FINITE ELEMENT ANALYSIS OF GAS-LUBRICATED FLAT MECHANICAL FACE SEALS WITH PRESSURE DAMS

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Abstract

A study of the influence of the sealing dam geometry on gas lubricated step face seal performance characteristics is carried out by using a finite element procedure based on the Galerkin weighted residual method. The finite element procedure is implemented to solve the Reynolds equation for compressible fluids. The variation of the seal opening force, flow leakage rate and frequency-dependent force coefficients is depicted in relation to some seal geometric and operating parameters, such as pressure ratio, clearance ratio, seal width and pressure dam extent.

Keywords: Gas Lubrication, Face Seals, Step Seals, Sealing Dam

1. INTRODUCTION

The development of efficient sealing technology for rotating machinery has been driven by strict environmental laws, which require virtual elimination of leakage of harmful process fluids into atmosphere. Due to their low leakage rate, wear-free operation and low power consumption, gas lubricated mechanical face seals have been widely employed to design very efficient sealing systems for machine components with rotating shafts, such as compressors, pumps, agitators and turbines (Burgmann, 1997).

Uni-directional and bi-directional gas face seals usually have grooves etched on either of their mating faces (Burgmann, 1997). The grooves provide a pumping effect that generates an increase in the hydrodynamic pressure (Gabriel, 1994). Typical gas face seal geometries possess an unprofiled area, the sealing dam, which can be located at either the seal inner or outer radius (Wasser, 1994). This pressure dam is introduced into the seal face to reduce leakage and increase axial stiffness. Most of the studies performed on gas face seals deal with the analysis of the influence of the seal-mating plate patterns, which generally include various types of grooves, face waviness, radial taper and face texture, on the seal performance (Tournerie *et al.* (1994); Zirkelback & San Andrés (1998); Hernandez & Boudet (1995); Salant & Homiller (1992); Etsion *et al.* (1998); Basu (1992); Wasser (1994); Burgmann (1997)). In the analyses of grooved face seals, the main concern has been the parametric study of the influence of the groove geometry, in conjunction with or without the sealing dam, on the seal behavior. The technical literature lacks studies more detailed about the influence of the pressure dam geometry on the performance of gas face seals.

A finite element analysis of gas-lubricated step mechanical face seals is performed to study the influence of the pressure dam geometry on gas seal performance. The hydrodynamic pressure within the gas flow domain is computed by solving the Reynolds equation for compressible fluids. The Galerkin weighted residual method is employed in the finite element formulation. A linearized perturbation method is used to obtain the zero-th and first order lubrication equations for the problem. Seal performance characteristics, such as seal opening force, flow leakage, and frequency dependent stiffness and damping force coefficients, are computed in relation to some seal parameters, such as dam extent, clearance ratio and seal width. The model validation is performed by comparing the finite element predictions with approximated analytical solutions for gas-lubricated flat mechanical face seals.

2. PROBLEM DEFINITION

A schematic view of the geometry of a flat mechanical face seal with outer pressure dam is shown in Figure 1. The sealing dam can be located at either the seal inner radius or outer radius, depending on the radial pressure ratio.

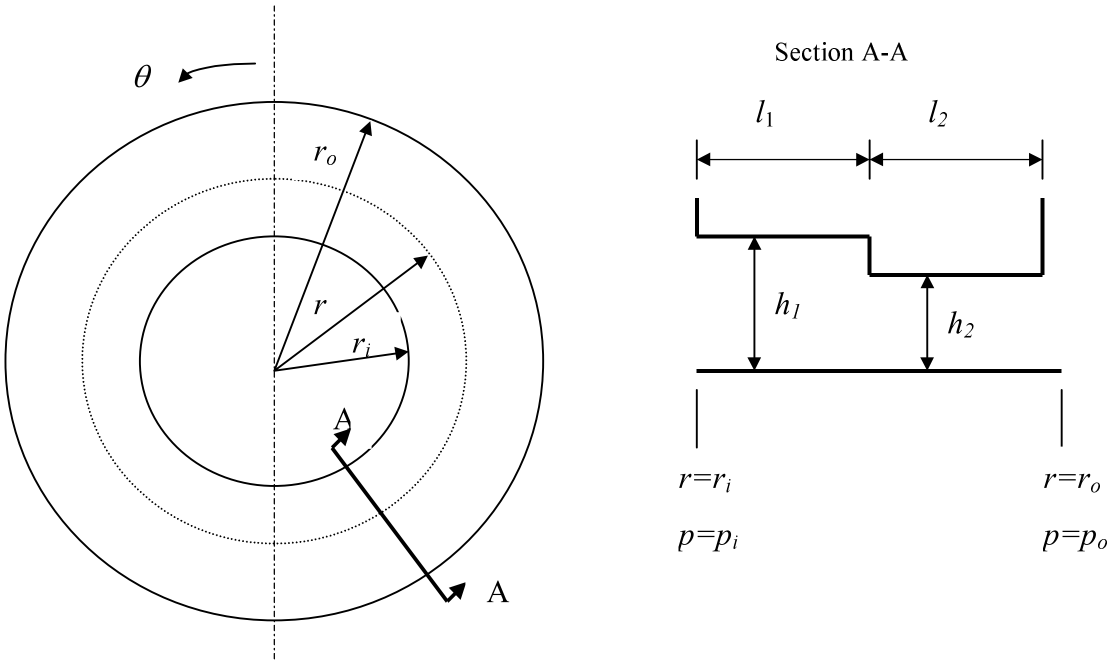


Figure 1. Schematic view of a flat mechanical face seal with pressure dam.

The inner, groove-ridge boundary, and outer radii are given by r_i, r_s, and r_o, respectively. The pressure dam extent is represented by l_2 and the seal extent by l_1. The seal length is $l=l_1+l_2$. The fluid film thicknesses at the dam and at the seal are expressed by h_2 and h_1, respectively. Generally the stepped face is rotating, while the flat face is non-rotating.

The lubricant flow in the face seal is described by the Reynolds equation for an isothermal, isoviscous, ideal gas (Zirkelback and San Andrés,1998)

$$\frac{1}{r}\frac{\partial}{\partial\theta}\left(\frac{ph^3}{12\mu}\frac{1}{r}\frac{\partial p}{\partial\theta}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{ph^3}{12\mu}\frac{\partial p}{\partial r}\right)=\frac{\Omega r}{2}\frac{\partial}{\partial\theta}(ph)+\frac{\partial}{\partial t}(ph) \tag{1}$$

where p describes the pressure field over the seal, h is the fluid film thickness, μ is the fluid viscosity, and Ω is the rotational speed of the seal. The boundary conditions for the system are given by $p(r_i)=p_i$ and $p(r_o)=p_o$. The pressures p_i and p_o are the inner and outer pressures acting on the inner and outer seal radius, respectively.

Due to the axi-symmetric nature of the gas flow, the pressure field is independent of the circumferential coordinate θ. Thus, equation (1) simplifies to the following form

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{ph^3}{12\mu}\frac{\partial p}{\partial r}\right)=\frac{\partial}{\partial t}(ph) \quad . \tag{2}$$

A dimensionless form of equation (2) is expressed as

$$\frac{1}{R}\frac{\partial}{\partial R}\left(RH^3P\frac{\partial P}{\partial R}\right)=\sigma\frac{\partial}{\partial\tau}(PH) \tag{3}$$

where the dimensionless variables are given by $R=\frac{r}{r_o}$, $P=\frac{p}{p_a}$, $H=\frac{h}{h_{\min}}$, $\tau=t\omega$, and $\sigma=\frac{12\mu\omega}{p_a}\left(\frac{r_o}{h_2}\right)^2$. In these variables, ω is the axial excitation frequency, p_a is the ambient pressure, and h_{min} is the minimum value of thickness. σ is the frequency or squeeze number.

3. PERTURBATION ANALYSIS

The rotating face seal is usually subjected to small axial perturbations from an equilibrium position. The effect of these small axial motions is to cause small perturbations in the pressure field. The expressions for the perturbed thickness and pressure field generated by dimensionless axial perturbations ΔZ are given in the following form.

$$H(R,\tau)=H_0(R)+\Delta Z.e^{i\tau} \tag{4.a}$$

$$P(R,\tau)=P_0(R)+P_1\Delta Z.e^{i\tau}\ ,\ \ i=\sqrt{-1}\ . \tag{4.b}$$

where P_0 and P_1 represent the zero-th and first order pressure fields, respectively. H_0 is the film thickness at equilibrium position. Expressions (4.a) and (4.b) are substituted into equation (3) to render the zero-th and first order lubrication equations,

$$\frac{1}{R}\frac{\partial}{\partial R}\left(RH_0^3P_0\frac{\partial P_0}{\partial R}\right)=0 \tag{5}$$

$$\frac{1}{R}\frac{\partial}{\partial R}\left(RH_0^3\frac{\partial(P_0P_1)}{\partial R}+3RH_0^2P_0\frac{\partial P_0}{\partial R}\right)=i\sigma(P_0+H_0P_1) \tag{6}$$

The zero-th order pressure field can be integrated over the seal domain to render the seal dimensionless opening force *Fz*.

$$Fz = \int_{R_i}^{R_o} (P_0 - 1) 2\pi R . dR \tag{7}$$

The complex first-order pressure field will render the dimensionless stiffness *Kzz* and damping *Czz* force coefficients for the seal.

$$Kzz + i\sigma Czz = \int_{R_i}^{R_o} 2\pi R P_1 \, dR \tag{8}$$

4. FINITE ELEMENT MODELING

The gas flow domain is modeled by using two-node linear finite elements of length L_e. A direct Galerkin weighted residual method is employed to render the element zero-th and first order lubrication equations. The zero-th and first order pressure fields depend only on *R*.

The zero-th order lubrication for a finite element *e* is written as

$$K_{ji}^e P_i^e = q_j^e \quad , (i,j=1,2) \tag{9}$$

where

$$K_{ji}^e = \int_0^{L_e} R H_e^3 P_o^e \frac{d\psi_i^e}{ds} \frac{d\psi_j^e}{ds} ds \tag{10}$$

$$q_j^e = \left[R \psi_j^e m_e \right]_{s=0} - \left[R \psi_j^e m_e \right]_{s=L_e} \tag{11}$$

P_o^e is the known value of pressure prior to the computation. The dimensionless flow rate is represented by $m_e = H_e^3 P_e \frac{dP_e}{dR}$. The dimensionless flux balance over an element *e* is given by q_j^e. K_{ji}^e represents the zero-th order fluidity matrix coefficients. The system of finite element equations (9) renders the zero-th order pressure distribution over the seal domain. An iterative procedure based on successive approximations, using the inner pressure p_i as initial guess, is used to compute the pressure field over the seal domain.

The first order finite element system of equations is obtained by using the same Galerkin procedure used in the derivation of the zero-th order equation. The first order lubrication equation for an element *e* is given by

$$K_{1\,ji}^e P_{1\,i}^e = q_{1\,j}^e + f_{1\,j}^e \quad , (i,j=1,2) \tag{12}$$

where

$$K_{1\,ji}^e = \int_0^{L_e} \left(R H_e^3 P_e \frac{d\psi_j^e}{ds} \frac{d\psi_i^e}{ds} + R H_e^3 \frac{dP_e}{ds} \frac{d\psi_j^e}{ds} \psi_i^e + iR\sigma H_e \psi_i^e \psi_j^e \right) ds \tag{13}$$

$$q_{1\ j}^{e} = \left[R\psi_j^e m_1^e\right]_{s=0} - \left[R\psi_j^e m_1^e\right]_{s=L_e} \tag{14}$$

$$f_{1\ j}^{e} = -\int_0^{L_e} iR\sigma P_e \psi_j^e ds - \int_0^{L_e} 3RH_e^2 P_e \frac{dP_e}{dR}\frac{d\psi_j^e}{ds} ds \tag{15}$$

P_e is the zero-th order pressure given by equation (9). $K_{1\ ji}^{e}$ represents the complex first-order fluidity matrix coefficients, $q_{1\ j}^{e}$ is the first-order flux balance over element *e*, and $f_{1\ j}^{e}$ represents a complex right-hand side vector. The first-order dimensionless flow rate is given by $m_1^e = \left(H_e^3 P_e \frac{dP_1^e}{dR} + H_e^3 \frac{dP_e}{dR} P_1^e + 3H_e^2 P_e \frac{dP_e}{dR} \right)$.

5. VALIDATION

Firstly, the zero-th order finite element (FEM) solution for gas lubricated step face seals is compared with the analytical steady-state solution derived for equation (3). The steady-state form of equation (3) is given by equation (5). The computation of the analytical zero-th order pressure field from equation (5) is straightforward. On the other hand, it is not so simple to obtain a closed-form solution for the first-order lubrication equation (6).

Figure 2 depicts the comparative results for dimensionless zero-th order pressure for a case of gas step face seal. The solid line represents the finite element solution obtained with 50 elements, while the dotted line represents the analytical solution. It is shown in Figure 2 that the solutions are practically the same.

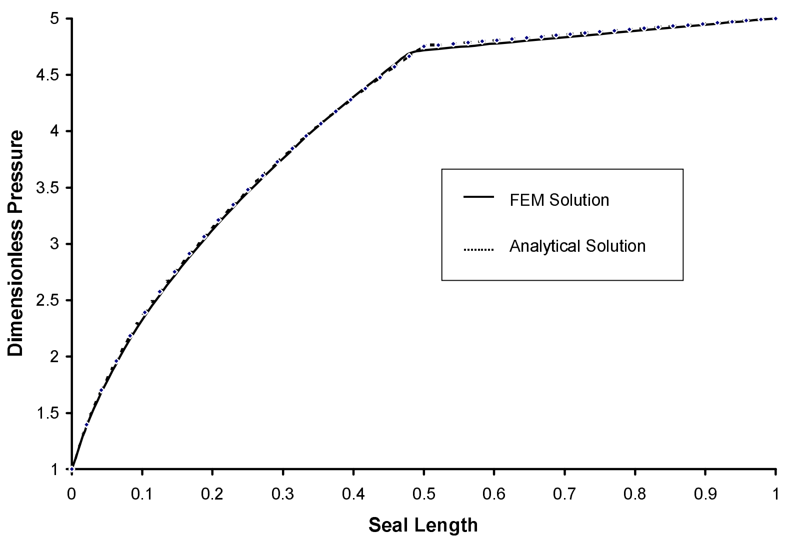


Figure 2. Comparative results for dimensionless pressure distribution in a step face seal ($P_o/P_i=5$; $H_1/H_2=0.5$; $R_i=0.8$; $R_s=0.9$; $R_o=1$).

6. SEAL PERFORMANCE CHARACTERISTICS

The dimensionless opening force (*Fz*), flow leakage rate (*Q*), stiffness coefficient (*Kzz*) and damping coefficient divided by the frequency number (*Czz/σ*) are computed for several cases of gas step face seals with inner pressure dam. The predicted values of seal performance characteristics are shown for the following parameters: i. Seal radial width ($SRW = (R_o - R_i)/R_o$); ii. Seal dam extent ($\delta = (R_o - R_s)/(R_o - R_i)$); iii. Seal pressure ratio ($PR = P_o/P_i$); iv. Seal clearance ratio ($CR = H_2/H_1$).

Figure 3 depicts the variation of *Fz* and *Q* with increasing values of *δ*. Solid lines represent the predicted values obtained for *SRW*=0.1, while the dotted lines are for *SRW*=0.2. Squares indicate the values computed for *CR*=2.5, while triangles are for *CR*=5. Figures 3(a) and 3(c) are computed for *PR*=5, while Figure 3(b) and 3(d) are for *PR*=10. The results show that opening force and leakage flow decrease as the seal dam extent increases. Wider sealing dams (larger *SRW*) lead to lower leakage flows and higher opening forces. Smaller clearance ratios (smaller *CR*) produce lower leakage flows mainly for small dam extent.

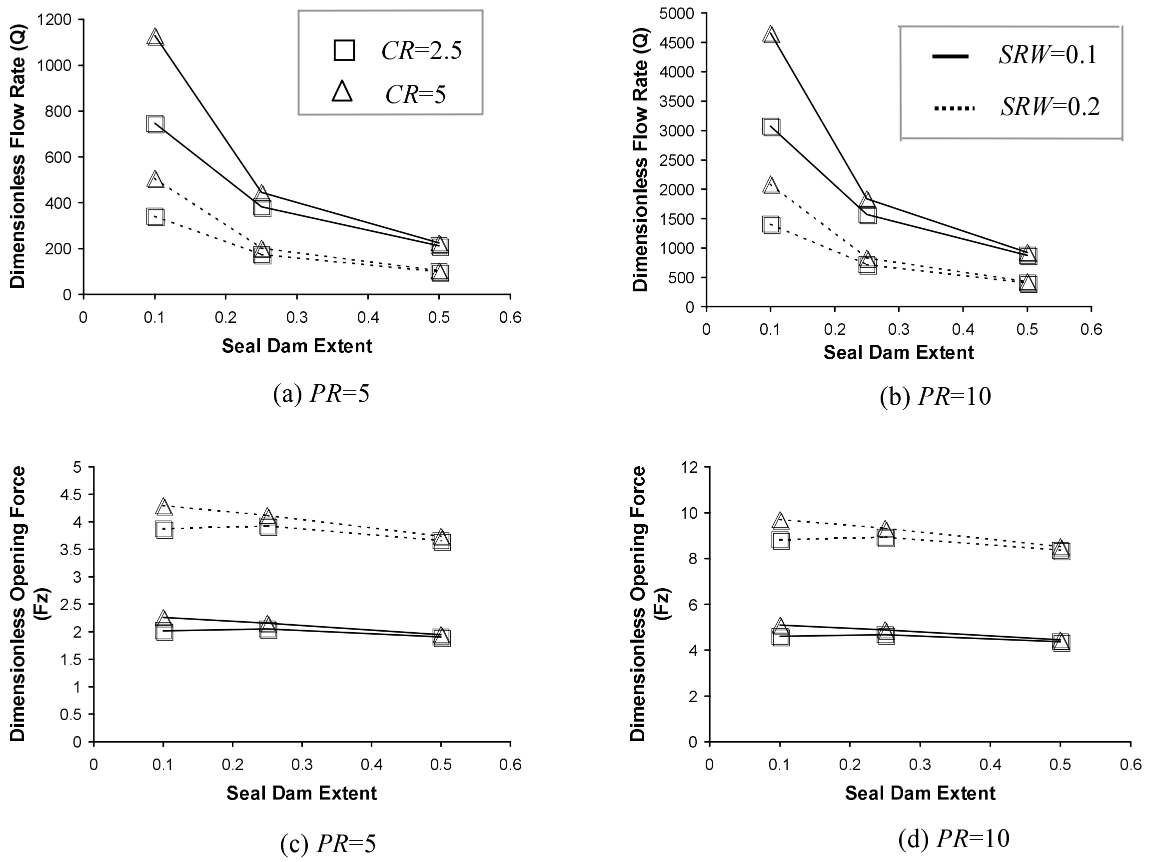
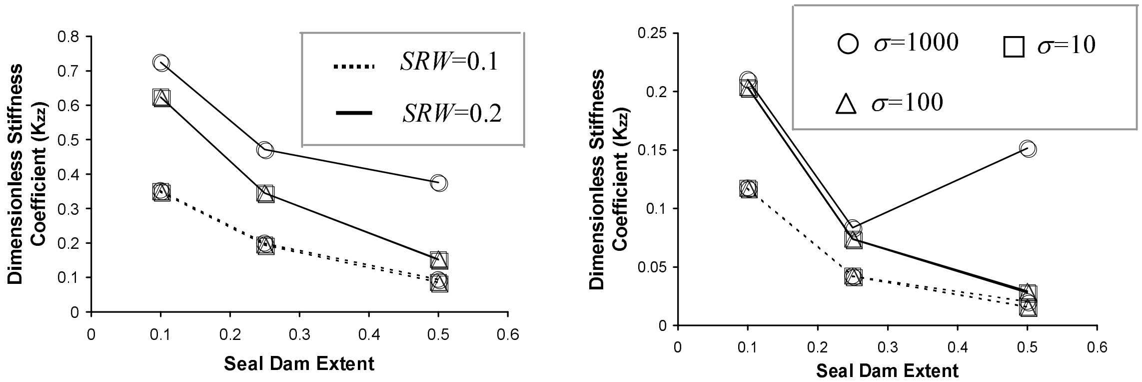


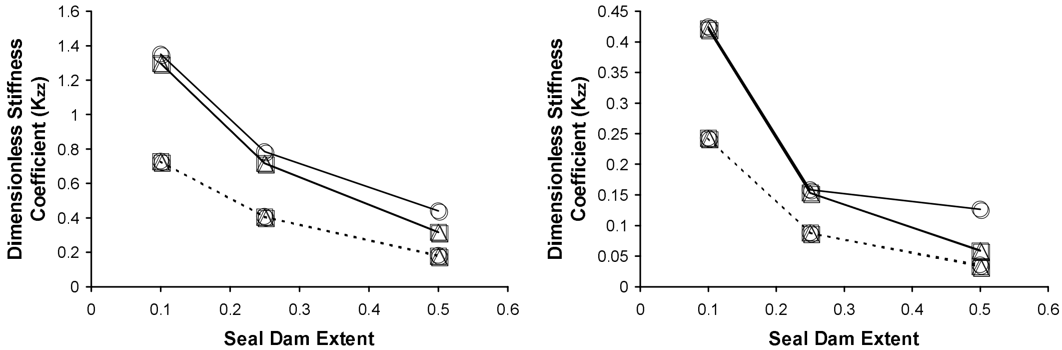
Figure 3. Dimensionless opening force and flow leakage rate versus the seal dam extent.

Figure 4 shows the dimensionless frequency-dependent stiffness coefficient (*Kzz*) versus the seal dam extent determined at three values of frequency number (low, medium and high). Solid lines represent the values of *Kzz* computed for *SRW*=0.2, while the dotted lines are associated with *SRW*=0.1. Squares indicate the values computed at *σ*=10, triangles are for *σ*=100, and circles are for *σ*=1000. *Kzz* generally increases as *σ* increases. Narrower seals (smaller *SRW*) lead to more flexible seal designs.



(a) *CR*=2,5 and *PR*=5

(b) *CR*=5 and *PR*=5



(c) *CR*=2,5 and *PR*=10

(d) *CR*=5 and *PR*=10

Figure 4. Dimensionless stiffness coefficients versus the seal dam extent at different values of frequency number.

Curves of the dimensionless damping coefficient (Czz/σ) versus the seal dam extent for different values of seal width and frequency number are depicted in Figure 5. As expected, (Czz/σ) increases as either the seal width or the sealing dam extent increases. The seal damping coefficient presents a decreasing trend for increasing values of frequency number.

7. CONCLUSIONS

A finite element procedure is specially devised to analyze the behavior of gas lubricated flat mechanical face seals with pressure dams. Seal opening force, flow leakage rate and dynamic force coefficients are computed in relation to some seal parameters, such as dam extent, clearance ratio and seal width. The analysis shows that the larger values of seal opening force, flow leakage rate and stiffness coefficient are obtained for smaller dam extents. Furthermore, the seal damping coefficients increase as the dam extent increases.

8. REFERENCES

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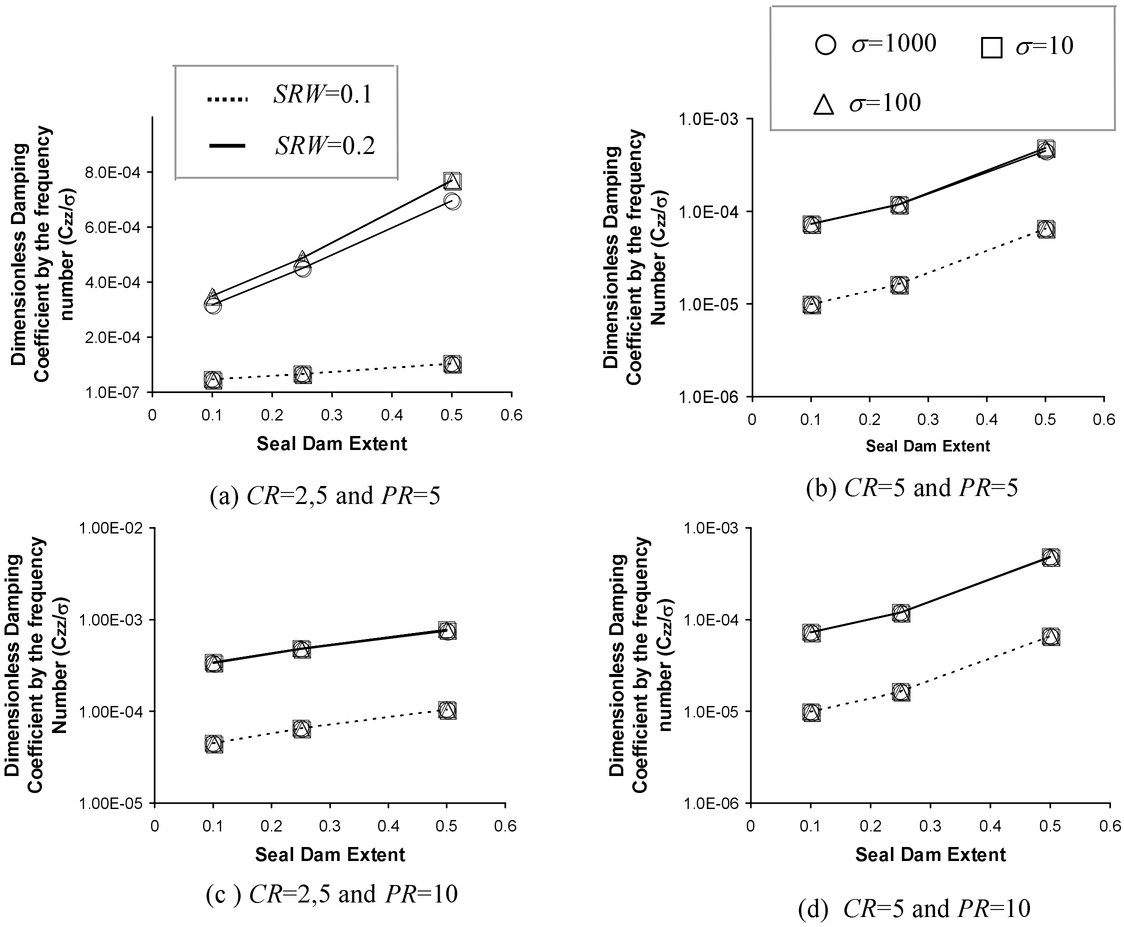

(a) *CR*=2,5 and *PR*=5

(b) *CR*=5 and *PR*=5

(c) *CR*=2,5 and *PR*=10

(d) *CR*=5 and *PR*=10

Figure 5. Dimensionless damping coefficient by the frequency number (Czz/σ) versus the seal dam extent at different values of frequency number.