UKRAINIAN JOURNAL OF MECHANICAL ENGINEERING AND MATERIALS SCIENCE
Vol. 11, No. 3, 2025

Mehmet Bozca

Machine Design Division, Mechanical Engineering Department, Mechanical Engineering Faculty, Yildiz Technical University, Yildiz, Istanbul, Turkey, E-mail: mbozca@yildiz.edu.tr, ORCID 0000-0002-2620-6053

STRUCTURAL OPTIMISATION OF THE DRY CLUTCH WITH CONICAL FRICTION SURFACE WITH MULTI-OBJECTIVE FUNCTIONS

Received: August 25, 2025 / Revised: September 11, 2025 / Accepted: October 06, 2025

https://doi.org/10.23939/ujmems2025.03.049

Abstract. The structural optimisation of the dry clutch with a conical friction surface is presented, with multi-objective functions such as contact pressure and surface temperature. A program utilizing MATLAB for conducting both simulation and optimization calculations was developed. The variation of clutch torque, normal force, friction surface width, and pressure on the contact surface for contact pressure is simulated according to the inner and outer diameter dimensions of the pressure plate. Furthermore, the simulation takes into account the effects of friction work, outer surface area, and temperature on the friction surface, with these parameters derived from the inner and outer diameter dimensions of the pressure plate, aiming to generate heat. The results of the simulation indicate a direct correlation between the increase in the outer radius of the friction surface (ro) and the subsequent rise in clutch torque (MK), the decrease in normal force (FN), the increase in friction surface width (b), and the reduction in contact pressure (p). Furthermore, the simulation results demonstrated that as the outer radius of the friction surface, designated as Ro, increased, there was a concomitant increase in clutch torque, denoted as MK, as well as an increase in both friction work, denoted as WS, and outer surface area, denoted as AK. Concurrently, the temperature of the friction surface, denoted as T, decreased. Following the optimisation process, it was determined that all design parameters were found to be in accordance with the established pressure and surface temperature constraints. It is possible to achieve a lightweight clutch structure and minimise costs by optimising the design parameters.

Keywords. Dry clutch, Friction surface, Optimisation, Pressure constraint, Temperature constraint.

Introduction

A dry clutch with a friction surface is a crucial machine element that transmits torque from the driving part to the driven part in power transmission systems. Clutches are widely used between the engine and the gearbox in automotive transmissions and all types of manufacturing machines.

This study aims to present the structural optimization of a dry clutch with a conical friction surface, considering multiple objective functions, including contact pressure and surface temperature.

The following studies focus on the surface stress, surface deformation, surface pressure, and thermal behaviour of the dry clutch:

The analysis of stresses and deformations in a dry friction clutch system is presented. The finite element method is employed to investigate the stress and deformation of the dry clutch system under the influence of contact pressure and centrifugal force during full engagement of the clutch disc. It is concluded that the maximum and minimum values of contact pressure and total contact stress occur at the outer disc radius and the inner disc radius, respectively. Moreover, the permanent deformations and thermal cracks on the contact surfaces of the clutch affect the contact pressure distribution and the actual contact area [1].

A finite element analysis of the temperature field in an automotive dry friction clutch is presented. The finite element analysis is used to investigate the effect of thermal load type on the temperature field of the clutch system. It is concluded that slipping occurs between contact surfaces due to the difference in speed between the two contacting surfaces during engagement. The frictional heat generation reduces the performance of the clutch system and leads to premature failure in some cases [2].

Three-dimensional finite element analysis of the contact problem in dry friction clutches is presented. A finite element model of a single-disc dry friction clutch system is developed to estimate the distribution of contact pressure between the clutch system's contact elements. The effect of the elasticity modulus on the distribution of contact pressure of mating surfaces is investigated. It is concluded that heat generation leads to a drastic reduction in performance during the slipping phase, resulting in premature failure of the contacting surfaces, including the formation of cracks and plastic deformation. Moreover, the structural stiffness is proportional directly to the elasticity modulus of the contacting elements. A reduction in the magnitude of the elasticity modulus results in a decrease in structural stiffness. As a result, the contact pressure decreased with a reduction of the structural stiffness, whereas the displacements increased [3].

Analysis of the thermos-mechanical behaviour of a multi-plate clutch during transient operation conditions using the FE method is presented. A parametric two-dimensional finite element model is developed and validated for damage prevention and to analyze the thermomechanical behavior of a clutch in transient operation. It is concluded that the maximum temperatures tend to occur at the outer diameter of the friction area. The pressure distribution is very homogeneous. Although the elasticity modulus of the friction lining influences the pressure distribution in the friction contact, the temperature behavior is only slightly altered by variations in elastic modulus due to the load case [4].

When the clutch starts to engage, two physical events occur: i. contact pressure is applied to ***the contact surfaces.*** This contact pressure must be less than the permissible pressure value of the material of the pressure plates. The surface area of the pressure plates in the clutch must be designed according to this permissible pressure requirement. The appropriately sized pressure plate area can be determined by optimization under a contact pressure constraint. ***ii. Slippage occurs at the contact surfaces, generating heat.*** This heat generation must be less than the permissible heat value of the material of the pressure plates. The surface area of the pressure plates in the clutch must be designed according to this allowable temperature rise. The appropriately sized pressure plate area can be determined by optimization under surface temperature constraint.

Problem Statement

A dry clutch with a conical friction surface consists of the clutch cup, the clutch cone, the shifting groove, the spring, and the pressure plate. The main parts of the dry clutch with a conical friction surface are shown in Fig. 1. When the clutch is engaged, the clutch cone, mounted on the driven shaft by means of a spring, is pressed against the clutch cup. Thus, the motion is transmitted from the driving shaft to the driven shaft by means of clutches. When the clutch cup and clutch cone come into contact, surface pressure is generated. When the clutch is to be disengaged, the spring is pulled back through the shifting groove, thereby disconnecting the clutch cup and the clutch cone.

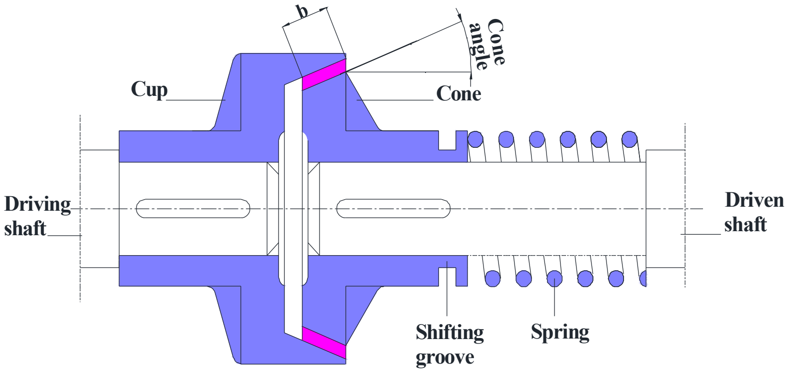

Fig. 1. Main parts of the dry clutch with conical friction surface

Methodology

Dry Clutch Calculation for Contact Pressure

Total clutch torque M_K is calculated by Equation (1) and Equation (2) [5–11]:

$$M_K = \left(\frac{2.\pi}{\sin\alpha}\right).p.\mu.\int_{r_i}^{ro} r^2.dr \; ; \tag{1}$$

$$M_K = \frac{2.\pi}{3.\sin\alpha}.\mu.p.\left(r_o^3 - r_i^3\right), \tag{2}$$

where α is cone half angle [°], μ is friction coefficient [-], p is contact pressure [N/mm²], r_i is friction surface inner radius [mm], and r_o is friction surface outer radius [mm].

Surface pressure p is calculated by Equation (3) and Equation (4) [5–11]:

$$p = \frac{F_N}{\pi.d_m.b} \leq p_{per} \; ; \tag{3}$$

$$p = \frac{F_e}{\sin\alpha\pi.d_m.b} \leq p_{per} \; , \tag{4}$$

where d_m is the average friction surface diameter [mm], b is the friction surface width [mm], and p_{per} is the permissible surface pressure value of the friction plate material [N/mm²].

$$d_m = r_o + r_i \; ; \tag{5}$$

$$b = \frac{r_o - r_i}{\sin\alpha} \; . \tag{6}$$

By substituting Equations (5) and Equation (6) into Equation (4), the surface pressure p is calculated by Equation (7):

$$p = \frac{F_e}{\pi.\left(r_o^2 - r_i^2\right)} \leq p_{per} \, . \tag{7}$$

By substituting Equation (7) into Equation (2), the total clutch torque is calculated with Equation (8):

$$M_K = \frac{F_e}{\sin\alpha}.\mu.r_m = F_N.\mu.r_m \, . \tag{8}$$

From Equation (8), the normal force is calculated according to the total clutch torque as follows:

$$F_N = \frac{M_K}{\mu.r_m} \, . \tag{9}$$

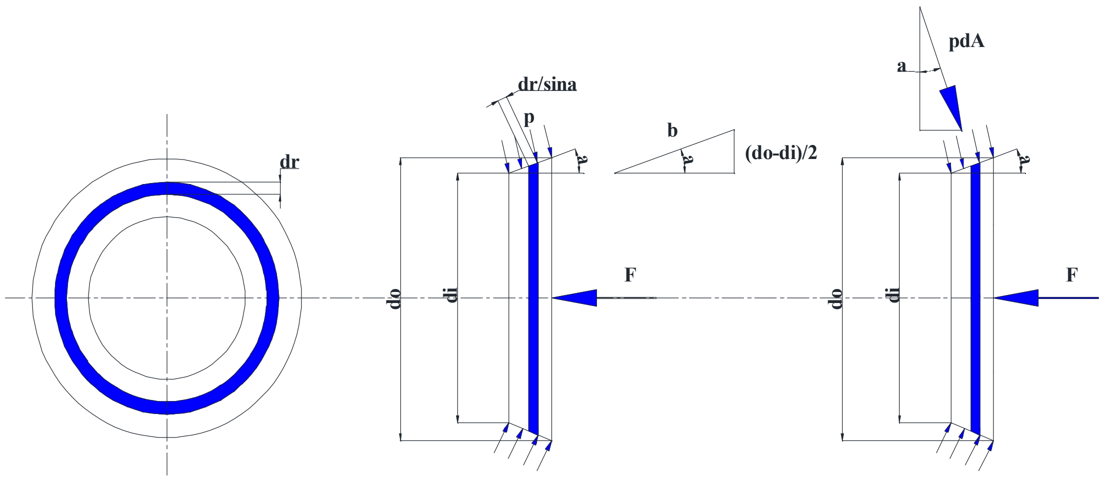

Fig. 2. Dimensions of dry clutch with conical friction surface

Dry Clutch Calculation for Temperature Rise

The temperature rise of the clutch-friction surface is calculated by Equation (10) [5–11]:

$$T = T_0 + \frac{W_S . z_h}{\alpha_K . A_K} \leq T_{per}, \tag{10}$$

where T_0 is temperature of the clutch before it engages [°C], W_S is friction work [J], z_h is number of times the clutch engages and exits per unit time [-], α_K is heat conduction coefficient [-], A_K is outer surface area of the clutch [m^2]. Tper is total permissible temperature [°C].

Friction work is calculated by Equation (11):

$$W_S = \frac{1}{2} M_K . \omega . t_S, \tag{11}$$

where M_K is clutch torque [N · m], ω is angular speed [1/s], and t_s is clutch engagement time [s].

Clutch torque is calculated by Equation (12) [5–11]:

$$M_K = \frac{F_e}{\sin\alpha} . \mu . r_m = F_N . \mu . r_m . \tag{12}$$

Heat conduction coefficient is calculated by Equation (13):

$$\alpha_K = 5{,}22 + (v_K)^{0{,}75} . 6{,}94, \tag{13}$$

where v_K is tangential speed [m/s].

Numerical Example

The numerical example consists of the simulation of parameters related to contact pressure and heat generation, and optimisation examples, respectively.

Simulation of Dry Clutch Mechanism for Contact Pressure

The variation of pressure on the contact surface is simulated according to the inner and outer diameter dimensions of the pressure plate. The friction surface outer radius diameter and cone semi-angle were assumed to be constant values during the simulation. The phenolic plastic /steel pair is considered as the friction material pair. Simulation parameters and simulation results for contact pressure are presented in Tables 1 and 2, respectively.

Table 1

Simulation parameters for contact pressure

Parameters	**Unit**	**Values**
Transmitted power P	kW	20
Number of revolutions of shaft n	[1/minute]	1000
Clutch engagement time t_s	[s]	4
Angular acceleration ε	[1/s^2]	26
Cone half angle α	[°]	20
Coefficient of friction for the phenolic plastic /steel pair μ	[-]	0.25
Permissible surface pressure for phenolic plastic/steel pair	[N/mm^2]	0.8

Table 2

Simulation results for contact pressure

No	**A [°C]**	**r_i [mm]**	**r_o [mm]**	**M_K [N.mm]**	**F_N [N]**	**b [mm]**	**p [N/mm^2]**
1	**20**	**100**	120	300990	10915	58.48	0.80
2	**20**	**100**	125	301020	10659	73.10	0.61
3	**20**	**100**	130	301050	10412	87.71	0.49
4	**20**	**100**	135	301100	10175	102.33	0.40
5	**20**	**100**	140	301160	9947	116.95	0.33
6	**20**	**100**	145	301230	9727	131.57	0.29

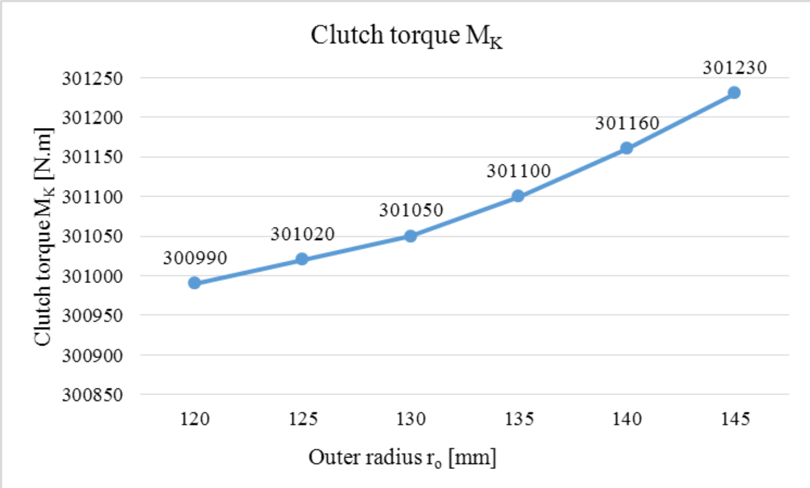

Fig. 3. Variation of clutch torque

As the outer radius of the friction surface r_o increases, the clutch torque M_K increases. Variation of clutch torque is shown in Fig. 3.

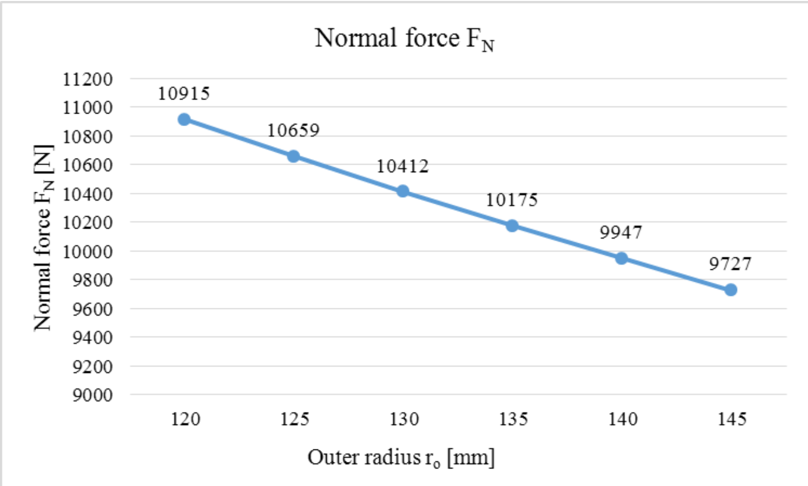

Fig. 4. Variation of normal force

As the outer radius of the friction surface r_o increases, the normal force F_N decreases. Variation of normal force is shown in Fig. 4.

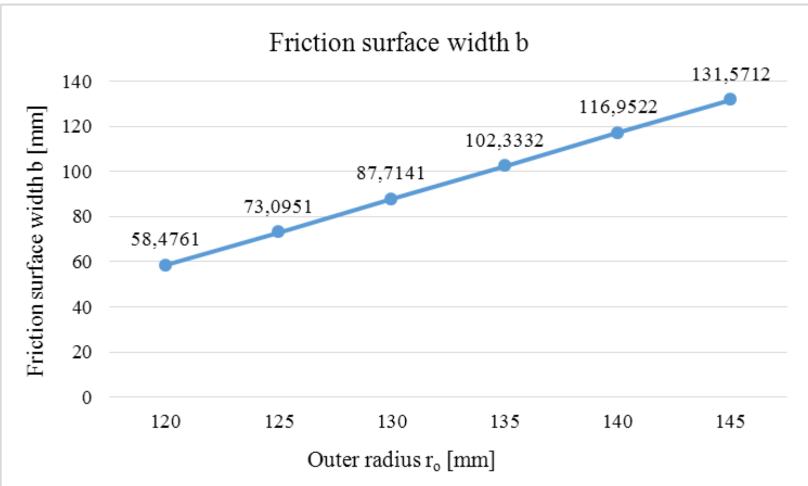

Fig. 5. Variation of friction surface width

As the radius of the friction surface increases, the friction surface width b increases. The variation in friction surface width is shown in Fig. 5.

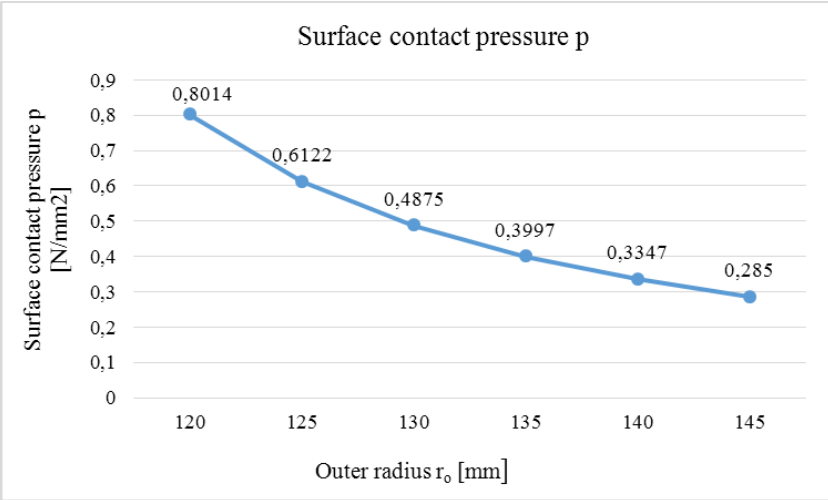

Fig. 6. Variation of contact pressure

As the outer radius of the friction surface r_o increases, the surface pressure p decreases. The variation in contact pressure is shown in Fig. 6.

Simulation of Dry Clutch Mechanism for Surface Temperature

The variation of temperature rise on the contact surface is simulated according to the inner and outer diameter dimensions of the pressure plate. The friction surface outer radius diameter and cone semi-angle were assumed to be constant values during the simulation. The phenolic plastic /steel pair is considered as the friction material pair. Simulation parameters and simulation results for surface temperature are presented in Tables 3 and 4, respectively.

Table 3

Simulation parameters for surface temperature

Parameters	**Unit**	**Values**
Transmitted power P	kW	20
Number of revolutions of shaft n	[1/minute]	1000
Angular speed ω	[1s]	105
Clutch engagement time t_s	[s]	4
Cone half angle α	[^{o}]	20
Coefficient of friction for the phenolic plastic/steel pair μ	[-]	0,25
Number of the clutch engages and exits per unit time z_h	[1/h]	60
Permissible surface temperature for phenolic plastic/steel pair	[^{o}C]	150
Temperature of the clutch before it engages T_0	[^{o}C]	20

Table 4

Simulation results for surface temperature

No	**α [^{o}C]**	**r_i [mm]**	**r_o [mm]**	**M_K [N.mm]**	**W_S [J]**	**A_K [m^2]**	**T [^{o}C]**
1	**20**	**100**	120	300990	60198	0.11	203.56
2	**20**	**100**	125	301020	60203	0.12	182.61
3	**20**	**100**	130	301050	60210	0.13	164.95
4	**20**	**100**	135	301100	60220	0.15	149.89
5	**20**	**100**	140	301160	60232	0.16	136.93
6	**20**	**100**	145	301230	60247	0.17	125.68

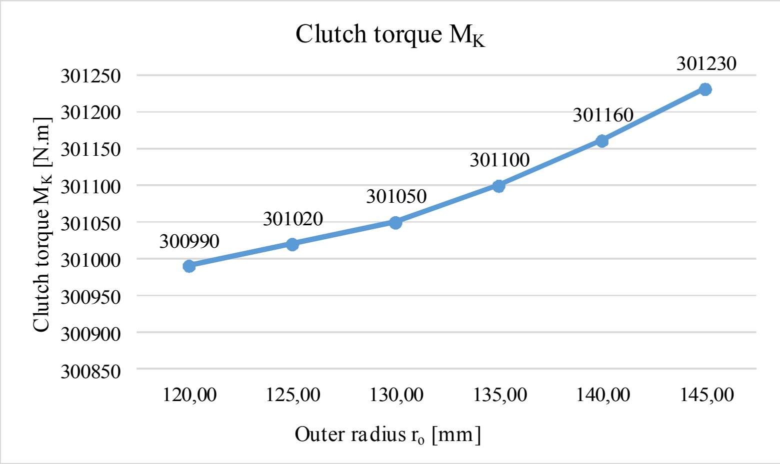

Fig. 7. Variation of clutch torque

As the outer radius of the friction surface r_o increases, clutch torque M_K increases. Variation of clutch torque is shown in Fig. 7.

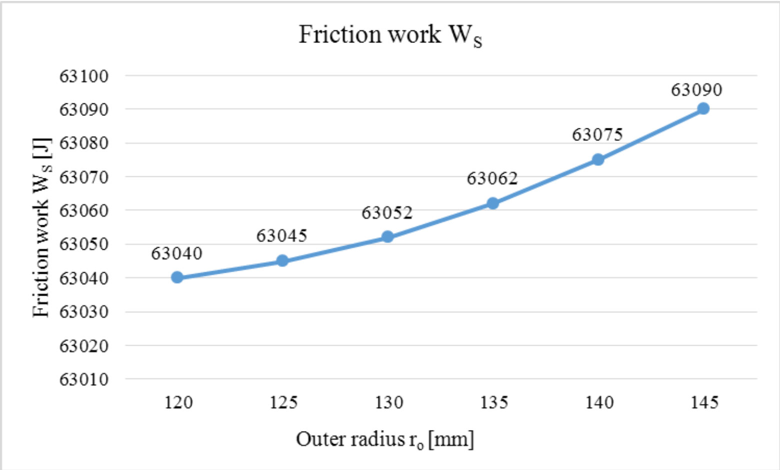

Fig. 8. Variation of friction work

As the outer radius of the friction surface r_o increases, friction work W_S increases. Variation of friction work is shown in Fig. 8.

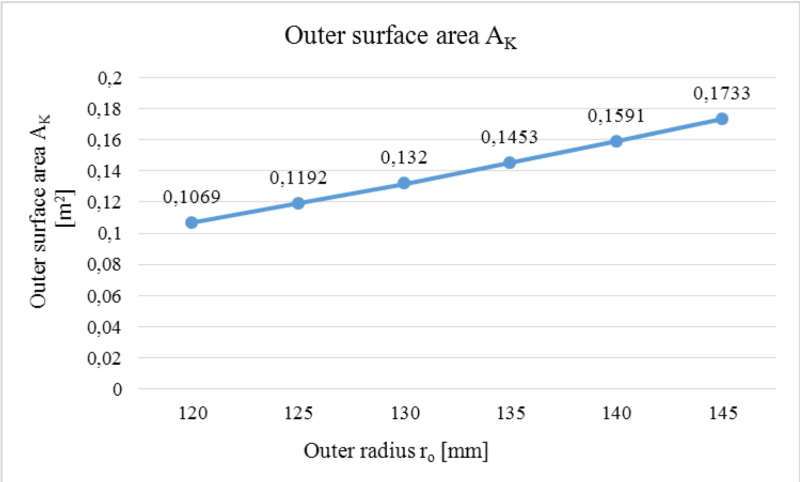

Fig. 9. Variation of outer surface area

As the outer radius of the friction surface r_o increases, outer surface area A_K increases. The variation in outer surface area is shown in Fig. 9.

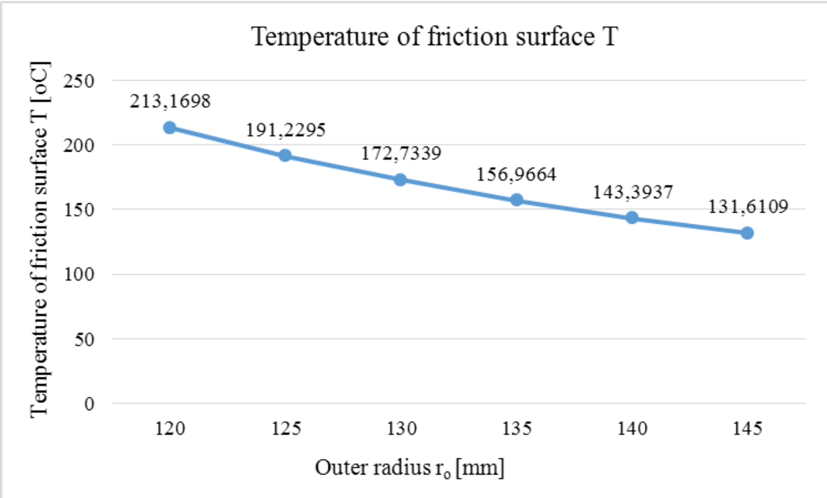

Fig. 10. Variation of temperature of the friction surface

As the outer radius of the friction surface r_o increases, the temperature T decreases. The variation in temperature of the friction surface is shown in Fig. 10.

Optimisation of Dry Clutch Mechanism for Contact Pressure

In optimization, the goal is usually to minimize the cost of a structure while satisfying the design specifications. By optimizing the design parameters, it is possible to obtain a light-weight clutch structure and reduce the cost.

Let F(X) denote the objective function to be minimized, where X is the design parameter vector to be determined. Then, to find the constrained minimum of F(X), the following optimization problem is solved:

$$\text{Min } F(X), \tag{14}$$

$$\text{Subject to: } \mathbf{LB} \leq \mathbf{X} \leq \mathbf{UB} \text{ and } G(X) \leq 0, \tag{15}$$

where, **LB** and **UB** define, the sets of lower and upper bounds on the design parameters **X**. Iterations start with the initial design parameter vector $\mathbf{X_0}$ and a solution vector **X** is found that minimizes the objective function F(X) subject to the nonlinear inequalities

$$G(X) \leq 0. \tag{16}$$

Where contact pressure p is considered as the objective function. The fact that the contact pressure is less than the permissible surface pressure is considered the constraint function.

$$F(X) = p\,, \tag{17}$$

$$G(X) = p - p_{per} \leq 0\,. \tag{18}$$

During optimization, the inner radius of the friction surface ri and the outer radius of the friction surface r_o values were determined. According to the determined ri and r_o values, the pressure constraint was calculated. All of the design parameters specified by optimisation satisfy the pressure constraint. Optimisation results are presented in Table 5.

Variation of inner radius r_i under pressure constraint is shown in Fig. 11. All the obtained optimum results satisfy the pressure constraint.

Variation of outer radius r_o under pressure constraint is shown in Fig. 12. All the obtained optimum results satisfy the pressure constraint.

Although all the obtained optimum results satisfy the constraint, the best solution is the choice of the design engineer.

Table 5

Optimisation results for contact pressure

No	LB	UB	X_0	r_i [mm]	r_o [mm]	$G(X) \le 0$ $p - p_{per} \le 0$
1	[90 110]	[110 130]	[100 120]	90	130	-0.5733
2	[100 120]	[120 140]	[105 125]	100	140	-0.6089
3	[110 130]	[130 150]	[115 130]	110	150	-0.6367
4	[120 140]	[140 160]	[125 135]	120	160	-0.6587
5	[130 150]	[150 170]	[135 140]	130	170	-0.6765
6	[140 160]	[160 180]	[145 145]	140	180	-0.6910

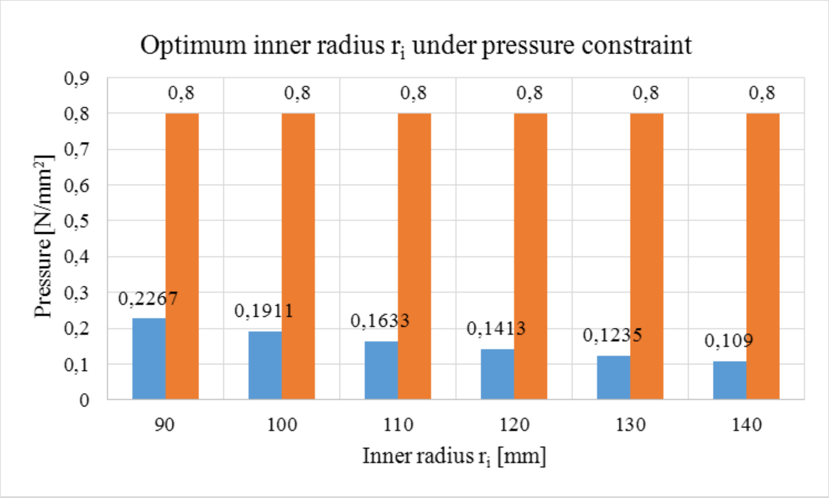

Fig. 11. Variation of inner radius under pressure constraint

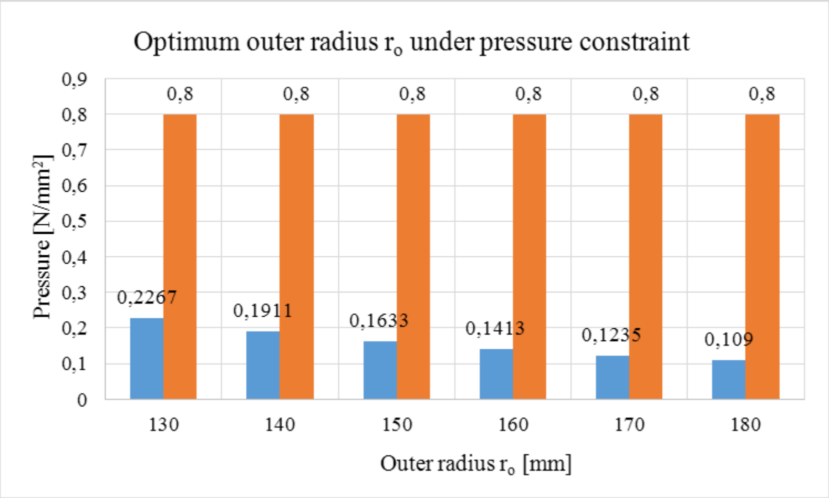

Fig. 12. Variation of outer radius under pressure constraint

Optimisation of Dry Clutch Mechanism for Surface Temperature

In optimization, the goal is usually to minimize the cost of a structure while satisfying the design specifications. By optimizing the design parameters, it is possible to achieve a lightweight clutch structure and minimize costs.

Let F(X) denote the objective function to be minimized, where X is the design parameter vector to be determined. Then, to find the constrained minimum of F(X), the following optimization problem is solved:

$$\text{Min } F(X), \tag{19}$$

$$\text{Subject to: } \mathbf{LB} \leq \mathbf{X} \leq \mathbf{UB} \text{ and } G(X) \leq 0, \tag{20}$$

where **LB** and **UB** define, the sets of lower and upper bounds on the design parameters **X**. Iterations start with the initial design parameter vector $\mathbf{X_0}$ and a solution vector **X** is found that minimizes the objective function F(X) subject to the nonlinear inequalities

$$G(X) \leq 0, \tag{21}$$

where the temperature of the friction surfaces T is considered as the objective function. The fact that the temperature of friction surfaces is less than the permissible surface temperature is considered the constraint function.

$$F(X) = T\ , \tag{22}$$

$$G(X) = T - T_{per} \leq 0\ . \tag{23}$$

During optimization, the inner radius of the friction surface r_i and the outer radius of the friction surface r_o values were determined. According to the determined r_i and r_d values, the temperature constraint was calculated. All of the design parameters defined by optimisation satisfy the temperature constraint. Optimisation results are presented in Table 6.

Table 6

Optimisation results for surface temperature

No	**LB**	**UB**	$\mathbf{X_0}$	$\mathbf{r_i}$ [mm]	$\mathbf{r_o}$ [mm]	$G(X) \leq 0$ $T - T_{per} \leq 0$ [^{o}C]
1	[90 110]	[110 130]	[100 120]	90	130	-54.5875
2	[100 120]	[120 140]	[105 125]	100	140	-64.9715
3	[110 130]	[130 150]	[115 130]	110	150	-73.4583
4	[120 140]	[140 160]	[125 135]	120	160	-80.6136
5	[130 150]	[150 170]	[135 140]	130	170	-86.7058
6	[140 160]	[160 180]	[145 145]	140	180	-91.9362

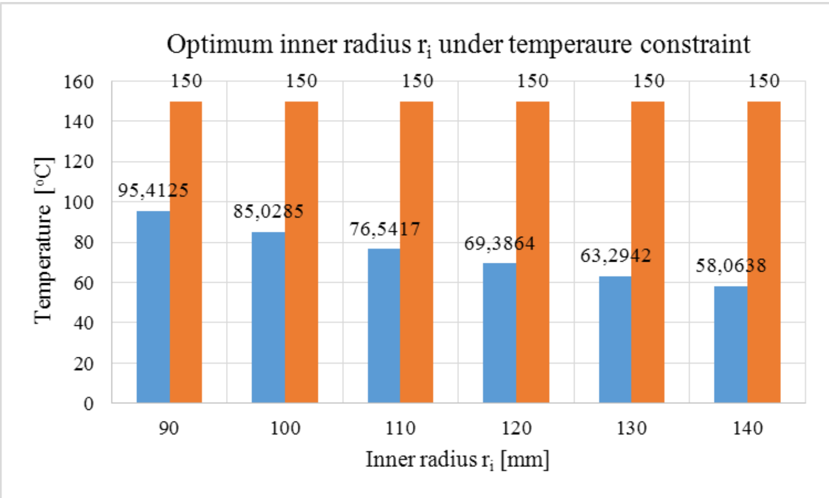

Fig. 13. Variation of inner radius under temperature constraint

Variation of inner radius r_i under temperature constraint is shown in Fig. 13. All the obtained optimum results satisfy the temperature constraint.

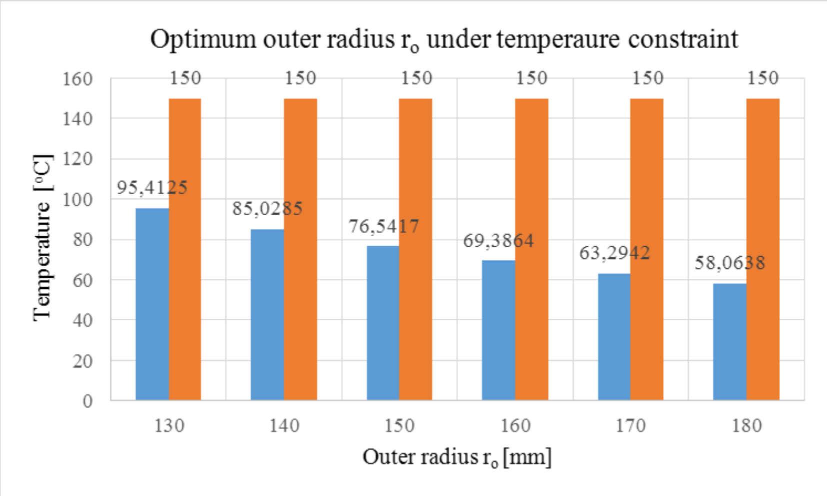

Fig. 14. Variation of outer radius under temperature constraint

Variation of outer radius r_o under temperature constraint is shown in Fig. 14. All the obtained optimum results satisfy the temperature constraint.

Although all the obtained optimum results satisfy the constraint, the best solution is the choice of the design engineer.

Results

In Table 2, as the outer radius of the friction surface r_o increases from 120 [mm] to 145 [mm],

i. Clutch torque M_K increases from 300990 [N.mm] to 301230 [N.mm]

ii. Normal force F_N decreases from 10915 [N] to 9727 [N].

iii. Friction surface width b increases from 58.48 [mm] to 131.57 [mm].

iv. Surface pressure p decreases from 0.80 [N/mm^2] to 0.29 [N/mm^2].

In Table 4, as the outer radius of the friction surface r_o increases from 120 [mm] to 145 [mm],

v. Clutch torque M_K increases from 300990 [N.mm] to 301230 [N.mm].

vi. Friction work W_S increases from 63040 [J] to 63090 [J].

vii. Outer surface area A_K increases from 0.11 [m^2] to 0.17 [m^2].

viii. Temperature T decreases from 213.17 [^{O}C] to 131.61 [^{O}C].

In Table 5, as the inner radius r_i increases from 90 [mm] to 140 [mm] and the outer radius r_o increases from 130 [mm] to 180 [mm],

ix. Pressure p decreases from 0.2267 [N/mm^2] to 0.109 [N/mm^2] and is less than the permissible pressure value 0.8 [N/mm^2].

In Table 6, as the inner radius r_i increases from 90 [mm] to 140 [mm] and the outer radius ro increases from 130 [mm] to 180 [mm],

x. Temperature T decreases from 95.4125 [^{O}C] to 58.0638 [^{O}C] and is less than the permissible temperature value 150 [^{O}C].

Conclusions

The structural optimisation of a dry clutch with a conical friction surface, considering multi-objective functions such as contact pressure and surface temperature, is presented.

The variation of clutch torque, normal force, friction surface width, and pressure on the contact surface is simulated according to the inner and outer diameter dimensions of the pressure plate, and the following results were obtained:

i. As the outer radius of the friction surface r_o increases, the clutch torque M_K increases.

ii. As the outer radius of the friction surface r_o increases, the normal force F_N decreases.

iii. As the outer radius of the friction surface increases, the friction surface width b increases.

iv. As the outer radius of the friction surface r_o increases, the contact pressure p decreases.

v. All of the design parameters determined by optimisation satisfy pressure constraints.

The variation of clutch torque, friction work, outer surface area, and temperature on the friction surface is simulated according to the inner and outer diameter dimensions of the pressure plate, and the following results were obtained:

i. As the outer radius of the friction surface r_o increases, clutch torque M_K increases.

ii. As the outer radius of the friction surface r_o increases, friction work W_S increases.

iii. As the outer radius of the friction surface r_o increases, outer surface area A_K increases.

iv. As the outer radius of the friction surface r_o increases, the temperature of the friction surface T decreases.

v. All of the design parameters determined by optimisation satisfy temperature constraints.

Although all the obtained optimum results satisfy the constraint, the best solution is the choice of the design engineer. By optimizing the design parameters, it is possible to achieve a lightweight clutch structure and minimize costs.

References

[1] O. I. Abdullah, J. Schlattman, A. M. Al-Shabibi, “Stresses and deformations analysis of a dry friction clutch system”, *Tribology in Industry,* vol. 35, No. 2, pp. 155–162, 2013.

[2] O. I. Abdullah, J. Schlattman, “Finite element analysis of temperature field in automotive dry friction clutch”, *Tribology in Industry,* vol. 34, No. 4 pp. 206–216, 2012.

[3] L. A. Sabri, N. Stojanovic, A. Senatore, M. J. Jweeg, A. M. Abed, O. I. Abdullah, “Three-dimensional finite element analysis of contact problem in dry friction clutches”, *Lubricants,* vol. 9, No. 115, pp. 1–10, 2021.

[4] T. Schneider, M. Dietsch, K. Voelkel, H. Pflaum, K. Stahl, “Analysis of the thermos-mechanical behaviour a multi-plate clutch during transient operation conditions using the FE method”, *Lubricant,* vol. 10, No. 76, pp. 1–22, 2022.

[5] H. Naunheimer, B. Bertsche, J. Ryborz, W. Novak, “Automotive Transmissions”, 2nd edition. Springer, 2011.

[6] R. Fischer, F. Küçükay, G. Jürgens, R. Najork, B. Pollak, “The Automotive Transmission Book”, 1st edition. Springer, 2015.

[7] R. G. Budynas, J. K. Nisbett, “Shigley’s Mechanical Engineering Design”, 9th edition. Mc Graw Hill, 2008.

[8] R. C. Juvinall, K. M. Marshek, “Fundamentals of Machine Component Design”, 4th edition. John Wiley& Sons Inc., 2006.

[9] Roloff/Matek, “Mashinenelemente”, 20th edition. Wieweg+Teubner, 2011.

[10] K. H. Decker, “Mashinenelemente”, 17th edition. Carl Hanser, 2009.

[11] Haberhauer/Bodenstein, “Mashinenelemente”, 15th edition. Springer, 2009.