한국기계가공학회지, 제18권, 제1호, pp.85∼93(2019.01) ISSN 1598-6721(Print)
Journal of the Korean Society of Manufacturing Process Engineers, Vol. 18, No. 1, pp.85~93(2019.01) ISSN 2288-0771(Online)

https://doi.org/10.14775/ksmpe.2019.18.1.085

A Study on the Design of Transmission Oil-Seal Using 2D Finite Element Analysis

Hyun-cheol Yoon*, Gi Hyun Jeon*, Ju Yong Choi**,#
*Department of Mechatronics Engineering, Kyungsung Univ.
**School of Mechanical&Mechatronics Engineering, Kyungsung Univ.

2D 유한요소해석을 이용한 트랜스미션 오일 씰 설계에 관한 연구

윤현철*, 전기현*, 최주용**,#
*경성대학교 메카트로닉스공학과, **경성대학교 기계메카트로닉스공학부

(Received 12 September 2018; received in revised form 16 October 2018; accepted 24 October 2018)

ABSTRACT

Oil seals are most essential parts in mechanical lubrication system to maintain the close gaps between stationary and high rotating components, and to help prevent oil leakages. Oil seals also can prevent harmful contaminants entering from outside to machinery, especially in severe environments. Therefore, the oil seals have an important performance in the machinery components. The performance of the oil seals are influenced by the design variables such as amount of interference gap between the main lip and shaft, the angle of main lip at air and oil sides and the distance between the garter spring and main lip. In the present study, a finite element analysis was performed to evaluate the oil seal performance with the considerations of number of oil seal dust lips and angle of the lip at oil side with the different design variables. As a result from the FEM analysis, the stress and contact pressure distributions was derived, based on this, performance of the sealing and durability were determined.

Key Words : Oil-Seal(오일씰), Finite Element Analysis(유한요소해석), Sealing(밀봉성), Durability(내구성)

1. Introduction

Oil seals prevent the leakage of oil or grease from the shaft and block pollutants such as dust from outside. They perform the sealing function while being tightly attached to the shaft by a constant tension and maintaining a torque. Furthermore, a convex oil film (meniscus) is formed between the shaft and the oil seal to block internal oil and external air. If the connection between the oil seal and shaft is faulty or has a durability problem, the oil is leaked to the outside, and thus, it cannot perform lubricating and cooling functions. This can drastically reduce the function of the transmission and cause serious damage such as fusion of the friction part. Therefore, the sealing

Corresponding Author : jychoi@ks.ac.kr
Tel: +82-51-663-4695, Fax: +82-51-626-4773

performance and durability of oil seals are key factors of oil seal performance, which are actively researched. Kim et al. [1] conducted a study on the optimal design of a double-lip-type oil seal. Lee et al. [2] researched the lip seal design by conducting an experiment. Kim et al. [3] performed a numerical study on the film thickness of single- and double-lip seals. Yoo et al. [4] investigated the relationship between the stress relaxation phenomenon and the performance of lip seals. Choi et al. [5] evaluated the performance of a PTFE (polytetrafluoroethylene) oil seal in the automotive engine front. Choi et al. [7] studied measures to improve the oil seal durability of automobile power steering.

The present study selected the number of dust lips and the oil-side angle of the main lip as design parameters and investigated the sealing performance and durability of transmission oil seals using finite element analysis.

2. Oil Seal Design

2.1 Oil seal structure

The elements that determine the shape of oil seals are largely divided into the fitting part and lip part, as shown in Fig. 1. The fitting part fixes the oil seal to the housing and prevents the infiltration of pollutants and the leakage of fluids between the inside of the housing and the contact surface. The lip part can be divided into the dust lip and main lip. The dust lip prevents the infiltration of foreign substances from outside, and the main lip prevents the leakage of fluids by maintaining contact with the rotation axis. In addition, there is a reinforcement band and a garter spring between the fitting part and lip part. The reinforcement band reinforces the strength of the oil seal made of rubber, and the garter spring increases the contact force of the oil seal and maintains it for a long time.

2.2 Design elements of oil seal

The factors that determine the shape of oil seals are shown in Fig. 2. The performance of oil seals is determined by the amount of interference between the main lip and shaft (δ) and between the main-lip angle alpha (α) and beta (β) and the distance between the garter spring and lip edge (d). These design elements have been researched for a long time through many empirical developments and experiments. As a result, the conditions for optimal sealing have been presented as a main-lip angle alpha (α) of the oil side of 40–60°, an air-side main-lip angle beta (β) of 20–35°, and a distance between the lip end and the center of the garter spring of 0.4–0.7 mm.

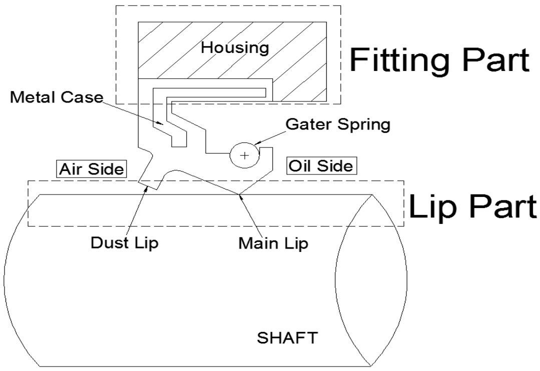


Fig. 1 Structure of oil-seal

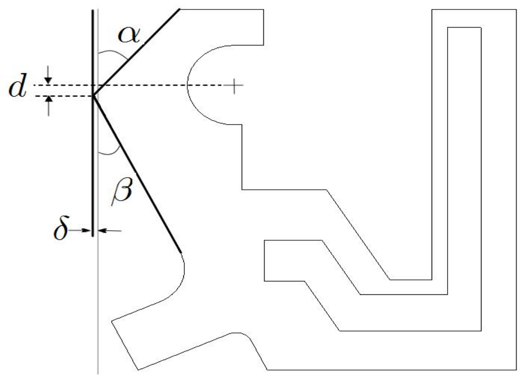


Fig. 2 Design parameters of oli-seal

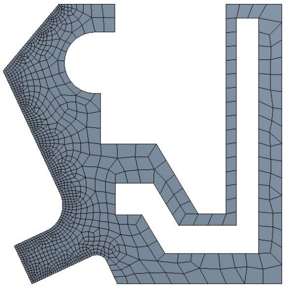
3. Finite Element Analysis

The analytical model has an axisymmetric shape. Hence, for efficient calculation, finite element analysis was performed using the commercial software ANSYS 19.0 with the 2D axisymmetric model.

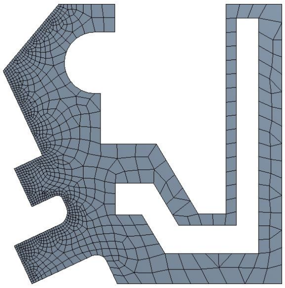
3.1 Analytical model and mesh

For the analytical model, the oil-side angle alphas (α) of the main lip for three numbers of lips (i.e., single, double, and triple) were selected, as shown in Fig. 3.

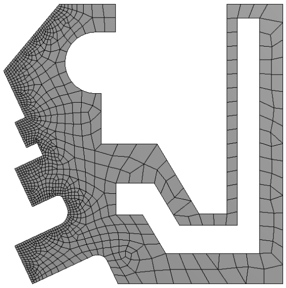
The mesh of the analytical model was set densely at 0.08 mm to improve the analysis accuracy for the shaft contact surface and relatively large at 0.5 mm for efficient analysis for the reinforcement band, because it does not have a significant effect on analysis.



(a) Single lip



(b) Double lip



(c) Triple lip

Fig. 3 Model and mesh

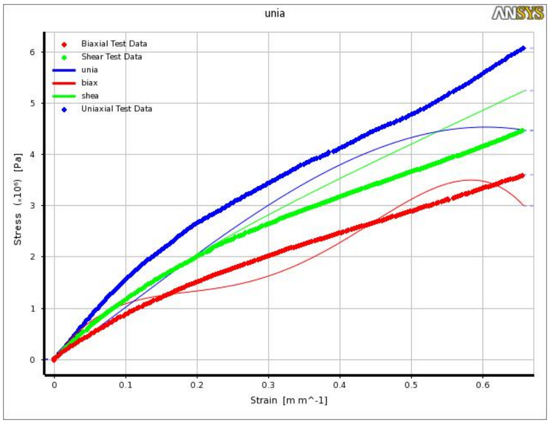


Fig. 4 Stress-Strain curve of fkm

3.2 Properties of FKM material

The rubber used in the analysis is FKM (fluorocarbon) rubber, which has excellent heat resistance and chemical resistance next to FFKM (perfluoroelastomer) among the currently commercialized rubbers. In this analysis, an S–S curve was derived based on the uniaxial, biaxial, and shear tensile test data using FKM. The Mooney–Rivlin model was applied to this analysis, which is generally used for the strains of 15–150% among the many models that have been proposed until now. The S–S curve shown in Fig. 4 was derived after curve fitting with a graph that is similar to actual data using five parameters.

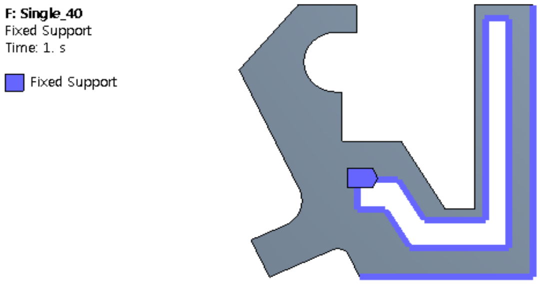
3.3 Analysis conditions

In general, the inner diameter of an oil seal is designed smaller than the diameter of the shaft to compress the shaft by a certain amount in order to maintain the fluid sealing performance. To implement this, the initial interference between the main lip and shaft was set at 0 mm with no load. Then, a positive (+) displacement was applied to the shaft in the x-axis so that the final interference between the main lip and shaft would be 0.5 mm and the interference between the dust lip and shaft would be 0.1 mm just like the model. Furthermore, the contact condition was set as Normal Lagrange, which does not permit the infiltration of oil seal and shaft, and the contact type was set as Frictional, considering the friction.

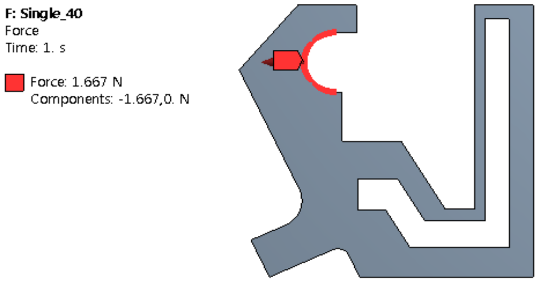
Fig. 5 shows the oil seal constraints. There is a reinforcement band for supporting the oil seal shape, but the simple fixed support condition was applied to shorten the analysis time. In addition, the fixed support condition was applied to the housing part as well because the oil seal is fixed to the housing in the outside. The load of the garter spring is determined by the inner diameter, outer diameter, and material. The initial load of the actual garter spring seal product used as an oil seal was approximately 130–220gf. A force of 170gf (approximately 1.667 N) was applied in the shaft direction to the circular shape of the spring diameter based on the literature [3].

3.4 Analysis results

The main concern of oil seal design is minimizing the oil leakage and friction loss. Durability can be verified by comparing the stress acting on the oil seal and the yield strength of the rubber. Furthermore, the sealing performance for the inflow of external foreign substances and oil leakage can be verified by comparing the maximum contact pressure acting on the lip part [3,4].



(a) Fixed support



(b) Gater spring Force

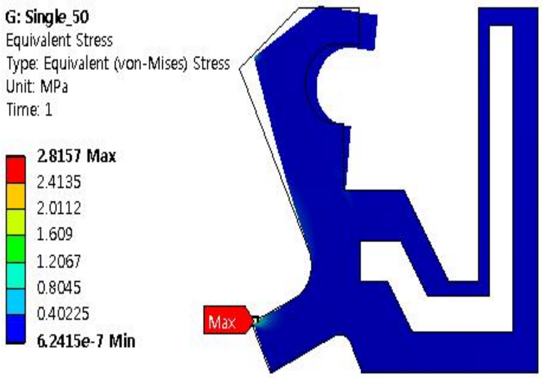
Fig. 5 Analysis condition of oil-seal

3.4.1 Maximum stress

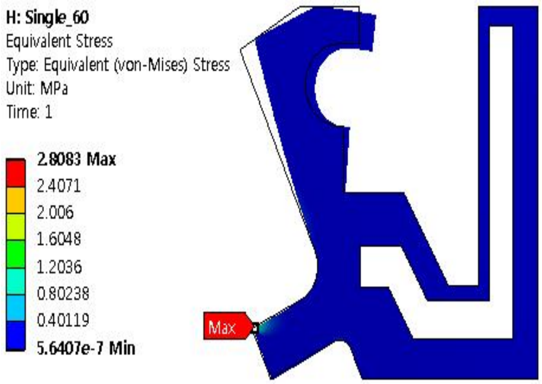
The analysis results for nine analyses in total are shown in Figs. 6–8 and Table 1. The maximum stress of the main lip according to the number of dust lips that had the same α angle did not show significant differences. The smaller the α angle was, the greater the maximum stress became. This is because as the α angle decreases, the position of the lip edge becomes farther from the reinforcement band and housing part, and the bending moment increases as the distance increases.

The main lip part receives a force for the shaft displacement as strain that is equal to the bending moment. Thus, the smaller the α angle is, the more stress the main lip receives. However, the durability performance is expected to be adequate because the stress applied to the main lip is less than the yield criterion of FKM (9.878 MPa) for all nine models.

(a) Main lip's α angle = 40°

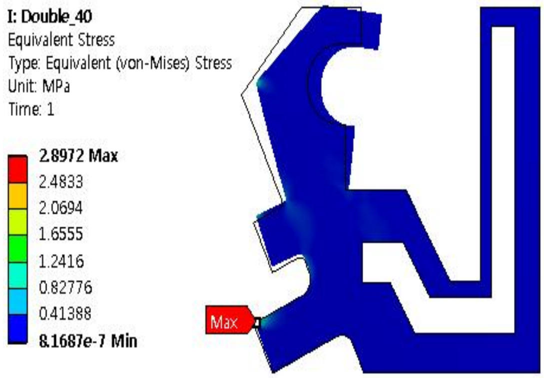


(b) Main lip's α angle = 50°

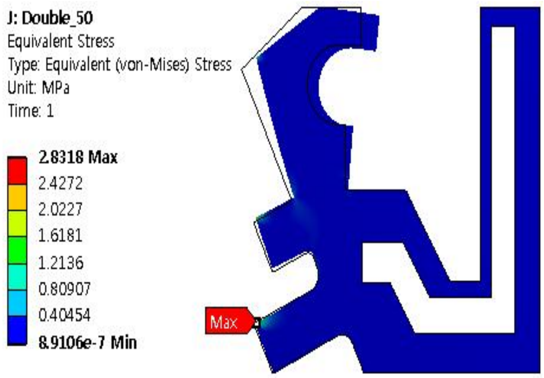


(c) Main lip's α angle = 60°

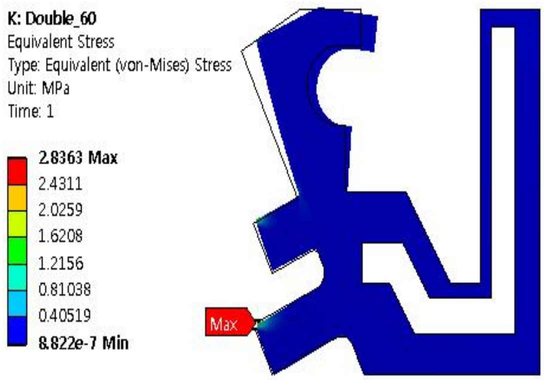
Fig. 6 Maximum stress of single lip



(a) Main lip's oil side angle α = 40°



(b) Main lip's oil side angle α = 50°



(c) Main lip's oil side angle α = 60°

Fig. 7 Maximum stress of double lip

(a) Main lip's oil side angle α= 40°

(b) Main lip's oil side angle α= 50°

(c) Main lip's oil side angle α= 60°

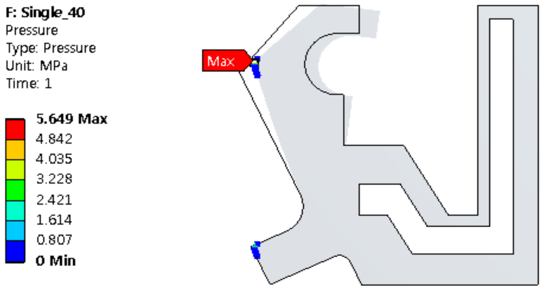
Fig. 8 Maximum stress of triple lip

Table 1 Result of maximum stress

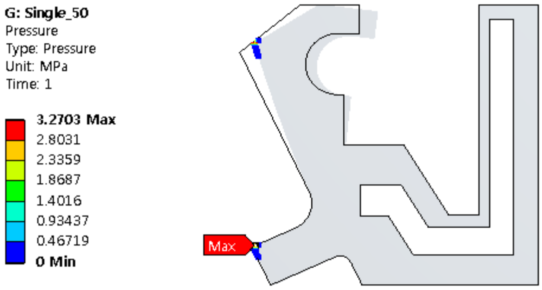
	Single lip (MPa)		Double lip (MPa)		Triple lip (MPa)	
	Main lip	Dust lip	Main lip	Dust lip	Main lip	Dust lip
40°	1.8862	2.8714	1.9423	2.8972	1.9682	2.8605
50°	1.4791	2.8157	1.535	2.8318	1.5371	2.8293
60°	1.227	2.8083	1.279	2.8363	1.3055	2.8304

Table 2 Result of maximum contact pressure

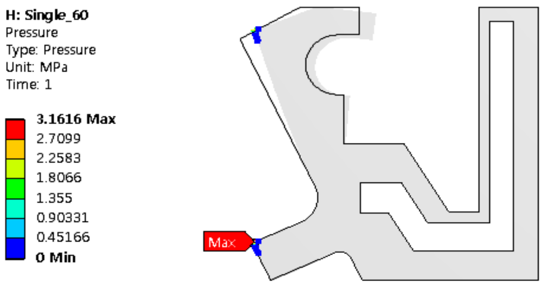
	Single lip (MPa)		Double lip (MPa)		Triple lip (MPa)	
	Main lip	Dust lip	Main lip	Dust lip	Main lip	Dust lip
40°	5.649	3.5266	5.817	3.4446	5.9323	3.4374
50°	2.9577	3.2703	3.0698	3.5056	3.0844	3.4696
60°	1.8514	3.166	1.9279	3.9599	1.9627	3.9501



(a) Main lip's oil side angle α= 40°

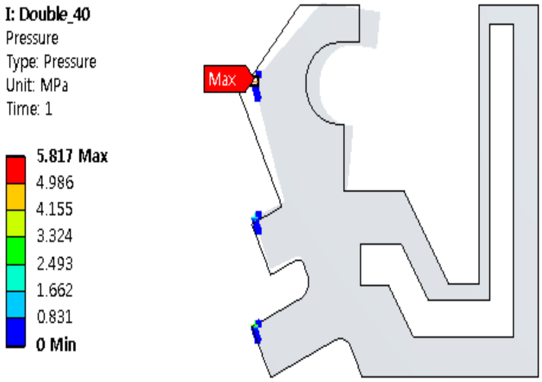


(b) Main lip's oil side angle α= 50°

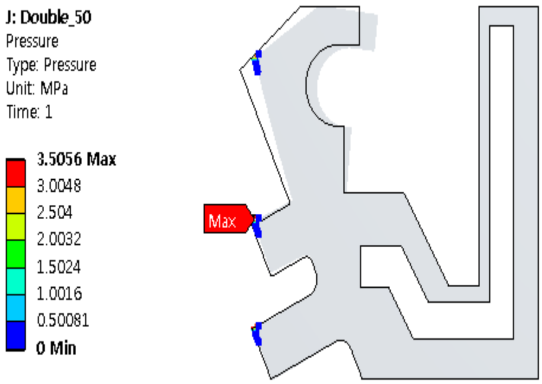


(c) Main lip's oil side angle α= 60°

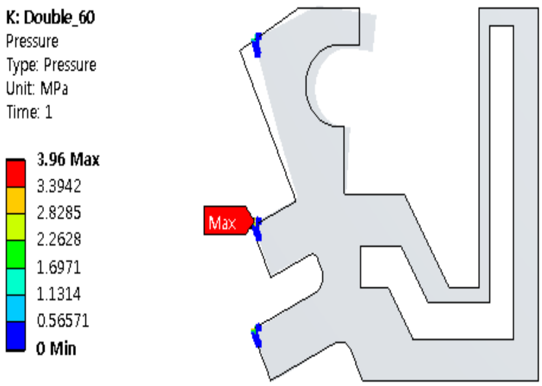
Fig. 9 Maximum contact pressure of single lip



(a) Main lip's oil side angle α = 40°

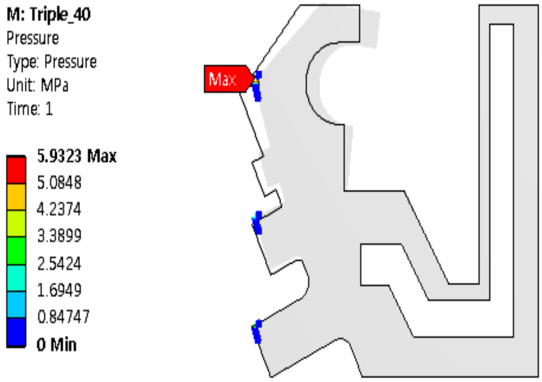


(b) Main lip's oil side angle α = 50°

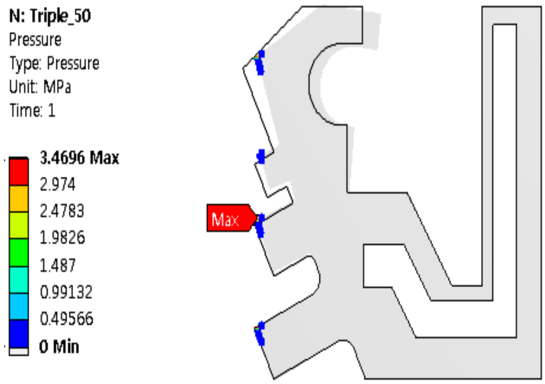


(c) Main lip's oil side angle α = 60°

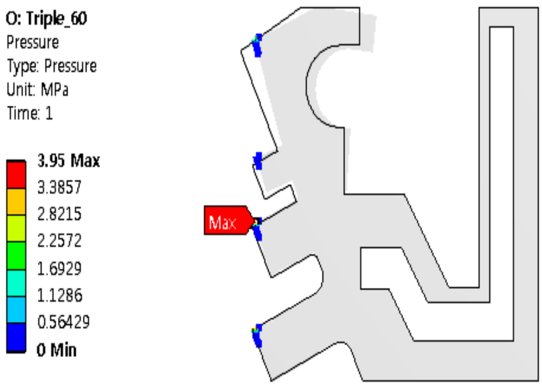
Fig. 10 Maximum contact pressure of double lip



(a) Main lip's oil side angle α = 40°



(b) Main lip's oil side angle α = 50°



(c) Main lip's oil side angle α = 60°

Fig. 11 Maximum contact pressure of triple lip

4. Conclusions

This study conducted finite element analysis for oil seals according to the design parameters, and the following conclusions regarding the sealing performance and durability were obtained.

1. The number of dust lips does not have a significant effect on the sealing performance and durability.
2. A change of the oil-side angle of the main lip (α) leads to a change in the distance between the fixed point and th lip edge
3. A change in the distance between the fixed point and lip edge has a significant effect on the bending moment, and this affects the strain and stress of the oil seal. However, this can be regarded as a stable behavior because the stress is lower than the allowable stress of FKM.
4. As the oil-side angle of the main lip (α) becomes smaller, the maximum contact pressure increases and the sealing performance improves. The sealing performance is adequate because the maximum contact pressure is higher than the maximum pressure applied by the oil.
5. However, excessive contact pressure can cause such problems as wear and deformation of the contact part of the oil seal, which directly reduce sealing performance. In particular, shaft wear can cause not only a sealing performance problem but also an economic loss due to replacement.
6. Therefore, the design parameters for oil seals that should be considered in relation to sealing performance and durability are the distance between the fixed support point and lip edge and the force of the garter spring. However, the number of dust lips is not a parameter for consideration. Furthermore, the main-lip angles α and β are parameters for selecting the position of the lip edge and affect the contact pressure, which is an indicator of sealing performance. Thus, the optimal angle should be determined considering the lip thickness and length.

Acknowledgement

This research was supported by The Leading Human Resource Training Program of Regional Neo industry through the National Research Foundation of Korea(NRF) funded by the Ministry of Science, ICT and future Planning(NRF-2016H1D5A1910536)

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