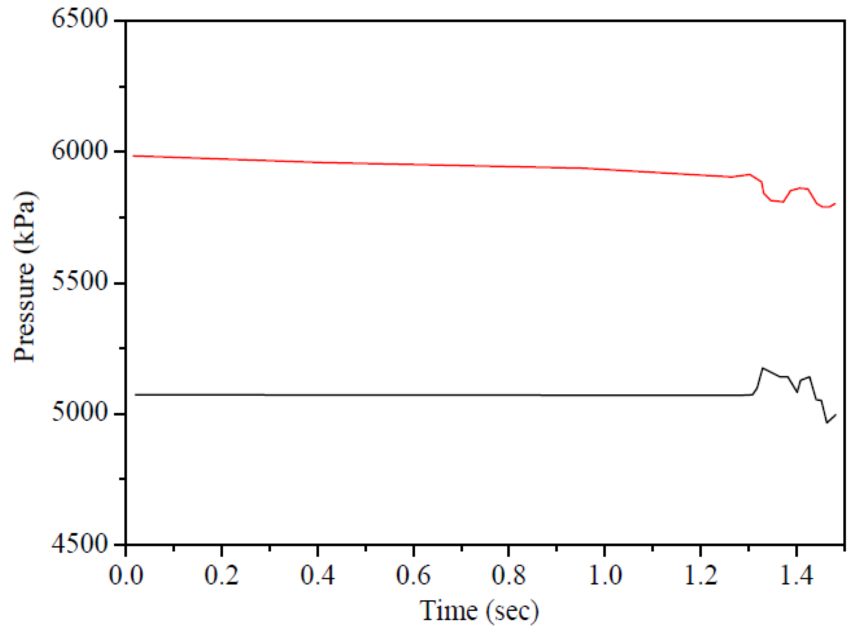
Figure 5.1 Compressor Characteristic Polytropic Head vs. Volume Flow Rate

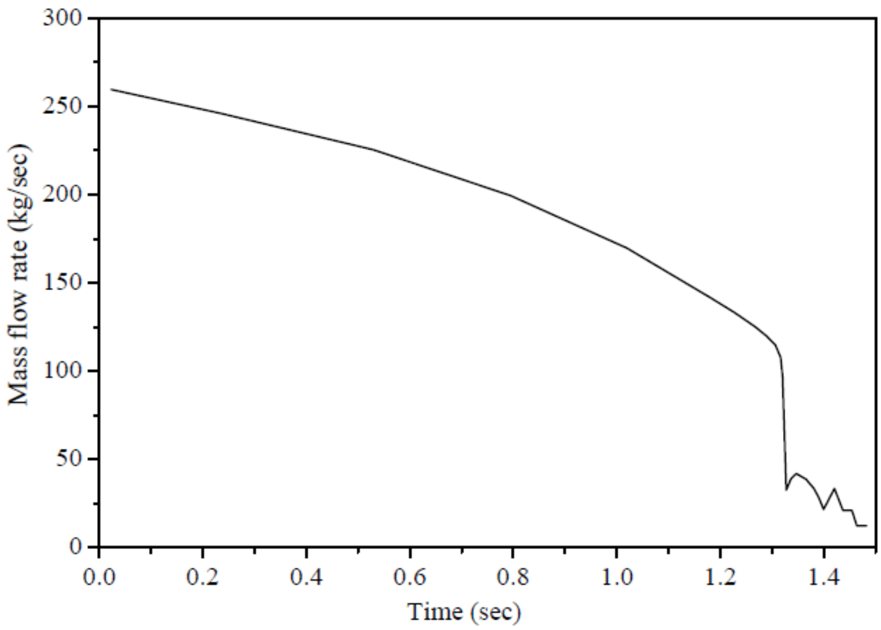
A series of validation tests were conducted.

6. Variable Speed Simulation with Further Duct Work Downstream

Further duct work was added in the simulation, also some of the upstream duct work was removed. This was necessary in order to have sufficient computer memory available. Furthermore the simulation with further duct work downstream was to be of much interest as it was here that transients were likely (in below Fig.) to occur. The patching of the stub pipes shown is similar to the case when the non-return valve was suppose closed irrespective of the pressure difference across it.

Again the deceleration was at 500rpm per second and the speed transient was introduced after 0.02 seconds of steady state working. The simulated results, where the pressure across the compressor element and the corresponding entry mass flow during the transient are shown in below Figs. respectively. The transient running line of the compressor characteristic is shown in below Fig. The compressor outlet pressure drop during the transient is larger when compared with the smaller system below fig. The transient working line on the compressor characteristic is also steeper for this case. This is due to the increased frictional loss that has resulted by the addition of further duct work downstream.





7. CONCLUSION

The Model behavior with respect to stable operation, transients from one stable point to another has been successfully analyzed. Other transients such as perturbations introduced into the system have also been reasonably analyzed. The phenomenon where the compressor becomes less stable as the compressor operating point approaches surge is clearly mentioned. The ability for the transient operating point to leave the steady state characteristic was also demonstrated.

The surge cycle simulations give us excellent results. The fall in the compressor outlet mean pressure, due to a non-linear characteristic was well given. The model also has ability to incorporate anti-surge controllers, and we may be check test the efficiency of control systems used in process plants.

Further the development of the boundary condition for use in an "in finite" system seen to have been very successful.

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