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INFLUENCE OF THE STIFFNESS ON THE FLUID-STRUCTURE INTERACTION PROBLEM IN A RADIAL DIFFUSER WITH MOVING FRONTAL DISK

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Abstract. *Hermetic reciprocating compressors are widely used in small and medium size vapor compression refrigeration systems. One of the main parts of this type of compressor is the automatic valve system used to control the suction and discharge processes. The experimental or analytical/numerical study of these processes is very complex, mainly because the occurrence of the fluid-structure interaction problem during the operation of the valves and due to the geometric complexity of the whole system. Thus, simplified models as radial diffusers have been used to represent the valve system. However, the previous works have further simplified the model by considering radial diffusers with fixed frontal disks. In this work, we have improved the radial diffuser model in which the real fluid-structure interaction problem has been experimentally studied by considering moving frontal disks. The flow was characterized by measuring the distance between the disks, the pressure acting on the frontal disk surface, and the Reynolds number of the flow. Tests for constant mass flow rates were accomplished for Reynolds number in the range of 2,000 to 20,000 in increments of 2,000. Three radial diffuser systems with stiffness equal to 1.2, 1.5 and 2.0 N/mm have been tested for a diameter ratio of 1.3. The effective force and flow areas were calculated for all tests. The results also show the relation between the Reynolds number and equilibrium positions, depending on the stiffness of the system. In addition, the effective force area decreases for lower Reynolds numbers. On the other hand, the discharge coefficient increases for increasing Reynolds numbers and the influence of the system stiffness is not much significant. The results can be used for validating numerical CFD codes developed to solve the FSI problem in refrigeration compressor valves.*

Keywords: *Reciprocation Compressors, Compressor Valves, Radial Diffuser*

1. INTRODUCTION

In the suction and discharge processes inside reciprocating hermetic compressors, the pressure field of the flow acting on the valves controls their opening and closing movement. For this reason, this type of valve is called automatic valve. The flow dynamics through these valves is very complex because involves complex geometries, turbulent flow, pulsatile flow, compressible effects, fast transient processes and with large variation in the local Mach number (Habing, 2005). Besides these features, the flow forces act on the valve body, which is deformed and reacts on the flow, resulting in the so-called Fluid-Structure Interaction (FSI) phenomenon.

Due these features, the radial diffuser has been widely used as a base model in order to understand the main characteristics of this problem. The Figure 1 shows a schematic of the radial diffuser geometry. It is composed of two concentric and parallel disks, the frontal disk with diameter D and the back disk which has a concentric orifice with diameter d. When it is used to model compressor valves, the frontal disk represents the valve reed and the back one the valve seat. The fluid flows axially through the feeding orifice and is deflected by the frontal disk, flowing radially toward the exit through the channel formed between the disks. This channel is called radial diffuser. The distance between the disks, s, is a very important parameter to characterize the flow as is evidenced on the works of the Ferreira *et al.* (1989) and Gasche *et al.* (2015).

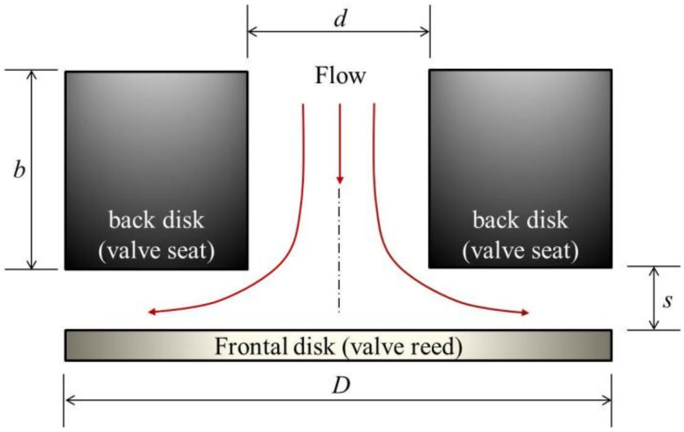


Figure 1. Radial diffuser

The frontal disk of the radial diffuser in static conditions has been widely investigated by different techniques, such as numerical/analytical and experimental. The experimental investigations in this condition can be seen in the following works: Moller (1963), Jackson & Simmons (1965), Wark & Foss (1984), Ferreira & Driessen (1986), Tabakabai & Pollard (1987), Ervin et al. (1989), Gasche *et al.* (1992) and Souto (2002).

The concern to model dynamic behaviour of the frontal disk of the radial diffusers has been already conceived by Costagliola (1950), which can be considered the first success analytical development for predicting the valve behaviour of reciprocating compressors. Costagliola has used the one degree of freedom model for predicting the suction and discharge valve dynamics. This concept has been widely used in many numerical investigations, such as the works of Matos *et al.* (2002), Pereira *et al.* (2008), Pereira *et al.* (2012) and Myrria *et al.* (2012). On the other hand, experimental works regarding fluid-structure interaction are very rare and most of them use invasive techniques in order to obtain the results, like Shiva Prasad & Woollatt (2000) and Ma *et al.* (2008).

The main purpose of this work is to provide experimental results for the FSI problem in a radial diffuser system, which is able to allow analyse the influence of three different stiffness applied on the frontal disk. The tests have been accomplished for a radial diffuser with diameter ratio, D/d, equal to 1.3 and Reynolds number varying from 2,000 to 20,000. These experimental data can be very useful to validate numerical codes developed to solve the FSI problem in refrigeration compressor valves.

2. EXPERIMENTAL APPARATUS

The experimental apparatus shown in Figure 2 has been designed to provide a constant mass flow rate of air in the test section, which is a radial diffuser with diameter ratio equal to $D/d = 1.3$, with $d = 34.9$ mm. The experimental test rig comprises the following: two 500 liter reservoirs connected in parallel, a filter, a pressure control valve, a mass flow rate control valve, a Coriolis mass flow meter, a flexible tube, an aluminum tube containing two fine nets at the inlet, and finally the test section, as can be seen in the sketch of the Figure 3. The test section was installed on a large concrete block to avoid vibrations transmitted by the external environment. As the distance between the disks, s, is the most critical parameter of the flow, the test section was designed to have high stiffness in order to reduce its measurement uncertainty.

In a typical test, the air in the reservoir flows to the filter, where it is cleaned and dehumidified, and then to the pressure control valve, which ensure a constant downstream pressure. Therefore, the mass flow rate at the test section is maintained constant in the desired set value, which is given by the mass flow rate control valve installed downstream of the pressure control valve. Then, the air flows through the Coriolis mass flow meter, flexible tube, and through a 2 m aluminum tube, which ensures a completely developed flow at the inlet of the test section. Two fine nets were installed at inlet of the aluminum tube in order to regularize the flow velocity profile.

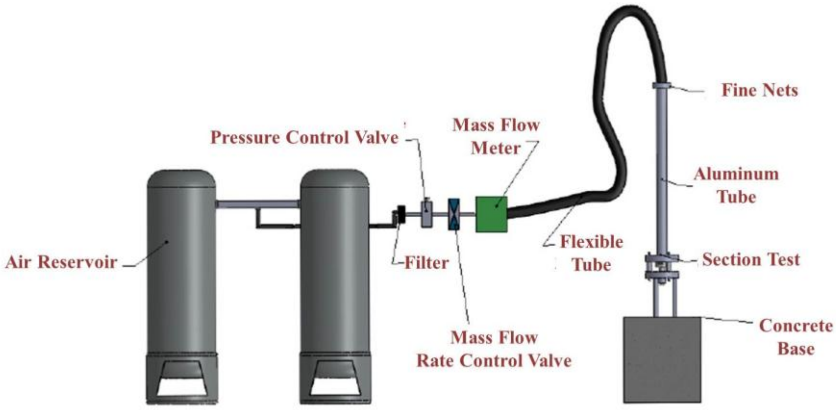
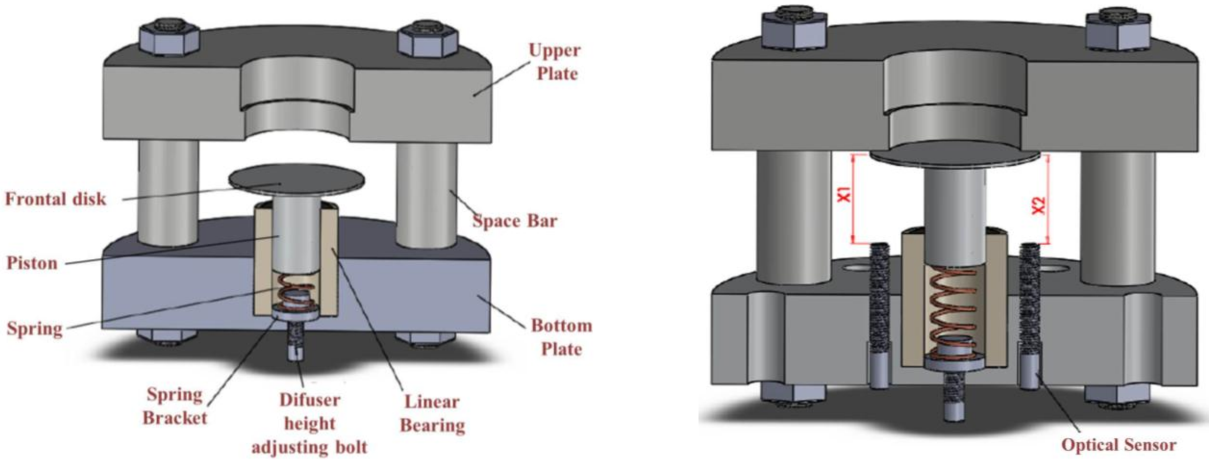


Figure 2. Sketch of the experimental apparatus

Basically, the test section consists of two stiff circular plates. The bottom plate is fixed in the concrete block through three spacer bars and is used as both a support for the frontal disk and the optical sensors used to measure the distance between the disks, *s*. The upper plate is also fixed to bottom plate through three similar spacer bars, displaced 120° from each other and is used as the back disk for the radial diffuser geometry. As can be seen in Figure 3, the frontal disk is installed within a linear bearing which is used to reduce the friction forces during the frontal disk movement. A spring is installed within the piston and the whole kit (radial diffuser/piston/spring) can be moved by an adjusting bolt, without adding any additional stiffness to the kit. Thus, this adjusting bolt is used to position the frontal disk on its start position that is leaned against the upper plate.



(a) (b)

Figure 3. Test section

The distance between the frontal disk and the upper plate was measured by two optical sensors installed 180° from each other, allowing to verify the quality of the parallelism between them, The manufacturer's calibration chart for a standard reflexive surface (retro tape) glued on the frontal disk surface provided an uncertainty of ±1µm, and it was used to convert the signals.

The other parameters measured during the experiments were the temperature and mass flow rate of the flow, upstream pressure (at the inlet of the section test) and the atmospheric pressure. The mass flow rate was measured through a Coriolis mass flow meter with operating range of 5.0 kg/min and maximum uncertainty equal to ±0.2% of the reading. The temperature of the flow was measured by a PT100 sensor (as part of the Coriolis mass flow meter) with an estimate uncertainty of ±0.1°C. The upstream pressure was measured by an inductive pressure transmitter with 1 bar full scale (FS) and uncertainty of ±0.1% FS. Finally, the atmospheric pressure was measured by a barometer with 0.05 kPa resolution. All analogic signals were converted to digital signals and wired to a computer where they were processed by a LabView program.

3. EXPERIMENTAL PROCEDURE

A typical test starts with some important settings in the experimental apparatus, of which the initial position of the frontal disk is the most important to ensure accurate experimental results. The first step is to displace upwards the frontal disk until it remains in contact with the upper plate, as shown in Figure 3(b), without adding any additional stiffness to the system. In this position, the distance between the disks is considered to be equal to zero. The readings of both optical sensors are considered as references for accounting the distance s, that is, $s_1 = 0$ for x_1 and $s_2 = 0$ for x_2. Then, when the frontal disk is moved (displacement x'_1 for sensor 1 and x'_2 for sensor 2) due to the pressure of the flow, the user interface of the LabView can provide the following displacements: $s_1 = x_1 - x'_1$ and $s_2 = x_2 - x'_2$, respectively. The average value from s_1 and s_2 during the test is taken to be the distance between the disks, s.

After adjusting the initial position of the frontal disk, the next step is to set the pressure downstream the pressure control valve in order to guarantee a constant mass flow rate in the test. This pressure must be always lower than the pressure within the reservoirs, usually from 8 to 9 bar. Then, the mass flow rate control valve is opened to provide the desired mass flow rate for the test, which is characterized by the Reynolds number of the flow, given by:

$$\mathrm{Re} = \frac{\rho V d}{\mu} = \frac{4\dot{m}}{\pi \mu d} \tag{1}$$

where ρ and μ are the fluid properties, density and dynamic viscosity, respectively. The average velocity of the flow is V and $\dot{m}$ is the mass flow rate of the flow.

The density of the air was calculated by using the equation of state for ideal gas, considering the measured values of the upstream pressure and temperature of the flow. The dynamic viscosity of the air was calculated by using equation adapted from Bean (1971) in function of absolute temperature given by:

$$\mu = \left(0.872 + 7.029\times10^{-2}T - 3.81\times10^{-5}T^2\right)\times10^{-6} \tag{2}$$

Once the steady state regime was reached, the distance between the disks, s, the upstream pressure, p_u, and the Reynolds number of the flow, *Re*, were measured for Reynolds numbers varying from 2,000 to 20,000. This is considered to be the first run. Then, the frontal disk was displaced to a random position and repositioned to its initial position again, that is, in contact with the upper plate, and the new measurements were taken for the same Reynolds numbers like in the first run. The same procedure was made five times in order to verify the repeatability of the data.

3.1 Spring Calibration

In order to obtain the stiffness (k) of the springs the same test section presented in Figure 3(b) was used, but without the aluminum tube. First of all, the frontal disk was set on its initial position following the same procedure used in the tests. Then, a piece of metal with known mass was placed on the frontal disk and the displacement was measured by both optical sensors. This procedure was repeated for increasing masses. For each spring the calibration procedure was repeated 10 times. The graph in Figure 4 depicts the force [N] of those masses as a function of the displacements of the frontal disk [mm]. The frontal disk displacement was taken as an average of the both sensor measures. Using these results one can say that the maximum misalignment of the frontal disk is about 0.08 mm, which is the maximum difference obtained by both optical sensors for the whole range of operation. The maximum standard uncertainty for displacement measured was of 0.01mm considering all calibration tests. As can be seen in Figure 4, linear trend lines were plotted for the three springs. The stiffness of the system was taken as the slope of those straight lines. The stiffness of the springs were 2.0043, 1.5050, and 1.1886 N/mm.

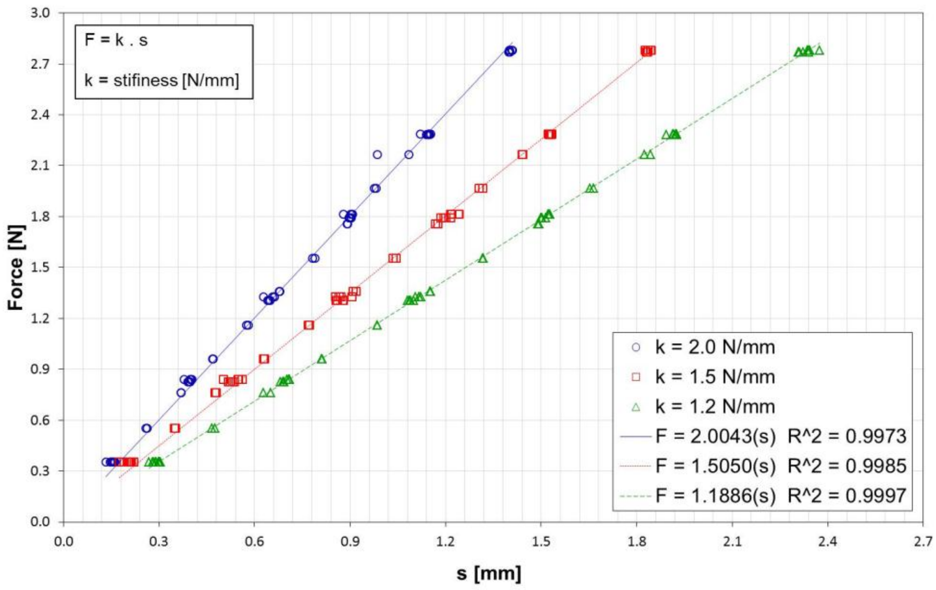


Figure 4. Calibration of the radial diffuser system

4. RESULTS AND DISCUSSION

The tests have been accomplished for Reynolds number ranging from 2,000 to 20,000 in increments of 2,000. For each Reynolds number, the test was repeated five times. The average of these data measured was plotted in the graph and the maximum standard uncertainty was calculated for each parameter.

Figure 5 shows the behavior of the dimensionless equilibrium position of the frontal disk, s/d, as a function of the Reynolds number. The maximum standard uncertainties for s/d measures were reached 0.4, 2.9 and 1.3% for tests with stiffness of 2.0, 1.5 and 1.2 N/mm, respectively. For a constant stiffness, the s/d increases with increment of the Reynolds numbers, which means that the force caused by the flow also increases, as can be seen in Figure 6. The graph in Figure 5 also shows that the Reynolds number necessary to cause the same equilibrium position is larger as larger the stiffness of the system, which means that the force of the flow to provide the same equilibrium position also increases for increasing stiffness of the system. In addition, we can notice that the equilibrium position reached by the frontal disk decreases for increasing stiffness of the system for the same Reynolds number. This behavior was expected, because the equilibrium position is inversely proportional to the stiffness of the system.

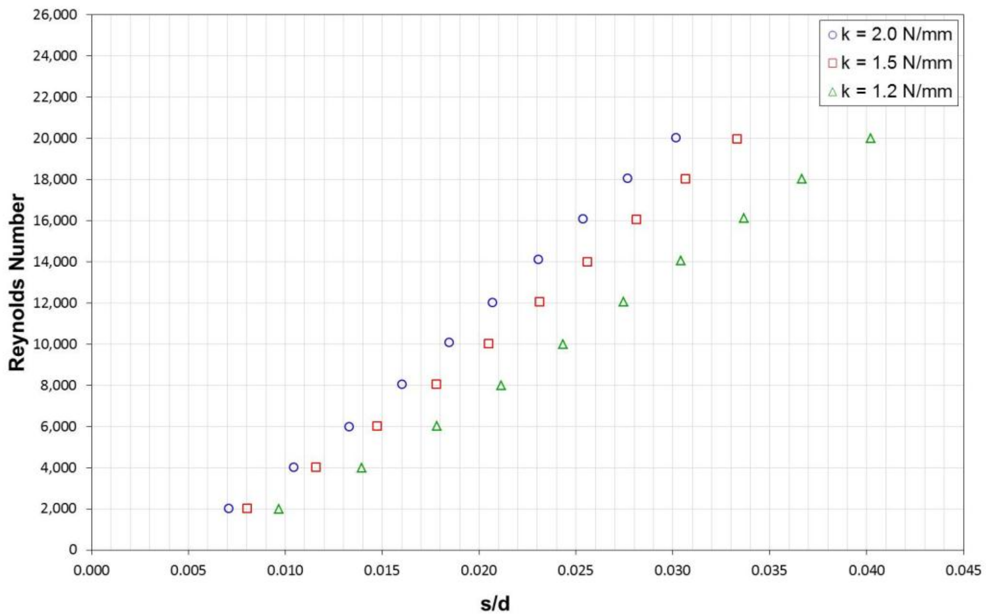


Figure 5. Dimensionless equilibrium position of the frontal disk as a function of the Reynolds number

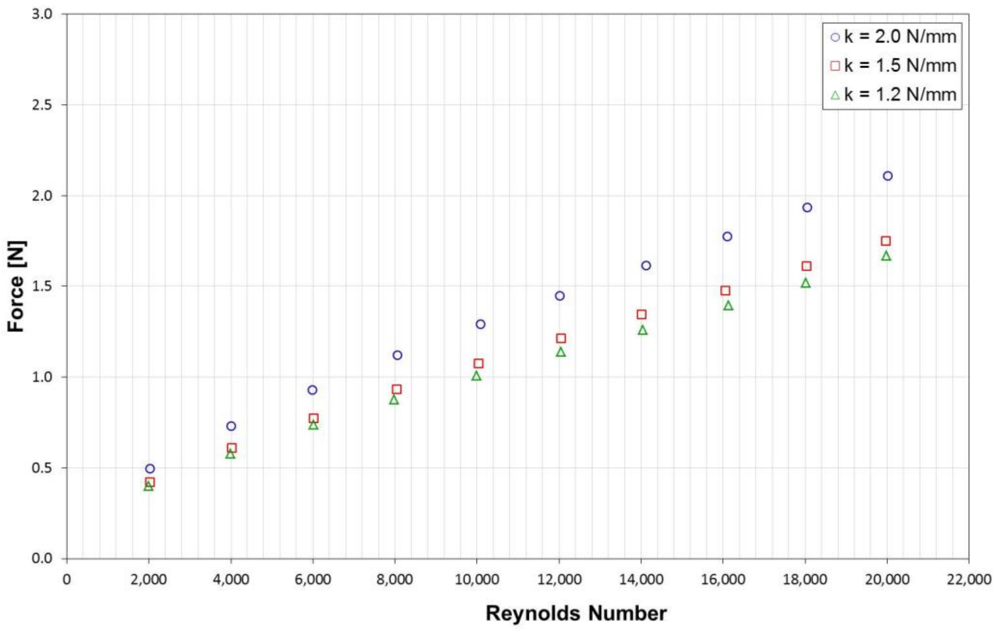


Figure 6. Force of the flow on the frontal disk as a function of the Reynolds number

Figure 6 allow us to analyze the behavior of the force of the flow as a function of the Reynolds number for the three stiffness of the system. As expected, the larger the Reynolds numbers means the larger the force of the flow, if the stiffness is fixed. The graph shows that the systems with lower stiffness would require larger *s/d*, and thus Re, in order to maintain the same force of the flow.

The effective flow and force areas are important parameters for the numerical simulation of reciprocating hermetic compressors, which is a fundamental step in the optimization process of compressors (Ferreira and Driessen, 1986). The effective force area is defined as the ratio between the force of the flow acting on the frontal disk and the pressure drop through the flow, $A_F = F/\Delta P_v$. This parameter can be understood as how efficiently a differential pressure is to open a valve. Using this parameter, compressor designers can estimate the force acting on the valve by knowing only the pressure drop through, without solving the complex flow taking place in it. The dimensionless effective force area, which is defined as the effective force area divided by the feeding orifice area, A_F*, is used in this work. The effective flow area is defined by Equation (3) as the ratio between the actual mass flow rate of the flow and the product of the velocity by the density of an isentropic flow through the feeding orifice. This parameter is used by compressor designers to determine the mass flow rate through the valve. The dimensionless effective flow area is used to analyze the results. It is also used as discharge coefficient of the flow, C_d, which is defined as the effective flow area divided by the feeding orifice area.

$$A_v = \frac{\dot{m}}{P_u\sqrt{\dfrac{2k}{(k-1)RT_u}\left(r^{2/k} - r^{k+1/k}\right)}} \tag{3}$$

P_u and T_u are the upstream absolute pressure and the upstream absolute temperature, respectively. The R is the gas constant, k the gas relation of specific heats and r is the relation between atmospheric pressure with upstream absolute pressure.

Figure 7 shows the dimensionless effective force area behavior for different flow regimes. The maximum standard uncertainty for A_F* measures reached 1.94, 2.42 and 1.35% for tests with stiffness of 2.0, 1.5 and 1.2 N/mm, respectively. As can be observed, for the lower Reynolds number, the reduction of the A_F* is stronger for the system with stiffness of 2.0 and 1.2 N/mm than 1.5 N/mm. Figure 8 shows the discharge coefficient increasing as the Reynolds number of the flow increases. As can be seen, the influence of the stiffness of the system is small. The maximum standard uncertainties for discharge coefficient measures were reached 3.19, 0.61 and 0.47% for tests with stiffness of 2.0, 1.5 and 1.2 N/mm, respectively.

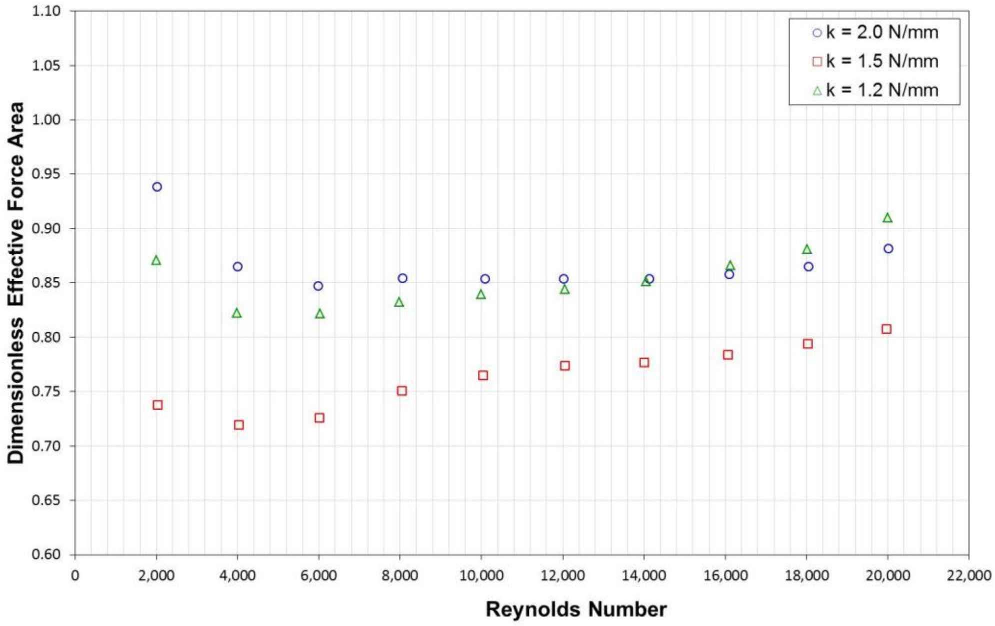


Figure 7. Dimensionless effective force area as a function of the Reynolds number

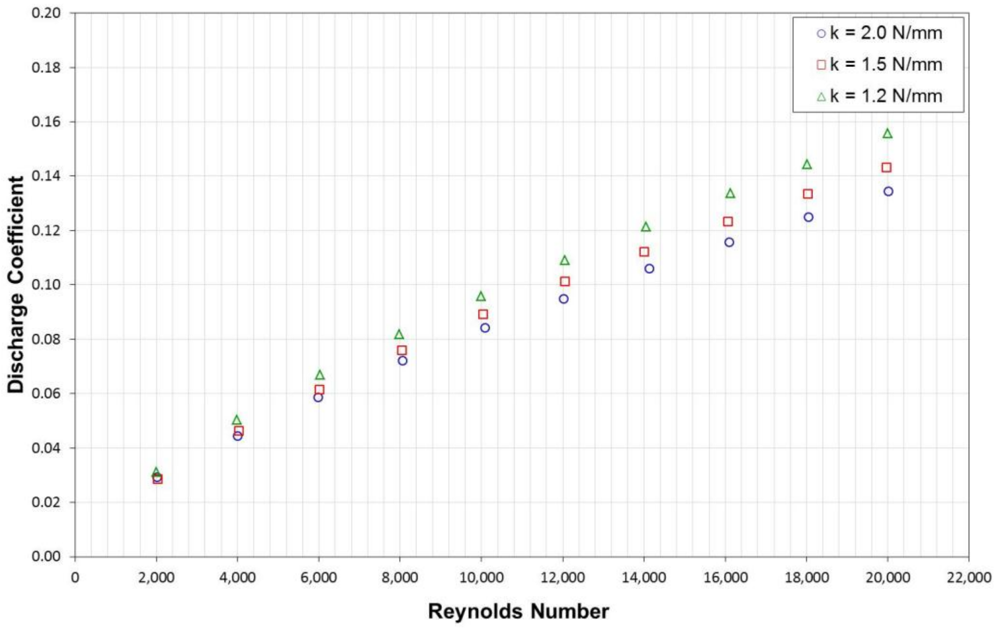


Figure 8. Discharge coefficient as a function of the Reynolds number

5. CONCLUSSIONS

An experimental investigation of the fluid-structure interaction problem in the flow through a radial diffuser with diameter ratio equal to 1.3 has been carried out in order to evaluate the influence of stiffness of the system on the equilibrium position of the frontal disk. In addition, the effective force and flow areas have been also evaluated. The results show the behaviour of the system equilibrium position and valve body force as a function of the Reynolds number. These results allows us to conclude that the systems with lower stiffness would require larger equilibrium positions and thus Reynolds number, in order to maintain the same force of the flow. Evaluating the stiffness effect in the system for the effective force area, we conclude that it has similar behaviour. However, for lower Reynolds number, the effective force area reduce stronger for system with k = 2.0 and k = 1.2 N/mm than k = 1.5 N/mm. On the contrary, the discharge coefficient increases as the Reynolds numbers is enhanced and the influence of the stiffness of the system is not much significant. The present experimental results can be useful for validating numerical codes developed to solve the FSI problem in refrigeration compressor valves.

6. ACKNOWLEDGEMENTS

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