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Modeling of Forced Convection Heat Transfer from the Swing Plate on an Industrial Robot

Qi Lu, *Member, IEEE* and Dexian Wang

***Abstract*— The swing plate on an industrial robot is a sort of disk-like component through which an amount of heat generated inside the robot axis 1 drivetrain will transfer to the ambient. This raises a need on modeling and design of the swing plate from a thermal perspective. However, it still remains questionable whether or not the thermal models developed for conventional type of rotating disks are applicable to the swing plate on a robot. This is because, not like the conventional spinning disk which rotates in a single direction continuously, the swing plate is driven back-and-forth in a bi-directional mode, called rotary reciprocation. This paper aims to understand the convection heat transfer properties of such a swing plate by conducting a set of experiments with a plate undergoing rotary reciprocation. A set of empirical models are then assessed by comparing against the experimental results, and eventually, the model that best describes forced convection of the swing plate is obtained.**

I. Introduction

Thermal issue of an industrial robot has been of great interest for long time because unexpected temperature rise will not only have a negative effect on robot accuracy [1-3], but also result in failure of the temperature-sensitive components in a robot, for instance, motors and sealing [4,5]. This is particularly true for the cases of small robots which are usually limited in size, but having higher energy density than the large ones. This will result in the fact that temperature rise becomes more critical to small robots. Because of this, there exists a need in robotics society to better understand the heat transfer properties of an industrial robot, so that it will be possible to analyze, design and optimize the robot from a thermal perspective.

Despite of the century-long history of the heat transfer research, it still remains practically difficult to conduct quantitative analysis of robot heat transfer due to its unique motion pattern. Specifically, each axis of a robot always repetitively rotates back-and-forth in a bi-directional fashion, called rotary reciprocation. Such a motion pattern is quite different from the study objects in the conventional heat transfer research most of which are considered as stationary or spinning continuously in one direction such as a spinning hard disk [6]. The difference between the motion patterns makes it questionable to apply the forced convection models developed in the conventional studies directly onto the analysis of industrial robots. The presented study tries to look into the forced convection modeling of an industrial robot by taking a swing plate on the robot as an example.

Qi Lu is with Mechtronics Department, ABB Corporate Research China, Shanghai, 201319, P.R. China (Tel: +86(21)6129-8459; fax: +86(21)6105-6905; e-mail: qi.lu@cn.abb.com).

Dexian Wang is a master student of Shanghai University, Shanghai, 20072, P.R. China (e-mail: kangjiayiren@shu.edu.cn).

As shown in Figure 1, a swing plate is a disk-like component mounted on the output end of the axis 1 drivetrain of a robot. It is driven by the drivetrain, and brings along with it the other axes and arms of the robot rotating about the robot base. It should be noted that the swing plate always performs rotary reciprocation in all kinds of robot application. In practice, the swing plate has to transfer to ambient a great amount of heat energy generated in axis 1 drivetrain; otherwise the drivetrain will be at risks of overheating. This leads to a desire to model and design the swing plate from a thermal perspective. However, such attempt is not feasible before we quantitatively understand the heat transfer, especially the forced convection, from the swing plate when undergoing rotary reciprocation.

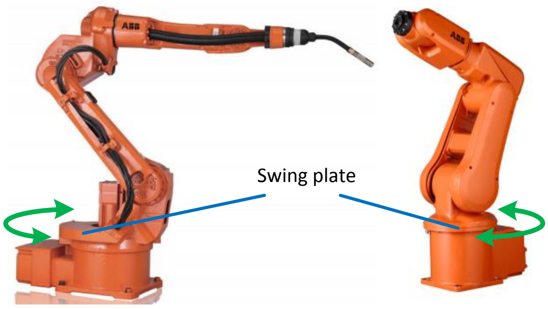

Figure 1 The swing plates on industrial robots

This paper aims to addressing the forced convection heat transfer from a swing plate. Instead of directly measuring the heat transfer coefficient of the swing plate, the presented study begins with answering a question that whether or not the existing empirical models developed for the forced convection from a continuously-spinning disk are still valid for a disk undergoing rotary reciprocation. The rest of the paper is organized as follows. Section II briefly reviews the existing study on forced convection in the scenario of continuous spinning. They will be slightly revised and then implemented into a thermal network model for the swing plate. In Section III, a set of experimental studies are conducted with a disk undergoing rotary reciprocation, just like a swing plate in a robot. A discussion will be given in Section IV by comparing the experimental results against the model outcomes. Finally, the paper will be concluded in Section V.

II. Models

In this study, the air flow around the swing plate is assumed laminar in large scale because the swing plate rotates at a rather low speed in practice. Therefore, only the models of laminar flow will be discussed and implemented.

A. Existing empirical models for forced convection in continuous spinning

Numerous effects have been spent on the theoretical and experimental studies on the forced convection heat transfer from a continuously-spinning disk. As in many other studies on convection, the key is to determine an appropriate dimensionless Nusselt number Nu and then the convective heat transfer coefficient h, as shown Eq. (1) [7].

$$h = \lambda Nu/r \tag{1}$$

where λ represents the conductivity of fluid (i.e. still air in this case); and r is the radius of the local position towards to the spinning center. Wagner [8] proposed an approximated solution to determine the local Nusselt by the local Reynolds number Re as

$$Nu = KRe^n \tag{2}$$

$$Re = \omega r^2/\nu \tag{3}$$

where $n = 0.5$ for laminar flow; ω refers to the angular velocity of the disk; and ν is the kinematic viscosity of the fluid. It was further proved by Millsaps and Pohlhausen [9] that ideally the parameter $K = 0.326$ for the fluid with Prandtl number $Pr = 0.71$.

Since then, experimental studies has also been conducted to address the forced convection form a spinning disk. Basically, two types of approaches were adopted in the studies. One is to calculate convective heat transfer coefficient and Nusselt number by measuring temperature of the disk with techniques such like inferred camera [10, 11]. The other is to study the convective mass transfer on the disk and then deduce Nusselt number based on Sherwood number by utilizing the analogy between heat and mass transfer [12-14].

It is interesting that studies showed that Eq. (1) actually agrees with the experimental data quite well [14, 20]. The Nusselt number in forced convection of a spinning disk does follow the behavior described in Eq. (1) with $n = 0.5$ in laminar region. Yet, the value of K varies in different studies as shown in Table 1

TABLE 1 EXISTING MODEL OF NUSSULT NUMBER BASED ON EQ.(1)

K	*n*	*Pr* [a]	*Remark*
0.326	0.5	0.71	[9]
0.330	0.5	0.72	[15,16]
0.333	0.5	N/A	[10]
0.335	0.5	0.74	[1,11]
0.341	0.5	0.72	[17]
0.370	0.5	0.71	[14, 18]
0.379	0.5	0.71	[6]
0.384	0.5	0.71	[13]
0.417	0.5	0.71	[14,19]

a. Pr stands for Prandtl number which is a physical property of the fluid. $Pr = 0.71$ for still air at room temperature

In addition, both theoretical and practical studies indicate that K value in Eq. (1) seems relevant to the Prandtl number Pr [20,16]. In general, K rises up when Pr increases. Equations (4) ~ (8) have been proposed respectively to quantitatively describe this relation.

$$K = 0.308 \times (2Pr)^{1/2} \qquad [21]\ (4)$$

$$K = \frac{0.585}{0.6/Pr + 0.95/Pr^{1/3}} \qquad [22]\ (5)$$

$$K = \frac{0.6109Pr}{(0.5301 + 0.3996Pr^{1/2} + Pr)^{2/3}} \qquad [23]\ (6)$$

$$K = \frac{0.6Pr}{(0.56 + 0.26Pr^{1/2} + Pr)^{2/3}} \qquad [24]\ (7)$$

$$\frac{1}{K} = \left[\left(\frac{1}{0.88447Pr}\right)^{1.077} + \left(\frac{1}{0.62048Pr^{1/3}}\right)^{1.077}\right]^{1/1.077} \qquad [25]\ (8)$$

In short, Eq. (1) is found to well represent convection heat transfer of a continuously-spinning disk in laminar region. Various K values have been found in the literature while $n = 0.5$ has been agreed in both thematically and experimental studies.

B. Simulation model for a swing plate

For a swing plate on an industrial robot, heat energy will transfer out through the plate to its neighbor parts and ambient by means of conduction, convection and radiation. However in this study, we will model only convection and radiation to ambient and leave conduction between the plate and other parts out of the discussion. Such simplification will not invalid the study. This is because we wish to understand the effects caused by unique motion pattern of the plate, but conductive heat transfer is not motion-dependent compared to forced convection. In addition, it is always possible to establish a throughout heat transfer model of a swing plate considering all three heat transfer modes as soon as its forced convection property is fully revealed.

With respect to the law of energy conservation, we will have the mathematical model for the swing plate in steady state as follows.

$$q = q_{conv} + q_{rad} \tag{9}$$

with

$$q_{conv} = hA(T - T_{amb}) \tag{10}$$

$$q_{rad} = \sigma A\epsilon(T^4 - T_{amb}^4) \tag{11}$$

where q, q_{conv}, and q_{rad} are respectively the total energy, energy transferred by forced convection, and by radiation; T and T_{amb} represent the temperature of the swing plate and the ambient temperature. In Eqs. (9) ~ (11), σ the Stefan-Boltzmann constant, which is $5.67 \times 10^{-8} W/m^2K^4$; and ϵ the emissivity determined by materials and surface properties. In addition A stands for the equivalent area of heat transfer.

One may notice that Eqs. (1) ~ (3) are not directly implementable to Eq. (10) as the angular speed of the swing plate is not a constant, but varies periodically. To simplify the model, we introduce the root-mean-square angular velocity ω_{rms} which is root-mean-square value of the angular speed in one period of motion. Consequently, Eqs. (1) ~ (3) have to be slightly revised correspondingly.

$$h = \lambda Nu_{av}/R \tag{12}$$

$$Nu_{av} = KRe_{rms}^{0.5} \tag{13}$$

$$Re_{rms} = \omega_{rms}R^2/\nu \tag{14}$$

where Nu_{av} and Re_{rms} refer to average Nusselt number and Reynolds number defined by ω_{rms}; R stands for the radius of the swing plate. In Eqs. (13) ~ (15), average Nusselt number Nu_{av} is employed instead of local Nusselt number Nu. Yet, such simplification should be valid. This is because the air around the swing plate is laminar in our cases, which will result in uniform heat transfer on the surface. This can be proved by submitting Eqs. (1) and (2) into Eq. (12), and has been shown in literature [10] and in our experiments. Thus there is no need to differentiate local and average Nusselt number.

Then, one can easily implement Eqs. (9) ~ (14) as a thermal network model as illustrated in Fig. Figure 2. For the completeness of model, thermal capacity of the swing plate has been considered, so that this model is also capable of modeling the transient behavior of the swing plate. Yet, only steady state temperature is of interest in this study. With such a model, it becomes straightforward to solve for the temperature of the swing plate in a certain ambient at a given heat level, as long as the material and heat transfer properties are known. This simulation model will run with various settings of K values proposed by different studies. The outcome of each simulation will be compared against the experimental results. Then, an assessment of all values of K will be made, so that we can determine whether or not the convection heat transfer model for continuously-spinning disk is still valid for the swing plate.

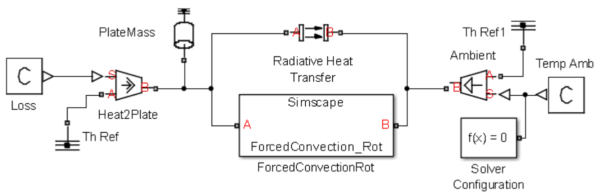

Figure 2. Thermal network model of the swing plate for simualtion

III. Experiments

A set of experiments have been conducted to create the working scenario of a swing plate, i.e. rotating back-and-forth in a bi-directional way. The results will be collected to compare with the model outputs.

A. *Experimental facility*

The setup of the experiments has been shown in Fig.Figure 3. It should be noted that, instead of using a real swing plate, a uniform isothermal disk of aluminum alloy is employed to represent the swing plate. Such simplification helps to focus on the forced convection study by eliminating the effects of geometry (e.g. heat sink-like structure). The disk is mounted at the end of a robot drivetrain through an adaptor made by thermal insulation material which will greatly eliminate the heat leaking from the disk. Regulated by a robot controller, the drivetrain drives the disk rotating back-and-forth in a bi-directional way as if a swing plate behaves in the robot.

An electric heater at the disk center is used to heat up the disk. The heat energy generated by the heat can be easily computed by its voltage and current. Given the fact that some amount of heat may be transferred out through the adaptor, a heat flux sensor is placed behind the heater to measure the heat leaking. Hence, the net heat transferred by the disk becomes measurable which is the total energy generated by the heater minus the heat leaking. The disk temperature is recorded by six thermocouples evenly arranged on the front and back surfaces of the disk, and the disk temperature is computed by averaging the outputs of the thermocouples. An extra thermocouple is used to collect the time history of ambient temperature during the experiment.

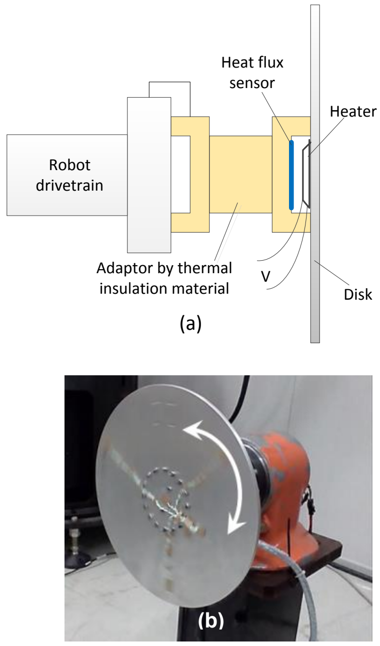

Figure 3. The setup of the experiments, (a) structure diagram and (b) its realization

B. *Design of Experiments*

In this study, the disk is subjected to experience four different motion cycles whose properties are presented in Table 2. It should be mentioned that the cycles in Table 2 are close to the motion limits of the industrial robot of interest, which indicates these cycles well represent the working scenarios of a swing plate in practice.

Table 2 The Motion Cycles of The Disk

	Cycle 1	Cycle 2	Cycle 3	Cycle 4
Motion[a]	-120° → 120°		160°→-120°→ -160°→120°	
Peak speed	Vmax[b]	150 rad/s	Vmax	150 rad/s
Disk ω_{rms}	3.670 rad/s	2.457 rad/s	3.795 rad/s	2.495 rad/s
Heat level	L/H[c]	L/H	L/H	L/H

a. The limit of the motor is ±165°

b. Vmax is approximately 520 rad/s for the motor

c. $L \approx 10W; H \approx 15W$

In addition, as shown in Table 2, the disk in each motion cycle will also experience two heating levels, namely 10W and 15W, which approximates the gross heat loss of the drivetrain of the robot of interest. This means 8 cases in total will be conducted in the experiment.

C. Results of experiments

In each round of experiments, the disk temperature will rise up since cycle begins, and eventually approach to its steady state. As mentioned before, we will focus only on the steady state temperature of the disk which is mostly of concern in practice. The experimental results of all cases are summarized in Table 3.

In each case, the net heat is regarded as the heat transferred out through the disk. It is calculated by subtracting the leaking measured by the heat flux sensor from the heat supplied by the electric heater. The steady-state readings of the 6 thermocouples on the disk are recorded, and the mean value of them is presented in Table 3 as the average steady-state temperature of the disk. It should be pointed out that, because the air around the disk flows in its laminar region in all cases, the temperature is almost uniform on the disk surface, which makes it meaningful to average the readings of the thermocouples.

TABLE 3 THE SUMMARY OF THE EXPERIMENTAL RESULTS

Case #	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Cycle #	Cycle 1		Cycle 2		Cycle 3		Cycle 4	
Net heat (W)	9.63	15.0	9.53	14.39	9.86	14.68	9.57	14.4
Amb.(°C)	22.1	23.5	24.4	24	22.9	23.9	22.6	24.4
S.S. Exp. Temp. (°C)	35.8	42.8	38.5	43.6	36.3	43.1	36.5	43.9

IV. RESULTS AND DISCUSSION

A. Simulation outcomes vs. experimental results

As discussed above, a simulation model (as shown in Figure 2) has been built up for the disk in the experiments. In the simulation, all parameters, except for K value, are configured correspondingly to the measured values in Case 1~8 in the experiments. K values presented in Table 1 and Eqs.(4)~(8) will be assessed one after the other by submitting them into the simulation. The simulation results with different K values are presented in Table 4. One may already noticed that some K values (i.e. $K \geq 0.43$) in Table 4 were not proposed by any previous studies. They are considered here in order to have a better coverage for comparison.

In Table 4, "Sim." represents the simulated results, and the simulation error, which is defined as the differences between the simulated results the experimental results, is denoted by "Δ" in Kelvin degree. In order to explicitly present the effect of K values on the simulation error, we also defined an error index as follows.

$$Err|_K = \sqrt{\sum \Delta_i^2}, \quad i = 1,2,\dots 8 \tag{15}$$

where $Err|_K$ represents the error index at a given K; $i = 1,2,\dots 8$ is the case number, and Δ_i denotes the simulation error in $Case\ i$ at the given value of K. Figure 4 illustrates the changes of the error index with respect to K.

TABLE 4 THE SIMULATION RESUTLS WITH VARIOUS K VALUES

Case #		**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**Ref.**
Net heat (W)		9.63	15.0	9.53	14.39	9.86	14.68	9.57	14.4	Table 3
Amb. (°C)		22.1	23.5	24.4	24.0	22.9	23.9	22.6	24.4	
S.S. Exp. Temp. (°C)		35.8	42.8	38.5	43.6	36.3	43.1	36.5	43.9	
K=0.306	Sim. (°C)	38.5	49.8	43.5	52.7	39.4	48.3	41.8	52.9	Eq. (5) [22][a]
	Δ (K)	2.7	7.0	5.0	9.0	3.1	5.2	5.3	9.0	
K=0.309	Sim. (°C)	38.4	49.6	43.4	52.4	39.3	48.1	41.7	52.7	Eq. (8) [25][a]
	Δ (K)	2.6	6.8	4.9	8.8	3.0	5.0	5.2	8.8	
K=0.32	Sim. (°C)	37.9	48.9	42.8	51.6	38.8	47.4	41.1	51.9	Eq. (6) [23][a]
	Δ (K)	2.1	6.0	4.3	8.0	2.5	4.3	4.6	8.0	
K=0.326	Sim. (°C)	37.6	48.5	42.6	51.2	38.6	47.0	40.8	51.4	[9]
	Δ (K)	1.9	5.6	4.0	7.6	2.2	3.9	4.3	7.6	
K=0.327	Sim. (°C)	37.6	48.4	42.5	51.1	38.5	47.0	40.8	51.4	Eq. (7) [24][a]
	Δ (K)	1.8	5.6	4.0	7.5	2.2	3.9	4.3	7.5	
K=0.330	Sim. (°C)	37.5	48.2	42.4	50.9	38.4	46.8	40.6	51.2	[15,16]
	Δ (K)	1.7	5.4	3.8	7.3	2.1	3.7	4.1	7.3	
K=0.333	Sim. (°C)	37.3	48.0	42.2	50.7	38.3	46.6	40.5	51.0	[10]
	Δ (K)	1.6	5.2	3.7	7.1	1.9	3.5	4.0	7.1	
K=0.341	Sim. (°C)	37.0	47.5	41.9	50.2	38.0	46.1	40.1	50.5	[17]
	Δ (K)	1.3	4.7	3.4	6.6	1.6	3.1	3.6	6.6	
K=0.367	Sim. (°C)	36.1	46.1	40.8	48.7	37.0	44.8	39.1	48.9	Eq. (4) [21][a]
	Δ (K)	0.3	3.2	2.3	5.0	0.7	1.7	2.6	5.0	
K=0.370	Sim. (°C)	36.0	45.9	40.7	48.5	36.9	44.6	39.0	48.7	[14,18]
	Δ (K)	0.2	3.1	2.2	4.9	0.6	1.5	2.5	4.9	
K=0.379	Sim. (°C)	35.7	45.5	40.4	48.0	36.6	44.2	38.6	48.3	[6]
	Δ (K)	-0.1	2.6	1.9	4.4	0.3	1.1	2.1	4.4	
K=0.384	Sim. (°C)	35.5	45.2	40.2	47.7	36.5	44.0	38.4	48.0	[13]
	Δ (K)	-0.2	2.4	1.7	4.1	0.1	0.9	1.9	4.1	
K=0.417	Sim. (°C)	34.6	43.7	39.1	46.1	35.5	42.5	37.3	46.4	[14,19]
	Δ (K)	-1.2	0.9	0.6	2.5	-0.8	-0.5	0.8	2.5	
K=0.43	Sim. (°C)	34.3	43.2	38.7	45.5	35.2	42.0	36.9	45.8	N/A
	Δ (K)	-1.5	0.3	0.2	1.9	-1.2	-1.0	0.4	1.9	
K=0.44	Sim. (°C)	34.0	42.8	38.5	45.1	34.9	41.7	36.6	45.4	
	Δ (K)	-1.7	-0.1	-0.1	1.5	-1.4	-1.4	0.1	1.5	
K=0.45	Sim. (°C)	33.8	42.4	38.2	44.7	34.7	41.3	36.4	45.0	
	Δ (K)	-2.0	-0.5	-0.3	1.1	-1.6	-1.8	-0.1	1.1	
K=0.46	Sim. (°C)	33.5	42.0	37.9	44.3	34.5	41.0	36.1	44.6	
	Δ (K)	-2.2	-0.8	-0.6	0.7	-1.9	-2.1	-0.4	0.7	

a. K values are computed based on Eqs. (4) ~ (8) with $Pr = 0.71$.

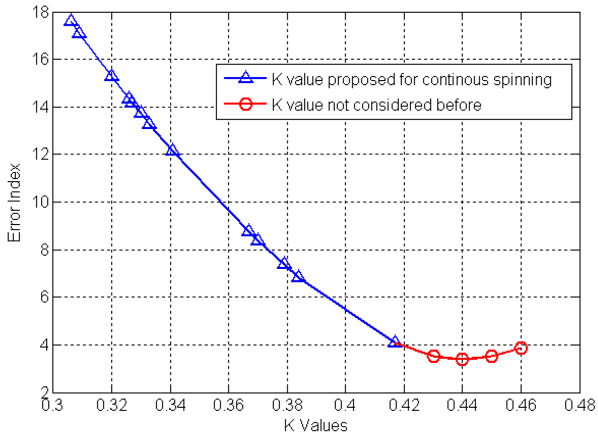

Figure 4. The effect of K on error index

B. Discussion

By closely looking at Table 4 and Figure 4, we may have the following observations.

1. Rotary reciprocation enhances heat transfer.

Consider the cases when K equals 0.326 which is the theoretical value of the continuously-spinning disk. The simulated temperatures in all eight cases are at least 1.3°C higher than what are measured in our experiments of rotary reciprocation. Similar trends can also be observed on other K values, especially in the low speed cases (i.e. Case 3, 4, 7, and 8). Consider these K values are actually measured and proposed based on the previous studies on continuous spinning, it suggests that rotary reciprocation may enhance the heat transfer by resulting in stronger forced convection and thus result in less temperature rises.

2. Forced convection in rotary reciprocation seems less sensitive to speed change.

In our experiments of rotary reciprocation, four pairs of cases, namely, Case 1 vs. 3, Case 2 vs. 4, Case 5 vs. 7, and Case 6 vs.8, are examined. In each pair, the cases are with equivalent net heat and motion trajectory, but different speed. Despite that the speed drops by about 1/3, it is found that the measured temperature rises in each pair of the cases are very close. For example, the measured temperature rises in our experiments of Case 6 and 8 are 19.2 °C and 19.5 °C respectively while the RMS speed drops from 3.795 rad/s in Case 6 to 2.495 rad/s in Case 8. In other words, the heat transfer enhancement caused by rotary reciprocation becomes even more obvious in low speed cases.

3. The forced convection model for rotary reciprocation

Figure 4 illustrates that error index first drops with increase of K values, and then rises up if further increasing the values of K. It reaches its minimum when K equals 0.440. This means the simulation results of the model with $K = 0.440$ seems best match with our experimental data, and thus, the following equation of Nusselt number should best describe the forced convection due to rotary reciprocation.

$$Nu_{av} = 0.440Re_{rms}^{0.5} \tag{16}$$

Compared to the models proposed for continuous spinning, this model will result in a larger Nusselt number, as well as a greater heat transfer coefficient. This suggests again that forced convection will be enhanced in the scenario of rotary reciprocation.

One possible interpretation of such enhancement may be as follows. In the scenario of rotary reciprocation, due to its viscosity, a layer of air should closely follow the motion of the disk moving back-and-forth. On the other hand, the inertia of the air would drive the air away from the disk when the angular speed of the disk changes. The effects of viscosity and inertia conflict with each other. When they play their roles simultaneously, there may be some part of the air (e.g. at disk boundary) becoming turbulent in some time of the cycle, which enhances the heat transfer.

However, in the scenario of continuous spinning, the disk spins at a constant angular speed. Therefore, the effect of inertia, together with viscosity, always makes the air follow the disk, and thus the air flow is laminar everywhere on the disk in all time. As a result, the heat transfer in continuous spinning becomes lower than that in rotary reciprocation.

V. Conclusion

The presented study aims to figure out an appropriate forced convection model for the swing plate on an industrial robot. Such a model should benefit robot design from a thermal perspective, which will potentially prevent the robot drivetrain from overheating. Driven by the robot drivetrain, the swing plate always rotates about its axis back-and-forth periodically in a way of rotary reciprocation. This unique type of motion makes the forced convection from the swing plate quite different from that in the case of a continuously-spinning disk.

In this paper, the existing empirical models developed for continuous spinning are reviewed. It seems that Eq.(2) of Nusselt number are widely agreed in heat transfer society except for its coefficient of K. These empirical models are then implemented into a thermal network simulation for assessment. Meanwhile, a set of experiments are conducted with a disk undergoing rotary reciprocation just as a swing plate on the robot. In our experiments, the resulting temperature in all eight cases are lower than what are predicted by the simulations with various setting of K. In other words, the existing models for continuous spinning seem inapplicable to rotary reciprocation directly. This is very likely because rotary reciprocation results in a stronger forced convection than continuous spinning does, and thus enhances the heat transfer from the disk in our experiments. From the experiment data, it also turns out that the model that best describes the forced convection in rotary reciprocation is the one with $K = 0.440$. Since this is concluded using a simplified disk in lab environment, it is desired to validate this forced convection model practically by a swing plate on a real robot considering all three modes of heat transfer. This will be part of the future works.

In addition, it should be pointed out that the experiments conducted in this study only covers the working conditions (motion, speed, heat level, etc.) that could be found in the application of industrial robots. From theoretic point of view, it should be interest to look into the forced convection in rotary reciprocation with much larger speed that is beyond robot speed limit. When speed is large enough, the laminar assumption will be invalid and hence models for turbulent flow should be more applicable to describe forced convection in such cases. We will also leave this study to the future works.

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