INFLUENCE OF THE HEAT TRANSFER ON THE PRESSURE DISTRIBUTION ON THE FRONTAL DISK OF A RADIAL DIFFUSER REPRESENTING A REFRIGERATION COMPRESSOR VALVE

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Abstract. *In refrigeration system compressors, the suction and blowout valves are responsible for, respectively, the retention and passage of the refrigerant fluid from the suction chamber to the cylinder and from the cylinder to the blowout chamber. Valves with small opening and closing times, low back pressure and with low return of the gas are required to increase the efficiency of the compressor. As the opening and closing of the valves are caused by the force of the gas flow, the understanding of the flow through the valve is of fundamental importance. The numerical simulation of the flow is an efficient method to perform this task. Due to the complex geometry usually found in this type of valve, simplified geometries have been used to represent the valve, particularly the radial diffuser geometry. Most researchers have performed the numerical simulation of the isothermal flow through the radial diffuser. The study of the heat transfer in this geometry is rare in the literature. In the present work, an analysis of the non-isothermal, incompressible, and laminar flow through a radial diffuser representing the valve is performed numerically using the Finite Volume Methodology. The SIMPLE algorithm applied to a staggered mesh was used for solving the pressure-velocity coupling problem. The power-law scheme was used as the interpolation function for the convective-diffusive terms, and the TDMA algorithm was used to solve the systems of algebraic equations. The main goal of the work is to investigate the effect of the heat transfer process on the pressure profile on the frontal disk of the radial diffuser. The results showed that the dimensionless pressure in the stagnation region of the flow increases for increasing surface temperatures, mainly for low Reynolds numbers.*

Keywords: *Heat transfer, compressor valve, refrigeration, radial diffuser*

1. INTRODUCTION

The compression process in refrigeration reciprocating compressors is developed by the linear alternate piston displacement, as can be seen schematically in Fig. 1. The suction and discharge valves are responsible for the refrigerant gas retention and flow from the suction chambers to the compressor cylinder and from cylinder to the discharge chamber.

The appropriate design of the valve system is of fundamental importance for elevating the compressor efficiency. Valves with small opening and closing times, small pressure drops, and those restricting the gas backflow are required. As the valve opening and closing movements are caused by the refrigerant gas flow force, a comprehensive understanding of the flow through the valve is essential in order to enhance the compressor efficiency.

The numerical simulation of this type of flow is an efficient method to perform this task. Due to the complex geometry usually found in this type of valve, simplified geometries have been used to represent the valve, particularly the radial diffuser geometry shown in Fig. 2. The fluid enters the passage orifice with diameter d and, after being deflected by the frontal disk of diameter D, flows through the gap between the frontal disk (reed) and the valve seat.

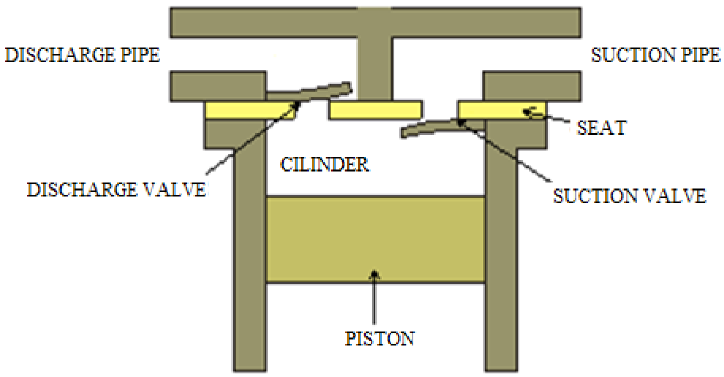

Figure 1. Schematic representation of the valve system in hermetic compressors.

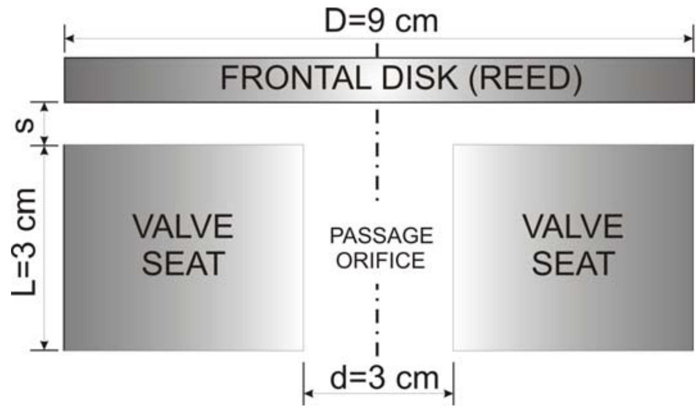

Figure 2. Radial diffuser used for modeling the compressor valve.

Souto (2002) has presented an extensive review on radial diffuser flows with applications in compressor valves. Works on analytical solutions for laminar, incompressible, and steady flows have been performed by Woolard (1957), Livesey (1960), Savage (1964) and Killmann (1972), while Ishizawa *et al.* (1987) have presented analytical solution for the incompressible, laminar and unsteady flows. Numerical solution for laminar and incompressible flows has been accomplished by Hayashi *et al.* (1975), Raal (1978), Piechna and Meier (1986), Ferreira *et al.* (1987), Deschamps *et al.* (1989), Ferreira *et al.* (1989), Langer *et al.* (1990), Gasche (1992) and Possamai *et al.* (1995). On the other hand, numerical solutions for turbulent and incompressible flows have been obtained by Deschamps *et al.* (1988) and Deschamps *et al.* (1996). Experimental works on the subject have been developed by Jackson and Simmons (1965), Wark and Foss (1984), Ferreira and Driessen (1986), Tabatabai and Pollard (1987), Ervin *et al.* (1989), and Gasche (1992).

Some researchers also have obtained numerical solutions for laminar and incompressible flows including the frontal disk dynamics: Lopes (1994), Matos *et al.* (1999), Matos *et al.* (2000), Matos *et al.* (2001), and Salinas-Casanova (2001). Souto (2002) has experimentally studied turbulent flows through the radial diffuser considering steady and unsteady flows.

The literature review reveals that the most works have considered the flow through the radial diffuser as isothermal flow. Prata *et al.* (1995) have been one of the few researchers that have included the heat transfer process in the radial diffuser flow modeling. The local Nusselt number profile on the frontal disk surface has been obtained experimentally using the naphthalene sublimation technique. The experimental data have been compared with numerical results obtained by using the Finite Volume methodology.

The objective of this work is to solve numerically the flow through the radial diffuser including the heat transfer process in order to verify the influence of the heat transfer on the pressure distribution on the frontal disk surface.

2. METODOLOGY

The governing equations for the incompressible flow of a Newtonian fluid through the radial diffuser (mass conservation, momentum and energy equations) are given by:

$$\vec{\nabla} \bullet \vec{V} = 0 \tag{1}$$

$$\rho\left[\frac{\partial \vec{V}}{\partial t} + \vec{V} \bullet \vec{\nabla}\vec{V}\right] = -\vec{\nabla}p + \vec{\nabla} \bullet \left[\mu\left(\vec{\nabla}\vec{V} + \vec{\nabla}^T\vec{V}\right)\right] \tag{2}$$

$$\rho C_p\left[\frac{\partial T}{\partial t} + \vec{\nabla} \bullet (\vec{V}\,T)\right] = \vec{\nabla} \bullet (k\vec{\nabla}T) \tag{3}$$

where ρ is the fluid density, μ is the absolute viscosity, C_p is the specific heat at constant pressure, k is the thermal conductivity, p is the pressure, V is the velocity vector, and T is the temperature. The necessary boundary conditions to solve the problem are indicated in Fig. 3.

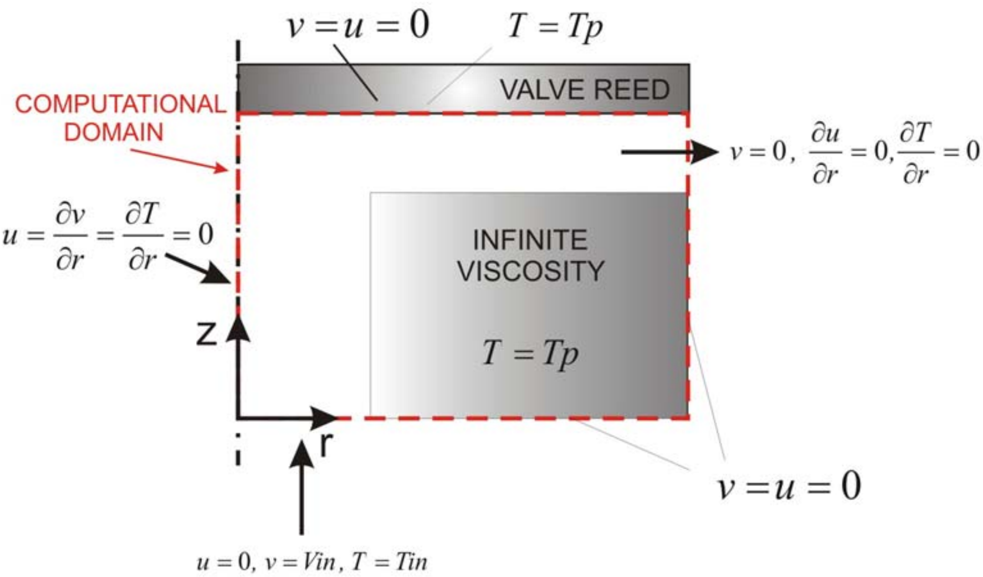

Figure 3. Boundary conditions.

The influence of the heat transfer process in the fluid flow is considered through the variation of the absolute viscosity with temperature, which is given by the equation proposed by the following equation (The U.S. Standard Atmosphere, 1962):

$$\mu = \frac{bT^{\frac{3}{2}}}{S+T} \tag{4}$$

where b=1.458 10^{-6} kg/(m s $K^{1/2}$) and S=110.4 K

The governing equations and boundary conditions are discretised using the Finite Volume method. The pressure-velocity coupling is treated by the SIMPLE – Semi-Implicit Method for Pressure Linked Equations algorithm, while the algebraic equation systems are solved by the TDMA – Tri-Diagonal Matrix Algorithm. The power-law relation is used as interpolation scheme for the advective/diffusive terms of the Navier-Stokes equations. The final results were obtained by using a non-uniform mesh composed by 240 volumes in the z direction and 360 volumes in the r direction, which results in a total of 86400 volumes in the whole domain. All numerical results were considered converged for mass conservation residue lower than 10^{-13}.

3. RESULTS

Before obtaining the final results, a grid convergence test was performed. Figure 4 shows the results for the worse case for four different grids. As can be seen, the difference between the results of the two finer grids is very small. Therefore, the finest grid (240x360) was used to generate the final results.

In order to validate the computational procedure, dimensionless pressure profiles acting on the frontal disk surface for isothermal flow were compared with experimental data obtained by Gasche (1992) for two flow configurations. Figure 5 shows the comparison for Re=1491 and s/d=0.025 and Fig. 6 presents similar results for Re=2033 and s/d=0.02. The dimensionless pressure, p_{adm}, is defined as:

$$P_{adm} = P/(\frac{\rho v_{in}^2}{2}) \tag{5}$$

and the Reynolds number is given by:

$$\mathrm{Re} = \frac{\rho v_{in} d}{\mu} \tag{6}$$

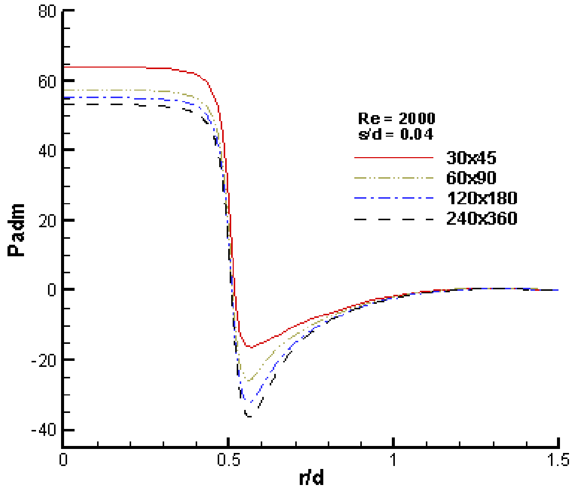

Figure 4. Grid convergence test.

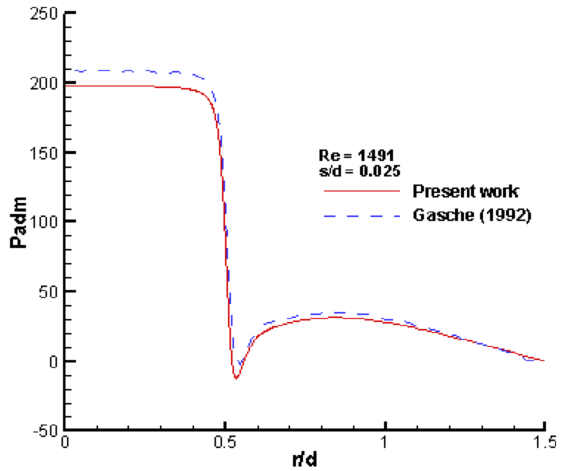

Figure 5. Comparison between the numerical and experimental results for Re=1491 and s/d=0.025

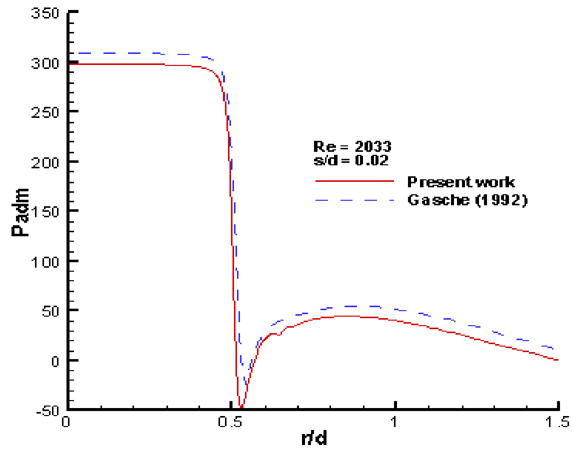

Figure 6. Comparison between the numerical and experimental results for Re=2033 and s/d=0.02

As can be noted in those figures, the numerical results agree very well with the experimental data, indicating that the numerical procedure can be considered validated at least for isothermal flows.

After validating the numerical procedure, one can obtain numerical results for other flow configurations, including the heat transfer process. Figures 7 to 14 present numerical results in order to verify the influence of heat transfer in the pressure profile acting on the frontal disk. The inlet temperature of the flow is zero Celsius degree for all cases. The influence of three surface temperatures (T_p=70ºC, 100ºC and 130ºC) is studied for two Reynolds number (Re=600 and 2000), considering four gaps between the frontal disk and the valve seat (s/d=0.005, 0.01, 0.02, and 0.04).

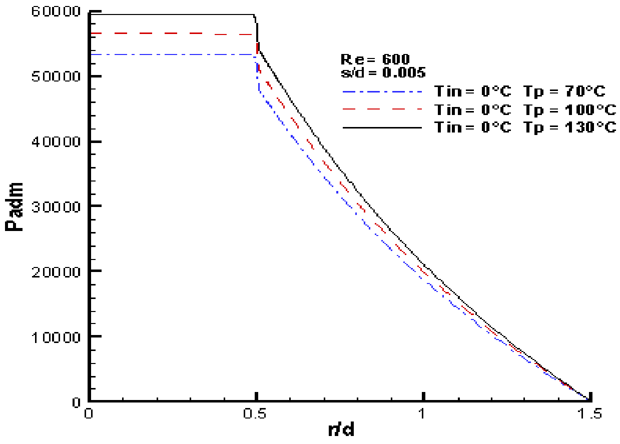

Figure 7. Pressure profile for s/d=0.005 and Re=600.

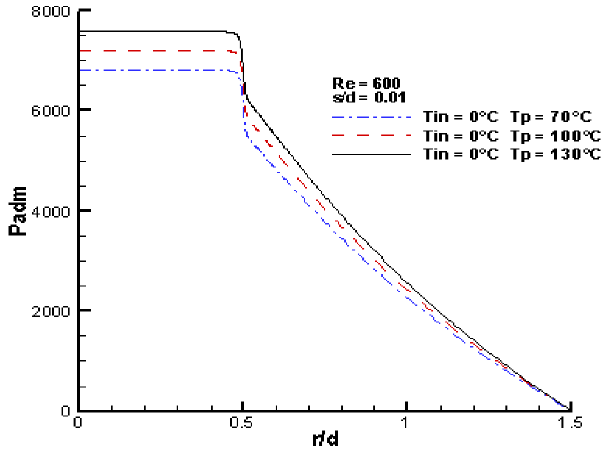

Figure 8. Pressure profile for s/d=0.01 and Re=600.

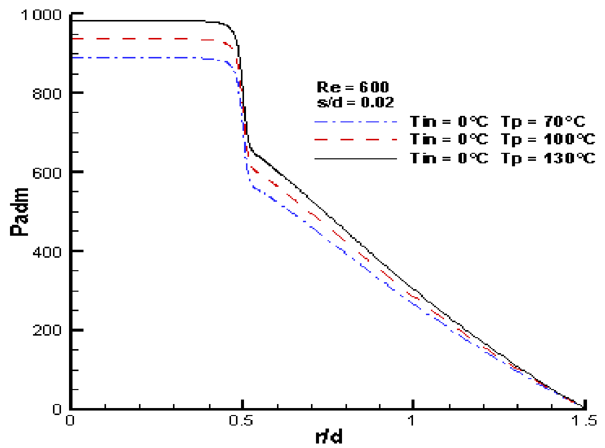

Figure 9. Pressure profile for s/d=0.02 and Re=600.

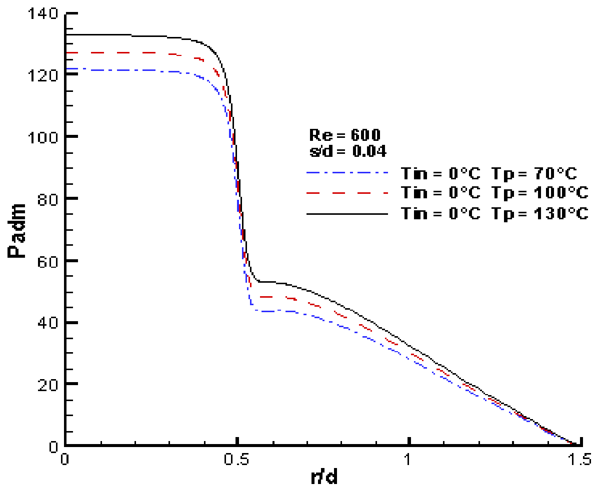

Figure 10. Pressure profile for s/d=0.04 and Re=600.

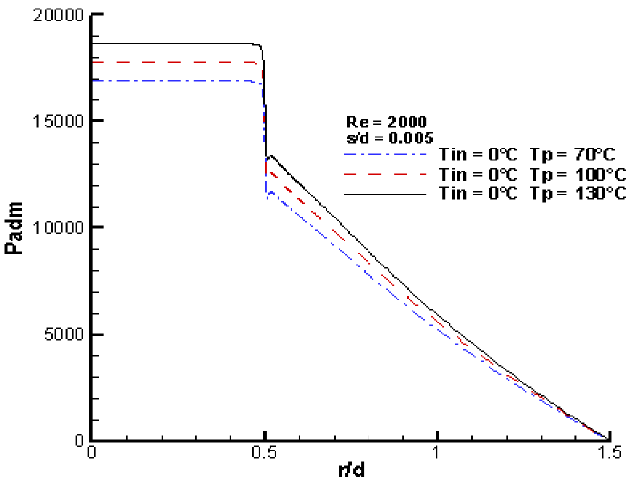

Figure 11. Pressure profile for s/d=0.005 and Re=2000.

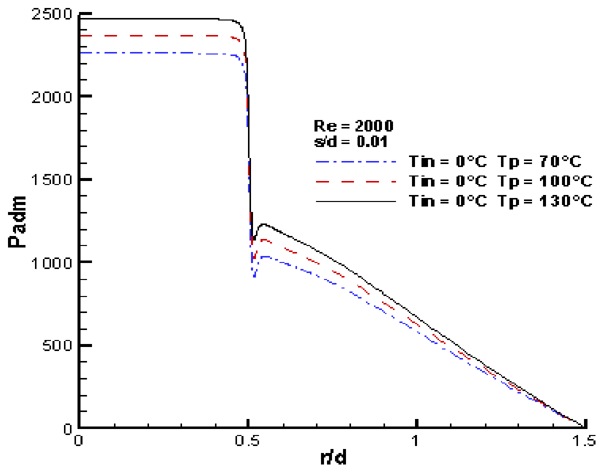

Figure 12. Pressure profile for s/d=0.01 and Re=2000.

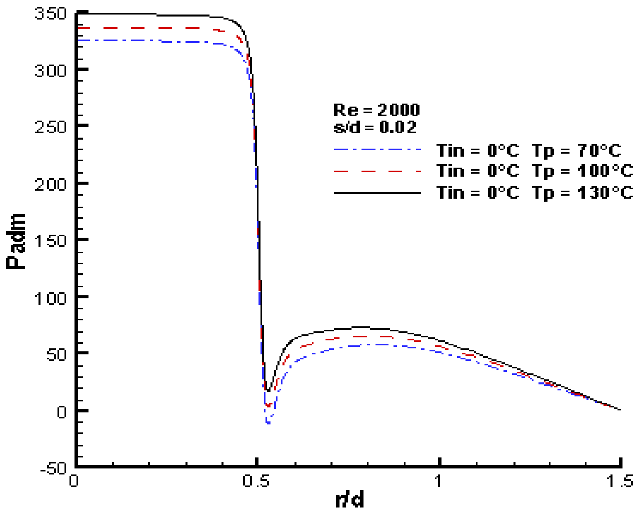

Figure 13. Pressure profile for s/d=0.02 and Re=2000.

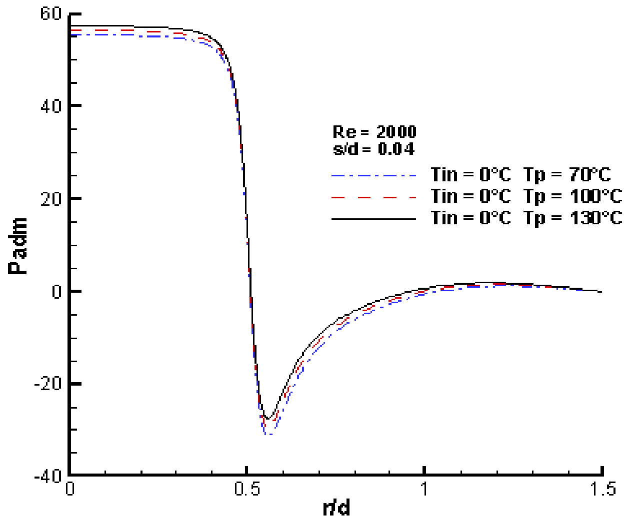

Figure 14. Pressure profile for s/d=0.04 and Re=2000.

First of all, it can be observed in all cases the existence of a flat pressure distribution in the region of the passage orifice, which is caused by the stagnation of the fluid at this region. In addition, as the fluid enters the gap between the frontal disk and the valve seat (radial diffuser region), there is a large pressure decrease due to the acceleration of the fluid. In most cases, the pressure continues to decrease due to friction forces. However, in some cases, mainly for Re=2000 for s/d=0.02 and s/d=0.04, the pressure starts increasing after the pressure drop due to the cross section area increase.

For the same Reynolds number it can be noticed that the dimensionless pressure decreases for increasing gaps, s/d, owing to the decrease of the friction forces in the radial diffuser region. Comparing the pressure profiles for the same gap and increasing Reynolds number, one can observe the reduction of the dimensionless pressure. In fact, the absolute pressure also increases, but as the dimensionless pressure is divided by the kinetic energy by unit volume, the resultant effect is the reduction of the dimensionless pressure.

The influence of the surface temperature, T_p, on the pressure distribution on the frontal disk also can be analyzed in those figures. For low Reynolds number (Re=600) one can note that the effect of varying T_p from 70ºC to 130ºC is small as the gap varies from 0.005 to 0.04 (11 to 10%). However, the influence of T_p is larger for Re=2000, diminishing as the s/d increases (10.5 to 3.5%). For s/d=0.04 (Fig. 14) there is practically no difference among the results (3.5%). In addition, it can be noticed the increment of the dimensionless pressure for increasing surface temperatures. The augmentation of the fluid viscosity for increasing temperatures explains the pressure enlargement. This effect diminishes as the Reynolds number increases.

5. CONCLUSION

The results showed that the dimensionless pressure increases for increasing surface temperatures, mainly for low Reynolds numbers. For Re=600 and s/d varying from 0.005 to 0.04, the pressure increases 11 to 9%, respectively. For Re=2000 and the same s/d variation, the pressure augmentation is larger, varying from 10.5 to 3.5%.

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5. RESPONSIBILITY NOTICE