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Optimal Design of an In-flight Refueling Door Mechanism

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ABSTRACT

In this study, the preliminary design of an in-flight refueling door mechanism is performed. A systematic design methodology is introduced and used in the design of the refueling door mechanism. The design is divided into two sub-functions: door opening and actuation. Nine different mechanism concepts are created for the door opening function and eight different concepts are created for the actuation function. Pugh decision matrix method is used to evaluate and select the most feasible options. Six experienced engineers scored the option set, resultantly two concepts for the door opening and three concepts for the actuation sub-function are selected. Kinematic synthesis of these concepts is performed and used to determine the upper and lower bounds during optimization. Kinematic and force analysis of the concepts are performed and utilized for the constraints and cost function calculations of the optimization algorithm. Multi-objective Genetic Algorithm optimization technique is used to optimize the parameters of the selected mechanisms. The best mechanism for each sub-function is selected and combined to reach the final design. It was shown that through optimization, the required input torque decreased approximately 20% for the door opening mechanism and the required input force decreased approximately 42% for the actuation mechanism when compared to the graphical synthesis results.

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INTRODUCTION

Aerial refueling is the process of transferring fuel from a tanker aircraft to a receiving aircraft when both aircrafts are flying [1]. The purpose of this operation is to extend the operation time and range of aircrafts. There are mainly two types of refueling systems used in modern aircrafts. One is the probe-and-drogue type [12] and the other is the flying boom type [13]. In probe-and-drogue type refueling system, there is a flexible hose on the tanker aircraft and a probe on the receiving aircraft that is inserted to the hose through the drogue for refueling. In the flying boom type refueling system, there is a rigid telescopic tube that extends from the tanker aircraft to the receiving aircraft and the tube is inserted to a receptacle on the receiving aircraft. In most of the modern aircrafts and in this paper, due to faster fuel transfer, flying boom type refueling system is used.

In the flying boom type refueling system, there is a receptacle that receives the fuel and this receptacle is protected by the in-flight refueling door. Before refueling, the door, mostly made in two parts, opens symmetrically so that the telescopic tube engages with the receptacle on the receiving aircraft. Different door opening mechanisms in different contexts have been studied in the literature. One of them was the swing plug door [2]. It had a four-bar mechanism that opened laterally and occupied small space when fully opened. The same mechanism was also used for luggage door mechanisms on commercial vehicles [3]. It was mentioned that the parallel-hinged system has a narrow and safe trajectory and takes up less space when fully open. Several different door hinge mechanisms have been designed for different applications such as cabinet doors [14] and garage doors [4]. Another multi-link door mechanism was the invisible hinge [5]. The design allowed the door to open up to 180° and did not show the hinge externally when the door is in the closed position. Toropov and Robertis [6] proposed an analytical approach for the design of invisible hinge mechanisms.

To obtain symmetric motion [15] as in the case of two-part refueling doors, different types of mechanisms are used. Gripper mechanisms are one of the examples of this type of mechanism. Lanni and Ceccarelli used an industrial two-finger gripper, which was powered and controlled by one actuator. A prismatic joint and revolute joint were combined to actuate the gripper mechanism symmetrically [7]. Another two-finger gripper mechanism was introduced by Nuttall and Breteler [8]. One inverted-crank mechanism was used in a piston-cylinder arrangement, and two four-bar mechanisms were used to transmit the motion to two sides of the gripper.

Different methodologies have been employed in the design of mechanisms. In [16] different mechanisms are created based on degree of freedom requirements and evaluated systematically based on the functional requirements. Automatic synthesis of three degrees of freedom closed-loop mechanisms are performed based on contracted graphs and topological graphs in [17, 18]. In [19] functional requirements, structural requirement, and design constraints are considered. Atlas of mechanism is used to find the compatible kinematic structure in a systematically.

The aim of this study is to synthesize an optimal in-flight refueling door mechanism for a flying boom type refueling system with a systematic design methodology as in [16, 17, 18]. The mechanism should provide the required clearance when refueling door is open, should close firmly in its place and should occupy as small space as possible.

Different studies have been performed to optimize mechanism with Genetic Algorithm [20, 21]. In this study, a design methodology is introduced to synthesis mechanism for different problems and optimize these mechanisms with Genetic Algorithm. Therefore, contribution of this study is developing a design methodology, which is combination of a systematic way of mechanism synthesis and optimization with using Genetic Algorithm.

METHOD

Design Methodology

In this paper, a systematic design methodology is proposed and followed in the design of the in-flight refueling door mechanism as illustrated in Fig. 1.

First, the problem is defined, and the main function is subdivided into sub-functions. Following that concepts are created for each sub-function. Then, the concepts are evaluated based on the evaluation criteria by experienced designers such that two or three best mechanisms are selected for each sub-function. Kinematic synthesis for each mechanism is performed and these results are used to determine the upper and lower boundaries of design parameters for the optimization. Following that, position and force analysis of each concept is performed and used during optimization for the constraints and cost function. Each concept is then

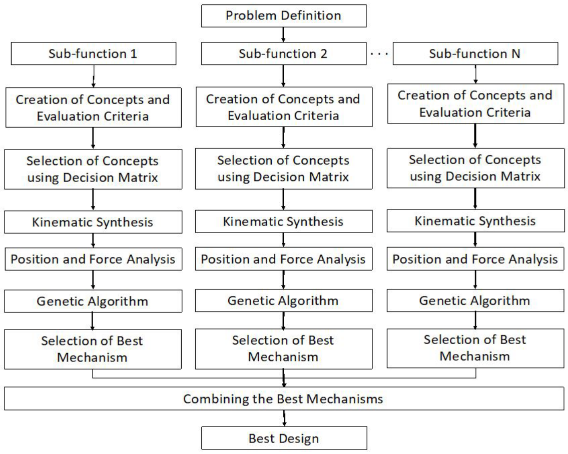

Figure 1. The systematic design methodology.

optimized using Genetic Algorithm (GA) method [20, 21]. Finally, based on the optimization results, the best mechanism for each sub-function is selected and then combined to form the final design.

Design of the Refueling Door Mechanism

In this paper, the problem is to design an in-flight refueling door mechanism for a flying boom type refueling system. The design of the door actuation mechanism is performed based on the design methodology presented. The main function is divided into two sub-functions: door opening, and actuation as shown in Fig. 2. The door might open later-ally or 180° and the actuation might be via a rotary actuator or a linear actuator. Different concepts are created for each sub-function they are evaluated by weighted decision matrix method. The selected concepts are synthesized using graphical methods [9-10]. After synthesis of the mechanisms, position and force analyses are performed. Each mechanism is then optimized using multi-objective GA method. The best concepts for each sub-problem is combined to form the final design.

THEORY/CALCULATION

Conceptual Design

In the conceptual design, different concepts are created for the door opening and actuation sub-functions.

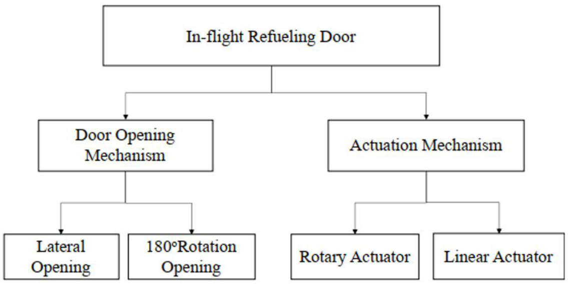

Figure 2. In-flight refueling door sub-functions and different ways of satisfying these functions

Door Opening Mechanism Concepts

Due to aerodynamic effects and space limitations, the door is designed such that it has two parts and opens symmetrically outwards. For the sake of convenience, only one part of the door and its opening mechanism is shown here. Nine different concepts are developed (only five of these concepts are depicted in Fig. 3); some opening the door laterally and some opening 180°. These concepts are: D1: A four-bar mechanism. The door opens laterally, and it is rigidly connected to the coupler link (Fig. 3a). D2: A Watt I type six-bar mechanism. The door opens laterally (Fig. 3b). D3: A Stephenson III type six-bar mechanism (Fig. 3c). The door opens laterally. D4: A four-bar mechanism. The door opens up to 180° and connected to the follower link. D5: A four-bar mechanism. The door opens 180° and connected to the coupler link (Fig. 3d). D6: A Watt I type six-bar mechanism. The door opens up to 180°. D7: A Watt I type six-bar mechanism. The door opens up to 180°. D8: A Watt II type six-bar mechanism. The door opens up to 180°. D9: An invisible hinge mechanism [5, 6]. Revolute and sliding joints are used. The door opens up to 180° (Fig. 3e).

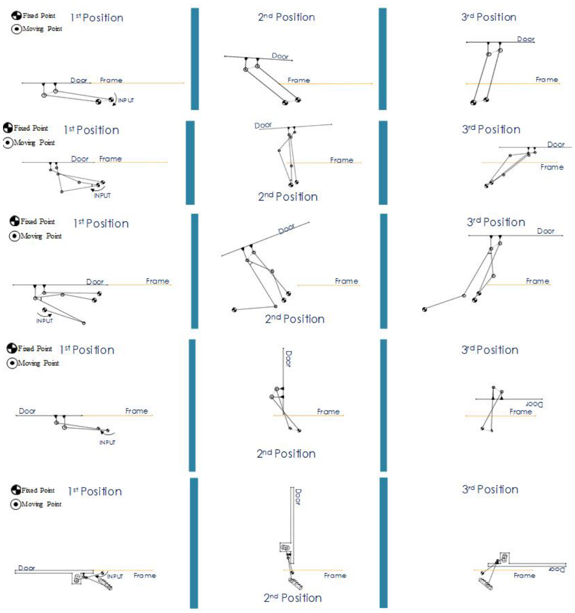

Figure 3. Optional concept sets for door opening mechanism. Only five concepts are shown. (a) D1, (b) D2, (c) D3, (d) D5 and (e) D9.

Actuation Mechanism Concepts

Eight different concepts (one with rotary actuator and the rest with linear actuators) are developed for the actuation mechanism (five of these concepts are depicted in Fig.4). Some of these concepts are designed to be 1-DOF and the others are 2-DOF.

Those optional concepts are A1: A slider-crank mechanism actuated by two linear motors. The followers are connected to the door opening mechanism (Fig. 4a). A2: A planar mechanism actuated by single linear motor as in [7]. The followers are connected to the door opening mechanism as in A1. The slider-crank is in the form of piston-cylinder arrangement. A3: A planar mechanism actuated by a linear motor formed by two four-bar and an inverted slider crank mechanism as in [8]. The slider-crank is arranged as in A2. The followers of the four-bar mechanism are connected to the door opening mechanism and cranks are used to transmit the motion as in A2 (Fig. 4b). A4: A planar mechanism actuated by a single linear motor formed by a Watt II type six-bar and an inverted slider-crank mechanism. The six-bar mechanism transmits the motion to the other side to provide symmetrical motion. The followers are connected to the door opening mechanism as in A1 (Fig. 4c). A5: A planar mechanism actuated by a single linear motor

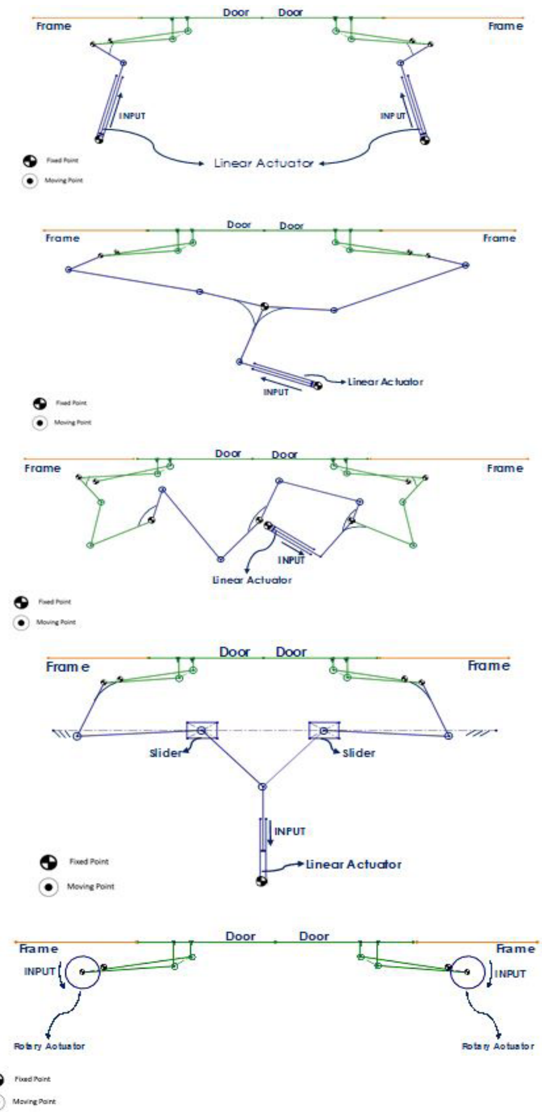

Figure 4. Optional concept sets for the actuation mechanism. Only five concepts are shown. All concepts are integrated to D5 for demonstration without the loss of generality. (a) A1, (b) A3, (c) A4, (d) A6, (e) A8.

Table 1. Weighted decision matrix of the door opening concepts. Simplicity, maintainability, simplicity of assembly, reliability, rigidity, mobility and design flexibility, space utilization, and force characteristics are used as the evaluation criteria. Weighting factors are given in the second column. The other columns represent the average score of six engineers for the corresponding evaluation criteria.

Evaluation Criteria	Wgh.	D1	D2	D3	D4	D5	D6	D7	D8	D9
Simplicity	0.161	8.7	6.0	5.5	7.5	8.1	5.3	5.1	4.7	3
Maintain.	0.125	8.8	5.3	6.3	7.8	8.2	5.7	5.3	4.6	3.3
Simp. of Assembly	0.089	8.8	5.3	5.7	8.3	8.1	5.1	5.2	5.2	4.0
Reliability	0.143	8.8	7.0	6.7	8.7	7.9	5.8	6.3	6.0	4.6
Rigidity	0.125	8.2	5.9	6.1	7.0	6.8	5.8	5.5	5.5	3.8
Mob./Flex.	0.089	5.3	7.7	7.4	5.7	5.8	7.6	7.8	7.2	6.3
Space Util.	0.107	2.1	3.9	8.4	4.7	2.8	8.3	9.4	1.0	10
Force Cha.	0.161	9.1	9.9	1.0	6.1	7.7	10	7.8	2.4	9.8
TOTAL	1	**7.8**	6.5	5.6	7.0	**7.1**	6.7	6.5	4.5	5.6

formed by a four-bar, a Watt II type six-bar, and an inverted slider-crank mechanism. The symmetrical motion is obtained by the six-bar mechanism as in A4. The followers of the four-bar and six-bar mechanisms are connected to the door opening mechanism. A6: A planar mechanism actuated by a single linear motor formed by two double slider-crank mechanisms. Symmetrical motion is provided by the two sides of the slider-crank mechanisms. The output links of the slider-crank mechanism is connected to the door opening mechanism (Fig. 4d). A7: A planar mechanism actuated by two motors using a rack-pinion arrangement. Two motors are synchronously driven. A8: Two rotary actuators are directly coupled to the door opening mechanism (Fig. 4e).

Evaluation of Concepts

The door opening and actuation concepts are evaluated separately based on different evaluation criteria. Weighting factors of these evaluation criteria are determined according to problem needs. These weighting factors can be changed for different problems. The evaluation is performed by six experienced engineers based on a value scale from 0 to 10.

Table 2. Weighted decision matrix of the actuation concepts. Simplicity, maintainability, cost, long life, simplicity of assembly, reliability, rigidity, design flexibility and space utilization are the design criteria. Weighting factor is given in the second column. The following columns show the average scores of six engineers for the corresponding evaluation criteria.

Evaluation Criteria	Wgh.	A1	A2	A3	A4	A5	A6	A7	A8
Simplicity	0.148	8.5	8	7	5	5.2	5.2	6.3	9
Maintain.	0.115	7.8	7.3	7	5.2	5.3	4.7	4.2	8
Cost	0.098	6.5	6.5	7.4	5.5	5.5	4.7	3.2	6.2
Serv. Life	0.131	8.2	7	7.7	5.7	5.7	4.8	4.5	7.3
Simp. of Assembly	0.082	8.2	7.8	7.3	4.8	4.8	4.5	4.8	8.8
Reliability.	0.131	8.2	7.2	7.2	6	6	4.8	5.2	8.5
Rigidity.	0.115	8.2	6.5	6.8	5.2	5.2	5.5	5.5	9
Flex.	0.082	7.3	6.5	6.7	6	6	5.7	6	8
Space Util.	0.098	7.3	5.7	6.3	5.2	5.2	4.7	7.3	9.2
TOTAL	1	**7.9**	7	**7.1**	5.4	5.4	4.9	5.2	**8.2**

The criteria for the door opening concepts are simplicity, maintainability, simplicity of assembly, reliability, rigidity, mobility and design flexibility, space utilization, and force characteristics. After the evaluation, shown in Table 1, D1 and D5 are selected as the best concepts.

The evaluation criteria for the actuation concepts are simplicity, maintainability, cost, long life, simplicity of assembly, reliability, rigidity, design flexibility and space utilization. After the evaluation, as shown in Table 2, A1, A3, and A8 are chosen to be the best concepts.

Kinematic Synthesis and Analysis

Kinematic synthesis, kinematic analysis and force analysis of all the best concepts D1, D5, A1 and A3 are performed (kinematic synthesis and analysis have not been performed for A8 since it only consists two rotary actuators directly coupled to the driving link of the door mechanism). Since the procedure is the same for all the mechanisms, only the kinematic synthesis and the analysis of concept D1 are presented in this paper.

Kinematic Synthesis of D1

D1, shown in Fig. 5, is designed using two position graphical synthesis method [9].

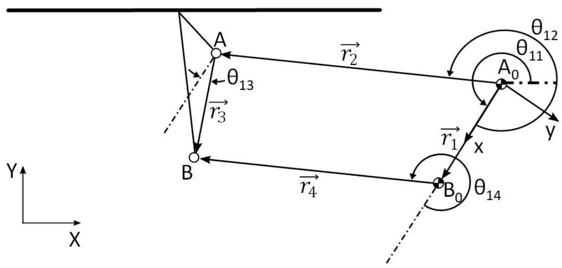

Figure 5. Schematic drawing of concept D1.

First, the initial and final positions of the door are selected then two moving points are determined as A1, A2, B1, and B2. Then these points are connected, and two perpendicular lines are drawn at the mid-points of A1A2 and B1B2 as shown in Fig. 6. Two fixed pivot points, A0 and B0 are selected at any place on the perpendicular lines. The mechanism is then synthesized joining A0, B0, B1 and A1. The parameters found after the synthesis are shown in Table 3.

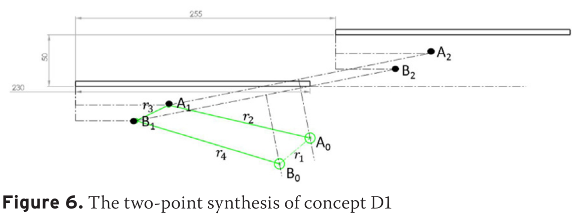

Figure 6. The two-point synthesis of concept D1

Table 3. Calculated parameters for concept D1.

1	2	3	4	5	6
r_1 (mm)	r_2 (mm)	r_3 (mm)	r_4 (mm)	Q_{12}(deg)	Q_{11}(deg)
39.05	142.24	35.32	146.65	-52.73	219.81

Kinematic Analysis of D1

Freudenstein's equation is used to perform the kinematic analysis. The loop closure equation in complex notation is given by:

$$r_2 e^{i\theta_{12}} + r_3 e^{i\theta_{13}} - r_4 e^{i\theta_{14}} - r_1 = 0 \qquad (1)$$

By solving Eq. (1), the unknown joint variables are found as:

$$\theta_{13} = 2 \cdot atan\left(\frac{(-B + \sigma\sqrt{B^2 - 4AC}}{2A}\right) \qquad (2)$$

$$\theta_{14} = 2 \cdot atan\left(\frac{r_2 \sin\theta_{12} + r_3 \sin\theta_{13}}{r_2 \cos\theta_{12} + r_3 \cos\theta_{13} - r_1}\right) \qquad (3)$$

where

$$A = (K_1 + K_2 \cos\theta_{12} - K_3 + \cos\theta_{12}) \qquad (4)$$

$$B = (-2 \cdot \sin\theta_{12}) \qquad (5)$$

$$C = (K_1 + K_2 \cos\theta_{12} + K_3 - \cos\theta_{12}) \qquad (6)$$

$$K_1 = \frac{r_4^2 - r_1^2 - r_2^2 - r_3^2}{2 r_3 r_2} \qquad (7)$$

$$K_2 = \frac{r_1}{r_3}, K_3 = \frac{r_1}{r_2}, \sigma = -1 \qquad (8)$$

Eq. (2) and (3) are used to find the joint variables for every crank angle.

Force Analysis of D1

In the refueling applications high accelerations are not required, furthermore link masses are relatively small. Therefore, inertial forces are ignored, and quasi-static force analysis is performed to calculate the required driving force. An approximate value is taken for the external force and the same external force is applied to all the door opening mechanisms. The free-body diagram of each link is drawn (not shown), and unknown forces are found. The system is assumed to be in equilibrium under the action of the external force, F, and the driving torque, T_{input} as shown in Fig. 7.

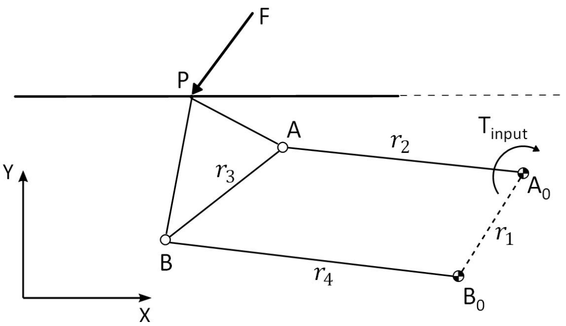

Figure 7. External force and the driving torque acting on D1.

The equilibrium equations for each link are not presented here (refer to [11] for details). Only the final results are given. In matrix form to find the forces:

$$[x] = [A]^{-1} \cdot [b] \qquad (9)$$

where

$$[x] = \begin{bmatrix} F_{23x} \\ F_{23y} \\ F_{43} \\ T_{input} \end{bmatrix} \qquad (10)$$

$$[A] = \begin{bmatrix} 1 & 0 & \cos(\theta_{11} + \theta_{14}) & 0 \\ 0 & 1 & \sin(\theta_{11} + \theta_{14}) & 0 \\ r_3 \sin(-\gamma) & r_3 \sin(\frac{\pi}{2} - \gamma) & 0 & 0 \\ -r_2 \sin(\pi - \theta_{11} - \theta_{12}) & -r_2 \sin(\frac{3\pi}{2} - \theta_{11} - \theta_{12}) & 0 & 0 \end{bmatrix} \qquad (11)$$

$$[b] = \begin{bmatrix} -F\cos(\theta_f) \\ -F\sin(\theta_f) \\ -M - F\frac{r_3}{2}\sin(\theta_f - \gamma) \\ 0 \end{bmatrix} \qquad (12)$$

The variable vector, [x] can be calculated using the known [A] matrix and [b] to find the unknown forces and the required driving torque.

Kinematic Synthesis of A1

An iterative graphical approach [10] is used to synthesize the concept A1, as shown in Fig. 8.

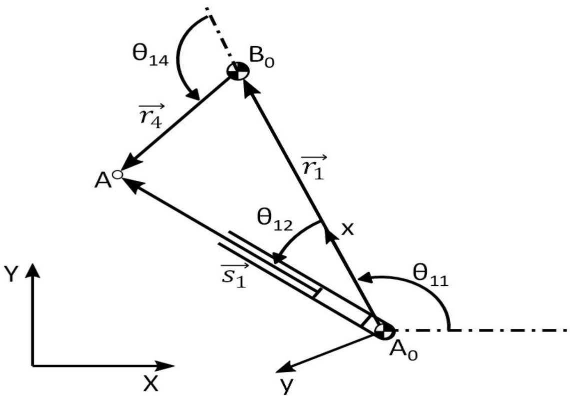

Figure 8. Schematic drawing of Concept A1.

The required rotation of the output link should be equal to the rotation of the drive link of the door opening/closing mechanism. Therefore, without loss of generality concept D1 is used for the synthesis of A1. The required driving link rotation is calculated as -131.96°. After using the graphical synthesis as in Sect. 4.1, the parameters of A1 are found as in Table 4.

Table 4.Calculated parameters for concept A1.

1	2	3	4	5
r_1 (mm)	r_4 (mm)	s_1 (mm)	Q_{11}(deg)	Δ_{stroke} (mm)
230.00	70.00	174.00	132.50	123.70

Kinematic Analysis of A1

Freudenstein's equations are used to perform the kinematic analysis of A1. The loop closure equation in complex form is given by:

$$s_1 e^{i\theta_{12}} - r_1 - r_4 e^{i\theta_{14}} = 0 \tag{13}$$

Solving the loop closure equation yields:

$$\theta_{12} = 2 \cdot \mathrm{atan}\left(\sigma\sqrt{-\frac{C}{A}}\right) \tag{14}$$

$$\theta_{14} = \mathrm{atan}\left(\frac{s_1 \sin\theta_1}{s_1 \cos\theta_1 - r_1}\right) \tag{15}$$

where all the required variables are defined in Eq. 4-8.

Force Analysis of A1

Quasi-static force analysis is performed to find the required driving force. The freebody diagram of each link is drawn (not shown, refer to [11]), and all the unknown forces acting on the links are calculated. The external torque is assumed to be acting from the door opening mechanism as shown in Fig. 9.

The forces can be calculated as in Eq. 9 using:

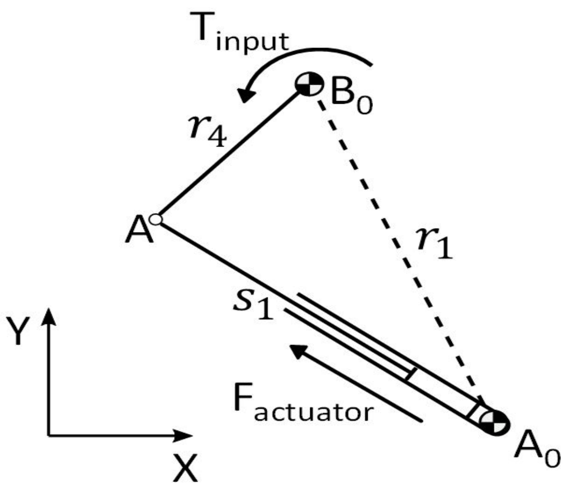

Figure 9. External torque and input force acting on A1

$$[x] = \begin{bmatrix} F_{14x} \\ F_{14y} \\ F_{34} \\ F_{actuator} \\ F_{12x} \\ F_{12y} \end{bmatrix} \tag{16}$$

$$[A] = \begin{bmatrix} 1 & 0 & \cos(\theta_{11}+\theta_{12}) & 0 & 0 & 0 \\ 0 & 1 & \sin(\theta_{11}+\theta_{12}) & 0 & 0 & 0 \\ 0 & 0 & r_4\sin(\theta_{12}-\theta_{14}) & 0 & 0 & 0 \\ 0 & 0 & 1 & -1 & 0 & 0 \\ -1 & 0 & -\cos(\theta_{12}+\theta_{11}+\pi) & 1 & 0 & 0 \\ 0 & 1 & -\sin(\theta_{12}+\theta_{11}+\pi) & 0 & 0 & 0 \end{bmatrix} \tag{17}$$

$$[b] = \begin{bmatrix} 0 \\ 0 \\ -T_{input} \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{18}$$

Optimization of the Mechanisms

All the door opening (D1, D5) and actuation mechanisms (A1, A3, A8) are optimized using the GA method with MATLAB software using the parameters shown in Table 5. These GA parameters are determined by using trial-error method after several iterations. Best results are obtained by using GA parameters, which are given in the Table 5. Different parameters can be used for different problems.

Table 5.GA parameters used during optimization

Population Size	Maximum Generation	Crossover Function	Crossover Fraction	Mutation Function
2250	500	Heuristic 1.2	0.8	Uniform 0.1

Only the optimization of D1 and A1 are presented in this paper (refer to [11] for the rest).

Optimization of D1

The door opening/closing mechanism, D1 is optimized based on a cost function and several geometric constraints

Variables: r1, r2, r3, r4, θ_{12}, initial, Δ_θ. Δ_θ is the angle per step and the other variables are shown in Fig. 5.

Cost Function: (a) A laterally opening and closing door is desired, therefore the first cost function is:

$$f(1) = \left|\theta_{3,initial} - \theta_{3,final}\right| \cdot G \tag{19}$$

(b) In order to minimize the input torque, the second cost function is defined as:

$$f(2) = \max(T_{input}) \cdot G \tag{20}$$

Where G is a parameter set to 1 if the constraints are satisfied, if not it is set to a very large number to increase the cost and penalize the solution.

Constraints: (a) The first constraint is related to the initial orientation of the mechanism given as:

$$175° \geq \theta_{11} + \theta_{1,2,initial} \geq 165° \tag{21}$$

$$absolute(\theta_{12,initial} - \theta_{13,initial}) > 5° \tag{22}$$

(b) The second constraint is related to minimum clearance being greater than 200mm when the door is open. It is defined as:

$$fr_2 \cdot \cos(\theta_{11} + \theta_{12,final}) - r_2 \cdot \cos(\theta_{11} + \theta_{12,initial}) - 200mm > 0 \tag{23}$$

(c) The third constraint is related to the maximum allowable space envelope that is a rectangle having a width of 300mm and a height of 100mm. The point A0 is fixed in space. The constraint about the initial position of the fixed point B0 is given by:

$$r_1 \cdot \sin(\theta_{11}) > -50mm \tag{24}$$

Constraints related to the initial and final position of point A to prevent collisions are:

$$r_2 \cdot \sin(\theta_{11} + \theta_{12,initial}) < 30mm \tag{25}$$

$$120mm > r_2 \cdot \sin(\theta_{11} + \theta_{12,final}) > 70mm \tag{26}$$

Constraints related to initial and final position of point B to prevent collisions are:

$$r_1 \cdot \sin(\theta_{11}) + r_4 \cdot \sin(\theta_{11} + \theta_{14,initial}) < 30mm \tag{27}$$

$$r_1 \cdot \cos(\theta_{11}) + r_4 \cdot \cos(\theta_{11} + \theta_{14,initial}) > -180mm \tag{28}$$

$$170mm > r_1 \cdot \sin(\theta_{11}) + r_4 \cdot \cos(\theta_{11} + \theta_{14,initial}) > 70mm \tag{29}$$

The upper and lower boundaries of the variables are given in Table 6.

Table 6.GA parameters used during optimization.

	1	2	3	4	5	6	7
	r_1 (mm)	r_2 (mm)	r_3 (mm)	r_4 (mm)	Q_{12} (deg)	Q_{11} (deg)	Δ_θ (deg)
Minimum	20	100	30	100	-65	180	-2.90
Maksimum	100	200	125	200	65	230	-1.60

At each iteration step of the optimization, Eq. (2-3) are used to find the unknowns, θ_{13}, θ_{14} then Eq. (9) and Eq. (10-12) are used to find the forces. The input variable θ_{12}, k at every iteration, k is calculated as:

$$\theta_{12,k} = \theta_{12,initial} + \Delta_\theta \cdot (k-1) \tag{30}$$

where k=2, 3…, 50.

Results: The computation time of the optimization is 637s (Core i7-4700HQ, 2.40GHz CPU) and the maximum required input torque is calculated as 24.48Nm (20% decrease with respect to the graphical synthesis). The parameters of the mechanism are found as in Table 7.

Table 7.Parameters of D1 after optimization

1	2	3	4	5	6	7
r_1 (mm)	r_2 (mm)	r_3 (mm)	r_4 (mm)	Q_{12}(deg)	Q_{11}(deg)	Δ_θ (deg)
42.2	119.89	36.57	139.39	-63.31	228.92	-2.40

An MSC ADAMS model is created to verify the calculation of the input torque as shown in Fig. 10. The error is found to be less than 0.0016%.

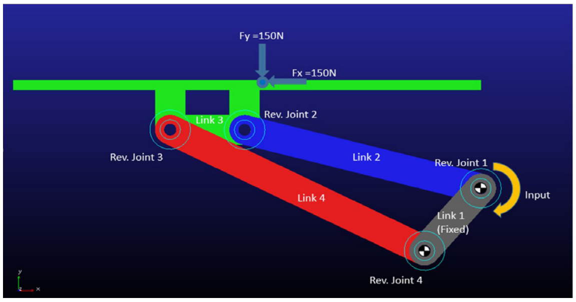

Figure 10. MSC ADAMS model of the optimized D1.

Optimization is also performed for D5 (not shown). Maximum input torque is selected as the selection criteria and it is found that maximum torque for D5 is 43% greater than D1, as shown in Figure 11. Therefore, for the door opening mechanism the best concept is D1.

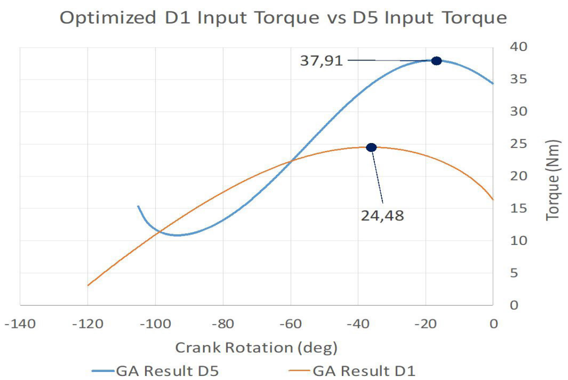

Figure 11. Input torques for the optimized D1 and D5 mechanisms versus the crank angle calculated using MSC ADAMS software.

Optimization of A1

The mechanism A1 is optimized based on the cost function and the constraints. In the optimization of A1, without loss of generality, D1 is used as the door opening/closing mechanism.

Variables: r1, r4, s0, θ_{11}, Δ_{stroke}. Δ_{stroke} is the stroke per step and the other variables are shown in Fig. 8.

Cost function: (a) The mechanism should be able to rotate the crank of the door opening/closing mechanism to the required degree, $(\Delta_{\theta_{12}})_{Design1}$, calculated before.

The first cost function is:

$$f(1) = abs(abs(\theta_{14,initial} - \theta_{14,final}) - (\Delta_{\theta_{12}})_{Design1})G \tag{31}$$

(b) In order to decrease the input force, the second cost function is defined as:

$$f(2) = \max(F_{actuator}) \cdot G \tag{32}$$

Constraints: (a) The first constraint is related to the transmission angle given as:

$$140° \geq \mu \geq 20° \quad (33)$$

(b) The second constraint is related to the space envelope defined as a rectangle with a width of 600mm and a height of 100mm. The pivot, A0, of the door opening mechanism is fixed in space and the following constraints are obtained:

$$r_1 \cdot \sin(\theta_{11}) < 300mm \quad (34)$$

$$r_1 \cdot \cos(\theta_{11}) < 170mm \quad (35)$$

$$s_{1,final} \cdot \sin(\theta_{11} + \theta_{12,final}) - r_1 \cdot \sin(\theta_{11}) < 45 \quad (36)$$

The upper and lower boundaries of the variables are shown in Table 8.

Table 8.Upper and lower boundaries of variables for A1.

	1	4	5	6	7
	r_1 (mm)	r_4 (mm)	s_0(mm)	Q_{11}(deg)	Δ_{stroke} (mm)
Minimum	50	30	50	50	1
Maksimum	300	130	250	200	3.2

At each iteration, θ_{13} and θ_{14} are found using Eq. (14-15) and the forces are found using Eq. (9) and Eq. (16-18). The input variable, $s_{1,k}$ is calculated at every iteration, k as:

$$s_{1,k} = s_0 + \Delta_{stroke} \cdot (k-1) \quad (37)$$

Where, k=2,3…,50.

Results: The computation time of the optimization is 193s (Core i7-4700HQ, 2.40GHz CPU) and the maximum required input force is found to be 313.18N (42% less than the graphical synthesis). The variables are found as in Table 9. Simulation of the synthesized mechanisms A1 and D1 in MS Excel are shown in Figure 12.

Table 9.Optimized variables of A1

1	2	3	4	5
r_1 (mm)	r_4 (mm)	s_0(mm)	Q_{11}(deg)	Δ_{stroke} (mm)
249.99	78.39	189.94	131.44	2.65

An MSC ADAMS model, shown in Fig. 13, is created to verify the calculation of the input force and the error is found to be less than 0.1%. Optimization is also performed for A3 and A8 (not shown). Power is used as the selection criteria since it is the determining factor for the size of the actuator so that it must be minimized to decrease the size and weight of the actuator. The other point is when power is used, mechanisms with linear actuators (A1, A3) and rotary actuators (A8) can be com-pared. The door opening time is assumed to be 5s and the required instantaneous power is calculated for the optimized

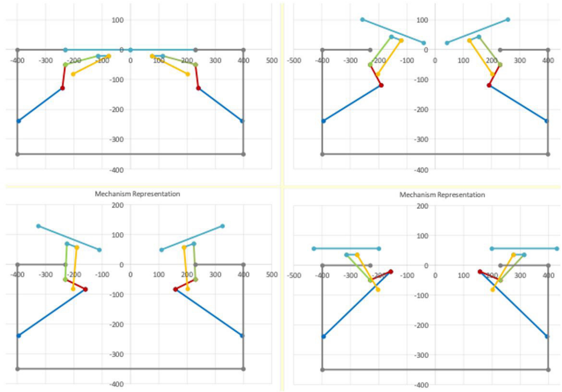

Figure 12. Simulation of concept A1 combined with D1 using MS Excel.

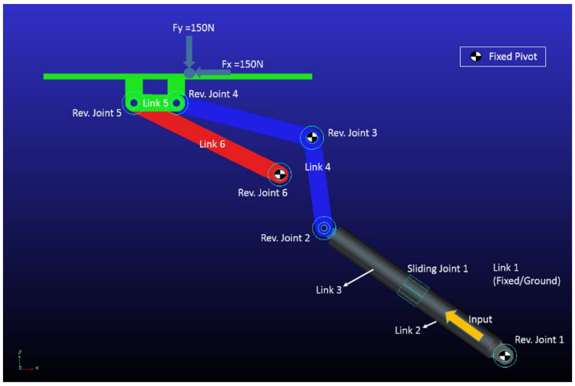

Figure 13. MSC ADAMS model of concept A1 and D1.

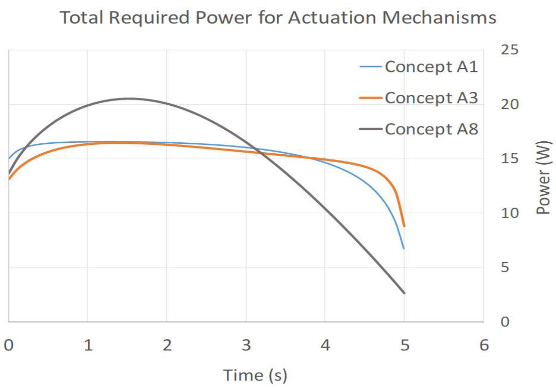

Figure 14. The required power versus time for the optimized A1, A3 and A8 mechanisms calculated with MSC ADAMS software.

A1, A3, and A8 mechanisms. The results are depicted in Fig. 14. A8 requires the highest power, whereas the power requirement of A1 and A3 are almost the same. To select the best alternative, simplicity and rigidity are considered as the evaluation criteria and A1 is selected as the best alternative.

DISCUSSION AND CONCLUSION

In this paper, the preliminary design of an in-flight refueling door actuation mechanism for flying boom type refueling system is presented. A systematic design methodology is followed. The main function is divided into two sub-functions:

H. Akman et al./ Hittite J Sci Eng, 2022, 9 (1) 27–36

door opening and actuation sub-functions. Different concepts are developed for each sub-function based on the requirements and constraints. A heuristic approach is used for evaluating the concepts based on the evaluation criteria. After the evaluation process, two concepts are selected for the door opening/closing mechanism, and three concepts are selected for the actuation mechanism. Kinematic synthesis for one concept for each sub-function is performed by using graphical methods to obtain suitable mechanisms. In addition, the obtained results are used to determine the upper and lower boundaries of design parameters for the optimization process. Thereafter, position analysis is executed by using Freudenstein's equation to check the motion of the synthesized mechanisms. Additionally, force analysis is performed to obtain the required actuation force and joint forces of the concepts. Finally, chosen concepts are optimized by using the multi-objective Genetic Algorithm. Be-fore the optimization process, the lower and upper boundaries of design parameters, objective functions, and constraints are specified. Then, for each sub-problem concept, the optimization process is performed, and optimized concepts are compared with each other to select the best concept. In this study, Concept D1 and Concept A1 are chosen as the best concepts. After determining the best concepts for each sub-problem, these concepts are combined to obtain the preliminary design of the in-flight refueling door actuation system mechanism.

As a future work, aerodynamic analysis can be performed to estimate the external force more precisely. The effect of inertial forces can be taken into consideration in the force analysis. Based on the force analysis, detailed design of the mechanism can be performed, and a prototype can be built to validate the design.

NOMENCLATURE

Δ_θ : Angle per step of the Door Concepts
Δ_{stroke}: Stroke per step
r_i : Link Lengths
θ_{ij} : Link Angles
μ : Transmission Angle
T : Torque
F : Force
M : Moment

CONFLICT OF INTEREST

Authors approve that to the best of their knowledge, there is not any conflict of interest or common interest with an institution/organization or a person that may affect the review process of the paper.

AUTHOR CONTRIBUTION

All sections including conceptualisation, methodology, software, analysis, writing, review and editing were equally organised and performed by Hasan Akman, Ali Emre Turgut and Hakan Çalışkan.

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