Influence of liquid in clearances on the operational behaviour of twin screw expanders

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Abstract. A lot of effort has been expended on understanding the influences of an injected auxiliary liquid on a twin screw expander’s performance. Sealed clearances improve performance on the one hand, but involve considerable frictional losses on the other hand. This paper contributes to an evaluation of these opposing effects with regard to the efficiency of screw expanders. First, thermodynamic analyses using the multi-chamber model-based simulation tool KaSim, developed at the Chair of Fluidics, are presented for a test screw expander in order to show the maximum potential of clearance sealing. This analysis involves thermodynamic simulations for sealed and unsealed clearances and leads to an order of priorities for different clearance types. Second, hydraulic losses within front and housing clearances are calculated, applying an analytical model of incompressible one-phase clearance flow. Subsequently dry and wet screw expanders are evaluated while both clearance sealing and frictional losses are considered for the simulation of a liquid-injected machine.

1. Introduction
Rotary positive displacement machines are designed to be either dry or wet running. While dry running twin screw machines can be calculated well by chamber model based simulations, no general model for liquid injected machines, including all significant auxiliary liquid effects, exists. Consequently simulating these machine types to produce a satisfactory match with experimental results is not possible at present. Initial investigations in understanding the operational performance of wet running screw machines are presented by Kauder [1], who deduced the "Oil-Surge-Theory" by experiments with an oil injected screw compressor. According to this approach, the injected oil leads, besides other effects, to sealed clearances but also to higher frictional losses. The ratio of these two effects mainly depends on the volume of oil. Kauder and Deipenwisch [2] point out that oil has a considerable influence within the working chambers of screw compressors: cooling the working fluid, sealing clearances, rotor lubrication and noise reduction. However, there are also dissipative processes such as friction and liquid acceleration. An analytical model of dissipation is developed, including oil acceleration during injection and within clearances, as well as frictional losses due to clearance flows. The results show clearance priority, and indicate that more than 90% of power losses within the screw compressor investigated result from front and housing clearances. An initial model for oil distribution covering droplet trajectory within working chambers, film flow on rotor lobes and clearance flows is introduced by Harling [3]. Harling succeeded in verifying his model partially by visual experiments; however, also unconsidered issues such as oil surge or gas bubbles within the liquid...
flow are apparent. Oil distribution within working chambers is extremely inhomogeneous, as confirmed by Rinder and Moser [4] who took samples of an oil-gas-mixture to determine the amount of oil. They relate irregular spreading to insufficient distribution of the injected oil jet. Previously described research work deals with screw compressors exclusively and only a few research papers focus on screw expanders. Zellermann [5] provides a first method for the optimal design of liquid-injected machines. His theoretical approach for calculating oil distribution within working chambers is similar to Harling’s [3]. Using analytical relations, droplet trajectory, film flow and clearance flow are calculated based on oil injection. Experimental approaches also show inhomogeneous oil distribution. Moreover, the considerable influence of liquid injection is demonstrated by varying injection location, auxiliary liquid (oil and water), auxiliary liquid volume flow and operating parameters, such as rotational speed and pressure ratio. Further investigations presented by Nikolov [6], demonstrate measured deviations of effective isentropic efficiency in dry and wet running screw expanders. In these machines, isentropic change of state in unsaturated humid air is shown as a function of circumferential speed. At a low circumferential speed, the efficiency of a wet running machine rises due to the sealing of clearances; however, with rising circumferential speed frictional losses within clearances become predominant and therefore efficiency decreases. In addition, effective isentropic efficiency is analysed, based on an isentropic change of state of a humid air-water mixture. The advantages of water injection at low circumferential speeds are not very significant. For a small water volume flow, water injected machines are even less efficient than dry running expanders. Differences in isentropic efficiency compared with unsaturated humid air result from continual interaction between water and air during expansion. The research already illustrates the main effects of liquid within screw machines. In order to provide further understanding of liquid-injected machines, this paper focuses on comparing the potential of clearance-sealing with frictional losses in an oil-injected screw expander.

2. Test screw expander

In this paper a liquid injected, unsynchronised test screw expander is simulated. The expander has an inner volume ratio of 3.5 and is operated at rotational speeds up to 16000 min$^{-1}$. The tests were performed at an external industry test rig. Due to privacy restrictions no exact absolute values are specified.

|                           | male | female |
|---------------------------|------|--------|
| number of lobes           | 5    | 7      |
| wrap angle                | 300° | 214°   |
| length / diameter ratio   | 1.6  | 1.74   |
| inner volume ratio        | 3.5  |        |
| distance between shafts   | ≈ 60 mm |       |
| normalized housing clearance height | 3    | 4.67   |
| normalized profile clearance height (mean height) | 1.67 |
| normalized front clearance height (high pressure side) | 1    |
| normalized front clearance height (low pressure side) | 5    |
| working fluid             | ethanol |
| injected auxiliary liquid | oil    |
Geometrical data is given in Table 1. The distance between the shafts is approximately 60 mm. All clearance heights are declared in relation to minimum clearance height, which arises at high pressure front clearance. In this paper clearance height is defined by two solid boundaries (rotors or housing) whereas clearance length is given by direction of flow. Clearance width indicates the third dimension which is normal to flow direction. Figure 1 illustrates the different clearance types. The expander is investigated with organic fluid ethanol which is superheated during the whole process for all operating points. For simulation the injected auxiliary liquid (oil) is assumed to have a constant temperature of 95°C. The fluid parameters used to calculate clearance losses are dynamic viscosity $\mu = 0.0265$ Pa·s and density $\rho = 791$ kg·m$^{-3}$. All simulations are carried out for the same operation point. The inlet pressure of the expander is set to $15 \times 10^5$ Pa and the working fluid is superheated by 30°C.

3. Potential of clearance sealing regarding the expander’s efficiency

In this section the influence of different clearance types on the operational behaviour of the test screw expander is investigated. In order to determine internal isentropic efficiency

$$\eta_{is} = \frac{w_i}{w_s} = \frac{P_i}{w_s m} = \frac{P_i}{(h_i - h_{O,s})m}$$

mass flow $\dot{m}$ and internal power $P_i$ must be calculated taking all internal loss mechanisms such as inlet throttling, non-isentropic change of state and clearance mass flow of the working fluid into account. These parameters are computed with the multi-chamber simulation program KaSim using the main characteristics of positive displacement machines, which are cyclically changing working chambers [7]. Homogeneous fluid states within each chamber are determined for discrete points in time. The extensive state variables, i.e. mass, internal energy and the specific volume of the working fluid are calculated based on the chamber volume curve and conservation of mass and energy. Subsequently, intensive parameters, such as pressure and temperature, are determined according to the defined thermodynamic state. In order to compute these state properties with maximum accuracy, KaSim accesses the NIST Reference Fluid Thermodynamic and Transport Properties Database ( REFPROP), providing equations of state for various fluids [8]. Equations used for ethanol are given in [9]. The injection of an auxiliary liquid is not simulated explicitly;
however, the effect of clearance sealing can be included in this modelling. The assumed changes of state for the reference cycle under consideration are isobar filling, isentropic expansion and isobar discharge. Thus, the specific efficiency of this ideal process \( \eta_s \) is determined directly by the difference between inlet (I) and isentropic outlet (O,s) enthalpy.

Simulated differences in internal, isentropic efficiency \( \Delta \eta_{i,s} \) between a dry running expander with unsealed clearances and completely sealed clearances (overall) are shown in Figure 2. Furthermore, different clearance types are closed separately in order to show their individual effect on operational behaviour (housing clearance, blow holes, profile clearance, front clearance (high and low pressure side)). The values presented demonstrate an increase in internal isentropic efficiency due to clearance sealing. They are therefore interpreted as representing the maximum potential. Apparently this potential is largest for small rotational speeds. If all clearances are closed, internal isentropic efficiency increases by 40 % at \( U_{mr} = 10 \text{ m·s}^{-1} \) which is remarkable. Decreasing potential at higher rotational speed can be traced back to the declining influence of clearances. At high rotational speed working cycles are considerably shorter, leading to less mass transportation through clearances. Due to this effect the internal isentropic efficiency of machines with unsealed clearances converges with that for machines with sealed clearances and therefore potentially decreases; however, for the investigated screw expander, even at a high circumferential speed of 90 m·s\(^{-1}\) a significant potential of 10 % exists.

As far as clearance priority is concerned, the influences of housing and profile clearances are significant, whereas front clearances (high and low pressure side) and blow holes have a negligible effect on internal isentropic efficiency. The almost non-existent impact of high pressure front clearances is explained by the relatively small clearance height and the resulting small cross-section area in the direction of flow. At the male rotor, the high pressure front clearance cross-section area is only 4.5% of the mean cross-section area of the housing clearance. This approach applies also for the small blow hole cross-section areas of this expander. At the low pressure side, front clearances generally show small mass flows due to almost vanishing pressure differences between clearance inlet and outlet. Thus neither blow holes nor high and low pressure side front clearance sealing lead to significant changes in the operational behaviour of the investigated expander, as even for unsealed clearances mass flows are negligible. In contrast, high pressure differences combined with large cross-section areas lead to significant influences of housing and profile clearances.

![Figure 2. potential of clearance sealing](image-url)
Note that the delta internal isentropic efficiencies $\Delta \eta_{i,s}$ of each clearance type cannot be added together for an overall result, as there are interdependencies. Sealing a single clearance type affects chamber pressures, which leads to different mass flows in all other clearances. Therefore, linear combination is not possible and each clearance configuration has to be simulated separately. Furthermore, the potential for clearance sealing might change in the screw expander under investigation, as shown, for example, by Dreißig [10], who points out significant influences of high pressure front clearances and blow holes. Nevertheless, two main tendencies which are a major influence on profile and housing clearance and a minor influence on the other two types concerning the efficiency of screw expanders correspond to Figure 2. Generally speaking, the major potential of clearance sealing is demonstrated; however, liquid injection in order to take advantage of this potential always involves frictional losses which will be analysed for high pressure front and housing clearances in the next section.

4. Frictional losses within front and housing clearances of liquid injected screw expanders

In this section frictional losses within front and housing clearances are calculated based on the analytical model presented in [11]. Assuming completely sealed clearances, a steady one-phase flow of auxiliary liquid is calculated based on the superposition of Couette and Poiseuille flow. In both cases either laminar or turbulent flow type is considered, depending on boundary conditions. Front clearance geometry is modelled by finite elements which are connected in parallel and do not interact with each other. Calculation is done separately for each element and the solution is deduced explicitly. In contrast, finite elements of housing clearance are arranged in series leading to additional boundary conditions between these elements. Thus, the resulting equations must be solved iteratively. Moreover, flow direction has to be taken into account due to the clearance geometry corresponding to nozzle and diffuser respectively, Figure 3. During simulation, calculations are performed for both flow directions; however, if the flow is defined, only one valid solution exists. If more or even no solutions exist, the flow is undefined and cannot be obtained by this model. This usually occurs with an almost vanishing mass flow. Therefore, mass flow is set to zero for all undefined states of flow, whereas hydraulic power is interpolated by limiting valid solutions. Furthermore it can be shown that the large opening angle of the diffuser geometry leads to flow separation. Therefore, the clearance outlet will always correspond with the minimum clearance height. For more details see [11]. Chamber pressure is determined for each time step by $\text{KaSim}$. Thus, pressure differences attached to all clearances are given, assuming chamber pressure at clearance in- and outlet. Using these boundary conditions, quasi-steady flow is calculated for each time step. Before analysing integral hydraulic losses depending on the expander’s circumferential speed, hydraulic power and clearance mass flow during the working cycle are analysed as an example for male rotor housing clearance.

![simulated clearance geometry](image)

**Figure 3.** simulated geometry at male rotor housing clearance according to [11]
Figure 4 presents parameters of male rotor housing clearance as functions of normalized phase angle $\phi$, defined as

$$\phi = \frac{\alpha_{mr} z_{mr}}{360^\circ}$$

(2)

with rotational angle $\alpha$ and lobe number $z$. Thus, the thermodynamic state of the expander is equal at the start and end of each phase. All parameters are normalized to their maximum absolute values which are

$$\Delta \bar{p} = \frac{\Delta p}{\Delta p_{\text{max}}}, \bar{w}_c = \frac{w_c}{w_{c,\text{max}}}, \bar{m}_c = \frac{\dot{m}_c}{\dot{m}_{c,\text{max}}} \text{ and } \bar{P}_c = \frac{P_c}{P_{c,\text{max}}}.$$  

(3)

The upper diagram of Figure 4 shows normalized pressure difference $\Delta \bar{p}(\phi)$ at housing clearance, which is calculated from the pressure difference between two consecutive chambers. Furthermore, normalized housing clearance width $\bar{w}_c(\phi)$ is given, resulting from the screw expander’s geometry. Both parameters are used to compute clearance mass flow $\bar{m}_c$ and

![Diagram showing normalized parameters](image)

Figure 4. normalized housing clearance parameters during working cycle of screw expander
hydraulic power $\bar{P}_t$, as indicated in the lower diagram. As presented in [11], negative pressure differences correspond to lower pressures in the subsequent chamber (chamber 2 in Figure 3). As there are no positive pressure differences shown in Figure 4, overexpansion, that is pressure at the end of expansion below low pressure, does not occur at the investigated operating point. During charge, the working chamber under consideration (chamber 1 in Figure 3) increases, while the subsequent chamber is already expanding. Thus pressure difference at the housing clearance increases first ($0.4 < \phi < 1.7$) and decreases with the expansion of the working chamber ($1.7 < \phi < 4.0$). Underexpansion of the working fluid leads to a sudden pressure drop within the subsequent chamber at $\phi = 4.0$. Thus the pressure difference between the working chamber and the subsequent chamber shows an abrupt rise. Finally, pressure in both chambers corresponds to the outlet pressure ($\phi > 5.3$) resulting in vanishing pressure difference at the housing clearance during discharge. Clearance width shows a linear increase up to maximum value, and then decreases.

In the following, resulting clearance mass flow and hydraulic power are analysed. As shown in Figure 3 positive mass flow corresponds to mass transported from considered (1) to subsequent (2) chamber whereas a negative sign indicates the opposite direction. First ($0.4 < \phi < 0.8$) no mass flow arises due to an undefined state of flow that is explained above. For this set up (geometry, fluid and rotational speed) normalized pressure differences between $-0.05$ and $-0.14$ lead to clearance flow which cannot be calculated explicitly with the described analytical model. Subsequently, mass flow increases due to rising absolute values of pressure difference and clearance width ($0.8 < \phi < 1.7$). With decreasing pressure difference, mass flow drops although clearance width is still increasing ($1.7 < \phi < 4.0$). Increases and decreases at ($4.0 < \phi < 5.1$) result from changing pressure difference. Again, clearance width also has an impact on gradients. At $5.1 < \phi < 5.2$ the flow is undefined, again. For vanishing pressure difference negative mass flow results ($5.2 < \phi$). At a constant pressure difference the transported mass decreases linearly due to decrease in clearance width.

Hydraulic power can be either negative or positive. Frictional loss is indicated by a positive sign, while negative power corresponds to driving power. If driving power is transferred to the rotors, Poiseuille flow dominates over Couette flow, which always causes frictional loss. These simulations are computed at a moderate rotational speed, leading to low Couette flow. Therefore normalized pressure differences below $-0.007$ result in additional power. Increase and decrease in hydraulic power ($0.4 < \phi < 5.1$) can be traced back to pressure difference and clearance width, analogous to mass flow. For undefined states of flow ($(0.4 < \phi < 0.8)$ and $(5.1 < \phi < 5.2)$) indirect calculation by linear interpolation seems to be reasonable. For pressure differences above $-0.02$, that occur at $\phi > 5.2$, infinitesimal frictional loss arises which decreases linearly due to declining clearance width. Local maximums at $\phi = 1.7$ and $\phi = 4.3$ are nearly equal although pressure differences vary by 75%. This points out a dramatic influence of clearance width on hydraulic power. These results indicate that incompressible clearance flow might lead to additional power averaged over the working cycle. The behaviour of mass flow and power within front clearances is not analysed during the working cycle as their clearance geometry is kept constant and pressure difference is the only changing parameter. Furthermore, undefined states of flow do not exist for front clearances. Therefore qualitative functions of mass flow and power always follow the change in pressure difference.

In order to determine the averaged hydraulic power, incompressible flow is simulated for all high pressure front and housing clearances during one phase. Simulation of front clearances is only possible if just two chambers are connected. Figure 5 shows a sample male rotor front clearance connecting chambers a, b and c. At present, calculation of this flow situation is not straightforward with the analytical model. Furthermore, inlet area has to be considered as front clearances exist only partially at this point. Thus, hydraulic loss of high pressure front clearances is calculated for the rotational angles $\alpha_{mr}$ and $\alpha_{fr}$, respectively. Calculations start
when the ultimate lobe point passes the first dashed line and ends at the second line. Thus rotational angles around the inlet area are neglected.

Normalized hydraulic power of the housing and high pressure front clearances is presented in Figure 6 as a function of the expander’s rotational speed. The power of both clearance types is normalized, with maximum power occurring at high pressure front clearances in order to illustrate proportions. Furthermore, overall power is split between the male and female rotor. Simulations with KaSim are carried out assuming the clearance type to be sealed, whereas all other clearances are unsealed. First of all, a different indication of power at the high pressure front and housing clearance is apparent. While the frictional effects of oil cause significant losses within front clearance, flow within the housing clearance leads to additional power. With rising circumferential speed, driving power at the housing clearance actually increases, which is unexpected at first due to greater Couette flow; however, Figure 2 already indicates significant influence of circumferential speed on the performance of this screw expander. As only the housing clearance is sealed, rising rotational speed affects transported mass in all other clearances, which is explained in section 3. Therefore chamber pressures are influenced and pressure difference at the housing clearance changes, which affects Poiseuille flow. Rotors are driven more effectively as long as Couette flow is dominated by opposed Poiseuille flow. Thus the impact of changing pressure difference exceeds the increase in Couette flow. However, additional power arising from flow in housing clearances is negligible compared to hydraulic losses within high pressure front clearances. Obviously, even for small circumferential speeds, Couette flow is not dominated by Poiseuille flow as pressure differences are too small. This effect increases with the circumferential speed of the expander. In particular, high frictional losses within the male rotor front clearance are remarkable. They can be traced back to a large transfer area in the direction of flow. At the male rotor this area is approximately double in size compared with the female rotor.

Generally speaking, the analysis of frictional losses within the front and housing clearances confirms the predominant influence of front clearances. At the operation points investigated, housing clearance flow leads to additional power that is, however, negligible compared to front

**Figure 5.** rotational angle at the sample rotor profile and the inlet area for calculation of hydraulic power at high pressure front clearance
clearances. Section 3 has already illustrated the thermodynamic potential of clearance sealing while not considering losses. In order to evaluate the influence of liquid on the operational behaviour of screw expanders within clearances, these two influences have to be contrasted, as outlined in the next section.

5. Comparison of potential by clearance sealing and frictional losses

In previous sections, the potential of clearance sealing and hydraulic losses has been analysed. In order to investigate the overall effect of the mechanisms described, the expander is simulated for both, dry and wet running conditions. Internal isentropic efficiency for dry screw expanders is determined according to equation (1). Liquid injected machines are modelled with sealed high pressure front and housing clearances. Profile clearance and blow holes are kept unsealed as no incompressible flow model exists at present. Therefore, hydraulic isentropic efficiency is defined as

\[
\eta_{\text{hyd}, s} = \frac{|P_i| - P_c}{\dot{w}_s \dot{m}}
\]

\( \dot{w}_s \)
where $P_i$ is the internal power of an expander with sealed clearances and $P_c$ is the hydraulic power of these sealed clearances (high pressure front clearances and housing clearances).

**Figure 7** shows the isentropic hydraulic efficiency for wet running expanders with two different clearance configurations. While scaling factor $f = 1$ corresponds to relative clearance heights according to Table 1, high pressure front and housing clearance are scaled by 0.5 for the second configuration. All other clearance types are kept constant. Apparently, both types of hydraulic efficiency presented in Figure 7 at first increase with circumferential speed. At low circumferential speeds hydraulic losses are small compared to the expander’s internal power. Improved efficiency can be traced back to sealing effects within the profile clearance, which are described in section 3. With rising circumferential speeds these sealing effects decrease while the hydraulic power of high pressure front clearances increases. According to Figure 6 friction within housing clearances is negligible. Therefore, the gradients of both types of hydraulic efficiency decline and even reverse their signs. Due to higher frictional losses, the hydraulic efficiency of the expander with reduced clearance heights is smaller compared to the unscaled machine for all operating points. Furthermore, the positive gradient of the scaled machine is smaller and the maximum hydraulic efficiency occurs at a lower circumferential speed ($U_{mr} = 50 \text{ m} \cdot \text{s}^{-1}$).

Moreover, isentropic efficiency for dry running expanders is indicated in Figure 7 in order to compare both applications. For small circumferential speeds liquid injection leads to higher efficiency. On the one hand, the potential of sealed high pressure front and housing clearances is remarkable. Note that the potential of these combined clearance types in not shown in Figure 2 and cannot be calculated from a linear combination of the values presented. On the other hand, frictional losses are small. Thus improved thermodynamics due to reduced clearance mass flow dominate frictional losses. With increasing circumferential speed, friction in high pressure front clearances rises, which leads to smaller positive gradients. At the same time, the internal isentropic efficiency of a dry running machine rises due to the sealing effects of all clearance types, which were described in section 3. Therefore, at high circumferential speeds, clearance friction dominates thermodynamic advantages and liquid injection is unfavourable.

It is clear that especially the hydraulic efficiency of liquid injected machines depends substantially on clearance heights. With decreasing clearance heights, frictional losses
increase rapidly, leading to reduced hydraulic efficiency. The different intersection points at \( (U_{mr} = 37 \text{ m·s}^{-1}) \) for \( f = 0.5 \) and \( (U_{mr} = 76 \text{ m·s}^{-1}) \) for \( f = 1 \) point out the necessity for precisely defined hot clearance heights, as shown for example by Nikolov et al. [12].

6. Conclusion
This paper contributes to the understanding of liquid within twin screw expanders, focusing on the opposed effects of clearance sealing and frictional losses. First the maximum potential of clearance sealing is demonstrated. The predominant influence of housing and profile clearances on the thermodynamic behaviour of the investigated geometry is demonstrated. Especially at low circumferential speeds, the sealing of clearances leads to higher internal isentropic efficiency. Besides sealing effects, frictional effects within high pressure front and housing clearances were analysed. It was demonstrated that additional power is generated within housing clearances. However, this is negligible compared with losses within high pressure front clearances. Subsequently, dry and wet running screw expanders were analysed, taking both effects into account. At low circumferential speeds clearance sealing outweighs frictional losses, leading to higher hydraulic isentropic efficiency compared to the internal isentropic efficiency of a dry running expander. With increasing circumferential speed this order changes and dry running expanders are favourable in terms of efficiency. These results illustrate the influence of liquid injection in principle.

An analysis of the thermodynamic potential of clearance sealing shows significant effects of profile clearance. Thus a model for incompressible flow for this clearance type has to be developed in order to determine the intersection point of internal and hydraulic isentropic efficiency more precisely. Moreover, further effects, such as oil surges or cooling of the working fluid, have to be investigated. Nevertheless, two basic effects have been demonstrated. On the one hand, sealing the housing clearance involves remarkable potential for operational behaviour of investigated screw expander; however, friction within this clearance type is infinitesimal. On the other hand, high pressure front clearance shows considerable hydraulic losses while thermodynamic advantages are small. Both effects apply for a wide range of screw expanders. Certainly, the influence of different clearance types changes with varying geometry, for example clearance heights, as demonstrated in this paper; however, for standard applications basic relations between flow cross-section areas and frictional areas in flow direction do not change.
## Notation

| symbol | dimensions | meaning                  | index | meaning   |
|--------|------------|--------------------------|-------|-----------|
| α      | [°]        | rotational angle         | c     | clearance |
| Δ      | [-]        | difference               | hyd   | hydraulic |
| h      | [N·m·kg⁻¹] | specific enthalpy        | i     | internal  |
| η      | [-]        | efficiency               | I     | inlet     |
| \(\dot{m}\) | [kg·s⁻¹] | mass flow                | max   | maximum   |
| \(\mu\) | [Pa·s⁻¹]  | kinematic viscosity      | mr    | male rotor|
| p      | [N·m⁻²]    | pressure                 | O     | outlet     |
| P      | [N·m·s⁻¹]  | power                    | s     | isentropic|
| \(\rho\) | [kg·m⁻³] | density                  | -     | normalized|
| U      | [m·s⁻¹]    | circumferential speed    |       |           |
| φ      | [-]        | normalized phase angle   |       |           |
| w      | [N·m·kg⁻¹] | specific work            |       |           |
| \(w_{c}\) | [m]  | clearance width          |       |           |

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