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Linking Power and Flow in Rotor-Stator mixers

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Abstract

The relationship between power and flow characteristics of batch rotor-stator mixers has been studied using CFD simulations with experimental power validation. The mixer studied was the Silverson L5M batch mixer with the standard emulsor head. The size of the holes in the screen and the constriction of the base hole were changed in small increments. The MRF technique was used to model rotor rotation. A model is developed in this study which links the power and flow numbers of the mixer. Since power is easy to measure experimentally, one can use this model to predict the flow number by measuring torque. A second model is also developed which allows one to predict the flow number using solely the geometry of the mixing head. This study greatly enhances our understanding of the relationship between power, flow and mixer geometry in rotor-stator mixers.

Keywords: Rotor-stator. Mixing. Power number. Flow number. CFD.

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Nomenclature

Symbols

$\dot{m}$	Mass flow rate	kg s^{-1}
B	Empirical constant	—
C	Empirical constant	—
Po_1	Power number constant	—
Po_Z	Power number constant	—
A	Area	m^2
a	Empirical exponent	—
D	Diameter	m
F	Fill ratio	U_Q/U_{tip}
H	Screen height	m
k	Empirical constant	—
k_1	Empirical constant	—
N	Rotor speed	s^{-1}
Q	Volumetric flow rate	$\text{m}^3 \text{s}^{-1}$
R	Radius	m
T	Stator base thickness	m
U_Q	Average fluid velocity through stator holes	m s^{-1}
U_{tip}	Rotor tip speed	m s^{-1}
v_n	Normal velocity	m s^{-1}

Greek symbols

Δ	Rotor-stator gap	m
Δ_{row}	Vertical distance between row centres	m
ϵ	Turbulent dissipation rate	$\text{m}^2 \text{s}^{-3}$

ϵ_{max}	Maximum turbulent dissipation rate	$\text{m}^2 \text{s}^{-3}$
α	Empirical constant	—
β	Empirical constant	—
ρ	Density	kg m^{-3}

Sub- and super-scripts

b	Stator base hole
h	Screen holes
i	Inner screen wall
o	Outer screen wall
r	Rotor
s	Whole stator

Dimensionless numbers

Eu	Euler number	$\Delta p / \rho v^2$
N_Q	Flow number	$Q / N D_r^3$
$N_{Q,0}$	Flow number at $A_b = 0$	$Q / N D_r^3$
Po	Power number	$P / \rho N^3 D_r^5$

Abbreviations

CFD	Computational Fluid Dynamics
MRF	Multiple Reference Frame
PIV	Particle Image Velocimetry

1. Introduction

Rotor-stator mixers, also known as high shear mixers, are mixing devices consisting of high speed rotors surrounded by stationary screens, or stators. They are widely used for processes which require intense mixing such as de-agglomeration and emulsification, since they generate very high shear rates in the region of the mixing head and also provide highly localised energy dissipation. For lab-scale operation, they are usually operated in a batch configuration, with the mixing head positioned on the end of a shaft which can be lowered into or raised from a mixing vessel. For larger scale processing, a continuous configuration is typically used, in which the mixing head is encased within a pipeline.

Despite their widespread use, there is a significant lack of theoretical understanding around rotor-stator mixers (Atiemo-obeng et al., 2004). In particular, there is a lack of understanding of how the design of the stator affects the agitation power and the hydrodynamics of the mixer. The agitation power is an important process variable, since it determines how much energy the mixer will use. It also provides valuable information related to the amount of energy available for droplet break-up during emulsification. Mechanistic models are available to predict the maximum droplet size during emulsification in the turbulent flow regime, which are based on the assumption that droplet break-up is caused by the interactions between the turbulent eddies and the dispersed droplets, and also that the droplets are larger than the smallest eddies at the Kolmogorov length scale (Calabrese et al., 1986; Davies, 1985, 1987; Hinze, 1955). One of the variables needed to predict the maximum droplet size is the maximum turbulent energy dissipa-

tion rate, ϵ_{max} . ϵ_{max} is particularly difficult to measure experimentally, and so instead the average energy dissipation around the impeller can be used. This means that

$$\epsilon_{max} \propto \text{Po} N^3 D_r^2 \quad (1)$$

where Po is the dimensionless power number for the impeller, N is the rotational speed of the impeller, and D_r is the diameter of the impeller (Carrillo De Hert, 2017). Po can be measured using experimental torque measurements, however, for standard impellers used in industry it is usually just found from literature. Obtaining Po for rotor-stators without using torque measurements is much more complex since there are many different designs of screen, each of which results in a different value of Po. It has been reported that droplet break-up in rotor-stators is caused by shear rather than turbulence (Rodgers and Cooke, 2012), in which case the above information regarding energy dissipation and droplet break-up models is not so relevant. In any case though, it is still highly desirable to understand how the screen design affects the agitation power. By doing so, it would therefore be possible to predict the power number based on the geometry of the mixing head, and so experimental torque measurements would not be required each time the screen design is changed. Optimisation of the screen design for emulsification processes would also be much easier with more knowledge in this area.

For in-line rotor stators with a given screen design, the power number has been found to be given by

$$\text{Po} = \text{Po}_Z + k_1 N_Q \quad (2)$$

where P_{OZ} is a constant which can be thought of as the power required to overcome the drag force as the rotor moves in the fluid, k_1 is a constant which depends on the mixing head geometry, and N_Q is the dimensionless flow number of the mixer (Kowalski, 2009). N_Q is calculated as follows

$$N_Q = \frac{Q}{ND_r^3} \quad (3)$$

where Q is the volumetric flow rate. Equation (2) has been proven valid by both experimental means and simulation for many different designs of in-line mixer (Cooke et al., 2011; Hall et al., 2011; Kowalski et al., 2011; Özcan-Taşkın et al., 2011; Jasińska et al., 2015). Equation (2) has been applied for in-line mixers due to the fact that N_Q can be measured and varied with ease, since the flow rate in an in-line mixer is essentially flow through a pipeline. No such model has been applied or tested by other authors for batch mixers since, for a given mixing head design, N_Q can not easily be changed or measured. It has already been noted in literature that to investigate the effect of mixing head design on k_1 would be highly beneficial in the development of our understanding of these mixers (Håkansson, 2018).

Padron (2001) studied power draw in various batch-rotor stator mixers, each with a different mixing head design, and concluded that the stator geometry, ie. number and size of holes in the screen did have an effect on the agitation power and therefore also the power number. However no correlation was found between stator geometry and power number. These conclusions are similar to many other studies which have also been conducted in this field of research (Utomo et al., 2008, 2009; Özcan-Taşkın et al., 2011; Mortensen et al., 2017). It is likely that no correlation has been found between the stator

70 geometry and the agitation power due to the fact that previous studies have,
 71 for the majority, been conducted using drastically different designs of mixing
 72 head for each experiment. Essentially, more than one geometrical variable
 73 is changing between designs. It would be much more beneficial to perform
 74 experiments in which only one geometrical parameter changes systematically.

Figure 1: Power number against (a) base hole radius and (b) flow number predicted by CFD for the Silverson L5M with standard emulsor head and various hole sizes, taken from John et al. (2019).

75 John et al. (2019) studied power and flow in the Silverson L5M operating
 76 in both batch and in-line modes using CFD with experimental torque mea-
 77 surements used as validation. It was found that by changing the constriction
 78 of the base holes, N_Q could be changed for a given screen design. The results
 79 also suggested that for the batch mixer, the power number could be pre-
 80 dicted using the model typically used for in-line mixers shown in Equation
 81 (2), something which had not previously been explicitly suggested in litera-
 82 ture. This is only the case when flow does not exit the base of the stator,
 83 something which will be explained in Section 3. It was suggested that the

84 rise in power number at large base radiuses, as seen in Figure 1(a) is due to
 85 flow begining to exit through the base. Since the study was conducted in a
 86 systematic manner, with stator geometries differing only in small increments,
 87 the effect of screen design on the constant k_1 could be seen much more clearly
 88 than from previous studies. The effect of screen design on k_1 was explained
 89 using the Euler number, Eu, which is the ratio between pressure and inertial
 90 forces. The equation developed by Kowalski (2009) shown in Equation (2)
 91 was derived by considering the acceleration of the fluid by the rotor in the
 92 mixing head. John et al. (2019) showed that by instead considering the pres-
 93 sure drop created by the rotor, k_1 in Equation (2) can be replaced with Eu. It
 94 is also recorded in literature that for fluid flowing through a perforated plate
 95 in a pipe, Eu is dependent on the design of the holes (Malavasi et al., 2012),
 96 which potentially explains why the screen design affects k_1 in a rotor-stator
 97 mixer.

98 The aims of this study were to investigate how the stator geometry af-
 99 fects agitation power in batch rotor-stator mixers. In this study, stator ge-
 100 ometries are changed in small systematic increments in order to enhance our
 101 understanding of which geometrical parameters affect the power and flow
 102 characteristics.

103 **2. Materials and Methods**

104 *2.1. Stator geometries*

105 The rotor-stator mixer used in this study was the Silverson L5M batch
 106 lab-scale mixer. There are various designs of stator for this mixer, including
 107 the disintegrating head, slotted head, square hole head, and the standard
 108 emulsor head. The standard emulsor head was used as the stator design in

109 this study. The standard emulsor head consists of staggered rows of small
110 closely spaces circular holes (see Figure 3).

111 As previously mentioned, N_Q is not easily changed in a batch rotor-stator
112 with a given stator design due to the fact that the speed and the diameter
113 of the rotor both affect the flow rate, ie. the flow rate can not be varied
114 independently using the rotor. To vary the flow rate of a particular screen
115 design, the hole in the base of the stator can be constricted. This has a
116 similar effect on the flow rate as when one uses a valve to change the flow
117 in an in-line mixer. The constriction of the base hole can be seen in Figures
118 3(d) and 3(e). As can be seen from Figure 3 and Table 1, the base hole was
119 varied from fully open, 3(a), to fully closed, 3(e). The effect of varying N_Q by
120 changing the base hole size on agitation power was investigated for screens
121 with various hole sizes, as can be seen in Figure 3. Figures 3(a), 3(b), and
122 3(c) show stators with the same number of holes but different sized holes.
123 Figures 3(f) and 3(g) show stators with the same total area of holes as Figure
124 3(a), but with different sized holes, ie. the number of holes was varied to
125 keep the area roughly constant. These experiments were designed to allow
126 us to determine whether it is simply the area of the holes which affects k_1
127 or whether it is something more complex such as the hydraulic radius of the
128 holes. The dimensions of the stator defined in this investigation are shown
129 in Figure 2. Table 1 lists range of values of the stator dimensions used in
130 this study.

Figure 2: General stator dimensions.

Table 1: Range of values of stator dimensions used in this investigation. All values given for standard scale simulations.

Symbol	Meaning	Values (mm)
D_b	Base hole diameter	$0 \leq D_b \leq 31.50$
D_h	Screen hole diameter	$1.00 \leq D_h \leq 2.00$
D_i	Inside screen diameter	31.50
D_o	Outside screen diameter	34.10
D_s	Whole stator diameter	51.00
H_s	Screen height	19.55
S	Screen thickness	1.25
T	Base thickness	3.15
Δ_{row}	Vertical distance between row centres	$2.14 \leq \Delta_{row} \leq 2.50$
* note that for simulations run at different scales, all values of all dimensions are just scaled by the factor chosen.		

131 The rotor used was a 4 bladed rotor in all cases. The rotor for the
132 standard scale simulations, where $T = 0.24$ m, was 31 mm in diameter, and
133 12.5 mm in height. The centre of the rotor was always positioned vertically
134 at the centre of the stator. This ensured, at least for most simulations, that
135 all the rows of holes remained fully covered by the rotor.

Figure 3: Examples of stator geometries.

136 2.2. CFD model

137 CFD simulations were carried out using ANSYS Fluent version 18, with
 138 each simulation being run on 12 cores. To model the rotation of the rotor
 139 one can either use the transient sliding mesh technique or the steady-state
 140 multiple reference frame technique. Sliding mesh has previously been the
 141 most popular choice for CFD simulations of rotor-stator mixers due to the
 142 fact that transient simulations are needed to account for the time dependent
 143 interaction between the rotor and the stator holes. This time dependency

144 is more pronounced with some stator designs than others. For example, the
145 Silverson disintegrating head causes a large time dependency due to the fact
146 that the holes are very large, relatively speaking, and spaced very sparsely.
147 As mentioned, the stator used in this study is the standard emulsor head,
148 which has small holes which are closely spaced and staggered between rows.
149 This means that the time dependent fluctuations in power and flow will be
150 very small and therefore the MRF technique can be used, which saves a large
151 amount of simulation time and computational expense. To prove that the
152 MRF technique is sufficient for this design, multiple rotor simulations were
153 tested. The interaction between the rotor and the baffles can be assumed to
154 be negligible since a rotor-stator naturally dissipates energy in a small region
155 around the mixing head, and the distance between the rotor and the baffles
156 is relatively large.

157 Turbulence was modelled using the standard $k - \epsilon$ model, which has been
158 widely used for the simulation of rotor-stator mixers (Håkansson et al., 2017;
159 Jasińska et al., 2015; Minnick et al., 2018; Mortensen et al., 2018; Özcan-
160 Taşkın et al., 2011; Utomo et al., 2008, 2009). All constants in the k and ϵ
161 transport equations were left as their default values, and the standard wall
162 function was employed to model near-wall viscosity-dependent effects. This
163 combination of model and wall function was found to give the best agreement
164 between the agitation power predicted from torque on the rotor and shaft
165 and the power predicted from the turbulent dissipation. To follow from this,
166 power was always calculated using the torque on the rotor and the shaft.
167 This method is more reliable and accurate than taking the volume integral
168 of ϵ , since all RANS turbulence models inherently under-predict turbulent

169 dissipation (Rielly and Gimbun, 2009). All solid boundaries were given the
 170 no-slip condition whilst the shear stress at the free surface was specified as 0
 171 in all directions. For spacial discretisation of the convective flow terms, the
 172 third order MUSCL scheme was used for momentum and turbulence, and the
 173 PRESTO! scheme was used for pressure. The COUPLED solver was used
 174 for the coupling of the pressure and the velocity.

175 The flow rate used in the calculation of the flow number, Equation (3),
 176 was taken at the inside wall of the stator. Flow through the holes was
 177 calculated by integrating the radial velocity across the inside wall of the
 178 screen, ie on a cylindrical surface with center at the center of the rotor and a
 179 radius of 15.75 mm. The flow through the base of the stator was calculated
 180 by integrating the axial velocity across a circular surface at the top of the
 181 base hole, ie. on a circular surface with center at the center of the stator
 182 and a height of 3.15 mm which is the thickness of the base of the stator. The
 183 flows into (+ve) and out of (-ve) the mixing head can then be calculated as
 184 follows

$$Q_{+ve} = \frac{\iint_S |v_n| \, dS + \iint_S v_n \, dS}{2} \quad (4)$$

$$Q_{-ve} = \frac{\iint_S |v_n| \, dS - \iint_S v_n \, dS}{2} \quad (5)$$

185 where v_n is the velocity normal to the surface across which flow is being
 186 calculated, ie. radial for the holes and axial for the base.

187 Convergence was said to be reached when the torque on the rotor and the
 188 flow through the stator holes did not change with regards to the iteration

189 number.

190 *2.3. Experimental procedure*

191 Experimental torque measurements were used to validate the power num-
192 ber predicted by the CFD simulations. The experimental procedure followed
193 to measure the power in this study is identical to that described in the study
194 by John et al. (2019). Experimental validation of the flow number predicted
195 by our CFD simulations of batch rotor-stators is very difficult to obtain, since
196 even with techniques such as 2D-PIV one must make various assumptions re-
197 lating to the way that the flow changes throughout each hole or slot in the
198 screen. This is relatively simple for stators with rectangular slots such as
199 the slotted head Mortensen et al. (2011), however it is much more difficult
200 for stators with circular holes such as the stator used in this study, espe-
201 cially since the flow is also found to vary in-between the rows of holes (this
202 will be discussed in Section 3). Realistically it is only possible to accurately
203 validate velocity profiles rather than the whole flow rates. For this reason,
204 experimental validation was only carried out for the power number.

205 **3. Results and discussion**

206 Results regarding the validation of the CFD model can be found from the
207 study of John et al. (2019). These results include the comparison between the
208 predicted power number from CFD and the experimental one, the effect of
209 grid refinement on the CFD results (ie, the establishment of grid independent
210 results), and also the effect of rotor position on power and flow numbers. The
211 results presented from here on in are presented under the assumption that
212 the CFD model used is now well validated.

Figure 4: Contours showing the radial velocity normalised by the rotor tip speed on 6 planes at the heights of the 6 rows of holes in the screen. Velocity vectors are overlapped to show the direction of flow. Stator is the original standard emulsor head from Silverson.

214 Figure 4 shows the radial velocity normalised by the tip speed of the rotor
 215 at planes of heights equal to the center of each row of holes. Each velocity
 216 contour is very similar to those obtained previously by both experimental
 217 means and CFD simulations, in that on the leading edge of each hole there
 218 is a high velocity jet which extends from the inside of the hole to the bulk
 219 of the tank, and behind this the fluid recirculates back into the hole towards

the rotor swept region (Utomo et al., 2008; Mortensen et al., 2011). The maximum normalised radial velocity reached is, quite understandably, close to unity and the highest radial velocities reached generally are located in the high velocity jets on the leading edge of the holes. It can also be seen from these results that the velocity contour is different for each row of holes in the screen, more specifically the flow appears to become more positive from rows 1 to 5 and then drop slightly at row 6. Figure 5 shows the mass flow rate of the positive radial component leaving the holes against row number (Figure 5(a)) and also the mass flow rate of the negative radial component recirculating back into the holes against row number (Figure 5(b)). These results show, quantitatively, the information displayed in Figure 4. The results suggest that there is not only non-uniformity in the flow rates across the various rows of holes, but also that this non-uniformity actually increases as the size of the holes in the screen increases. For the 2.00 mm holes, the flow through the lowest and highest rows appears to be quite extreme, that is there is a large amount of flow entering the lowest row and a small amount of flow exiting the top row relative to the other hole sizes. This could be due to the fact that for this particular hole size, the rotor doesn't entirely cover the top and bottom rows of holes. Similar results to this have been found by Espinoza et al. (2018), who used particle image velocimetry (PIV) to study the flow field of an in-line Silverson mixer. They also concluded that the flow was non-uniform across the rows of holes in the stator screen.

Figure 5: Mass flow rates (a) out of and (b) into each of the rows of holes, with 1 being the lowest with regards to the y-axis, for various sizes of holes in the screen of the stator.

242 3.2. Effect of stator geometry on power and flow characteristics.

Figure 6: Flow number components against base hole radius for screens with holes sizes of (a) 1.00 mm, (b) 1.59 mm, and (c) 2.00 mm.

243 Figure 6 shows each component of the flow number as a function of the
 244 base hole radius for screens with holes of various sizes. It is useful here to
 245 mention that the total flow number can be calculated as either the sum of the
 246 total inwards flow into the mixing head (N_Q holes -ve and N_Q base +ve) or
 247 as the sum of the total outwards flow out of the stator (N_Q holes +ve and N_Q
 248 base -ve). Theoretically both of these these should give the exact same total

N_Q , however since CFD is a numerical approach, convergence issues may give
 rise to slight discrepancies. For each screen hole size, the total flow number
 appears to follow a similar trend, that is an increase in flow number as the
 base hole begins to open followed by a plateau and then another increase as
 the base hole becomes fully open. This trend is similar to that of the power
 number as shown in Figure 1(a), which is expected since Figure 1(b) suggests
 that $Po \propto N_Q$. The trend of the total flow number can be explained using
 each of the flow number components. The total flow number increases as the
 base begins to open due to the fact that the flow enters through the base and
 exits through the holes in the screen. The amount of flow entering through
 the holes in the screen also contributes to this total flow number, hence why
 the total flow number is generally larger for screens with larger holes. This
 also explains why, for $D_h = 2.00$ mm, there is a much larger total flow number
 at $R_b = 0$. The increase of the total flow number after the plateau has been
 reached appears to be caused by flow exiting the base. Barailler et al. (2006)
 used CFD to simulate a batch rotor-stator mixer and also found that with a
 fully open base there was flow leaving the mixing head through the base of
 the stator. From Figure 6 it can be seen that this behaviour is more prolific
 for screens with smaller holes, likely since there is less area for the fluid to
 leave through the holes, and so the total flow rises more sharply after the
 plateau when the holes in the screen are smaller. This was also suggested by
 Barailler et al. (2006). The explanation as to why the plateau occurs will be
 given later on in this section.

As previously mentioned, it is likely that k_1 can be replaced with Eu which
 is affected by the size of the holes in the screen. It is unknown whether it

Figure 7: Power number as a function of flow number for stators with fixed A_h but varying R_h .

would likely be the area of the holes, A_h , or the hydraulic radius of the holes which affects k_1 . Figure 7 shows the power number as a function of the flow number for three stators whose hole size varies but total area of holes is constant. The results suggest that k_1 is not affected by the hydraulic radius of the holes but the total area of the holes. This conclusion can only be made for changes in hole size similar to that investigated in this study. Further changes in hole size were not investigated in this study as the increased fluctuations in power and flow would require transient simulations, or at least multiple steady state simulations with varying rotor positions. Mortensen et al. (2017) conducted a similar study with a slotted head stator and found that the hydraulic radius did affect the flow number of this mixer for a certain range of slot widths, but also didn't affect the flow number in another range of slot widths. It would make sense that the flow number is somewhat affected by the hydraulic radius of the stator holes, since the high velocity jets form on the leading edge of the stator holes. For stators with the same total hole area, a larger hydraulic radius means that the perimeter

290 of the holes is smaller and hence there is less room for the high velocity jets
 291 to form. It might also be the case that the area of the holes has an individual
 292 effect on the flow number, since it has been shown in the study of Malavasi
 293 et al. (2012) that for a perforated plate in a pipe, there are multiple terms
 294 used in the calculation of the Euler number, some of which are purely area
 295 effects. It is therefore evident that to truly understand how the area and
 296 the hydraulic radius of the holes affects flow characterisation in rotor-stator
 297 mixers is highly complex. Using all of the previous information, we propose
 298 here that for the stators used in this study

$$\text{Eu} = k \left(\frac{A_h}{D_r^2} \right)^a \quad (6)$$

299 where k is a pre-exponential empirical constant and a is an empirical expo-
 300 nent. It was also found that Po_Z followed a similar trend so that

$$\text{Po}_Z = \text{Po}_1 \left(\frac{A_h}{D_r^2} \right)^a \quad (7)$$

301 where Po_1 is another pre-exponential constant. Figure 8 shows all of the
 302 data obtained in this study collapsed using Equations (6) and (7). It can be
 303 seen that all of the data fits onto the correlation almost perfectly with an r^2
 304 value of 0.996. The power number for a batch rotor-stator with a standard
 305 emulsor head can therefore be predicted using the following

$$\frac{\text{Po}}{\left(\frac{A_h}{D_r^2} \right)^a} = \text{Po}_1 + k\text{N}_Q \quad (8)$$

306 for various hole sizes.

307 Of course, it would be ideal to be able to predict the power number not
308 just for stators of a particular design, but many designs. Figure 9 shows
309 data obtained from CFD simulations of a Silverson L4M with various stator
310 designs, namely the slotted head, disintegrating head, and square hole head
311 from Utomo et al. (2008), and also data from PIV experiments on a custom
312 built mixer with a slotted head stator from Mortensen et al. (2017). The
313 correlation proposed in Equation (8) is compared with the data for other
314 studies in Figure 9. It can be seen that the correlation appears to work
315 for not just the standard emulsor head, but many designs of stator. This
316 may seem contradictory to some of the previous conclusions in this study.
317 It has been mentioned that only stators with no negative axial flow in the
318 base were considered for the data analysis. Stators with fully open bases
319 sometimes exhibit axially negative flow in the base, and the magnitude of
320 this depends on the size of the holes in the screen. It is therefore likely that
321 the areas of the holes in the stators used in the studies by Utomo et al.
322 (2008) and Mortensen et al. (2017) were large enough so that there was little
323 or no negative axial flow through their bases. The results from the study
324 of Mortensen et al. (2017) also provide some validation to our CFD results
325 since they are experimental PIV results.

326 The results from Figures 8 and 9 are particularly useful, since they also
327 suggest that it is possible to predict the flow number for a batch rotor-
328 stator using the power number, which can be measured easily using torque
329 measurements. It is far easier and simpler to measure the power in a batch-
330 rotor stator than it is to measure the flow rate, especially since it has already
331 been shown in this study and the study of John et al. (2019) that the flow

332 which needs to be measured is the actual flow through the holes in the screen,
333 not just the flow in the pipe in the case of the in-line mixer.

334 The results from John et al. (2019) suggested that for a given screen de-
335 sign, a batch and in-line mixer share the same characterisation for power and
336 flow, ie. k_1 and Po_Z are the same. This would also mean that Equation (8)
337 should work for multiple stator geometries regardless of mode of operation. A
338 correlation was developed in John et al. (2019) which allowed the prediction
339 of the true flow rate through the holes in the screen based on the geometry
340 of the stator. Therefore, for an in-line mixer, Po can be predicted using only
341 the geometry of the stator.

342 It should be noted here that all of the mixers investigated in this study
343 are single rotor/single stator mixers. It is likely the same conclusions can not
344 be made for mixers with more than one rotor or stator.

Figure 8: Plot showing how Equation (8) can be used to predict power number for a given flow number for multiple stator geometries.

Figure 9: Plot showing how data from previous experimental and simulation based studies using dramatically different stator designs compare with the results presented in this study.

3.3. Prediction of flow number

Although Equation 8 now allows us to predict the power number using the mixer geometry and the flow number, there is still no way of easily knowing the flow number in a batch rotor-stator. As already mentioned, the power can be measured using a torque meter and Equation (8) can be used to predict the flow number, but it would be more useful to be able to predict the flow number without having to run any experiments. Using the data from this study, we can propose a new correlation which allows us to also predict the flow number based on the mixer geometry. It must be noted here that this correlation only will work for stators who fall on the linear correlation shown in Equation (2), or more specifically, only for mixers who do not exhibit any axially negative flow through the base of the stator. These stators with negative flow through the base were ignored here as it is not yet understood whether this type of flow has an effect on the effectiveness of the rotor-stator in terms of emulsification. It could potentially be found that for optimal performance, stators should be constricted to prevent negative flow through the base.

Figure 10: Flow number as a function of the area of the base hole for various screen hole sizes.

Figure 10 shows the flow number as a function of the area of the base hole for three different screen hole sizes. It can be seen that at low base hole areas, the flow number is proportional to the area of the base. This would be expected since the base hole is restricting the flow. At higher base hole sizes, the flow number begins to level off to a plateau since the holes in the screen are now restricting the flow and for a given hole size, the area of the holes is constant. If the flow through the holes was plug flow, it would be possible to predict the height of the plateau using the tip speed of the rotor and the area of the holes. However, as already mentioned, the high speed jets in the holes do not occupy the total area of the holes. It is therefore necessary to study how much of total area of the holes is actually taken up by the jets and how this changes with hole size. To achieve this, the fill ratio of the rotor stator will be used. The fill ratio is calculated using

$$F = \frac{U_Q}{U_{tip}} = \frac{Q/A_h}{\pi N D_r} \quad (9)$$

375 where F is the fill ratio, U_Q is the average radial velocity through the
 376 stator holes, U_{tip} is the tip speed of the rotor, and Q is the volumetric flow
 377 rate of fluid in the high velocity jets. The fill ratio has been used in previous
 378 rotor-stator studies to investigate hydrodynamic differences between batch
 379 and in-line configurations (Mortensen et al., 2017). Figure 11 shows the fill
 380 ratio as a function of the area of the holes for various stators investigated
 381 in this study. The stators here are stators who exist on the plateaus shown
 382 in Figure 10, ie the holes in the screen are restricting the flow, not the base
 383 hole. It can clearly be seen that fill ratio is proportional to $A_h^{-1/2}$. This tells
 384 us that as we increase the size of the holes, less of the area of the holes is
 385 actually used for the radial flow generated by the mixer. It is likely that the
 386 hydraulic radius of the holes, which takes into account the wetted perimeter
 387 of the holes, will have an impact on this relationship. The effect of hydraulic
 388 radius on mixer hydrodynamics was investigated in this study (see Figure 7),
 389 however the hole size could only be varied by relatively small amounts due
 390 to the fact that MRF was used in this study. Of course, one could create a
 391 stator with a very small number of very large holes, which would have the
 392 same area as the stators used in this study, but then periodicity in power and
 393 flow would require transient simulations. One would expect that for stators
 394 with a very large number of very small holes, the fill ratio would be nearer to
 395 unity than for stators with a small number of large holes. Ergo, the hydraulic
 396 radius of the holes in the screen should be considered when extreme changes
 397 in stator geometry are compared.

Figure 11: Fill ratio as a function of A_h/D_r^2 for stators with hole sizes of 1.00 mm, 1.59 mm, and 2.00 mm. Dotted line is a power law fit with exponent of $-1/2$. $r^2 = 0.95$.

398 Using Equation, (9), it is now possible to know the level of the plateau
 399 relative to the flow number at a base hole area of 0. At a base hole area of 0,
 400 it can be seen that stators with larger holes have a higher flow number since
 401 all of the flow is entering and leaving through the holes. This flow number,
 402 $N_{Q,0}$ can therefore be calculated by

$$N_{Q,0} = B \frac{A_h}{D_r^2} \quad (10)$$

403 where B is an empirical constant. Using Equations (9) and (10) it is now
 404 possible to know the height of the plateau for each stator. Again, if flow
 405 through the holes in the screen were plug flow, the base hole area at which
 406 the plateau occurs would be equal to the area of the holes in the screen, and
 407 so to predict the flow number for all geometries the ratio of the base hole
 408 area and the total screen holes area could be used. Using the fill ratio again,
 409 we can account for the deviation caused by the low fill ratio. Figure 12 shows

the curves obtained and displayed in Figure 10 collapsed using the fill ratio
and Equation (10).

Figure 12: Curves from Figure 10 collapsed using Equations (9) and (10). Dotted line shows line of best fit whose format is displayed in the figure.

The flow number of a batch rotor stator can therefore be calculated using
the following

$$N_Q = \frac{B \left(\frac{A_h}{A_b F} \right)}{1 + C \left(\frac{A_h}{A_b F} \right)} \times \frac{1}{F} - \beta \left(\frac{A_h}{D_r^2} \right) \quad (11)$$

where B, C, and β are empirical constants. Equation (11) can be rearranged
using the results displayed in Figure 11 to give the following

$$N_Q = \frac{B \left(\frac{A_b}{\alpha \sqrt{A_h} D_r} \right)}{\frac{\alpha D_r}{\sqrt{A_h}} + \frac{C A_b}{A_h}} + \beta \left(\frac{A_h}{D_r^2} \right) \quad (12)$$

where α is a fitting constant. The values of the fitting constants can be seen in Table 2.

Table 2: Values of fitting constants used in Equation (12)

Constant	Value
B	0.00011
C	0.03
α	0.008
β	0.1

Figure 13 shows the flow number calculated from the CFD data in this study against that predicted by Equation (12). It can be seen that Equation (12) is able to accurately predict the flow number in a batch rotor-stator with a standard emulsor head for various geometries.

Figure 13: Parity plot showing CFD flow number against predicted one. Dashed line represents $Y=X$. Grey area represents $\pm 10\%$ relative error. r^2 is 0.99. Mean absolute error is 0.008.

3.4. Effect of rotor-stator gap on power and flow characteristics.

It is currently unknown how the magnitude of the gap between rotor and the stator affects the power and flow characteristics of rotor-stator mixers, however, this is a variable which commonly changes between different designs of rotor-stator available for companies to purchase.

Figure 14: Power number as a function of flow number for various rotor-stator gaps. Dotted line represents a linear fit of the concatenated data. Screen hole size is 1.59 mm in all cases.

Figure 15: Flow number as a function of rotor-stator gap normalised by inner stator screen diameter for various base sizes. Dotted lines represent the average value of first three data points in each series.

427 Figure 14 shows the power number as a function of the flow number for
 428 rotor-stators with various base diameters and rotor-stator gaps. It can be

429 seen that the distance between the rotor and the stator does not appear to
 430 change the relationship between the power number and the flow number (ie,
 431 it does not affect k_1 and Po_Z). It does, however, appear to affect the flow
 432 number and this effect seems to be more prominent for stators with larger
 433 base hole areas. This can also be seen in Figure 15 which shows the flow
 434 number as a function of the rotor-stator gap normalised by the inner stator
 435 screen diameter for various base hole sizes. At large base hole areas, the flow
 436 number appears to increase slightly with the magnitude of the rotor-stator
 437 gap. The results from Padron (2001) show a decreasing power number as
 438 the width of the shear gap is increased, this contradicts the results presented
 439 here. However, the screen design used in their study was the slotted head
 440 design. This could indicate that the effect of varying shear gap width actually
 441 changes with the design of the screen. Simulation results from Ghaderzadeh
 442 (2018) agree with the results shown in Figures 14 and 15. They found that
 443 the power number of an in-line Silverston L4M with a standard emulsor screen
 444 increases as the shear gap width is increased. This might be expected, since
 445 it is the low pressure behind the rotor blade which is responsible for the
 446 recirculation of flow into the holes. At smaller rotor-stator gaps, the low
 447 pressure behind the rotor blade is closer to the holes and so fluid will be
 448 pulled into the holes at a faster rate. At larger gaps, the low pressure behind
 449 the blade is further from the hole and so there is more area available in the
 450 holes for the high velocity radial jet, since less fluid is being drawn in, ergo
 451 the flow number increases. At small base hole areas, the large majority of
 452 the flow enters and leaves through the holes in the screen. Therefore, the gap
 453 between the rotor and the stator does not have an effect on the flow number,

454 since the amount of fluid entering and leaving through the holes always have
455 to be roughly equal, and since the flow number is normalised by the rotor-
456 diameter, the flow number is constant. It can also be seen in Figure 15 that
457 the model shown in Equation (12) is slightly less accurate when predicting
458 flow numbers in stators with small base holes.

459 The rotor-stator gaps investigated in this study range from the more
460 commonly used gap sizes in industry to gap sizes which are much larger than
461 that typically used in industry. The above conclusions are therefore mainly
462 hypothetical and in reality the gap between the rotor and the stator will
463 not affect power and flow characteristics in most industrial mixers, since the
464 gap is nearly always very small. It can therefore be concluded that when
465 predicting the flow number, the rotor-stator gap does not need to be taken
466 into account.

467 4. Conclusions

468 This study presents findings from an investigation into how stator geome-
469 try affects power and flow characteristics in batch rotor-stator mixers. It was
470 found that in a batch rotor-stator with a standard emulsor head, the flow rate
471 varies through each of the rows of holes in the screen. The non-uniformity in
472 flow between rows increases as the holes size is increased. This is something
473 which has also been observed experimentally for an in-line rotor-stator in a
474 previous study. The empirical constant, k_1 , used in the typical power char-
475 acterisation of in-line mixers is shown to vary with the ratio of the area of
476 the holes in the screen and the rotor diameter squared. Using this new rela-
477 tionship, we have created a correlation which allows for the prediction of the
478 power number provided that one knows the flow number. This correlation

479 is proven to be accurate at predicting power number for various geometries
480 and designs. Data from previous studies, both experimental and simulation
481 based, matched exceptionally well with the model proposed in this study.
482 This correlation is extremely useful to us, since it is far easier to measure the
483 power draw of a batch rotor-stator mixer than it is the flow number. One
484 can measure the power number using experimental torque measurements, and
485 predict the flow number using the new correlation developed in this study.
486 However, it would be even more useful to be able to predict the flow num-
487 ber of the mixer without any experimentation. We therefore have developed
488 another new correlation which enables the prediction of the flow number in
489 batch-rotor stator mixers with the standard emulsor head. This correlation
490 uses a dimensionless number known as the fill ratio to relate the amount of
491 radial flow generated by the mixer to the mixer geometry. This correlation is
492 proven to be accurate at predicting flow numbers for a wide range of stator
493 geometries. It was concluded that the gap between the rotor and the stator
494 had a negligible effect on the flow number and so this variable does not need
495 to be taken into account when trying to predict the flow number. Overall,
496 this study greatly enhances our understanding of the relationship between
497 agitation power, flow, and mixing head geometry in rotor-stator mixers.

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